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Simulation of concentrated photovoltaic cooling system

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SIMULATION OF CONCENTRATED PHOTOVOLTAIC COOLING SYSTEM

by

Christopher K. Halford

Bachelor of Science
University of Nevada, Las Vegas
2004

A thesis submitted in partial fulfillment
of the requirements for the

Master of Science Degree in Mechanical Engineering
Department of Mechanical Engineering
Howard R. Hughes College of Engineering

Graduate College
University of Nevada, Las Vegas
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ABSTRACT

Simulation of Concentrated Photovoltaic Cooling System

by

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Dr. Robert F. Boehm, Examination Committee Chair
Professor of Mechanical Engineering
University of Nevada, Las Vegas

This paper reports work underway in converting a dish-Stirling system to a dish-PV system at UNLV. The existing SAIC dish – Stirling system is being retrofitted with new fixed-focus facets and an Amonix photovoltaic receiver to replace the Stirling-engine/generator package. As is the case with photovoltaic systems generally, the Amonix cells being used in this application tend to lose efficiency as their temperatures increase. To combat this effect, cooling is provided by circulating liquid through channels in the backing plate. The liquid is then pumped through an array of automotive type radiators and the excess heat is rejected to the ambient air.

Reported here is the development of a numerical model for the cooling system. Experimental data are taken to determine the various properties of the individual components to be used in the system and these data are used in a MATLAB-based simulation. The cooling system model can then be linked to a similar model for the receiver and cell assembly and the optimization functions included in MATLAB can be used to select the input parameters (liquid flow rate, number of radiators, air flow rate, etc.) that maximize the overall efficiency of the system. The predictions of the model can

be used in the selection of the final cooling system design and the validity of the model can be checked against the actual performance of the unit.

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This work is dedicated to the memory of my beloved sister, Stacey Kaplan (1969-2003). I miss you.

CHAPTER 1

INTRODUCTION

System Description

In the Summer of 2001 UNLV and Science Applications International Corporation (SAIC) completed the construction of a 25 kW dish-Stirling system at the UNLV test facility [2]. The system used an array of variable focus circular reflectors to concentrate the available solar energy on a central receiver containing the working fluid for the Stirling cycle. This setup was in use for about a year and a half until engine reliability issues forced a halt in operation. See Figure 1.

Figure 1 SAIC dish-Stirling system in operation at UNLV test facility

Since this time the focus of SAIC has shifted away from electro-mechanical systems like the dish-Stirling and more towards concentrated photovoltaics. Currently, efforts are being made to convert the existing system to this type. The project involves replacing the old engine/generator package with an array of single crystal silicon PV cells and a heat rejection system. In addition to this, the variable focus facets will be replaced by a more accurate fixed focus design. Figure 2 shows the general configuration of the proposed system.

Figure 2 Schematic of proposed concentrated PV system

The available solar flux is collected and concentrated by the array of mirrors mounted at a fixed distance from a central receiver. The photovoltaic panels within the receiver unit convert a portion of this energy into electrical energy. The remainder of this energy must be dissipated by the cooling system. This is accomplished using automotive

type radiators mounted in a steel square tube frame. A single fan is used to pull air through the heat exchangers and coolant is circulated through the system using a centrifugal type pump.

Although the main focus of this thesis is the modeling of the cooling system, a brief description of the PV receiver is given here. The receiver is made up of 24 rectangular (28.5cm X 5.02cm) cells. Each of these cells is attached to a copper plate, which conducts the excess heat out of the cell material. Soldered to the back of each copper plate is a finned copper heat sink through which coolant is circulated. See Figure 3. The high temperature water is then pumped through the cooling system where the excess heat is rejected to the ambient. The PV cells being considered for use in this project are the AMNX 1805 single crystal silicon conversion unit. As is the case with most photovoltaics, there is an inverse relationship between the temperature of the cell material and its efficiency. See Figure 4.

Figure 3 Cross section of PV cell assembly

Figure 4 Efficiency of AMNX 1805 as a function of temperature [1]

Purpose of Study

With the continued availability of fossil fuels more and more in question, the importance of utilizing renewable resources cannot be understated. Currently the cost per kWh for most forms of solar generated energy is still significantly higher than it is for fossil fuels. In order to make these systems more economically viable, design parameters should be selected which will maximize the overall efficiency of the unit. For this study the system efficiency is defined as:

(1)

$$\eta = \frac{Q_{el} - W_p - W_f}{Q_{sol}}$$

Inspection of equation (1) reveals that increasing the system efficiency may be accomplished by either increasing the electrical output of the cells or by decreasing the parasitic losses (i.e. the power used by the fan and the pump). These parameters, however, do not vary independently of one another. For a given solar flux, the power output of the cells may only be increased by decreasing the temperature. This decrease in cell temperature, however, will require an increase in either pump power, fan power, or both.

The problem then becomes one of determining the combination of cell cooling and parasitic losses that yield the maximum efficiency for a given set of ambient conditions. The purpose of this study is the determination of this optimum design.

Methodology

Before the system optimization can be performed, it is necessary to create numerical models for the individual components of the system, namely the cooling system and the PV cells themselves. These two models will then be linked to determine the steady state efficiency of the system as a function of the design parameters. The complete system model will then be optimized for efficiency using the multi-variable constrained optimization routine included in MATLAB. The development of the model for the cells is documented in [5]. This work will focus mainly on the development of the cooling system model, the linking of the models, and finally the optimization of the complete system. Before the specifics of these tasks are examined, however, it is considered worthwhile to discuss the design parameters that will define the system performance.

Inspection of equation (1) reveals that for a given level of solar flux, knowledge of the overall system efficiency requires that Q_{cl} , W_p , and W_f also be known. The goal of this analysis is to express these quantities (and hence the efficiency) as functions of one or more of the following design variables.

- (1) Liquid flow rate through system (q_L).
- (2) Composition of coolant (mr). It was originally assumed that the system would run a mixture of ethylene glycol and water. The thermo-physical properties of the mix can then be expressed as a function of the volume fraction of glycol in the mix.
- (3) Air flow rate through the cores (q_a).
- (4) Number of radiators in parallel configuration (nr). (It should be noted that due to its inferior heat transfer and pressure drop characteristics, the series configuration is not investigated in this study.) Figure 5 shows the liquid side loop with 4 radiators.
- (5) Number of 1mm wide channels in the copper heat sink (nc). See Figures 3 and 5. The significance of the number and dimensions of these channels is discussed in detail in [5]. Generally increasing the number of channels will produce lower cell temperatures (higher electrical output) at the expense of a higher liquid side pressure drop. The numerical results of this separate analysis are included in the MATLAB function called “module” which is included in appendix B. This function takes as its arguments q_L , mr , Q_{sol} , nc and the temperature of the liquid entering the modules. The inlet liquid temperature is then eliminated when this function is linked to the heat exchanger function. Based upon these inputs, the

function predicts the electrical output of the cells, the pressure drop, and the outlet water temperature.

Figure 5 Liquid side loop for 4 radiators. (Note heat sink fins run horizontally across receiver modules.)

Once equation (1) is reduced to a known function of these 5 design variables, The optimization routine can be used to select the values of these which result in the highest efficiency. It is worthwhile to note that while the 5 aforementioned parameters are the only design parameters being considered, the final system model will also contain the variables Q_{sol} and T_{amb} . Because these values cannot be controlled they are treated as constants. The effects of variations of these parameters on the optimization results will be discussed in Chapter 5.

CHAPTER 2

REVIEW OF RELATED LITERATURE

Before any attempt was made to model the heat exchangers selected for this project, a search of the related literature was performed. While many papers on modeling compact heat exchangers were found, they were generally limited to a specific type and were not applicable to the units selected for this project. For this reason it was decided that an empirical approach would be the best course of action. Extensive empirical data for a wide range of core geometries are given by Kayes and London [4]. These data are generally presented in a non-dimensional form for a given tube and fin arrangement. In order for these data to be used directly it would be necessary that all dimensions of the selected radiators either match exactly the geometry for which the data was originally taken, or scale by a constant factor. While this was not the case, the experimental methods used by Kayes and London could be used as a guideline to generate the required data for the selected core.

CHAPTER 3

DEVELOPMENT OF HEAT EXCHANGER MODEL

Geometry of Selected Heat Exchangers

The heat exchangers selected for this project are a cross flow finned tube type commonly used in automotive applications. Each unit consists of an inlet tank and an outlet tank connected by 3 rows of 55 oval cross-section tubes, which are 24.5" in length. See Figures 6 and 7.

Figure 6 Schematic of selected radiator

Figure 7 Cross section of liquid flow tubes. ($a=.395''$, $b=.075''$, $c=.038''$)

The airflow channels are created by a flat 2" wide serpentine fin, which runs the length of the core. See Figure 8.

Figure 8 Airflow channels

The fin pitch is 12 fins per inch and the overall width of the core is 26.5"

Liquid Side Pressure Drop

To predict the pump work for the entire system, the relationship between pressure drop and flow rate for the liquid side of a single core must be determined. The experimental setup is shown in Figure 9.

Figure 9 Experimental setup used to measure liquid side pressure drop and flow rate

The flow rate was varied between 0 to 14.5 GPM, and the corresponding pressure drop through the core was recorded. For this test the liquid being used was water, with the thermo-physical properties being evaluated at 20 c. These results were then non-dimensionalized so that they may be used for any mixture of ethylene glycol and water selected. The Reynolds numbers (based on hydraulic diameter) being considered here are low enough that the flow within the individual tubes may be considered laminar. For laminar flow through a non-circular tube, the expected relationship between the friction coefficient and the Reynolds number is given as [8] ;

(2)

$$C_{fL} \cdot Re_L = \text{Constant}$$

This was consistent with the data and the average value of the constant was determined to be 37.016. Figure 10 shows a plot of this relation.

Figure 10 Experimentally derived relationship between Reynolds number and friction coefficient

Figure 11 shows a plot of the actual data as well as the predicted pressure drop for a 50% water glycol mix.

Figure 11 Pressure drop vs. flow rate. Blue points are actual measured data using pure water. Red line is the predicted relation for a 50% water/glycol mix.

Based upon this analysis, the pressure drop through the core for a given mix of glycol and water is known as a function of flow rate. From this, the required pumping power can be determined using the relation.

(3)

$$W_P = \frac{q_L \cdot \Delta P_L}{\eta_P}$$

Test Apparatus for Air Side Pressure Drop and Thermal Measurements

Measurement of the pressure drop for the air side as well as the thermal properties of the core required the construction of the special test bed as shown in Figure 12.

Figure 12 Schematic of test setup

The apparatus is essentially a rectangular plywood enclosure with the fan fitting tightly at one end. See Figure 13.

Figure 13 Fan and intake side of enclosure.

The fan speed and hence the air flow rate is varied using the variable frequency drive shown in Figure 14.

Figure 14 Variable frequency drive

The air flows through a straightener made of 0.25-inch honeycomb and then through a converging section, which forces the air through the fins of the core. See Figures 15 and 16.

Figure 15 Flow straightener downstream of fan

Figure 16 Converging section and core

Along the length of the enclosure, holes are drilled through which probes may be inserted to measure pressure, temperature and velocity. The device used for these measurements is the VelociCalc Plus model 8384 shown in Figure 17.

Figure 17 VelociCalc Plus model 8384

This instrument uses a hot wire anemometer probe to measure air velocity and temperature. In addition to this, two barb type connectors located at the top of the unit can be attached to Pitot tubes to measure the change in total pressure between two points. The liquid side of the core is connected to the hot water supply in the lab with a flow meter and an RTD placed in series. Another RTD is placed downstream of the core to measure the outlet temperature of the liquid. See Figure 18. A butterfly valve is placed in series to allow adjustment of the liquid flow rate.

Figure 18 RTD and flow meter at core outlet

The described setup allows for measurement of the following parameters.

- (1) Mass flow rate of liquid and air.
- (2) Inlet temperatures of liquid and air.
- (3) Outlet temperatures of liquid and air.
- (4) Pressure drop for the air side of the core.

Air Side Pressure Drop

Pressure drop through the air side was determined in much the same way as it was for the liquid side. Using the setup described, the airflow rate through the core was varied using the variable frequency drive. The corrected volumetric flow rate was measured downstream of the core using the anemometer. It should be noted that this instrument calculates the volumetric flow rate by measuring the velocity at one point and multiplying by the cross sectional area. The validity of assuming constant velocity across

the core was inspected by taking measurements at many different locations and comparing them. For the flow rates being considered here, the spatial variation of the velocity was not significant. The corresponding pressure drop was measured using pitot tubes placed upstream and downstream of the core. Figure 19 shows the downstream pitot tube and the anemometer probe in front of the core.

