ABSTRACT

Forrest, Andrew Ryan. Optimization of a Geothermal Heat Pump System with Aboveground Water Storage. (Under the direction of Dr. James W. Leach)

This study investigates and recommends design improvements for a geothermal heat pump system with aboveground water storage. It builds on a previous study that tested a 3-ton geothermal heat pump on a mobile classroom at Wilson Mills Elementary School in Johnston County, North Carolina. The previous experiment used two 1,000 gallon polyethylene bladders filled with saltwater for freeze protection.

Using TRNSYS, a model of the original system was constructed and validated by comparing model predictions to measured performance. TRNSYS models of several new designs and theories were constructed to evaluate potential design improvements. The system models were evaluated based on predicted performance for a typical meteorological year, and on other criteria such as initial cost, maintenance, and portability. This resulted in a new optimized system design in which the water storage volume is reduced to 120 gallons, and the predicted electrical energy requirements are about two-thirds of those of an air source heat pump. The predominant design improvement to the system is the implementation of a heat exchanger constructed of PVC pipe. Detailed design, costs, and assembly procedures for the PVC heat exchanger are presented in this study.

Optimization of a Geothermal Heat Pump System with Aboveground Water Storage

by

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BIOGRAPHY

Andrew Ryan Forrest was born in Kitty Hawk, North Carolina on June 15, 1980. He is the son of James and Diane Forrest, and he has one brother, Jimmy Forrest. He was raised in the area, enjoying the leisure activities the beach has to offer while growing up. He was also an avid participant in organized sports such as basketball, baseball, and soccer during his childhood. He attended Manteo High School where he participated in tennis, soccer, and cross-country. He graduated high school in the spring of 1998 and enrolled at North Carolina State University the following fall.

Andrew chose to major in mechanical engineering soon after he enrolled at N.C. State. In the fall of 2001 Andrew began an accelerated Master's program, which allowed for the completion of a master's degree one year after completing the requirements for a bachelor's degree. Andrew received a Bachelor of Science degree in Mechanical Engineering in May 2002. Immediately following this degree, Andrew began working under the direction of Dr. James Leach on research for geothermal heat pumps in modular classrooms. He was also employed by the University's Industrial Assessment Center where he assisted in performing energy assessments of manufacturing facilities plants and helped write reports that include recommendations on improving energy conservation, productivity, and reducing waste.

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I want to emphasize my gratitude to Dr. leach for his constant support for my research. His patience and knowledge during this project were invaluable resources. Working under the direction of Dr. Leach has truly been a rewarding experience.

A special thanks goes to Eric Soderberg for answering questions about the previous study he performed two years ago. I would also like to thank Stephen Terry for his support and knowledge either on IAC assessments or in the lab, which was instrumental in my development as a better engineer. Thanks to Robert Hughes for his time in making hastily requested parts for the heat exchanger, and for providing the much needed space to assemble the heat exchangers. A special thanks goes to David Bradley of Thermal Energy Systems Specialists for assisting me with questions regarding TRNSYS. The work was sponsored by the North Carolina State Energy Office and Program Manager, Bob Leker. Finally, I would like to thank my parents for their unwavering love, support, and encouragement throughout my academic career. I could not have been as successful without them. I would also like to thank all of my friends for their support and patience as I worked to complete my degrees.

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1 INTRODUCTION

The purpose of this project is to optimize the operating parameters and design of an earth-coupled heat pump system with aboveground water storage. A computer model was created in a Transient Systems Simulation Program (TRNSYS), to predict the best geothermal heat pump design. Available experimental results from the previous project were compared to predicted results in order to validate the model. Afterwards, the computer model was used to predict how changes in system parameters would affect the energy use of the water source heat pump.

1.1 Project Background

Two previous studies were performed to apply geothermal heat pump technology to mobile classrooms, homes, and offices. Researchers at Progress Energy Carolinas, formally Carolina Power & Light, conceived the original design and funded the first study, which was conducted by a graduate student at North Carolina State University. In this study, a geothermal heat pump was installed on a mobile office building, and the water in the system circulated through a plastic bladder that rested on top of the ground. The bladder was insulated to provide some freeze protection, and heat strips were installed in the bladder to heat the water if temperatures approached the freezing point.

The second study, which was funded by the North Carolina State Energy Office, was two-fold, containing a theoretical analysis and experimental validation of a new geothermal heat pump design. The new design was installed on a modular classroom in Johnston County, North Carolina at Wilson Mills Elementary School (WMES). The bladders in this study were not insulated and saltwater was used to prevent freezing. The theoretical analysis was performed by a graduate student at North Carolina State University, and was entitled "Geothermal Heat Pumps and Modular Classroom Units". The theoretical study predicted that the geothermal heat pump would use approximately one-half the energy that a similar air-source heat pump would use on the same classroom.

The experimental part of the study was also performed by a graduate student at North Carolina State University, and was entitled "Test of an Earth-Coupled Heat Pump With Aboveground Water Storage". This study compared the results of the geothermal heat pump system to the results of a nearly identical air source heat pump on an adjacent classroom. The results of this study showed that the heat pump did use about one-half as much electricity as the air-source heat pump while in heating mode. However, in the cooling mode, the geothermal heat pump actually used about 80% as much energy as the air-source heat pump. Subsequent study showed that the ventilation rate in the classroom with the air source heat pump was well below the value recommended by ASHRAE. This changed the building load in favor of the air source heat pump.

The purpose of the current study is to improve the design of the water source heat pump so that it is more energy efficient, and more reliable. Also, a reduction in the initial cost of the heat pump system and additional system mobility is desired. The study will also demonstrate the performance advantages for water source heat pumps by comparing two water source heat pumps to two air source heat pumps in modular classrooms. To improve heat pump energy efficiency, alternatives to the bladder storage system and the use of solar collectors are considered.

In the first phase of the study, candidate design improvements were evaluated, new heat exchangers were designed and fabricated, and a test site was selected. A reliable computer model was needed to evaluate potential design improvements. TRNSYS was selected to model the modular classroom and heat pump. The model was calibrated by comparing predicted performance to experimental data from WMES. Wake County Public School Systems agreed to allow their mobile classrooms to be used to demonstrate the water source heat pumps for a period of one year. Several potential test sites were considered, but Davis Drive Elementary School (DDES) was selected because the site contained multiple school trailers with similar building loads. Several data loggers were left at selected trailers to confirm this assertion. Measurements from the crawl space from two of these trailers were used to design the PVC heat exchangers.

The report begins with a description of the TRNSYS computer model of the water source heat pump used during the WMES experiment. Subsequently, a TRNSYS model of an air source heat pump similar to the one used in the WMES experiment is described. In addition to both of the models, modifications made and errors found in TRNSYS can be found in Section 2. TRNSYS models of new designs and theories (including solar collectors and alternatives to the bladders) are included in Section 3. Results of the TRNSYS models are presented and discussed in Section 4. Based on the results from the TRNSYS computer models, new heat exchangers and a water storage system were developed to replace the large bladders used in the previous experiment. Section 5 describes the PVC heat exchanger design and construction.

1.2 Previous Project Problems

The experiment at WMES provided valuable experimental results to validate previous theories. However, several problems with the water source heat pump design became evident as the experiment was being conducted. Initially, one of the bladders split at the seam during filling, and required replacement. In addition, the saltwater used for freeze protection was corrosive to the pump and heat pump. The volume of the brine solution (2,000 gallons) created disposal and portability problems.

The original bladder that split near the seam was made from a high-density polyethylene material, and cost $250. The bladder that was used as a replacement was constructed of a vinyl coated woven polyester fabric, and cost $750. While the original bladder was a low cost option, it did not meet reliability standards for the project.

The use of saltwater as a heat transfer fluid provided a low-cost option compared to using anti-freeze for freeze protection in 2,000 gallons. However, the saltwater corroded the direct drive pump and required the use of a more expensive magnetic drive pump. Furthermore, the leakage of saltwater around the connections caused the base of the heat pump enclosure to rust. This corrosion would require portions of the heat pump enclosure to be periodically replaced.

2,000 gallons of saltwater cannot be easily transported or easily disposed of if the modular classroom is relocated to another site. To ensure freeze protection, salt was added to the water in the bladders to a 15% solution [1], or about 2,940 pounds [2]. The volume of saltwater would make transportation unlikely, which would raise operational costs and would require disposal.

Based on these problems, a further investigation into improving the geothermal heat pump system was required. In addition to solving these problems, other theories needed to be analyzed to determine if they would improve the system performance.

1.3 TRNSYS Description

TRNSYS is a widely used computer code developed at the University of Wisconsin. It uses a system-specific language that allows the user to program the components that make up the system and the configuration in which they are connected. TRNSYS contains many components that are regularly found in thermal energy systems, along with components to handle input of weather data or external data files. The modular nature of the program allows the user to implement numerical models not included in the standard libraries. TRNSYS is well suited to model thermal systems that are time-dependent.

TRNSYS was chosen as the software to model the geothermal heat pump system for three reasons. TRNSYS is a Fortran language based program, which allowed for a ground temperature distribution and an earth-coupled heat exchanger to be modeled in conjunction with the existing TRNSYS components. TRNSYS is well known for its ability to model solar radiation and solar collectors in thermal systems. The software was used to predict the performance of solar collectors in the geothermal heat pump system. Since the system being modeled in TRNSYS consists of multiple components, it allowed a large, complex system to be broken into smaller parts. Thus, it was convenient to analyze the effects of modifying various parameters in the system.

Two versions of TRNSYS were used for modeling the heat pump system: TRNSYS 14 and TRNSYS 15. TRNSYS 14 was the only version available while the initial models were being constructed. Funding issues held back the purchase of TRNSYS 15 until late October, 2002. TRNSYS 15 was used exclusively after it was purchased.

2 COMPUTER MODEL DESCRIPTION AND VALIDATION

2.1 TRNSYS Operation Overview

TRNSYS components are built with a modular design so that the user has the greatest amount of flexibility in modeling the desired thermal system. Each component is assigned a particular “type” number. In a simulation, multiple “types” may be used, but need to be distinguished by different “unit” numbers. Each component has three major parts: parameters, inputs, and outputs. The parameters are distinct to each component and allow the user to specify static variables in the component. The inputs and outputs are dynamic links between individual components. The inputs for a particular component are linked to the outputs from various other components in the model by specifying the “unit” number and desired output number. Each input must be given an initial value in order for the simulation to begin. Occasionally, when an input does not change with time it is denoted by “0,0” and the initial value is set to the constant value.

2.2 Data Files Description

TRNSYS allows the use of data files as input in various components in the model. Three sets of data files were used to predict the behavior of the water source heat pump system. Building loads that were created to model the modular classroom in the previous experiment were used in this model. In addition, heat pump performance files (released by the manufacturer) were used to model the water source heat pump and air source heat pump. One year of Typical Meteorological Year (TMY) data was used to simulate the ambient conditions in the system.

2.2.1 Building Load Data

Pre-determined building loads, excluding the ventilation loads, were used as designated inputs to the model since Smith and Soderberg had already done extensive work in this area. Smith composed annual building load files for modular classroom with very similar characteristics to the classroom that was modeled in his study. Two years later Soderberg improved the accuracy of the annual building loads with ventilation, infiltration, and internal load adjustments.

The building loads used in the current model exclude ventilation loads since this load can be actively controlled in TRNSYS. In the model, the ventilation flow rate is zero unless the heat pump or electrical strip heat are engaged. Additional changes were made to the infiltration load and infiltration flow rate. In the previous study, the infiltration load, q_{infl}, was calculated by:

$$q_{infl} = 4.5*Q*(h_{HW} + h_o - h_i)$$

where:

Q = Infiltration flow rate (cfm)

h_{HW} = Enthalpy added to ambient air from heat wheel (Btu/lb)

h_o = Enthalpy of ambient air (Btu/lb)

h_i = Enthalpy of air in conditioned space at 70°F db, 60 % RH, 28.8 Btu/lb

The revised infiltration load does not include the enthalpy added from the heat wheel, h_{HW}. The indoor enthalpy, h_i, was changed to a quadratic function, which is written as:

$$h_i = 19.7375 - 0.4167*T_{set}+0.007167*T_{set}^2$$

where:

T_{set} = Indoor thermostat set point, with 50% RH

The other modification to the infiltration load is with respect to the infiltration flow rate, Q. The equation was specified in the previous two studies. In the building load files spreadsheet, a typo had caused the infiltration flow rate to approach zero during the summer months. This was corrected, making the infiltration flow rate equation consistent throughout the year, which is based on temperature difference (between thermostat set point and ambient temperature) and wind speed.

Two building load files were created and are used in all TRNSYS models. The file that contains all building load data except for ventilation loads is "bldloadsi_novent.dat". In order to run simulations to examine the effect of night set back temperature difference, "bldloadsi_novent_setbk.dat" was generated.

2.2.2 Heat Pump Performance Data

Heat pump performance data files are required as inputs for the TRNSYS water source heat pump component. Heating and cooling capacity files were not available for the air source heat pump retrofitted to a water source heat pump. Therefore, detailed heating and cooling capacity files for a range of inlet water temperatures, water flow rates, and air flow rates were obtained from the manufacturer, Bard, for a similar 3-ton water source heat pump, model GSV-361 [4]. The difference between the two heat pumps is that the one used for modeling was not a wall-mount water source heat pump. Both heat pumps use the same High Efficiency Cupro-Nickel Coaxial Water Coil.

In addition to heating and cooling capacity files, heating and cooling correction files were required by the TRNSYS water source heat pump component to account for variations in return air conditions entering the heat pump. Bard supplied very limited data that accounted for a range of dry-bulb and wet-bulb temperatures. Therefore, Performance Data Correction Factors available from Trane were used [5]. As seen in Figure 2.1, this set of correction factors was used because they corresponded closely (within 1%) to the available Bard correction factors when the heat pump was in cooling mode. The heating correction factors could not be compared because Bard did not provide any data for heat pump performance other than for a return air temperature of 70°F. The Trane Performance Data Correction Factors were assumed to be acceptable in this case.

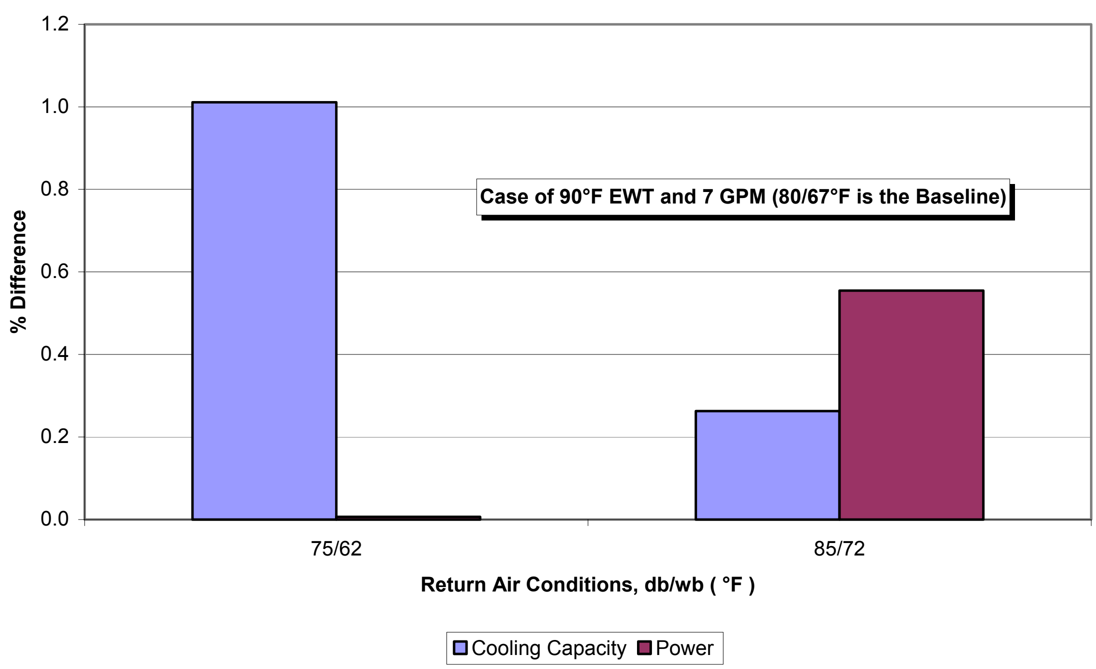

Figure 2.1 – Correlation of Trane Performance Data Correction Factors to Bard Data

The TRNSYS air source heat pump component also required performance data files. The air source heat pump modeled was the WH-361, which is a 3-ton unit. Heating performance data was available from Bard, Inc. for a range of outdoor temperatures. The cooling performance data was also available, but the data does not contain an Energy Efficiency Ratio (EER). The unit was specified as a 10 SEER unit, but clearly the heat pump will be used in ambient conditions above 82°F. Bard was contacted regarding this issue and stated that the EER information was not published. A "ballpark" EER of 9.2 was obtained from Bard Technical Support. The SEER was used to determine the heat pump energy consumption for outdoor temperatures of 75°F-85°F. The EER was used use to model heat pump energy consumption for outdoor temperatures greater than 85°F. The Performance Data Correction Factors from Trane were again used to account for variations in return air temperatures.

2.2.3 TMY Data

The TMY data used by TRNSYS is required to be in a specific format, unique to TRNSYS. It is a fraction of the size of standard TMY data, containing only commonly used information by TRNSYS. The data is corrected for known errors and is in a format easily accessible to TRNSYS. Thermal Energy System Specialists (TESS) provided the data for Raleigh, NC, which is used in the TRNSYS simulation. Table 2.1 shows the format of the unique TMY data files required by TRNSYS.

Table 2.1 – TMY Data Format

TMY FIELD NUMBER	COLUMN NUMBER IN TMY FILE	FIELD DESCRIPTION
1	2-3	Month of the Year
2	5-7	Hour of the Month
3	10-13	Direct normal solar radiation (integrated over pervious hour) kJ/m^2
4	15-18	Global solar radiation on horizontal (integrated over previous hour) kJ/m^2 (1)
5	20-23	Dry Bulb Temperature (degrees *10, C)
-	25-30	Humidity Ratio (kg H_2O/kg air)*10^4
6	32-33	Wind velocity (m/s)
7	35-36	Wind direction (degrees *10) (2)

Notes:
(1) Note that this is not horizontal surface beam radiation.
(2) Wind direction expressed as 0 for wind from north, 9 for east, 18 for south, etc.

2.3 Model of Geothermal Heat Pump System at Wilson Mills Elementary School

The first goal of the project was to model the geothermal heat pump system's performance at WMES using TRNSYS. Descriptions of the TRNSYS components used in this simulation are provided to help clarify the model [6]. Once a component has been introduced for one simulation, it will not be restated unless significant changes are made.

The components that are generally used in TRNSYS models include components relaying input and output data for the model or simple components that are useful in modeling. The input components include a TMY Data Reader and a Building Load Reader. The components commonly used within most models include a user-defined Forcing Function and a Quantity Integrator. The output components that are typically used within most models are a Printer and Online Graphics. Parameters will only be explained for general and output components. The distinctive components used in this model include the

Ground Heat Exchanger (bladder version), Water Source Heat Pump, Pump, Building Load, and Three-Stage Thermostat. Specific parameters and inputs are explained in detail for each component. Figure 2.2 shows the basic energy flow path.

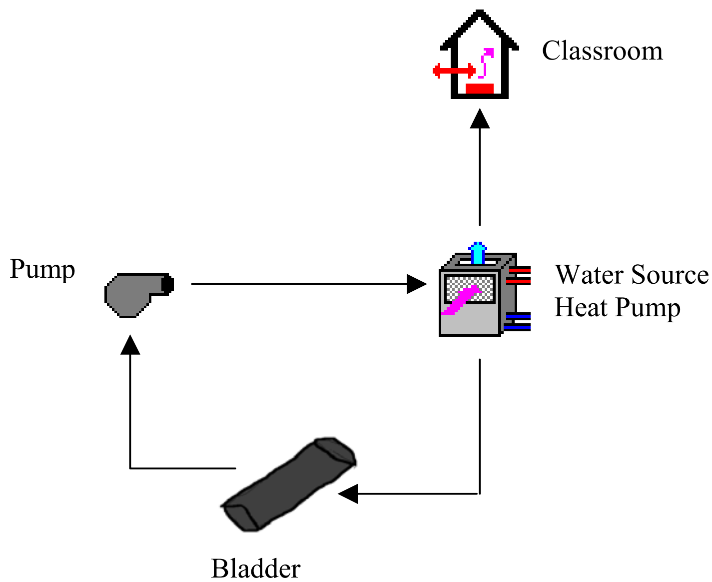

Figure 2.2 – Layout of Geothermal Heat Pump System with Bladders

2.3.1 Input Components

2.3.1.1 TMY Data Reader

The TMY Data Reader, type 89, is known as a standard data reader and reads meteorological data at regular time intervals from a logical unit number (external data file). It is used to read the "Raleigh.dat" TRNSYS TMY data file. This data is converted into a desired system of units, and is assigned to specific outputs. The outputs are available for other TRNSYS components as inputs at the specified simulation time step.

