EML 4905 Senior Design Project

A B.S. THESIS

PREPARED IN PARTIAL FULFILLMENT OF THE

REQUIREMENT FOR THE DEGREE OF

BACHELOR OF SCIENCE

IN

MECHANICAL ENGINEERING

HYBRID GEOTHERMAL HEAT PUMP SYSTEM

100%

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This B.S. thesis is written in partial fulfillment of the requirements in EML 4905. The contents represent the opinion of the authors and not the Department of Mechanical and Materials Engineering.

Ethics Statement and Signatures

The work submitted in this B.S. thesis is solely prepared by a team consisting of Henry Gutierrez, Miguel Freire, Santiago Paz and it is original. Excerpts from others' work have been clearly identified, their work acknowledged within the text and listed in the list of references. All of the engineering drawings, computer programs, formulations, design work, prototype development and testing reported in this document are also original and prepared by the same team of students.

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Team Leader	Team Member	Team Member

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Abstract

A geothermal heat pump system uses thermal energy to provide cooling during the summer or heating during the winter. It works under the same principles as an air conditioner but it utilizes constant underground temperature to efficiently perform this (McQuay, 2002). Around the earth, about 5 to 6 feet (2 meters) under the ground and regardless of the outside temperature, there is a moderate temperature range from 74°F to 77°F. This range allows the geothermal pump to act as a heat sink during the summer and as a heat source during the winter. The Florida International University (FIU) Solar House shown in Figure 1 (Garciage, 2009) is connected to this type of pump. A cooling tower will be incorporated to balance the heat rejected during time. Furthermore, a cost analysis will be included in detail showing cost and prices before and after the cooling. Therefore, the cooling tower will be introduced to efficiently remove heat leaving the pump lines prior to entering the geothermal loops. This auxiliary heat rejecting system is the reason for which the geothermal heat pump is a hybrid one.

Figure 1 FIU Solar House

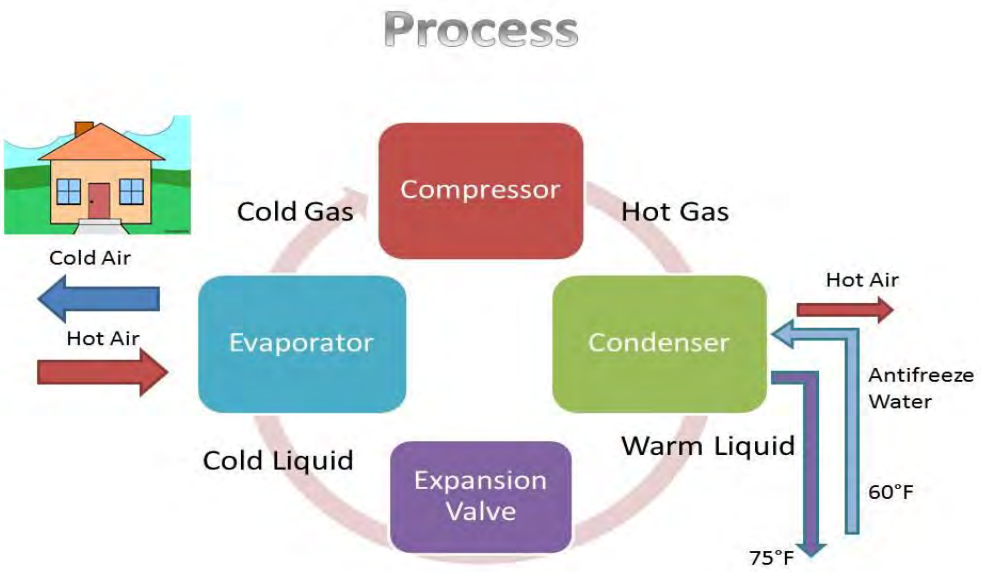


Figure 2 Geothermal Heat Pump Process

Figure 3 Florida Heat Pump Aquarius Geothermal Heat Pump

1. Introduction

1.1 Problem Statement

The solar house at FIU uses a geothermal heat pump for cooling and heating process. Geothermal energy is used to reject the heat from the house and back into the ground. Many slinky tube loops that contain refrigerant are laid six feet under the ground to accomplish this process. (See Reference A3, pg. 78). Inside the tubes, refrigerant R410-A circulates changing the temperature from one point to the other as it travels underground. Therefore it acts as a heat exchanger. One of the problems with the solar house heat pump is that in Miami, especially during the months of June, July, August and September, the average temperature above 90 degrees Fahrenheit is very common and doesn't allow the heat pump to work as effectively as it should and to dissipate the heat quickly enough. The number of days with a max temperature of 90°F or higher is shown in Table 4.

As the temperatures rises during the summer, the heat flow from the inside the house to the ground is huge. During the process of heat balance on summer days, the ground becomes hotter and the process cannot complete appropriately. The area where the pipes were originally buried, due to space restrictions, is not large enough to dissipate the heat required to cool the house.

There is a solution to this problem. This is to increase the area of pipes under the ground and incorporate a cooling tower. Adding more pipes will dissipate the extra amount of heat needed to cool the house. In addition, during the winter time the extra energy needed will be lost. A cooling tower is an auxiliary heat rejecting system to dissipate the extra heat during the summer time. With the cooling tower, a shut off valve

will be designed to activate the tower during the summer time when temperature is high facilitating the heat transfer in an efficient way. During the winter, the cooling tower will be off in order to save energy and not energy loss.

1.2 Motivation

Geothermal heat pump industries have been increasing for recent years. The market is growing and more people are changing from a traditional air condition units to a geothermal heat pump saving money for during time. In addition, the geothermal pump is the most effective environmentally friendly solution. With the new way of cooling or heating process, issues will arise in each situation that will require different approaches. The solution to this problem is a cooling tower. This new system has the most effective of heat removal and low operation cost. Furthermore, it reduce the area for an underground pipes required to remove the total heat needed during the cooling process saving work and money. The geothermal pump uses different type of energy from the ground like heat in case of volcanic activity, water flow in a river and the different temperature between the ground and the surface. The applications of the geothermal pump are extending for all country making a big market to exploit using the new technology. Figure 4 shows the hottest know geothermal regions in the world include the west coast of North and South America, between Africa and Europe and the Middle as well as pacific near Asia

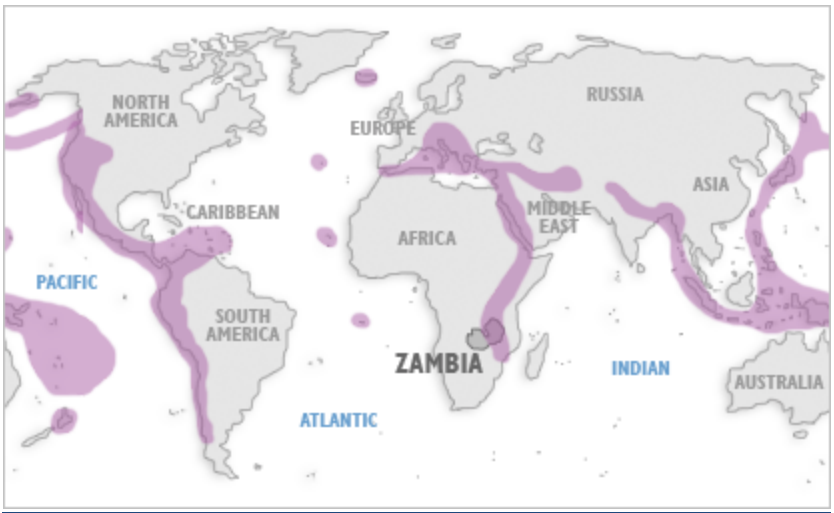


Figure 4 Hottest Known Geothermal Regions in the World

1.3 Literature Survey

One of the journals read was**:** *Optimal Sizing of Hybrid Ground-Source Heat Pump Systems That Use a Cooling Pond as a Supplemental Heat Rejecter— A System Simulation Approach*

Name. The author's names were:

Mahadevan Ramamoorthy *Student Member ASHRAE*

Hui Jin *Student Member ASHRAE*

Andrew D. Chiasson *Associate Member ASHRAE*

Jeffrey D. Spitler, Ph.D., P.E *Member ASHRAE*

2. Project Formulation

2.1 Project Objectives

The hybrid geothermal heat pump will consist of several objectives. The first one is to assess the working state of the geothermal heat pump. It needs to be working properly in order to measure the efficiency of the heat pump (Coefficient of Performance), its Environmentally Efficient Ratio and how effectively it accommodates the FIU Solar House's cooling load. Once all the data has been acquired, further calculations will be performed to better understand the design of the air cooling tower that will be combined with the geothermal system. Due to the small size of the Solar House (755 square feet) a compact cooling tower should be designed. In implementing the cooling tower, the geothermal heat pump should reject heat with even more efficiency during extremely hot seasons such as the summer. Therefore, the cooling load will increase. Finally, it is important to implement a design that will work effectively, comply with the Department of Energy Safety Codes and Regulations and remain under budget.

2.2 Conceptual Design

These two alternative designs will benefit geothermal heat pump to be more efficient by reducing heat, at the same time it will prevent any possible over heat situation and the pump will keep performing at its designed function. The first alternative design is to collect rain water into a container, these will help to cold down the tubes from the geothermal pump the only problem of this design is that in South Florida, five-month of rainy season from June through October, when 70 percent of the year's rain falls, and the seven-month dry season from November through May, probably in those months from

November to May there is no guarantee that it will be any rain. The second alternative design is to construct a dry cooling tower to increase the heat transfer balance rejecting heat from the geothermal pump to the ground. A dry cooling tower is the most inexpensive design due that in Miami, FL is not needed energy taken from the ground for heating process; the energy is always reject.

2.3 Proposed Design

A small cooling tower will be incorporated into the geothermal heat pump system via the pump lines. As the hot refrigerant from the geothermal loops leave the condenser, it will enter a cooling tower. This cooling tower will use evaporation to remove the excess heat into the atmosphere. After cooling, the refrigerant lines will leave the cooling tower and enter again through the geothermal tubes to continue its cooling cycle. The cooling tower should be small because the FIU Solar House is small (only 755 square feet). In addition, the design should be the proper size to suit the year round hot weather in South Florida as well. It will be a compact system to be placed either on the ground next the house or on the roof. Also, the cooling tower will comply with the Department of Energy (DOE) codes and regulations. The water cooling tower will also have an automatic shut off-valve to close the line during the cooler seasons such as the fall and winter. This will allow the geothermal heat pump to work alone without the water cooling tower.

3. Project Management

3.1 Time Line

Table 1 Time Line

	Spring								Fall									
Months	September		October		November		December		January		February		March		April		May	
Definition																		
Initation																		
Planing																		
Execution																		
Equipment Research																		
Choose Final Equipment																		
Device assembly																		
Device testing																		
Solid Work																		

3.2 Member Responsibilities

- Henry Gutierrez: CAD models of heat exchanger components. Air heat exchanger design calculations. Electric motor, fan, thermostat research and design development. Manufacturing of the heat exchanger housing.
- Miguel Freire: CAD models of heat exchanger components. Air heat exchanger design calculations. Air heat exchanger design calculations. Electric Motor, fan thermocouple research and design development. Manufacturing of the heat exchanger housing.
- Santiago Paz: Fan calculation. Cooling tower cooling load calculations including HVAC Load Explorer. Thermostat, thermocouple and shut-off valve research and design development. Manufacturing of the heat exchanger housing.

3.3 Hours Spent

Table 2 Hours Spent (Fall 2012 Semester, Aug – Dec)

HOURS SPENT (August-December)		
DATE	HOURS	ACTIVITIES
8/23/2012	3	Searching a topic
8/27/2012	3	Choosing a topic
8/28/2012	4	Research of geothermal heat pump
8/30/2012	2	Splitting responsibilities
9/4/2012	5	Research geothermal heat pump
9/13/2012	4	Working for team power point
9/18/2012	2	Meeting with Dr. Lin
9/25/2012	4	Reading and searching about geothermal heat pump
9/26/2012	2	Meeting with Dr. Lin and visiting the Solar House
10/11/2012	3	Working on the power point presentation for GL description
10/16/2012	1	Meeting Dr. Lin
10/16/2012	5	Project Synopsis
10/23/2012	2	Softcopy team poster
10/30/2012	4	Working on the Final report 10%
11/1/2012	4	Working on the Final report 25%
11/6/2012	1.5	Meeting with Dr. Lin
11/6/2012	6	Working on the Final report 25%
Senior 1 Total	55.5	

Table 3 Hours Spent (Spring 2013 Semester, Jan – Apr)

HOURS SPENT (January-April)		
DATE	HOURS	ACTIVITIES
1/2/2013	6	Searching topics
1/2/2013	1	Meeting with Dr. Lin
1/3/2013	3	Calculation
1/9/2013	6	Searching Material
1/9/2013	1	Meeting with Dr. Lin
1/10/2013	3	Contacting Vendors
1/16/2013	6	Searching topics & Materials
1/16/2013	1	Meeting with Dr. Lin
1/23/2013	6	Calculation
1/24/2013	3	Working on the Report
1/30/2013	6	Working on the Final report 50%
2/6/2013	5	Searching Material
2/6/2013	1	Meeting with Dr. Lin
2/13/2013	5	Buying Material
2/13/2013	1	Meeting with Dr. Lin
2/14/2013	4	Working on the Report
2/20/2013	6	Working on the Final report 75%
2/27/2013	5	Testing the Motor & Blade
2/28/2013	2	Working on the Report
3/6/2013	5	Working on Solid Works
3/6/2013	1	Meeting with Dr. Lin
3/7/2013	3	Buying Material
3/10/2013	5	Testing the prototype
3/13/2013	4	Working on the report & prototype
3/14/2013	3	Assembling Prototype
3/17/2013	5	Testing the prototype
3/20/2013	4	Working on the Report
3/20/2013	1	Meeting with Dr. Lin
3/27/2013	6	Working on the Final report 100%
3/28/2013	2	Working on the Final report 100%
Senior 2 Total	110	

4. Design Analysis

4.1 Overall Analytical Calculations

The analytical design involved using of the theory related to air conditioning design, heat transfer and thermodynamics laws. It was divided into three parts.

The first part is the analysis of the cooling load from the solar house at FIU described in detail. The total cooling load was calculated using HVAC load calculation program and several considerations were taken into account. First, the solar house has an area of 755 square feet (Gonzalez, 2004) which is usually the same dimensions of a one bedroom apartment. It is assumed that not more than three people will be living inside due to the space restrictions. Second, the total appliances for the FIU solar house are a refrigerator, microwave, three incandescent lights bulbs, and an electrical kitchen. There are some other appliances not taken into consideration due the small amount of heat transfer like two radios and one electrical watch. Third, temperature in Miami, for one year, was found in Figure 5. Every month has different temperature ranging anywhere from roughly 58 degrees Fahrenheit to past 97 degrees Fahrenheit. Furthermore the properties of the material and the dimensions are listed in Table 16.

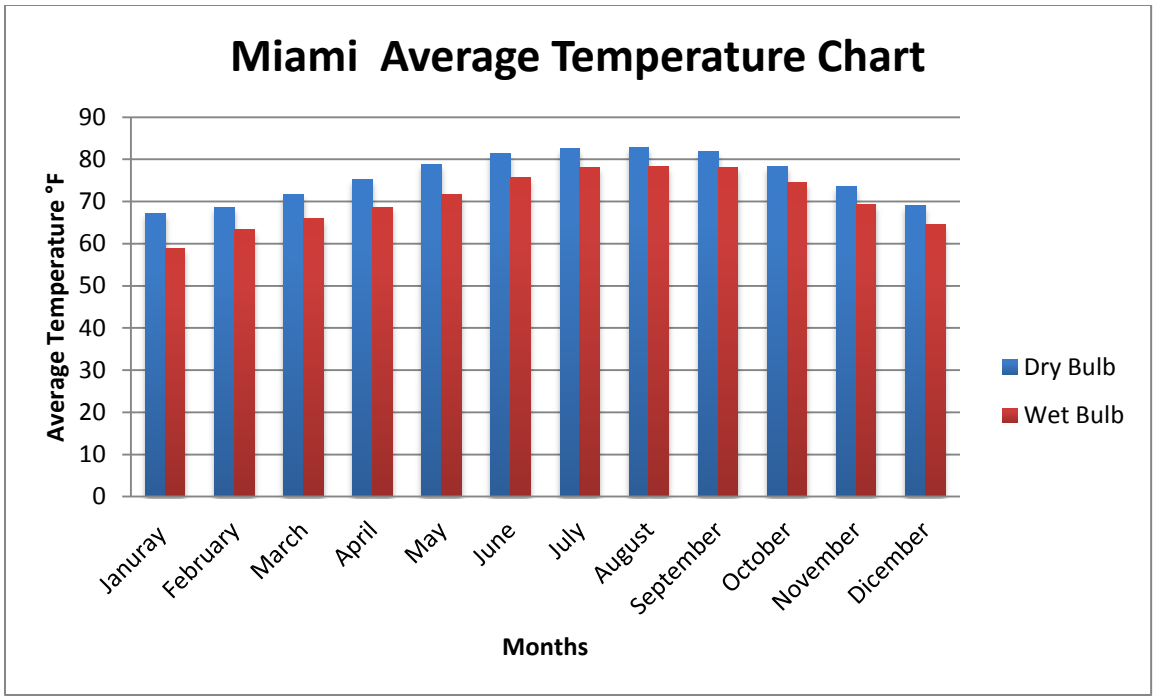


Figure 5 Miami Average Temperature Graph

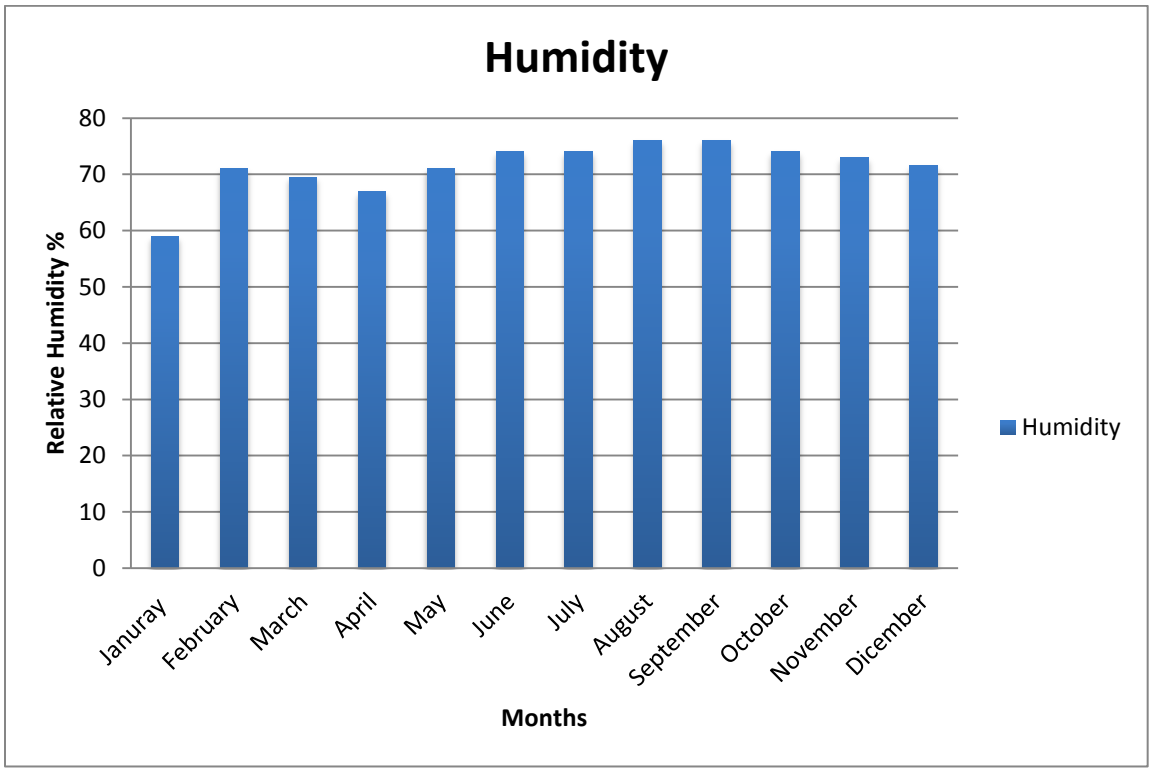


Figure 6 Miami Climate Temperature

The second part is to calculate the total length buried underground. Also, it is analyzed if the length was long enough to remove of the heat coming from the solar house at extremely hot days once the heat is rejected from the solar house. A thermal resister is analyzed to approach the heat transfer problem.

Table 4 Miami - Days with Highest Max Temp

Month	Jan	Feb	Mar	April	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Miami Days with max temperature of 90ºF or higher	0	0	0	1	4	10	16	16	10	2	0	0

The last part is to incorporate a dry cooling tower into the system. By adding the cooling tower a portion of the pipes will be removed and a heat transfer will be dissipated effectively. Different types of cooling tower will be analyzed to establish the best suit for the operations.

5. Cooling Load

5.1 Cooling Load Analysis

The design for the geothermal loop system will be used for cooling the FIU solar house. The definition for the cooling load is the rate at which energy must be removed from a space to preserve the temperature and humidity at the design temperature. Also heat gain should be considered in the space. Heat gain is the increase of heat in a given space as a result of direct heating by solar radiation and heat radiated by other sources like openings, lights, equipment and people (Cengel, 2003).

Due to heat gained in the space, heat should be removed in order to keep comfortable indoor conditions. The heat extraction rate is the rate which energy is removed from space by cooling and dehumidifying equipment (Casa, Alonzo and Cifuentes, 2010).

The calculation of the cooling load involves transient state heat transfer, as the immediate heat gain into an acclimatized space is quite flexible with time; mostly because of the strong effect created hourly by the solar radiation. The transient nature for the cooling load problem can be determined of the heat conduction thought the wall or roof f with a variable outdoor temperature and with a variable solar radiation input on the outside surface (Spitler, McQuiston, and Parker, 2005) in the case the wall or roof is a single homogeneous slab, the governing differential equation is:

$$\frac{\partial t}{\partial \theta} = \frac{k}{\rho c_p} * \frac{\partial^2 t}{\partial x^2}$$

t	Local temperature at a point in the slab, F or C
θ	Time, hr or s
$k/\rho c_p$	Thermal diffusivity of the slab, ft^2/hr or m^2/s
$\boldsymbol{x}$	Length, ft or m

The cooling load can be resolve by complex of boundary conditions; in that case the outdoor and indoor surface temperatures can be determined with conduction solution. Heat gain depends on the zone air and surface temperature.

To compute the cooling load, the Heat Balance Method will be used. These Method involves balancing the energy into the space of interest. An energy balance equation can it explained for any surface as well for the zone air. This equation can be

combined with the transient conduction equation thought external wall and roof.

In order to calculate the cooling load, the design conditions need to be determined and the internal heat gains through using Heat Balance Method. In the design conditions it is include the dry bulb and wet bulb temperatures and the wind speed.

The internal heat gain are due to lights, equipment and people, they represent a significant component for the cooling load in the space. In order to estimate the heat gain from people in the space, the amount occupants in the space and the period of occupancy need to be known. The heat gain depends of how much occupancy is in the space and also depends of the activity of the occupants. The sensible heat gain is to be 30 percent convective (instant cooling load) and 70 percent radiation (delayed) (Spitler, McQuiston, and Parker, 2005). The heat gain due to light depends on the installation, if there is air distribution system and the mass of the structure. The rate of gain at any known period can differentiate from the heat power equivalent to those lights parts of the lights emitted by the energy in a form of radiation and are absorbed by the building. The energy absorbed is transformed to the air by convection. In order to calculate the instantaneous rate of heat gain from electric lighting, the following equation will be used:

$$\dot{q} = 3.41 * W F_u F_s$$

$\dot{\mathbf{q}}$	Heat gain, $\mathbf{Btu/hr}$ (to obtain heat gain in W, eliminate 3.41)
W	Total installed light wattage, W
$\mathbf{F_u}$	Use factor, ratio of wattage in use to total installed wattage
$\mathbf{F_s}$	Special allowance factor (ballast factor in the case of fluorescent and metal halide fixtures)

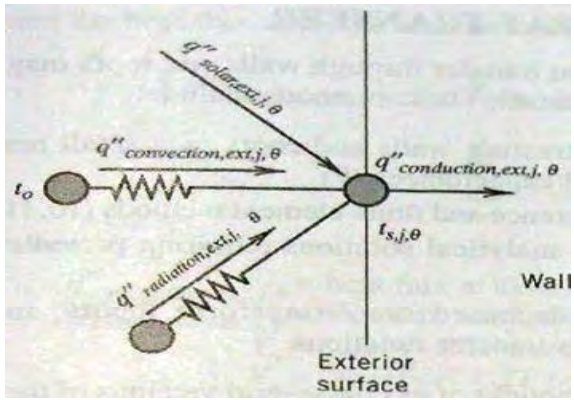


Figure 7 Exterior Surface Heat Balance [31]

The heat balance on the exterior surface j at time θ is given by

$$q''_{conduction,ext,j,\theta} = q''_{solar,ext,j,\theta} + q''_{convection,ext,j,\theta} + q''_{radiation,ext,j,\theta}$$

Table 5 Exterior Surface Heat Balance Components

Heat Balance		
Components	**Descriptions**	**Units**
$\mathbf{q''_{conduction,ext,j,\theta}}$	Conduction heat flux	$Btu/(hr-ft^2)$ or W/m^2
$\mathbf{q''_{solar,ext,j,\theta}}$	Absorbed solar heat flux	$Btu/(hr-ft^2)$ or W/m^2
$\mathbf{q''_{convection,ext,j,\theta}}$	Convection heat flux	$Btu/(hr-ft^2)$ or W/m^2
$\mathbf{q''_{radiation,ext,j,\theta}}$	Thermal radiation heat flux	$Btu/(hr-ft^2)$ or W/m^2

The absorbed solar heat gain is computed through use of the following relation:

$$q''_{solar,ext,j,\theta} = \alpha G_t$$

The absorbed solar heat gain is computed through use of the following relation:

$$q''_{solar,ext,j,\theta} = \alpha G_t$$

$\boldsymbol{\alpha}$	Solar absorptivity of the surface, dimensionless
$\boldsymbol{G_t}$	Total solar irradiation incident on the surface, $Btu/(hr-ft^2)$ or W/m^2

$$G_{ND} = \frac{A}{\exp(B/_{sin\beta})} * C_N$$

Table 6 Nominal Direct Irradiation

Normal Direct Irradiation		
Components	**Descriptions**	**Units**
A	Apparent solar irrational at air ass equal to zero	$Btu/(hr-ft^2)$
B	Atmospheric extinction coefficient	Dimensionless
β	Solar altitude	Dimensionless
$\mathbf{C_N}$	Clearness number	Dimensionless

$\mathbf{h_c}$	Convection coefficient

The exterior surfaces may be represented in the exterior heat balance equation is

$$q''_{conduction,ext,j,\theta} = h_c(t_o - t_{es,j,\theta})$$

A relationship established by Yazdanian and Klems is used in the calculation the convection heat transfer coefficient for low-rise structures

$$h_c = \sqrt{\left[C_t(\Delta t)^{\frac{1}{3}}\right]^2 + \left[aV_0{}^b\right]^2}$$

Table 7 Exterior Convection

External confection		
Components	**Descriptions**	**Units**
C_t	Turbulent natural convection constant	Dimensionless
Δt	Temperature different between the exterior surface and the outside air	°F
V_0	Wind speed at standard conditions	mph
a,b	Constants	Dimensionless

Long wavelength radiation was taken to and from exterior surface. It was assumed that each surface was opaque, diffuse and isothermal have uniform radiosity and irradiation. In addition the surface is assumed to be gray and having a single value of absorptive and emissivity. From radiation of the sky of the atmosphere also participate medium, modeled by heat transfer affecting the sky temperature. Also is expected that the building sits on a smooth, featureless plain. In this case a vertical wall has a view factor between the wall and the ground of 0.5, and between the wall and the sky of 0.5 (Spitler, McQuiston, and Parker, 2005).

$$q''_{radiation,ext,j,\theta} = \epsilon\sigma\left[F_{s-g}\left(t_g^4 - t_{es,j,\theta}^4\right) + F_{s-sky}\left(t_{sky}^4 - t_{es,j,\theta}^4\right)\right]$$