Figure 19 Pitot tube and anemometer probe downstream of the core

As with the liquid side, the results are non-dimensionalized so that the effects of temperature variations on the thermo-physical properties of the air may be taken into account in the final model. Figure 20 shows the experimental relationship between the friction coefficient and the Reynolds number (based upon fin depth).

Figure 20 Experimental relation between air side Reynolds number and friction coefficient

Based upon this analysis, the pressure drop through the air side of the core is known as a function of flow rate. From this, the required fan power can be determined using the relation.

(4)

$$W_f = \frac{q_a \cdot \Delta P_a}{\eta_f}$$

Thermal Analysis

The ultimate goal with respect to the thermal analysis of the radiator is to predict the liquid side outlet temperature based upon the flow rates of the two streams, the liquid side inlet temperature, and the ambient air temperature. In order to accomplish this, an effectiveness – NTU analysis is performed. For a given heat exchanger, the effectiveness is defined as the ratio of the actual rate of heat transfer to the maximum rate possible [3].

This maximum occurs when the hot fluid exits the unit at the inlet temperature of the cold fluid. This is the theoretical limit for an exchanger of infinite area. For the case being considered here the effectiveness is given by the expression.

(5)

$$\varepsilon = \frac{C_h(T_{L_in} - T_{L_out})}{C_{min}(T_{L_in} - T_{amb})}$$

Inspection of equation 5 reveals that it contains 2 unknown quantities, ε and the liquid outlet temperature. In order to express the effectiveness in terms of known quantities (radiator geometry, air and liquid flow rates), a dimensionless quantity known as the “number of transfer units” or NTU is introduced. This parameter is defined as.

(6)

$$NTU = \frac{U \cdot A}{C_{min}}$$

Where it can be shown that [4]

(7)

$$\varepsilon = f(NTU, Cr)$$

The radiators being used are of the unmixed cross-flow configuration. For this type, the empirical relation between effectiveness and NTU suggested by Kayes and London [4] is

(8)

$$\varepsilon = 1 - \exp\left[\frac{NTU^{.22}}{Cr} \cdot (\exp(-NTU^{.78} \cdot Cr) - 1)\right]$$

Given these relationships, the problem becomes one of determining U as a function of the known values. The overall heat transfer coefficient is defined as

(9)

$$\frac{1}{UA} = \frac{1}{h_L \cdot A_{wL} \cdot n} + \frac{t}{k \cdot A_{wL} \cdot n} + \frac{1}{h_a \cdot A_{wa} \cdot n f \cdot \eta_0}$$

The first term on the right hand side of equation (9) represents the thermal resistance between the liquid and the inner surface of the tube, the second term is the resistance between the inner and outer tube surfaces, and the last term is the resistance between the fin and the ambient air. For this case, the tube wall thickness is very small and the thermal conductivity of the radiator material (copper) is relatively high. Based upon this, the wall resistance term may be neglected and the equation reduces to

(10)

$$\frac{1}{UA} = \frac{1}{h_L \cdot A_{wL} \cdot n} + \frac{1}{h_a \cdot A_{wa} \cdot n f \cdot \eta_0}$$

It should be noted at this point that the individual values of U and A are unimportant. Only the product of the two has any significance in this case. Equation (10) has two unknown values h_L and the product $h_a \eta_0$. An important point to mention is the fact that these two quantities are independent of one another. That is to say that the heat transfer coefficient for the liquid side is unaffected by any changes in airflow rate or air properties. Similarly, the coefficient for the air side is independent of the flow characteristics on the liquid side. This being the case two sets of tests may be conducted to determine each of these quantities experimentally.

Experimental Determination of Convection

Coefficient for the Liquid Side

The purpose of this section is the development of an experimental relationship between the liquid side Reynolds and Nusselt numbers. Once this is done, h_L will be a known function of the liquid side flow rate and the thermo-physical properties of the fluid being used. The analysis is based upon the basic convection heat transfer equation for fluid moving through a tube with an imposed wall temperature distribution. See Figure 21.

Figure 21 Basic geometry for liquid side heat transfer analysis

The heat transfer for the jth tube is given by

(11)

$$Q_j = h_L \cdot p \cdot \int_0^L (T_L(y) - T_{w_j}(y)) dy$$

If the convection coefficient is assumed constant across the core, the total heat transfer for a radiator of n tubes is given by.

(12)

$$Q_{\text{total}} = h_L \cdot P \cdot r \sum_{j=1}^n \int_0^L (T_{L_j}(y) - T_{w_j}(y)) dy$$

Where

(13)

$$Q_{\text{total}} = m_L \cdot c_{pL} (T_{L_in} - T_{L_out})$$

The experimental procedure is as follows:

The liquid flow rate through the core is varied from 0 to 12.6 GPM with the airflow rate being held constant. These tests are run using pure water, although the results will later be non-dimensionalized to be used for any mixture of water and glycol. The thermo-physical properties of the water are assumed to be temperature dependent and are evaluated at the bulk liquid temperature given by

(14)

$$T_{\text{bulk}_L} = \frac{T_{L_in} + T_{L_out}}{2}$$

For each value of flow rate tested, tube temperatures are measured at 25 points on the core using the probe shown in figure 22. The liquid temperature at the inlet and outlet are measured and Q_{total} is calculated using equation (13). These data are included in appendix C.

Figure 22 Temperature probe used to measure tube wall temperatures

Using linear interpolation, a continuous function $T_w(y)$ may be created for tubes at any value of x . The inlet and outlet temperatures are also recorded. Substituting equation (13) into equation (12), the only unknowns in are the water temperature $T_L(y)$ and the liquid side convection coefficient h_L . Due to the small dimensions of the flow channels, there is no effective way to measure the water temperature inside of the tubes without disrupting the flow. To circumvent this problem, the water temperature is assumed to vary as a second order polynomial fitted to the conditions $T_L(L) = T_{L_in}$, $T_L(0) = T_{L_out}$, and $dT/dy = 0$ at $y=0$, Figure 23 shows a typical profile for the tube (dotted line) and water temperatures (solid line).

Figure 23 Typical profiles for tube wall (dashed) and liquid temperatures (solid).

Using this approximation, equation (12) can be solved for h_L for each value of liquid flow rate tested. The program “hal”, included in appendix B, was written to perform the interpolation and numerical integration required to solve this equation. Figure 24 shows the experimental relationship between the liquid side Reynolds and Nusselt numbers that was generated by this program.

Figure 24 Experimental relation between liquid side Reynolds and Nusselt numbers

Experimental Determination of Convection

Coefficient for the Air Side

The purpose of this section is the development of an experimental relationship between the air side Reynolds and Nusselt numbers. Once this is done, the product $h_a \eta_0$ will be a known function of the air side flow rate and the temperature dependent thermo-physical properties of the air. The convection coefficient for the air side is determined in much the same manner as was the liquid side. The analysis is based upon the convection heat transfer equation for a fluid moving over a fin with an imposed base temperature. See Figure 25.

Figure 25 Basic geometry for air side heat transfer analysis.

The fins are assumed small enough to neglect any variations in temperature along the surface of an individual fin. This assumption is made due to the fact that a classical flat

fin analysis where the temperature does vary spatially on individual fins would greatly increase the complexity of the problem for marginal gains in accuracy. This is especially true considering the fact that the diameter of the probe being used to measure the base temperatures is one quarter of the length of the fin itself. The temperature of the fin $T_{f,j,k}$ is measured at the center point $d/2$ in the z direction (normal to page). The j in the subscript refers to the j_{th} vertical column of fins and the k refers to the k_{th} horizontal row. By measuring at 25 points across the core surface and interpolating, temperatures may be found for a fin at any location (x,y) . The air temperature over the fins varies in both the y and z directions, while changes in the x direction were found to be minimal. The air is assumed to vary as a second order polynomial fitted to the conditions, $T_a(y,0) = T_{amb}$, $T_a(y,d) = T_{a_out}(y)$, and $dT/dz = 0$ at $z=d$. Where $T_{a_out}(y)$ is determined by measuring the outlet air temperature at several y values and fitting a function to the data. From this, the heat transfer from fin j, k is given by;

$$Q_{jk} = h_a \cdot L_f \cdot \eta_0 \cdot \int_0^d (T_{f,j,k} - T_a(y_k, z)) dz \quad (15)$$

Where y_k is the y value of the k_{th} row of fins. At this point it is worthwhile to note that by assuming that the entire fin is at the base temperature, by definition the fin efficiency η_0 is equal to unity. It is carried through the calculations for the sake of completeness. Assuming that h_a is constant across the core, the rate of heat rejection for the entire radiator is given by

$$Q_{total} = h_a \cdot L_f \cdot \eta_0 \cdot \sum_{j=1}^u \sum_{k=1}^v \int_0^d (T_{f,j,k} - T_a(y_k, z)) dz \quad (16)$$

Where as before

(17)

$$Q_{\text{total}} = m_L \cdot c_{pL} \cdot (T_{L_in} - T_{L_out})$$

Where this quantity is measured as in the previous section. For each value of air flow rate tested equations (16) and (17) may be combined and the product $h_a \eta_0$ solved for. The program “haa” was created to perform the interpolation and numerical integration required to solve the resulting equation for each airflow rate tested. This code is included in appendix B. The experimental results generated by this program are shown in Figure 26.

Figure 26 Experimental relationship between air side Reynolds and Nusselt numbers.

CHAPTER 4

PROGRAMMING

(All programs referred to in this section are included in appendix B)

Rad_SI Function

The function “rad_SI” was created to predict the air and liquid side pressure drops as well as the liquid outlet temperature for a given combination of air flow rate, liquid flow rate, mix ratio, ambient air temperature, and number of radiators. The routine uses the data collected in Chapter 3 to estimate the values of the air and liquid side Nusselt numbers and friction coefficients based upon the Reynolds numbers for the given flow rates through the radiator. The properties of both fluids are assumed temperature dependent, with those for the liquid side also dependent upon the selected volume fraction of glycol. For the air side, the functions “VisAir”, “CpAir”, “CondAir”, and “DenseAir” use data given in [3] to predict the properties (viscosity, constant pressure specific heat, thermal conductivity, density) for air at a given temperature. For the liquid side, the functions “VisMix”, “CpMix”, “CondMix”, and “DensMix”, use data for water and glycol also given in [3]. For a given temperature these functions interpolate values for the various properties (viscosity, constant pressure specific heat, thermal conductivity, density) for both water and glycol. The mass fraction of glycol can then be calculated based upon the given volume fraction. The properties of the mix may then be calculated using

$$P_{\text{mix}} = \text{mf} \cdot P_g + (1 - \text{mf}) \cdot P_w$$

Where P refers to any of the aforementioned thermo-physical properties and mf is the mass fraction of glycol. All fluid properties are evaluated at a bulk temperature which is taken as simply the average between the inlet and the outlet temperatures. Since the outlet temperature is not known a priori, the function rad_SI uses the iterative routine described below.

- (1) Reynolds numbers are calculated for both the air and liquid sides based upon the input variables and an initial guess value for the bulk temperatures.
- (2) The corresponding Nusselt numbers are estimated based upon the empirical data.
- (3) With the Nusselt numbers known the outlet temperatures for both streams are calculated using the effectiveness-NTU relations described in Chapter 2.
- (4) The bulk temperatures for both fluids are recalculated by taking the average of the inlet temperatures and the new outlet temperatures.
- (5) Reynolds numbers are recalculated based upon the new bulk temperatures and the corresponding Nusselt numbers are selected.
- (6) New outlet temperatures are generated based upon the new Nusselt numbers and the process is repeated until the difference in the change in both bulk temperatures between iterations reaches a preset limit.

Module Function

The function “module” was created to predict the pressure drop, total cell electrical output, outlet liquid temperature, and maximum cell temperature for a given combination of liquid flow rate, mix ratio, inlet liquid temperature, and number of channels per module. This program is based upon the analysis described in [5] and uses the finite element routine described below.

- (1) The module is divided into sections $\frac{1}{2}$ ” in length. For the first section, the Reynolds number is calculated for the given number of channels, liquid flow, rate, and mix ratio. The liquid temperature is at the specified inlet temperature.
- (2) The corresponding Nusselt number and friction coefficient are calculated using the empirical models presented in [3]. These values are used to calculate the local convection coefficient and the pressure drop across the finite element.
- (3) A guess value is given for the cell temperature and the corresponding cell efficiency estimated using the data in Figure 4. The net energy flux into the element is then the difference between the incident solar flux and the electrical output. Using this estimated value, an energy balance is constructed for the element and a new cell temperature is calculated. The cell efficiency is then recalculated based upon this new temperature. This is repeated for the element until the difference between iterations falls within the specified tolerance.
- (4) The liquid temperature for the next element is calculated based upon the net energy flux into the previous element and the entire process is repeated until the cell and liquid temperatures are known for the entire module.