Table 2.2 – Parameter Description for Type 89

Parameter No.	Value	Description
1	-1	Mode - Read TRNSYS TMY data file where T_{on}-1 data lines are skipped before the simulation begins.
2	10	L_{unit} - logical unit number of input TMY data file

2.3.1.2 Building Load Reader

The Building Load Reader, type 9, is known as a standard data reader and reads data at regular time intervals from a logical unit number. It is used to read the "bldloadsi_novent.dat" file described in Section 2.2.1. The data is assigned to specific outputs, which are available for other TRNSYS components as inputs varying over time. The data can be interpolated for time steps less than one hour based on parameters specified by the user.

Table 2.3 – Parameter Description for Type 9

Parameter No.	Value	Description
1	-1	Mode - read user-supplied data file where ($T_{on}/\Delta t_d$-1) data lines are skipped before the simulation begins.
2	0	Skip – number of header lines in data file to skip before data begins
3	1	N – total number of columns to be read from the data file
4	1	Δt_d – time interval at which data is provided (hrs)
5	1	output column to interpolate using following two parameters
6	1	m_i – multiplication factor
7	0	a_i – addition factor
8	18	L_{unit} - logical unit number of input data
9	-1	FRMT – formatted reading (FRMT > 0 specifies formatted reading)

2.3.2 General Components

2.3.2.1 Time Dependent Forcing Function

The Time Dependent Forcing Function, type 14, is a convenient way to simulate behavior that is characterized by a repeated pattern. The type contains no inputs since the pattern is specified in the parameters declaration. Several units of this type are used in the model to simulate daily occupancy patterns, night set back temperature difference, and seasonal changes in the thermostat set points. The forcing function will repeat whatever pattern is specified. The pattern periods can range from hourly to annually. The specifications for other forcing functions can be found in Appendix A and Appendix B. The parameters to model a night set back temperature difference are provided in Table 2.4.

Table 2.4 – Parameter Description for Type 14

Parameter No.	Value	Description
1	0	Initial time of time (hr)
2	1	Initial value of the function
3	6	Second value of time (hr)
4	1	Second value of the function
5	6	Third value of time (hr)
6	0	Third value of the function
7	16	Fourth value of time (hr)
8	0	Fourth value of the function
9	16	Fifth value of time (hr)
10	1	Fifth value of the function
11	24	Final value of time (hr)
12	1	Final value of the function

2.3.2.2 Quantity Integrator

The Quantity Integrator, type 24, is used to integrate quantities in the simulation for convenient display. It is an aid in data manipulation and organization. The Integrator was used to tally the energy used by the heat pump on a daily basis. An additional benefit of the Integrator is to reduce the size of the output file generated by a TRNSYS simulation.

Several units of this type were used in the simulation so a detailed description of inputs and parameters will not be provided. The inputs to the Integrator are the outputs from various components that one wants to tally. The only parameter required for this type is the time interval in hours over which the inputs are to be integrated. Appendix A and Appendix B contain examples of this component.

2.3.3 Output Components

2.3.3.1 Printer

The Printer, type 25, is used to output (or print) selected system variables at specified time intervals. The Printer prints the data to a logical unit number, which has an associated data file. The values in Table 2.5 represent an hourly printer for the entire time of the simulation.

Table 2.5 – Parameter Description of Type 25

Parameter No.	Value	Description
1	1	Δt_p – time interval at which printing is to occur (hr)
2	0	t_{on} – the time at which printing is to start (hr)
3	8,760	t_{off} – the time at which printing is to stop
4	11	L_{unit} - logical unit number of output file
5 (optional)	2	Units – print TRNSYS supplied units for each input

2.3.3.2 Online Graphics

The Online Graphics, type 65, is used to display selected system variables at specified time intervals while the simulation is in progress. The component allows the user to observe multiple variables during the simulation time. This component is valuable because it allows the user to immediately see if the system is not performing as expected. The variables typically displayed by this component were room temperature and outlet fluid temperatures of various components.

Table 2.6 – Parameter Description for Type 65

Parameter No.	Value	Description
1	3	N_{top} - Number of variables displayed on left axis
2	0	N_{bot} - Number of variables displayed on right axis
3	0	$Y_{min,1}$ – Left axis minimum
4	120	$Y_{max,1}$ – Left axis maximum
5	0	$Y_{min,2}$ – Right axis minimum
6	120	$Y_{max,2}$ – Right axis maximum
7	1	N_{pic} – Number of plots per simulation
8	10	Grid - number of x axis grid points
9	1	On/off – Display plot online
10	-1	LU – don't print output file

2.3.4 Distinctive Components

2.3.4.1 Water Source Heat Pump

The Water Source Heat Pump, type 504, is a recent addition to the TRNSYS components, and was purchased separately from TESS as part of the Geothermal Heat Pump Library. It is used to predict the energy consumption, the outlet water temperatures, as well as the heating and cooling performance of the heat pump. Modifications to the source code were made for this component, and are explained in Section 2.5.

Table 2.7 – Parameter Description for Type 504

Parameter No.	Value	Description
1,2	14,15	Logical unit No. of heating/cooling performance data
3,4	16,17	Logical unit No. of heating/cooling correction factor data
5-12	------	Numbers of columns and rows of data in heating/cooling performance and correction factor files
13	1,130	Density of liquid stream (kg/m^3)
14	3.48	Specific heat of liquid stream (kJ/(kg-°C))
15	1.007	Specific heat of air stream (kJ/(kg-°C))
16	N/A	Specific heat of DHW fluid (kJ/(kg-°C))
17	1,343	Blower power (kJ/hr) (1)
18	36	Controller power (kJ/hr) (1)
19	36,000	Capacity of stage-1 auxiliary heating device (kJ/hr) (1)
20	0	Capacity of stage-2 auxiliary heating device (kJ/hr)
21	566	Total air flow rate (l/s) (1)
22	153	Outside air flow rate-325 CFM (l/s) (1)

Note:
(1) Values were obtained from [4]

Table 2.8 – Input Description for Type 504

Input No.	Value	Description
1	3,1	Inlet liquid temperature (°C)
2	3,2	Inlet liquid flow rate (kg/hr)
3	12,4	Return air temperature (°C)
4	WTOT2	Return air humidity ratio (kg H_2O/kg dry air) (1)
5	9,5	Ambient air temperature (°C)
6	9,6	Ambient humidity ratio (kg H_2O/kg dry air)
7	N/A	Inlet DHW temperature (°C)
8	N/A	Inlet DHW flow rate (kg/hr)
9	8,3	Cooling control signal
10	8,1	Heating control signal
11	8,2	Stage 1 auxiliary heating signal (10 kW strip heaters)
12	0	Stage 2 auxiliary heating signal
13	0	Control signal for operation of ventilation fan when heat pump is not operating in heating or cooling mode
14	1	Control signal for mixing of the outside air with return air before sending it to the heat pump
15	N/A	Cooling desuperheater temperature (°C)
16	N/A	Heating desuperheater temperature (°C)
17	1	Fraction of rated catalog cooling power used
18	1	Fraction of rated catalog cooling capacity used
19	1	Fraction of rated catalog heating power used
20	1	Fraction of rated catalog heating capacity used
21	N/A	Desuperheater UA in cooling mode (kJ/hr-K)
21	N/A	Desuperheater UA in heating mode (kJ/hr-K)

Note:
(1) WTOT2 is an equation that controls return air humidity ratio

2.3.4.2 Three-stage thermostat

The Three Stage Thermostat, type 8, is modeled to output three on/off signals that are used to control the heating or cooling mode on the water source heat pump and the auxiliary electrical resistance heaters. This controller is to be used to control systems based on temperature-rate control. When a thermal system containing a thermostat is to be modeled, it is much more accurate to model the system using temperature-level control as opposed to energy-rate control. Modifications were made to this component, and can be found in Section 2.5.

Table 2.9 – Parameter Description of Type 8

Parameter No.	Value	Description
1	5	No. of oscillations in a time step before outputs "stick"
2	1	Enables first stage heating during second stage
3	-3.89	Min. source temp. for source utilization (minimum inlet fluid temperature of heat pump) (° C) [5]
4	23.89	Room temp. above which room is to be cooled (°C)
5	21.11	Room temp. below which first stage heating is used (°C)
6	20.0	Room temp. below which second stage heating is used (°C)
7	8.33	Heating set back temp. difference (°C)
8	1	Dead band temp. difference (°C)

Table 2.10 – Input Description for Type 8

Input No.	Value	Description
1	12,4	Room Temperature (°C)
2	3,1	Liquid source temperature (°C)
3	0	Heating set back control function (not used in this simulation)

2.3.4.3 Building Load

The Building Load component, type 12, is used to calculate the room temperature in the model. Because extensive research had already been done to calculate building loads for the mobile classroom at WMES, this component was not used in its traditional manner. Therefore, most inputs and parameters required by the component were not used and are set to 0. Modifications were made to this component, and can be found in Section 2.5. In the component, the room temperature was determined by:

$$T_{r,final} = T_{r,initial} + (Q_{ld} + Q_{hp})*dt/CAP$$

where:

$T_{r,final}$ = Room temperature at end of time step (°C)

$T_{r,initial}$ = Room temperature at beginning of time step (°C)

Q_{ld} = Energy transferred to the air by heat pump (kJ/hr)

Q_{ld} = Estimated Building load for specified hour of simulation (kJ/hr)

dt = time step of simulation (0.05 hrs)

Table 2.11 – Parameter Description of Type 12

Parameter No.	Value	Description
1	4	Lumped capacitance with temperature level control
2	N/A	UA – overall conductance for heat loss from classroom (kJ/hr -°C)
3	4,000	CAP – lumped thermal capacitance of classroom (kJ/°C)
4	22.2	T_{RI} – initial room temperature (°C)
5	3.48	C_{Pf} – specific heat of heat source fluid (kJ/kg -°C)
6	N/A	εC_{min} – product of the effectiveness and min. capacitance rate of load heat exchanger

Table 2.12 – Input Description for Type 12

Input No.	Value	Description
1	N/A	Temperature of fluid from heat source (°C)
2	N/A	Mass flow rate of fluid from heat source (kg/hr)
3	N/A	Ambient temperature (°C)
4	29,1	Time variant heat gains (pre-determined hourly building load) (kJ/hr)
5	104,8	Auxiliary heating input to space (energy transferred to the air from heat pump and ventilation load) (kJ/hr)
6	N/A	Rate of cooling energy removed from space (kJ/hr)

2.3.4.4 Pump

The Pump, type 3, is a simple model of the pump in the actual system. It is primarily used to estimate the annual power consumption of the pump and to control the fluid flow through the system. The component assumes the pump uses its rated horsepower whenever it is operating. The size of the pump is one-third horsepower. The Pump component also accounts for the fraction of pump power converted to fluid thermal energy. This fraction is neglected because thermal losses in the pipes in the actual system are neglected.

Table 2.13 – Parameter Description of Type 3

Parameter No.	Value	Description
1	2567	Mass flow rate of liquid (Q ~10 gpm) (kg/hr)
2	3.48	Fluid specific heat (kJ/(kg-°C))
3	895.2	Maximum power consumption (1/3 hp) (kJ/hr)
4	0	Fraction of pump power converted to fluid thermal energy ($0 \leq f_{par} \leq 1$)

Table 2.14 – Input Description of Type 3

Input No.	Value	Description
1	116,1	Inlet liquid temperature (°C)
2	116,2	Inlet liquid flow rate (kg/hr)
3	8,1 + 8,3	Control function for pump (indicates when compressor is operating)

2.3.4.5 Ground Heat Exchanger and Ground Temperatures

The Ground Heat Exchanger is used to model the bladders (type 117) or the PVC heat exchangers (type 118) in the actual system. The component also predicts temperatures in the ground underneath the heat exchanger. The major function of the component is to predict the outlet fluid temperature of the heat exchanger. The model also accounts for moisture freezing in the ground underneath the bladder. A detailed explanation of fundamental theories and assumptions will follow. Type 117 and type 118 are supplemental (designed and written for this simulation) components, and the source code is included in Appendix C.

There are two differences between type 117 and type 118. The differences are in the allocation of surface area for heat transfer to the air and to the ground. The other difference is in the heat transfer coefficient between the air and the heat exchanger. The variations were implemented to realistically model each type of ground heat exchanger. The analysis is provided in Section 2.3.4.5.3

The bladder version of this component, type 117, is used in this model. Two units of type 117 are used in the simulation to model the two 1,000 gallon bladders. The area for heat transfer to the air is specified to be about 20% larger than the area for heat transfer to the ground. The following tables provide a parameter and input description for the bladder version of the heat exchanger (type 117).

Table 2.15 – Parameter Description for Type 117

Parameter No.	Value	Description
1	640	Surface Area of heat exchanger (ft^2)
2	2,000	Volume of fluid in heat exchanger (gals)
3	3.48	Specific heat of liquid (KJ/kg-°C)
4	1,130	Density of liquid (kg/m^3)
5	10	Heat transfer coefficient for water and heat exchanger (BTU/(hr-ft^2-°F)
6	2.0	Heat transfer coefficient for air and heat exchanger (BTU/(hr-ft^2-°F)

Table 2.16 – Input Description for Type 117

Input No.	Value	Description
1	104,1	Inlet liquid temperature (°C)
2	104,2	Inlet liquid flow rate (kg/hr)
3	9,5	Ambient air temperature (°C)

2.3.4.5.1 Methods and Assumptions

The temperature distribution in the ground is modeled using the Explicit Finite Difference Model for one-dimensional transient heat conduction. This is a conservative estimate in the model because heat transfer in the y and z direction are neglected. This assumption decreases the heat transfer between the heat exchanger and the ground. The heat exchanger is modeled as a lumped capacitance.

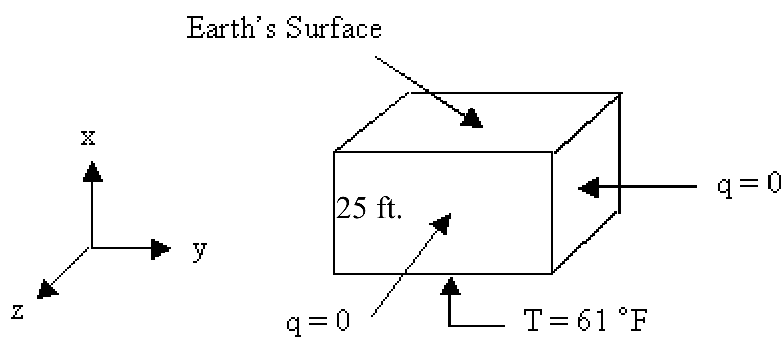

Figure 2.3 – Boundary Conditions for Ground Temperatures

The energy balance equations were determined based on several assumptions and basic calculations for boundary conditions and parameters:

1. The 50^{th} node is 25 ft below the surface of the earth and is approximated at 61 °F.
2. An initial heat exchanger water temperature is determined by the average of the ambient air temperature and 61 °F.
3. The depth of the nodes is 1/2 ft.
4. Heat transfer in the y and z direction is neglected.
5. The temperature on the surface of the ground is the same as the temperature of the surface of the heat exchanger that is in contact with the ground.

6. The temperature of the fluid in the heat exchanger is constant throughout the heat exchanger during one time step.
7. Properties of the soils

Table 2.17 – Soil Properties Used in Ground Temperature Model [7]

Description	Value	Units
Thermal Conductivity of soil with 25% water	1.15	Btu/(hr-ft-°F)
Density of Soil with 25% water	93.6	lb/ft^3
Specific Heat of soil with 25% water	0.60	Btu/(lb-°F)
Thermal conductivity of soil with 25% ice	1.0	Btu/(hr-ft-°F)
Density of soil with 25% ice	84.6	lb/ft^3
Specific heat of soil with 25% ice	0.46	Btu/(lb-°F)

2.3.4.5.2 Frozen Ground Assessment

During the previous experiment, ice formed underneath the bladders on the surface of the ground. As a result, the ground heat exchanger was designed to model the phase change of moisture in the ground underneath the heat exchanger. This was determined to be a worthwhile modeling consideration because the soil nodes would remain at 32°F while a phase change takes place and the fluid temperature in the heat exchangers is expected to drop to approximately 25°F. This extra consideration was expected to more accurately model the performance of the heat exchanger in the winter.

By simulating temperatures in the ground beneath the heat exchanger, it was observed that during the winter, when freezing of the ground takes place, it does not occur deeper than 1.5 ft. Since the nodes are 1/2 ft deep, a phase change energy balance is only considered for three soil nodes. The basic equation for determining soil node temperature at the end of a time step is given below:

$$a*T+b = m*c_p*((T_p-T)/dt)$$

where:

a	=	dependent variables for soil nodes (refer to Appendix C)
b	=	independent variables for soil nodes (refer to Appendix C)
dt	=	time step (hrs)
T	=	temperature of node at beginning of time step (°F)
T_p	=	temperature of node at end of time step (°F)
m	=	mass of soil nodes (lbm)
c_p	=	specific heat of soil with constant pressure (Btu/hr/lbm/°F)

After the temperature at the end of the time step is determined, a check is performed to see if the temperature is below 32°F. If the temperature is below freezing, the count of the heat of fusion of water is incrementally increased, and an appropriate mass fraction of ice and water is determined. Soil properties are altered depending on the state of moisture in the soil. Once the mass fraction of ice becomes one (heat of fusion is reached), the temperature of the soil nodes changes according to the explicit finite difference equations, and corresponding soil properties are determined. A similar process occurs in reverse if the temperature in the soil node is above freezing. The model accounts for sub-freezing and above-freezing conditions during the "thawing" or "freezing" states, respectively, and adjusts the count of the heat of fusion appropriately.

2.3.4.5.3 Heat Transfer Coefficient Calculations

The heat transfer coefficient to the air, h_{air}, is an intriguing parameter in the model that required calculations for both type 117 and type 118. As the WMES geothermal heat pump system model was being constructed it became apparent that the heat transfer coefficient between the bladder and air, $h_{air,Blad}$, must be approximately 2 Btu/hr/ft^2/°F.

Under this assumption, the predicted bladder temperatures tracked very closely to the actual bladder temperatures as shown in Figure 2.4. This value is used in all TRNSYS simulations that include type 117.

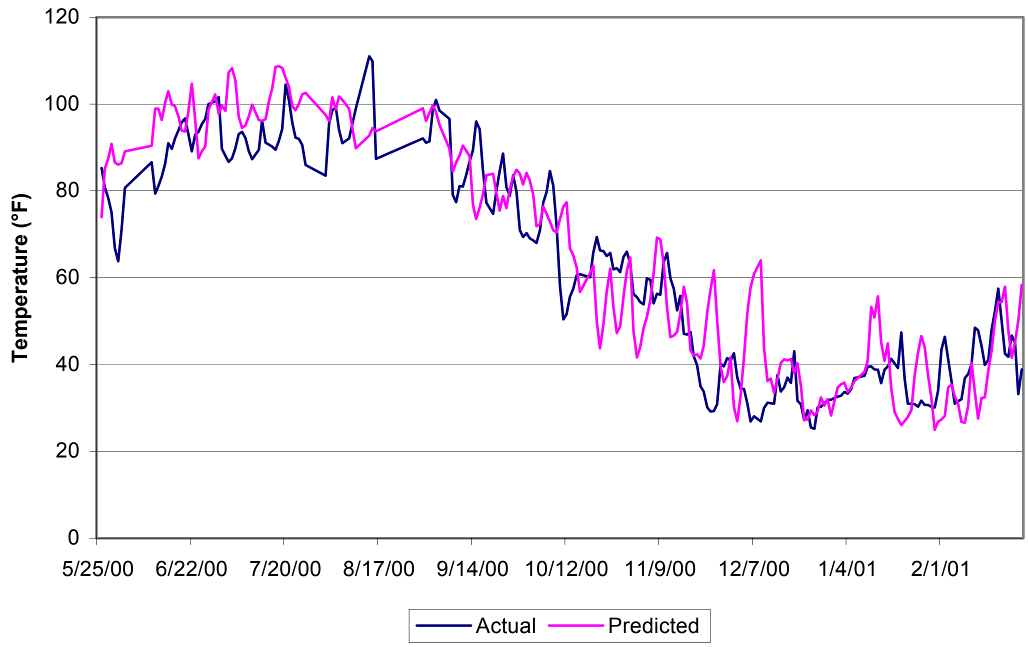

Figure 2.4 – Actual and Predicted Bladder Outlet Water Temperatures

In order to calculate h_{air} for the PVC heat exchangers a detailed heat transfer analysis was done for both the bladder and PVC heat exchangers. The effective heat transfer coefficient for the bladder is a combination of the external and internal heat transfer coefficients. Due to the size of the bladders, the conductive resistance of the bladders can be neglected.