Table 8 External Convection

External Convection			
Components	**Descriptions**		**Units**
ϵ	Surface long wavelength emissivity		Dimensionless
σ	Stefan-Boltzmann constant	$= 0.1714x10^{-8}$	$Btu/(hr - ft^2 - R^4)$
		$= 5.67x10^{-8}$	$W/(m^2 - K^4)$
F_{s-g}	View factor from the surface to the ground		Dimensionless
F_{s-sky}	View factor from the surface to the sky		Dimensionless
t_g	Ground temperature		R or K
t_{sky}	Effective sky temperature		R or K
$t_{es,j,\theta}$	Surface temperature		R or K

To building sits on a featureless plain, In the following equations:

$$F_{s-g} = \frac{1 - \cos\alpha}{2}$$

$$F_{s-sky} = \frac{1 + \cos\alpha}{2}$$

$\boldsymbol{\alpha}$	The tilt angle of the surface from horizontal

To liberalize this equation, radiation heat transfer coefficient is used:

$$h_{r,g} = \epsilon\sigma\left[\frac{F_{s-g}\left(t_g^4 - t_{es,j,\theta}^4\right)}{t_{sky} - t_{es,j,\theta}}\right]$$

$$h_{r,sky} = \epsilon\sigma\left[\frac{F_{s-sky}\left(t_g^4 - t_{es,j,\theta}^4\right)}{t_{sky} - t_{es,j,\theta}}\right]$$

The net long wavelength radiation equation

$$q''_{radiation,ext,j,\theta} = h_{r,g}\left(t_g - t_{es,j,\theta}\right) + h_{r-sky}\left(t_{sky} - t_{es,j,\theta}\right)$$

The efficiency and accuracy of the z-transform methods have been design it for load calculation. The Z-transform methods use one or two formulations, response factor and conduction factor. The response factor is series of time in a coefficient that connected to heat flux. In this situation construction transfer function (CFT) coefficient are used. While determines the conduction transfer function coefficient involved using it straightforward. The CTF coefficients multiply the current values of interior and exterior

surface temperature and past values surface heat flux at the exterior surface (Spitler, McQuiston, and Parker, 2005).

$$q''_{conduction,ext,j,\theta} = -Y_0 t_{is,j,\theta} - \sum_{n=1}^{N_y} Y_n t_{is,j,\theta-n\delta} + X_0 t_{es,j,\theta} + \sum_{n=1}^{N_x} X_n t_{es,j,\theta-n\delta} + \sum_{n=1}^{N_q} \Phi_n q''_{conduction,ext,j,\theta-n\delta}$$

Table 9 External Convections

External Convection		
Components	**Descriptions**	**Units**
$\boldsymbol{q''_{radiation,ext,j,\theta}}$	Heat flux at exterior surface	$Btu/(hr-ft^2)$ or W/m^2
$\boldsymbol{q''_{conduction,in,j,\theta}}$	Heat flux at interior surface	$Btu/(hr-ft^2)$ or W/m^2
$\mathbf{Y_n}$	"cross" CTF coefficient	$Btu/(hr-ft^2-F)$ or $W/m^2\,K$
$\mathbf{X_n}$	"exterior" CTF coefficient	$Btu/(hr-ft^2-F)$ or $W/m^2\,K$
$\mathbf{Z_n}$	"interior" CTF coefficient	$Btu/(hr-ft^2-F)$ or $W/m^2\,K$
$\mathbf{t_{is,j,\theta}}$	Interior surface temperature	F or C
$\mathbf{t_{es,j,\theta}}$	Exterior surface temperature	F or C
$\boldsymbol{\Phi_n}$	Flux coefficient	Dimensionless

Exterior surface heat balance

$$H_{ext,j,\theta} = -\sum_{n=1}^{N_y} Y_n t_{is,j,\theta-n\delta} + \sum_{n=1}^{N_x} X_n t_{es,j,\theta-n\delta} + \sum_{n=1}^{N_q} \Phi_n q''_{conduction,ext,j,\theta-n\delta}$$

Also it can be represented as:

$$q''_{conduction,j,\theta} = -Y_0 t_{is,j,\theta} + X_0 t_{es,j,\theta} + H_{ext,j,\theta}$$

The interior surface heat balance on the j surface at time θ represented by:

$$q''_{conduction,in,j,\theta} = q''_{solar} + q''_{convection,in,j,\theta} + q''_{radiation,in,j,\theta}$$

Table 10 Interior Surface Heat Balance Components

Interior Surface Heat Balance		
Components	**Descriptions**	**Units**
$\mathbf{q''_{conduction,in,j,\theta}}$	Conduction heat flux	$Btu/(hr-ft^2)$
$\mathbf{q''_{solar}}$	Absorbed solar heat flux	$Btu/(hr-ft^2)$
$\mathbf{q''_{convection,in,j,\theta}}$	Convection heat flux	$Btu/(hr-ft^2)$
$\mathbf{q''_{radiation,in,j,\theta}}$	Thermal radiation heat flux	$Btu/(hr-ft^2)$

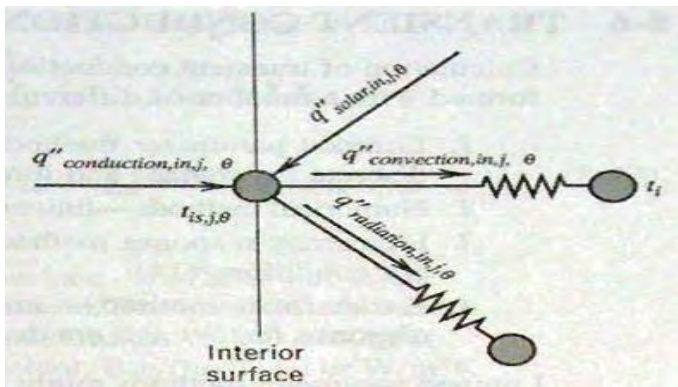


Figure 8 Interior Surface Heat Balance

Also the heat balance on the zone air negligible thermal storage may be represented as:

$$\sum_{j=1}^{N} A_j q''_{conduction,in,j,\theta} + \dot{q}_{infiltration,\theta} + \dot{q}_{system,\theta} + \dot{q}_{internal,conv,\theta} = 0$$

Table 11 Zone Air Heat Balance

Zone Air Heat Balance		
Components	**Descriptions**	**Units**
$\boldsymbol{A_j}$	Area of the j surface	ft^2
$\boldsymbol{\dot{q}_{infiltration,\theta}}$	Heat gain due to infiltration	Btu/hr
$\boldsymbol{\dot{q}_{system,\theta}}$	Heat gain due to the heating/cooling system	Btu/hr
$\mathbf{\dot{q}_{internal,conv,\theta}}$	Convective portion of internal heat gains due to people	Btu/hr

$$q''_{conduction,ext,j,\theta} = -Y_0 t_{is,j,\theta} - \sum_{n=1}^{N_y} Y_n t_{is,j,\theta-n\delta} + X_0 t_{es,j,\theta} + \sum_{n=1}^{N_x} X_n t_{es,j,\theta-n\delta} + \sum_{n=1}^{N_q} \Phi_n q''_{conduction,ext,j,\theta-n\delta}$$

5.2 Monthly Cooling Load Calculations

The cooling load was calculated for the twelve months using the HVAC Load Explorer program. In addition, a monthly study was performed to know in which months the cooling load was the highest. As expected, the maximum cooling loads occur from May through September where the rejection almost reaches 27,000 BTU/H; during the summer, the heat rejection through the ground is greater than that of the rest of the other months.

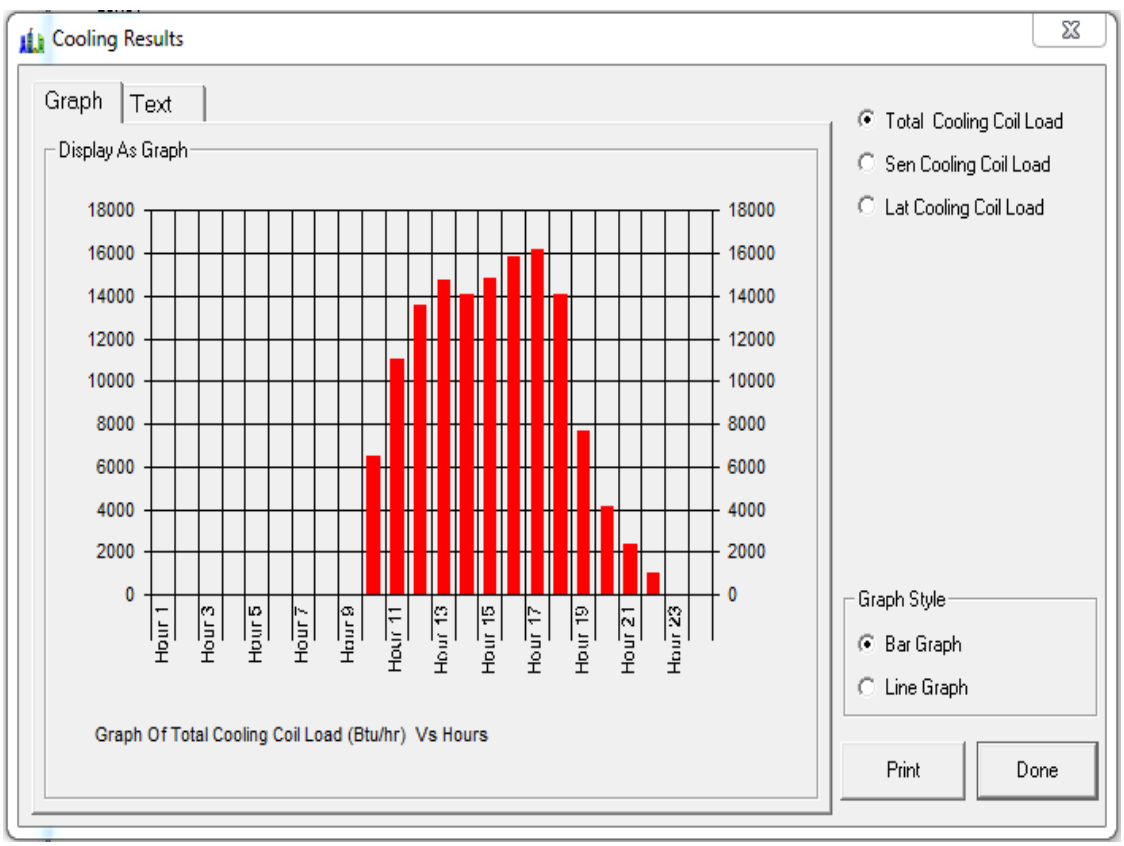
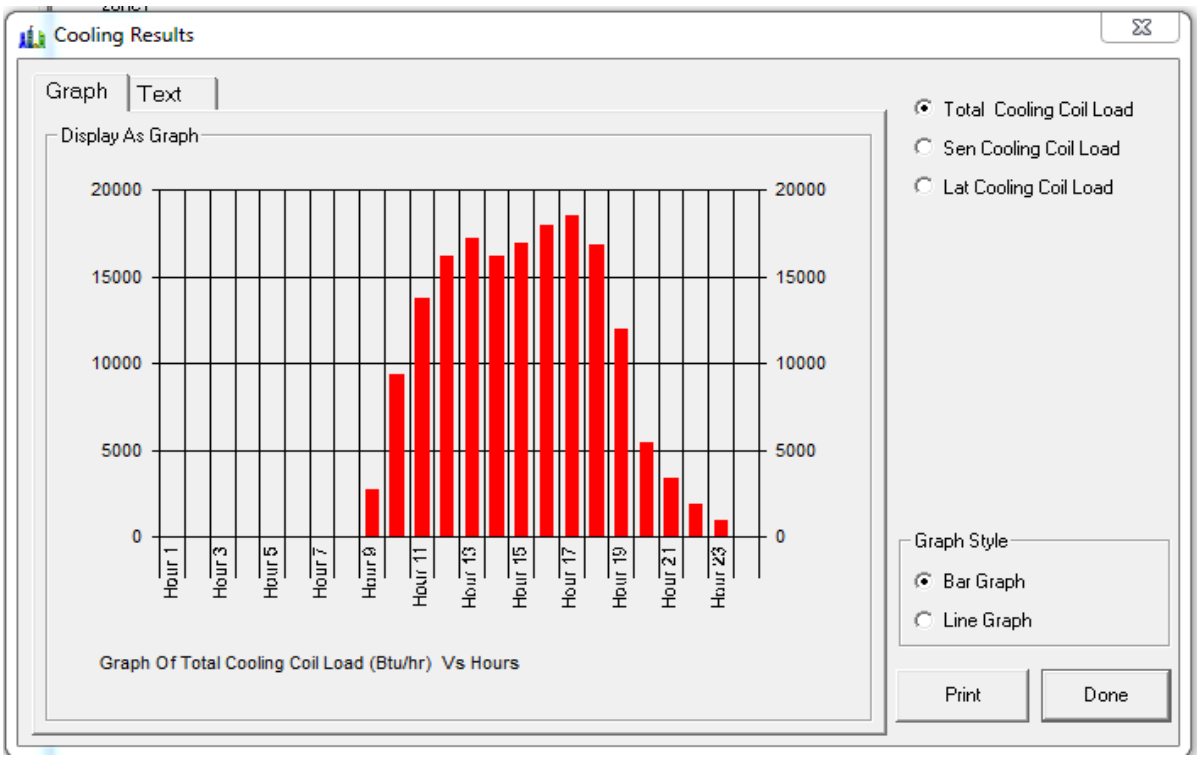


Figure 9 January



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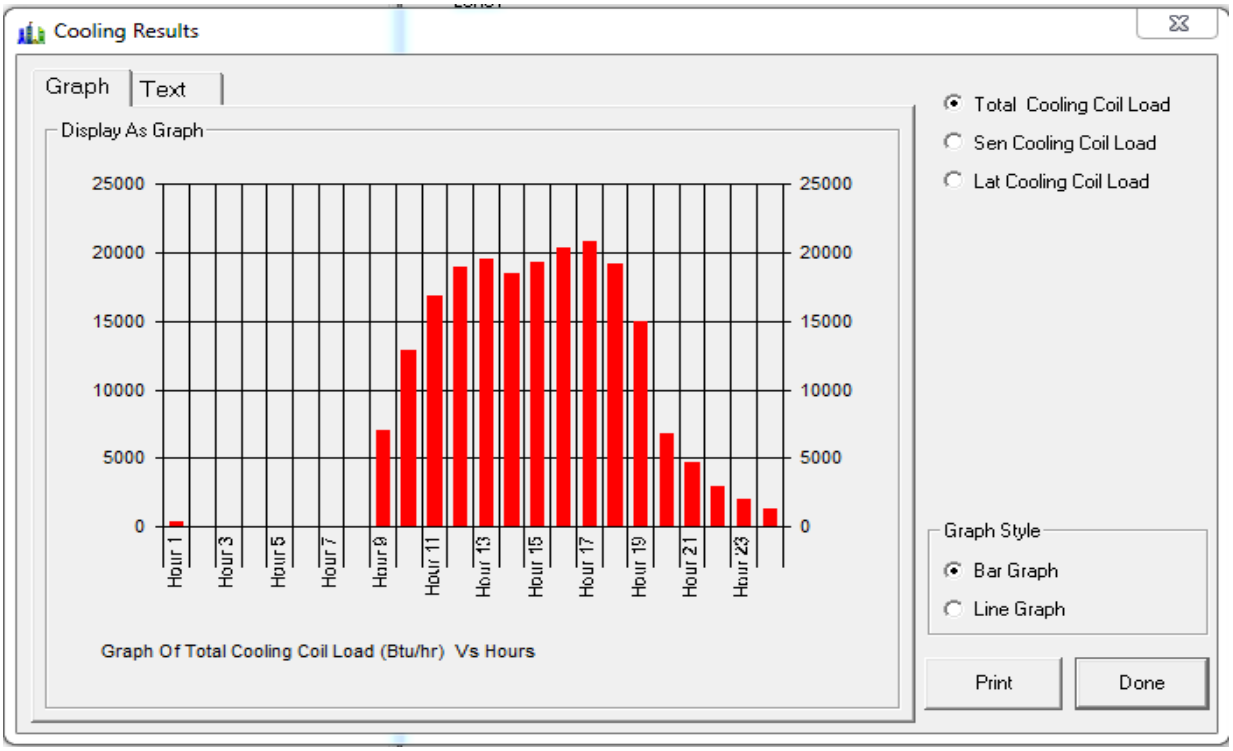

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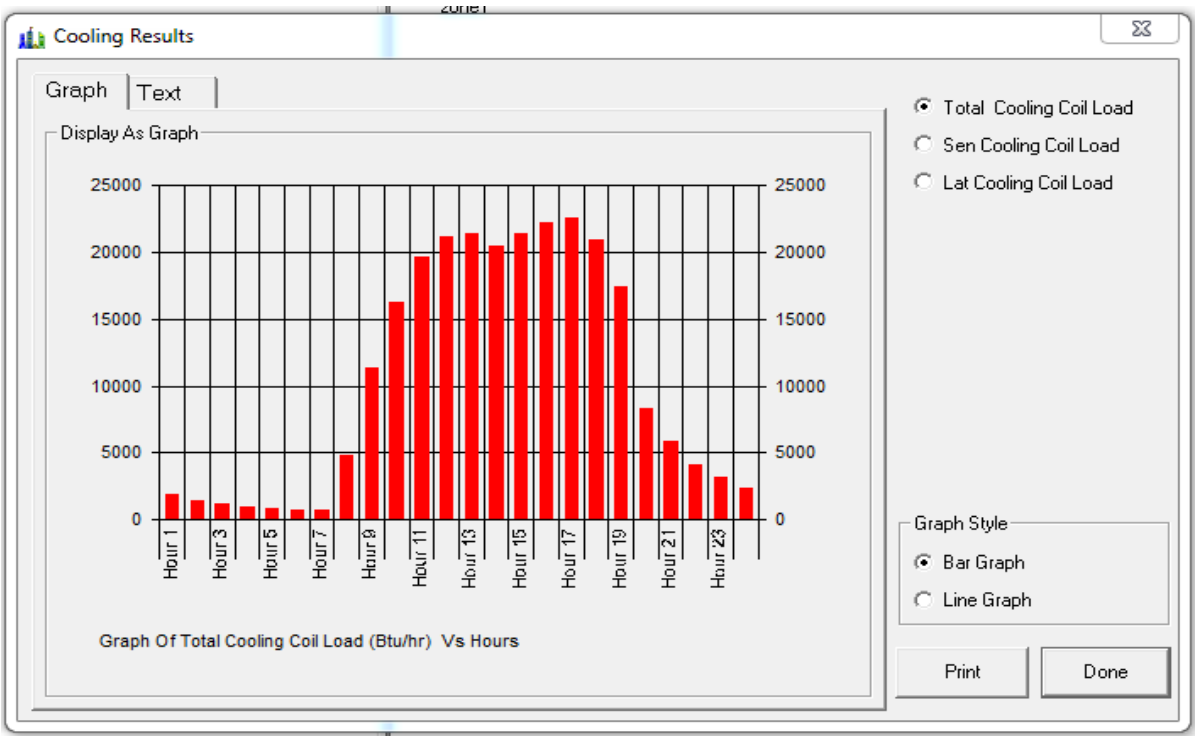

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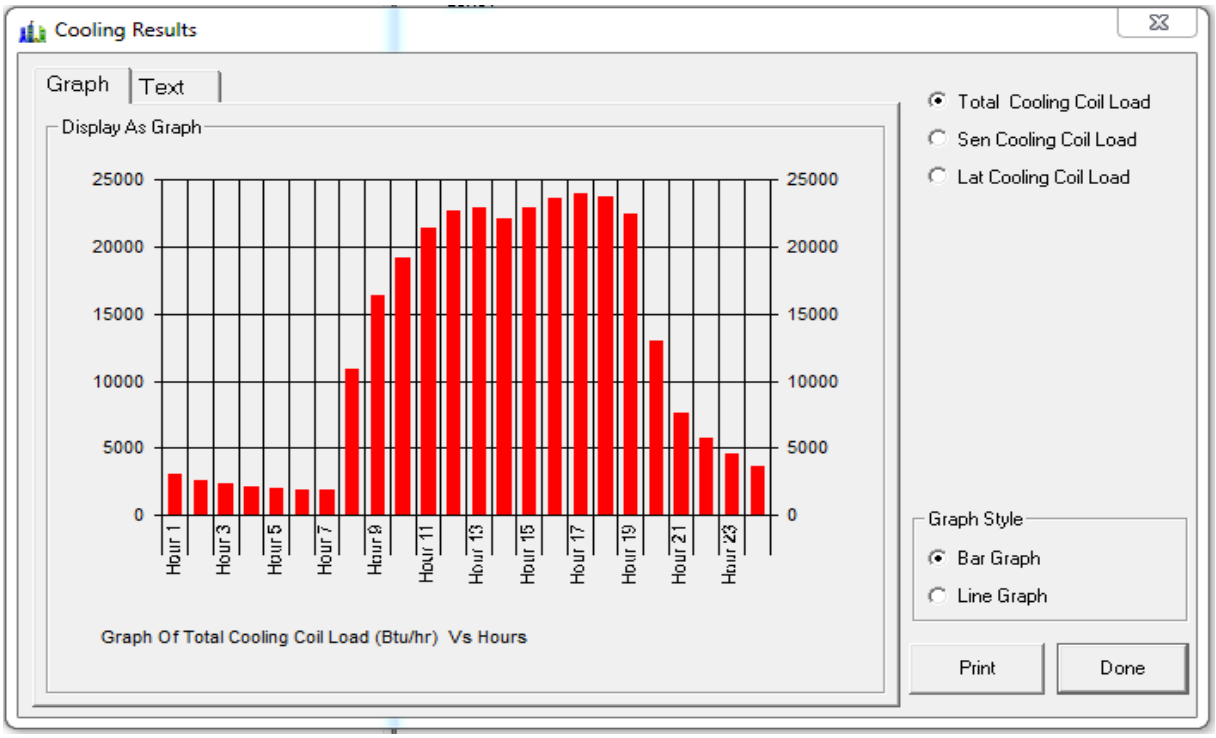

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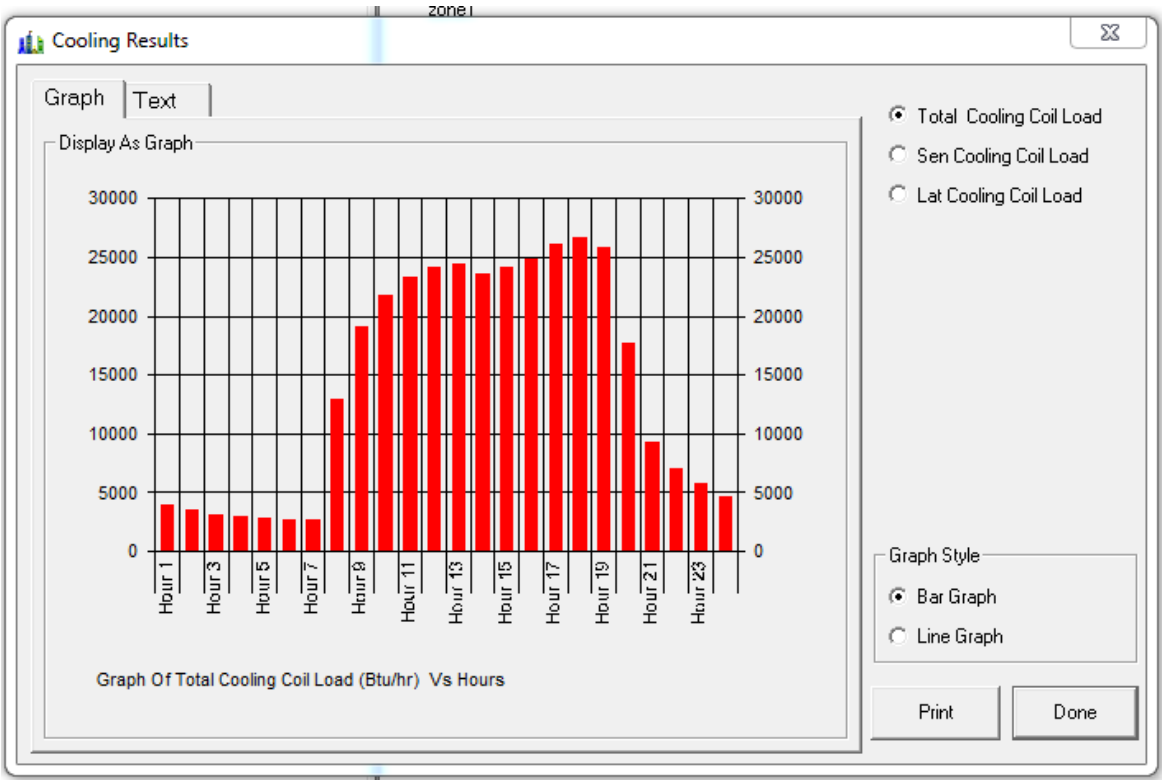

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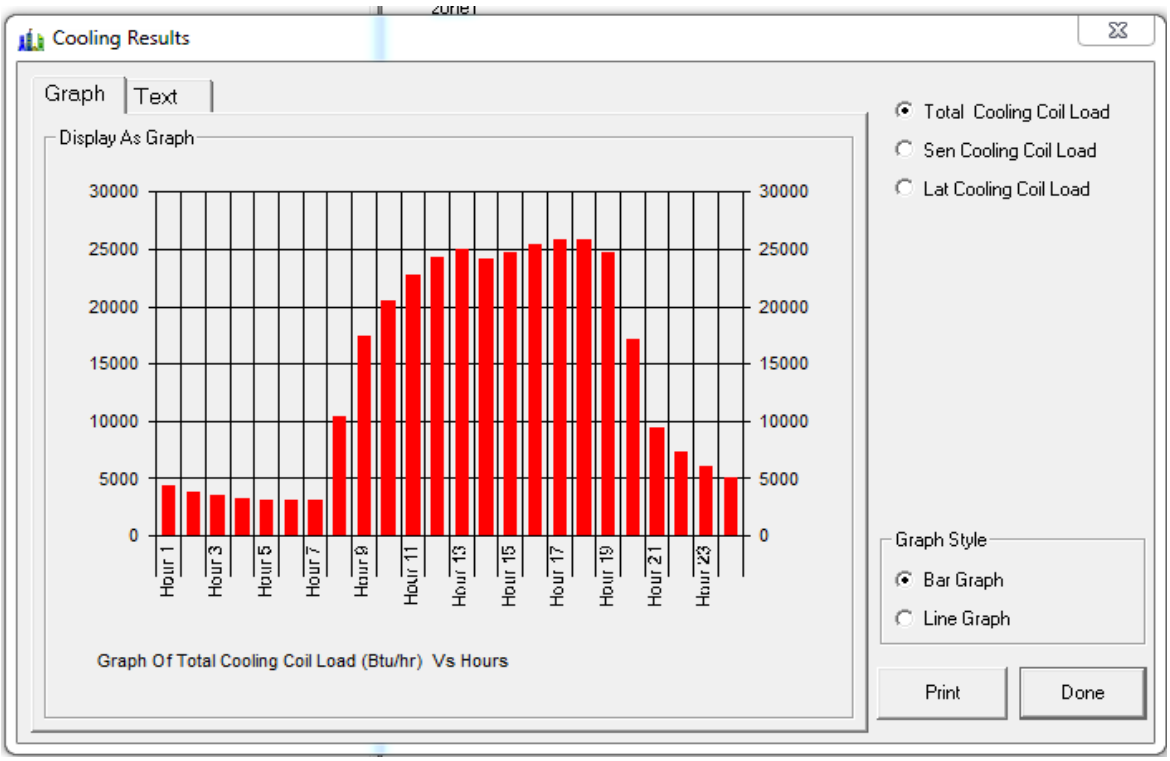