- (5) The electrical output and pressure drop for the individual elements are summed to give the total values for the modules.

PV Function

The function “PV” is used to link the functions module and rad_SI to simulate the performance of the entire system. This function takes the arguments air flow rate, liquid flow rate, number of radiators, and the number of channels per module and outputs the overall system efficiency as defined by equation (1). A guess value is given for the inlet temperature to the cells. The module function calculates an outlet value based upon this guess. This then becomes the inlet temperature for the function rad_SI, which predicts the temperature of the liquid leaving the radiators. This becomes the new module inlet temperature. This process is repeated until the temperature difference between iterations falls within a specified tolerance. The outputs of this function are the steady state electrical output, pump work, fan work, system efficiency, maximum cell temperature, air side pressure drop, and liquid side pressure drop. It should be noted that this model does not take into account any liquid side losses in the plumbing or any air side losses due to obstructions (pipes, radiator overflow tank, etc.) within the receiver itself.

Eff_Inv function

The function “Eff_inv” takes the same inputs as PV. The output is the reciprocal of the total system efficiency for all positive net electrical outputs, and infinity for all negative values.

Opt_PV function

The function “Opt_PV” calls the function Eff_Inv and determines the combination of air flow rate, liquid flow rate, number of radiators, and number of channels which produce a minimum in Eff_inv. This routine uses a modified version of the MATLAB multi-variable constrained optimization function “fmincon” called “fminconset” [7] This function was designed by Mr. Ingar Solberg to perform optimization for the case where one or more of the variables are limited to discrete values. This is the situation for the number of radiators and the number of channels.

CHAPTER 5

RESULTS

Radiator Model

To verify the accuracy of the radiator model, the test setup described in Chapter 3 was run for various combinations of air flow rate, liquid flow rate, and ambient temperature. The outlet liquid temperature was then compared to the value predicted by the model. These results are shown in Table 1.

Air flow rate (ft³/s) STD	Liquid flow rate (GPM)	Ambient air temp (F)	Inlet water temp (F)	Measured outlet water temp (F)	Predicted outlet water temp (F)	Percent error
1855	1.72	78	118.6	86.3	82.3	12.3839
1915	3.91	76.4	120.4	97.6	93.5	17.9825
1040	2.55	55.1	116.4	85.8	81.1	15.3595
1495	2.55	51.6	119.4	76.2	71.5	10.8796
384	2.55	69	112.3	98.95	99.5	4.11985
1960	2.9	76.2	119.55	94.3	87.6	26.5347
3300	2.55	45.3	125.2	64.65	54.6	16.5979
2230	2.55	51.2	117.7	71.55	63.9	16.5764
1920	0.79	73.1	121.55	77.15	73.4	8.44595
1970	5.56	75.9	117	102.3	96.9	36.7347
1905	4.54	79	120.6	101.2	97.5	19.0722
1710	8.1	62.5	121.55	103.75	101.6	12.0787

Table 1 Results of Radiator Testing. % error calculated as

$$\frac{|T_{out\ meas} - T_{out\ theo}|}{|T_{in\ meas} - T_{out\ meas}|}$$

For the points tested the maximum error was 37 % with most values significantly less than this. This is within the range of acceptable error. For all of the cases tested the model tends to predict a temperature lower than the actual value. It is believed that this is most likely due to an overestimation of the air side heat transfer coefficient due to the assumed quadratic variation of the air moving over the fins. In future models a linear profile could be used to correct this.

Optimization and Selection of Design

In order to select the best design for the location selected for this project, the optimization function described in the previous chapter was linked to a file containing typical hourly solar flux and temperature data for the entire year. [6] For this simulation it is assumed that the pump and fan speeds can be varied independently to produce the combination of air and liquid flow rates which produce the maximum efficiency for the ambient conditions at any given hour. The pump and fan efficiencies are assumed constant over their respective operating ranges and are set at 0.7. The mix ratio is set at 0. The other two variables cannot be changed after the system is built. In order to address this, the yearly simulation is run with three of the variables (air flow rate, liquid flow rate, number of radiators) allowed to float over a specified range. The number of channels per module is fixed at 1 and the net energy output for each hour is calculated. The number of channels is then increased to 2 and the process is repeated for up to 15 channels. The yearly net electrical output can then be summed for each value of n_c . Figure 27 shows the maximum yearly energy production as a function of number of channels.

Figure 27 Maximum yearly energy production vs. nc

Inspection of the graph shows that the maximum energy is produced when $nc=6$ channels per module. Because the model for the system is generally a very strong function of the number of radiators, the expected result for each run is that there will be strong preference for one value of nr. Figure 28 shows the number of times each value of nr was selected as the optimum for $nc=6$.

Figure 28 Histogram of nr for $nc=6$

Based upon this, there is clearly a strong preference for $nr=4$. Figures 29-40 shows the predicted hourly net energy production for the optimized system ($nc=6$, $nr=4$) for each month of the year.

Figure 29 Optimized system output for January

Figure 30 Optimized system output for February

Figure 31 Optimized system output for March

Figure 32 Optimized system output for April

Figure 33 Optimized system output for May

Figure 34 Optimized system output for June

Figure 35 Optimized system output for July

Figure 36 Optimized system output for August

Figure 37 Optimized system output for September

Figure 38 Optimized system output for October

Figure 39 Optimized system output for November

Figure 40 Optimized system output for December

Constraints on the Optimization

The optimization routine described in the previous chapter requires that limits be placed on the variables. All efforts were made to ensure that the limits selected reflect an actual physical restriction of the system. By doing this, the situation is avoided in which the optimization routine always selects the maximum allowed value for a certain variable, but this maximum is an artificially imposed constraint. For this reason, these limits and some explanation as to the selection of each are presented here.

The maximum value for the air flow rate was set at $1.7 \text{ m}^3/\text{s}$ std. This is the maximum measured output through the air side of a single core for the selected fan running at 60 Hz. The theoretical minimum for this parameter is 0. In order to reduce the run time on

for the program, however, a value of $0.2 \text{ m}^3/\text{s}$ std was used. This value is low enough that the optimum value selected by the program is always higher than this limit.

The maximum value for the number of radiators was set as 4. Because the receiver and cooling system are to be mounted on an existing system, they must be of similar weight and geometry as the original engine/generator package. Based upon this, it was determined that this is the maximum number of radiators that could be mounted in the given space. The minimum value for this parameter was set as 1.

The maximum number of channels was set as 15. Due to the difficulty involved in machining very small channels, manufacturing a heat sink with more channels than this would be impractical. The minimum value for this variable was set at 1.

The maximum value for the liquid flow rate was set at $.007 \text{ m}^3/\text{s}$. This is based upon the initial design specifications presented by SAIC. This value is high enough that the optimum value selected by the program is always lower than this maximum. The theoretical minimum for this parameter is 0. In order to reduce the run time on for the program, a value of $.0015 \text{ m}^3/\text{s}$ was used. This value is low enough that the optimum value selected by the program is always higher than this minimum.

CHAPTER 6

CONCLUSIONS

The predictions made by the radiator model seemed to agree reasonably well with the actual performance of the unit. In the future the code could possibly be modified to accept data for other compact heat exchangers. The test apparatus described in Chapter 3 could be modified to accept other cores and the instrumentation could be improved to increase precision and reduce the time needed to collect the data. Later versions of the model would most likely use a linear air temperature profile to correct for the over prediction of the air side heat transfer coefficient evident in this version.

Based upon the results of the yearly simulation presented in Chapter 5, the optimum system design would consist of 4 radiators and 6 channels per module with the air and liquid flow rates being allowed to vary. It is worthwhile to note, however, that while Figure 27 shows a maximum at $nc=6$, the variation in net the energy production over the entire range of nc is less than 1%. Due to this, the value of this variable would most likely be determined by other factors (cost, ease of manufacture, etc.).

Because the actual system is still under construction the accuracy of the complete system model cannot at this time be verified. When it is completed, however, the actual output can be compared to the predicted values and if necessary changes made. Once the output of the system model has been verified, the optimization routine could then be modified to be used as the logic for a closed loop control system for the fan and pump. If

this could be done successfully the result would be a system which runs near its maximum efficiency for any set of ambient conditions.

APPENDIX A

NOMENCLATURE

A_{wL} = Wall surface area of liquid flow channels (m^2)

A_{wa} = Wall surface area of air flow channels (m^2)

C_{fL} = Friction factor for liquid side

C_h = Heat capacity of the hot fluid (W/kg)

C_{max} = Higher of the two fluid heat capacities (W/kg)

C_{min} = Lower of the two fluid heat capacities (W/kg)

c_{pL} = Constant pressure specific heat for water glycol mix (J/kg K)

Cr = Ratio of heat capacities = C_{min} / C_{max}

d = Depth of core (m)

h_a = Average convection coefficient for the air side of the core ($W/m^2 K$)

h_L = Average convection coefficient for the liquid side of the core ($W/m^2 K$)

k = Thermal conductivity of radiator tube material (W/m K)

L = Length of core (m)

L_f = Length of air side fins (m)

mf = Mass fraction of glycol in mix

m_L = Mass flow rate of water glycol mix through the core (kg/s)

mr = The volume fraction of glycol

n = Number of tubes on the liquid side of the core
 n_c = Number of channels in module heat sink
 n_f = Number of fins per radiator
 n_r = Number of radiators in parallel
 NTU = Number of transfer units
 P = Generic thermo-physical property of mix
 P_g = Generic thermo-physical property of glycol
 P_w = Generic thermo-physical property of water
 p_t = Perimeter of liquid side tubes (m)
 ΔP_a = Pressure drop through the air side of the core (Pa)
 ΔP_L = Pressure drop through the liquid side of the core (Pa)
 q_a = Volume flow rate of air through the core (m^3/s)
 Q_{cl} = Electrical power produced by cell array (W)
 q_L = Volume flow rate of water glycol mix through the core (m^3/s)
 Q_{sol} = Solar power available at receiver (W)
 Q_{total} = Power dissipated by single radiator (W)
 Re_L = Reynolds number for the liquid side of the core
 t = Thickness of liquid side tube wall (m)
 T_a = Air temperature at arbitrary position (K)
 T_{amb} = Ambient air temperature (K)
 T_{a_out} = Temperature of air exiting core (K)
 T_{bulk_a} = Bulk temperature of air (K)
 T_{bulk_L} = Bulk temperature of water glycol mix (K)

T_f = Air side fin temperature at arbitrary position (K)
 T_L = Liquid temperature at arbitrary position (K)
 T_{L_in} = Temperature of water glycol mix entering core (K)
 T_{L_out} = Temperature of water glycol mix exiting core (K)
 T_w = Wall temperature of liquid side tube at arbitrary position (K)
 u = Number of vertical columns of fins on the air side of the core
 v = Number of horizontal rows of fins on the air side of the core
 W_f = Fan power (W)
 W_p = Pump power (W)
 x = Spatial coordinate
 y = Spatial coordinate
 z = Spatial coordinate

Greek Symbols

ε = Overall effectiveness of heat exchanger
 η = Overall system efficiency
 η_f = Rated fan efficiency
 η_p = Rated pump efficiency
 η_0 = Fin efficiency

APPENDIX B

MATLAB PROGRAMS

(MATLAB was used extensively in this simulation. This section includes all codes produced for this project.)