The internal heat transfer coefficient of the bladders is based on natural convection only because ($Gr_D/Re_D^2 >> 1$), where the Grashof number and Reynolds numbers are defined below. The bladder was assumed to be a circular cylinder with a hydraulic diameter of 2.3 ft.

$$Gr = g*\beta*\Delta T*D_h^3/\nu^2$$

where:

g = acceleration due to gravity, 32.2 ft/sec^2

β = thermal expansion coefficient of water, 55.55 x 10^{-6} R

ΔT = average temperature difference between bladder surface and fluid, 9°F (conservative estimate)

D_h = hydraulic diameter of bladder, 2.3 ft

ν = kinematic viscosity of water, 0.016 x 10^{-3} ft^2/sec

The Reynolds number is defined by the following equation:

$$Re_D = 4*\dot{m}/(\pi*\mu*D_h)$$

where:

$\dot{m}$ = mass flow rate of fluid in bladder, 13.33 lb/sec (10 gpm)

μ = viscosity of water, 1.5 x 10^{-3} lbf/(ft-sec)

The Rayleigh Number, Ra, is required to determine the internal heat transfer coefficient, h_i, due to natural convection. It is defined by the following equation:

$$Ra = Gr*Pr$$

where:

Gr = Grashof number, 760 x 10^6

Pr = Prandtl number for water, 6

For internal natural convection, the heat transfer in the bladder to be the presumed to behave as a flat plate. The empirical correlation for the average Nusselt number (suggested by McAdams) for a Ra_D of 4.56 x 10^9 is:

$$Nu_D = 0.15*Ra_D^{.333} = h_i*D_h/k_{water} \quad [8]$$

This results in an average internal heat transfer coefficient of about 37 Btu/h/ft^2/°F.

The external heat transfer coefficient is due to radiation and convection. An average radiation heat transfer coefficient, $h_{R,Blad}$, was calculated to be approximately 0.6 Btu/h/ft^2/°F using the following equation:

$$h_{R,Blad} = F_{eff,Blad}*\varepsilon*\sigma*(T_1^2+T_2^2)*(T_1+T_2)$$

where:

ε = emissivity of bladder, 0.9 [9]

σ = Stephan-Boltzmann Constant (0.1714 x 10^{-8} Btu/h/ft^2/°R^4)

T_1 = Temperature of Bladder (510°R, average annual temperature)

T_2 = Temperature of Surroundings (520°R, average annual temperature)

$F_{eff,Blad}$ = View Factor based on Radiation analysis of bladder to surroundings and underside of the classroom, 0.7

Then, using the determined $h_{air,Blad}$, the convective heat transfer coefficient, $h_{c,Blad}$, is 1.5 Btu/h/ft^2/°F. Assuming that the Reynolds number for the bladder in cross flow is between 40,000 and 400,000, the empirical correlation for the average Nusselt number, Nu_D, (due to Hilpert) is:

$$Nu_D = 0.027*Re_D^{.805}*Pr^{(1/3)} = h_{c,Blad}*D_h/k_{air} \quad [8]$$

where:

k_{air} = 0.0152 Btu/h/ft/°F

Pr = 0.7

Therefore, Re_D is 87,471. The corresponding wind velocity for this Reynolds number is about 5 mph. According to the Southeast Regional Climate Center, the average wind speed for Raleigh, NC is about 7.5 mph, which makes 5 mph a reasonable estimate for average wind speed around and under the mobile classrooms [10].

The effective heat transfer coefficient for the PVC heat exchangers only includes the external heat transfer coefficient and the thermal conductive resistance of the PVC pipe. The internal heat transfer coefficient is dominated by forced convection ($Gr/Re^2 \ll 1$), which only applies when fluid is flowing. According to the model predictions, the heat pump runs about 50% of the time during the year. Therefore, during periods of no-flow conditions, a transient conduction analysis of the stagnant fluid is required to accurately predict the heat transfer in the pipes. Consequently, the heat transfer inside the pipes was neglected.

The external heat transfer coefficient is a combination of radiation and convection. An average radiation heat transfer coefficient, $h_{R,PVC}$, was calculated to be approximately 0.63 Btu/h/ft²/°F using the following equation:

$$h_{R,PVC} = F_{eff,PVC} * \varepsilon * \sigma * (T_1^2 + T_2^2) * (T_1 + T_2)$$

where:

ε = emissivity of PVC, 0.84 [9]

σ = Stephan-Boltzmann Constant (0.1714×10^{-8} Btu/h/ft²/°R⁴)

T_1 = Temperature of Bladder (510°R, average annual temperature)

T_2 = Temperature of Surroundings (520°R, average annual temperature)

$F_{eff,PVC}$ = View Factor based on Radiation analysis of PVC heat exchanger to surroundings and underside of the classroom, 0.75

The same average wind speed (5 mph) was used for the PVC heat exchanger, which resulted in a convective heat transfer coefficient, $h_{c,PVC}$, of 5.5 Btu/h/ft^2/°F. The Reynolds number for flow over the 1/2 in PVC pipe is about 3,000. For a circular cylinder in cross flow with Reynolds number of about 3,000, the empirical correlation (from Hilpert) for the average Nusselt number, Nu_D, is:

$$Nu_D = 0.683 * Re_D^{.466} * Pr^{(1/3)} = h_{c,PVC} * D_{PVC} / k_{air} \quad [8]$$

where:

k_{air} = 0.0152 Btu/h/ft/°F

Pr = 0.7

The thermal resistance of the PVC pipe, $R_{t,cond}'$ is equal to 0.0324 ft-h-°F/Btu and is governed by the following equation:

$$R_{t,cond}' = \ln(D_o/D_i)/(2*\pi*k)$$

where:

D_o = Outer diameter of 1/2 in PVC pipe, 0.84 in

D_i = Average inner diameter of 1/2 in PVC pipe, 0.70 in

k_{PVC} = 0.896 Btu/h/ft/°F (average thermal conductivity of PVC) [11]

The normalized conductive resistance, $R_{t,cond}''$, is 0.0855 ft^2-h-°F/Btu.

With the aforementioned considerations, the overall heat transfer coefficient, $h_{air,PVC}$ is approximately 4 Btu/h/ft^2/°F. This value is used in all TRNSYS simulations that include type 118.

2.4 Model of Air Source Heat Pump System at Wilson Mills Elementary School

In an effort to better understand the results from the geothermal heat pump model a model of the air source heat pump system was generated. Actual data was collected for an air source heat pump on an adjacent classroom during the WMES experiment. Nevertheless, the model of the air source heat pump was not compared to the data from the WMES experiment because crucial information for the model was unknown or unavailable. The model was created using the same building loads and thermostat settings that were specified for the geothermal heat pump model. The only new component was the air source heat pump itself.

The computer model was not correlated to the experimental data from the previous experiment because there were two distinct unknowns. First, the Wall-Mount™ Energy Recovery Ventilator (WERV) was not working for an unknown duration, but was estimated to have spanned several months. Soderberg stated that the disconnected WERV indicated that the classroom was not receiving any forced ventilation air, which would substantially reduce the overall building loads. Secondly, during the winter months, the air source heat pump was frequently observed in the defrost cycle. This heat pump function was not included in type 665. With these differences known, the annual energy consumption of the heat pump could not be compared to the experimental data collected during the previous experiment.

The model does provide a basis for a prediction of energy savings using the specified geothermal heat pump system compared with the specified air source heat pump. The air source heat pump model contains code to simulate an outdoor thermostat on the heat pump.

This provides some comparison for the two systems. Figure 4.3 shows the annual energy comparison for the two systems.

2.4.1 Air Source Heat Pump

The Air Source Heat Pump, type 665, is a recent addition to the TRNSYS components, and was obtained separately from TESS. It is used to predict the energy consumption as well as the heating and cooling performance of the heat pump. Basic modifications were made to this component and include the implementation of ventilation loads that are the same as the loads prescribed for the water source heat pump.

Table 2.18 – Parameter Descriptions for Type 665

Parameter No.	Value	Description
1	14	Logical unit No. cooling performance data
2	15	Logical unit No. heating performance data
3-8	------	Numbers of columns and rows of data in heating/cooling performance files
9	1.007	Specific heat of air stream (kJ/(kg-°C))
10	36,000	Capacity of stage-1 auxiliary heating device (kJ/hr) (1)
11	0	Capacity of stage-2 auxiliary heating device (kJ/hr)
12	1.0	Scale factor for heat pump
13	566	Total air flow rate (l/s) (1)
14	153	Outside air flow rate-325 CFM (l/s) (1)
15	1,343	Blower power (kJ/hr) (1)

Note:
(1) Values were obtained from [13]

Table 2.19 – Input Description for Type 665

Input No.	Value	Description
1	12,4	Return air temperature (°C)
2	WTOT2	Return air humidity ratio
3	9,5	Ambient air temperature (°C)
4	9,6	Ambient humidity ratio
5	8,3	Cooling control signal
6	8,1	Heating control signal
7	8,2	Stage 1 auxiliary heating signal (10 kW strip heaters)
8	0	Stage 2 auxiliary heating signal
9	0	Control signal for operation of ventilation fan when heat pump is not operating in heating or cooling mode
10	OA_SIG	Control signal for mixing of the outside air with return air before sending it to the heat pump

2.5 Modifications for Current Study and Errors Found

It was mentioned in the overview of TRNSYS that two versions were used to create the current model. Changes were made to source-code of the Water Source Heat Pump, Air Source Heat Pump, Three-Stage Thermostat, and Building Load components. The changes were implemented over the model construction period. Although some of the discrepancies were found in TRNSYS 14, the changes made after October 2002 were only made to TRNSYS 15 components. The changes or corrections were made to more accurately represent the systems being modeled, with the majority of the changes pertaining to the water source heat pump component. The corrections were discussed with TRNSYS technical support. Table 2.20 shows a summary of changes made to TRNSYS source code.

Table 2.20 – Modifications to TRNSYS Components

Type	Line(s)	Previous Equation	Reason
504	416	X(2) = WFLOW **3.7854/60.	Conversion should be multiplied, not squared.
504	557	ELSE IF (ONHEAT) THEN	Code to assign auxiliary energy use to total heat pump energy use is nested inside this condition statement.
504	489	N/A (added equations)	Ventilation loads are actively calculated using WERV efficiency
504	509, 523	QTOTC=FLOW* (HAIRIN-HAIRO)	The ventilation loads are being independently calculated in the component so this line is not necessary.
504	331-360	N/A (added equations)	The strip heating was being bypassed when no-flow conditions were present.
665	268	N/A (added equations)	Ventilation loads are actively calculated using WERV efficiency
665	326, 328	QTOTC=FLOW* (HAIRIN-HAIRO)	The ventilation loads are being independently calculated in the component so this line is not necessary.
8	93	IF((IGAM2.EQ.1). AND.(ISTG.EQ.1)) THEN	Component description states that heat pump should only run if the fluid temperature (TS) is above a specified value (TMIN)
8	77	TMIN = TMIN - OUT(1)*PAR (IDB)	The minimum water temperature (TMIN) should not contain a dead band temperature difference
12	160	N/A (added equations)	The type was using an average room temperature instead of final room temperature for the temperature of the room at the end of the time step.

2.5.1 Type 504 – Water Source Heat Pump

The first change to type 504 corrected a typo. The code in the first row of Table 2.20 is intended to convert the inlet water flow rate from gallons per minute to liters per second. The conversion should be multiplied not raised to a power.

The second change reflects a revision in the way the component is being used for the models that were created. The original code failed to include electrical resistance heat in the total heat pump power use if the heat pump was not operating because the assignment statement was nested inside this condition statement. The condition statement was changed to include the auxiliary power signal.

The third change reflects the inclusion of the actively controlled ventilation loads, which also accounts for the WERV. Although Bard states that the WERV efficiencies were as high as 72% in the winter, the efficiency was conservatively specified to be 50% in the model, which is in line with previous work [1],[12]. The ventilation load is specified by the following equation:

$$Q_{vent} = flow\_oa/2*(h_{air\_amb}-h_{air\_return})$$

where:

flow_oa/2 = Outside air mass flow rate specified by the user when the blower is on, divided by 2 to account for the efficiency of the WERV (kg/hr)

h_{air_amb} = Enthalpy of the return air, based on temperature and humidity (kJ/kg)

h_{air_return} = Enthalpy of the return air, based on temperature and humidity (kJ/kg)

The fourth change is related to the previously described addition of ventilation air. The component originally modified the energy transferred to the air in cooling mode based on enthalpy of the return air and enthalpy of the outdoor air. However, the heating load was not being adjusted at all. This problem was resolved by independently calculating the ventilation loads, and then adding the loads to the energy transferred to the air from the heat pump. Therefore, the lines specified in Table 2.20 should be eliminated.

The fifth change to the component is addition of code to include electrical resistance heat in the energy transferred to the air during conditions when the water flow to the heat pump was zero (heat pump was not running). A conditional statement set several heat pump power values to zero either when a heating or cooling signal was not received, or inlet water flow rate was zero. This is similar to the second change made to this component, but the two sets of code were in different parts of the Fortran file.

2.5.2 Type 665 – Air Source Heat Pump

The changes made to type 665 were the same changes made to the water source heat pump regarding ventilation loads. The changes include the inclusion of actively controlled ventilation loads and by eliminating lines that altered the heat pump performance when in cooling mode.

2.5.3 Type 8 – Three Stage Room Thermostat

The first change to the type 8 was made to reflect the description of the component in the "TRNSYS 15 User's Manual". The description states that the heat pump should only run if the fluid entering the heat pump is greater than a minimum temperature, which was specified in Section 2.3.4.2. The Fortran code for the component did not include a comparison of the specified minimum source temperature (TMIN) to the temperature of the fluid entering the heat pump.

The second change was regarding the controller's dead band temperature difference settings. The dead band temperature difference, PAR(IDB), had been multiplied to the minimum entering fluid temperature, resulting in a minimum water temperature that was lower than the user specified.

2.5.4 Type 12 – Building Load

The change made to this component modified the manner in which the room temperature was calculated. The original component set the average room temperature over the simulation time step to the outlet room temperature. A change was implemented to relay the room temperature at the end of the time step to the outlet room temperature.

3 COMPUTER MODEL NEW DESIGNS AND THEORIES

With the validity and capability of the TRNSYS model understood, the model was used to help estimate how changes to the original heat pump system would affect heat pump energy use. Two major theories were investigated along with several secondary theories. The first of the two major theories studied included how surface area and volume of the heat exchanger would affect heat pump performance. The other substantial theory investigated was how the use of solar collectors would affect the system. Other system parameters that were investigated included variations in how fluid flow rate, freeze protection, and night setback temperature difference would affect heat pump energy consumption.

3.1 Surface Area and Volume Investigation

Before any new designs for a heat exchanger could be pursued, the energy use of the heat pump based on heat exchanger volume and surface area needed to be determined. The PVC heat exchanger model (type 118) was used for this analysis. The theories stated in section 2.3.4.5 apply to this component. The area for heat transfer to the air and the area for heat transfer to the ground were specified to be 70% and 30% of the total surface area, respectively. This specification was made in the source code for the PVC heat exchanger version of this component. Table 3.1 shows the parameters used in this model, and are different than with the model containing type 117. The results of the simulation are presented in Figure 4.4.

Table 3.1 – Parameter Description for Type 118

Parameter No.	Value	Description
1	varies	Surface Area of heat exchanger (ft^2)
2	varies	Volume of fluid in heat exchanger (gals)
3	3.75	Specific heat of liquid (KJ/kg-°C)
4	1,048	Density of liquid (kg/m^3)
5	10	Heat transfer coefficient for water and heat exchanger (BTU/(hr-ft^2-°F) (**)
6	4.0	Heat transfer coefficient for air and heat exchanger (BTU/(hr-ft^2-°F) (**)

3.2 Impact of Solar Collectors

The choice to investigate solar collectors stemmed from the theory that changes in inlet fluid temperature to the water source heat pump would improve heat pump performance. Unglazed flat-plate solar collectors were determined to be the most likely type of solar collector that would help in this application because of their known performance in lower temperature (swimming pool heating) applications. The collectors would absorb heat in the winter and reject heat in the summer on cloudy days from the circulating fluid. In addition, these collectors cost approximately \$5-6/ft^2 (average materials cost from solar collector vendors).

The investigation into the use of solar collectors requires definitions of several other components in TRNSYS. The additional components used in this model are the Solar Radiation Processor, Solar Collector, Flow Diverter/Mixer, and Solar Collector Control. Figure 3.1 shows the basic flow path of the components for the model that includes solar collectors and PVC heat exchangers.

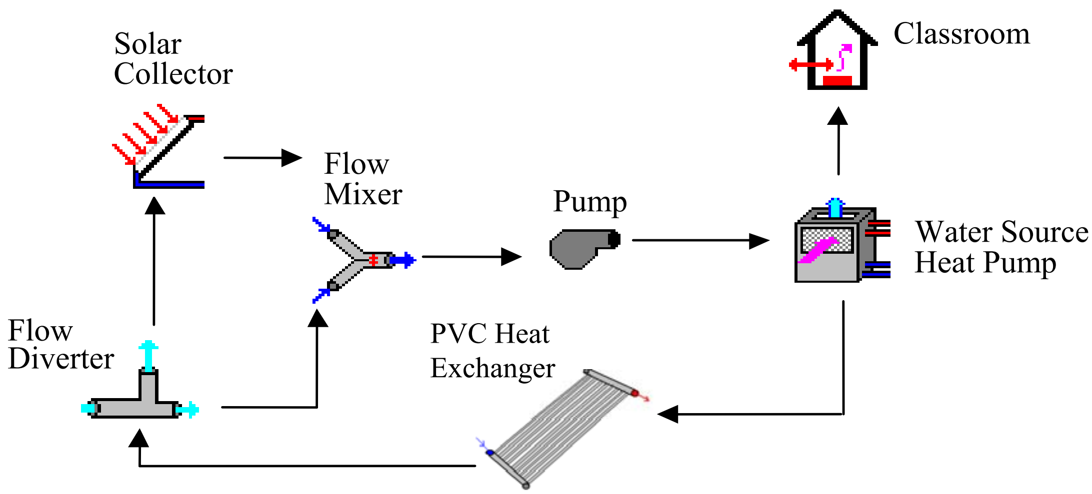

Figure 3.1 – Layout of TRNSYS model with Solar Collectors

3.2.1 Solar Radiation Processor

The Solar Radiation Processor, type 16, is used to interpolate radiation data and calculate several quantities related to the position of the sun. Its purpose is to convert data into a useful form that can be used by the Solar Collector component, type 1. Parameters 5 and 7 are specified for the Raleigh, NC. Inputs 6 and 7 are specified for mobile classrooms at DDES.

Table 3.2 – Parameter Description for Type 16

Parameter No.	Value	Description
1	2	Horizontal Radiation Mode- Reindl (Full Correlation)
2	1	Tracking Mode- Fixed Surface
3	3	Tilted Surface Radiation Mode- Reindl Model
4	1	Day of the year of the start of simulation
5	35.87	Latitude (degrees)
6	4,870.8	Solar Constant (kJ/(hr-m^2))
7	-3.783	Shift in solar time hour angle (degrees)
8	1	Enable Radiation Smoothing

Table 3.3 – Input Description for Type 16

Input No.	Value	Description
1	9,4	Global radiation on horizontal surface (kJ/m^2-hr)
2	9,5	Ambient temperature (°C)
3	9,10	Ambient relative humidity (%)
4	9,99	t_{d1} – time of last radiation data reading (hr)
5	9,100	t_{d2} – time of next radiation data reading (hr)
6	0.2	ρ_g – ground reflectance (common reflectance used Raleigh, NC)
7	14	β_i – slope of surface from ground (slope of roof) (degrees)
8	-30	γ_i – azimuth of surface or tracking axis (Collectors would be positioned 30° East of South) (degrees)
9	9,24	Radiation on horizontal surface at next time step- only if radiation smoothing is required (kJ/m^2-hr)

3.2.2 Solar Collector

The Solar Collector, type 1, is used to model the thermal performance of the unglazed solar collectors in the trial. User-supplied results from standard tests of collector efficiency are the driving force behind the Solar Collector component. The collector requires radiation data received from the Solar Radiation Processor, type 16. The purpose of this component is to predict the outlet fluid temperature from the collector.