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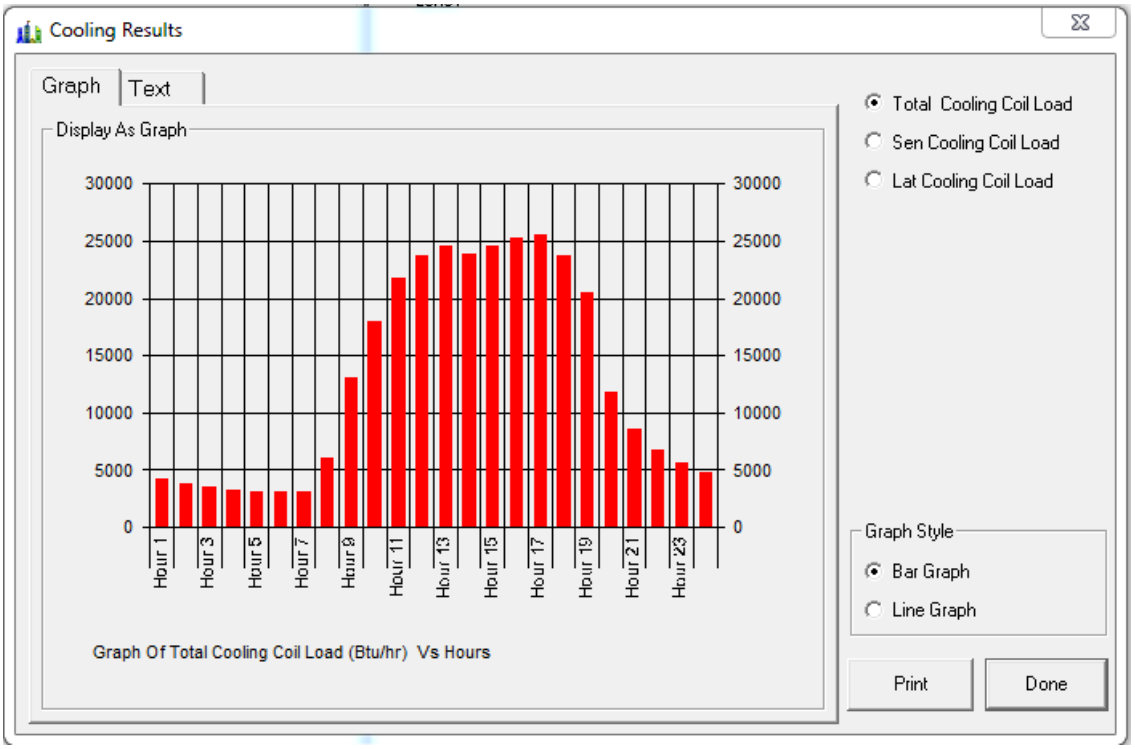

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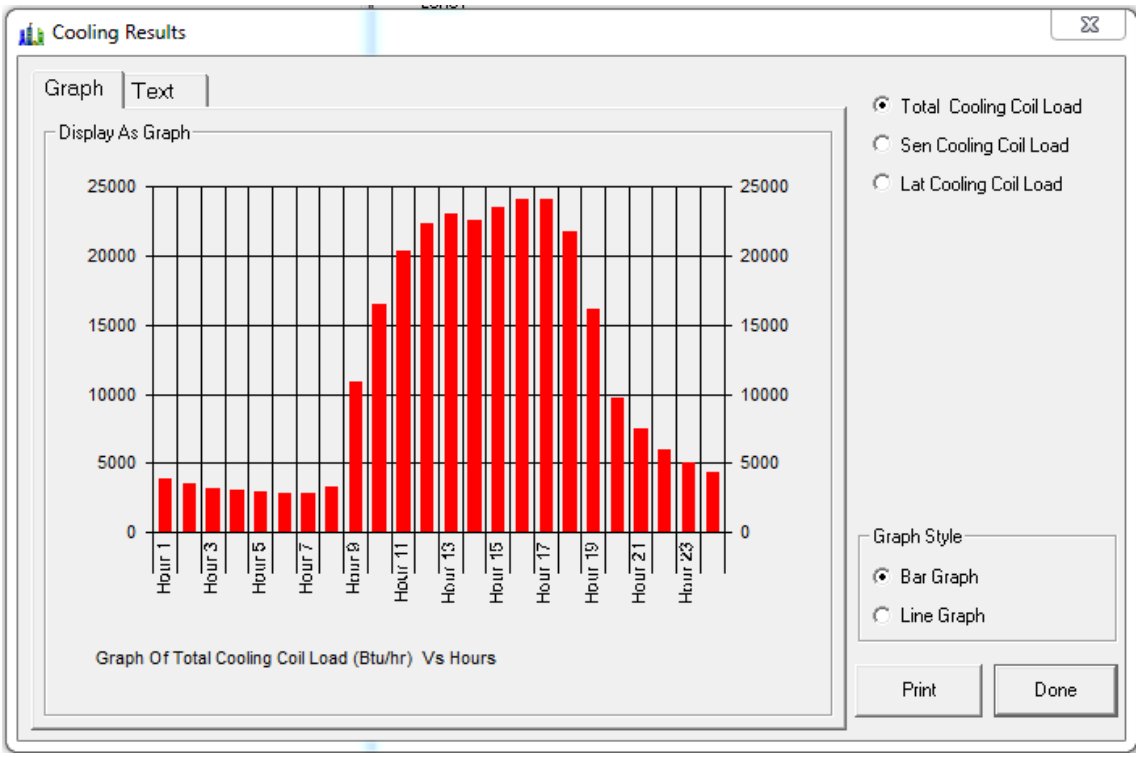


Figure 17 September

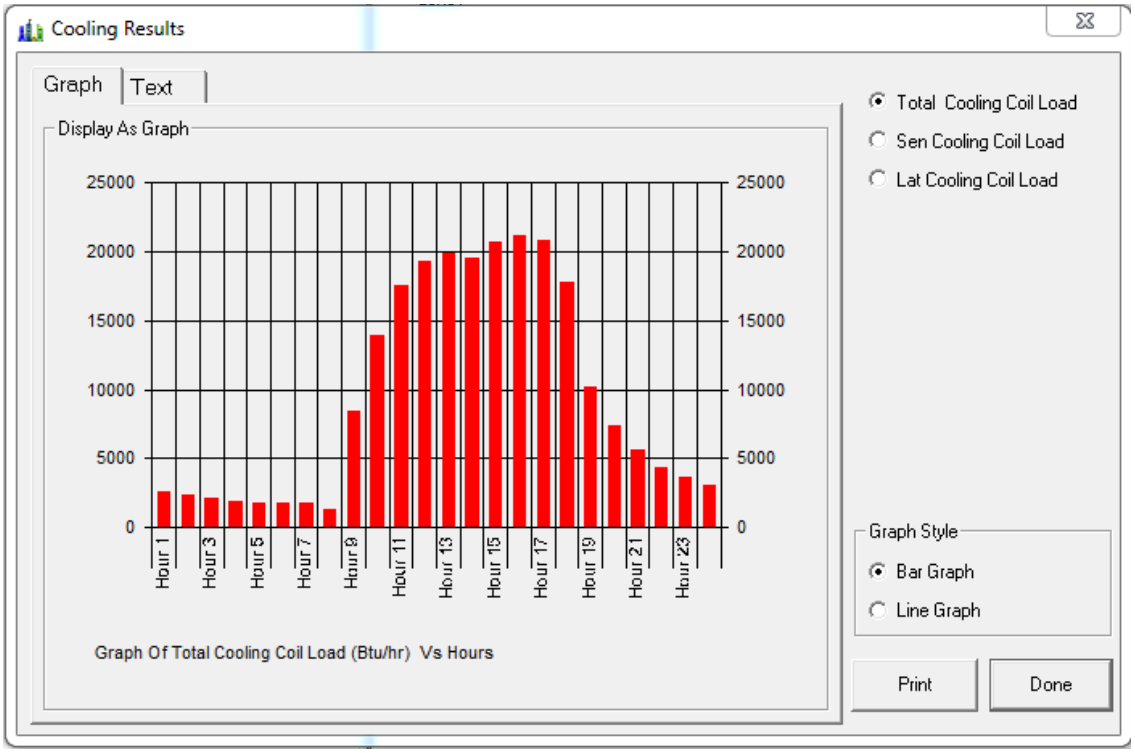
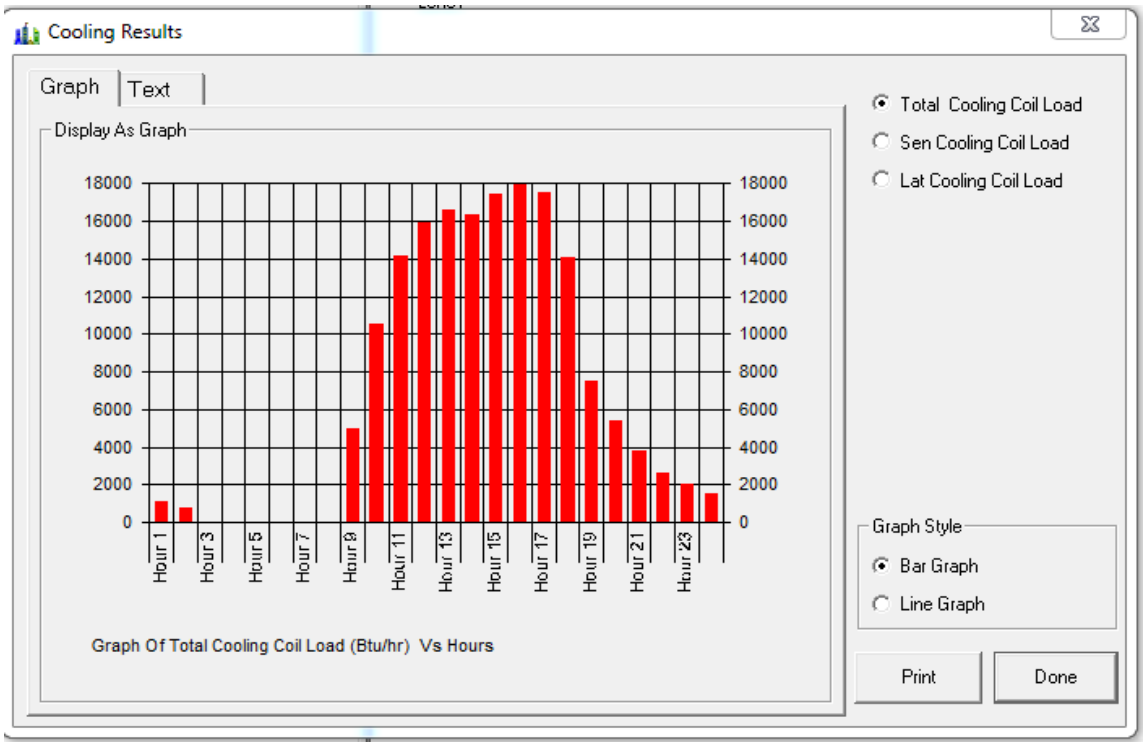


Figure 18 October



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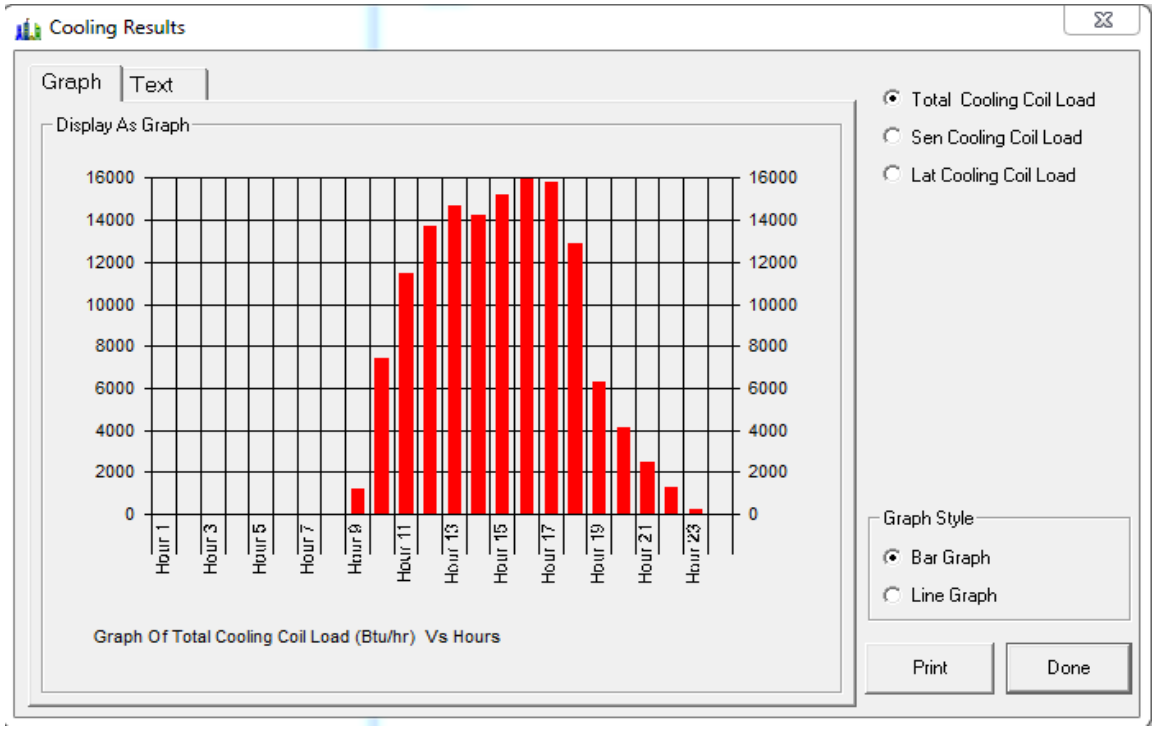

Figure 20 December

6. Heat Transfer Analysis

6.1 Heat Transfer Approach

The design of the system required pipes of polyethylene pipes buried on the ground to dissipate the heat needed to decrease the water temperature from 94°F to 85 °F. These pipes act as a heat exchanger dissipating energy to the ground as water flows from the pipes lowering the temperature. On the analysis different type of heat transfer were performed to know the total length needed to accomplish the goal. For example, the convection, conduction and the total resistance are the principal cases of study. The analyses were done by separated the problem into three main parts. The first part of the analyses was on the evaporator. On the evaporator, the inlet water has a temperature of 85°F with a heat rejection of 35.35 MBTUH; the exit temperature is unknown.

Inside it the water never mixes with the Freon; however, the water absolves the energy as it passes through by increasing the temperature. The second part was the calculation of the total thermal resistance of the system including the convection, conduction and the ground resistance. Finally, the last part was the total heat transferred from the water through the ground as it flows. Also, the total length was calculated.

There are several assumptions on the procedure during the calculations. For example, the system was looked as a steady state with no variation of temperature over time; also, it was assumed that at any point the surface temperature of the pipe remains the same for the entire length. There is not variation of temperature on cylindrical shape. Furthermore, the heat transfer on the cylindrical pipe occurs only in radial direction.

6.2 Temperature of the Water Leaving Condenser

The heat pump is two tons units taken from the cooling load calculations. The specifications of the AC unit shows water in at 85°F with a heat rejection of 35.35 MBTUH and the volumetric flow is 8 GPM as shown in Figure 15**.** As the main fluid is water which is incompressible its properties are found on the water table. Also, the volumetric flow is constant then the conservation of energy was applied.

$$dq_{conv} + \dot{m}(c_v T_m + pv) - \left[\dot{m}(c_v T_m + pv) + \dot{m}\frac{d(c_v T_m + pv)}{dx}dx\right] = 0$$

$$dq_{conv} = \dot{m}d(c_v T_m + pv)$$

The specific heat at constant pressure remains constant and the equation becomes

$$dq_{conv} = \dot{m}c_p dT_m$$

6.3 Total Thermal Resistance

The calculation of the total thermal resistance involved several parameters to consider. The thermal resistance model was used to determine the rate of the steady heat transfer through composites medium. For example, the complete system medium involved the convection from the water, the conduction of the polyethylene cylindrical pipe and the geometry of a large medium in this case which is the ground. Such geometries can be approximate as one dimension with a simple analytical solution. All equations are established below.

$$q = UA\Delta T$$

$$UA = \frac{1}{R_{tot}}$$

6.4 Convention Thermal Resistance of the Water

The analysis of convection of the water requires the determination of some of the dimensionless number like the Reynolds number and the Nusselt number to find the overall heat transfer coefficient using the properties of the water and the volume flow from the specifications.

$$Re = \frac{4Q\rho}{\pi D_i \mu g_c}$$

The Nusselt number depends on the Reynolds number and the Prandtl number to use the right equation in an internal flow. After solving for the Reynolds number, the fluid resulted turbulent given the Nusselt equation:

$$Nu_{D,1} = 0.023 {Re_D}^{4/5} Pr^n$$

Where n = 0.4 for heating and n = 0.3 for cooling; the equation is valid for:

$$\begin{bmatrix} 0.7 \leq \mathrm{Pr} \leq 16,700 \\ Re_D \geq 10,000 \\ \frac{L}{D} \geq 10 \end{bmatrix}$$

The heat transfer coefficient was calculated using the water properties using the results on the previous step. The equation is given by:

$$Nu = \frac{h_w D_i}{k_w}$$

Finally, the convection thermal resistance of the water flowing on an internal cylindrical pipe was calculated using the equation:

$$R_{conv} \cdot L = \frac{1}{\pi D_i h}$$

$$q = UA\Delta T$$

6.5 Conduction Thermal Resistance of the Cylindrical Wall

The heat transfer on a cylindrical pipe can be modeled as steady and one dimension. The temperature depends in one direction, radial direction, and there is no change of temperature at any point. Furthermore, the rate of heat transfer remains constant for the entire pipe. Solving the equation for a cylindrical pipe gives the equation for thermal resister as follow.

$$R'_P = R_p \cdot L = \frac{\ln\left(\frac{r_o}{r_i}\right)}{2\pi k_p}$$

6.6 Conductive Heat Transfer through the Ground

Many problems require the analysis of complicate geometries which no simple solutions are available. In those cases where the temperature of the two medium are constant the rate of heat transfer is expressed as

$$Q = SK(T_1 - T_2)$$

Where S is the conduction shape factor and K is the thermal conductivity of the medium. Solving the equation into the thermal resistance of the ground is given by.

$$R_{g.cond} = \frac{1}{SK_g}$$

The conduction factor S is a circular isothermal cylinder of length L in the mid-plane of an infinite wall where Z is greater than 0.5 D.

$$S = \frac{2\pi L}{\ln(8z/\pi D)}$$

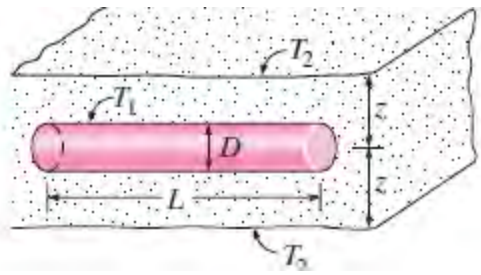

Figure 21 S Factor

7. Heat Transfer Calculations

7.1 Calculation of Temperature Leaving the Condenser

Using heat transfer equations, the first calculation was to find out the temperature leaving the condenser inside the geothermal heat pump. At maximum working capacity the heat pump has a heat rejection of 35,350 BTUH. The nominal pipe diameter at this location is 1 ¼". The temperature of the water that leaves the ground coils to come into the heat pump is at 85°F. In order to solve for the temperature of the water leaving the condenser a series of a trial and errors were calculated. First, a temperature of 100°F was assumed with its properties. After three trials, the final temperature was roughly 94F. Therefore, in order to solve for the total heat rejection, the properties of the water was taken from the bulk temperature of 90°F which is in between the inlet at 85°F and the outlet at 94°F.

Table 12 Condenser Data

Condenser Data 1 ¼" Nominal Diameter		
Volume Flow (V)	8	GPM
Temp Water in (Ti)	85	°F
Diameter (Din)	0.11	ft
Total Heat Reject (Q)	35350.350	BTU/hr

Table 13 Trial and Error

Trial & Error to Determine the Outlet Water Temperature							
Temp out [°F]	Temp in [°F]	Cp [BTU/lbm*°F]	Density [lbm/ft^3]	Volumetric Flow [m^3/s]	Mass Flow [kg/s]	Const. Heat Rejection [BTU/H]	Tout [°F]
100	85	0.999	62.12	8	3989.2073	35350	93.8703
93.87028002	85	0.999	62.22	8	3995.629	35350	93.856
93.85602371	85	0.999	62.23	8	3996.2712	35350	93.8546

Table 14 Water Properties in a Condenser

Water Properties at 90 °F		
Density (ρ)	62.12	Lbm/ft^3
Specific Heat(Cp)	0.999	BTU/Lbm* °F
Pressure (P)	0.6988	PSI

Table 15 Outlet Temperature Calculations

Calculation at 1 1/4 Nominal Diameter		
Volume Flow (V)	1.070296	ft/min
Area (A)	0.00950334	ft^2
Velocity (Ve)	112.6231409	ft/min
Mass Flow Rate (m)	3989.207251	Lbm/hr
Temp Water out (Te)	93.87028002	°F

7.2 Calculations for the Ground Coil ¾ Nominal Diameter

The second set of calculations was found for the ground coils of the geothermal tubing with a nominal pipe size of ¾". This pipe is buried underground at a depth of 6 feet where the temperature is constant at 77°F. This is the temperature of the soil that essentially removes the heat from the water to cool it from 94°F to 85°F. The properties of the water were taken from the water table [37] at the bulk temperature of 90°F. The

results for the volumetric flow rate, cross sectional area and velocity are established in Table 16,17, 18.

Table 16 Ground Coil Data ¾" Nom. Dia.

Data from the specifications ¾" Nom. Dia.		
Volume Flow (V)	1.33	GPM
Temp Water in (Ti)	93.87028	°F
Temp Water in (Te)	85	°F
Temp Ground	77	°F
Diameter (Di)	0.068	ft
Diameter (Do)	0.0875	ft
Surface Area (As)*L	0.21352	ft^2

Table 17 Water Properties at 90 F

Water Properties at 90 °F		
Density (ρ)	62.12	Lbm/ft^3
Specific Heat(Cp)	0.999	BTU/Lbm* °F
Dynamic Viscosity (μ)	1.842	Lbm/ft*hr
Thermal Conduct (K)	0.358	BTU/hr*ft*°F
Prandtl Number (Pr)	5.14	

Table 18 Calculation Results for Ground Coil

Calculation Results for Ground Coil		
Volume Flow (V)	0.17793671	feet/min
Area (A)	0.00363169	ft^2
Velocity (Ve)	48.9955722	ft/min

7.3 Overall Thermal Resistance

The next calculation was to determine the total thermal resistance acting on the system. The overall thermal resistance in this case consists of first the convection heat transfer from the water to the inside the ¾" pipe. The second is the conduction from the polyethylene pipe and the third is the resistance of the ground. Every thermal resistor was

solved in function of the length using their corresponding properties and their specific case as established below.

Table 19 Thermal Resistance of the Water

Thermal Resistance - Convection from the Water		
Reynolds Number (Rey)	6548.920086	
Nusselt Number (Nu)	42.45731486	
Heat Coefficient (h)	217.1383456	BTU/hr*ft*°F
Convection Resistance (Rw)*L	**0.020952449**	hr*°F/BTU

Table 20 Thermal Resistance of the 3/4 Polyethylene Pipe

Thermal Resistance - Conduction of Polyethylene Pipe		
Thermal Conduct Pipe (K)	0.225	BTU/hr*ft*°F
Conduction Resist (Rcc)*L	**0.157841**	hr*°F/BTU

Table 21 Thermal Resistance of the Ground

Thermal Resistance of the Ground		
Deep of the Pipes (Z)	0.75	ft
Conduction Shape Factor (S)*L	2.037919912	ft
Thermal Conduct Ground (Kg)	1.215	BTU/hr*ft*°F
Ground Resister (Rg)*L	**0.40386536**	hr*°F/BTU

7.4 Total Heat Reject to the Ground

The mass flow of the water was calculated with the volumetric flow of the water and the density. The water enters the ground at 94°F and exits at 85°F; the properties of the water were taken from water Table 18. The total heat rejection through the ground was calculated and established below.

Table 22 Total Heat Transfer

Total Heat Transfer from the Water		
Mass Flow Rate (mw)	663.2057055	Lbm/hr
Total Heat Transfer one Loop (Qw)	3889.310001	BTU/hr
Total Heat Transfer 6 Loop (Qw)	23335.86	BTU/hr

7.5 Total Length

Finally, the total length needed to decrease the water temperature from 94°F to 85°F was calculated. The total length was calculated to know the amount of pipe required to dissipate 5876.935 BTU/H with a temperature of the ground at 77 °F all year. The analysis was done by taking one loop into consideration; however, the completed system has 6 loops buried underground as shown in Figure 32. The total length of the one loop has to be multiplied by 6 to get the correct value. The total length is 2400 ft.

Table 23 Total Length

Total Length		
Total Resister (Rt*L)	0.582665219	hr*°F/BTU
Length (L)	386.0369495	ft
Total Length (L)	2316.218973	ft

8. Cooling Tower Design

8.1 Cooling Tower Introduction

Cooling towers are used to reject heat into the atmosphere. They may be either a wet cooling system or dry cooling system. The wet system uses cool water to remove the heat from hot water by means of evaporation. The dry system uses ambient air to cool the

working fluid close to the dry bulb temperature. Cooling towers are usually used as auxiliary heat rejecting systems in combination with geothermal heat pumps and or other heat exchangers. Cooling towers are used for many processes and applications including but not limited to HVAC systems, chemical plants and oil refineries.

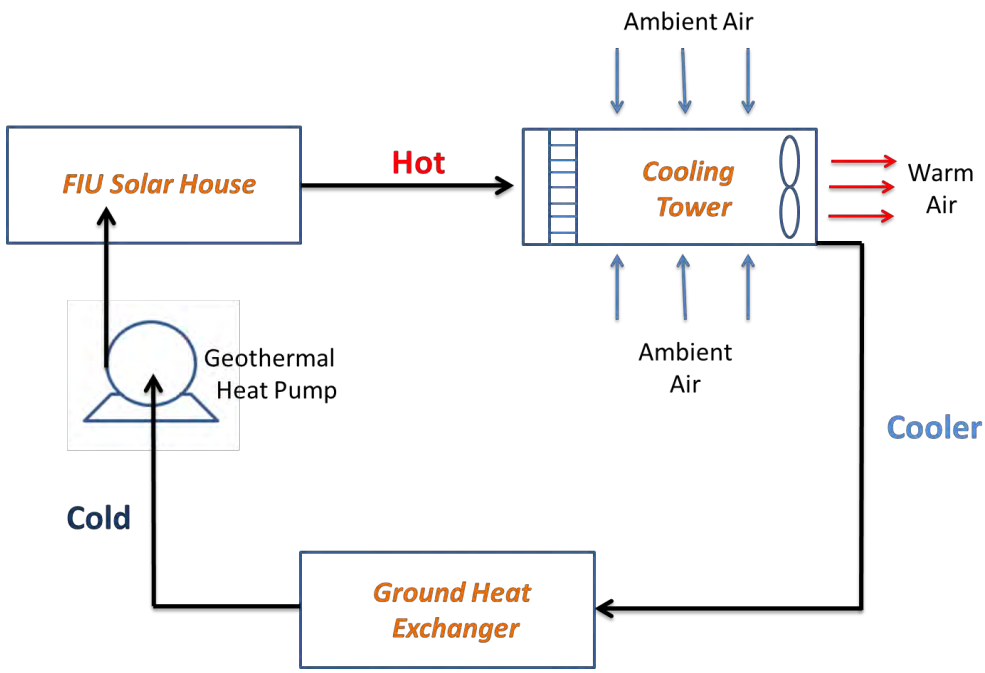

Figure 21 Cooling Water System

8.2 Types of Cooling Tower

There are two main categories that cooling towers fall into. The first is natural draft and the second is mechanical draft. The natural draft cooling towers are extremely large concrete chimneys that allow a large amount of air to run through. These cooling towers are used to cool water rates that are greater than 26,486 ft^3/min and are primarily used by utility power stations (Bureau of energy efficiency India).

The second type is the mechanical draft which is more commonly used and is composes of a large fan to suck the hot air from the working water. The cooling rates of these types of tower depend on the diameter of the fan, the power of the motor, speed of operation, static pressure and volumetric flow rate in cubic feet per minute.

Mechanical draft towers are available in three different types of configurations including counter-flow induced draft, counter-flow forced draft and cross flow induced draft (Bureau of energy efficiency India).

Counter flow induced draft and forced draft use the same technique. The hot water enters at the top and the fan induces cool air from the bottom to the top. For the cross flow induced draft, the hot water is brought from the top and a fan induces the air from the opposite side.