Code for Air Properties Functions

```
function VisAir=f(T);  
%This function takes the following arguments  
%T = Average air temp in K  
%It returns the kinematic viscosity of the air in (W/mK)  
VisAir=-.00000000002*T^2+.000000006*T+.000003;
```

```
function CondAir=f(T);  
%This function takes the following arguments  
%T = Average air temp in K  
%It returns the Thermal conductivity of the air in (W/mK)  
CondAir=-.00000007*T^2+.0001*T-.0001;
```

```
function CpAir=f(T);  
%This function takes the following arguments  
%T = Average air temp in K  
%It returns the Cp of the air in (J/kg K)  
CpAir=.0004*T^2-.2438*T+1043.6;
```

```
function DenseAir=f(T);  
%This function takes the following arguments  
%T = Average air temp in K  
%It returns the density of the air in (W/mK)  
DenseAir=.000004*T^2-.0058*T+2.5321;
```

Code for Liquid Properties Functions

```

function CondMix=f(Tav,mix_ratio)
%This function takes the following arguments
%Tav = Average water temp in K
%mix_ratio = Volume fraction of glycol
%It returns the Thermal conductivity of the mix in (W/mK)
% Density of glycol (kg/m^3) as a function of Temp (K)
DensG=3488.8*Tav^-.2006;
% The mass fraction of glycol is
mg=mix_ratio*(DensG/DensMix(Tav,mix_ratio));
CondW=-.000007*Tav^2+.0057*Tav-.464;
CondG=-.000003*Tav^2+.0018*Tav-.0713;
CondMix=mg*CondG+(1-mg)*CondW

function DensMix=f(Tav,mix_ratio)
%This function takes the following arguments
%Tav = Average water temp in K
%mix_ratio = Volume fraction of glycol
%It returns the density of the mix in (kg/m^3)
DensW=-.0032*Tav^2+1.6094*Tav+796.7;
DensG=3488.8*Tav^-.2006;
DensMix=mix_ratio*DensG+(1-mix_ratio)*DensW;

function VisMix=f(Tav,mix_ratio)
format long
%This function takes the following arguments
%Tav = Average water temp in K
%mix_ratio = Volume fraction of glycol
%It returns the viscosity of the mix in (N s/m^2)
% Density of glycol (kg/m^3) as a function of Temp (K)
DensG=3488.8*Tav^-.2006;
VISG=[6.51 4.2 2.47 1.57 1.07 .757 .561 .431 .342 .278 .228 .215]/100;
T=[273 280 290 300 310 320 330 340 350 360 370 373];
% The mass fraction of glycol is
mg=mix_ratio*(DensG/DensMix(Tav,mix_ratio));
VisW=20000000000*Tav^(-5.4286);
VisG=(3*10^24)*Tav^(-10.577);
VisMix=mg*VisG+(1-mg)*VisW;

function CpMix=f(Tav,mix_ratio)
%This function takes the following arguments
%Tav = Average water temp in K
%mix_ratio = Volume fraction of glycol
%It returns constant pressure spec heat of the mix in (J/kg K)
% Density of glycol (kg/m^3) as a function of Temp (K)

```

```

DensG=3488.8*Tav^-.2006;
% The mass fraction of glycol is
mg=mix_ratio*(DensG/DensMix(Tav,mix_ratio));
Cpw=.0121*Tav^2-7.7068*Tav+5408.6;
Cpg=92.006*Tav^(.573);
CpMix=mg*Cpg+(1-mg)*Cpw;

```

Code for Program “hal”

```

%Program hal
L=24.5*2.54/100; %Core Length in meters
W=26.5*2.54/100; %Core Width in meters
y=[0:.001:L];
%number of tube banks
N=55;
%Tube cross sectional area (m^2)
At=.00002196;
%Tube wall area (m^2)
Aw=.016;
% The tube perimeter is (m)
pt=.026;
%The hydraulic diameter is (m)
Dh=.003372;
X=[0 6.75 13.25 20 26.5]*(2.54/100);
Y=[0 6 12 18 24.5]*(2.54/100);
%distance between tube banks
delta_x=.5*2.54/100;
%The data taken is as follows
%Test 1
Tin1=((((123.3+119.8)/2)-32)/1.8)+273.15; %inlet temp in K
Tout1=((((77.1+77.2)/2)-32)/1.8)+273.15; %outlet temp in K
Tbulk1=(Tin1+Tout1)/2; %bulk temp K
Q1=.00004984; %vol flow rate (m^3/s)
mdot1=DensMix(Tbulk1,0)*Q1; %mass flow (kg/s)
qdot1=mdot1*Cpmix(Tbulk1,0)*(Tin1-Tout1); %measured heat transfer rate in W

Tt1=[74.5 76.8 73.9 73.8 73.6; %measured tube temps (F)
     75.4 79.0 76.7 74.5 73.8;
     76.0 85.8 79.0 79.2 74.4;
     87.5 95.7 89.3 88.6 77.8;
     99.1 105.2 113.0 112.6 95.6;];

```

```

Tt1K=((Tt1-32)/1.8)+273.15; %measured tube temps (K)

Temp_int_sum=0;
n=0;

while n<=(N-2);
    x=n*delta_x;
    T=[interp1(X,Tt1K(1,:),x) interp1(X,Tt1K(2,:),x) interp1(X,Tt1K(3,:),x)
    interp1(X,Tt1K(4,:),x) interp1(X,Tt1K(5,:),x)];
    TofY=interp1(Y,T,y);
    TwofY=Tout1+((Tin1-Tout1)/L^2)*y.^2;
    Temp_int=trapz(y,TwofY-TofY);
    Temp_int_sum=Temp_int_sum+Temp_int;
    n=n+1;

end

%plot(Y,Tt1K(:,4),y,TwofY)
h0=0;
Re0=0;
Nu0=0;
h1=qdot1/(pt*Temp_int_sum);
Re1=(DensMix(Tbulk1,0)*Dh)*(Q1)/(VisMix(Tbulk1,0)*At*3*N);
Nu1=(Dh*h1)/(CondMix(Tbulk1,0));

%Test 2
Tin2=((((116.8+120.4)/2)-32)/1.8)+273.15; %inlet temp in K
Tout2=((((86.5+86.2)/2)-32)/1.8)+273.15; %outlet temp in K
Tbulk2=(Tin2+Tout2)/2; %bulk temp K
Q2=.0001085; %vol flow rate (m^3/s)
mdot2=DensMix(Tbulk2,0)*Q2; %mass flow (kg/s)
qdot2=mdot2*Cpmix(Tbulk2,0)*(Tin2-Tout2); %measured heat transfer rate in W

Tt2=[80.1 83.4 79.5 80.4 78.4; %measured tube temps (F)
    77.8 84.4 83.3 82.7 78.6;
    81.7 90.6 84.8 87.2 80.0;
    91.9 99.7 96.4 95.3 82.4;
    98.3 112.6 113.8 109.2 99.0;];

Tt2K=((Tt2-32)/1.8)+273.15; %measured tube temps (K)

Temp_int_sum=0;
n=0;

while n<=(N-2);
    x=n*delta_x;

```



```

    T=[interp1(X,Tt2K(1,:),x) interp1(X,Tt2K(2,:),x) interp1(X,Tt2K(3,:),x)
    interp1(X,Tt2K(4,:),x) interp1(X,Tt2K(5,:),x)];
    TofY=interp1(Y,T,y);
    TwofY=Tout2+((Tin2-Tout2)/L^2)*y.^2;
    Temp_int=trapz(y,TwofY-TofY);
    Temp_int_sum=Temp_int_sum+Temp_int;
    n=n+1;

end
%plot(Y,Tt2K(:,3),y,TwofY)
h2=qdot1/(pt*Temp_int_sum);

Re2=(DensMix(Tbulk2,0)*Dh)*(Q2)/(VisMix(Tbulk2,0)*At*3*N);
Nu2=(Dh*h2)/(CondMix(Tbulk2,0));

%Test 3

Tin3=(((121.4+117.7)/2)-32)/1.8)+273.15; %inlet temp in K
Tout3=(((94.9+93.7)/2)-32)/1.8)+273.15; %outlet temp in K
Tbulk3=(Tin3+Tout3)/2; %bulk temp K
Q3=.000183; %vol flow rate (m^3/s)
mdot3=DensMix(Tbulk3,0)*Q3; %mass flow (kg/s)
qdot3=mdot3*Cpmix(Tbulk3,0)*(Tin3-Tout3); %measured heat transfer rate in W

Tt3=[83.6 88.6 85.0 89.3 83.2; %measured tube temps (F)
    81.6 92.4 91.7 90.5 83.3;
    85.1 95.6 94.3 94.7 85.8;
    95.0 102.2 97.8 100.2 84.9;
    100.7 111.2 113.4 109.2 99.2;];

Tt3K=((Tt3-32)/1.8)+273.15; %measured tube temps (K)

Temp_int_sum=0;
n=0;

while n<=(N-2);
    x=n*delta_x;
    T=[interp1(X,Tt3K(1,:),x) interp1(X,Tt3K(2,:),x) interp1(X,Tt3K(3,:),x)
    interp1(X,Tt3K(4,:),x) interp1(X,Tt3K(5,:),x)];
    TofY=interp1(Y,T,y);
    TwofY=Tout3+((Tin3-Tout3)/L^2)*y.^2;
    Temp_int=trapz(y,TwofY-TofY);
    Temp_int_sum=Temp_int_sum+Temp_int;
    n=n+1;
end

```

```

%plot(Y,Tt3K(:,5),y,TwofY)
h3=qdot3/(pt*Temp_int_sum);
Re3=(DensMix(Tbulk3,0)*Dh)*(Q3)/(VisMix(Tbulk3,0)*At*3*N);
Nu3=(Dh*h3)/(CondMix(Tbulk3,0));

%Test 4
Tin4=((((127.8+113.0)/2)-32)/1.8)+273.15; %inlet temp in K
Tout4=((((101.2+94.0)/2)-32)/1.8)+273.15; %outlet temp in K

Tbulk4=(Tin4+Tout4)/2; %bulk temp K
Q4=.0002467; %vol flow rate (m^3/s)
mdot4=DensMix(Tbulk4,0)*Q4; %mass flow (kg/s)
qdot4=mdot4*Cpmix(Tbulk4,0)*(Tin4-Tout4); %measured heat transfer rate in W

Tt4=[85.2 90.6 91.3 92.1 83.7; %measured tube temps (F)
      80.7 91.6 97.8 92.7 90.4;
      95.2 94.0 94.8 94.8 93.4;
      98.4 97.7 98.2 99.2 93.6;
      97.0 108.6 109.4 102.1 95.7;];

Tt4K=((Tt4-32)/1.8)+273.15; %measured tube temps (K)

Temp_int_sum=0;
n=0;

while n<=(N-2);
    x=n*delta_x;
    T=[interp1(X,Tt4K(1,:),x) interp1(X,Tt4K(2,:),x) interp1(X,Tt4K(3,:),x)
      interp1(X,Tt4K(4,:),x) interp1(X,Tt4K(5,:),x)];
    TofY=interp1(Y,T,y);
    TwofY=Tout4+((Tin4-Tout4)/L^2)*y.^2;
    Temp_int=trapz(y,TwofY-TofY);
    Temp_int_sum=Temp_int_sum+Temp_int;
    n=n+1;

end

%plot(Y,Tt4K(:,5),y,TwofY)
h4=qdot4/(pt*Temp_int_sum);
Re4=(DensMix(Tbulk4,0)*Dh)*(Q4)/(VisMix(Tbulk4,0)*At*3*N);
Nu4=(Dh*h4)/(CondMix(Tbulk4,0));

%Test 5
Tin5=((((125+116.2)/2)-32)/1.8)+273.15; %inlet temp in K

```

```

Tout5=((((103.4+99)/2)-32)/1.8)+273.15; %outlet temp in K
Tbulk5=(Tin5+Tout5)/2; %bulk temp K
Q5=.0002864; %vol flow rate (m^3/s)
mdot5=DensMix(Tbulk5,0)*Q5; %mass flow (kg/s)
qdot5=mdot5*Cpmix(Tbulk5,0)*(Tin5-Tout5); %measured heat transfer rate in W

Tt5=[88.4 90.4 89.2 94.5 85.4; %measured tube temps (F)
     85.2 94.0 98.0 95.5 84.3;
     88.6 95.4 99.9 100.9 95.8;
     101.8 102.9 104.6 104 87.5;
     106.9 112.2 112.9 108.4 96.8;];

Tt5K=((Tt5-32)/1.8)+273.15; %measured tube temps (K)

Temp_int_sum=0;
n=0;

while n<=(N-2);
    x=n*delta_x;
    T=[interp1(X,Tt5K(1,:),x) interp1(X,Tt5K(2,:),x) interp1(X,Tt5K(3,:),x)
    interp1(X,Tt5K(4,:),x) interp1(X,Tt5K(5,:),x)];
    TofY=interp1(Y,T,y);
    TwofY=Tout5+((Tin5-Tout5)/L^2)*y.^2;
    Temp_int=trapz(y,TwofY-TofY);
    Temp_int_sum=Temp_int_sum+Temp_int;
    n=n+1;

end

%plot(Y,Tt5K(:,5),y,TwofY)
h5=qdot5/(pt*Temp_int_sum);
Re5=(DensMix(Tbulk5,0)*Dh)*(Q5)/(VisMix(Tbulk5,0)*At^3*N);
Nu5=(Dh*h5)/(CondMix(Tbulk5,0));

%Test 6
Tin6=((((121.0+113.0)/2)-32)/1.8)+273.15; %inlet temp in K
Tout6=((((104.9+99.7)/2)-32)/1.8)+273.15; %outlet temp in K
Tbulk6=(Tin6+Tout6)/2; %bulk temp K
Q6=.0003508; %vol flow rate (m^3/s)
mdot6=DensMix(Tbulk6,0)*Q6; %mass flow (kg/s)
qdot6=mdot6*Cpmix(Tbulk6,0)*(Tin6-Tout6); %measured heat transfer rate in W

Tt6=[89.4 97.4 94.3 96.4 87.2; %measured tube temps (F)
     96.4 99.4 98.4 96.6 87.9;
     97.1 99.4 99.7 101.1 88.8;
     101.8 102.4 101.5 102.3 90.8;