Table 3.4 – Parameter Description for Type 1

Parameter No.	Value	Description
1	1	N_s - Number of collectors in series
2	14.7	A - Total Collector Area (m^2) (Area for 3 HC-40 collectors)
3	Cpl	C_{pc} - Specific heat of collector fluid (kJ/kg-°C)
4	1	Efficiency Mode- η vs. $(T_i-T_a)/I_T$
5	257.25	G_{test} - Glow rate per unit area at test conditions (kg/hr-m^2) (1)
6	0.838	a_o - Intercept efficiency (1)
7	65.88	a_1 - 1st order coefficient of the efficiency (kJ/(hr-m^2-°C)) (1)
8	0.2246	a_2 – 2nd order coefficient of the efficiency (kJ/(hr-m^2-°C^2)) (1)
9	2	Optical Mode- Use incidence modifiers from ASHRAE test $K_{\alpha\tau} = 1- \beta_0(S) - \beta_1(S)^2$ (1)
10	0.0316	β_0 - First-order angle modifier constant from ASHRAE test (1)
11	0.104	β_1 - Second-order angle modifier from ASHRAE test (1)

Note:
(1) Values are obtained from [14]

Table 3.5 – Input Description for Type 1

Input No.	Value	Description
1	11,3	Inlet liquid temperature (°C)
2	11,4	Inlet liquid flow rate (kg/hr)
3	9,5	Ambient air temperature (°C)
4	16,7	Incident Radiation (kJ/m^2-hr)
5	16,4	Total horizontal radiation (kJ/m^2-hr)
6	16,6	Horizontal Diffuse Radiation (kJ/m^2-hr)
7	0.2	ρ_g – ground reflectance (common reflectance for environment similar to Raleigh, NC)
8	16,10	θ - incidence angle (degrees)
9	16,11	β_i – slope of surface from ground (slope of roof) (degrees)

3.2.3 Flow Diverter/Mixer

The Tee-Piece, Flow Diverter, Flow Mixer, and Tempering Valve are all under the umbrella of type 11. Type 11 has ten modes operation, which allow it to model various types flow equipment. In the solar collector study a flow diverter and flow mixer were used to simulate a controlled three-way ball valve. These components operated from a signal received from a controller, type 98, to the appropriate component. In the parameters declaration, mode 2 specifies a controlled flow diverter, and mode 3 specifies a controlled flow mixer.

Table 3.6 – Input Description for Type 11, Mode 2

Input No.	Value	Description
1	104,1	Inlet liquid temperature (°C)
2	104,2	Inlet liquid flow rate (kg/hr)
3	99,1	Control function from solar collector controller (type 98)

Table 3.7 – Input Description for Type 11, Mode 3

Input No.	Value	Description
1	11,1	Inlet 1 liquid temperature (°C)
2	11,2	Inlet 1 liquid flow rate (kg/hr)
3	1,1	Inlet 2 liquid temperature (°C)
4	1,2	Inlet 2 liquid flow rate (kg/hr)
5	99,1	Control function from solar collector controller (type 98)

3.2.4 Solar Collector Control

The solar collector controller, type 98, was designed to model a photo sensor, which would control a mechanical 3-way valve. The construction of the component allowed the flow from the heat pump to either circulate through or to by-pass the solar collectors. The controller signal depended on what mode the heat pump was operating, and whether the sun was shining or not. In an attempt to gain full use of the solar collector, a condition statement was added to direct the fluid flow through the collector if the ambient air temperature was greater than the fluid temperature and the heat pump was in heating mode. Type 98 is a supplemental component, and the source code is included in Appendix C.

Parameters are not required for this component, but the inputs are provided in the Table 3.8 below.

Table 3.8 – Input Description for Type 98

Input No.	Value	Description
1	8,1	Control signal for 1st stage heating
2	8,3	Control signal for cooling
3	16,7	Total radiation on mobile classroom roof (kJ/hr)
4	104,1	Outlet fluid temperature from heat pump (°C)
5	9,5	Ambient air temperature (°C)

4 RESULTS AND DISCUSSION

4.1 Geothermal Heat Pump Study at Wilson Mills Elementary School

It was necessary to construct a TRNSYS model of the previous experiment to validate the calculations for the new design. This model was honed until it most accurately predicted the performance of the previous experiment. The theoretical data in Figure 4.1 is organized to correspond directly to the days that actual heat pump energy use was recorded. The model correlates very closely to recorded data during the spring and fall months. When divergences do occur, the majority of the differences in experimental and theoretical energy use can be attributed to differences in actual Degree Days and TMY Degree Days as shown in Figure 4.2. The two exceptions occur during the summer months. In the summer 2000, the Degree Days difference accounts for some of the divergence, while the remainder could be attributed to larger infiltration and internal loads due to increased student activity levels. In the summer 2001, the classroom was not used for summer school, as it was the previous summer. The disparity in energy use in the winter months can be explained by differences in Degree Days. The predicted annual energy use was 9,337 kWh, and the actual annual energy use during the previous study totaled 10,783 kWh. This underestimation should be taken into account when viewing the subsequent model predictions.

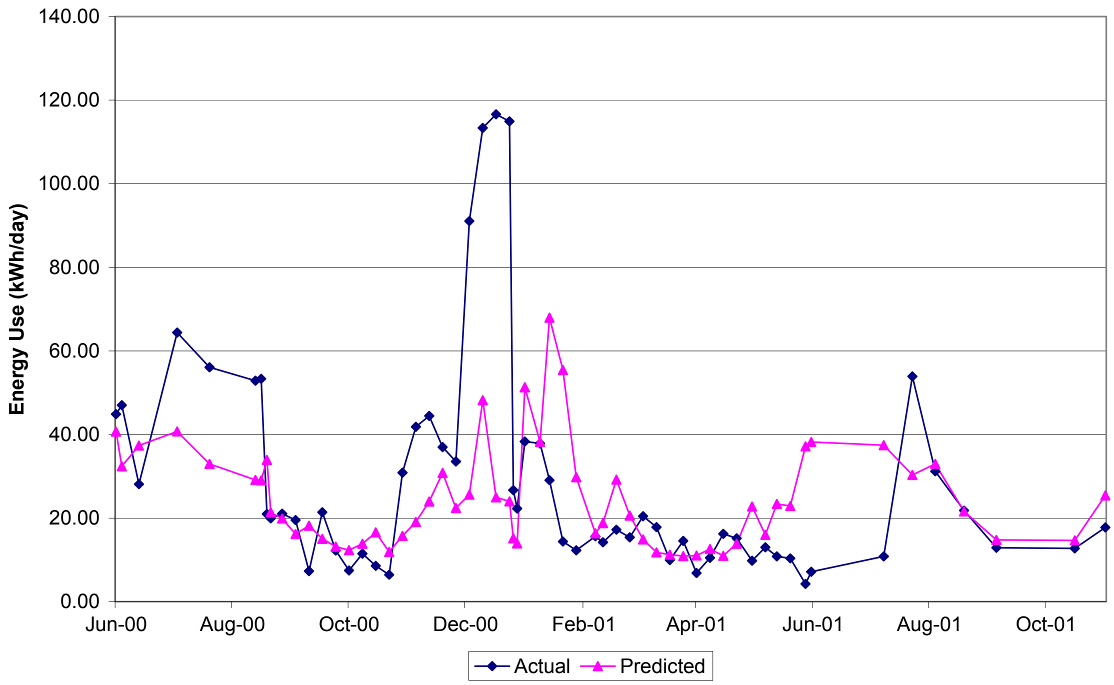

Figure 4.1 – Predicted Vs. Experimental Energy Use for the Geothermal Heat Pump

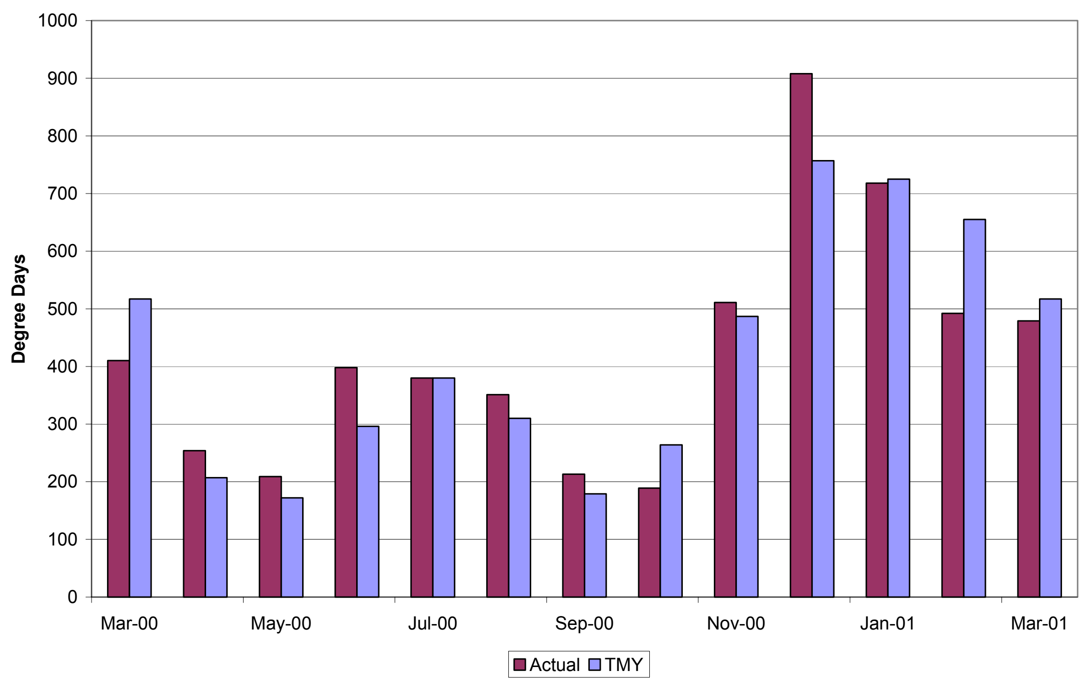

Figure 4.2 – TMY Vs. Actual Degree Days for the WMES Experimental Study

4.2 Air Source Heat Pump Study at Wilson Mills Elementary School

As mentioned in Section 2.4.1, TRNSYS predictions for the air source heat pump are not compared to the available experimental data. The air source heat pump model does not account for the defrost cycles that occurred during cold ambient air temperatures. However, the TRNSYS model of an air source heat pump will be useful in the present work because outdoor thermostats are installed on all air source heat pumps in Wake County, NC. Bard Technical Support claims the thermostat saves energy by keeping the heat pump from cycling between the heating mode and the defrost mode. In an effort to make the TRNSYS model more realistic, controls were implemented to force the air source component to engage electrical resistance heating and not use the heat pump if ambient air temperatures were less than 30°F. The daily energy use of the geothermal model at WMES and an air source model with an outdoor thermostat are compared in Figure 4.3. The TRNSYS model predicted the electrical energy use of the air source heat pump to be 13,498 kWh/yr, a 45% increase over the water source heat pump.

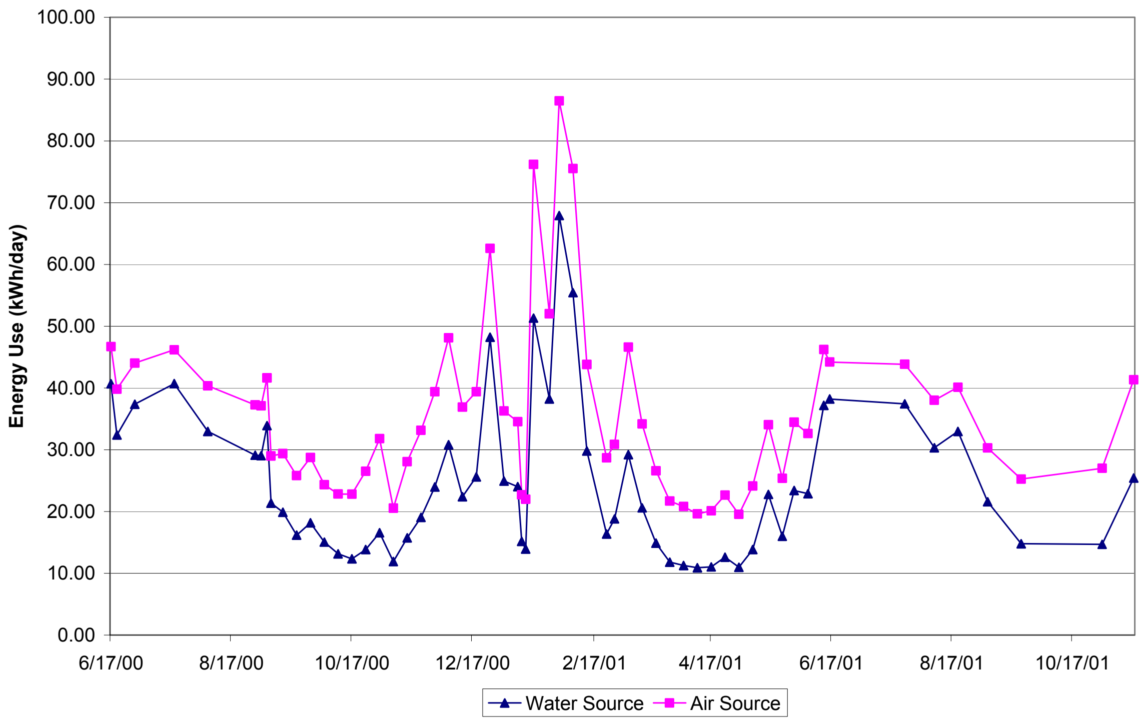

Figure 4.3 – TRNSYS Comparison of Water Source and Air Source Heat Pumps at WMES

4.3 New Designs and Theories

4.3.1 Surface Area and Volume Investigation

The trends predicted in Figure 4.4 are the fundamental reason for using a heat exchanger design in place of the bladders. The predicted annual energy use of the heat pump in the bladder system (with 2,000 gallons and 640 ft^2 of surface area) is included for comparison. The difference in energy use is due to different percentages of surface area allocated for heat transfer to air and water between the bladder and heat exchanger models. Furthermore, the heat transfer coefficient for the heat exchangers and air is twice as large as the heat transfer coefficient for the bladders and air. The heat transfer for the heat exchanger was affected much more by changing surface area rather than by modifying the volume.

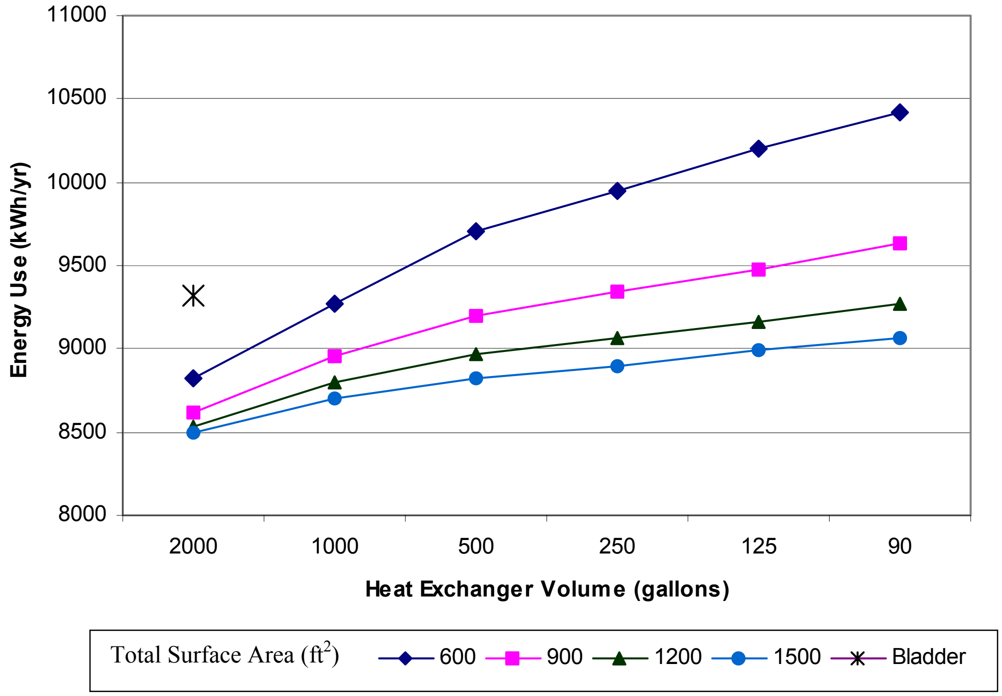

Figure 4.4 – Water Source Heat Pump Annual Energy use for Various Surface Areas and Volumes

4.3.2 Solar Collector Investigation

The interest in modeling solar collectors as an energy savings measure motivated the decision to use TRNSYS. However, the TRNSYS code proved to be useful for modeling systems without solar collectors. The predicted electrical energy savings yielded by the solar collectors was much less than expected. The model predicts that solar collectors provide more savings for a system with bladders than for a system with PVC heat exchangers. Still, the system with solar collectors and PVC heat exchangers uses less energy. The larger storage capacity of the bladders enabled them to store the energy gained by the solar collectors. As seen in Figure 4.5 and Figure 4.6, the temperatures in the heat exchangers fluctuate much more with the ambient air temperature than the bladder temperatures do. The average amplitude of temperature fluctuation in the bladders and PVC heat exchangers was predicted to be about 10°F and 20°F, respectively.

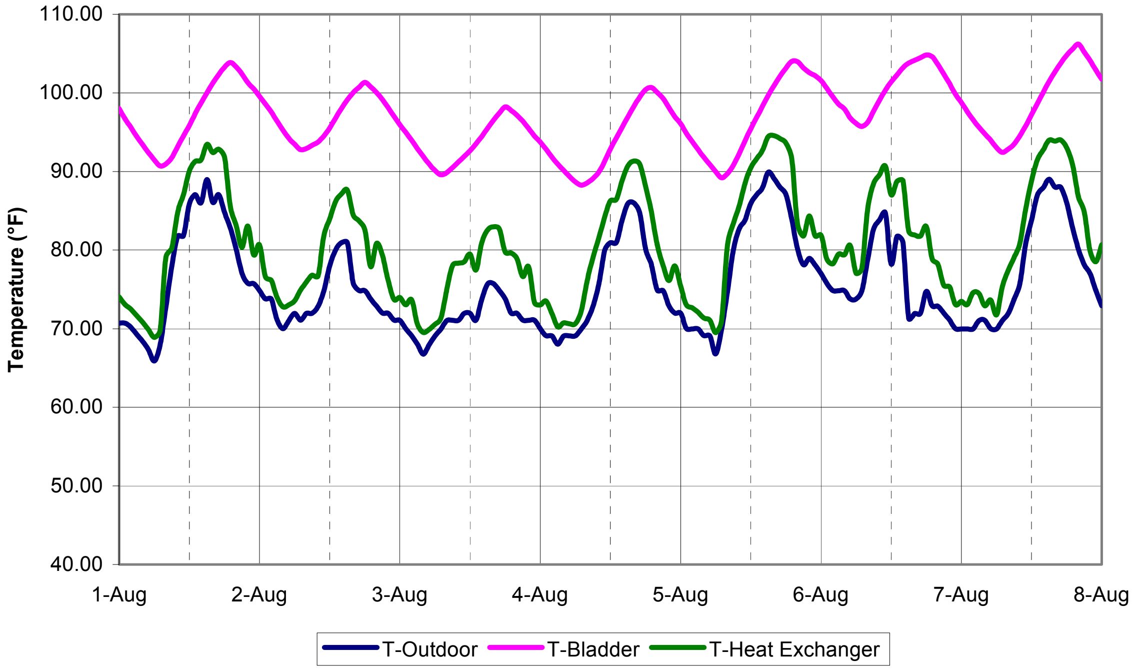

Figure 4.5 – Typical Predicted Summer Temperatures

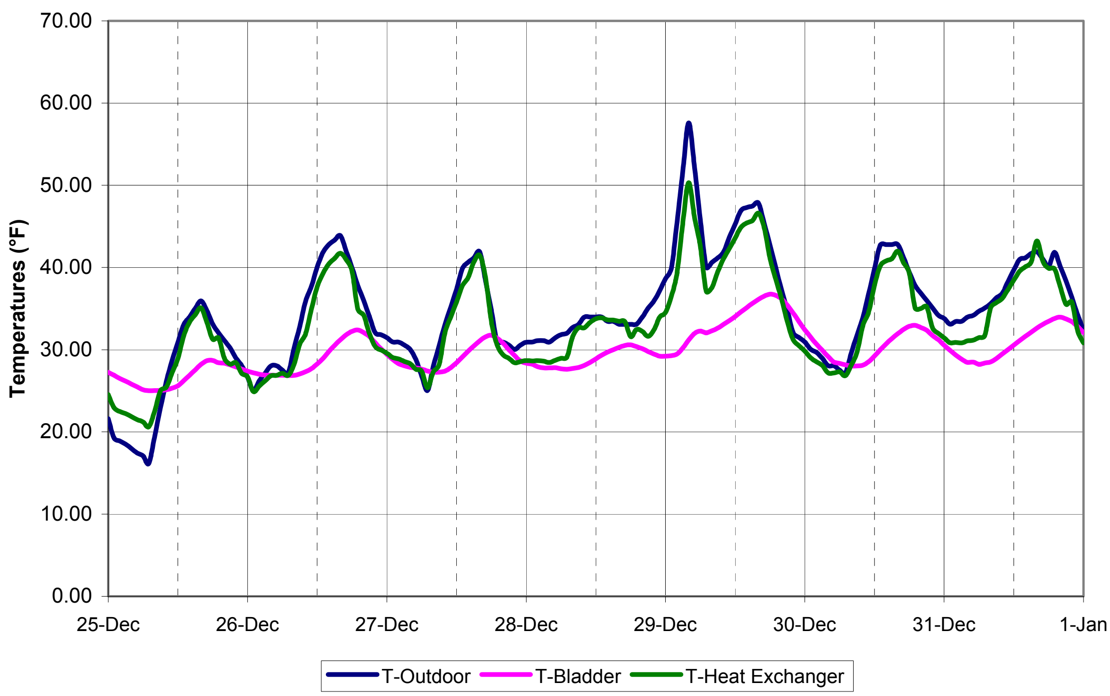

Figure 4.6 – Typical Predicted Winter Temperatures

Figure 4.7 shows the energy savings from using solar collectors with bladders and with heat exchangers. Despite the increased energy savings from using the solar collectors with the bladders as opposed to heat exchangers, the predicted payback period for the three solar collectors being modeled would greatly exceed ten years. Further evidence of the solar collectors ineffectiveness can be seen by the fraction of time the fluid flows through the solar collectors. The simplest control strategy causes fluid to flow though the collectors when the sun is shining and the heat pump is heating, or when the sun is not shining and the heat pump is cooling. In this case the solar collectors are used only about 15% of the time. More complex strategies increase the time that the collectors can be used, but do not have much effect on energy usage.

