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Solar Evaporative Fan Coil Unit

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Evaporative Chiller

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ABSTRACT

The purpose of any engineering project is to anticipate a need and meet that need through prediction analysis and design. Over 70% of the nation's energy is consumed by building infrastructure such as HVAC systems, electrical, etc. HVAC systems use boilers to generate hot water or steam to heat buildings and chillers to provide cold water for cooling. The overall solar evaporative fan coil unit project involves the design and construction of a system that will heat and cool air using a solar collector and an evaporative chiller. This report covers the design and construction of the evaporative chiller specifically. The evaporator harnesses the latent heat of vaporization to chill a fin tube heat exchanger which can then provide chilled water to an air handling device called a fan coil unit. Testing will consider input and output water temperature, relative humidity, as well as input and output air temperature in order to compare the changes and develop a value for efficiency of the system.

INTRODUCTION

Motivation

With the cost of heating and cooling rising with the cost of energy – and concerns with environmental impact becoming increasingly important – a solution is needed to provide these functions at a lower cost using renewable resources. As an entry in the University of Washington Environmental Innovation Challenge, our project is designed to be a product, process or service that reduces waste, minimizes energy consumption, and decreases dependence on non-renewable resources.

Function Statement

The evaporative chiller will provide the necessary heat transfer capabilities to supply chilled water to the fan coil unit cooling the room. Water will be sprayed into the evaporator via misters driving down the temperature of the air passing through the evaporator which will then transfer heat away from the heat exchanger supplying cold water to the fan coil unit.

Requirements

In order to satisfy the function statement, the project must meet the following requirements:

- A pipe material, size, and length must be determined that allows the necessary heat transfer from the air going through the evaporator to drive down the temperature of the water in the heat exchanger by 20˚ F.
- The water distribution system (misters) must cause the water to evaporate into the air and drive the air temperature down enough to provide the necessary heat transfer to the water in the heat exchanger.
- The fan coil unit will provide the correct CFM of 55˚ of air to a 1000 cubic foot space to maintain a 75˚ temperature setpoint. This is determined by the heat transfer capability of the evaporator and fan coil unit.
- The evaporator shall be no larger than 3 feet long by 2 feet wide by 2 feet tall.
- The system layout shall be conducive to testing in the provided facilities.

Success Criteria

The success of the project will be defined by how efficiently the system can transfer heat away from a room and how long it can maintain a temperature setpoint. Success for this will be measured by the change in air temperature across the fan coil unit and how close the discharge air is to the setpoint. Another criteria is how efficiently the evaporator can transfer heat away from the heat exchanger contained within it. Success for this will be measured in the change in temperature of the air being passed through the evaporator and how well it can transfer heat away from the heat exchanger. Our system will be at least 20% efficient in its heat transfer capabilities and at least 15% efficient in its energy usage. To be competitive in the UW Environmental Innovation Challenge, our project will be a marketable option for reducing waste, minimizing energy consumption, and decreasing dependence on non-renewable resources.

Scope

As this scope of this project is quite large, the system will be separated in three parts. The solar collector will be designed and spec'd by Sam Budnick, the fan coil unit will be designed and spec'd by Kyle Kluever, and the evaporative chiller will be designed and spec'd by Jeremy Dickson. This document will cover the evaporative chiller that transfers the heat from the room to the air in the evaporator. The heat transfer capacity of the evaporator will be determined so a suitably sized heat exchanger can be specified for purchase.

DESIGN AND ANALYSIS

The design for this project was originally conceived by our group as a way to make heating and cooling more efficient by using renewable resources. Meant as a supplement to current heating and cooling methods, our system works to use the energy of the sun to provide heating during the day as well as improve the efficiency of evaporative chilling by using air from the space being cooled.

Approach

The solar water heater and the evaporator are used to provide hot and cold water through copper coils in a "fan coil unit" (FCU) which handles the air being recirculated to the space. Temperature of the air exiting the FCU is monitored and the recirculation fan is adjusted to provide the correct mass air flow to the FCU and/or evaporator to keep a temperature setpoint.

The design for the evaporator uses the latent heat of evaporation to cool a fluid in the heat exchanger contained within the evaporative chamber. A misting device sprays water into the chamber through which air is passed. Assuming the humidity of the air is low enough to begin with, the water vapor will evaporate into the air driving the dry bulb temperature of the air down. This works because the air will keep a constant enthalpy, and as the water vapor is evaporated into the air, the humidity will increase until it reaches a maximum saturation state. This increase in humidity of the air corresponds to a decrease in its dry bulb temperature (see psychometric chart, Appendix A-4) [1].