```

```

106.3 110.3 110.6 106.7 99.0;];

Tt6K=((Tt6-32)/1.8)+273.15; %measured tube temps (K)
Temp_int_sum=0;
n=0;

while n<=(N-2);
    x=n*delta_x;
    T=[interp1(X,Tt6K(1,:),x) interp1(X,Tt6K(2,:),x) interp1(X,Tt6K(3,:),x)
    interp1(X,Tt6K(4,:),x) interp1(X,Tt6K(5,:),x)];
    TofY=interp1(Y,T,y);
    TwofY=Tout6+((Tin6-Tout6)/L^2)*y.^2;
    Temp_int=trapz(y,TwofY-TofY);
    Temp_int_sum=Temp_int_sum+Temp_int;
    n=n+1;

end

%plot(Y,Tt6K(:,5),y,TwofY)
h6=qdot6/(pt*Temp_int_sum);
Re6=(DensMix(Tbulk6,0)*Dh)*(Q6)/(VisMix(Tbulk6,0)*At*3*N);
Nu6=(Dh*h6)/(CondMix(Tbulk6,0));

%Test 7
Tin7=((((123.5+119.6)/2)-32)/1.8)+273.15 %inlet temp in K
Tout7=((((105.4+102.1)/2)-32)/1.8)+273.15 %outlet temp in K
Tbulk7=(Tin7+Tout7)/2; %bulk temp K
Q7=.000511; %vol flow rate (m^3/s)
mdot7=DensMix(Tbulk7,0)*Q7; %mass flow (kg/s)
qdot7=mdot7*Cpmix(Tbulk7,0)*(Tin7-Tout7); %measured heat transfer rate in W

Tt7=[77 93.6 92.6 94.5 83.5; %measured tube temps (F)
94.0 97.7 97.8 95.8 79.9;
94.2 94.5 100.2 99.2 81.1;
93.3 97.8 102.6 101.6 83.2;
111.1 112.3 115.7 110.2 97.5];

Tt7K=((Tt7-32)/1.8)+273.15; %measured tube temps (K)
Temp_int_sum=0;
n=0;

while n<=(N-2);
    x=n*delta_x;
    T=[interp1(X,Tt7K(1,:),x) interp1(X,Tt7K(2,:),x) interp1(X,Tt7K(3,:),x)
    interp1(X,Tt7K(4,:),x) interp1(X,Tt7K(5,:),x)];
    TofY=interp1(Y,T,y);

```

```

TwofY=Tout7+((Tin7-Tout7)/L^2)*y.^2;
Temp_int=trapz(y,TwofY-TofY);
Temp_int_sum=Temp_int_sum+Temp_int;
n=n+1;

end

%plot(Y,Tt7K(:,5),y,TwofY)
h7=qdot7/(pt*Temp_int_sum);
Re7=(DensMix(Tbulk7,0)*Dh)*(Q7)/(VisMix(Tbulk7,0)*At*3*N);
Nu7=(Dh*h7)/(CondMix(Tbulk7,0));

%Test 8
Tin8=((((123.1+116.4)/2)-32)/1.8)+273.15; %inlet temp in K
Tout8=((((110.5+106.3)/2)-32)/1.8)+273.15; %outlet temp in K
Tbulk8=(Tin8+Tout8)/2; %bulk temp K
Q8=.0007949; %vol flow rate (m^3/s)
mdot8=DensMix(Tbulk8,0)*Q8; %mass flow (kg/s)
qdot8=mdot8*Cpmix(Tbulk8,0)*(Tin8-Tout8); %measured heat transfer rate in W

Tt8=[97.1 99.0 97.6 97.5 84.3; %measured tube temps (F)
     97.3 98.5 101.3 95.3 86.6;
     99.8 99.4 101.2 104.8 85.8;
     103 105.2 106.0 107.0 85.5;
     109.7 112.3 112.8 110.5 97.7;];

Tt8K=((Tt8-32)/1.8)+273.15; %measured tube temps (K)
Temp_int_sum=0;
n=0;

while n<=(N-2);
    x=n*delta_x;
    T=[interp1(X,Tt8K(1,:),x) interp1(X,Tt8K(2,:),x) interp1(X,Tt8K(3,:),x)
    interp1(X,Tt8K(4,:),x) interp1(X,Tt8K(5,:),x)];
    TofY=interp1(Y,T,y);
    TwofY=Tout8+((Tin8-Tout8)/L^2)*y.^2;
    Temp_int=trapz(y,TwofY-TofY);
    Temp_int_sum=Temp_int_sum+Temp_int;
    n=n+1;

end

h8=qdot8/(pt*Temp_int_sum);
Re8=(DensMix(Tbulk8,0)*Dh)*(Q8)/(VisMix(Tbulk8,0)*At*3*N);
Nu8=(Dh*h8)/(CondMix(Tbulk8,0));
Re=0:1:1000;
Nu_Test=[Nu0 Nu1 Nu2 Nu3 Nu4 Nu5 Nu6 Nu7 Nu8]

```

```
Re_Test=[Re0 Re1 Re2 Re3 Re4 Re5 Re6 Re7 Re8]
```

```
plot(Re_Test,Nu_Test);
```

End of hal

Code for Program “haa”

```
%Program haa
%This program takes the numerical data collected and calculates the air
%side Nusselt Number as a function of Reynolds Number
Aff=.334; %free flow area of the air side of the core (m^2)
Lf=(3/8)*2.54/100; %Fin length (m)
L=24.5*2.54/100; %Core Length in meters
W=26.5*2.54/100; %Core Width in meters
d=2*2.54/100; %Core depth in meters
z=[0:.001:d];
X=[0 6.75 13.25 20 26.5]*(2.54/100);
Y=[0 6 12 18 24.5]*(2.54/100);
Yao=[0,12,24.5]*(2.54/100);
Fp=472.4; %number of fins per meter
%horizontal distance between vertical columns of fins
delta_x=.5*2.54/100;
%vertical distance between horizontal rows of fins
delta_y=(1/Fp);
%The data taken is as follows
ha0=0;
Nu0=0;
Re0=0;

%Test 1
Tamb1=((69.0-32)/1.8)+273.15; %ambient temp (K)
Tin1_L=(((113.0+111.6)/2)-32)/1.8+273.15; %inlet liquid temp in K
Tout1_L=(((99+98.9)/2)-32)/1.8+273.15; %outlet liquid temp in K
Tbulk1_L=(Tin1_L+Tout1_L)/2; %bulk liquid temp K
Q1_A_std=.181; %vol flow rate of air (m^3/s)(std)
Q1_L=.0001609; %vol flow rate of liquid (m^3/s)
mdot1_L=DensMix(Tbulk1_L,0)*Q1_L; %mass flow of liquid (kg/s)
qdot1=mdot1_L*Cpmix(Tbulk1_L,0)*(Tin1_L-Tout1_L); %measured heat transfer rate
in W

Tf1=[74.6 82.8 91.3 80.0 80.5; %measured fin temps (F)
```

```

73.5 88.1 96.3 84.5 85.8;
97.8 100.6 99.8 101.6 97.3;
102.1 106.8 105.9 102.8 98.9;
104.4 108.6 109.9 108.0 99.2;];

Tf1K=((Tf1-32)/1.8)+273.15; %measured fin temps (K)

%Outlet air temps
Ta_out1=[71,100.6,106.3];
Ta_out1K=((Ta_out1-32)/1.8)+273.15;
TbulkA1=(Tamb1+Ta_out1K(2))/2;
k=0;
y=k*delta_y;
Temp_int_sum=0;
Q1_A=Q1_A_std*(TbulkA1/293.15); %vol flow rate of air (m^3/s)

while y<(L-1*delta_y);
    y=k*delta_y;
    n=0;
    x=n*delta_x;
    while x < (W-.5*delta_x);
        x=n*delta_x;
        Ta=Tamb1+2*z*((interp1(Yao,Ta_out1K,y)-Tamb1)/d)-((interp1(Yao,Ta_out1K,y)-
Tamb1)/d^2)*z.^2;
        Tf_of_y=[interp1(X,Tf1K(1,:),x) interp1(X,Tf1K(2,:),x) interp1(X,Tf1K(3,:),x)
interp1(X,Tf1K(4,:),x) interp1(X,Tf1K(5,:),x)];
        Tf=interp1(Y,Tf_of_y,y);
        Temp_int=trapz(z,Tf-Ta);
        Temp_int_sum=Temp_int_sum+Temp_int;
        n=n+1;
    end

    k=k+1;
end
ha1=qdot1/(Lf* Temp_int_sum);
Nu1=ha1*d/CondAir(TbulkA1);
Re1=(DenseAir(TbulkA1)*(Q1_A/Aff)*d)/VisAir(TbulkA1);

%Test 2
Tamb2=((55.1-32)/1.8)+273.15; %ambient temp (K)
Tin2_L=(((116.8+116.0)/2)-32)/1.8)+273.15; %inlet liquid temp in K
Tout2_L=(((85.8+85.8)/2)-32)/1.8)+273.15; %outlet liquid temp in K
Tbulk2_L=(Tin2_L+Tout2_L)/2; %bulk liquid temp K
Q2_A_std=.491; %vol flow rate of air (m^3/s)(std)

```

```

Q2_L=.0001609; %vol flow rate of liquid (m^3/s)
mdot2_L=DensMix(Tbulk2_L,0)*Q2_L; %mass flow of liquid (kg/s)
qdot2=mdot2_L*Cpmix(Tbulk2_L,0)*(Tin2_L-Tout2_L); %measured heat transfer rate
in W

Tf2=[78.8 81.8 77.6 71.6 66.0; %measured fin temps (F)
    80.7 82.4 80.6 75.5 68.8;
    86.0 96.0 92.0 87.6 73.0;
    94.4 99.0 102.3 91.9 78.5;
    105.5 112.6 112.5 104.6 89.0;];

Tf2K=((Tf2-32)/1.8)+273.15; %measured fin temps (K)
%Outlet air temps

Ta_out2=[58.3,90.3,100.1];
Ta_out2K=((Ta_out2-32)/1.8)+273.15;
TbulkA2=(Tamb2+Ta_out2K(2))/2;
k=0;
y=k*delta_y;
Temp_int_sum=0;
Q2_A=Q2_A_std*(TbulkA2/293.15); %vol flow rate of air (m^3/s)

while y<(L-1*delta_y);
    y=k*delta_y;
    n=0;
    x=n*delta_x;
    while x < (W-.5*delta_x);
        x=n*delta_x;
        Ta=Tamb2+2*z*((interp1(Yao,Ta_out2K,y)-Tamb2)/d)-((interp1(Yao,Ta_out2K,y)-
Tamb2)/d^2)*z.^2;
        Tf_of_y=[interp1(X,Tf2K(1,:),x) interp1(X,Tf2K(2,:),x) interp1(X,Tf2K(3,:),x)
interp1(X,Tf2K(4,:),x) interp1(X,Tf2K(5,:),x)];
        Tf=interp1(Y,Tf_of_y,y);
        Temp_int=trapz(z,Tf-Ta);
        Temp_int_sum=Temp_int_sum+Temp_int;
        n=n+1;
    end

    k=k+1;
end
ha2=qdot2/(Lf* Temp_int_sum);
Nu2=ha2*d/CondAir(TbulkA2);
Re2=(DenseAir(TbulkA2)*(Q2_A/Aff)*d)/VisAir(TbulkA2);