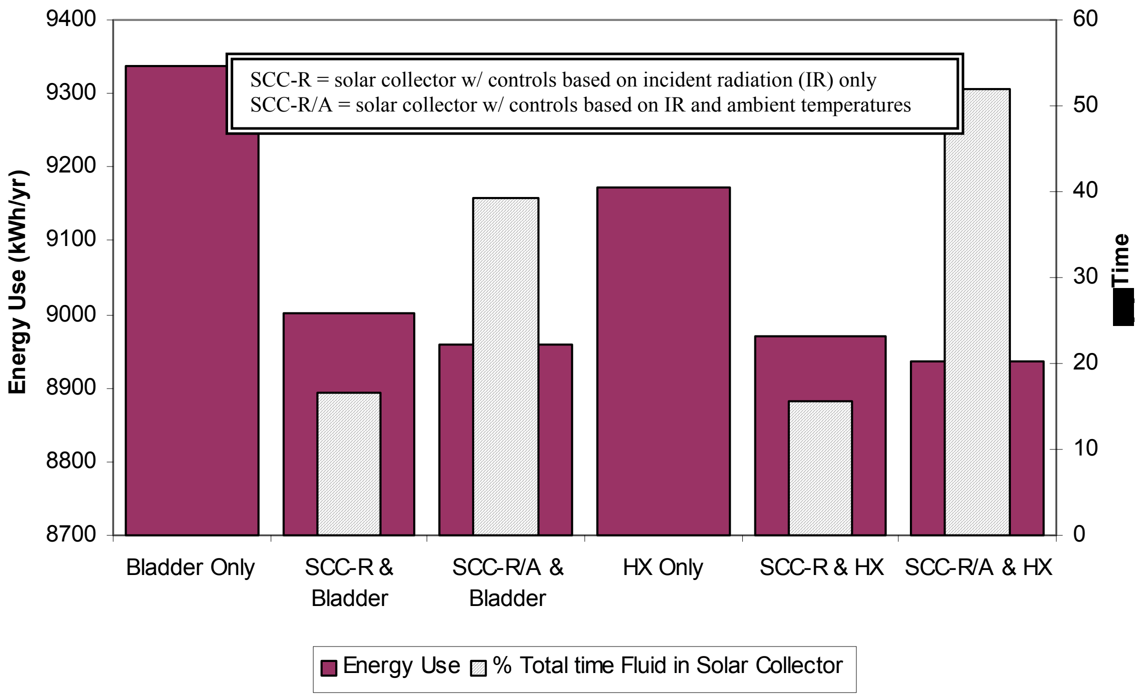

Figure 4.7 – Geothermal Heat Pump Energy Use with Solar Collectors

Another issue to be examined is the optimum orientation of the solar collectors. For appearance, it is desirable to have the collectors lay flat on the roof. Measurements were taken at DDES to determine the slope of the classroom roof, β, and the azimuth, γ, of the most southern facing roof. β was measured to be about 14°, and γ was measured to be about 30° East of South. The optimum case for solar collector orientation was determined to be at a slope of 47 degrees and facing south. According to the Florida Solar Energy Center, a solar collector performs best facing South, but a 30° variation in either direction will still allow the solar collector to capture 90 percent of the maximum solar energy available [15]. The recommended optimum slope a solar collector should be positioned for winter use is the location's latitude plus 15 degrees. In Raleigh, NC this would equate to about 51 degrees. The TRNSYS model predicted that energy savings associated with changing the orientation of the collectors are very small. Thus, the solar collectors could lie flat on the roof at DDES.

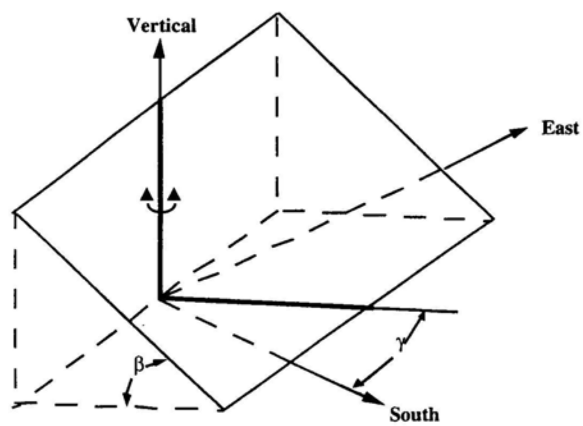

Figure 4.8 – Surface Orientation for Solar Collectors in TRNSYS [6]

4.3.3 Other Variables

The other parameters and investigations explored to conserve energy included a flow rate analysis and an analysis of the effect of different types of freeze protection. In addition, the effect of an automated night setback temperature difference on heat pump energy use was checked.

Parametric analyses were performed to determine whether an optimum flow rate exists, and to confirm that it coincides with the manufacturer's recommended flow rate of 7 gpm. The simulation was carried out over the range of flow rates for which Bard provided data [4]. The optimum flow rate for heat pumps with solar collectors and heat pumps without solar collectors was 13 gpm, which differs from the manufacturer's recommended flow rate. As seen in Figure 4.9, the differences in energy consumption between a system with and with out solar collectors dwindle as the flow rates are increased.

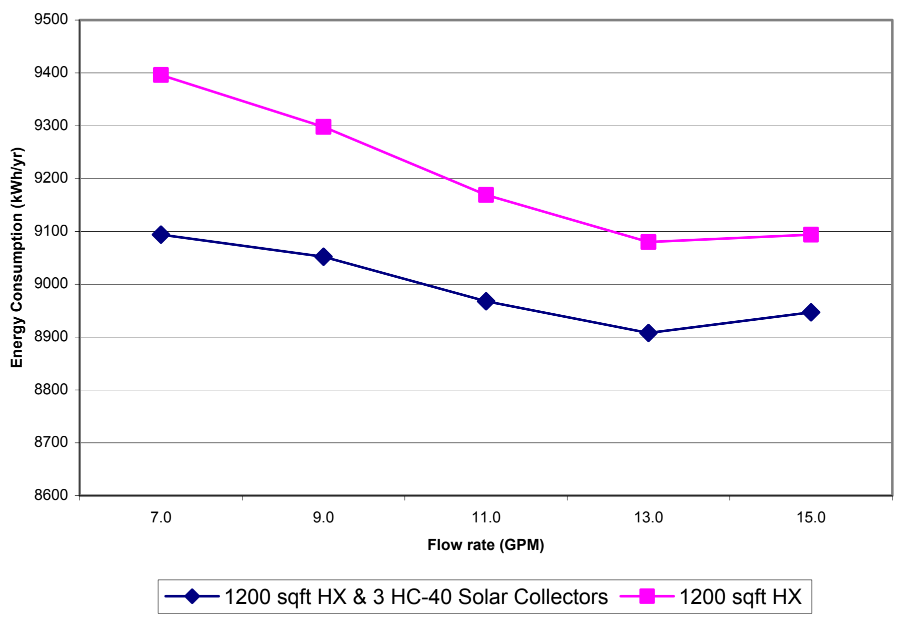

Figure 4.9 – Heat Pump Annual Energy Use for Flow Rates Provided by Bard

Three different types of fluids were modeled for the same heat pump: Pure water, 25% ethylene glycol, and 25% brine solution. The system containing the brine solution properties performed the worst. The system containing the ethylene glycol solution yielded about a 1/2% energy savings when compared to the brine solution. The system containing pure water performed the best (about 1/3% savings over the ethylene glycol solution). The results of the simulations were in line with expectations due to the variations in specific heat of the three solutions.

All mobile classrooms in Wake County, NC are equipped with radio-controlled night set back controls. During the winter months, the thermostat is supposed to be set back to 55°F typically from about 4:00 p.m. until 6:00 a.m. In the summer months, the heat pump is supposed to be turned off during a similar time frame. Some of the night setback thermostat controls were observed not functioning as previously described during visits made to mobile classrooms. In addition, these controls can be manually overridden with the thermostat inside the classroom. The night setback feature was modeled in TRNSYS to predict the energy savings generated by the use of these controls for a geothermal heat pump system with PVC heat exchangers and for an air source heat pump. The results generated by the TRNSYS model, which are summarized in Figure 4.10, show that the night set back controls reduce electrical energy consumption of about 50%.

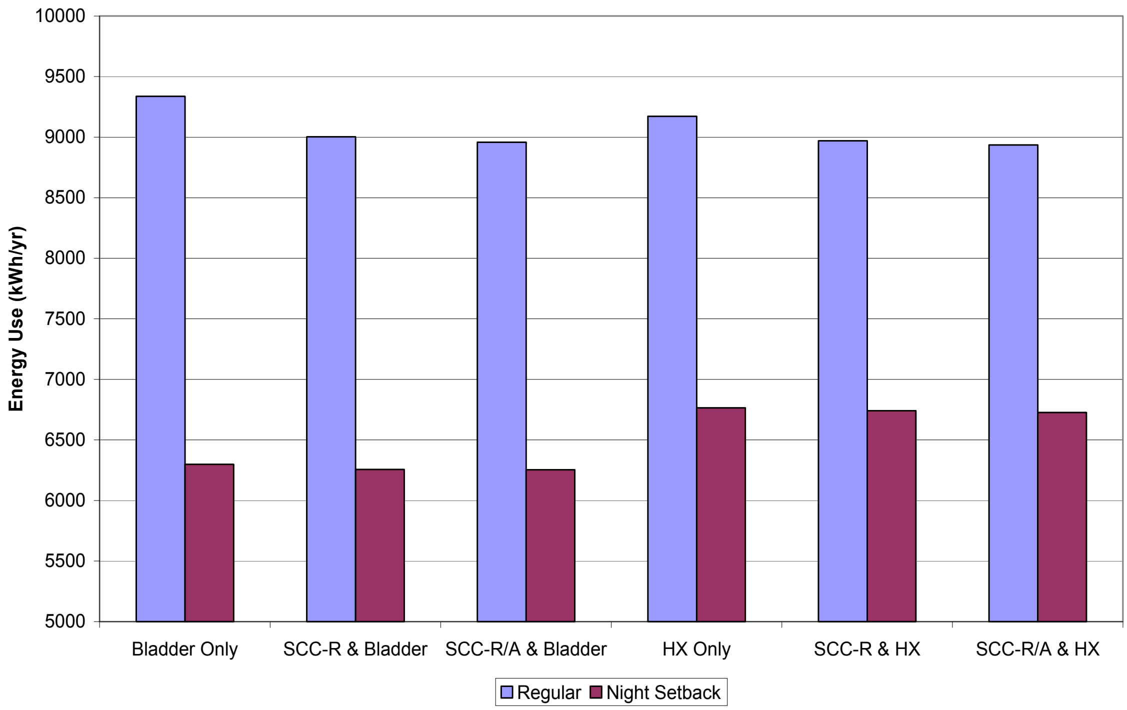

Figure 4.10 – Energy Use With Night Setback Controls

4.3.4 Optimum System

An optimum system was designed based on the analysis presented in the previous sections and on what the actual site could handle (in terms of space under the classroom). The optimum system was determined to have a surface area of 1200 ft^2 and a volume of 120 gallons. The flow rate of the ethylene glycol solution in the system was prescribed to be 13 gpm. Figure 4.11 compares the daily energy use of the original bladder system and the PVC heat exchanger system. The total predicted electrical energy consumption of the optimized heat pump system was 9,069 kWh/yr, a predicted energy savings of 3% compared to the original system. The optimum system used much less energy in the summer months, but used slightly more or about the same throughout the rest of the year.

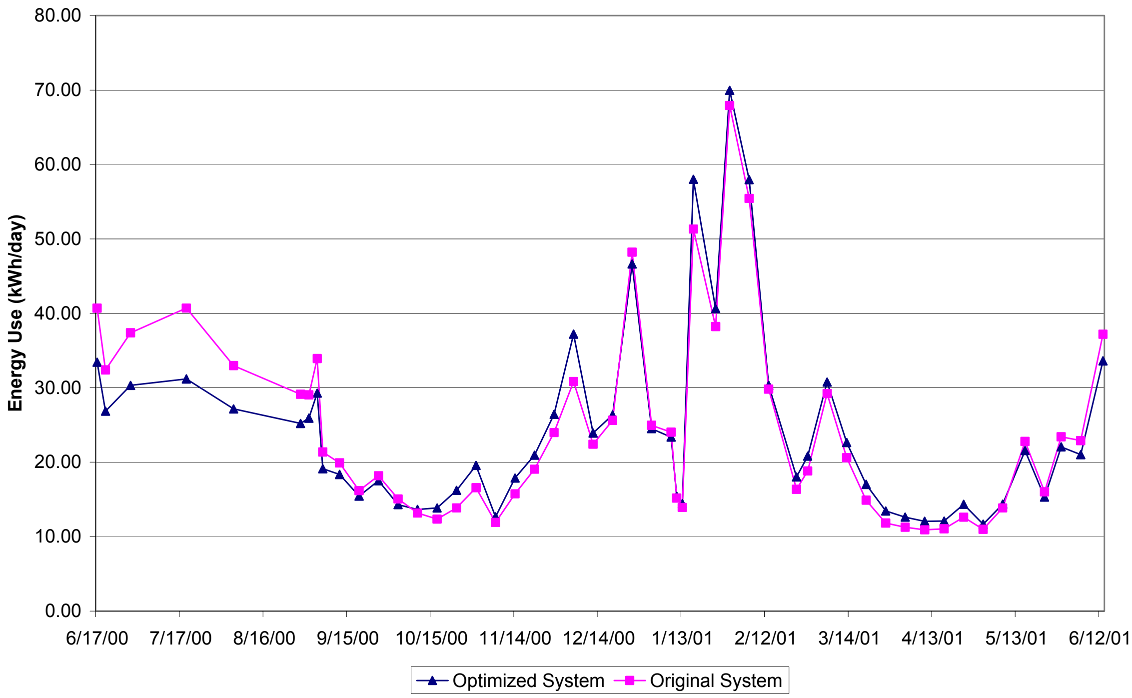

Figure 4.11 – Annual Energy Comparison of Original and Optimized Systems

5 PVC HEAT EXCHANGER DESIGN

The problems with the large bladders described in Section 1.2 indicated a need to pursue alternative designs for an earth-coupled heat exchanger. As shown in Figure 4.4, the volume of the heat exchanger is not as important as the surface area in reducing heat pump energy consumption. The PVC heat exchangers should be more durable and less likely to leak than the bladders were. The significant reduction in volume of the PVC heat exchangers will allow an ethylene glycol solution to be used instead of a brine solution, eliminating the problems caused by saltwater. A less-expensive, direct drive pump can be used to circulate the ethylene glycol solution. In addition, the ethylene glycol solution can be reused when the mobile classrooms are moved, further reducing the overall cost of the system.

The heat exchanger that evolved as a result of this work is made from PVC pipe, and is comprised of a header and footer, and 1/2 in PVC pipe. A desired velocity flow rate of approximately 1 ft/sec in each tube and space limitations influenced the design of the heat exchanger. The heat exchanger was designed to be approximately 2 ft wide, and run the length of the space it is intended to occupy. The heat exchangers were designed to this width to ease manufacturing and handling of the headers, and to maintain the desired velocity of about 1 ft/sec. An additional benefit from using the multiple 2 ft heat exchangers would be realized if a replacement were required. The headers are connected with a temporary fitting, which was selected because it would allow for easier movement of the heat exchangers and it would allow small sections to be replaced if a leak developed. Figure 5.1 shows the design of the heat exchanger header.

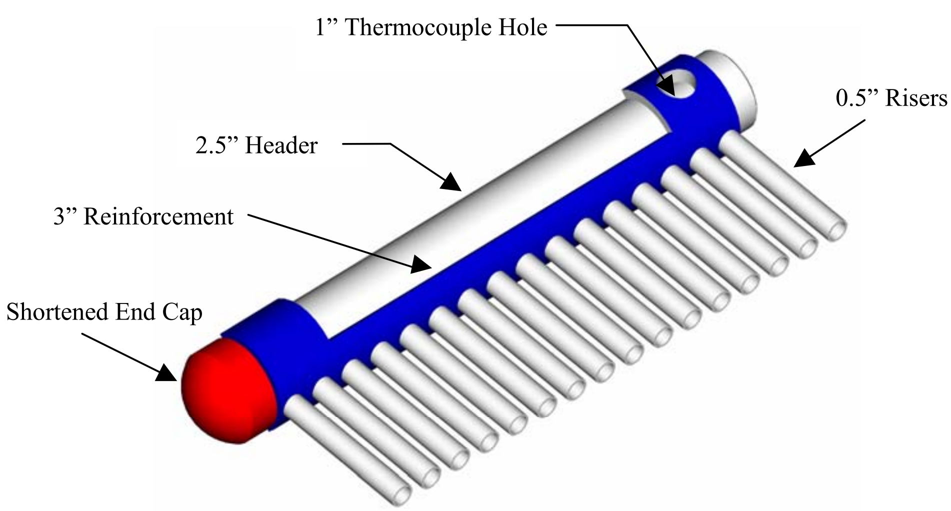

Figure 5.1 – PVC Heat Exchanger Header Design

The available area for heat exchanger placement varies between mobile classrooms. Figure 5.2 shows a proposed layout that would be for a typical classroom.