8.3 Cooling Tower Considerations

For the hybrid system in this project, a dry cooling tower was designed with a mechanical induced draft in a cross flow arrangement. A wet cooling system was rejected for several reasons that made it less cost effective. The dry cooling tower, for example, eliminates the use of water all together therefore saving water, electricity and staying greener. Consequently fewer components are required such as an extra water pump and several nozzles. The water pump is used to direct the water towards the top of the tower and nozzles to spray it down.

Furthermore the advantages of a cooling tower is not only that its more cost effective in removing low grade heat but also more compact in size, maximizing space. The dry cooling system will only consists of the electric motor, fan, air-cool louvered fin

heat exchanger and the housing. The figure below demonstrates how the cooling tower should work like.

9. Heat Exchangers

9.1 Heat Exchanger Introduction

Heat exchangers are used to facilitate the exchange of heat between two fluids that are not mixed. Both fluids are of two different temperatures. Heat exchangers are seen in many mechanical applications such as car radiators, in the HVAC industries and chemical processing plants.

The transfer of heat will be analyzed through the convection of the fluids and the conduction of the fin walls. Several parameters that will be analyzed in order to effectively design the dry cooling tower will be the overall heat transfer coefficient U and the logarithmic mean temperature difference LMTD.

9.2 Heat Exchanger Analysis

The dry cooling tower is a water-to-air heat exchanger in which the two fluids will be separated by a solid copper wall. The heat from the hot water will be transferred to the wall by convection, then through the wall by conduction and then from the wall to the cold air again by convection. This is shown in figure [10].

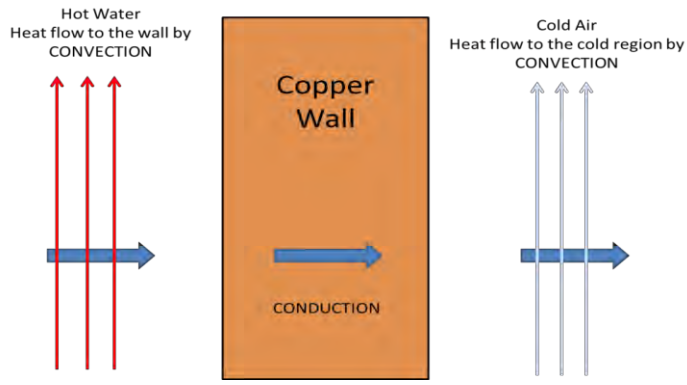

Figure 22 Heat Transfer through Wall

The overall heat transfer coefficient U accounts for all the effects on the transfer of the heat from convection to conduction and back to convection. These are measured in what is called thermal resistances. For convection inside and outside the copper tube the following equations will be used:

Inside Convective Thermal Resistance

$$R_i = \frac{1}{h_i A_i}$$

Outside Convective Thermal Resistance

$$R_0 = \frac{1}{h_0 A_i}$$

For the copper tube, the conduction formula is used and depends on the outer and inner diameter of the tube as well as its thermal conductivity *k* and total length *L*.

Conduction Thermal Resistance

$$R_{\substack{copper \\ Wall}} = \frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi kL}$$

The overall heat transfer coefficient U is dominated by the smaller convection coefficient which is the air side. As a consequence the air side impedes on the overall UAs. In order to fix this issue, fins are added to increase the overall heat transfer in the air side—especially for gas and fluid heat exchangers such as the dry cooling tower. The overall heat transfer coefficient in an air cooled heat exchanger was approximated to about 738 W/(m^2-°C) or 130 BTU/(h-ft^2-°F). This number was approximated from the range of Water-to-air in finned tubes (water in tubes) 400 < U < 850 BTU/(h-ft^2-°F) (Cengel, 2003).

Representative values of the overall heat transfer coefficients in heat exchangers

Type of heat exchanger	U, W/m^2 · °C*
Water-to-water	850–1700
Water-to-oil	100–350
Water-to-gasoline or kerosene	300–1000
Feedwater heaters	1000–8500
Steam-to-light fuel oil	200–400
Steam-to-heavy fuel oil	50–200
Steam condenser	1000–6000
Freon condenser (water cooled)	300–1000
Ammonia condenser (water cooled)	800–1400
Alcohol condensers (water cooled)	250–700
Gas-to-gas	10–40
Water-to-air in finned tubes (water in tubes)	30–60†
	400–850†
Steam-to-air in finned tubes (steam in tubes)	30–300†
	400–4000‡

*Multiply the listed values by 0.176 to convert them to Btu/h · ft² · °F.
†Based on air-side surface area.
‡Based on water- or steam-side surface area.

Figure 23 Cengel's Overall Heat Transfer Coefficient Heat Exchanger

For the dry cooling tower, a seamless copper tube, nominal ¼ Type K will be used with an inside diameter of .775 cm and wall thickness of 0.178 cm (Janna, 2011).

$$A_i = \frac{\pi {D_i}^2}{4} = 4.717x10^{-5}\, m^2 = 5.08x10^{-4}\, ft^2$$

$$A_0 = \frac{\pi {D_0}^2}{4} = 7.133x10^{-5}\, m^2 = 7.68x10^{-4}\, ft^2$$

9.3 Types of Heat Exchangers

There are several types of heat exchangers and flows that are associated with each other. Two types of heat exchangers are the double pipe and the plate fin heat exchangers. In the double pipe heat exchanger the parallel flow and the counter flow arrangements are possible. In the parallel flow heat exchanger, the hot and cold fluids enter at the same time and move in the same direction. In the counter flow heat exchanger, the hot and cold fluid enters at opposite ends and move towards each other. The plate fin heat exchanger has a cross flow arrangement in which the two fluids move perpendicular to each other. The cross flow head exchanger can either be mixed or unmixed as shown in Figure 25. In a heat exchanger where fins are present, the fluids are said to be unmixed because the plate fins force the fluid to flow between two specific fins and prevent the air from moving parallel to the tubes. In a mixed cross flow heat exchanger, the fluid is unrestricted to move in the transverse direction.

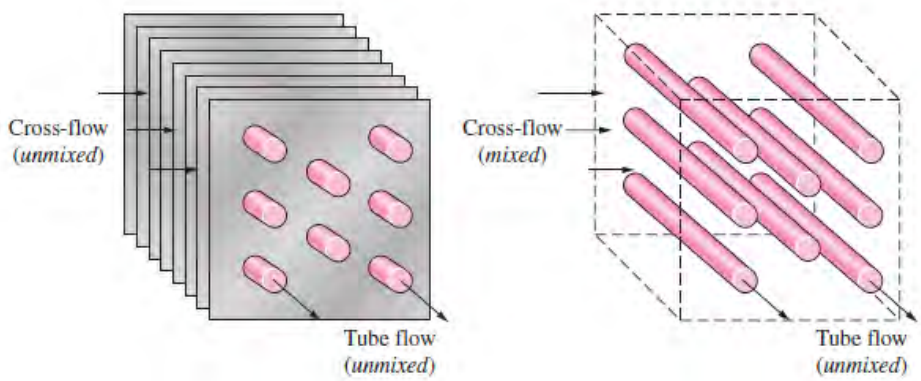

Figure 24 Flow Configurations in Cross-Flow Heat Exchanger

9.4 Heat Exchangers Considerations

For the purpose of designing a dry cooling tower, the heat exchanger will consist of plate fins and a cross flow configuration. This will allow the cooling tower to be compact in size. A heat exchanger with an area density or $\beta > 700\ m^2/m^3$ is considered compact. This ratio determines the effectiveness of heat transfer in a small or condensed volume. Several examples of a compact heat exchanger include car radiators ($\beta \approx 1000\ m^2/m^3$) and the human lung ($\beta \approx 20{,}000\ m^2/m^3$) (Cengel, 2003).

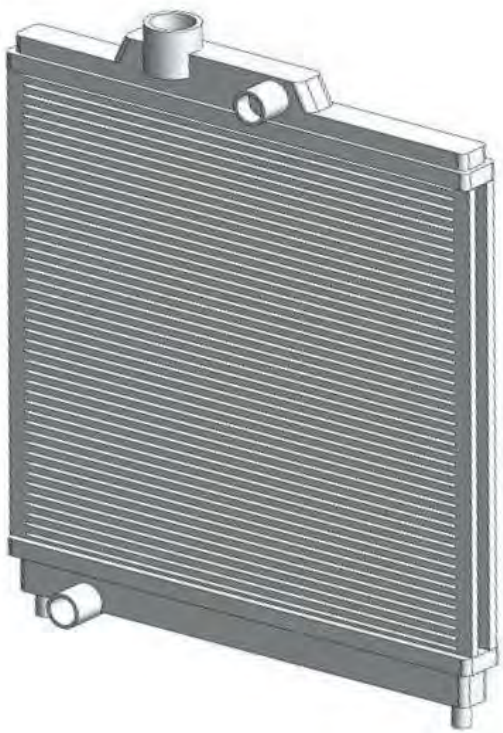

Figure 25 Louvered Fin Heat Exchanger

10. Heat Exchanger Calculations

10.1 Introduction

The analysis was based into the fined- tube heat exchangers in order to know the size. The conditions are established as followed. For example, air enters at constant volume of 5638.86 ft^3/min [Table 32] with a temperature of 87 °F on the air side, the volumetric flow of the water is constant at 8GPM, a desired amount of 4°F will be used to decrease the water temperature. The temperature of the air (87°F) was taken from the average air temperature from April through September in Miami, FL [Appendix C1 Average Monthly Climate and Weather Indicators in Miami Florida]. The two methods that can be used are: the Effectiveness- NTU and the LMTD methods. Both methods are correct but using the Effectiveness- NTU a trial and error is eliminated.

10.2 Total Heat Rejected by Water

The first part of the calculation is to know the total heat rejected to the air in order to balance effectively the heat transfer. The formula used was the conservation of energy and the amount of temperature that the cooling tower will be reduced is 4°F. The total system for the underground geothermal loops releases about 8°F. The other 4°F will be released as water circulate through the underground pipe .The temperature of the water on the heat pump inlet is 85°F and the exit temperature is 93.87°F [Table 15] releasing a total heat of 35350.00 BTU/hr [Table 12]. This temperature of 4°F was chosen as a reference to balance the heat transfer. The properties of the water and calculations are established below.

$$Q = m * Cp * \Delta t$$

10.3 Final Temperature of the Air

Again using the conservation of energy, the final temperature of the air was found. The total heat rejected by the water was calculated by applying the conservation of energy; also, the final temperature of the air was calculated. The volumetric flow and properties of the air are established below with those conditions.

The mass flow rate of the air was calculated using the Ideal Gas law. The properties of the air were taken using the air properties at 87°F. The properties and calculations are established below.

$$PV = MRT$$

$$Q = m * Cp * \Delta t$$

Table 24 Air Mass Flow Rate

Air Mass Flow Rate (ma)		
Temp in (Tin)	550.67	R
Pressure (P)	14.7	Lbf/inch^2
Constant (R)	53.35	ft*Lbf/Lbm*R
Volume Flow (V air)	4000	ft/min
Mass Flow Rate (ma)	**17292.79638**	Lbm/hr

Table 25 Temperature of Air Exiting the Radiator

Temperature of Air Exiting the Radiator		
Specific Heat(Cp)	0.24	BTU/Lbm* °F
Temp in (Tin)	87	°F
Total Heat Reject (Q)	15940.87218	BTU/hr
Mass Flow Rate (ma)	17292.79638	Lbm/hr
Temp Tout (To)	**90.84092308**	°F

10.4 Inner Tube Diameter Calculations

The objective of the calculation is to size the appropriate heat exchanger. As known, the water enters at 8GPM equal to 1.07 ft^3/min and the water velocity inside the heat exchanger cannot be more than 4 ft/s because it will results in very high head losses. The nominal diameters choose for the heat exchanger was a 1/5” cooper with specifications below.

The material of the inside pipe was chosen to be copper for its high thermal conductivity. From the result above the diameter is a 0.5 inch nominal size with specification below.

Table 26 Copper Tube Dimensions

Copper Tube Dimension								
Nominal Pipe Size, in	Diameter				Wall Thickness		Inside Cross-Sectional Area	
	O.D.		I.D					
	in	mm	in	mm	in	mm	ft^2	10^-3 m^2
½	0.525	15.9	0.4831	13.8	0.020	1.02	0.00125	0.150

10.5 Water Heat Transfer Coefficient

The Reynolds number is needed to find the inside heat transfer coefficient. There are three types of flows depending on the Reynolds number. A flow could be laminar, transitional or turbulent. If the Reynolds number is less than 2,300 is laminar, if it is more than 10,000 turbulent and transitional in between. Depending on the Reynolds number, the formula for the inside heat transfer coefficient is used. The formula used are establish below in addition with final results.

$$Re_D = \frac{\rho\, VD}{\mu}$$

$$h_i = 0.0023\, \frac{K}{D}\, (Rey)^{0.8}\, (Pr^{0.3})$$

Table 27 Inside Heat Transfer Coefficient

Inside Heat Transfer Coefficient		
Density (ρ)	62.12	Lbm/ft³
Thermal Conduct (K)	0.358	BTU/hr*ft*°F
Prandtl Number (Pr)	5.14	
Dynamic Visc (μ)	1.842	Lbm/ft*hr
Velocity in	240	ft/min
Inside Diameter(Di)	0.4831	inch
Inside Diameter(Di)	0.0402583	ft
Reynolds (Rey)	19550.601	
inside heat transfer cohef (hi)	905.6744	BTU/hr-ft^2-°F

10.6 Air Side Heat Transfer Coefficient

To find the air side heat transfer coefficient, several data are available for different types of fin configurations that are established in the manufacturing table. Some of the parameters are the friction factor, j factor, fin spacing and the jp in function of the Reynolds number. The table used was the Heat transfer and fanning friction factor data for the plate-fin-tube coil with various fin spacing as shown in Appendix D1 - Spitler, 2003, HVAC - Figure 14-12 Fanning Friction Factor.

To compute the air side heat transfer coefficient it is necessary to know the air velocity. The air velocity cannot exceed 1000 ft/min (Spitler, 2005). Consequently, an air face velocity of 900ft/min is assumed. The subscript *fr* refers to face coil, *c* refers to minimum flow and σ to ratio of minimum flow area to frontal area; all formulas and results are states below and can be referenced from Appendix D1 Fanning Friction Factor.

$$G_c = \frac{G_{fr}}{\sigma}$$

$$Re_D = \frac{G_c D}{\mu}$$

$$\frac{A}{A_t} = \frac{4}{\pi}\frac{X_b}{D_h}\frac{x_a}{D}\sigma$$

$$Jp = Rey^{-0.4}\left[\frac{A}{A_t}\right]^{-0.15}$$

$$j = \frac{h_o}{G_c C_p}\left[\frac{\mu C_p}{k}\right]^{(\frac{2}{3})}$$

Table 28 Outside Heat Transfer Coefficient

Outside Heat Transfer Coefficient		
Out Diameter (Do)	0.04375	ft
Density (ρ)	0.0728	Lbm/ft³
Specific Heat(Cp)	0.24	BTU/Lbm*-°F
Thermal Conduct (K)	0.01505	BTU/hr*ft*°F
Prandtl Number (Pr)	0.7275	
Dynamic Visc (μ)	0.04554	Lbm/ft-hr
Face Velocity (Vf)	900	ft/min
Gfr	3931.2	Lbm/hr-ft^2
Minimum Front Area(σ)	0.55	
Gc	7147.636364	Lbm/hr-ft^2
Reynolds (Rey)	6866.690622	
Xa	0.09025	ft
Xb	0.104166667	ft
Dh	0.0111	ft
A/At	13.55647364	
Jp	0.019745994	
Value (j)	0.0062	
Outside heat transfer coefficient (ho)	**13.16396954**	BTU/hr-ft^2-°F

10.7 Overall Heat Transfer Coefficient

The overall heat transfer coefficient is the combination of the inside heat transfer coefficient (water side) and the outside heat transfer coefficient (air side). Also, it is influenced by the thickness (Δx), thermal conductivity (K) and the fin efficiency (η). The larger the coefficient, the easier heat is transferred from its source to the product being heated. The equations and results are represented by the following.

$$\frac{1}{U_o} = \frac{1}{h_o \eta_{so}} + \frac{\Delta x}{K\left(\frac{A_m}{A_o}\right)} + \frac{1}{h_i \eta_{si}\left(\frac{A_i}{A_o}\right)}$$

$$\frac{A_i}{A_o} = \frac{D_i}{x_a}\frac{\pi}{x_b}$$

In which subscript *o* refers to air side and subscript *i* refers to water side. Several assumptions were established in order to solve for the overall heat transfer coefficient. For example, the second term on the equation is zero due to the high conductivity of the copper, the outside fin efficiency (η_{so}) is 0.75 and the inside fin efficiency $\eta_{si}$is 1. Solving the equation gives the following results which can also be referenced from Appendix D1 Fanning Friction Factor.

Table 29 Overall Heat Transfer Coefficient

Overrall Heat Transfer Coheficient		
Inside Diameter(Di)	0.040258333	ft
Xa	0.09025	ft
Xb	0.104166667	ft
Outside Efficiency	0.75	
Inside Efficiency	1	
Alfa	170	ft^-1
Ai/Ao	0.079137401	
Overall Heat Transfer Coefficient (Uo)	8.677626767	BTU/hr-ft^2-°F

10.8 Geometric Configuration

As mentioned before the Effectiveness- NTU method has certain advantages over the LMTD. Referring to effectiveness-NTU curved the final configuration as shown in Appendix D2 Effectiveness of Cross Flow Exchanger with unmixed fluids. Formulas used and results are established below.

$$C_{air} = C_{coul} = C_p\, m_{air}$$

$$C_{water} = C_{hot} = C_p\, m_{water}$$

$$\epsilon = \frac{t_{hi} - t_{ho}}{t_{hi} - t_{ci}}$$

$$NTU = \frac{AU}{C_{min}}$$

Table 30 Effectiveness NTU Method for Outside Area

Effectiveness NTU to obtain Outside Area		
Mass Flow Rate Water (mw)	3989.207251	Lbm/ft³
Specific Heat Water(Cp)	0.999	BTU/Lbm- °F
Mass Flow Rate Air (ma)	17292.79638	ft
Specific Heat Air(Cp)	0.24	BTU/Lbm- °F
C-water (hot)	3985.218044	BTU/(hr-°F)
C-air (Cool)	4150.27113	BTU/(hr-°F)
Cmin/Cmax	0.96023077	
T(h-in)	93.87028002	°F
T(h-out)	89.87028002	°F
T(c-in)	87	°F
T(c-out)	90.84092308	°F
Effectiveness	0.582217899	
NTU	1.23	
Area (Ao)	564.8800444	ft^2

The total volume of the heat exchanger (V), face area (A_{fr}), depth (L) and numbers of rows (Nr) are given by the equations and results below.

$$V = \frac{A_o}{\alpha}$$

$$A_{fr} = \frac{Q}{f_r}$$

$$L = \frac{V}{A_{fr}}$$

$$Nr = \frac{L}{x_b}$$

Table 31 Heat Exchanger Geometric Configurations

Geometric Configurations		
Area (Ao)	564.88004	ft^2
Alfa	170	
Total Volume	3.3228238	ft^3
Face Area	4.4444444	ft^2
Depth (L)	0.7476354	ft
Numbers of Rows (Nr)	7.1772994	
Numbers of Rows (Nr)	8	
Numbers of Rows (Nr)*4	32	
Height (H)	2.888	ft
Width	1.5389351	ft

Since the N must be an integer and a multiple of two for the flow arrangement of fig 14.18., eight rows must be used. However another possibility was also analyzed. Recall from the stamen below that velocity cannot exceed 4 ft/s at least eight rows are required and the water connections must be on the same end of the heat exchangers. The problem is that the velocity still high and the configuration were so staggered so the number of rows was increased to 32. The new configuration obtained was over the parameters.

10.9 Louvered Fin

A louvered fin heat exchanger was also considered for the project. For this type of radiator, the water needs to enter at a constant volumetric flow. Then, the volumetric flow is divided into the numbers of channels contained inside. Usually the volumetric flow inside the tubes or channels is laminar. In addition, the geometrical configurations of the louver fins increase the velocity of the air which increases the efficiency of the heat

exchanger. Such geometrical parameters are defined as the fin pitch, louver pitch, and louver angle. The combinations of these three parameters give a higher heat transfer rate.

11. Fans

11.1 Introduction

A fan is a type of mechanical device that produces a flow mainly used to move a fluid (air) from one side to another. Most of the cases fans are powered with an electric motor. In general, fans produce an air flow with high volume and low pressure. There are three different types of fans design depending of the application the axial, the centrifugal and the cross flow.

The blade or propellers of the fan are chosen in relation with the electric motor; the combination of these characteristics is known as fans law or affinity laws. These laws apply to the design of centrifugal and axial flows.

11.2 Centrifugal Fans

The first consideration for the project was the centrifugal fan. These fans are very similar to a scroll cage or a hamster wheel; for this similitude it is the common name. These fans consist of rotating impellers about a central shaft. As the impellers rotates the air enter near the shaft and moves through the duct; the air blows at angle of 90 degree to the intake. A centrifugal fans produce more pressure for a given air volume than the axial flow. It is mainly used in leaf blowers, blow-dryers and on the interior of houses for air conditioning applications. These types of fans produce less noise, but are more expensive.

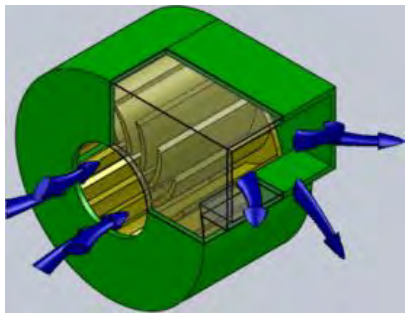

Figure 26 Centrifugal

11.3 Cross Flow Fans

The second consideration was the cross flow fan. This fan is very similar to centrifugal flow; however, it is usually long in relation to the diameter. It also uses an impeller but with forward curves blade to moves the air. The main flow moves in an inward direction and then in an outward radial direction. These type of fan the can supply large volumes of air and is therefore suitable for building into small ventilation units, such as air curtains for example.

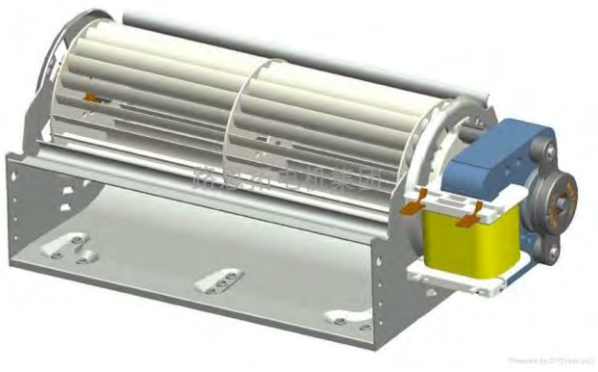

Figure 27 Cross Flow Fan

11.4 Axial Fans

The last consideration was the axial fan. On the axial fan the air moves parallel to the shaft about the blade rotates. These fans are incapable of developing high pressure but they are suitable for moves large volume of air. They are low in cost and have a good efficiency. They have a several configurations depending of the blade to use. Axial fans are better suited for low resistance; they have a huge application in industry, automobile and outside houses for air conditioning application due to its lower cost.

Figure 28 Axial Fans

11.5 Fans Selection

The three types of fans were analyzed in detail to choose the best for the applications. From the calculations of the heat exchanger, it is needed a huge amount of volumetric air flow to cool the water from 93°F to 90° F. from the analyses above the best option is the axial fans. These fan moves a huge amount of air a lower efficiency, the operation cost is lower and it can be manipulate to obtain a better results changing the blade or the motor. It has a disadvantage in relation with the other; the noise tends to be

louder than the centrifugal fans. However, the fan will be installed at the outside of the house for which the noise is not an issue and the cost is much lower.

11.6 Motor Selection

An electric motor is an electromechanical device that converts electrical energy into mechanical energy. There are different types of electric motor depending of the source and running conditions like DC and AC. On the design it was choose to operate an AC motor to generate enough RPM to maximize the volumetric flow of the air that was fixed to 3000 cfm. Also, the motor has to be 120 V due to the conditions of the house.

The motor choose was an AC motor 120 V with four speeds, ¾ horse power and 1075 rpm. The four speeds were used to generate different solutions to produce the required cfm needed in the operation.

12. Fans Calculations

12.1 Blade Selection

There a relationship between RPM, HP from the motor and the type of fan selected like the pitch diameter, total diameter of the propeller and static pressure. Those characteristics are established in tables from different manufactures. Due to the lack of the tables, several tests were done for different blades to choose the most appropriate one. The speed of the motor is 1075 rpm (maximum velocity); from the motor's specification the appropriate current's reading should be about **8.1** amps or less for these conditions. The reading was performed using a Clamp-on Amp Probes. The three blades tested are shown below:

Table 32 Blade Selection

Blade Selection			
Blade Part Number	Diameter(inches)	Pitch Diameter (degrees)	Reading Amps (W)
60804101	18	23	3.2
60559701	20	27	7.5
60761201	26	27	8.8

The relationship between hp and amperage is that if the amps' reading in a motor is too low there is a loss of energy. On the contrary, if the amps' reading on the motor is too high the motor will burn. From the experimental data, the first blade (Part Number 60804101) has a reading of 3.2 amps which is too low to use. The third (Part Number 60761201) has a reading of 8.8 amps which is too high for the motor. The optimal choice would be the second option with a reading of 7.5 amps.

12.2 Air Calculations

The objective of the air calculations is to know the volumetric flow rate than the fan can produce at the four speeds. To know the volumetric flow a velocity has to be measured. The velocity was measure using the Pitot tube.

A Pitot tube is an instrument to measure the total pressure and the static pressure in an air flow. This pressure is given in terms of height inches of water and can be related of the difference of the two pressures by:

$$V = \sqrt{\frac{2\,(P_{total} - P_{static})}{\rho_{air}}}$$

Solving the equations using the Ideal Gas Law it becomes:

$$V = \sqrt{\frac{2\ (H_{total} - H_{static})\rho_{water}\, g}{P_{ambient} / R_g T_{ambient}}}$$

In order to solve the equation the temperature of the air was recorded to be 70°F and the R constant is 287 (J/kg-K). Several convention factors were done to calculate the air velocity. It was assumed the area where the air is flowing to 20" by 20" the outside diameter of the blade. The results are established at the table below.

Table 33 Data for Velocity Calculations

Data for Velocity Calculations								
Water Density (kg/m^3)	Temp °F	Temp K	R (J/Kg*K)	Ambient Pressure (pa)	Gravity (m/s^2)	Blade Area (ft^2)	Motor Area (ft^2)	Total Area (Ft^2)
1000	70	294.2611	287	101325	9.81	2.77	0.16	2.61

Table 34 Fan Air Velocity

Fan Air Velocity					
Motor Speed	Total Pressure in Inches of Water	Total Pressure in meters of Water	Air Velocity (m/s)	Air Velocity (ft/min)	Volume Flow(ft^3/min)
1 - Low	0.11	0.002794	6.759456536	1330.60172	3472.870497
2 - Med Low	0.15	0.00381	7.893344664	1553.80805	4055.439022
3 - Med High	0.22	0.005588	9.559315107	1881.755	4911.380557
4 - High	0.29	0.007366	10.97525037	2160.48243	5638.85913

13. Pressure Loss on the System

13.1 Introduction

The total pressure loss in the system consists of the pressure lost by the fans and the pressure lost by the water as it flows through the cooling tower pipes. The first analysis was with the total pressure loss by the fan. The total pressure in a fan is composed of static and dynamic pressures. The dynamic pressure (also known as the velocity pressure) is dependent on the density of the fluid and the velocity. The static pressure is the total pressure minus the velocity pressure. A pressure measurement device known as a pitot tube is used to measure the air static pressure. And then using Bernoulli's equation, the static pressure is converted into the total volumetric flow rate. The static pressure loss essentially indicates the amount of negative pressure a fan is momentarily creating and how effective it will be drawing air through a the duct. If a fan is operating at a high speed, the amount of pressure lost is higher than a fan operating at lower speeds.