```



```

%Test 3
Tamb3=((51.6-32)/1.8)+273.15; %ambient temp (K)
Tin3_L=(((120.0+118.8)/2)-32)/1.8)+273.15; %inlet liquid temp in K
Tout3_L=(((74.2+78.2)/2)-32)/1.8)+273.15; %outlet liquid temp in K
Tbulk3_L=(Tin3_L+Tout3_L)/2; %bulk liquid temp K
Q3_A_std=.706; %vol flow rate of air (m^3/s)(std)
Q3_L=.0001609; %vol flow rate of liquid (m^3/s)
mdot3_L=DensMix(Tbulk3_L,0)*Q3_L; %mass flow of liquid (kg/s)

qdot3=mdot3_L*Cpmix(Tbulk3_L,0)*(Tin3_L-Tout3_L); %measured heat transfer rate
in W

Tf3=[59.6 66.2 66.6 61.8 58.4; %measured fin temps (F)
    74.9 76.9 75.9 65.9 62.2;
    79.0 87.0 77.8 78.4 66.3;
    79.9 93.4 86.8 87.2 68.7;
    87.0 109.0 109.7 109.0 90.0;];

Tf3K=((Tf3-32)/1.8)+273.15; %measured fin temps (K)

%Outlet air temps

Ta_out3=[53.9,77.5,88.3];
Ta_out3K=((Ta_out3-32)/1.8)+273.15;
TbulkA3=(Tamb3+Ta_out3K(2))/2;
k=0;
y=k*delta_y;
Temp_int_sum=0;
Q3_A=Q3_A_std*(TbulkA3/293.15); %vol flow rate of air (m^3/s)

while y<(L-1*delta_y);
    y=k*delta_y;
    n=0;
    x=n*delta_x;
    while x < (W-.5*delta_x);
        x=n*delta_x;
        Ta=Tamb3+2*z*((interp1(Yao,Ta_out3K,y)-Tamb3)/d)-((interp1(Yao,Ta_out3K,y)-
        Tamb3)/d^2)*z.^2;
        Tf_of_y=[interp1(X,Tf3K(1,:),x) interp1(X,Tf3K(2,:),x) interp1(X,Tf3K(3,:),x)
        interp1(X,Tf3K(4,:),x) interp1(X,Tf3K(5,:),x)];
        Tf=interp1(Y,Tf_of_y,y);
        Temp_int=trapz(z,Tf-Ta);
        Temp_int_sum=Temp_int_sum+Temp_int;
        n=n+1;
    end
end

```

```

k=k+1;
end

ha3=qdot3/(Lf* Temp_int_sum);
Nu3=ha3*d/CondAir(TbulkA3);
Re3=(DenseAir(TbulkA3)*(Q3_A/Aff)*d)/VisAir(TbulkA3);

%Test 4
Tamb4=((51.2-32)/1.8)+273.15; %ambient temp (K)
Tin4_L=(((118.6+116.8)/2)-32)/1.8)+273.15; %inlet liquid temp in K
Tout4_L=(((70.7+72.4)/2)-32)/1.8)+273.15; %outlet liquid temp in K
Tbulk4_L=(Tin4_L+Tout4_L)/2; %bulk liquid temp K
Q4_A_std=1.052; %vol flow rate of air (m^3/s)(std)
Q4_L=.0001609; %vol flow rate of liquid (m^3/s)
mdot4_L=DensMix(Tbulk4_L,0)*Q4_L; %mass flow of liquid (kg/s)
qdot4=mdot4_L*Cpmix(Tbulk4_L,0)*(Tin4_L-Tout4_L); %measured heat transfer rate
in W

Tf4=[60.7 67.8 64.2 62.0 58.8; %measured fin temps (F)
     68.2 69.2 68.4 66.2 60.0;
     72.6 74.9 72.2 68.0 66.3;
     81.1 84.8 84.5 80.0 68.7;
     94.1 103.4 102.4 94.9 73.6;];

Tf4K=((Tf4-32)/1.8)+273.15; %measured fin temps (K)
%Outlet air temps
Ta_out4=[54.9,71.6,79.2];
Ta_out4K=((Ta_out4-32)/1.8)+273.15;
TbulkA4=(Tamb4+Ta_out4K(2))/2;
k=0;
y=k*delta_y;
Temp_int_sum=0;
Q4_A=Q4_A_std*(TbulkA4/293.15); %vol flow rate of air (m^3/s)

while y<(L-1*delta_y);
    y=k*delta_y;
    n=0;
    x=n*delta_x;
    while x < (W-.5*delta_x);
        x=n*delta_x;
        Ta=Tamb4+2*z*((interp1(Yao,Ta_out4K,y)-Tamb4)/d)-((interp1(Yao,Ta_out4K,y)-
Tamb4)/d^2)*z.^2;
        Tf_of_y=[interp1(X,Tf4K(1,:),x) interp1(X,Tf4K(2,:),x) interp1(X,Tf4K(3,:),x)
interp1(X,Tf4K(4,:),x) interp1(X,Tf4K(5,:),x)];
        Tf=interp1(Y,Tf_of_y,y);
    end
end

```

```

    Temp_int=trapz(z,Tf-Ta);
    Temp_int_sum=Temp_int_sum+Temp_int;
    n=n+1;
end

k=k+1;
end

ha4=qdot4/(Lf* Temp_int_sum);
Nu4=ha4*d/CondAir(TbulkA4);
Re4=(DenseAir(TbulkA4)*(Q4_A/Aff)*d)/VisAir(TbulkA4);

%Test 5
Tamb5=((45.3-32)/1.8)+273.15; %ambient temp (K)
Tin5_L=(((126.4+124.0)/2)-32)/1.8)+273.15; %inlet liquid temp in K
Tout5_L=(((64.9+64.4)/2)-32)/1.8)+273.15; %outlet liquid temp in K
Tbulk5_L=(Tin5_L+Tout5_L)/2; %bulk liquid temp K
Q5_A_std=1.557; %vol flow rate of air (m^3/s)(std)
Q5_L=.0001609; %vol flow rate of liquid (m^3/s)
mdot5_L=DensMix(Tbulk5_L,0)*Q5_L; %mass flow of liquid (kg/s)
qdot5=mdot5_L*Cpmix(Tbulk5_L,0)*(Tin5_L-Tout5_L); %measured heat transfer rate
in W

Tf5=[53.4 62.5 57.8 53.5 50.1; %measured fin temps (F)
    62.8 63.5 62.2 56.3 52.3;
    66.0 68.3 62.7 60.2 59.2;
    70.7 73.6 74.6 71.4 70.9;
    80.6 100.4 103.4 98.1 76.9;];

Tf5K=((Tf5-32)/1.8)+273.15; %measured fin temps (K)

%Outlet air temps
Ta_out5=[45.5,63.2,71.8];
Ta_out5K=((Ta_out5-32)/1.8)+273.15;
TbulkA5=(Tamb5+Ta_out5K(2))/2;
k=0;
y=k*delta_y;
Temp_int_sum=0;
Q5_A=Q5_A_std*(TbulkA5/293.15); %vol flow rate of air (m^3/s)

while y<(L-1*delta_y);
    y=k*delta_y;
    n=0;
    x=n*delta_x;
    while x < (W-.5*delta_x);

```

```

    x=n*delta_x;
    Ta=Tamb5+2*z*((interp1(Yao,Ta_out5K,y)-Tamb5)/d)-((interp1(Yao,Ta_out5K,y)-
Tamb5)/d^2)*z.^2;
    Tf_of_y=[interp1(X,Tf5K(1,:),x) interp1(X,Tf5K(2,:),x) interp1(X,Tf5K(3,:),x)
interp1(X,Tf5K(4,:),x) interp1(X,Tf5K(5,:),x)];
    Tf=interp1(Y,Tf_of_y,y);
    Temp_int=trapz(z,Tf-Ta);
    Temp_int_sum=Temp_int_sum+Temp_int;
    n=n+1;
end

```

```

k=k+1;
end
ha5=qdot5/(Lf* Temp_int_sum);
Nu5=ha5*d/CondAir(TbulkA5);
Re5=(DenseAir(TbulkA5)*(Q5_A/Aff)*d)/VisAir(TbulkA5);

```

```

%Test 6
Tamb6=((52.6-32)/1.8)+273.15; %ambient temp (K)
Tin6_L=((((118.2+114)/2)-32)/1.8)+273.15; %inlet liquid temp in K
Tout6_L=((((65.5+66.6)/2)-32)/1.8)+273.15; %outlet liquid temp in K
Tbulk6_L=(Tin6_L+Tout6_L)/2; %bulk liquid temp K
Q6_A_std=1.77; %vol flow rate of air (m^3/s)(std)
Q6_L=.0001609; %vol flow rate of liquid (m^3/s)
mdot6_L=DensMix(Tbulk5_L,0)*Q5_L; %mass flow of liquid (kg/s)
qdot6=mdot6_L*Cpmix(Tbulk6_L,0)*(Tin6_L-Tout6_L); %measured heat transfer rate
in W

```

```

Tf6=[58.7 63.2 60.4 58.8 57.0; %measured fin temps (F)
    64.8 65.1 64.2 61.6 57.5;
    66.9 68.8 64.9 63.5 59.2;
    69.3 73.2 70.8 71.3 60.6;
    91.4 96.4 96.6 85.4 76.8;];

```

```

Tf6K=((Tf6-32)/1.8)+273.15; %measured fin temps (K)

```

```

%Outlet air temps

```

```

Ta_out6=[54.7,63.6,71.7];
Ta_out6K=((Ta_out6-32)/1.8)+273.15;
TbulkA6=(Tamb6+Ta_out6K(2))/2;
k=0;
y=k*delta_y;
Temp_int_sum=0;
Q6_A=Q6_A_std*(TbulkA6/293.15); %vol flow rate of air (m^3/s)

```

```

while y<(L-1*delta_y);
    y=k*delta_y;
    n=0;
    x=n*delta_x;
    while x < (W-.5*delta_x);
        x=n*delta_x;
        Ta=Tamb6+2*z*((interp1(Yao,Ta_out6K,y)-Tamb6)/d)-((interp1(Yao,Ta_out6K,y)-
Tamb6)/d^2)*z.^2;
        Tf_of_y=[interp1(X,Tf6K(1,:),x) interp1(X,Tf6K(2,:),x) interp1(X,Tf6K(3,:),x)
interp1(X,Tf6K(4,:),x) interp1(X,Tf6K(5,:),x)];
        Tf=interp1(Y,Tf_of_y,y);
        Temp_int=trapz(z,Tf-Ta);
        Temp_int_sum=Temp_int_sum+Temp_int;
        n=n+1;
    end
end

k=k+1;
end

ha6=qdot6/(Lf* Temp_int_sum);
Nu6=ha6*d/CondAir(TbulkA6);
Re6=(DenseAir(TbulkA6)*(Q6_A/Aff)*d)/VisAir(TbulkA6);
NU=[Nu0,Nu1,Nu2,Nu3,Nu4,Nu5,Nu6]
RE=[Re0,Re1,Re2,Re3,Re4,Re5,Re6]

plot(RE,NU)

```

End of haa

Code for Function “module”

```

function module=f(Q_L,TL_in,mix,qsol,nc)

clear clc;
% This function takes the following arguments
%Q_L=Volume flow rate of liquid (m^3/s)
%TL_in=Cell inlet water temp K
%mix=volume fraction of coolant if aqueous solution is used
%qsol=direct normal solar flux W/m^2 at receiver
%nc=# channels per module

%It outputs
%cell temp K
%electrical output W
%outlet temp (K)

% Cell module geometry
Cd=3/1000; % Channel depth (m)
ft=1/1000; % Fin thickness of inner fins(m)
fto=2/1000; % Fin thickness of outer fins(m)
wm=5/100; %module width (m)
Lm=54/100; % length of 2 modules (m)
wc=(wm-2*fto-(nc-1)*ft)/nc; %Channel width
Dhc=(4*Cd*wc)/(2*(Cd+wc)); % hydraulic diameter
nm=12; %#modules
td=.00008; %Thickness of dielectric (m)o
kd=2.2; %Thermal conductivity of dielectric (W/mK)
tcop=(1/8)*(2.54/100);%Thickness of copper (m)
kcop=401; %Thermal conductivity of copper
r=wc/Cd; %ratio of channel width and depth

CellEff=[.25 .2448 .223 .2064 .2016 .1937 .1920 .1776 .1749 .1642 .1550 .147 .14 .125
.111 .085 .063 .036 .021 0];

TCellEff=[-5 8.6 11.6 17.2 22.6 34.5 50.3 58.4 68.5 76.5 82.3 85 87 90 92 95 97 99 100
400]+273.15;

V=Q_L/(nm*nc*wc*Cd);
delta_x=.5*2.54/100;
n=1;
TL(1)=TL_in;
mdot=(Q_L/(nm*nc))*DensMix(TL(1),mix);

Tcell_guess=TL_in+10;