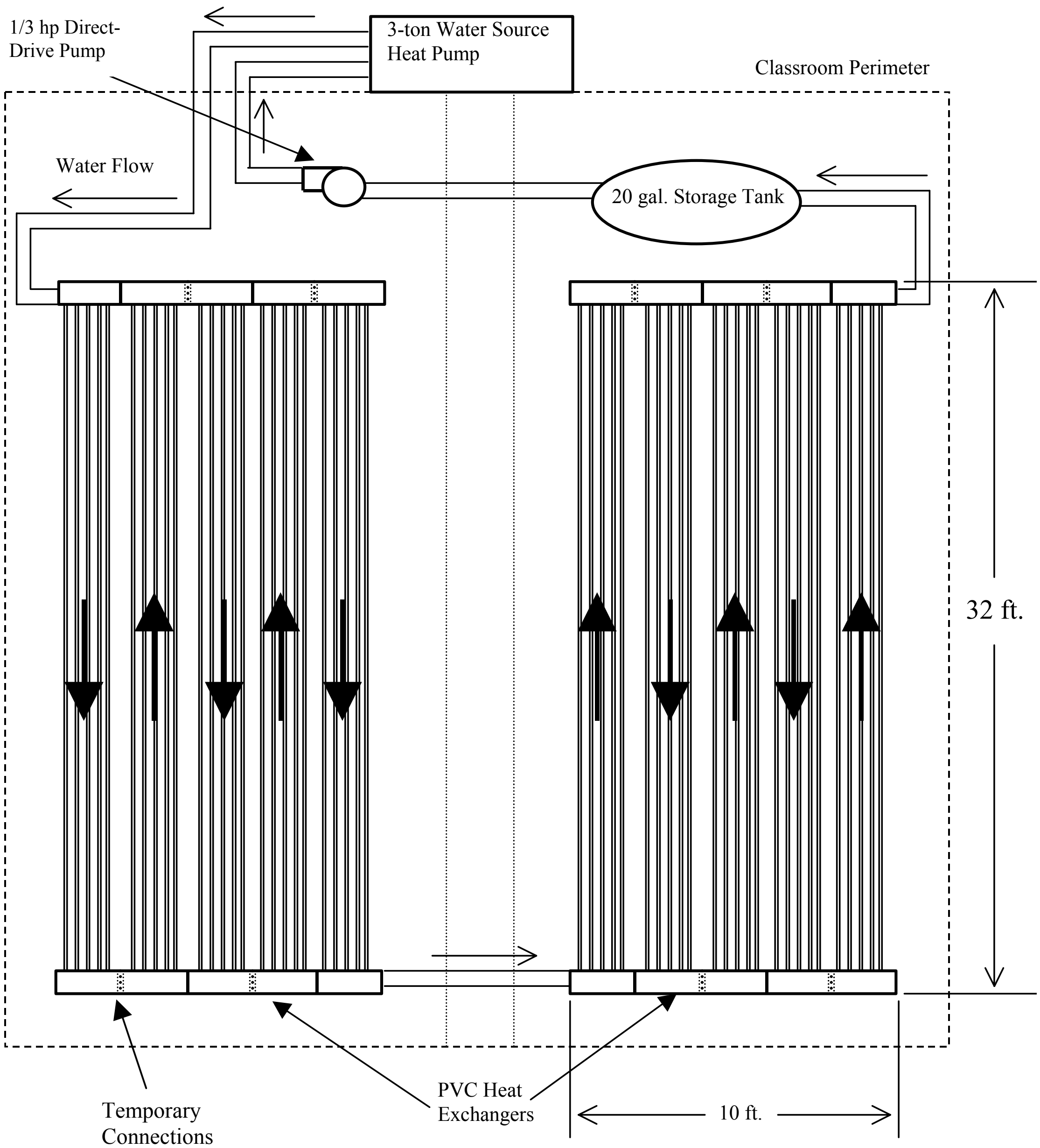

Figure 5.2 – Layout of Geothermal Heat Pump System

5.1 Pressure Drop Calculations

The heat exchanger was designed to have minimal pressure drop. The diameter of the headers ensured the pressure drop in each header could be neglected during the overall pressure drop calculations. Based on fifteen or sixteen tubes per heat exchanger and an optimum system flow rate of 13 gpm, the velocity through each of the tubes in the heat exchangers is about 0.75 ft/sec. Velocity of this magnitude in the 1/2 in PVC pipe yields a Reynolds number of about 2,800. For a tube of length 30 ft, the pressure drop, ΔP, was calculated to be 0.18 ft of water using the following equation:

$$\Delta P = 1/2*1/g*1/\rho*v^2*f*L/D$$

where:

g	=	acceleration due to gravity (32.2 ft/sec^2)
ρ	=	density of water (62.4 lbm/ft^3)
v	=	velocity per tube (0.75 ft/sec)
f	=	friction factor from Moody Chart, 0.04 [16]
L	=	Maximum length of pipe, 30 ft
D	=	Inner diameter of pipe, 0.69 in

The other source of pressure drop occurs with the entrance and exit losses from the 1/2 in pipe to the headers. These were considered sudden expansions and sudden contractions. Each tube contains a sudden expansion and sudden contraction, which together only equate to about 0.005 ft of water [16]. Since the heat exchangers are joined together by the same diameter tube as the headers the pressure drop between headers is assumed to be negligible. The total pressure drop across 10 heat exchangers 30 ft long is predicted to not exceed 2 ft of water.

5.2 Material Specification

The heat exchanger header is made of 2.5 in SCH-40 PVC pipe and 3 in SCH-80 reinforcement where holes for the risers are to be drilled. The reinforcement is made of quarter sections of the circumference of the 3 in SCH-80 PVC pipe. Using results from initial experiments and prototypes, a decision was made to add reinforcement to increase bonding surface area between the headers and risers. The minimum wall thickness for 2.5 in SCH-40 PVC and 3 in SCH-80 PVC is 0.203 in and 0.300 in, respectively. By adding the reinforcement, the wall thickness was increased by approximately 60%. These two pipe dimensions were selected because the inner diameter of the 3 in pipe was approximately equal to the outer diameter of the 2.5 in pipe, which ensured a natural fit. Additionally, the pressure drop in the header would be negligible using this size header. This extra step significantly decreases the likelihood that the heat exchanger will leak at the riser or thermocouple holes.

The reinforcements were assembled using the clamps as shown in Figure 5.3. Figure 5.3 shows the assembly of the reinforcements where a thermocouple would be mounted. This same procedure is performed for the larger reinforcement where the risers fit, except that the glued surfaces were immediately placed in a vice (centered), and allowed to bond for approximately forty-five minutes.

Figure 5.3 – Assembly of Header Reinforcements

The tubes of the heat exchanger were made from 1/2 in SDR-13.5 PVC pipe. This pipe was selected rather than 1/2 in SCH-40 for its lower cost, lower thermal resistance to heat transfer (thinner wall), and greater volume capacity. The result of using SDR-13.5 PVC instead of SCH-40 PVC yielded a cost savings of 30% and a volume capacity increase of 33%. Operating pressures should not exceed 50 psig in the system assuming flow rates do not exceed 15 gpm. As seen in Figure 5.4, if the operating pressure was 50 psig, the heat exchangers have a factor of safety of two at 125°F.

An additional 1 in hole was machined into the header assemblies so that temperatures could be monitored throughout the network of heat exchangers. Temperatures should be collected at regular intervals during this experiment so that the heat transfer in the heat exchangers can be better understood. 2 in pieces of 1 in diameter PVC SCH-40 are glued into the headers just as the 1/2 in risers were. Thermocouples are to be mounted in the 1 in diameter tubes. This feature would not be required in future installations.

The risers from each header will be joined by standard SCH-40 1/2 in couplings and standard 10 or 20 ft lengths of 1/2 in pipe. The 20 ft lengths contain a belled end, which reduce the number of couplings required. The use of standard PVC lengths and couplings will simplify the assembly of the heat exchangers during installation, and help to keep costs low.

The headers will be connected to each other via 3 in lengths of nitrile (Buna-N) hose and stainless steel worm gear hose clamps. Nitrile (Buna-N) hose was chosen because it will not deteriorate under the temperature range expected in the system (20°F - 120°F). The hose has "A" rating for transporting ethylene glycol (antifreeze) solutions [18].   This method of attachment is a much cheaper alternative to using 2.5 in PVC unions.

5.3 Manufacturing and Materials Cost

The holes for the 1/2 in PVC pipe were made in the header with a 27/32 (0.844) in diameter drill. The holes for the 1 in PVC pipe were made with a 1 5/16 (1.3125) in diameter drill. The spacing between the centers of the 27/32 in holes is 1.25 in. To allow for two additional tubes in the heat exchanger the couplings and end caps are shorted by 1.10 in. The end caps are SCH-40, and therefore can be shorted since operating pressures are much less than the maximum pressure of SCH-40 fittings.

Table 5.1 – Dimensions for Selected Heat Exchanger Components

Nominal Pipe Size (in.)	Outer Diameter (in.)	Ave. Inner Diameter (in.)	Min. Wall Thickness (in)	Max Working Pressure @ 73.4°F (PSI)
2.5 (SCH-40)	2.875	2.445	0.203	300
3 (SCH-80)	3.500	2.864	0.300	370
1/2 (SDR 13.5)	0.84	.700	0.062	315
1 (SCH-40)	1.315	1.033	0.133	450

Data presented in the Table 5.2 represents the price for two sets of 1200 ft^2 of PVC heat exchangers. The costs represent the North Carolina State University cost to build the heat exchangers for two experiments. Materials were purchased from Lowes, Inc. or Murray Supply Company in Raleigh, NC. The total cost in Table 5.2 is substantially higher than what the heat exchangers would cost if they were mass-produced. Retail price was paid for the PVC materials, and the labor time should be able to be reduced substantially in a production environment or even with useful fixtures or jigs.

Table 5.2 – Material and Assembly Cost of PVC Heat Exchangers for Two Trailers

Component	Material Cost ($)	Labor Time (hrs)			Labor Cost {$10/hr} ($)	Total Cost ($)
		Cut	Clean	Cement (1)		
2.5” Pipe	70	1	1	0	20	90
3” Pipe	70	4	1	9	140	210
2.5” Cap	75	1	0.5	0	15	90
2.5” Connection	108	1	0.5	0	15	123
1/2” Risers	30	2.5	1	13.5	170	200
1/2” Pipe	660	0	0	24 (2)	240	900
1/2” Coupling	90	0	0	0	0	90
Cement/Primer	50	0	0	0	0	50
Header Machining	0	15	0	0	150	150
Total	1,153				750	1,903

Notes:
(1) Cement time includes cementing and priming process time.
(2) Time is an estimate that includes final assembly at location.

The connection cost represents the cost of using the nitrile hose and hose clamps. If a standard 2.5 in union is used the connection cost would be $27 as opposed to $4.5. If the nitrile hose does not meet reliability standards, an alternative design would be required to reduce the cost of the headers. Section 5.5 provides detail regarding the alternative design that is tested along with the current design.

5.4 Temperature Limitations on PVC Pressure Rating and Impact Strength

PVC is pressure rated at 73.4 °F and has an operating range from 0°F to 140°F [18]. As the service temperature increases, the maximum pressure rating is reduced. As the service temperature falls below 73.4 °F the pressure capacity of PVC increases to a higher level than its pressure rating. It is practice to consider this increase as an unstated factor of safety. However, when handling PVC at very cold temperatures special caution must be taken because PVC pipe's impact strength is lower than its impact strength is 73.4 °F. Impact strength can vary significantly between manufacturers because different plasticizers are used in PVC pipe production [18]. This problem was evident in the Sanderson PVC pipe ordered for this project. The Sanderson brand pipe is much more brittle around 50°F than Silver-Line®. ASTM standard 2241 specifies that the impact resistance requirement for 1/2 in PVC pipe is 10ft-lbf [19].

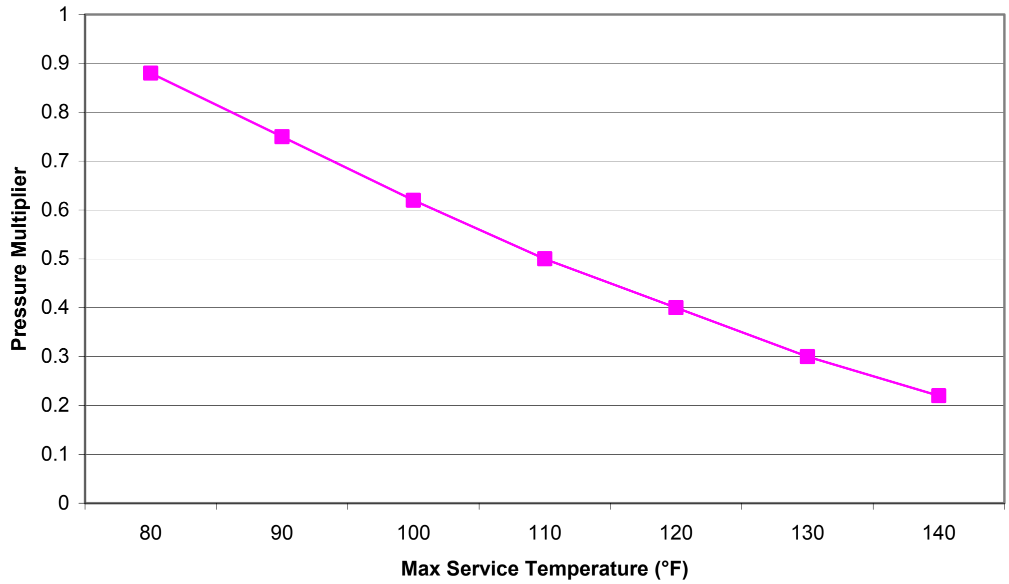

Figure 5.4 – Thermal De-rating Factors for PVC Pressure Pipe [60]

5.5 Alternative Header Design

The high cost of 2.5 in SCH-40 and 3 in SCH-80 PVC pipe and couplings encouraged the design of a cheaper heat exchanger. The same layout and header design was used, but 2 in SCH-80 PVC was used in place of 2.5 in SCH-40 and 2.5 in SCH-80 PVC was used in place of 3 in SCH-80. If the nitrile hose connections are not satisfactory connectors, and PVC unions are used, the overall cost of the heat exchangers could be reduced by 38%. Otherwise, the cost of the redesigned system would be about the same as the current design. Pressure drop in the headers is still negligible. However, the pipes did not fit together as well as the first header design did. Some gaps between the 2.5 in reinforcement and 2 in header can be seen and are estimated to be about 0.005 in. The gaps are present where direct pressure was not applied during the mating of the reinforcement and header. The same assembly procedure was used for the alternative design that was used for the primary header design. This is a concern with the implementation of the secondary header design, and should be closely monitored during the study. If a problem does occur, a possible solution could be to use a fixture to apply uniform loads over the entire reinforcement as it is being glued to the 2 in pipe.

6 CONCLUSIONS AND RECOMMENDATIONS

A TRNSYS computer model of a mobile classroom with a water source heat pump was constructed to predict the effects of proposed design modifications. Comparisons of predicted performance to experimental data from a previous project were used to validate the computer model. The computer model was then used to evaluate proposed design modifications. Parametric analyses showed that solar collectors are not cost effective in this system. However, the model made it possible to design PVC heat exchangers to replace large bladders employed in previous designs. The heat exchangers are expected to improve energy efficiency, to reduce initial costs, to increase mobility (in the event the system is moved), and to improve reliability.

The Wake County Public School System has agreed to allow two water source heat pumps and two air source heat pumps to be tested for a period of one year. Trailers with similar occupancy patterns have been identified at Davis Drive Elementary School, and initial data has been collected to confirm that building loads on selected trailers are similar. Heat pump designs developed in this work will be tested throughout the following year.

The predicted and actual water source heat pump electrical energy use correlated as well as could be expected using actual data and TMY data. Thus, the predictions and conclusions presented in this study should provide significant insight into the actual behavior of the enhanced geothermal heat pump system. The redesigned system includes a PVC heat exchanger with 25% ethylene glycol solution as the circulating fluid that should be pumped at an optimum flow rate of 13 gpm.

The redesigned geothermal heat pump system will be installed at Davis Drive Elementary School and will be monitored for a test period of one year. The test is designed to compare the heat pump performance under a wide variety of weather conditions. Data will be collected to determine the system's actual economic and physical performance.

The analysis raised new questions regarding the optimization of a geothermal heat pump with aboveground storage. During the experimental study temperature in the PVC heat exchanger should be closely monitored so that heat transfer to the heat exchanger can be more accurately modeled. Anemometers should also be installed in multiple locations underneath the mobile classroom to monitor wind velocities around the heat exchangers. Methods and means for mass-producing the PVC heat exchanger headers should be considered. With the performance of the original system known and the promise of this experiment, strategies should be developed to market the heat pump system.