13.2 Fan Considerations That Affect Pressure Loss

The characteristics of a fan play a crucial role in the amount of pressure that is lost as a result of air flow. Different fans shapes and sizes give off different reading for static pressure and volumetric flow rate depending on the power of the motor and the revolution it rotates in one minute. Although the fan's characteristics partly depend on the power of the motor, the physical characteristics of the fan play a crucial role. For example, the diameter of the fan and amount of blades influences the total frontal air face area. Furthermore, the pitch of the blades determines how swiftly the blade cuts through

air and whether it should be placed to rotate in a clockwise or counterclockwise motion. All of these parameters will influence the: outlet velocity measure in feet per minute; volumetric flow rate measured in cubic feet per minute; and the static and dynamic pressures measured in inch water gauge. Given that fan acts very similar to pumps, they can be thought of as a low pressure air pumps (Turner, 2002).

13.3 Measured Static Pressure Loss of the Fan

The fan used in the senior project will be of the propeller type, axial and will use three blades. It will rotate clockwise (when viewed opposite end to the motor) to pull the heat from the radiators and reject it towards the atmosphere parallel to the blade axis. To accomplish this task, the fan will produce a low static pressure loss and a relatively higher volumetric flow rate. The goal is to obtain roughly 4000 cubic feet per minute and 0.015 inch water gauge loss. These parameters are very important to consider because a low pressure fan with a high flow rate increases fan efficiency and reduces noise and stalling.

14.4 Pressure Loss of the Water

The pressure loss of the water was analyzed in order to know the total pressure of the system. The energy method was used to find the modified Bernoulli energy equation. The modified Bernoulli's equation involves the velocity, elevation, minor/major losses and pressure. It represents an energy balance between two points separated by a distance of L. the formula is state below as:

$$\frac{P_1 g_c}{\rho g} + \frac{v_1^2}{2g} + z_1 = \frac{P_2 g_c}{\rho g} + \frac{v_2^2}{2g} + z_2 + \frac{f L v^2}{D_h 2g} + \sum K \frac{v^2}{2g}$$

14.3 Total Pressure Loss

The modified Bernoulli's equation was solved using the Reynolds number and coefficient of loss of every union and pipes used on the completed system it has a total loss of 19.05 feet of head. Then the total loss has to be added to radiator loss to be 19 .09 feet of head (Casas, Alonzo & Cifuentes, 2010).

Table 35Total Head Loss Water Side

Total Head Loss Water Side		
Reynolds (Rey)	19550.60065	
friction factor (f)	0.028	
Loss Coefficient (K)	2.00	
Length (L)	12.31148046	
Head Loss Radiator	8.986266199	ft
Head Loss of Ground Coils	19.09	ft
Total Head Loss	28.07645656	ft

14. Major Components

14.1. Heat Pump

Florida Heat Pump (FHP) Manufacturing designed the geothermal heat pump that is currently in use in the solar house. The heat pump is called the Aquarius II, Premier Series as shown in the Figure 29 below. The model number of the heat pump is AP 025 – 1VTC. The specifications regarding the heat pump are located in Table 3. It is important

to note that these specifications were found on the actual heat pump, located inside the FIU Solar House.

Figure 29 FHP Manufacturing Aquarius II

Table 36 Aquarius Geothermal Heat Pump Specifications

FHP Manufacturing								
Model: AP025-1VTC								
Serial Number: 3540-003-TW0001-T111M00012								
Unit Volts	208 - 230 V		**Phase**	1	**Frequency**	60 HZ	**Min Volts**	197 V
Compressor (EA)	208 - 230 V		**Rated Load Amps RLA (max current)**	11.4 A	**LRA**	52.0 A		
Blower MTR (EA)	208 - 230 V		**Full Load Amps FLA (motor running at 100% Capacity)**	2.80 A	Power	.33 Hp		
Loop Pump (OPT)	208 - 230 V		**FLA**	1.75 A	**Power**	1/6 Hp		
Minimum Circuit Amplicity	18.8 A							
Max Fuse	30							
Refrigerant	R410A							
Factory Charged Per Circuit	73 OZ							
Design Pressure	HI - 450 PSIG LO - 175 PSIG							

The RLA stands for rated load amps. The rated load amp is the maximum current that the compressor will use up during any operating conditions. The LRA stands for Locked Rotor Amps and it depicts the current that the compressor will experience on initial startup. It is important to note that during starting conditions, the voltage that the compressor will experience is a high one. The FLA on the other hand stands for Full Load Amps and it has to do with the blower motor. In essence, the FLA refers to the blower motor's capacity to turn the fan.

The refrigerant that the geothermal heat pump uses is R410A. A refrigerant is used to improve efficiency in HVAC systems such as heat pumps. R410-A is used because it is ozone friendly because it does not contain chlorine and therefore provides environmental benefit (Lennox, 2013).

Another major component of the geothermal heat pump is the tubing laid out under the ground. The geothermal tubes are laid horizontally six feet under the ground as shown in Reference A1, pg. 77.

The geothermal tubes were laid out horizontally for several reasons. The first is one is that it saves more money. Laying tube vertically means digging a deeper hole which costs much more money (U.S. Department of Energy, 2001). Furthermore, it is practical to use horizontal tubes for residential areas where there are no water wells nearby. A slinky method was used because amount of ground available was limited between the two parking spaces. Therefore the slinky method allows more pipe to be fitted in a shorter length trench (Klaassen, 2006).

The hybrid component is the next major component. The auxiliary heat rejecting system will be a cooling tower that will cool with an axial air fan. This fan will be

located with a housing covering the motor and the radiator. It will be directly connected to the geothermal heat pump in the following manner: after the water leaves the house, prior to entering the geothermal loops all over again, it will get cooled by the cooling tower.

14.2 Automatic Shut-Off Valve

The shut off valve is an automated switch control that leads the water flow either directly to the ground coils or bypasses the latter to force the water through the cooling tower. The valve will open or close the system as needed during the operation. Its working condition is normally closed. This function means that the valve normally stays closed but opens when the temperature of the water is passing higher than 92°F. This feature lets the system dissipate more heat in hottest months of year thereby balancing the annual cooling load.

The shut off valve is an OMEGA SV6000 Series 2-way with solenoid valves that are internally piloted with assisted lift valves that have brass, stainless steel construction and FKM seal material. The temperature range from -10°C to 137°C (14°F to 280°F) and FKM O-ring material is ideal for neutral media such as compressed air, inert gases, synthetic oils and water. The specifications are established below in Table 37.

Figure 30 Belimo Shut-off Valve

Table 37 Shut-off Valve Specifications

Shut-off Valve	
Body	Brass
Temperature Range	10 to 137°C (14 to 280°F)
Pipe size	$1^1/_4$
Orifice Size	$1^3/_8$
Power Coil	8 Watts
Supply Power	120 Vac

14.3. Thermocouple

A thermocouple is a type of sensor that will read the temperature of the water flowing through the pipe. It is necessary to incorporate it into the system because it will send the temperature readings to the thermostat which will consequently give the shut-off valve the signal to open or stay closed. The thermocouple works with two different metals that make contact and provide a voltage reading related to the temperature. The type of thermocouple depends on the type of application, the material or metal and the maximum temperature it can read. For the air heat exchanger application, the

thermocouple to be used is a type K, ungrounded junction with the specifications is shown below:

Figure 31 Thermocouple

Table 38 Thermocouple Specifications

Thermocouple	
Mountaing Thread	$^1/_8$ NPT
Temperature Range	To 650 °C (1200°F)
Type	K
Rugged	304 SS

14.4. Thermostatic (DIM Ramp/ Soak Control)

The DIM Ramp/ Soak Control is an electronic control with a digital display that acts like a thermostat recording the temperature reading from the thermocouple and outputting it to the shut-off valve and the fan. The feature of this control is to regulate the desired operating temperature of the system. The specifications are shown below:

Figure 32 Thermostatic

Table 39 Thermostatic Specifications

Thermostatic		
Supply Voltage	100 to 240 Vac, 50/60nHz	
Dimension	Thickness	3.40mm (0.14")
	Depth	99.80mm (3.86")
Operating Temperature	0 to 50°C (32 to 132°F)	

14.5. Electric Motor

The motor chosen was an AC motor with 120 Volts and with four speeds. The four speeds were used to generate different volumetric flow rates for its operation. The specifications of the electric motor used are shown below:

Figure 33 Electric Motor

Table 40 Motor Specifications

Motor	
Horse Power	$^{3}/_{4}$
Imput Voltage	120V
RPM	1075
Watts	8.1 A
Speed	4 High Speed
Rotation Direction	Counter ClockWise
Shaft Diameter	$^{1}/_{2}\ in$
Frequency	60 Hz
Run Capacitor	$^{15}/_{370}\ uF/V$
Motor Body Diameter	$5\frac{5}{8}\ in$
Motor Body Length	$5\frac{13}{16}\ in$
Overall Shaft Length	$3\frac{3}{4}\ in$
Keyed/Flat Shaft Length	$3\frac{1}{16}\ in$
Overall Motor Length	$9\frac{9}{16}\ in$
Number of Phases	1

14.6. Blade Propeller

An axial flow fans have chosen to blow the greater concentration of flow. They also have a lower cost comparing with the other types of fans. These types of fans have a huge application in air condition design

Figure 34 Blade

Table 41 Blade Specifications

Blade	
Diameter	20"
Rotation	CW or CCW
Pitch Diameter	27°

14.7 Radiator

The small radiator that was used is shown below. It is a water to air heat exchanger that will use air as the cool fluid to warm the water which is the warm fluid. Its cross sectional area is 13” x 13” and two of these radiators were used to efficiently dissipate the heat.

Figure 35 Louvered Fin Radiator

15. Prototype System Analysis

15.1 Prototype System Considerations

Since the geothermal heat pump is already exist, the prototype will consist on building a Hybrid air cooling tower, these device will reject heat to the atmosphere, the goal is to balance out the annual load into the ground loop it will make the geothermal pump more efficiency, also by putting a shut off valve to the system, it will control when to turn on or off the cooling tower, only when its needed.

Two designs will be considered. The first one is one large radiator that measures 24” x 24” and the second one is two smaller radiators that measure 13”x13”. The pros of the first design are that it is simpler to build; however, it has no level of uniqueness. The second design, on the other hand, is more unique and will divide the volume flow rate into to two 4 GPM flows.

15.2 Prototype System Designs

The first prototype is shown in Figure 39. It consists of a 24” x 24” radiator that will receive a volumetric flow rate of eight gallons per minute. The same ¾ HP motor was going to be used but it was going to be placed horizontally. However, there were several reasons for rejecting this system. The first one was that the frame was going to very large. The second, it was going to be a plain design with not a lot of ingenuity. The third was that such a large radiator for the cooling purposes was going to cost too much money - $1,650 from Active Radiators. Finally, most louvered fin car radiators, when larger than 13” x 13” tend to be of the rectangular shape and a square shape was what was needed.

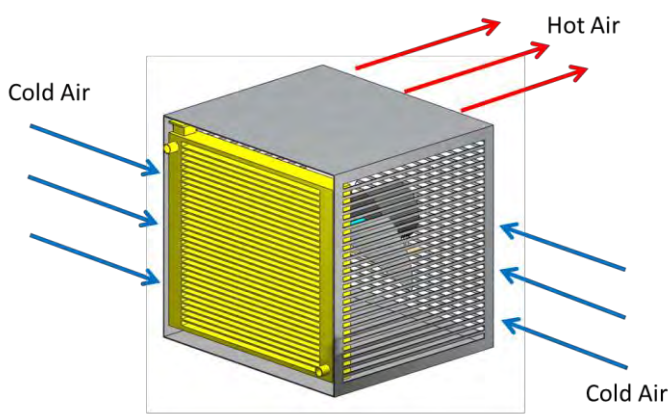

Figure 36 First Prototype Cooling Tower Design

The final design is shown in figure 41. This is the final prototype that was manufactured and chosen for several reasons. The first one was that size of the frame was going to be reduced because instead of using one large radiator, two smaller radiators were going to be used of the size 13” x 13” in the shape of a V. Second, the diagonally placed radiators were a more unique design and puts less stress on the radiators by divided the volume flow rate into four gallons per minute for each radiator. Finally, the electric motor was going to be placed in a vertical position to further remove even more heat.

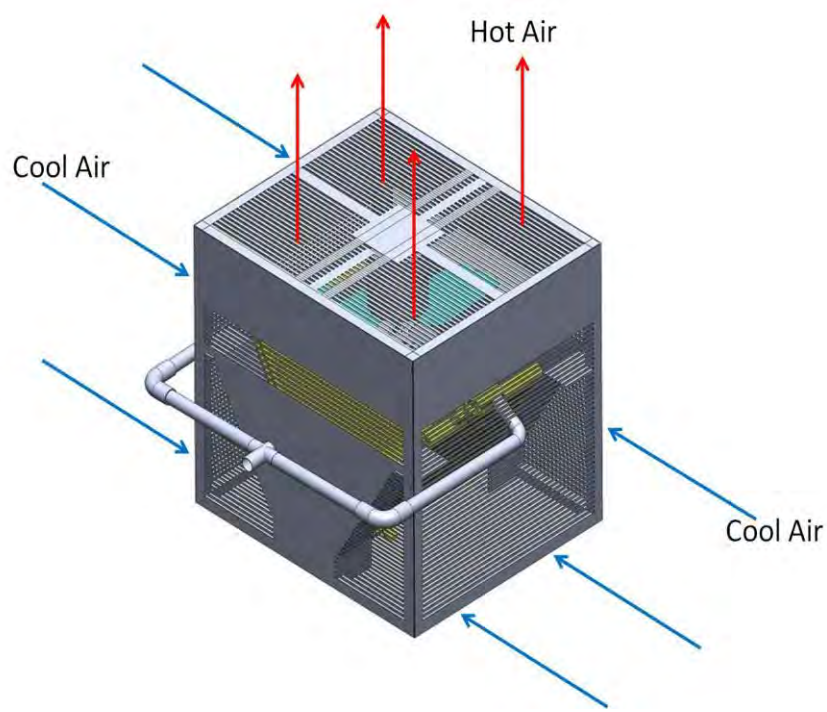

Figure 37 Second Prototype Cooling Tower Design

15.3 Prototype Testing

In order to implement the dry cooling tower to the geothermal heat pump, it needs to be tested before it can put into use. The cross flow heat exchanger will be assembled inside the housing with its electric motor and a 24" fan. A tank will be placed at an elevated position above the tower with hose connecting both the tank and the heat exchanger. Inside the tank, the water will be heated with an internal resistance up to 94°F and then let loose towards the cooling tower at an estimated 4 gallons per minute. While this is going on, another setup function is taking place. The thermocouple is connected to a thermostat that is wired to both the shut off valve and the fan's electric motor. The thermocouple will be placed inside the tank to read the water temperature and as soon as it senses 92°F or higher, it will send a signal to the shut-off valve to open and another signal to the electric motor to turn the fan on. At this point, the water will start flowing through the radiator, cooling off with the induced draft, and a second thermocouple will be placed on the outlet of the heat exchanger to read the temperature leaving. Several tests and studies will be conducted to determine all the necessary modifications to optimize the system. The diagram of what the prototype should look like is displayed below.

16. Frame Structural Design

16.1. Frame Considerations

The geothermal heat pump is already stationed in the FIU Solar House. However, in order to designs the auxiliary heat rejecting cooling tower several materials will be used. The frame of the cooling tower will made from galvanized steel or stainless steel to withstand corrosion from the environment (Bureau of energy efficiency India, 2011). For the fan, aluminum or galvanized steel will be used depending on whether the fan will be centrifugal or propeller (Bureau of energy efficiency India, 2011). For example, centrifugal fans usually are manufactured from galvanized steel whereas propeller fans are made with galvanized steel, aluminum, or reinforced plastic. These components are important because they will be chosen for enhanced corrosion resistance and promote reliability, service and require less maintenance (Bureau of energy efficiency India, 2011).

16.2. Frame Design

The frame will be constructed with aluminum 5052 – H32 in order to resist corrosion and maintain the ability operate in the outside cold and warm weather. The frame will be custom made and the cross section for each aluminum bar will be 1” x 1” with a thickness of 1/8”. Figure 35 depicts the overall frame design and Figure 36 shows the cross section dimensions of each bar. At the top of the frame two parallel bars were implemented to sustain the electric motor.

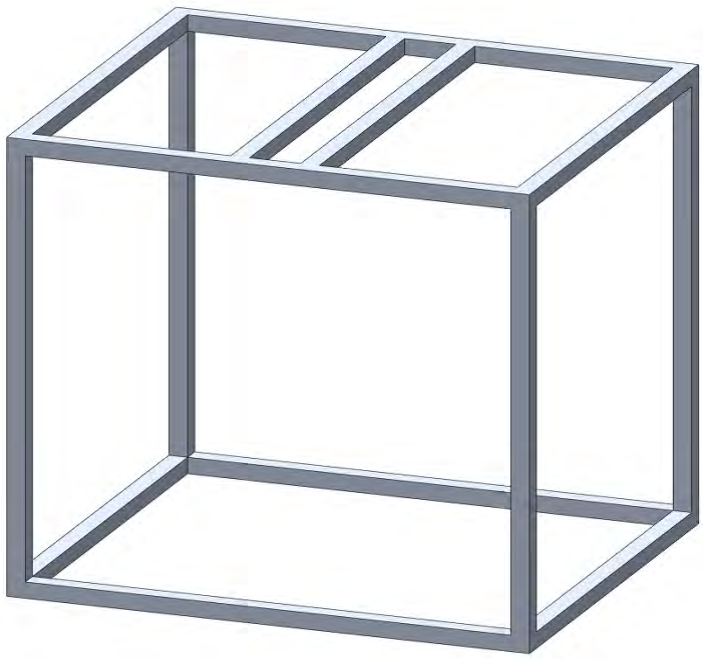

Figure 38 Overall Frame Design

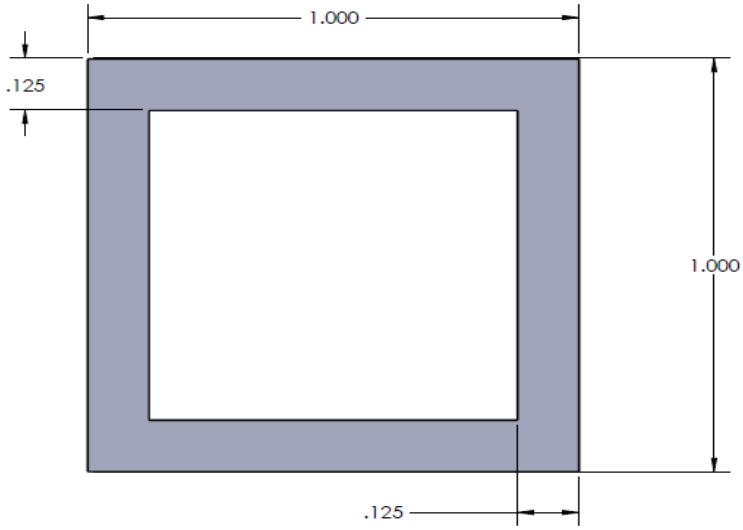

Figure 39 Cross Section of Aluminum Bar

The frame will also have front and rear hinges to hold the radiator at an angle of 60 degrees. Figure 38 shows the front and rear hinges that will be welded into the frame. Figure 39 Shows the entire frame fully built.

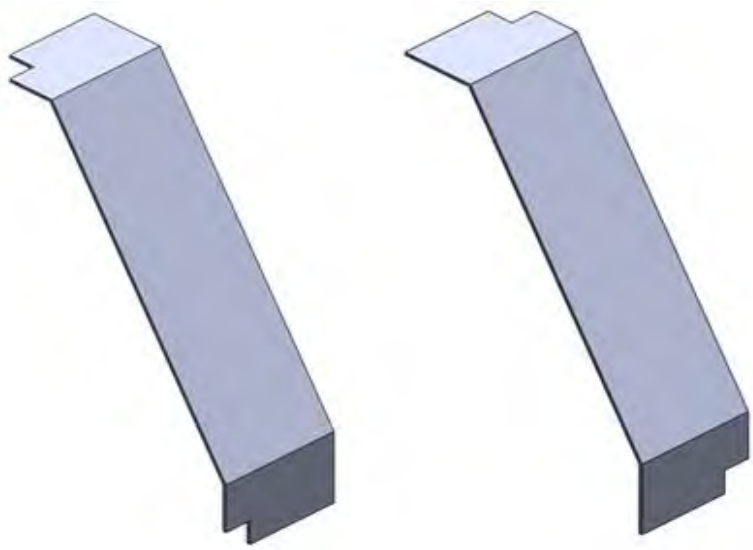

Figure 40 Front & Rear Hinges

Figure 41 Cooling Tower Frame - Final Design

16.3 Static Stress Analysis

The static analysis for the frame was performed using SolidWorks Simulations. The bottom four bars of the frame were fixed as shown in Figure 43. The bottom face of the top two parallel bars of the frame was given a load of 25 lbf that corresponds to the electric motor it will sustain. The material applied to the frame was Aluminum 1060 Alloy.

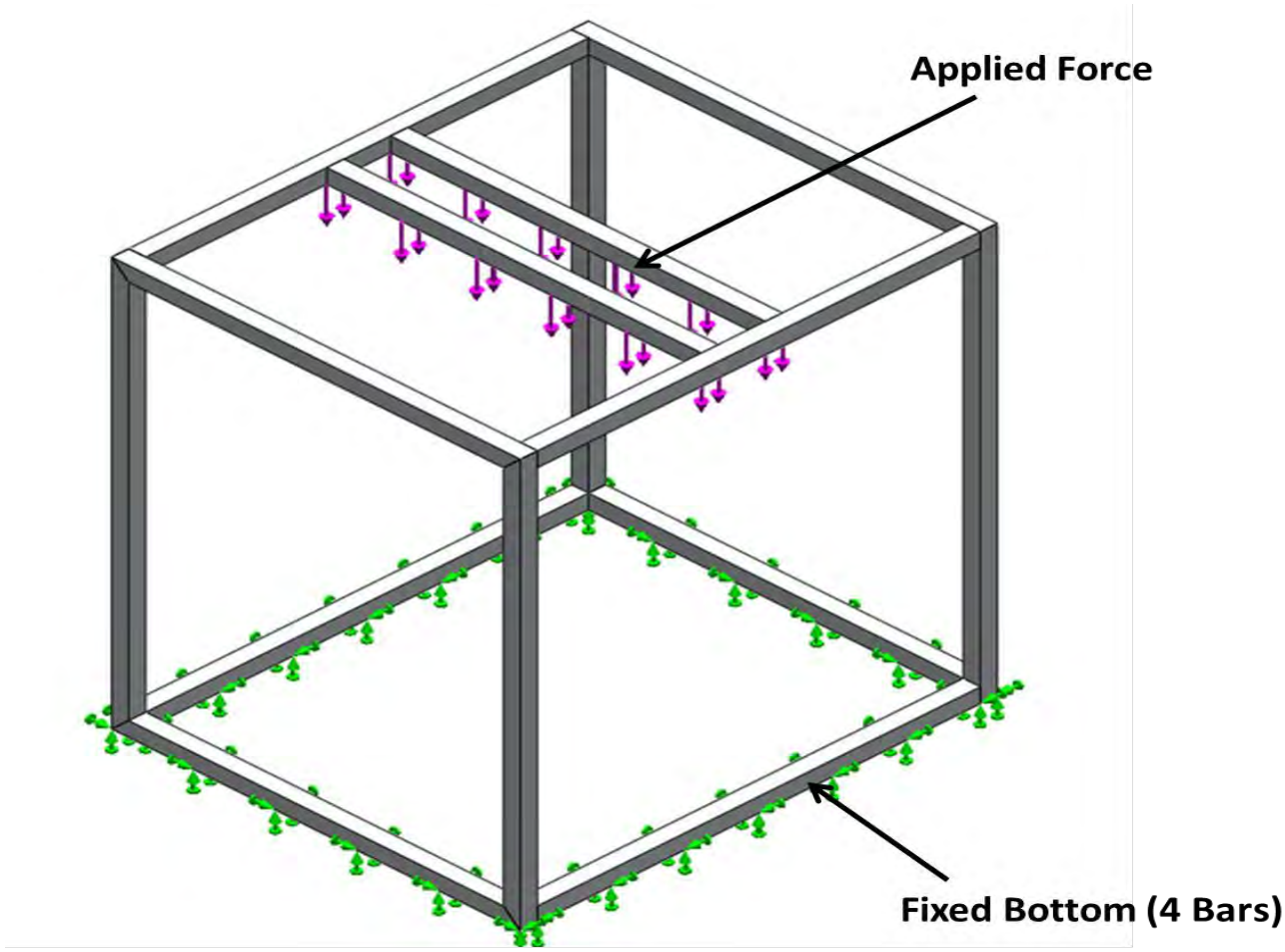

Figure 42 Forces & Constraints

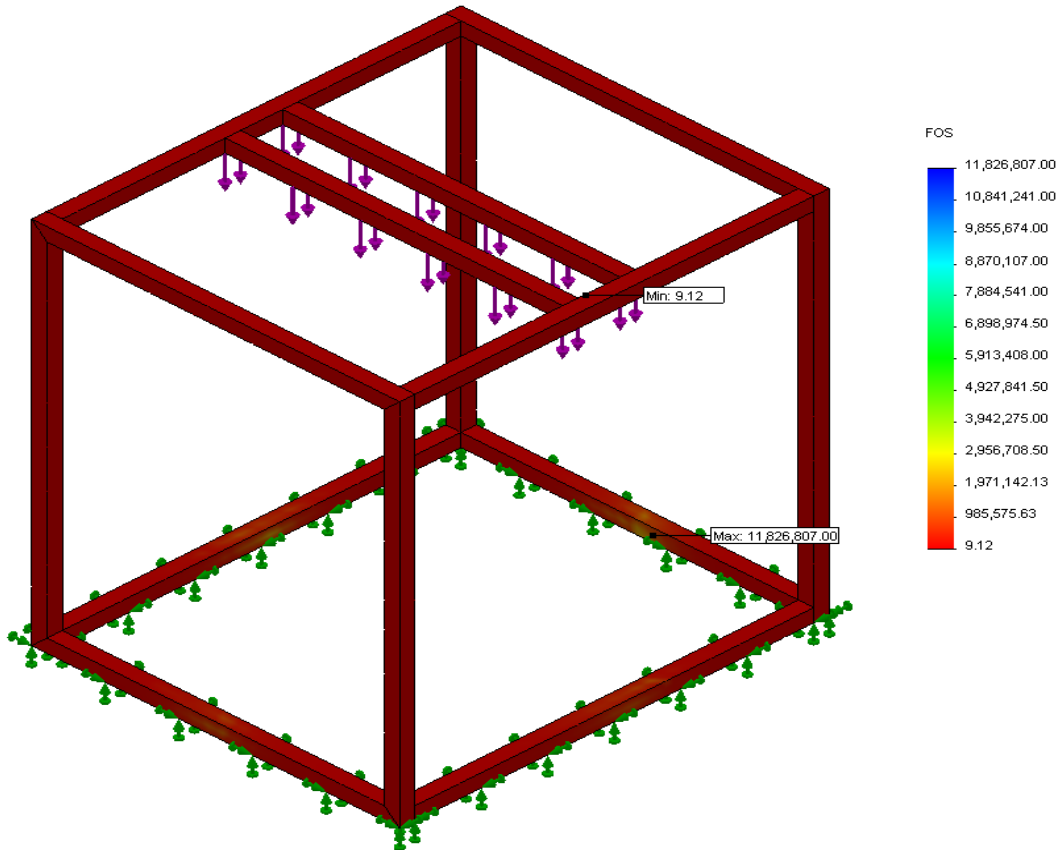

Figure 43 Factor of Safety Plot

The minimum factor of safety for the frame was 9.12. The frame is strong enough to hold the 25lbf motor and more. It has an acceptable factor of safety because the maximum the frame can hold is 228lbf. Although it is not the case, if it were to yield, the first sections to experience yielding would be at the corners of the parallel bars shown in Figure 45.