Benchmark

The most common type of air conditioning unit today uses a vapor-compression chiller. Vapor compressors use electrical motors and a refrigerant to produce their cooling effect via the reverse-Rankin cycle. Our system will try to replicate this by using a different method called evaporative cooling, sometimes called a swamp cooler. Instead of using a large amount of electricity to drive a refrigeration cycle, an evaporative cooler uses a much smaller amount of electricity and works by utilizing waters large enthalpy of vaporization and cools air through the evaporation of the water into the air. However the limitation of evaporative coolers is that they only work well in hot and dry environments of 40% humidity or less and do not cool as efficiently as a refrigerated air conditioner. An example is the Honeywell CSO71AE Indoor Portable Evaporative Cooler. With 150 CFM of air flow, a 50 watt motor, and a water tank capacity of 1.8 gallons, this unit can cool a room up to 100 ft^2 by 15˚ F [3]. In comparison, our unit will use 60W of power, but will work in environments of 40% humidity or more, and will cool 20˚ F.

Calculated Parameters

The main calculated design parameter for this proposal is sizing the amount of copper pipe needed in the fin tube heat exchanger contained within the evaporator for optimal heat transfer. Using the equation $\dot{Q} = \dot{m}C_p\Delta T$ the heat transfer rate of the working fluid in the heat exchanger can be determined [2]. With a flow rate of 0.214 GPM and a temperature differential of 20° F, the heat transfer rate is calculated to be 2120 BTU/hr. This heat transfer rate can then be used to calculate the surface area of copper pipe needed using the equation $\dot{Q} = UA\Delta T_{lm}$ [2]. First however, the log mean temperature differential must be calculated using the equation $\Delta T_{lm} = \frac{\Delta T_1 - \Delta t T_2}{ln\left({\Delta T_1}/{\Delta T_2}\right)}$ [2]. With the previously calculated heat transfer rate of 2120 BTU/hr, a heat transfer coefficient of 60 BTU/ft^2·hr·°F, and a log mean temperature of 12.4° F, the area of copper pipe needed is calculated to be 2.85 ft^2 (Appendix A-2). With this value the length of copper pipe can then be calculated. First the area of the pipe per 1 ft length must be calculated: $A = (1ft_L)\pi d$. With a nominal standard pipe size of 3/8" (1/2" OD) the area of 1ft of pipe is 0.131 ft^2. Next, taking the area of 1ft of 3/8" pipe and dividing it by the area of copper pipe needed, 2.85 ft^2, gives a length of copper pipe needed of 21.77ft (Appendix A-3).

Performance Prediction

After selecting a suitable heat exchanger that will satisfy the requirements set by the design parameters, a prediction can be made using the specs given by the heat exchanger manufacturer as to the performance of the unit for our specific use. The analysis starts with the specs of the heat exchanger which are a heat exchange rate of 54589 BTU/hr with an inlet water temp of 180˚F, a water flow rate of 10 GPM, and an air flow rate of 800 ft^3/min. First by assuming an air inlet temp of 70˚F we can calculate the air outlet temp by using the equation $\dot{Q}_{air} = \dot{m}_{air}C_p\left(T_f - T_i\right)$. This results in an outlet temp of 133.1˚F (Appendix A-5). Next we can calculate a water outlet temp using the same equation. This results in a water outlet temp of 168.9˚F. Using these temperatures we can find the log mean temperature difference using the equation $\Delta T_{lm} = \frac{\Delta T_{in} - \Delta T_{out}}{ln\left(\frac{\Delta T_{in}}{\Delta T_{out}}\right)}$. This results in a log mean temperature difference of 66.1˚F. Using this value we can predict the performance of the heat exchanger at the log mean temperature differential used for design by calculating the UA value for the heat exchanger. First an air temp correction factor is determined using an R value of .176, and a P value of .573. Using figure 22-18 of Thermo-fluids [3] an air temp correction factor of F=.97 was determined. Next, using the equation $\dot{Q}_{air} = [UA](\Delta T_{lm}F)$ a UA value of 851 BTU/hr-˚F was calculated. If we plug this value back into the equation $\dot{Q}_{air} = [UA](\Delta T_{lm}F)$, and use a log mean temperature difference of 12.4˚F from our design, the heat transfer rate is 10,236 BTU/hr, well above our design heat transfer rate of 2120 BTU/hr. At almost 4.5 times the design rate, there is plenty of room to accommodate possible efficiency losses incurred in the system.