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8 APPENDICES

8.1 Appendix A: TRNSYS Model of Geothermal Heat Pump System with PVC Heat Exchanger only

```
VERSION 15.0


ASSIGN C:\trnsys15\Heat_Pump_Project\MODEL915.LST    6
ASSIGN C:\trnsys15\Weather\RALEIGH.DAT    10
ASSIGN C:\trnsys15\Catalog_Data\bldloads\eric\bldloadsi_novent.dat  18
ASSIGN C:\trnsys15\Heat_Pump_Project\FINAL\MODEL915_HX_hrly.OUT    11
ASSIGN C:\trnsys15\Heat_Pump_Project\FINAL\MODEL915_HX_daily.OUT    13
ASSIGN C:\trnsys15\Heat_Pump_Project\FINAL\MODEL9WEEKLY15.OUT    12
ASSIGN C:\trnsys15\Catalog_Data\GHPs\WSHP\Samp_Cc.dat    14
ASSIGN C:\trnsys15\Catalog_Data\GHPs\WSHP\Samp_Hb.dat    15
ASSIGN C:\trnsys15\Catalog_Data\GHPs\WSHP\Samp_CTtrane.dat   16  ASSIGN
C:\trnsys15\Catalog_Data\GHPs\WSHP\Samp_HTa.dat   17
* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *


* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *

CONSTANTS 7
timestep =1/20
CAP = 4000
FLOW =3038 ! = 10GPM   2567 3336.5
rhol =1028.6  !1028.6 ethylene glycol, 1130 brine 10 deg F freeze pt.
cpl =  3.75   !3.75 ! 25% antifreeze, 75% h20, 3,48 brine 10 degree F freeze pt
timestart= 0 !2880 !0 !3960.
timestop= 14000 !7500.


SIMULATION timestart timestop  timestep
WIDTH 80

LIMITS 100 50
TOLERANCES -.01 .01

UNIT 9 TYPE 89 TMY DATA READER
PARAMETERS 2
-1  10

UNIT 29 TYPE 9 BUILDING LOAD READER
PARAMETERS 9
-1  !mode
```

```
0  !header lines to skip
1 !number of columns to read
1   !data file timestep
-1   ! >0: interpolate this column
1   !multiplication factor
0   !addition factor
18  !logical unit for data read
-1  ! >0: formatted read statement


********************MODEL COMPONENTS**************************

EQUATIONS 1
TIMESTART1=TIMESTART+1

UNIT 14 TYPE 14 FORCING FUNCTION FOR RADIATION
PARAMETERS  10
0 1 2160 1 2160 0 7000 0 7000 1

EQUATIONS 2
GSR= [9,3]*[14,2]
GSRNEXT=[9,24]*[14,2]

UNIT 16 TYPE 16 SOLAR RADIATION PROCESSOR USING Reindl model
*OPTIMUM ANGLE IS 45.7,*SLOPE OF ROOF=14, AZIMUTH=-30
PARAMETERS 8
2   1   3   timestart1 35.87 4870.8  -3.783  1
INPUTS 9
*GSR  9,5  9,10  9,99  9,100  0,0  0,0   0,0   GSRNEXT
9,3  9,5  9,10  9,99  9,100  0,0  0,0   0,0   9,24
 0       0   0       0       0        0.2  14 -30   0

UNIT 99 TYPE 98 CONTROLLER FOR COLLECTOR
PARAMETERS 0
INPUTS 5
8,1 8,3 16,7 116,1 9,5
0    0     0    0    0

UNIT 11 TYPE 11 CONTROLLED FLOW DIVERTER
PARAMETERS 1
2
INPUTS 3
116,1 116,2 99,1
10     0      0
```

```
UNIT 1 TYPE 1 FLAT-PLATE COLLECTOR-MODIFIED RADIATION TERM
*flow rate at test conditions is 3087 kg/hr w/ A=12.25 m2]
*ASHRAE efficiency in Cananda is: 0.873 74.23 0.155
*FSEC efficiency is .838 65.88 .2246
PARAMETERS 11
1 11.7  cpl 1 257.25   .838 65.88 .2246 2  0.0316 0.0104
INPUTS 9
11,3  11,4  9,5 16,7 16,4 16,6  0,0   16,10  16,11
10    FLOW  0     0      0       0      0.2     0        0

UNIT 10 TYPE 11 CONTROLLED FLOW MIXER
PARAMTERS 1
3
INPUTS 5
11,1 11,2 1,1 1,2 99,1
0    0    0     0      0

UNIT 114 TYPE 118 HX  MODEL
PARAMETERS 6
600 60 cpl rhol 10. 4.0
INPUTS 3
3,1 3,2 9,5
10 FLOW 5

UNIT 116 TYPE 118 HX MODEL
PARAMETERS 6
600 60 cpl rhol 10. 4.0
INPUTS 3
114,1 114,2 9,5
10 FLOW 5

UNIT 15 TYPE 14 Forcing Fcn for Setback and cooling
PARAMETERS  12
0 1 6 1 6 0 17 0 17 1 24 1

UNIT 49 TYPE 14 FFN FOR TSTAT (HEATING)
PARAMETERS 20
0 1.13 2160 1.13 2160 1. 3624 1. 3624 .93 5832 .93 5832 1. 7296 1. 7296 1.13 8760 1.13 !5
DEG F FLUCTUATION

UNIT 48 TYPE 14 FFN FOR TSTAT (COOLING)
PARAMETERS 20
0 1.07 2160 1.07 2160 1. 3624 1. 3624 .88 5832 .88 5832 1. 7296 1. 7296 1.07 8760 1.07 !5
DEG F FLUCTUATION
```

```
EQUATIONS 3
THEAT=21.11*[49,1]
TCOOL=23.89*[48,1]
TAUX=20*[49,1]

UNIT 8 TYPE 8  THREE STAGE ROOM THERMOSTAT
PARAMETERS 8
5 1 -3.89  23.89 21.11 20 8.33 0.5
INPUTS 6
*2ND INPUT IS 1ST STAGE SOURCE TEMP minimum source use is 25degF
12,4 3,1 0,0 TCOOL THEAT TAUX
22.2 10. 0. 23.89 21.11 20.

EQUATIONS 1
coolsig= [8,3] !*(1-[15,1])

UNIT 12 TYPE 12 SIMPLE BUILDING LOAD (DEGREE DAY MODEL) !!!still need e
for eCmin calc assume 80%
PARAMETERS 6
4 0 CAP 22.22 cpl 0
INPUTS 6
104,3 104,5 9,5  29,1 104,8 0,0
0    0.    0.    0   0    0.

UNIT 51 TYPE 14 Forcing Fcn for Humidity increase of occupants
PARAMETERS  12
0 0 7 0 7 1 15 1 15 0 24 0

UNIT 52 TYPE 14
PARAMETERS 12
0 1 3000 1 3000 0  6500 0 6500 1 8760 1

EQUATIONS 2
W_OCUP=[51,1]*.015
WTOT2= ([9,6]+20*.005+W_OCUP)/21

EQUATIONS 1
P_SIG=[8,1]+coolsig

EQUATIONS 1
OA_SIG=P_SIG+[8,2]


UNIT 104 TYPE 504 WATER SOURCE HEAT PUMP W/ HUMIDITY RATIO
PARAMETERS 22
```

```
*1  2   3   4   5  6  7  8  9  10  11  12  13    14   15    16   17    18  19 20
14  15  16  17  5  6  8  6  4  6 2 2  rhol cpl 1.007 4.19 1343. 36. 36000 0
*21  22
566. 153.4  !assuming 50% efficiency werv-eric's thesis,325 CFM
INPUTS 22
*1     2     3    4    5    6    7    8    9    10   11   12   13   14
116,1 116,2  12,4  WTOT2 9,5  9,6  0,0  0,0  coolsig  8,1  8,2  0,0  0,0  OA_SIG
*15  16   17   18   19   20   21   22
0,0  0,0  0,0  0,0  0,0  0,0  0,0  0,0
*1  2  3  4     5  6  7   8  9  10  11  12  13  14  15  16  17  18  19  20  21
 8. FLOW 20 .2 5  0  40  0  0  0.  0.  0.  0.  1.  60  55  1   1   1   1
*21   22
1500  1500

UNIT 3 TYPE 3 PUMP
PARAMETERS 4
FLOW cpl 895.2 0
INPUTS 3
104,1 104,2 P_SIG
10 FLOW 1
************************OUTPUTS***************************

UNIT 24 TYPE 24 INTEGRATOR
*DAILY
PARAMETERS 1
24.
INPUTS 11
P_SIG 116,1 104,1 104,3 12,4 104,14 104,17 104,13 3,3 8,2 9,5
0.0 0.0  0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0

UNIT 32 TYPE 24 INTEGRATOR
*HRLY
PARAMETERS 1
1.
INPUTS 2
104,14 8,2
0.0 0.0

EQUATIONS 12
TAMB=[9,5]*1.8+32
THP=[104,3]*1.8+32
Tmix=[10,1]*1.8+32
Tdiv=[11,1]*1.8+32
TBLAD=[116,1]*1.8+32
TCOL=[1,1]*1.8+32
```

```
TWHP=[104,1]*1.8+32
TROOM = [12,4]*1.8+32
TS=[116,3]*1.8+32
TR=[12,4]*1.8+32
QHPHRLY=[32,1]/3600
TAMB_AVE=[24,11]/24.

UNIT 25 TYPE 25 PRINTER 1
PARAMETERS 5
*WINTER STARTS AT 7272, EXP TEMP START AT 2868 (noon), END AT 10992
1 0. 17000. 11 2
*1 8592 8760 11 2
*1. 0. 8760. 11 2
INPUTS 19
TBLAD TWHP TCOL Tmix Tdiv TAMB TS TR  104,8 29,1   wtot2 coolsig 8,3 QHPHRLY
15,1 8,2 8,1 32,2 TAMB_AVE
Tbladout THPout TCOLout TtoHP Tfrombladd TAMB TSURF TROOM QHP QLD  wtot2
csigmod csig qhp ffn auxsig hsig auxtot TAMB_AVE


EQUATIONS 1
QWEEKLY=[30,1]/168/3600.

UNIT 27 TYPE 25 PRINTER 3
*WEEKLY PRINTER
PARAMETERS 5
24  12 17000 12 2
INPUTS 1
TBLAD
TBLAD

EQUATIONS 10
TAMBAV=[24,1]/24
TBLADAV=[24,2]/24
THPAV=[24,3]/24
TAIRAV =[24,4]/24
TRAV=[24,5]/24
QAIR=[104,8]
Qcomp=[24,8]/3600
QHPTOTKWH=[24,6]/3600
qauxtot=[24,7]/3600
pumptot=[24,9]/3600

UNIT 30 TYPE 24 INTEGRATOR
*DAILY
```

```
PARAMETERS 1
24.
INPUTS 6
QHPTOTKWH QAUXTOT QAIR 8,1 8,2 8,3
0. 0. 0. 0. 0. 0.

EQUATIONS 2
QCOMPSTRIP=Qcomp+QAUXTOT
QHP&P=pumptot+QHPTOTKWH

UNIT 26 TYPE 25 PRINTER 2
*PRINTS EVERY 24 HRS
PARAMETERS 5
24. 0 16980. 13 2
INPUTS 8 *13
*30,1 30,2 30,3 30,4 30,5 30,6 30,7 TAMB TROOM TBLAD 116,10 116,11 116,12
*QHP QSTRIP QAIR QBLD CH CHAUX CC TAMB TROOM TBLAD TWP TWIN TS
QHPTOTKWH QHP&P TBLAD qauxtot pumptot 24,1 32,2 TAMB_AVE
QHPDAY QHP&P TBLAD qauxday pumptot  psig auxsig TAMB_AVe


UNIT 65 TYPE 65 ONLINE PLOTTER
PARAMETERS 10
2  !number of left axis variables
0   !number of right axis variables
20  !left axis minimum
120  !left axis maximum
20  !right axis minimum
120  !right axis maximum
1  !number of plots per simulations
9 !number of x axis grid points
1   !on/off
-1  !logical unit for automatic output file
INPUTS 2
Tblad Troom
T_HX Troom
LABELS 5
[F]    [F]
TEMPS
TEMPS
Online Plot 1


END
```

8.2 Appendix B: TRNSYS Model of Geothermal Heat Pump System with PVC Heat Exchanger and Solar Collectors

```
VERSION 15.0

ASSIGN C:\trnsys15\Heat_Pump_Project\MODEL915.LST     6
ASSIGN C:\trnsys15\Weather\RALEIGH.DAT    10
ASSIGN C:\trnsys15\Catalog_Data\bldloads\eric\bldloadsi_novent.dat   18
ASSIGN C:\trnsys15\Heat_Pump_Project\FINAL\MODEL915_HX_hrly_contr_col.out  11
ASSIGN C:\trnsys15\Heat_Pump_Project\FINAL\MODEL915_HX_dly_contr_col.out    13
ASSIGN C:\trnsys15\Heat_Pump_Project\FINAL\MODEL9WEEKLY15.OUT    12
ASSIGN C:\trnsys15\Catalog_Data\GHPs\WSHP\Samp_Cc.dat    14
ASSIGN C:\trnsys15\Catalog_Data\GHPs\WSHP\Samp_Hb.dat    15
ASSIGN C:\trnsys15\Catalog_Data\GHPs\WSHP\Samp_CTtrane.dat   16
ASSIGN C:\trnsys15\Catalog_Data\GHPs\WSHP\Samp_HTa.dat   17
* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *


* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *

CONSTANTS 7
timestep =1/20
CAP = 4000
FLOW =2567  ! = 10GPM   2567 3336.5
rhol =1028.6  !1028.6 ethylene glycol, 1130 brine 10 deg F freeze pt.
cpl = 3.75 !3.75 ! 25% antifreeze, 75% h20, 3,48 brine 10 degree F freeze pt
timestart=2880 !3960.
timestop=14000.

SIMULATION timestart timestop  timestep
WIDTH 80

LIMITS 100 50
TOLERANCES -.01 .01

UNIT 9 TYPE 89 TMY DATA READER
PARAMETERS 2
-1  10

UNIT 29 TYPE 9 BUILDING LOAD READER
PARAMETERS 9
-1  !mode
0  !header lines to skip
1 !number of columns to read
1   !data file timestep
```

```
-1   ! >0: interpolate this column
1   !multiplication factor
0   !addition factor
18  !logical unit for data read
-1  ! >0: formatted read statement


*******************MODEL COMPONENTS*************************

EQUATIONS 1
TIMESTART1=TIMESTART+1

UNIT 14 TYPE 14 FORCING FUNCTION FOR RADIATION **shade from 3/1 to 10/15
**check actual forecast before making switch
PARAMETERS  18
0 1 2160 1 2160 0 7000 0 7000 1 10920 1 10920 0 15760 0 15760 1


EQUATIONS 2
GSR= [9,4]*[14,2]
GSRNEXT=[9,104]*[14,2]

UNIT 16 TYPE 16 SOLAR RADIATION PROCESSOR USING Reindl model
*OPTIMUM ANGLE IS 45.7,*SLOPE OF ROOF=14, AZIMUTH=-30
PARAMETERS 8
2   1   3   timestart1 35.87 4870.8  -3.783  1
INPUTS 9
*GSR  9,5  9,10  9,99  9,100  0,0  0,0   0,0   GSRNEXT
9,4  9,5  9,10  9,99  9,100  0,0  0,0   0,0   9,104
 0       0   0       0       0        0.2  14  -30  0


UNIT 99 TYPE 98 CONTROLLER FOR COLLECTOR
PARAMETERS 0
INPUTS 5
8,1 8,3 16,7 104,1 9,5
0    0     0    0    0

UNIT 11 TYPE 11 CONTROLLED FLOW DIVERTER
PARAMETERS 1
2
INPUTS 3
104,1 104,2 99,1
10     0      0
```

```
UNIT 1 TYPE 1 FLAT-PLATE COLLECTOR-MODIFIED RADIATION TERM
*flow rate at test conditions is 3087 kg/hr w/ A=12.25 m2]
*ASHRAE efficiency in Cananda is: 0.873 74.23 .155
*FSEC efficiency is .838 65.88 .2246
PARAMETERS 11
1 11.7 cpl 1 257.25 .838 65.88 .2246  2  0.0316 0.0104
INPUTS  9
11,3  11,4  9,5 16,7 16,4 16,6  0,0   16,10  16,11
10    FLOW  0     0      0       0      0.2     0        0


UNIT 10 TYPE 11 CONTROLLED FLOW MIXER
PARAMTERS 1
3
INPUTS 5
11,1 11,2 1,1 1,2 99,1
0     0     0      0       0

UNIT 114 TYPE 118 HX MODEL
PARAMETERS 6
600 60 cpl rhol 10. 4.0
INPUTS 3
10,1 10,2 9,5
10 FLOW 5

UNIT 116 TYPE 118 HX MODEL
PARAMETERS 6
600 60 cpl rhol 10. 4.0
INPUTS 3
114,1 114,2 9,5
10 FLOW 5

UNIT 15 TYPE 14 Forcing Fcn for Setback and cooling
PARAMETERS  12
0 1 6 1 6 0 17 0 17 1 24 1

UNIT 49 TYPE 14 FFN FOR TSTAT (HEATING)
PARAMETERS 20
0 1.13 2160 1.13 2160 1. 3624 1. 3624 .93 5832 .93 5832 1. 7296 1. 7296 1.13 8760 1.13 !5
DEG F FLUCTUATION

UNIT 48 TYPE 14 FFN FOR TSTAT (COOLING)
PARAMETERS 20
0 1.07 2160 1.07 2160 1. 3624 1. 3624 .88 5832 .88 5832 1. 7296 1. 7296 1.07 8760 1.07 !5
DEG F FLUCTUATION
```

```
EQUATIONS 3
THEAT=21.11*[49,1]
TCOOL=23.89*[48,1]
TAUX=20*[49,1]

UNIT 8 TYPE 8  THREE STAGE ROOM THERMOSTAT
PARAMETERS 8
5 1 -3.89  23.89 21.11 20 8.33 0.5
INPUTS 6
*2ND INPUT IS 1ST STAGE SOURCE TEMP minimum source use is 25degF
12,4 3,1 0,0 TCOOL THEAT TAUX
22.2 10. 0. 0 0 0

EQUATIONS 1
coolsig= [8,3] !*(1-[15,1])

UNIT 12 TYPE 12 SIMPLE BUILDING LOAD (DEGREE DAY MODEL)
PARAMETERS 6
4 0 CAP 22.22 cpl 0
INPUTS 6
104,3 104,5 9,5  29,1 104,8 0,0
0     0.    0.    0   0    0.

UNIT 51 TYPE 14 Forcing Fcn for Humidity increase of occupants
PARAMETERS  12
0 0 7 0 7 1 15 1 15 0 24 0

UNIT 52 TYPE 14
PARAMETERS 12
0 1 3000 1 3000 0  6500 0 6500 1 8760 1

EQUATIONS 2
W_OCUP=[51,1]*.015
WTOT2= ([9,6]+20*.005+W_OCUP)/21

EQUATIONS 1
P_SIG=[8,1]+coolsig

EQUATIONS 1
OA_SIG=P_SIG+[8,2]

UNIT 104 TYPE 504 WATER SOURCE HEAT PUMP W/ HUMIDITY RATIO
PARAMETERS 22
*1  2   3   4   5  6  7  8  9  10  11  12  13    14   15    16   17    18  19 20
```

```
14  15  16  17  5  6  8  6  4  6 2 2  rhol cpl 1.007 4.19 1343. 36. 36000 0
*21  22
566. 153.4  !assuming 50% efficiency werv-eric's thesis,325 CFM
INPUTS 22
*1     2     3    4    5    6    7    8    9    10   11   12   13   14
3,1 3,2  12,4  WTOT2 9,5  9,6  0,0  0,0  coolsig  8,1  8,2  0,0  0,0  OA_SIG
*15  16   17   18   19   20   21   22
0,0  0,0  0,0  0,0  0,0  0,0  0,0  0,0
*1  2  3  4      5  6  7   8  9  10  11  12  13  14  15  16  17  18  19  20  21
 8. FLOW 20 .2 5  0  40  0  0  0.  0.  0.  0.  1.  60  55  1   1   1   1
*21   22
1500  1500

UNIT 3 TYPE 3 PUMP
PARAMETERS 4
FLOW cpl 895.2 0
INPUTS 3
116,1 116,2 P_SIG
10 FLOW 1

************************OUTPUTS***************************

UNIT 24 TYPE 24 INTEGRATOR
*DAILY
PARAMETERS 1
24.   !************ changed to 24 hrs from 1 hr
INPUTS 9
P_SIG 116,1 104,1 104,3 12,4 104,14 104,17 104,13 3,3
0.0 0.0  0.0 0.0 0.0 0.0 0.0 0,0

UNIT 32 TYPE 24 INTEGRATOR
*HRLY
PARAMETERS 1
1.
INPUTS 4
104,14 3,3 99,1 p_sig
0.0 0.0  0 0 0

EQUATIONS 13
TAMB=[9,5]*1.8+32
THP=[104,3]*1.8+32
Tmix=[10,1]*1.8+32
Tdiv=[11,1]*1.8+32
TBLAD=[116,1]*1.8+32
TCOL=[1,1]*1.8+32
```

```
TWHP=[104,1]*1.8+32
TROOM = [12,4]*1.8+32
TS=[116,3]*1.8+32
TR=[12,4]*1.8+32
QHPHRLY=[32,1]/3600
PHRLY = [32,2]/3600
QVENT=[104,9]

UNIT 25 TYPE 25 PRINTER 1
PARAMETERS 5
*WINTER STARTS AT 7272, EXP TEMP START AT 2868 (noon), END AT 10992
1. 0. 17000. 11 2

INPUTS 26
TBLAD TWHP TCOL Tmix Tdiv 1,4 1,3 8,1 8,2 coolsig  99,1 TROOM 104,8 29,1  wtot2
104,6 QHPHRLY PHRLY QVENT 104,18 104,16 104,15 104,12 49,1 32,3 32,4
Tbladout THPout TCOLout Ttoblad Tfromhp eff col HSIG ASIG CSIG scsig TROOM QHP
QLD  wtot2  whp qhp PHRLY QVENT hreturn hairo hairin hamb FFN   scsig psig

EQUATIONS 1
QWEEKLY=[30,1]/168/3600.

UNIT 27 TYPE 25 PRINTER 3
*WEEKLY PRINTER
PARAMETERS 5
168  9516. 9550. 12 2
INPUTS 1
QWEEKLY
QWKLY

EQUATIONS 10
TAMBAV=[24,1]/24
TBLADAV=[24,2]/24
THPAV=[24,3]/24
TAIRAV =[24,4]/24
TRAV=[24,5]/24
QAIR=[104,8]
Qcomp=[24,8]/3600
QHPTOTKWH=[24,6]/3600
qauxtot=[24,7]/3600
pumptot=[24,9]/3600

UNIT 30 TYPE 24 INTEGRATOR
*DAILY
PARAMETERS 1
```

```
24.
INPUTS 6
QHPTOTKWH QAUXTOT QAIR 8,1 8,2 8,3
0. 0. 0. 0. 0. 0.

EQUATIONS 2
QCOMPSTRIP=Qcomp+QAUXTOT
QHP&P=pumptot+QHPTOTKWH

UNIT 26 TYPE 25 PRINTER 2
*PRINTS EVERY 24 HRS 3984 is 1am on june 17, 16980 is dec 8 EXP ENERGY 4016
16952
*annual data for 1416 10176
PARAMETERS 5
24. 3984 16980. 13 2
INPUTS 6 *13
*30,1 30,2 30,3 30,4 30,5 30,6 30,7 TAMB TROOM TBLAD 116,10 116,11 116,12
*QHP QSTRIP QAIR QBLD CH CHAUX CC TAMB TROOM TBLAD TWP TWIN TS
QHPTOTKWH QHP&P TBLAD qauxtot pumptot 24,1
QHPDAY QHP&P TBLAD qauxday pumptot  psig


UNIT 65 TYPE 65 ONLINE PLOTTER
PARAMETERS 10
3   !number of left axis variables
0   !number of right axis variables
20  !left axis minimum
120  !left axis maximum
20  !right axis minimum
120  !right axis maximum
1  !number of plots per simulations
9 !number of x axis grid points
1   !on/off
-1  !logical unit for automatic output file
INPUTS 3
Tblad  Tmix Troom
Tblad Tmix Troom
LABELS 5
[F]    [F]
TEMPS
TEMPS
Online Plot 1

END
```