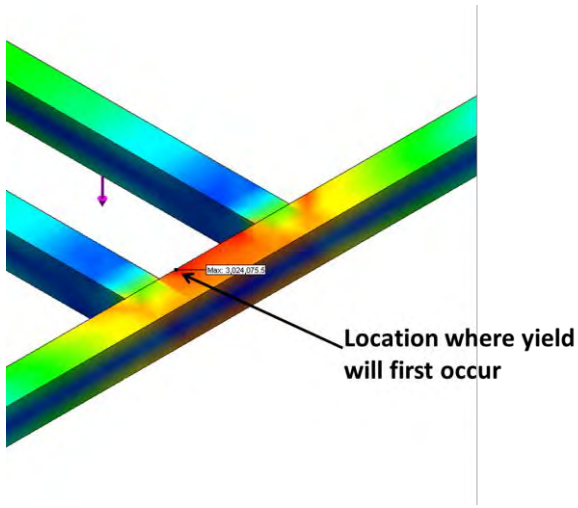

Figure 44 Location of yielding

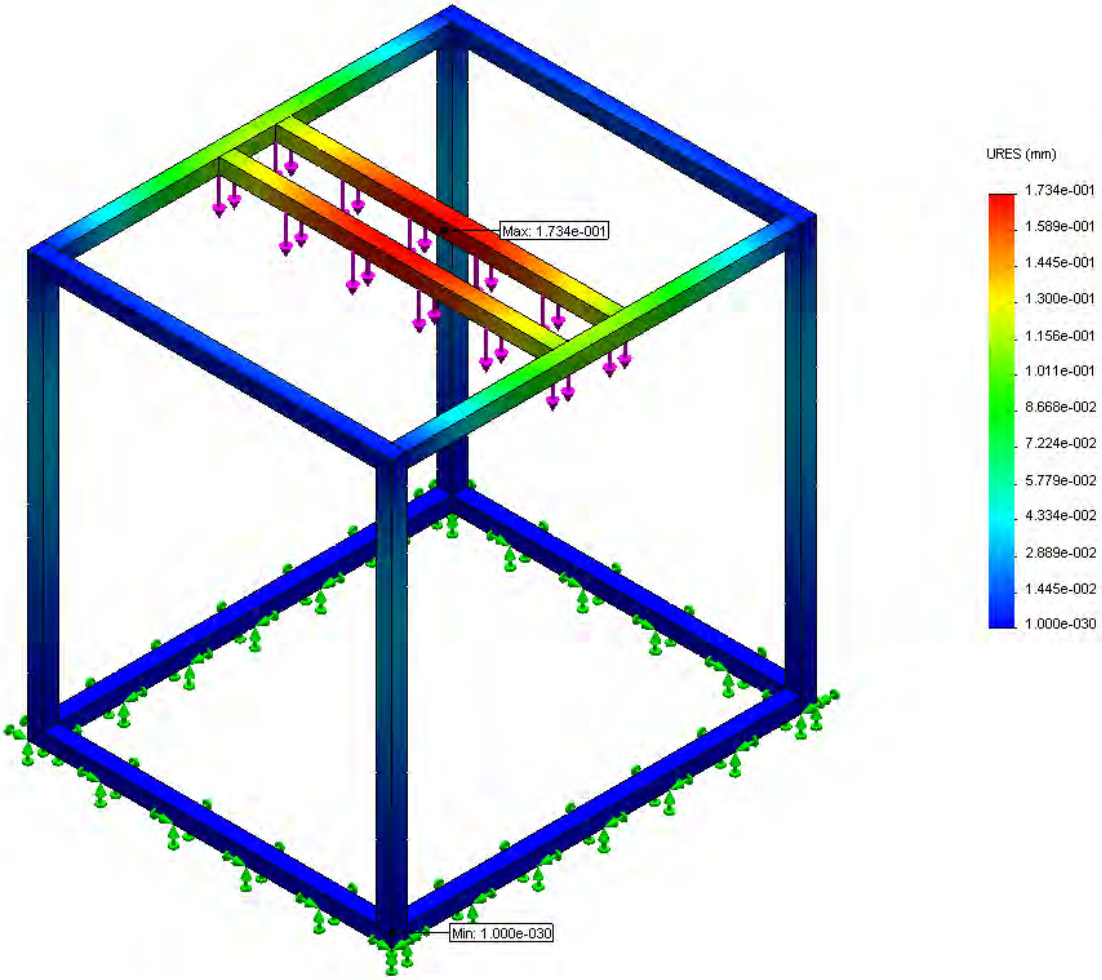

Figure 45 Displacement in mm

Figure 46 shows the maximum and minimum displacements that the frame will experience. The red segments show that top side of the frame will deform the most with a maximum of .1734 mm. The blue segments show the least deformation – 0mm. This study proves the initial assumption that the top parallel bars that hold the electric motor will deform the most.

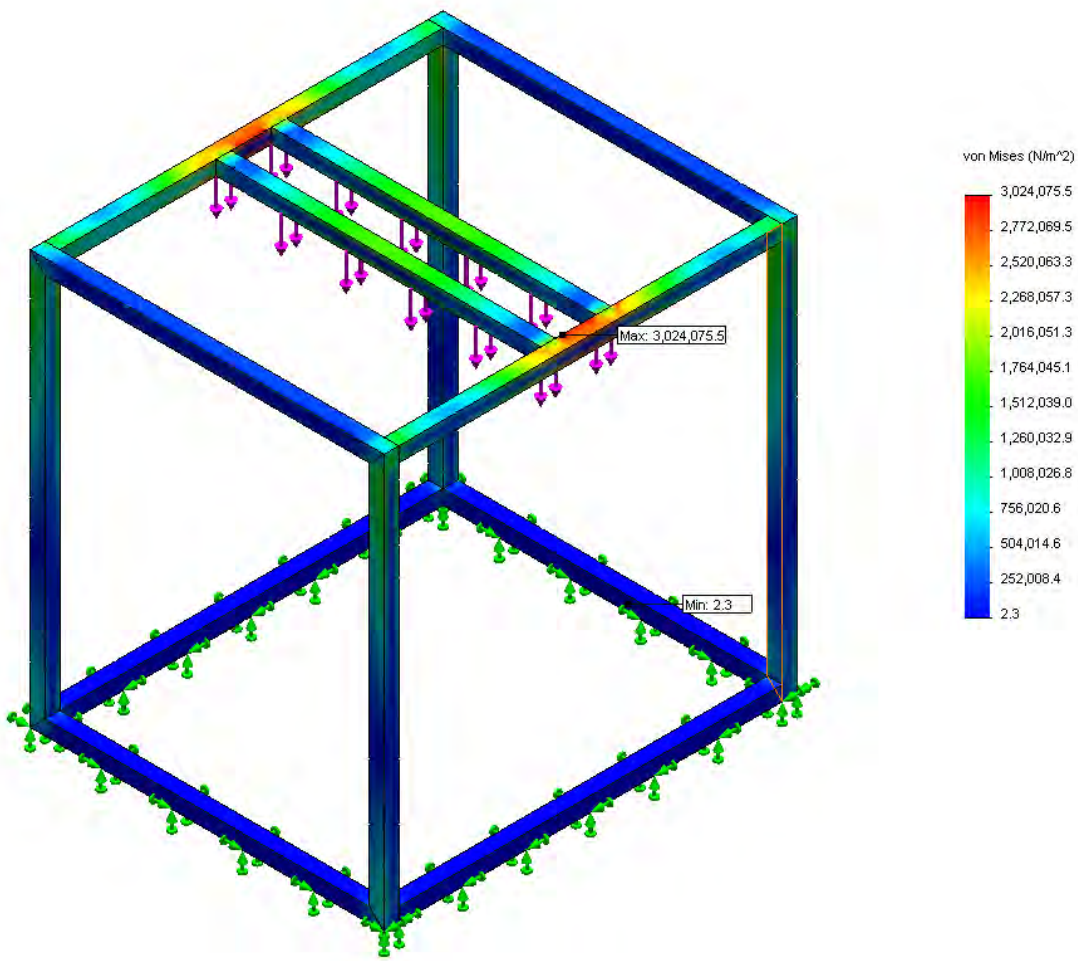

Figure 46 Von Mises Stress

Figure 46 shows the maximum and minimum von Mises stresses that the frame endures. The maximum stress will be 3.02 MPa and the minimum stress will be 2.3 Pa. It

is important to note that the von Mises not really a stress; instead it is criteria. The von Mises looks at the combination of all the 3 principle stresses at 1 point and compares it to the yield point. It is important to analyze the von Mises stress carefully because sometimes a principle stress is lower than the yield point yet the model still breaks. In this case, this will not happen because the FOS is still well into acceptable measures.

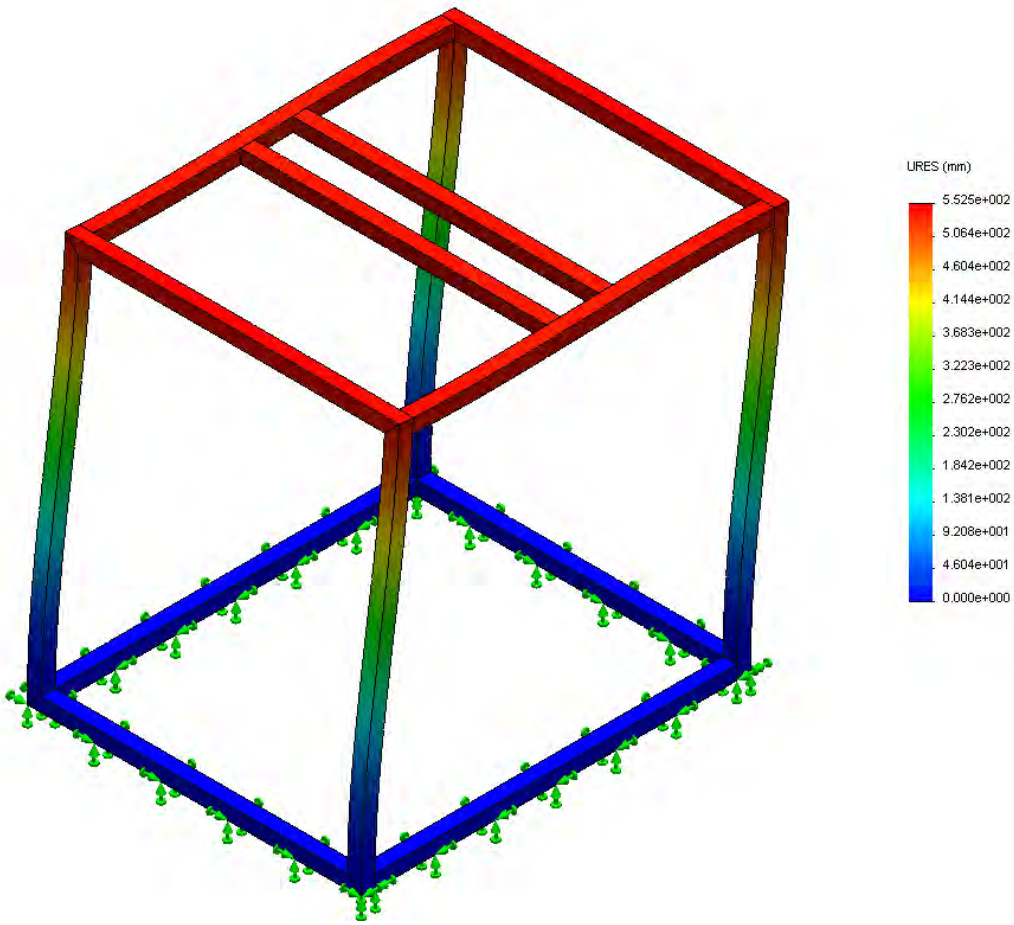

Figure 47 Frequency 1st Mode 0 Nodes

Table 42 Listed Resonance Frequencies

Study name: Frequency Analysis			
Mode No.	Frequency(Rad/sec)	Frequency(Hertz)	Period(Seconds)
1	223.27	35.535	0.028141
2	229.67	36.553	0.027358
3	327.17	52.071	0.019204
4	784.48	124.85	0.0080094
5	947.84	150.85	0.0066289

Every object has a natural frequency. Figure 47 and Table 42 shows the results of the frequency study. Only five modes were analyzed. Figure 28 shows mode number 1 in which the frame experiences a frequency of 223.27 rad/sec. The next four modes are listed as possible resonance frequencies. According to the static frequency results, the frame will completely deform when it experiences one of the five natural frequencies including 223 Hz, 229 Hz, 327 Hz, 784 Hz or 947 Hz. These natural frequencies might occur due to the vibration of the electric motor. Essentially, the frame will begin to vibrate and oscillate back and forth causing permanent damage.

17. Cost Analysis

17.1. Man-Hours

In this chart represent the man-hours with the cost and the percentage of each member of the group

Table 43 Cost Analyses

				Henry Gutierrez		Miguel Freire		Santiago Paz	
Task	Days	Man-Hours	Cost ($35/hr)	Percentage	Hours	Percentage	Hours	Percentage	Hours
Theoretical Research	55	60	$2,100	35%	20	25%	20	25%	20
Theoretical Analysis	35	50	$1,750	25%	16.666	20%	16.666	20%	16.666
Simulation	30	70	$2,450	25%	23.333	20%	23.333	25%	23.333
Prototyping	50	90	$3,150	25%	30	25%	30	25%	30
Testing	24	34	$1,190	25%	11.333	25%	11.333	25%	11.333
Total	194	304	$10,640		101.33		101.33		101.33

17.2. Total Cost Production

For the prototype, the following components will be implemented as part of the design and construction of the air-cooling tower. The cooling tower will be constructed with compactness in mind so that it sits aesthetically next to the FIU Solar House as well as any residential home.

For the material purchases, several months were spent researching the most cost effective materials in the market. Several calls were made to contact many suppliers such

as: OMEGA, Saez Distributor, Breeza Fans and Active Radiators to achieve a reasonable and affordable budget, at the same time finding the best products out there in the market.

Table 44 First Cost Analysis

First Cost Analysis				
	Part #	Units	Price	Amount
Thermocouple	TC-JNPT-G-72	1	38	38
Controller	CN7500	1	100	100
2 Way Valve	SV6006	1	440	440
Electric Motor	51-25023-01	1	90	90
Blade	60559701	1	40	40
Radiator	Online quotes	1	1650	1650
		Tax	7%	165.06
		Total		2523.06

Table 44 represents the first quotes that were found including a louvered fin radiator 24” x 24” from Active Radiator but it was highly priced at $1650. However, the price was considered extremely high due to the fact that it was a custom made piece. The shut-off valve was about $440 and also was over budget. As a result, a different approach was taken to reduce the amount of money spent on the materials. The search continued and expanded in the market to find affordable parts that were mass produced or factory made as opposed to custom in order to accommodate the budget and perform the required expectations.

Table 45 Final Cost Analysis

Cost Analysis Overall				
Description	Part #	Units	Price ($)	Amount
Motor	5470	1	84.41	84.41
Capacitor	CAP20X370	1	4.67	4.67
Fan Hub 1/2"	60-7658-04	1	6.13	6.13
Fan Blade 20"-CW-27-3	60-5567-01	2	30.39	60.78
Radiator (Sunbelt Rad.)	SBR129026MM	1	45	45
Radiator (EBay)	SBR129026MM	1	30	30
Alum Tube 1x1x1/8" 24'	TA1X1X1/8	2	24.05	48.1
Flat Plate 7 feet	----------	1	47.88	47.88
Digital Thermometer	67012055	2	14	28
PVC 10ft	6.11942E+11	1	3.67	3.67
PVC Bushing 1-1/4"x1"	12871626579	4	1.26	5.04
Elbows 90°	12871623356	8	0.66	5.28
1-1/4 Tee-PVC	12871620683	2	1.56	3.12
1"x3/4" PVC female adapter	12871625954	4	0.97	3.88
SS Clamp 3/4"	78575172057	4	1.15	4.6
Swivel PVC Fitting	46878533639	1	1.67	1.67
3/4 Nylon Barb	48643071766	1	3.24	3.24
Analog Thermometer	39953526862	1	9.97	9.97
Omega Thermostat	CN7500	1	101	101
Shut-Off Valve	B2229+LF120	1	161	161
Weather Shield Valve	ZS-CCV	1	62	62
Cap Radiator	17110029	2	9.49	18.98
Thermocouple	TC-K-NPT-G-72	1	38	38
Shipping 2 days	----------	----	22	22
Paint	----------	----	40	40
Flat plate 36"x1-1/2"x1/8"	30699448807	1	8.47	8.47
Mach screw #8-32	30699276318	3	1.18	3.54
Liquidate AC whip	32886895708	1	10.12	10.12
Alum sheet 3x3'	43374572087	2	32.98	65.96
Molding tape 5ft 3M	6012221	2	6.99	13.98
Molding tape 15ft 3M	7100055	3	12.99	38.97
		Tax	7%	60.3036
		Total		1009.7636

Table 45 represents all the materials that were acquired to construct and assemble the project. The aforementioned table has all the descriptions and the part numbers. Since the beginning of the semester, different quality products and alternatives were researched to build the most economical and optimal cooling tower. In addition, Table 45 shows other materials that were supplied free of charge which dramatically reduced the overall cost to remain under budget. When comparing Table 44 to Table 45, it is noticeable how much money was saved from the first quote of $2523.06 to a remarkable $805.79. Overall, more than $1700 dollars were saved with all the material chosen in order start constructing the FIU air-cooling tower

18. Conclusion

18.1 Problem Statement

The FIU Solar House is connected to a geothermal heat pump with horizontal loops laid 6 feet underground with a constant temperature of 77°F. This type of pump uses thermal energy from the ground to provide cooling during hot weather and heating during cold weather. One problem with this geothermal pump is that during extremely hot and humid days, Miami temperatures exceed 90°F and thus prevent the ground coils from sufficiently removing all of the excess heat. In order to solve this issue, a mini air cooling tower was designed to work in conjunction with the geothermal heat pump.

18.2 Experiments & Findings

For the first experiment, only one radiator and the electric motor with the propeller fan was used. There was no frame or enclosure. The radiator (SBR129026MM) has louvered fins with a cross sectional face area of 13” x 13”. Two parameters were kept constant to maintain steady state the water flow rate of 4 gallons per minute and the air flow rate of 4000 ft^3/min. The ambient temperature was roughly 82°F and the temperature of water flowing at the inlet of the radiator was 114°F. When the fan was turned on, the temperature of the water flowing through the exit of the radiator was reduced to 111°F. The system removed a total of 3°F which is 11955.65413 BTU/hr. Theoretically, if one radiator can remove 3 degrees with 4 gallons per minute passing through (keeping the same amount of air flow), then two radiator can remove 3 degrees with 4 gal/min of water passing through each one – for a total of 8 gal/min. For the second experiment, two of the same radiators were used that were placed inside the custom built enclosure. 8 gallons of water per minute was used and the cooling tower reduced the temperature by a total of 3°F which validated the initial assumption.

Another finding that is important to note is that two factory produced radiators were used instead of one custom made. One custom radiator constructed by Active Radiators was going to cost $1650. This radiator has a face area of 25” x 25” and designed to receive an air flow rate of 4000 ft^3/min and reject 17,750 BTU/hr of heat. In order to dissipate this amount of heat, the large radiator requires 8 gallons per minute. Therefore, in order to reduce cost and save space when designing the frame, it was decided that two factory produced radiators were going to be bought and that the flow rate was going to be divided by half for each one. The smaller radiators were bought from

Sunbelt Radiators (local area) and priced at $50 each for a total $100. The savings in the price was fantastic and both radiators were just as effective in that they were louvered finned and constructed with aluminum.

18.3 Approaches for Implementing Cooling Tower

From the results gained experimentally, there are two approaches that have been decided to implement the cooling tower and optimize the annual cooling load in the FIU Solar House. The first approach is incorporating the mini air cooling tower with the current geothermal heat pump situated inside the solar house and keeping all of the underground loops (a total of 2400 feet). The shut-off valve that redirects the water flow towards the cooling tower and the electric motor power are controlled by a thermostat controller. This controller will be set to 92°F and the cooling tower will commence operation when a k-type thermocouple senses that the temperature of the water leaving the heat pump is higher than 92°F. At this point, the shut off valve will close and force the water to take the path of least resistance towards the heat exchanger and the fan will turn on. The advantage to this approach is that right now the cooling tower can be integrated to the heat pump and the hybrid system will work. Furthermore, the hybrid heat exchanger will balance and optimize the heating and cooling load. The disadvantages, however, is that the cooling tower will most likely operate only 3 months of the year being the hottest months: June, July and August. Another disadvantage is that because the system will be in full use only ¼ of the year, the motor that powers that fan is more prone to mechanical failure. When the moving components of the electrical motor

sit around for too long without moving, lubrication is lost, moving components get stuck and become more resistant to moving.

The second approach is to remove a substantial amount of ground coil and replace it with the cooling tower to reject the same amount of heat. Since the cooling tower removes 3°F, it can be used to reduce the temperature from 94°F to 91°F. Through heat transfer calculations, it was determined that with water entering the ground loops at 91°F and returning at 85°F, only 1532 feet of ground coil is necessary. 2400 feet of coil is currently being used therefore a total of 868 feet can be removed to reject 11955 BTU/hr. The hybrid cooling tower will decrease the temperature of the water from 94°F to 91°F and the underground loops will continue to decrease it to 85°F. For this system to work, the controller will be set to 91°F to turn on the fan and signal the shut off valve to redirect the water.

There are several advantages to this approach. The first one is that the air cooling tower will optimize and balance the annual cooling load by working mostly all year around. Furthermore, the hybrid system will save money because fewer underground coils are required to reduce the temperature of a fluid. Finally, during the hottest months of the year, the geothermal heat pump will work the hardest by rejecting more heat constantly throughout the day with almost no rest to keep the occupants comfortable inside the house. The hybrid component will ensure that the heat pump will work with less stress by guaranteeing that the temperature of the water entering the loops will always be at 91°F. This means that the cooling tower will be designed to dissipate the excess heat and the ground loops will always receive water at the same temperature. Without the hybrid system, the water leaving the heat pump will be much higher than

94°F and the coils would not dissipate all of the heat, and the water returning to the heat pump would be higher than 85°F which is detrimental to the efficiency of the pump. The disadvantage to this approach is that it would be financially troublesome to spend more money to actually excavate the ground to remove ground coil.

18.4 Overall Advantages of Hybrid System

There are many advantages to the hybrid geothermal heating system including environmental friendliness, low cost and suitability for hotter climates. The dry cooling tower is more energy efficient than a water cooling tower because there is no water involved in cooling the warm fluid. This saves unnecessary water consumption and money because bringing water to a site is expensive. The water requires treatment and extra pumps and water lines are required to provide a water source for the cooling towers. This alone reduces the overall cost and will please more buyers who are environmentally oriented but are not necessarily willing to spend tons of money on expensive systems. Additionally, the only moving parts of the dry system are the fan and the motor which require little or no maintenance.

Furthermore, the installation and operating cost of a hybrid system will be less expensive than an entire geothermal heating system. The hybrid component – the water to air heat exchanger – will reduce the total cost by replacing a considerable amount of ground coil and reject the same of heat. This is important because the more cooling a fluid needs, the more expensive the geothermal system because more coils are required to complete the job.

Finally, the hybrid systems are ideal for areas that are very humid and have a hot climate such as Texas and South Florida. Hybrid system can also save plenty of space in residential locations. Generally speaking, homes are much smaller than industrial sites. Therefore, hybrid systems can save money and space by replacing horizontally laid loops that take up more ground area. It can also save more money by avoiding the need to dig deep wells for vertical loops which are more expensive projects than laying loops horizontally. For this reason, the hybrid systems can grow to become a potential application for current and future residences by placing them in areas with scarce ground space. The cooling tower is also very suitable for homes that are not nearby lakes or water reservoirs. Finally, the installation of a cooling tower to a geothermal heat pump is fairly easy.

18.5 Prototype versus Current Designs in the Market

The current dry cooling towers in the market are similar to the prototype that was designed but there are many significant differences that make the prototype unique. The current products in the market are mostly utilized for industrial purposes but the prototype was designed with residential purposes in mind. This is to provide a potential application to current and future homes that are going to use geothermal energy as a source for heating and cooling.

Current dry cooling towers use copper tubes with aluminum fins and the return bends for the coil are formed with thick walled tubes that are heavier than the standard tubing for the rest of the coils. Although heavier tubing does provides toughness and durability, it makes the tower much heavier and more expensive. The prototype on the other hand use aluminum louvered fin radiators (unmixed/unmixed) with micro channel

tubing that increase heat transfer efficiency and weighs a lot less. The tubes at the inlet and outlet of the cooling tower will be made of PVC as opposed to the heavy inlet and outlet headers made of wall steel pipes.

Current cooling towers use non-corrosive fiber glass and GI panels for the enclosure. The prototype designed uses aluminum sheet metals for the panels of the enclosure. The fiber glass is high strength, corrosion resistant and lightweight but still heavier than aluminum. The aluminum sheet metals on the other hand are also light weight and of excellent strength but more prone to corrosion with salt in the air. This is a problem for hybrid systems installed near the ocean.

Current cooling towers are also mounted on heavy duty base frames and the prototype was designed with an aluminum frame that that is heavy enough for its purpose but not as heavy as cooling towers in the market therefore much more light weight.

18.6 Recommendations

There are several physical changes that can be made to the current prototype to improve the design. For one, instead of using an axial propeller fan, a vane axial fan can be used because the flow is very efficient and the noise band is much lower than the propeller. The enclosure can be made aesthetically more pleasing by rerouting the tubes that exit the radiator through the inside of the frame as opposed to the outside. The frame can also be made lighter by constructing it with aluminum bars of ½” x ½” cross sections instead of 1” x1”. The thinner bars will also reduce the cost by decreasing the amount of material being used. Finally, instead of using metallic sheet metals, non-corrosive fiber glass can be used to reduce even more weight.

The motor inside the cooling tower is currently powered by the standard 120 volts AC outlet plug. In the future, the motor can be powered by using solar energy acquired by the solar panels located in front of the FIU Solar House.

The construction of the cooling tower began in late January of 2013 and it is currently due halfway through April. This means that the cooling tower was never tested during the hottest months of year being June, July and August. In order to obtain the most accurate temperature readings, the heat exchanger should be experimented during one of the hottest days of the year where the ambient temperature will be higher than 90°F. This will ensure that the cooling tower will still work efficiently during a hot day. Finally, the total heat transfer can be analyzed when angling the radiator less than 60° to optimize the amount of heat that the fan will suck from the radiators. Finally, the cooling tower can be studied using one radiator to verify its effectiveness in removing total heat as opposed to using two smaller radiators.

Global Learning

International Applications

Geothermal heat pump can be found in international countries, mostly found in the Pacific Rim. On figure 48 shows the hottest Geothermal Regions of the word. The first 1st Geothermal Power Plant was built in Lardarello, Italy (1904), also is helping the environment.

The energy of Europe is mostly provided by Italy (5200 GWh/yr), followed by Iceland (1500 GWh/yr) that supplies 18 % of his total energy.

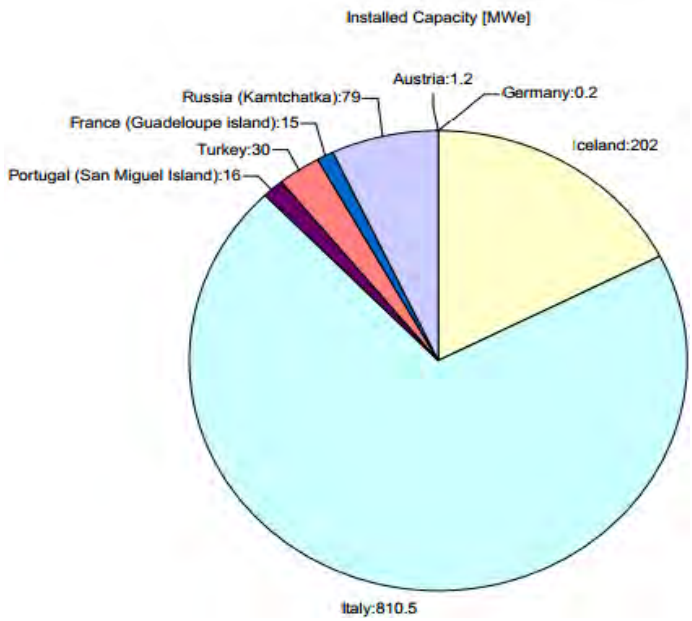

Figure 48 Geothermal Heat Pumps in Europe

These are the 20 leading Nationalities that use Geothermal:

- Australia
- Chile
- China
- Costa Rica
- El Salvador
- Ethiopia
- Germany
- Guatemala
- Iceland
- Indonesia
- Italy
- Japan
- Kenya
- Mexico
- New Zealand
- Nicaragua
- Philippines
- Russia
- Turkey
- United States

Warning Labels

Safety warning labels are going to placed in the machine, are going to be in different languages and colors that can be very noticeable.