```

```

eff_guess=interp1(TCellEff,CellEff,Tcell_guess);
q=(qsol-eff_guess*qsol)*((wc+ft)*delta_x);

while n*delta_x<=Lm
    qdiff=10;
    Re(n)=((DensMix(TL(n),mix))*Dhc*(Q_L/(nm*nc*wc*Cd)))/VisMix(TL(n),mix);
    Pr(n)=VisMix(TL(n),mix)*Cpmix(TL(n),mix)/CondMix(TL(n),mix);
    if Re(n)<=3000

        if r <=1
            F(n)=57/Re(n);
            Nu(n)=3.61;

        elseif 1<r<=1.43
            F(n)=59/Re(n);
            Nu(n)=3.73;
        elseif 1.43<r<=2
            F(n)=62/Re(n);
            Nu(n)=4.12;
        elseif 2<r<=3
            F(n)=69/Re(n);
            Nu(n)=4.79;
        elseif 3<r<=4
            F(n)=73/Re(n);
            Nu(n)=5.33;
        elseif r>4
            F(n)=82/Re(n);
            Nu(n)=6.49;

        end

    else

        F(n)=.316*(Re(n)^(-.25));
        Nu(n)=((F(n)/8)*(Re(n)-1000)*Pr(n))/(1+(12.7*(F(n)/8)^.5)*((Pr(n)^.667)-1));

    end

    cf(n)=F(n)/4;
    h(n)=Nu(n)*(CondMix(TL(n),mix))/Dhc;

    while qdiff>.001
        Tcop_in=(q+(h(n)*wc*delta_x*TL(n))+(h(n)*Cd*delta_x*TL(n)))/(h(n)*delta_x*(wc+C
        d));
    end
end

```

```

Tcop_out=(q*tcop/(kcop*wc*delta_x))+Tcop_in;
Tcell=(q*td/(kd*wc*delta_x))+Tcop_out;
ef=interp1(TCellEff,CelEff,Tcell);
qnew=(qsol-ef*qsol)*((wc+ft)*delta_x);
qdiff=abs(qnew-q);
q=qnew;

end

eff(n)=ef;
Tcopper_in(n)=Tcop_in;
Tcopper_out(n)= Tcop_out;
T_cell(n)= Tcell;
q_net(n)=q;
El(n)=qsol*eff(n)*((wc+ft)*delta_x);
delta_P(n)=(DensMix(TL(n),mix)*(V^2)*cf(n)*(delta_x*2*(Cd+wc)))/(2*Cd*wc);
TL(n+1)=TL(n)+(q_net(n)/(mdot*Cpmix(TL(n),mix)));
n=n+1;
end
edot=nc*nm*sum(El);
delta_P_total=sum(delta_P);
Wp=Q_L*delta_P_total;
q_in=qsol*Lm*wm*nm;
module=[delta_P_total,Wp,edot,TL(n-1),T_cell(n-1),q_in];

```

End Of Module

Code for Function “rad_SI”

```

function rad_SI=f(Q_A_std,QL,mix,Tamb,TL_in,nr)

%This function uses empirical data to predict the outlet liquid temperature
%for the radiators used in this project.
%It takes as arguments
%Q_A=volume flow rate of air (m^3/s) std
%Q_L=volume flow rate of liquid (m^3/s)
%mix=volume fraction of glycol
%Tamb=Ambient air temp (K)
%TL_in=Inlet liquid temp (K)
%nr=number of radiators
%Radiator geometry

L=24.5*2.54/100; %Core Length in meters
W=26.5*2.54/100; %Core Width in meters
d=2*2.54/100; %Core depth in meters
N=55; %number of tube banks
At=.00002196; %Tube cross sectional area (m^2)
Aw=.016; %Tube wall area (m^2)
pt=.026; % The tube perimeter is (m)
Dh=.003372; %The tube hydraulic diameter is (m)
Aff=.334; %free flow area of the air side of the core (m^2)
Lf=(3/8)*2.54/100; %Fin length (m)
Fp=472.4; %number of fins per meter

%Empirical relation for liquid side

ReL_em=1000*[0 0.0774 0.1736 0.3054 0.4197 0.4959 0.6004 0.8993
1.4175 ];
NuL_em=[0 7.2085 7.3008 13.6705 14.1779 14.0856 15.9166 17.8016
19.3592];

%Empirical relation for air side
Nua_em=1000*[0 0.23811 0.42930 0.62400 0.77014 0.95720 1.11964];
Rea_em=10000*[0 0.16816 0.46270 0.67163 1.00442 1.49878 1.69634];

Tbulk_a_diff=2;
Tbulk_a=Tamb;

Tbulk_L_diff=2;
Tbulk_L=TL_in;
Q_A=Q_A_std/nr;
Q_L=QL/nr;

```

Kav=37.02;

```
while abs(Tbulk_L_diff)>=1 | abs(Tbulk_a_diff)>=1
ReL=(DensMix(Tbulk_L,mix)*Dh)*(Q_L)/(VisMix(Tbulk_L,mix)*At*3*N);
```

```
NuL=(2.5125*log(ReL))-1.001; %best fit line for NuL_em
```

```
Rea=(DenseAir(Tbulk_a)*(Q_A/Aff)*d)/VisAir(Tbulk_a);
```

```
Nua=interp1(Rea_em,Nua_em,Rea);
```

```
ha_a=(Nua/d)*CondAir(Tbulk_a)*N*Fp*L*Lf*d;
```

```
ha_L=(NuL/Dh)*CondMix(Tbulk_L,mix)*N*L*3*pt;
```

```
UA_inv=(1/ha_a)+(1/ha_L);
```

```
UA=1/UA_inv;
```

```
Ch=Cpmix(Tbulk_L,mix)*DensMix(Tbulk_L,mix)*(Q_L);
```

```
Cc=CpAir(Tbulk_a)*DenseAir(Tbulk_a)*(Q_A);
```

```
if Ch >= Cc
```

```
    Cmax=Ch;
```

```
    Cmin=Cc;
```

```
else
```

```
    Cmax=Cc;
```

```
    Cmin=Ch;
```

```
end
```

```
NTU=UA/Cmin;
```

```
Cr=Cmin/Cmax;
```

```
VAR1=exp(-Cr*NTU^.78)-1;
```

```
VAR2=(1/Cr)*NTU^.22;
```

```
Eff=1-exp(VAR1*VAR2);
```

```
TL_out=(Eff*Cmin*(TL_in-Tamb)-(Ch*TL_in))/(-Ch);
```

```
qdot=Eff*Cmin*(TL_in-Tamb);
```

```
Ta_out=((Eff*Cmin*(TL_in-Tamb))+Cc*Tamb)/Cc;
```

```
Tbulk_a_new=(Tamb+Ta_out)/2;
```

```
Tbulk_a_diff=(Tbulk_a-Tbulk_a_new);
```

```
Tbulk_a=Tbulk_a_new;
```

```
Tbulk_L_new=(TL_in+TL_out)/2;
```

```
Tbulk_L_diff=(Tbulk_L-Tbulk_L_new);
```

```
Tbulk_L=Tbulk_L_new;
```

```
Q_A=Q_A_std*(Tbulk_a/293.15);
```

```
end
```

```
qdot_total=qdot*nr;
```

```
%Empirical relation for radiator air side (frictional)
```

```

CFA=[ 0.831 0.780 0.580 0.491 0.423 0.393 0.360 0.332 0.320 0.283 0.279 0.280 0.284
0.272 0.262 0.255 0.241 0.236 0.228 0.219 0.213 0.207 0.201 0.195 0.190 0.191 0.186
0.185 0.183 0.179 0.178 0.176 0.174 0.171];
RE_CFA=[ 1.399e3 1.667e3 2.161e3 2.779e3 3.519e3 4.012e3 4.751e3 5.267e3 5.917e3
6.970e3 7.463e3 7.822e3 8.517e3 9.144e3 1.013e4 1.080e4 1.161e4 1.255e4 1.340e4
1.434e4 1.520e4 1.596e4 1.694e4 1.784e4 1.838e4 1.865e4 1.910e4 1.954e4 2.004e4
2.057e4 2.089e4 2.125e4 2.205e4 2.286e4];

```

```

if ReL<=2300
    Cfl=Kav/ReL;
else
    Cfl=(.316/4)*(ReL^(-.25));
end

```

```

Cfa=interp1(RE_CFA,CFA,Rea);
delta_P_L=Aw*(DensMix(Tbulk_L,mix))*Cfl*((Q_L/(At*3*N))^2)/(2*At);
delta_P_a=(d*Lf*Fp*L*N*(DenseAir(Tbulk_a))*Cfa*((Q_A/Aff)^2)/(2*Aff);
Wp=Q_L*nr*delta_P_L;
Wf=Q_A*nr*delta_P_a;
rad_SI=[TL_out,Tbulk_a,delta_P_L,delta_P_a,Wp,Wf];

```

End Of rad_SI

Code for Function “PV”

```

function PV=f(x,q_norm,Tamb,nc)

%This function determines the steady state efficiency of the entire system

%It takes as arguments
%Q_A=volume flow rate of air (m^3/s) std
%Q_L=volume flow rate of liquid (m^3/s)
%nr=number of radiators in paralell
%nc=# channels per module
mix=0;
eta_f=.7;
eta_p=.7;
Q_A_std=x(1);
Q_L=x(2);
nr=x(3);

CR=250; %concentration ratio of collector

qsol=CR*q_norm;

TL_in_guess=315; %TL_in_guess=Initial guess for inlet liquid temp to module (K)
TL_in_diff=10;
TL_in=TL_in_guess;

while TL_in_diff>=.1
    X=module(Q_L,TL_in,mix,qsol,nc);
    TL_out=X(4);
    Y=rad_SI(Q_A_std,Q_L,mix,Tamb,TL_out,nr);
    TL_in_diff=abs(TL_in-Y(1));
    TL_in=Y(1);
end

Wp=(X(2)+Y(5))/eta_p;
Wf=Y(6)/eta_f;
El=X(3);
qin=X(6);
T_cell_max=X(5);
q_net=El-Wp-Wf;
Eff_sys=q_net/qin;
delta_P_L=X(1)+Y(3);
delta_P_A=Y(4);
PV=[El,Wp,Wf,Eff_sys,T_cell_max,delta_P_L,delta_P_A];

End Of PV

```

Code for Function “Eff_inv”

```
function Eff_inv=f(x,q_norm,Tamb,nc)
```

```
Y=PV(x,q_norm,Tamb,nc);
```

```
if Y(4)<=0
```

```
    B=1000000;
```

```
else
```

```
    B=1/Y(4);
```

```
end
```

```
Eff_inv=B;
```

```
End Of Eff_inv
```

Code for Function “Opt_PV”

```

%Program Opt_PV
% This program determines the values of the inputs to PV which produce the
% maximum efficiency
%clear all;
%load start
warning off
i=1;

q_norm=DNI;
Tamb=(T_ambient/10)+273.15;
f=size(DNI);
set={[],[],[1:1:4]};
lb = [.2,.0015,1]'; % Set lower bounds
ub = [1.7,.007,4]'; % Set upper bounds
x_guess=[.8,.0035,3]';
nc=3;

while i<=f(1)
    if q_norm(i)>200

%The parameters being varied are Air flow rate, Liquid flow rate, number of
%radiators, number of channels per module
%    [Q_A,Q_L,nr]

x_guess=[findone(Qa,'last'),findone(QL,'last'),findone(Num_rad,'last')];

options = optimset('LargeScale','off','Display','off','MaxFunEvals',5000,'TolX',.0001);
x = fminconset('Eff_inv',x_guess,[],[],[],[],lb,ub,[],options,set,[],q_norm(i),Tamb(i),nc);

B=PV(x,q_norm(i),Tamb(i),nc);

%disp(['Volume flow rate of air (m^3/s) = ', num2str(x(1))]);
%disp(['Volume flow rate of liquid (m^3/s) = ', num2str(x(2))]);
%disp(['Number of radiators = ', num2str(round(x(3)))]);
%disp(['Electrical output (W) = ', num2str(B(1))]);
%disp(['Pump power (W) = ', num2str(B(2))]);
%disp(['Fan power (W) = ', num2str(B(3))]);
%disp(['System efficiency % = ', num2str(B(4)*100)]);

```

```

%disp(['Max cell temperature (K) = ', num2str(B(5))]);
%disp(['Air side pressure drop (Pa) = ', num2str(B(7))]);
%disp(['Liquid side pressure drop (Pa) = ', num2str(B(6))]);

```

```

else

```

```

    x(1)=0;
    x(2)=0;
    x(3)=0;
    B(1)=0;
    B(2)=0;
    B(3)=0;
    B(4)=0;
    B(5)=Tamb(i);
    B(6)=0;
    B(7)=0;

```

```

end

```

```

    Qa(i)=x(1);
    QL(i)=x(2);
    El(i)=B(1);
    Wp(i)=B(2);
    Wf(i)=B(3);
    Delta_P_A(i)=B(7);
    Delta_P_L(i)=B(6);
    Eff(i)=100*B(4);
    Par(i)=Wp(i)+Wf(i);
    Tcell_max(i)=B(5);
    Num_rad(i)=round(x(3));
    net=El(i)-Wp(i)-Wf(i);