8.3 Appendix C: Supplemental TRNSYS Components

```
SUBROUTINE TYPE118(TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)

C*****************************************************************
C THIS SUBROUTINE MODELS A HX WITH 1-D GROUND TEMPERATURES.
C THE MODEL ACCOUNTS FOR FREEZING OF THE GROUND BENEATHE
C THE HEAT EXCHANGER.THIS COMPONENT MODELS A BLADDER SYSTEM.
C LAST MODIFIED ON 1/25/03 BY ARF
C*****************************************************************

c       INCLUDE '../kernal/param.inc'
        DOUBLE PRECISION XIN,OUT
        DIMENSION XIN(3),OUT(9),PAR(13),INFO(15)
        DIMENSION TEMP(101),TPRIME(100)
        CHARACTER*3 YCHECK(4),OCHECK(9)!number of inputs and outputs
        REAL DELX,AREAA,SA,AREAW,VOL,Z
        REAL CVW,THOUR,FLOW,TSTART, PL
        REAL TWIN,TAMB,TW,TWP
        REAL DELT,HW,HA,RHO,CV,COND
        REAL SLOPE,NUM,NUMMINUS
        REAL FROZE,COLD,HF1,HF2,HFDIFF,HFPREV,HFWATER,NEGHFWATER,
        CVGW
        INTEGER I,J,ISTORE,COUNT
        INTEGER*4 ICNTRL

         INCLUDE '../include/param.inc'
        !common block
        COMMON /SIM/ TIME0,TFINAL,DELT,IWARN
        COMMON/STORE/ NSTORE,IAV,S(NUMSTR)

    IF (INFO(7).GE.0) GO TO 100

C   FIRST CALL OF SIMULATION
    INFO(6)=9  !NUMBER OF OUTPUTS

    INFO(9)=1  ! INDICATES WHETEHR TYPE DEPENDS ON PASSAGE OF TIME
        INFO(10)=120 !number of storage spots
    CALL TYPECK(1,INFO,3,6,0)

    DATA YCHECK/'TE1','MF1','TE1','TD1'/
    DATA OCHECK/'TE1','MF1','TE1'/
    CALL RCHECK(INFO,YCHECK,OCHECK)

C   SET PARAMETER VARIABLES
```

```
100     ISTORE=INFO(10)-1

     SA=PAR(1) !ft2
        VOL=PAR(2)/7.48 !ft3
     CVW=PAR(3)/4.18812             !CONVERT FROM KJ/KG-C TO BTU/FT-F
        PL=PAR(4)*0.062422          !CONVERVERT FROM KG/M3 TO LBM/FT3
        HW=PAR(5) !BTU/(HR-FT2-F)
        HA=PAR(6) !BTU/(HR-FT2-F)
c       RHO=PAR(7) !LB/FT3
c       CV=PAR(8) !BTU/(LB-F)
c       COND=PAR(9) !BTU/(HR-FT-F)


C       ESTABLISH NUMBER OF NODES
        DELX=.5 !ft (depth of nodes) needs to be a constant because of storage
            !array is being used
        NUM=25/DELX
        NUMMINUS=NUM-1


C       READ IN INPUTS
     TWIN=XIN(1)*1.8+32 !CONVERT FROM C TO F
     FLOW=XIN(2)*2.20462   !CONVERT FROM KG/HR TO LBM/HR
        TAMB=XIN(3)*1.8+32 !CONVERT FROM C TO F

70      CONTINUE

        IF (INFO(7).GT.-1.) GO TO 10

C       SET INITIAL VALUES FOR TW AND TEMP(I) AND TPRIME(I)
        SLOPE=abs(61-TAMB)/NUM
        DO 60 I =1,NUM
        TEMP(I) =SLOPE*I+TAMB
60      TPRIME(I)=0.
        TW=(TAMB+61)/2  !61 deg F is constant temperature of soil 25 ft
        count=0.
        IF(INFO(7).EQ.-1) THEN
      DO 115 j=1,2*NUM,2
              count=count+1
115       S(ISTORE+j)=TEMP(count) !storage array (necessary for convergence)
          S(ISTORE+2*NUM+2)=TW
     ENDIF

C       AREA CALCULATIONS AND VOLUME CALCULATIONS
        AREAW= SA/2 -sa/5
```

```
      AREAA= SA/2 + SA/5


C     GROUND FREEZING ASSIGNMENTS FOR NODE 1,2,3
      HF1=0. !Talley of Freezing for node 1
      HF1A=0. !Talley of Freezing for node 2
      HF1B=0. !Talley of Freezing for node 3
      HF2=0.  !Talley of Thaw for node 1
      HF2A=0. !Talley of Thaw for node 2
      HF2B=0. !Talley of Thaw for node 3

      RHO=93.6 !LB/FT3 density of soil
      CV=0.60 !BTU/(LB-F) specific heat of soil with 25% moisture
      COND=1.15 !BTU/(HR-FT-F) thermal conductivity of soil w/ 25% moisture

      HFWATER=144 !BTU/LBM
      NEGHFWATER=-144
      HDIFFA=0.
      CVI=.5   !specific volume of ice
      RHOI=57.5 !density of ice
      CVGI=.47 !specific heat of ground when moisture is frozen
      RHOGI=84.6 !density of ground when moisture is frozen
      CONDICE=1.02
      HFPREV=0.
      CVGW=1.
      X=0.
      XA=0.

C     ON/OFF SIGNALS
      FROZE=0.
      FROZEA=0.
      FREEZING=0.
      FREEZINGA=0.
      FROZEB=0.
      FREEZINGB=0.
10   CONTINUE
      count=0.
      IF(INFO(7).EQ.0) THEN
            DO 116 j=1,2*NUM,2
116      S(ISTORE+j)=S(ISTORE+j+1)
             S(ISTORE+2*NUM+2)=S(ISTORE+2*NUM+3)
   ENDIF
      DO 117 j=1,2*NUM,2
            count=count+1.
117      TEMP(count)=S(ISTORE+j)
```

```
            TW=S(ISTORE+2*NUM+2)


C        Note: Ground is not preicted to freeze deeper than 1.5 ft into ground.
C        Therfore, only freezing of 2 nodes is accounted for.
C    GROUND TEMPERATURE EQNS
          TPRIME(1)=DELT/(RHO*DELX*CV)*(TEMP(1)*((RHO*DELX*CV)/DELT-
9*HW
    &   /(8+3*DELX*HW/COND)-
COND/DELX)+TEMP(2)*(HW/(8+3*DELX*HW/COND)
    &   +COND/DELX)+TW*(8*HW/(8+3*DELX*HW/COND)))

C************************START OF NODE 1 CHECK**********************
C        CHECK FOR GROUND FREEZING AND REDEFINE NODE 1 TEMPS
         IF((TPRIME(1).LE.32.).AND.(FREEZING.EQ.0).AND.(FROZE.NE.1)) THEN
                 HFDIFF=0.
                 THAW=0.
         ENDIF

         IF(TPRIME(1).LT.32.) THEN
                 HF1=(CVGW/DELT*(TPRIME(1)-32)*(1-X)+CVI/DELT*(TPRIME(1)-
32)*X)
    &   /.25
                 FREEZING=1.
                 HF2=0.
         ENDIF
         IF((TPRIME(1).LT.32.).AND.(HFDIFF.GT.0).AND.(THAW.EQ.0)) THEN
         HDIFF=0.
         ENDIF

         IF((TPRIME(1).GT.32.).AND.(FREEZING.EQ.1.)) THEN
                 HF2=(CVGW/DELT*(TPRIME(1)-32)*(1-X)+CVI/DELT*(TPRIME(1)-
32)*X)
    &   /.25
                 HF1=0.
         ENDIF

         IF((TPRIME(1).GT.32.).AND.(HFDIFF.GE.0)) THEN
                 FREEZING=0.
         ENDIF
         IF(THAW.EQ.1) THEN
         FREEZING=1.
         ENDIF
         IF(FROZE.EQ.1) THEN
         FREEZING=0.
```

```
        THAW=0.
        ENDIF

        IF((TPRIME(1).GT.32.).AND.(FROZE.EQ.1)) THEN
                HF2=(CVGW/DELT*(TPRIME(1)-32)*(1-X)+CVI/DELT*(TPRIME(1)-
32)*X)
    &   /.25
                HF1=0.
                HFDIFF=0.
                FREEZING=1.
                FROZE=0.
                THAW=1.
        ENDIF

        HFPREV=HFDIFF
        HFDIFF=HF1+HF2+HFPREV
        X=HFDIFF/HFWATER

C       X IS MASS FRAC OF ICE
        IF((X.GT.0).AND.(X.LE.1)) THEN
        X=1-X
        ENDIF
        IF((X.LT.0).AND.(X.GE.-1.)) THEN
        X=ABS(X)
        ENDIF

        IF(X.GT.1) THEN
        X=0.
        ENDIF
        IF(X.LT.-1) THEN
        X=1.
        ENDIF
        IF(FROZE.EQ.1) THEN
        X=1.
        ENDIF

        IF(HFDIFF.LE.NEGHFWATER) THEN
                FROZE=1.
                FREEZING=0.

TPRIME(1)=DELT/(RHOGI*DELX*CVGI)*(TEMP(1)*((RHOGI*DELX*CVGI)/DELT
    &-9*HW/(8+3*DELX*HW/CONDICE)-
CONDICE/DELX)+TEMP(2)*(HW/(8+3*DELX*HW
    &   /CONDICE) +CONDICE/DELX)+TW*(8*HW/(8+3*DELX*HW/CONDICE)))
        ENDIF
```

```
        IF((HFDIFF.LE.0).AND.(THAW.EQ.1)) THEN
                FROZE=1.
                FREEZING=0.

TPRIME(1)=DELT/(RHOGI*DELX*CVGI)*(TEMP(1)*((RHOGI*DELX*CVGI)/DELT
    &-9*HW/(8+3*DELX*HW/CONDICE)-
CONDICE/DELX)+TEMP(2)*(HW/(8+3*DELX*HW
    &   /CONDICE)+CONDICE/DELX)+TW*(8*HW/(8+3*DELX*HW/CONDICE)))
        ENDIF

        IF ((HFDIFF.GE.HFWATER).AND.(FREEZING.EQ.1)) THEN
                FREEZING=0.
                THAW=0.

        TPRIME(1)=DELT/(RHO*DELX*CV)*(TEMP(1)*((RHO*DELX*CV)/DELT-
9*HW
    &   /(8+3*DELX*HW/COND)-
COND/DELX)+TEMP(2)*(HW/(8+3*DELX*HW/COND)
    &   +COND/DELX)+TW*(8*HW/(8+3*DELX*HW/COND)))
        ENDIF

        IF (FREEZING.NE.0) THEN
                TPRIME(1)=32.
        ENDIF
C***********************END OF 1ST NODE
CHECK***************************

        TPRIME(2)=TEMP(2)*(1-2*DELT*COND/(RHO*DELX**2*CV))+DELT*COND
    &  /(RHO*DELX**2*CV)*(TEMP(1)+TEMP(3))

C***********************START OF 2ND NODE
CHECK**************************
        IF((TPRIME(2).LE.32.).AND.(FREEZINGA.EQ.0).AND.(FROZEA.NE.1)) THEN
                HFDIFFA=0.
                THAWA=0.
        ENDIF

        IF(TPRIME(2).LT.32.) THEN
          HF1A=(CVGW/DELT*(TPRIME(2)-32)*(1-XA)+CVI/DELT*(TPRIME(2)-32)
    &   *XA)/.25
                FREEZINGA=1.
                HF2A=0.
        ENDIF
        IF((TPRIME(2).LT.32.).AND.(HFDIFFA.GT.0).AND.(THAWA.EQ.0)) THEN
        HDIFFA=0.
```

```
      ENDIF

      IF((TPRIME(2).GT.32.).AND.(FREEZINGA.EQ.1.)) THEN
        HF2A=(CVGW/DELT*(TPRIME(2)-32)*(1-XA)+CVI/DELT*(TPRIME(2)-32)
     &  *XA)/.25
            HF1A=0.
      ENDIF

      IF((TPRIME(2).GT.32.).AND.(HFDIFFA.GE.0)) THEN
            FREEZINGA=0.
      ENDIF
      IF(THAWA.EQ.1) THEN
      FREEZINGA=1.
      ENDIF
      IF(FROZEA.EQ.1) THEN
      FREEZINGA=0.
      THAWA=0.
      ENDIF

      IF((TPRIME(2).GT.32.).AND.(FROZEA.EQ.1)) THEN
        HF2A=(CVGW/DELT*(TPRIME(2)-32)*(1-XA)+CVI/DELT*(TPRIME(2)-32)
     &  *XA)/.25
            HF1A=0.
            HFDIFFA=0.
            FREEZINGA=1.
            FROZEA=0.
            THAWA=1.
      ENDIF

      HFPREVA=HFDIFFA
      HFDIFFA=HF1A+HF2A+HFPREVA
      XA=HFDIFFA/HFWATER

C     XA IS MASS FRAC OF ICE
      IF((XA.GT.0).AND.(XA.LE.1)) THEN
      XA=1-XA
      ENDIF
      IF((XA.LT.0).AND.(XA.GE.-1.)) THEN
      XA=ABS(XA)
      ENDIF

      IF(XA.GT.1) THEN
      XA=0.
      ENDIF
      IF(XA.LT.-1) THEN
```

```
      XA=1.
      ENDIF
      IF(FROZEA.EQ.1) THEN
      XA=1.
      ENDIF

      IF(HFDIFFA.LE.NEGHFWATER) THEN
            FROZEA=1.
            FREEZINGA=0.
       TPRIME(2)=TEMP(2)*(1-2*DELT*CONDICE/(RHOGI*DELX**2*CVGI))
   &  +DELT*CONDICE/(RHOGI*DELX**2*CVGI)*(TEMP(1)+TEMP(3))
      ENDIF
      IF((HFDIFFA.LE.0).AND.(THAWA.EQ.1)) THEN
            FROZEA=1.
            FREEZINGA=0.
       TPRIME(2)=TEMP(2)*(1-2*DELT*CONDICE/(RHOGI*DELX**2*CVGI))
   &  +DELT*CONDICE/(RHOGI*DELX**2*CVGI)*(TEMP(1)+TEMP(3))
      ENDIF

      IF ((HFDIFFA.GE.HFWATER).AND.(FREEZINGA.EQ.1)) THEN
            FREEZINGA=0.
            THAWA=0.
            TPRIME(2)=TEMP(2)*(1-
2*DELT*COND/(RHO*DELX**2*CV))+DELT*COND
   &  /(RHO*DELX**2*CV)*(TEMP(1)+TEMP(3))
      ENDIF

      IF (FREEZINGA.NE.0) THEN
            TPRIME(2)=32.
      ENDIF

C**********************END OF 2ND NODE CHECK************************


      TPRIME(3)=TEMP(3)*(1-2*DELT*COND/(RHO*DELX**2*CV))+DELT*COND
   &  /(RHO*DELX**2*CV)*(TEMP(2)+TEMP(4))

C***********************START OF 3RD NODE
CHECK**************************
      IF((TPRIME(3).LE.32.).AND.(FREEZINGB.EQ.0).AND.(FROZEB.NE.1)) THEN
            HFDIFFB=0.
            THAWB=0.
      ENDIF

      IF(TPRIME(3).LT.32.) THEN
```

```
         HF1B=(CVGW/DELT*(TPRIME(3)-32)*(1-XB)+CVI/DELT*(TPRIME(3)-32)
     &  *XB)/.25
              FREEZINGB=1.
              HF2B=0.
        ENDIF
        IF((TPRIME(3).LT.32.).AND.(HFDIFFB.GT.0).AND.(THAWB.EQ.0)) THEN
        HDIFFB=0.
        ENDIF

        IF((TPRIME(3).GT.32.).AND.(FREEZINGB.EQ.1.)) THEN
         HF2B=(CVGW/DELT*(TPRIME(3)-32)*(1-XB)+CVI/DELT*(TPRIME(3)-32)
     &  *XB)/.25
              HF1B=0.
        ENDIF

        IF((TPRIME(3).GT.32.).AND.(HFDIFFB.GE.0)) THEN
              FREEZINGB=0.
        ENDIF
        IF(THAWB.EQ.1) THEN
        FREEZINGB=1.
        ENDIF
        IF(FROZEB.EQ.1) THEN
        FREEZINGB=0.
        THAWB=0.
        ENDIF

        IF((TPRIME(3).GT.32.).AND.(FROZEB.EQ.1)) THEN
         HF2B=(CVGW/DELT*(TPRIME(3)-32)*(1-XB)+CVI/DELT*(TPRIME(3)-32)
     &  *XB)/.25
              HF1B=0.
              HFDIFFB=0.
              FREEZINGB=1.
              FROZEB=0.
              THAWB=1.
        ENDIF

        HFPREVB=HFDIFFB
        HFDIFFB=HF1B+HF2B+HFPREVB
        XB=HFDIFFB/HFWATER

C       XB IS MASS FRAC OF ICE
        IF((XB.GT.0).AND.(XB.LE.1)) THEN
        XB=1-XB
        ENDIF
        IF((XB.LT.0).AND.(XB.GE.-1.)) THEN
```

```
        XB=ABS(XB)
        ENDIF

        IF(XB.GT.1) THEN
        XB=0.
        ENDIF
        IF(XB.LT.-1) THEN
        XB=1.
        ENDIF
        IF(FROZEB.EQ.1) THEN
        XB=1.
        ENDIF

        IF(HFDIFFB.LE.NEGHFWATER) THEN
                FROZEB=1.
                FREEZINGB=0.
         TPRIME(3)=TEMP(3)*(1-2*DELT*CONDICE/(RHOGI*DELX**2*CVGI))
     &  +DELT*CONDICE/(RHOGI*DELX**2*CVGI)*(TEMP(2)+TEMP(4))
        ENDIF
        IF((HFDIFFB.LE.0).AND.(THAWB.EQ.1)) THEN
                FROZEB=1.
                FREEZINGB=0.
         TPRIME(3)=TEMP(3)*(1-2*DELT*CONDICE/(RHOGI*DELX**2*CVGI))
     &  +DELT*CONDICE/(RHOGI*DELX**2*CVGI)*(TEMP(2)+TEMP(4))
        ENDIF

        IF ((HFDIFFB.GE.HFWATER).AND.(FREEZINGB.EQ.1)) THEN
                FREEZINGB=0.
                THAWB=0.
                TPRIME(3)=TEMP(3)*(1-
2*DELT*COND/(RHO*DELX**2*CV))+DELT*COND
     &  /(RHO*DELX**2*CV)*(TEMP(2)+TEMP(4))
        ENDIF

        IF (FREEZINGB.NE.0) THEN
                TPRIME(3)=32.
        ENDIF

C**********************END OF 3rd NODE CHECK***********************

C       Finds temps of ground after 3rd NODE
      DO 30 I=4,NUM
30       TPRIME(I)=TEMP(I)*(1-2*DELT*COND/(RHO*DELX**2*CV))+DELT*COND
     &  /(RHO*DELX**2*CV)*(TEMP(I-1)+TEMP(I+1))
```

```
          TS=(9*TEMP(1)-TEMP(2)+3*DELX*HW/COND*TW)/(8+3*DELX*HW/COND)

         TWP=TW*(1-
DELT/(PL*VOL*CVW)*(AREAW*HW+AREAA*HA+FLOW*CVW)) !AREAB FROM
AREAA
    &
+DELT/(PL*VOL*CVW)*(AREAW*HW*TS+FLOW*CVW*TWIN+AREAA*HA*TAM
B)

41      DO 40 J=1,NUMMINUS
40          TEMP(J)=TPRIME(J)
            TW =TWP

         HFDIFF=HFPREV
         HFDIFFA=HFPREVA
         HFDIFFB=HFPREVB

C    SET THE FINAL VALUE FOR THE NEXT TIMESTEP
         count=0.
                  DO 120 j=1,2*NUM,2
                  count=count+1
120               S(ISTORE+j+1)=TEMP(count)
                  S(ISTORE+2*NUM+3)=TW

C    SET OUTPUTS
50    OUT(1)=(TW-32.)/1.8
     OUT(2)=FLOW/2.20462
         OUT(3)=(TS-32.)/1.8

     RETURN 1
     END
```

```
      SUBROUTINE TYPE98(TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)

C************************************************************************
C THIS TYPE IS USED TO SIMULATE A SOLAR COLLECTOR CONTROLLER. IT
C USES A SIGNAL FROM THE CONTROLLER TO BE IN HEATING OR COOLING
C MODE. DEPEDING ON THE MODE OF THE HEAT PUMP, THE COMPONENT
C CHECKS TO SEE IF RADIATION IS IMPACTING THE SURFACE OF THE
C COLLECTOR AND THE AMBIENT TEMPERATURE.
C************************************************************************


      DOUBLE PRECISION XIN,OUT
      DIMENSION XIN(6),OUT(3),PAR(0),INFO(15)
      CHARACTER*3 YCHECK(5),OCHECK(1)!number of inputs and outputs


      INCLUDE '../include/param.inc'
      !common block
      COMMON /SIM/ TIME0,TFINAL,DELT,IWARN


   CALL TYPECK(1,INFO,5,0,0)

   DATA YCHECK/'CF1','CF1','IR1','TE1','TE1'/
   DATA OCHECK/'CF1'/
   CALL RCHECK(INFO,YCHECK,OCHECK)


C     READ IN INPUTS

      HSIG=XIN(1) !HEATING SIGNAL
   CSIG=XIN(2) !COOLING SIGNAL
      SRAD=XIN(3) !SOLAR RADIATION
      WTEMPI=xin(4) !entering water temp
      TAMB=xin(5)  !ambient temp


      COL_SIG=0 !DEFAULT SETTING

C     LOGIC

      IF((CSIG.EQ.1).AND.(SRAD.LT.0.1))THEN
      COL_SIG=1
      ENDIF
```

```
      IF((HSIG.EQ.1).AND.(SRAD.GT.0.1))THEN
      COL_SIG=1
      ENDIF

c     IF((HSIG.EQ.1).AND.(WTEMPI.LT.TAMB))THEN
c     COL_SIG=1
c     ENDIF



C     OUTPUTS
      OUT(1)=COL_SIG
      RETURN 1
  END
```