Figure 49Warning Label 1

The warning label on figure 49 is explaining in both English and Spanish that whoever is near the cooling tower should be careful with the spinning blades. They are made out of aluminum and at very fast speeds they can permanently remove someone's hands. The sign is also in very bright yellow to alert anybody looking at the tower. This sign can be further expanded by writing it in other languages such as German, Japanese and Russian as shown in the list of leading 20 nationalities that will possibly incorporate hybrid components with their geothermal systems.

Figure 50 Warning Label 2

The warning label on Figure 50 is to show that the cooling tower is operated automaotically.

Figure 51 Warning Label 3

The Figure 51 above is to prevent any injuries or fatalities because this machine start and stop automatically.

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Aquarius Geothermal Heat Pump Specification. Own Picture, taken with cellphone camera. 2012 October 11.

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APPENDICES

Appendix A: Solar House Plans & Components

A1 Geothermal Piping Layout

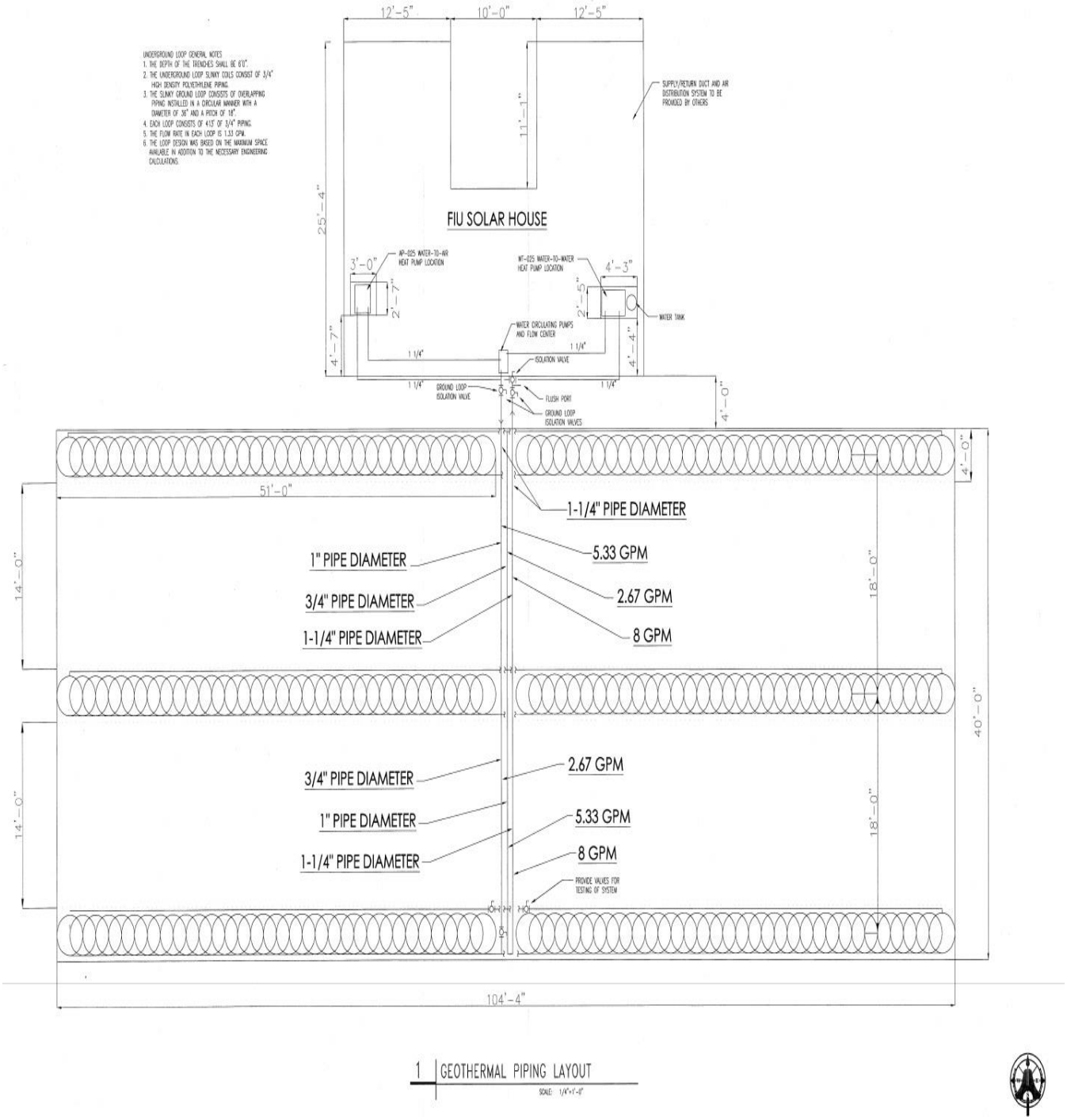
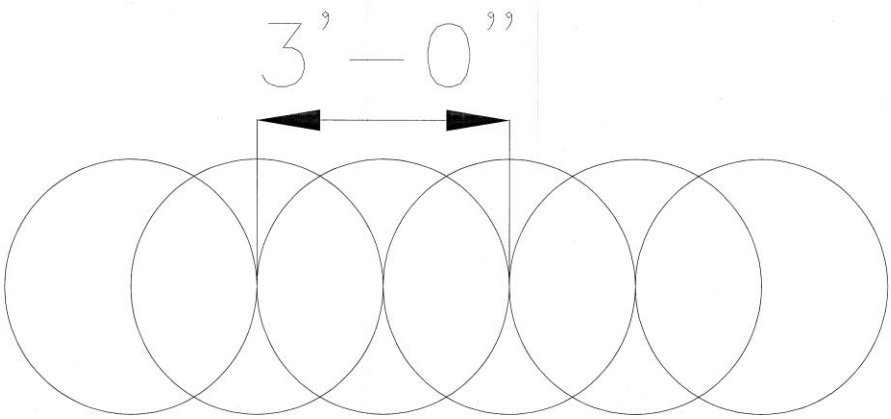


Figure 52 Geothermal Piping Layout

A2 Ground Loop Cross Section

UNDERGROUND LOOP GENERAL NOTES

1. THE DEPTH OF THE TRENCHES SHALL BE 6'0".
2. THE UNDERGROUND LOOP SLINKY COILS CONSISTS OF 3/4" HIGH DENSITY HIGH DENSITY POLYETHYLENE PIPING.
3. THE SLINKY GROUND LOOP CONSISTS OF OVERLAPPING PIPING INSTALLED IN A CIRCULAR MANNER WITH A DIAMETER OF 36" AND A SPACING OF 18".



1 GROUND LOOP CROSS SECTION
SCALE: 1 1/2"=1'-0"

Figure 53 Ground Loop Cross Section

A3 Trench Isometric View

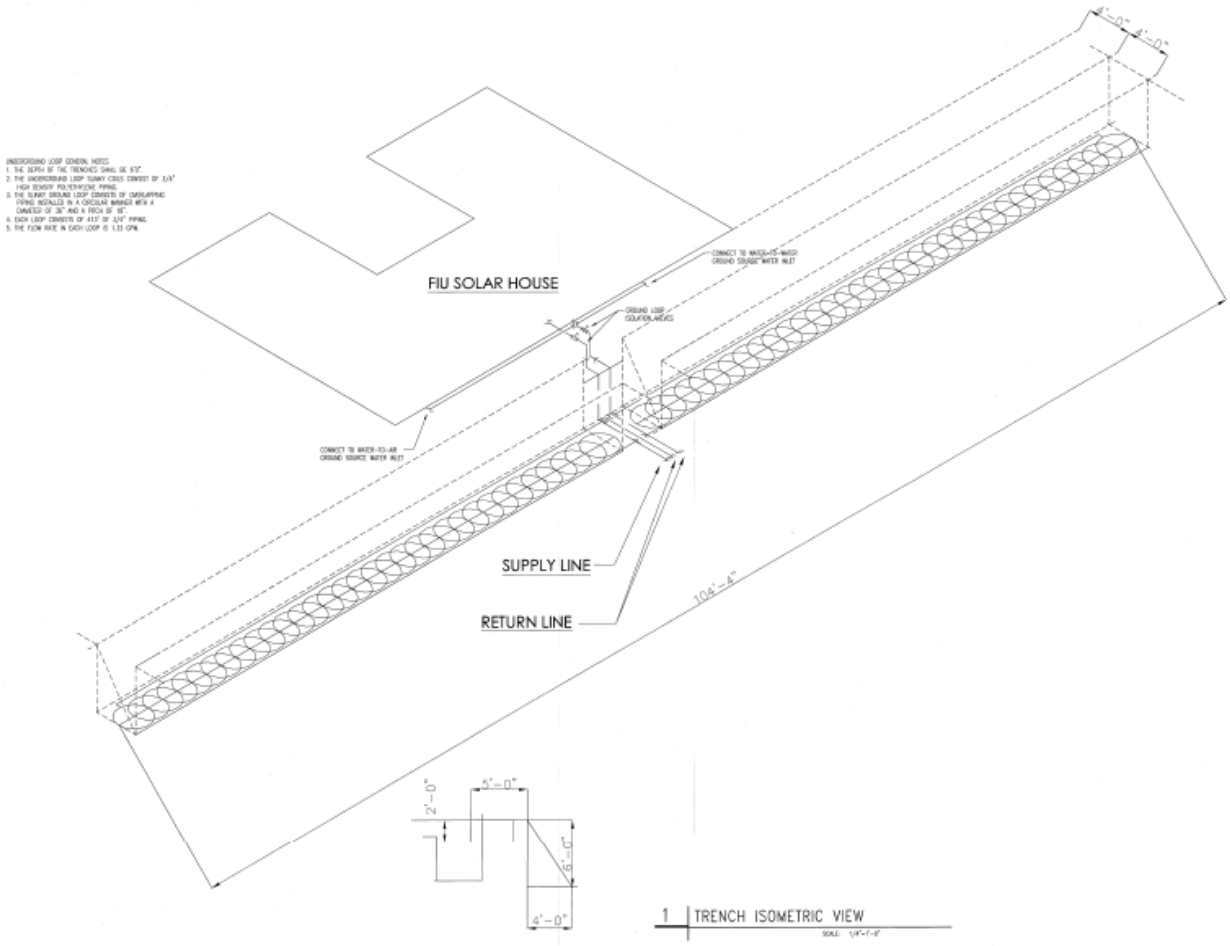


Figure 54 Solar House Trench Isometric View

A4 FHP Heat Pump Equipment Schedule

EQUIPMENT SCHEDULE			
WATER-TO-AIR HEAT PUMP		WATER-TO-WATER HEAT PUMP	
MANUFACTURER	FLORIDA HEAT PUMP	MANUFACTURER	FLORIDA HEAT PUMP
MODEL NUMBER	AP025	MODEL NUMBER	WT025
DIMENSION (WIDTH, DEPTH, HEIGHT) (IN)	21.50X26.00X47.25	DIMENSION (WIDTH, DEPTH, HEIGHT) (IN)	24.00X32.50X24.00
TOTAL AIR FLOW PERFORMANCE DATA RATING (CFM)	950	LOAD FLOW RATE PERFORMANCE DATA RATING (GPM)	5.0
FLOW RATE PERFORMANCE DATA RATING (GPM)	8.0	SOURCE FLOW RATE PERFORMANCE DATA RATING (GPM)	6.2
BLOWER HP	1/3		
ELECTRICAL (V,ø,Hz)	230/1/60	ELECTRICAL (V,ø,Hz)	230/1/60
MIN. CIRCUIT AMPACITY 208 V	18.8	MIN. CIRCUIT AMPACITY 208 V	14.3
MAX. FUSE/BREAKER	30.0	MAX. FUSE SIZE	25.0
FLUID PRESSURE DROP AT 7 GPM, 9 GPM (FOH)	3.6, 5.7		
CAPACITY DATA: ENTERING AIR AT 85 DEG F AND ENTERING WATER AT 85 DEG F		CAPACITY DATA: LEAVING LOAD FLUID AT 100 DEG F AND ENTERING SOURCE FLUID AT 70 DEG F	
COOLING TOTAL CAPACITY (MBTUH)	29.36	HEATING TOTAL CAPACITY (MBTUH)	33.08
POWER INPUT (KW)	1.76	POWER INPUT (KW)	1.65
REJECTED HEAT (MBTUH)	35.35	HEAT OF ABSORPTION (MBTUH)	27.44
EER	16.7	COP	5.90
CAPACITY DATA: ENTERING AIR AT 85 DEG F AND ENTERING WATER AT 100 DEG F		CAPACITY DATA: LEAVING LOAD FLUID AT 100 DEG F AND ENTERING SOURCE FLUID AT 60 DEG F	
COOLING TOTAL CAPACITY (MBTUH)	26.94	HEATING TOTAL CAPACITY (MBTUH)	33.08
POWER INPUT (KW)	1.99	POWER INPUT (KW)	1.65
REJECTED HEAT (MBTUH)	33.73	HEAT OF ABSORPTION (MBTUH)	27.44
EER	13.5	COP	5.90

Figure 55 FHP Heat Pump Equipment Schedule

A5 Water Heat Pump Schedule & Polyethylene Piping

WATER CIRCULATING PUMP SCHEDULE

MANUFACTURER	MODEL NO.	MAXIMUM FLOW RATE (GPM)	NO. OF PUMPS	VOLTAGE	AMPS	HP	INLET/ OUTLET SIZE (IN)
B&D MFG	1-1E-A-KFC-SS	18.0	2	230	2.14	1/6	1 1/4" / 1 1/4"

2 | WATER CIRCULATING PUMP SCHEDULE

SCALE: N.T.S

HIGH DENSITY POLYEHTYLENE PIPING

MANUFACTURER	MODEL NO.	NOMINAL DIAMETER (IN.)	INNER DIAMETER (FT.)	OUTER DIAMETER (FT.)	PRESSURE RATING (PSI)
ISCO PIPE	P11C35 5300 SERIES	3/4	0.860	1.050	SDR 11 160 PSI
ISCO PIPE	P112S 5300 SERIES	1	1.315	1.075	SDR 11 160 PSI
ISCO PIPE	P113C15 5300 SERIES	1 1/4	1.660	1.358	SDR 11 160 PSI

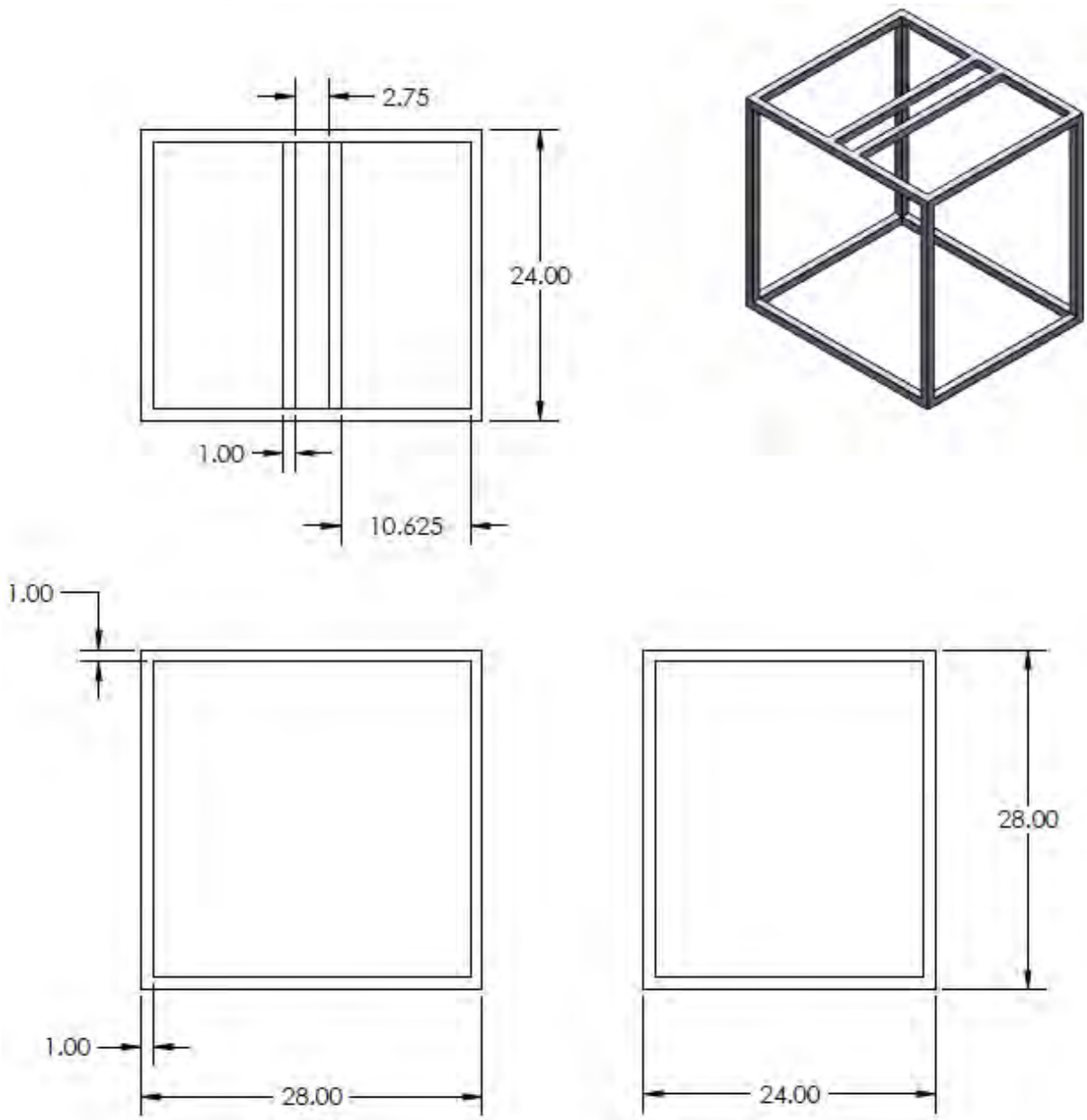
3 | HIGH DENSITY POLYETHYLENE PIPING

SCALE: N.T.S

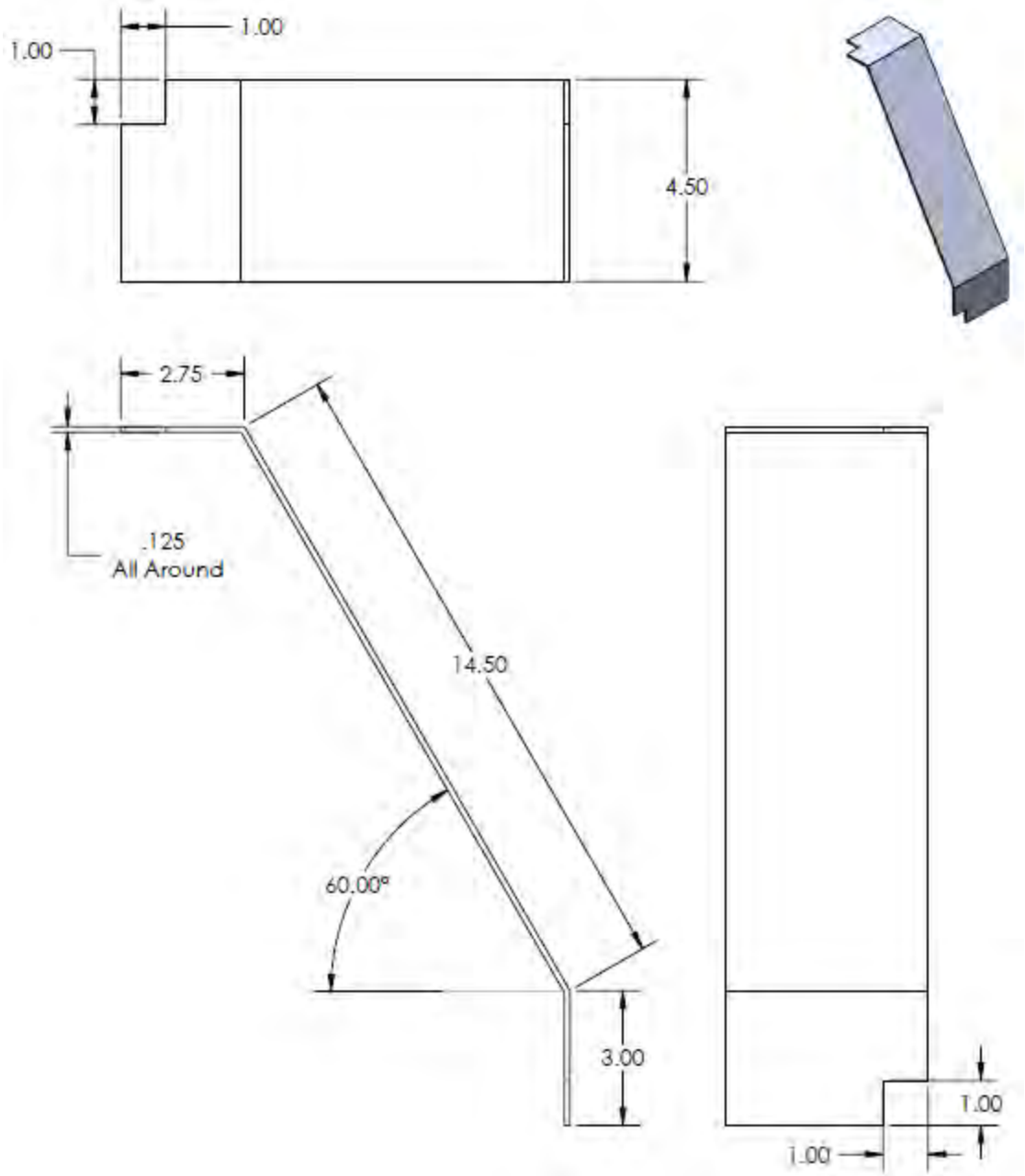
Figure 56 Pump Schedule & Polyethylene Piping

Appendix B: Drawing & Dimensions

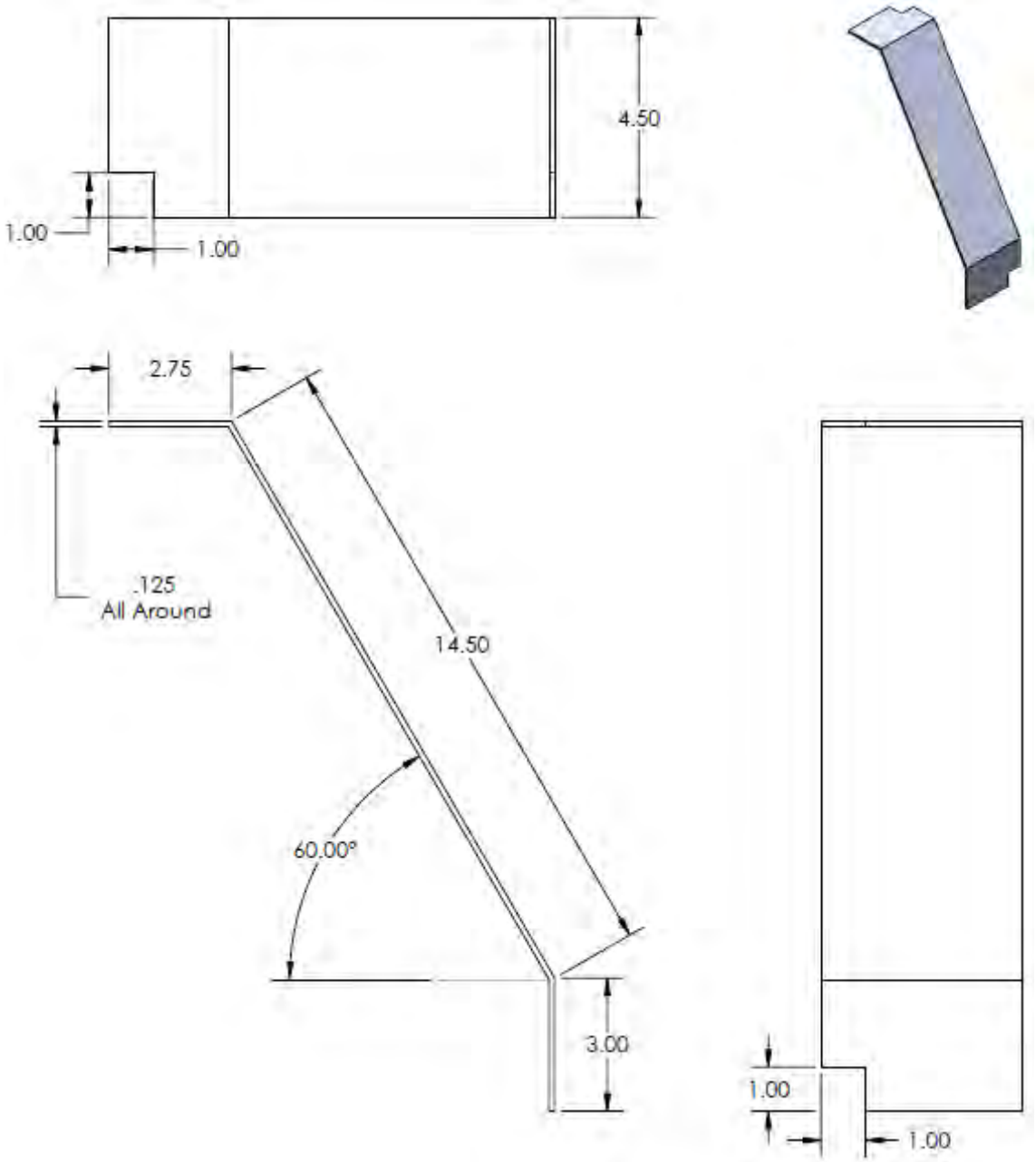
B1 Frame Enclosure Drawing



B2 Front Hinge Drawing



B3 Rear Hinge Drawing



Appendix C: Data & Component Specifications

C1 Average Monthly Climate and Weather Indicators in Miami Florida

Miami

The tables below display average monthly climate and weather indicators in Miami Florida.

Temperature by: Fahrenheit / Centigrade

Miami Temperature	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Annual
Avg. Temperature	67.2	68.5	71.7	75.2	78.7	81.4	82.6	82.8	81.9	78.3	73.6	69.1	75.9
Avg. Max Temperature	75.2	76.5	79.1	82.4	85.3	87.6	89.0	89.0	87.8	84.5	80.4	76.7	82.8
Avg. Min Temperature	59.2	60.4	64.2	67.8	72.1	75.1	76.2	76.7	75.9	72.1	66.7	61.5	69.0
Days with Max Temp of 90 F or Higher	0.0	0.0	< 0.5	1.0	4.0	10.0	16.0	16.0	10.0	2.0	0.0	0.0	61.0
Days with Min Temp Below Freezing	< 0.5	0.0	< 0.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	< 0.5	0.0

Miami Heating and Cooling	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Annual
Heating Degree Days	88.0	51.0	14.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	6.0	41.0	200
Cooling Degree Days	156	149	221	306	425	492	546	552	507	412	264	168	4198

Miami Precipitation	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Annual
Precipitation (inches)	2.0	2.1	2.4	2.9	6.2	9.3	5.7	7.6	7.6	5.6	2.7	1.8	55.9
Days with Precipitation 0.01 inch or More	7.0	6.0	6.0	6.0	10.0	15.0	16.0	17.0	17.0	14.0	8.0	7.0	131
Monthly Snowfall (inches)	0.0	0.0	0.0	0.0	< 0.05	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Other Miami Weather Indicators	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Annual
Average Wind Speed	9.5	10.1	10.5	10.5	9.5	8.3	7.9	7.9	8.2	9.2	9.7	9.2	9.2
Clear Days	9.0	8.0	9.0	8.0	6.0	3.0	3.0	2.0	2.0	7.0	8.0	9.0	74.0
Partly Cloudy Days	13.0	12.0	14.0	15.0	15.0	14.0	17.0	18.0	15.0	14.0	14.0	13.0	175
Cloudy Days	9.0	8.0	8.0	7.0	9.0	13.0	11.0	11.0	12.0	10.0	9.0	9.0	115
Percent of Possible Sunshine	66.0	68.0	74.0	76.0	72.0	68.0	72.0	71.0	70.0	70.0	67.0	63.0	70.0
Avg. Relative Humidity	59.0	71.0	69.5	67.5	67.0	71.0	74.0	74.0	76.0	76.0	74.0	73.0	71.5

Source from: Climate Zone (2012) Retrieved from:

http://www.climate-zone.com/climate/united-states/florida/miami/

C2 Belimo Shut-Off Valve Specifications

BELIMO

B2 Series, 2-Way, Characterized Control Valve Stainless Steel Ball and Stem

Application

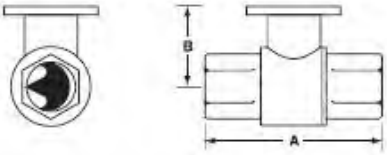
This valve is typically used in air handling units on heating or cooling coils, and fan coil unit heating or cooling coils. Some other common applications include Unit Ventilators, VAV box re-heat coils and bypass loops. This valve is suitable for use in a hydronic system with variable flow.