```

```

i=i+1

```

```

end

```

End Of Opt_PV

APPENDIX C

EXPERIMENTAL DATA

Liquid side

Liquid side		Test #			
		1			
Ambient temp (F)	Fan speed (Hz)	Air flow rate (cfm std)			
73.1	30.2	1920			
Liquid flow rate (gpm)	Inlet water temp start (F)	Inlet water temp end (F)	Outlet water temp start (F)	Outlet water temp end (F)	
0.79	123.3	119.8	77.1	77.2	
Tube Temps (F)					
Y (0= Top)	X=-13.25 in (Left edge)	X=-6.75 (in)	X=0 (center)	X=6.75 (in)	X=13.25 in (Right edge)
0 (in)	74.5	76.8	73.9	73.8	73.6
6 (in)	75.4	79	76.7	74.5	73.8
12 (in)	76	85.8	79	79.2	74.4
18 (in)	87.5	95.7	89.3	88.6	77.8
24 (in)	99.1	105.2	113	112.6	95.6

Liquid side		Test #			
		2			
Ambient temp (F)	Fan speed (Hz)	Air flow rate (cfm std)			
78	30.2	1855			
Liquid flow rate (gpm)	Inlet water temp start (F)	Inlet water temp end (F)	Outlet water temp start (F)	Outlet water temp end (F)	
1.72	116.8	120.4	86.2	86.5	
Tube Temps (F)					
Y (0= Top)	X=-13.25 in (Left edge)	X=-6.75 (in)	X=0 (center)	X=6.75 (in)	X=-13.25 in (Right edge)
0 (in)	80.1	83.4	79.5	80.4	78.4
6 (in)	77.8	84.4	83.3	82.7	78.6
12 (in)	81.7	90.6	84.8	87.2	80
18 (in)	91.9	99.7	96.4	95.3	82.4
24 (in)	98.3	112.6	112.8	109.2	99

Liquid side		Test #			
		3			
Ambient temp (F)	Fan speed (Hz)	Air flow rate (cfm std)			
76.3	30.1	1960			
Liquid flow rate (gpm)	Inlet water temp start (F)	Inlet water temp end (F)	Outlet water temp start (F)	Outlet water temp end (F)	
2.9	121.4	117.7	94.9	93.7	
Tube Temps (F)					
Y (0= Top)	X=-13.25 in (Left edge)	X=-6.75 (in)	X=0 (center)	X=6.75 (in)	X=-13.25 in (Right edge)
0 (in)	83.6	88.6	85	89.3	83.2
6 (in)	81.6	92.4	91.7	90.5	83.3
12 (in)	85.1	95.6	94.3	94.7	85.8
18 (in)	95	102.2	97.8	100.2	84.9
24 (in)	100.7	111.2	113.4	109.2	99.2

Liquid side		Test #			
		4			
Ambient temp (F)	Fan speed (Hz)	Air flow rate (cfm std)			
76.4	30.3	1915			
Liquid flow rate (gpm)	Inlet water temp start (F)	Inlet water temp end (F)	Outlet water temp start (F)	Outlet water temp end (F)	
3.91	127.8	113	101.2	94	
Tube Temps (F)					
Y (0= Top)	X=-13.25 in (Left edge)	X=-6.75 (in)	X=0 (center)	X=6.75 (in)	X=-13.25 in (Right edge)
0 (in)	85.2	90.6	91.3	92.1	83.7
6 (in)	80.7	91.6	97.8	92.7	90.4
12 (in)	95.2	94	94.8	94.8	93.4
18 (in)	98.4	97.7	98.2	99.7	93.6
24 (in)	97	108.6	109.4	102.1	95.7

Liquid side		Test #			
		5			
Ambient temp (F)	Fan speed (Hz)	Air flow rate (cfm std)			
79	30.3	1905			
Liquid flow rate (gpm)	Inlet water temp start (F)	Inlet water temp end (F)	Outlet water temp start (F)	Outlet water temp end (F)	
4.54	125	116.2	103.4	99	
Tube Temps (F)					
Y (0= Top)	X=-13.25 in (Left edge)	X=-6.75 (in)	X=0 (center)	X=6.75 (in)	X=-13.25 in (Right edge)
0 (in)	88.4	90.4	89.2	94.5	85.4
6 (in)	85.2	94	98	95.5	84.3
12 (in)	88.6	95.4	99.9	100.9	95.8
18 (in)	101.8	102.9	104.6	104	87.5
24 (in)	106.9	112.2	112.9	108.4	96.8

Liquid side		Test #			
		6			
Ambient temp (F)	Fan speed (Hz)	Air flow rate (cfm std)			
75.9	30.3	1970			
Liquid flow rate (gpm)	Inlet water temp start (F)	Inlet water temp end (F)	Outlet water temp start (F)	Outlet water temp end (F)	
5.56	121	113	104.9	99.7	
Tube Temps (F)					
Y (0= Top)	X=-13.25 in (Left edge)	X=-6.75 (in)	X=0 (center)	X=6.75 (in)	X=-13.25 in (Right edge)
0 (in)	89.4	97.4	94.3	96.4	87.2
6 (in)	96.4	99.4	98.4	96.6	87.9
12 (in)	97.1	99.4	99.7	101.1	88.8
18 (in)	101.8	102.4	101.5	102.3	90.8
24 (in)	106.3	110.3	110.6	106.7	99

Liquid side		Test #			
		7			
Ambient temp (F)	Fan speed (Hz)	Air flow rate (cfm std)			
62.5	30.3	1710			
Liquid flow rate (gpm)	Inlet water temp start (F)	Inlet water temp end (F)	Outlet water temp start (F)	Outlet water temp end (F)	
8.1	123.5	119.6	105.4	102.1	
Tube Temps (F)					
Y (0= Top)	X=-13.25 in (Left edge)	X=-6.75 (in)	X=0 (center)	X=6.75 (in)	X=-13.25 in (Right edge)
0 (in)	77	93.6	92.6	94.5	83.5
6 (in)	94.3	97.7	97.8	95.8	79.9
12 (in)	94.2	94.5	100.2	99.2	81.1
18 (in)	93.3	97.8	102.6	101.6	83.2
24 (in)	111.1	112.3	115.7	110.2	97.5

Liquid side		Test #			
		8			
Ambient temp (F)	Fan speed (Hz)	Air flow rate (cfm std)			
62.1	30.3	1705			
Liquid flow rate (gpm)	Inlet water temp start (F)	Inlet water temp end (F)	Outlet water temp start (F)	Outlet water temp end (F)	
12.6	123.1	116.4	110.5	106.3	
Tube Temps (F)					
Y (0= Top)	X=-13.25 In (Left edge)	X=-6.75 (in)	X=0 (center)	X=6.75 (in)	X=-13.25 In (Right edge)
0 (in)	97.1	93	97.6	97.5	89.3
6 (in)	97.3	98.5	101.3	95.3	86.6
12 (in)	99.8	99.4	101..2	104.8	85.8
18 (in)	103	105.2	106	107	85.5
24 (in)	109.7	112.3	112.8	110.5	97.7

Air side

Air side		Test #			
		1			
Air flow rate (cfm std)	Fan speed (Hz)	Ambient temp (F)			
384	9.8	69			
Liquid flow rate (gpm)	Inlet water temp start (F)	Inlet water temp end (F)	Outlet water temp start (F)	Outlet water temp end (F)	
2.55	113	111.6	99	98.9	
			Base Temps (F)		
Y (0= Top)	X=-13.25 in (Left edge)	X=-6.75 (in)	X=0 (center)	X=6.75 (in)	X=-13.25 in (Right edge)
0 (in)	74.6	82.8	91.3	80	80.5
6 (in)	73.5	88.1	96.3	84.5	85.8
12 (in)	97.8	100.6	99.8	101.6	97.3
18 (in)	102.1	106.8	105.9	102.8	98.4
24 (in)	104.4	108.6	109.9	108	99.2
Y	Outlet Air Temps (F)				
0	71				
12	100.6				
24	106.3				

Air side		Test #			
		2			
Air flow rate (cfm std)	Fan speed (Hz)	Ambient temp (F)			
1040	20	55.1			
Liquid flow rate (gpm)	Inlet water temp start (F)	Inlet water temp end (F)	Outlet water temp start (F)	Outlet water temp end (F)	
2.55	116.8	116	85.8	85.8	
			Base Temps (F)		
Y (0= Top)	X=-13.25 in (Left edge)	X=-6.75 (in)	X=0 (center)	X=6.75 (in)	X=-13.25 in (Right edge)
0 (in)	78.8	81.8	72.6	71.6	66
6 (in)	80.7	82.4	80.6	75.5	68.8
12 (in)	86	96	92	87.6	73
18 (in)	94.4	99	102.3	91.9	78.5
24 (in)	105.5	112.6	112.5	104.6	89
Y	Outlet Air Temps (F)				
0	58.3				
12	90.3				
24	100.1				

Air side		Test #			
		3			
Air flow rate (cfm std)	Fan speed (Hz)	Ambient temp (F)			
1495	29.6	51.6			
Liquid flow rate (gpm)	inlet water temp start (F)	Inlet water temp end (F)	Outlet water temp start (F)	Outlet water temp end (F)	
2.55	120	118.8	74.2	78.2	
			Base Temps (F)		
Y (0= Top)	X=-13.25 in (Left edge)	X=-6.75 (in)	X=0 (center)	X=6.75 (in)	X=-13.25 in (Right edge)
0 (in)	59.6	66.2	66.6	61.8	58.4
6 (in)	74.9	76.9	75.9	65.9	62.2
12 (in)	79	87	77.8	78.4	66.3
18 (in)	79.9	93.4	86.8	87.2	68.7
24 (in)	87	109	109.7	109	90
Y	Outlet Air Temps (F)				
0	53.9				
12	77.5				
24	88.3				

Air side		Test #			
		4			
Air flow rate (cfm std)	Fan speed (Hz)	Ambient temp (F)			
2230	40.1	51.2			
Liquid flow rate (gpm)	Inlet water temp start (F)	inlet water temp end (F)	Outlet water temp start (F)	Outlet water temp end (F)	
2.55	118.6	116.8	70.7	72.4	
			Base Temps (F)		
Y (0= Top)	X=-13.25 in (Left edge)	X=-6.75 (in)	X=0 (center)	X=6.75 (In)	X=-13.25 in (Right edge)
0 (in)	60.7	67.8	64.2	62	58.8
6 (in)	68.2	69.2	68.4	66.2	60
12 (in)	72.6	74.9	72.2	68	66.3
18 (in)	81.8	84.8	84.5	80	68.7
24 (in)	94.1	103.4	102.4	94.9	73.6
Y	Outlet Air Temps (F)				
0	54.9				
12	71.6				
24	79.2				

Air side		Test #			
		5			
Air flow rate (cfm std)	Fan speed (Hz)	Ambient temp (F)			
3300	50.1	45.3			
Liquid flow rate (gpm)	Inlet water temp start (F)	Inlet water temp end (F)	Outlet water temp start (F)	Outlet water temp end (F)	
2.55	126.4	124	64.9	64.4	
			Base Temps (F)		
Y (0= Top)	X=-13.25 in (Left edge)	X=-6.75 (in)	X=0 (center)	X=6.75 (in)	X=-13.25 in (Right edge)
0 (in)	53.4	62.5	57.8	53.5	50.1
6 (in)	62.8	63.5	62.2	56.3	52.3
12 (in)	66	68.3	62.7	60.2	59.2
18 (in)	70.7	73.6	74.6	71.4	70.9
24 (in)	80.6	100.4	103.4	98.1	76.9
Y	Outlet Air Temps (F)				
0	45.5				
12	63.2				
24	71.8				

Air side		Test #			
		6			
Air flow rate (cfm std)	Fan speed (Hz)	Ambient temp (F)			
3750	60.1	52.6			
Liquid flow rate (gpm)	inlet water temp start (F)	inlet water temp end (F)	Outlet water temp start (F)	Outlet water temp end (F)	
2.55	118.2	114	65.5	66.6	
			Base Temps (F)		
Y (0= Top)	X=-13.25 in (Left edge)	X=-6.75 (in)	X=0 (center)	X=6.75 (in)	X=-13.25 in (Right edge)
0 (in)	58.7	63.2	60.4	58.8	57
6 (in)	64.8	65.1	64.2	61.6	57.5
12 (in)	66.9	68.8	64.9	63.5	59.2
18 (in)	69.3	73.2	70.8	71.3	60.6
24 (in)	91.4	96.4	96.6	85.4	76.8
Y	Outlet Air Temps (F)				
0	54.7				
12	63.6				
24	71.7				

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