Technical Data

Service	chilled or hot water, 60% glycol
Flow characteristic	A-port equal percentage
Controllable Flow Range	75°
Sizes	½", ¾", 1", 1¼", 1½", 2", 2½", 3"
Type of end fitting	NPT female ends
Materials:	
Body	forged brass, nickel plated
Ball	stainless steel
Stem	stainless steel
Seats	PTFE
Characterizing disc	Tefzel®
Packing	2 EPDM O-rings, lubricated
Body pressure rating	
600 psi	½" - 1¼" (B230)
400 psi	1¼" (B231) - 3"
Media temp. range	0°F to 250°F [-18°C to 120°C]
Close off pressure	
200 psi	½" - 2" (B250)
100 psi	2" (B251) - 3"
Maximum differential pressure (ΔP)	50 psi for typical applications
Leakage	0% for A to AB
External leakage	according to EN 12266-1:2003
C_V rating	A-port: see product chart for values

Tefzel® is a registered trademark of DuPont

Dimensions



Valve Nominal Size			Type	Suitable Actuators				
C_V	Inches	DN [mm]	2-Way NPT	Non-Spring			Spring	
0.3	½	15	B207					
0.46	½	15	B208					
0.8	½	15	B209					
1.2	½	15	B210					
1.9	½	15	B211					
3	½	15	B212					
4.7	½	15	B213					
7.4	½	15	B214					
10	½	15	B215					
14	½	15	B216					
4.7	¾	20	B217					
7.4	¾	20	B218					
10	¾	20	B219					
14	¾	20	B220					
24	¾	20	B221*					
7.4	1	25	B222					
10	1	25	B223					
19	1	25	B224					
30	1	25	B225*					
10	1¼	32	B229					
19	1¼	32	B230*					
25	1¼	32	B231					
37	1¼	32	B232*					
19	1½	40	B238					
29	1½	40	B239					
37	1½	40	B240*					
29	2	50	B248					
46	2	50	B249					
57	2	50	B250*					
85	2	50	B251					
85	2	50	B252					
120	2	50	B253					
240	2	50	B254*					
60	2½	65	B261					
75	2½	65	B262					
110	2½	65	B263					
150	2½	65	B264					
210	2½	65	B265*					
70	3	80	B277					
130	3	80	B278					

C3 Belimo Actuator Specifications

LF Actuators, On/Off

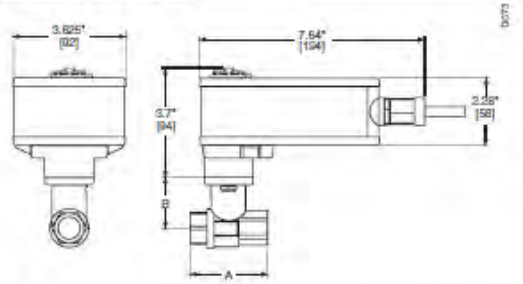
BELIMO

Models

LF24 US	
LF24-S US	w/built-in Aux. Switch
LF120 US	
LF120-S US	w/built-in Aux. Switch

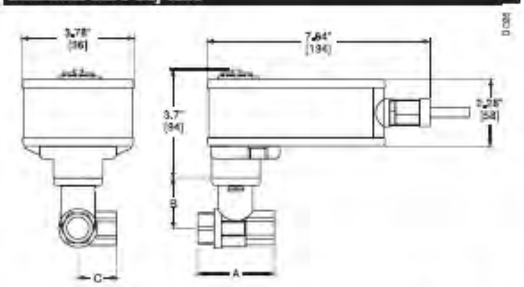
Technical Data		
Control		on/off
Power supply		
LF24(-S) US		24 VAC ± 20% 50/60 Hz 24 VDC ± 10%
LF120(-S) US		120 VAC ± 10% 50/60 Hz
Power consumption		
LF24(-S) US	running	5 W
	holding	2.5 W
LF120(-S) US	running	5.5 W
	holding	3.5 W
Transformer sizing		
LF24(-S) US		7 VA, class 2 power source
LF120(-S) US		7.5 VA, class 2 power source
Electrical connection (-S models have 2 cables)		½" conduit connector 3 ft [1m], 18 GA appliance cable
Electrical protection		120V actuators double insulated
Overload protection		electronic throughout rotation

Dimensions with 2-Way Valve



Valve Body	Valve Nominal Size Inches	DN [mm]	Dimensions (Inches [mm]) A	B
B207(B)-B211(B)	½"	15	2.41" [61.1]	1.39" [35.2]
B212(B)-B215(B)	½"	15	2.38" [60.4]	1.78" [45.2]
B217(B)-B220(B)	¾"	20	2.73" [69.3]	1.87" [47.4]
B222-B225	1"	25	3.09" [78.4]	1.87" [47.4]
B229-B230	1¼"	32	3.72" [94.6]	1.87" [47.4]

Dimensions with 3-Way Valve



Valve Nominal Size	Dimensions (Inches [mm])
Valve Body Inches DN [mm]	A B C

Appendix D: Calculation Reference Data

D1 Spitler, 2003, *HVAC* - Fig 14-12 Fanning Friction Factor

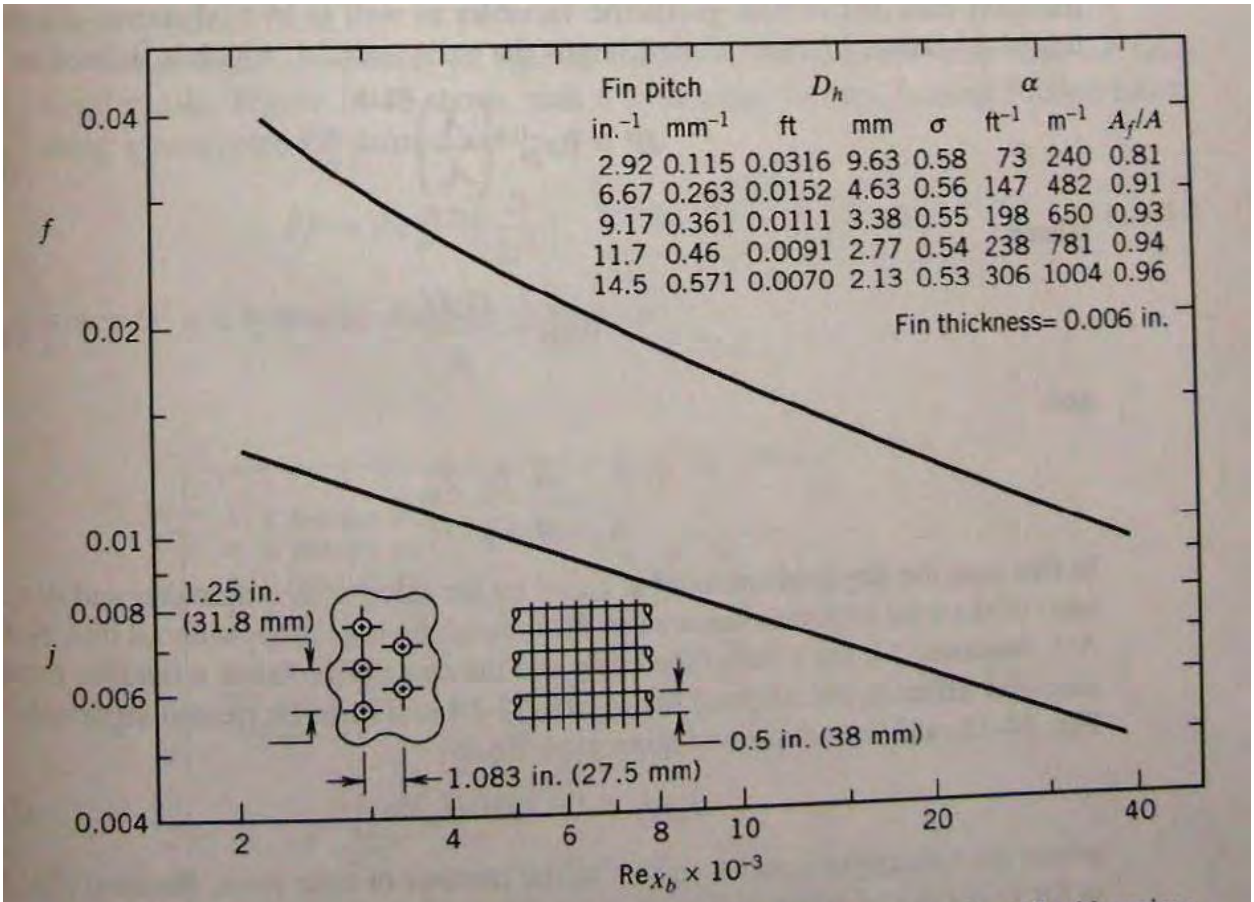


Figure 14-12 Heat-transfer and fanning friction factor data for a plate–fin–tube coil with various fin spacings and five rows of tubes. Fin thickness is 0.006 in. (0.024 mm). (Reprinted by permission from *ASHRAE Transactions*, Vol. 79, Part II, 1973.)

D2 Spitler, 2003, *HVAC* - Fig 14-18 Effectiveness Cross-Flow Unmixed Fluids

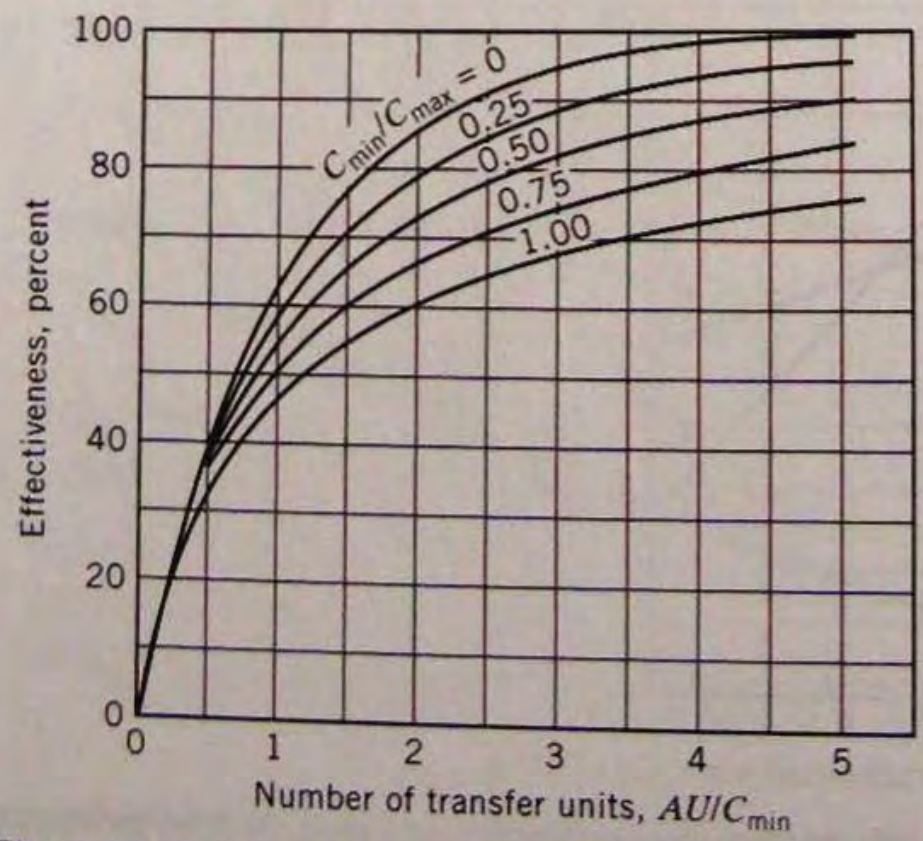


Figure 14-18 Effectiveness of cross-flow exchanger with fluids unmixed.

Appendix E: Heat Transfer Hand Calculations

E1 Ground Coil Calculations

(E) Heat transfer (1)

temp leaving the condense.

Data

$q = 35.35\ M\ BTU/hr$

$D_i = 1.34'' = 0.11'$

$D_o = 1.66'' = 0.13'$

} Pipe Nominal Diameter 1 1/4 DR 11 (160 PSI)

$V = 8\,GPM \left(\frac{3.79\,L}{\cancel{G}}\right)\left(\frac{0.035\,ft^3}{\cancel{L}}\right) = 1.07\ \frac{ft^3}{min}$ } Data.

* Velocity of the water

$\dot{V} = Vel \cdot Area$

$Vel = \frac{1.07}{0.0095}$

$\boxed{Vel = 112.62\ ft/min}$

$A = \frac{1}{4}\pi D_i^2$

$A = \frac{1}{4}\pi [0.11]^2$

$A = 0.0095\ ft^2$

* mass flow rate of the water

$m = \rho \dot{V}$

$m = 62.12$

$\dot{m} = \frac{LBM}{ft^3}\left[1.07\ \frac{ft^3}{min}\right]\left[\frac{60\,min}{hr}\right]$

$\boxed{\dot{m} = 3989.20\ LBM/hr}$

* temp leaving

$$q = m\,C_p\,\Delta t$$

$$t_f = \frac{q}{m C_p} + t_i$$

$$T_f = \frac{35350}{\left(\frac{Btu}{hr}\right) 3989.20 \left[0.99 \frac{Btu}{LbM}\right]} + 85\,F^\circ$$

$$\boxed{t_f = 93.87^\circ F}$$

(2) thermal Resistance Network.

the 1 1/4 Nominal Diameter change into 6 loops 3/4 Nominal Diameter.

$$\dot{V}_1 = 6\dot{V}_2$$

$$8GPM = 6\dot{V}_2$$

$$V_2 = 1.33\,GPM$$

Data

$T_{in} = 93.87^\circ F$ [temp Water in to the ground)

$t_{out} = 85^\circ F$ [temp Water leaving the ground]

$D_i = 0.848'' = 0.07'$

$D_o = 1.05'' = 0.0875'$

Surface Area in term of L

$$A = 2\pi r L$$

$$A = 2\pi \left[\frac{0.07}{2}\right] = 0.2199 \cdot L$$

* the Velocity of the water at 3/4 Nominal Diameter. (2)

$\dot{V} = Vel \cdot Area$

$A = \frac{1}{4} \pi D_i^2$

$Vel = \frac{1.33\ GPM \left| \frac{3.79\ L}{G} \right| \left| \frac{0.035\ ft^3}{L} \right|}{3.84 \cdot 10^{-3}\ ft^2}$

$A = \frac{1}{4} \pi [0.07]^2$

$\boxed{A = 3.84 \cdot 10^{-3}\ ft^2}$

$Vel = \frac{0.17}{3.84 \cdot 10^{-3}}$

$\boxed{Vel = 46.23\ ft/min}$

Convention at 3/4 Nominal Diameter

$Re_y = \frac{VD\rho}{\mu}$

$\rho = 62.12\ LBm/ft^3$

$\mu = 1.842\ LBm/ft\,h$

$D = 0.07$

$Re_y = \frac{46.23\ ft/min\ [0.07\ ft]\ [62.12]\ LBm/ft^3}{1.842\ \frac{LBm}{ft\ hr} \left| \frac{hr}{60\ m} \right|}$

$\boxed{Re_y = 6548.92}$

Nusset #

$Nu = 0.023\ Re_y^{0.8}\ Pr^{n}$ where $n = 0.3$ [cooling

$Nu = 0.023\ [6548.92]^{0.8} [5.14]^{0.3}$

$Nu = 42.45$

Inside heat transfer coheficient

$$Nu = \frac{h_i D_i}{K} = 42.45$$

$$h_i = \frac{42.45 \cdot K}{D_i}$$

$$h_i = \frac{42.45 \cdot 0.358 \ \frac{Btu}{hr\ ft\ F}}{0.07}$$

$$\boxed{h_i = 217.13 \ \frac{Btu}{hr\ ft\ F}}$$

Ressister Inside Convention

$$R_{conv} = \frac{1}{h_i A_i} \quad [\text{Pag } 151]$$

$$A_i = \pi D_i \cdot L$$

$$A_i = \pi [0.07]$$

$$R_{conv} = \frac{1}{217.13 [0.2198 \cdot L]}$$

$$\boxed{R_{conv} \cdot L = 0.020 \ \frac{hr \cdot F}{Btu}}$$

* thermal Resistance Conduction Cylinder. (3)

$$R_c = \frac{\ln(r_2/r_1)}{2\pi L k}$$

r_i $0.07/2 = 0.035'$

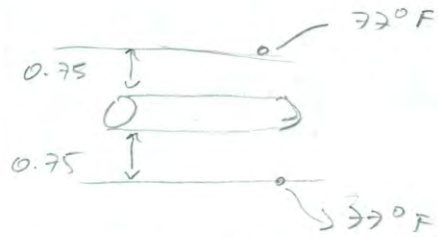
$r_o = \frac{0.0875'}{2} = 0.04375$

$$R_c \cdot L = \frac{\ln\left[\frac{0.04375}{0.035}\right]}{2\pi\,[0.225]}$$

$$\boxed{R_c \cdot L = 0.1578 \ \frac{hr\,F}{BtU}}$$

* Resistance of the Ground.

$$R_g = \frac{1}{S k_g}$$



$$g \cdot L \ \frac{1}{2.34\,[1.2150]}$$

$$\boxed{\cdot L = 0.40 \ \frac{hr\,F}{BtU}}$$

$$S = \frac{2\pi L}{\ln\left[\frac{8z}{\pi D}\right]}$$

$$S = \frac{2\pi L}{\ln\left[\frac{8(0.75)}{\pi(0.0875)}\right]}$$

$$S = \frac{6.28 \ L}{\ln[14.55]}$$

$$\boxed{S = 2.037 \ ft}$$

* total Heat transfer from water *

$m_w = \rho \dot{V}$

$\rho = 62.12 \frac{Lbm}{ft^3}$

$m_w = 62.12 \frac{Lbm}{ft^3} \left[0.1779 \frac{ft}{min} \right] \left[\frac{60 min}{hr} \right]$

$$\boxed{m_w = 663.06 \frac{Lbm}{hr}}$$

$q = \dot{m} C_p [\Delta t]$

$q = 663.06 \frac{Lbm}{hr} \left[0.999 \frac{BTU}{Lbm\, °F} \right] \left[93.87 - 85 \right] °F$

$$\boxed{q = 5875.53 \frac{BTU}{hr}}$$

* total length required to remove 5875.53 BTU/hr

$q = \frac{t_1 - t_2}{R_T}$

$R_T = R_{conv\ water} + R_{cond\ pipes} + R_{pipes}$

$R_T = 0.020 L^{-1} + 0.1578 L^{-1} + 0.43 L^{-1}$

$R_T = 0.6078 \cdot L^{-1} \left[\frac{hr\, °F}{BTU} \right]$

4

total longest.

$$q = \frac{t_i - t_t}{RT(L)}$$

$t_i = 93.87°F$

$t_t = 85°F$

$$RT(L)^{-1} = \frac{93.87 - 85°F}{5876.93} = 1.5 \cdot 10^{-3}$$

$$RT(L^{-1}) = 1.6 \cdot 10^{-3}$$

$$\frac{RT}{L} = 1.6 \cdot 10^{-3}$$

$$\frac{0.52}{L} = 1.6 \cdot 10^{-3}$$

$$L = \frac{0.52}{1.6 \cdot 10^{-3}}$$

$$\boxed{L = 386.03 \text{ ft}}$$

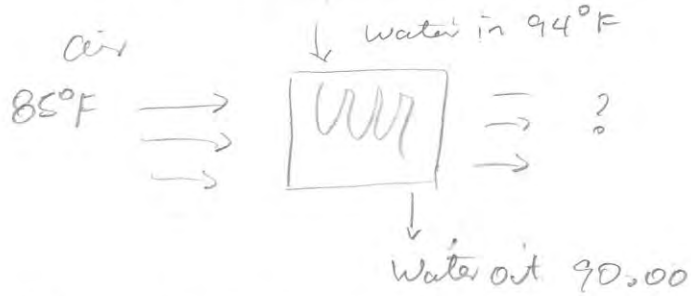
the total longest is $L \times 6$

$L_T = 2316.21$ ft.

E2 Heat Exchanger Calculations

(1)

Heat Echanger's Calculations



(I) total heat removed

$q = m C_p \Delta t$

$q = 3987.462(0.99)(4)$

$\boxed{q = 15940.78 \ BTU/h}$

Properties water

$C_p = 0.99 \ \frac{BTU}{LBM}$

$\rho = 62.11 \ \frac{LBM}{ft^3}$

$\dot{V} = 8GPM = 1.07 ft^3/min$

mass flow of the water

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

$\dot{m} = 62.11(1.07) * 60$

$\boxed{\dot{m} = 3987.462 \ \frac{LBM}{h}}$

II- Mass flow rate of the air

$PV = mRT$

$m = \frac{14.7[144][4000][60]}{53.35[87 + 463.5]}$

$m = 17292.79 \ \frac{LBM}{hr}$

Properties of air

$P = 14.7 \ \frac{LBf}{inch^2}$

$R = 53.35 \ \frac{LBf}{LBM R}$

III - temp out of the air

$\dot{Q} = mC_p\Delta T$

$C_p = 0.24 \frac{Btu}{LBm\,F}$

$15940.87 = 17355(0.24)(T_t - 87°F)$

$\boxed{T_t = 90.84\ °F}$

Assumptions

- water velocity 4 ft/sec = 240 ft/min
- Volumer flow air 4000 cfm
- Air face velocity 900 ft/min

Cooper Specifications

$ND = 0.5''$

$Id = 0.4831'' = 0.040'$

$OD = 0.525'' = 0.04375'$

$A_i = 1.25 \cdot 10^{-3}\ ft^2$

$A_o = 1.50 \cdot 10^{-3}\ ft^2$

Inside heat transfer coefficient

Reynolds.

$Re_y = \frac{\rho V D}{\mu}$

$\rho = 62.12 \frac{LBm}{ft^3}$

$\mu = 1.842 \frac{LBm}{ft\ hr}$

$D = 1.025'' \left(\frac{ft}{12''} \right)$

$Re_y = \frac{62.12 \frac{LBm}{ft^3} \quad 240 \frac{ft}{min} \quad 0.040\ ft}{1.842 \frac{LBm}{ft\ hr} \left[\frac{hr}{60\ min} \right]}$

$Re_y = 19550.60$ turbulent

$h_i = 0.023 \frac{k}{D} (Re_y)^{0.8} (Pr)^{0.3}$

$h_i = 0.023 \left[\frac{0.358}{0.040} \right] \left[19550.60 \right]^{0.8} \left[5.14 \right]^{0.3}$

$h_i = 905.67\ BTU/hr\ ft^2\ F^\circ$

Outside heat transfer coefficient

- Face Velocity 900 ft/min

$G_{fr} = \rho v$

$= 0.0728 \frac{LBm}{ft^3} \left[900 \frac{ft}{min} \right]$

$G_c = \frac{G_{fr}}{\sigma}$ (ratio minimum flow)

$G_{fr} = 65.52 \frac{LBm}{ft^2\ min} = 3931 \frac{LBm}{hr\ ft^2}$

$$G_c = \frac{G_{fr}}{\sigma \text{ (ratio minimum flow)}}$$

$$G_c = \frac{3931}{0.55}$$

$$\boxed{G_c = 7147.63 \ \frac{lbm}{hr\ ft^2}}$$

$$Re_y = \frac{G_c D}{\mu}$$

$$Re_y = \frac{7147.63 \ \frac{lbm}{hr\ ft^2} \left[0.043\right] ft}{0.0454 \ \frac{lbm}{ft\ hr}} \Rightarrow \boxed{Re_y = 6866.69}$$

Data from table 14.12

$\chi_a = 0.090$ ft
$\chi_B = 0.10$ ft
$D_h = 0.011$ ft
$A/A_t = 13.53$
$J_p = 0.019$
$J = 0.0062$

$$\frac{A}{A_t} = \frac{4}{\pi} \frac{\chi_b}{D_h} \frac{\chi_a}{D} \sigma$$

$$\frac{A}{A_t} = 6.32$$

$$J_p = Re_y^{-0.4} \left[\frac{A}{A_t}\right]^{-0.15}$$

$$J_p = \left[6866.69\right]^{-0.4} \left[6.32\right]^{-0.15}$$

$$J_p = 0.019$$

from these value look for the table (3) *

$St\, Pr^{2/3} = 0.0066$

solving for ho

$$0.0066 = \frac{ho}{G_c C_p}\left[\frac{\mu C_p}{k}\right]^{2/3}$$

$$ho = \frac{0.0066}{\left[\frac{\mu C_p}{k}\right]^{2/3}}\left[G_c C_p\right]$$

$$ho = 0.0066\left[\frac{\mu C_p}{k}\right]^{-2/3}\left[G_c C_p\right]$$

$$ho = 0.0066\left[\frac{0.045\ \frac{lbm}{ft\ hr}\left[0.24\ \frac{BTU}{lbm\ F}\right]}{0.015\ \frac{BTU}{hr\ ft\ F}}\right]^{-\frac{2}{3}}$$

$$\left[7147.63\ \frac{lbm}{hr\ ft^2}\left(0.24\ \frac{BTU}{lbm\ F}\right)\right]$$

$ho = 13.16\ BTU/hr\text{-}ft\text{-}F$

Overrall heat transfer coefficient

$$\frac{1}{U_O} = \frac{1}{h_o \eta_{so}} + \cancel{\frac{\Delta x}{k\left(\frac{A_m}{A_o}\right)}}^{\,0} + \frac{1}{h_i \eta_{si}\left(\frac{A_i}{A_o}\right)}$$

$$\frac{1}{U_O} = \frac{1}{13.16\,(0.75)} + \frac{1}{905.67\,(0.16)}$$

$$\boxed{U_O = 8.67 \ \text{Btu/hr ft}^2\,F}$$

Assumptions

$$\frac{\Delta x}{k\left(\frac{A_m}{A_o}\right)} = 0$$

$\eta_s = 0.75$

$\eta_{si} = 1$

(4)

NTU-method to find Geometric configurations

$\dot{P}\dot{V} = \dot{m} R T$

$\boxed{\dot{m}_{air} = 17292.79 \ \frac{lbm}{hr}}$ mass flow Air

$C_{air} = C_p \dot{m}$

$C_{air} = 0.24(17292.79)$

$\boxed{C_{air} = 4150.27 \ BTU/hrF}$ greater

$\boxed{\dot{m}_{w_c} = 3989.20 \ \frac{lbm}{hr}}$ mass flow Water

$C_w = C_p \dot{m}_w$

$C_w = 0.999(3989.20)$

$\boxed{C_{w_h} = 3985.21 \ \frac{BTU}{hrF}}$ minimum

$\varepsilon = \frac{t_{hi} - t_{ho}}{t_{hi} - t_{ci}}$ } Chot minimum

$\varepsilon = \frac{94 - 90}{94 - 85} = 0.44$

$$\frac{C_{min}}{C_{max}} = \frac{3985.20}{4150.27} = 0.96$$

$NTU = 1.23$ GS 14.18.

$$NTU = \frac{U_o A_o}{C_{min}}$$

$$A_o = \frac{NTU\ C_{min}}{U_o}$$

$$A_o = \frac{1.23\left[3985.21\right]}{8.67}$$

$$\boxed{A_o = 564.30\ ft^2}$$

the total Volumen

$$V = \frac{A_o}{\alpha}$$

$$V = \frac{564.30}{170}$$

$$\boxed{V = 3.32\ ft^3}$$

(4)*

face Velocity

$$A_{fr} = \frac{Q}{V_{fr}} = \frac{4000}{900} = 4.44\ ft^2$$

Depth

$$L = \frac{V}{A_{fr}} = \frac{3.32}{4.44} = 0.74$$

$$N_r = \frac{L}{x_5} = \frac{0.66}{0.10} = 7.17$$

the Adjust Number of row is 8 due to the Configurations. However, to get a good Reynolds number 32 rows were used.