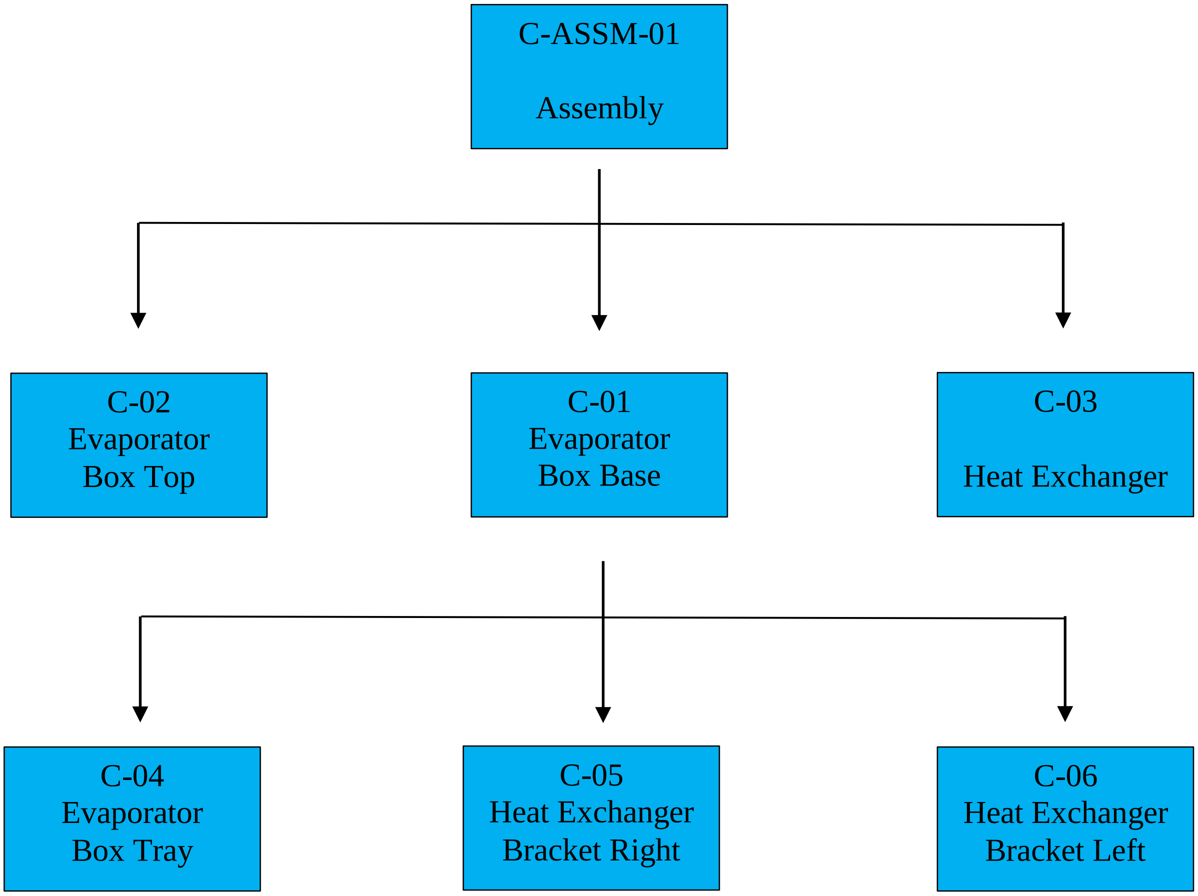
METHODS AND CONSTRUCTION

The design for the evaporator calls for a box to contain the heat exchanger and the misting device. This box will be constructed out of galvanized sheet metal for rust resistance since it will contain water. The sheet metal box will also serve as the structure for the devices contained within. The heat exchanger, misting device, and inlet ducting will all have attachment points built into the box.

Construction

The box will be built from two separate pieces: a base that serves as two walls and the water basin, and a "cap" that serves as the top as well as the inlet and outlet (Appendix B-1 and B-2). These two pieces will fit together and be screwed together with self-tapping sheet metal screws. Gaps in the sheet metal bends of the base that contain water will be sealed with caulking. The water reservoir in the base will be separated from the heat exchanger with another piece of sheet metal with a drain into the water reservoir. The sheet metal parts will be designed in SolidWorks which will produce a flat pattern which can be exported as a dxf file and cut on the CNC plasma table. The cut pattern will then be bent according to the drawings with a sheet metal brake. These are the only parts that will be designed and constructed from scratch.

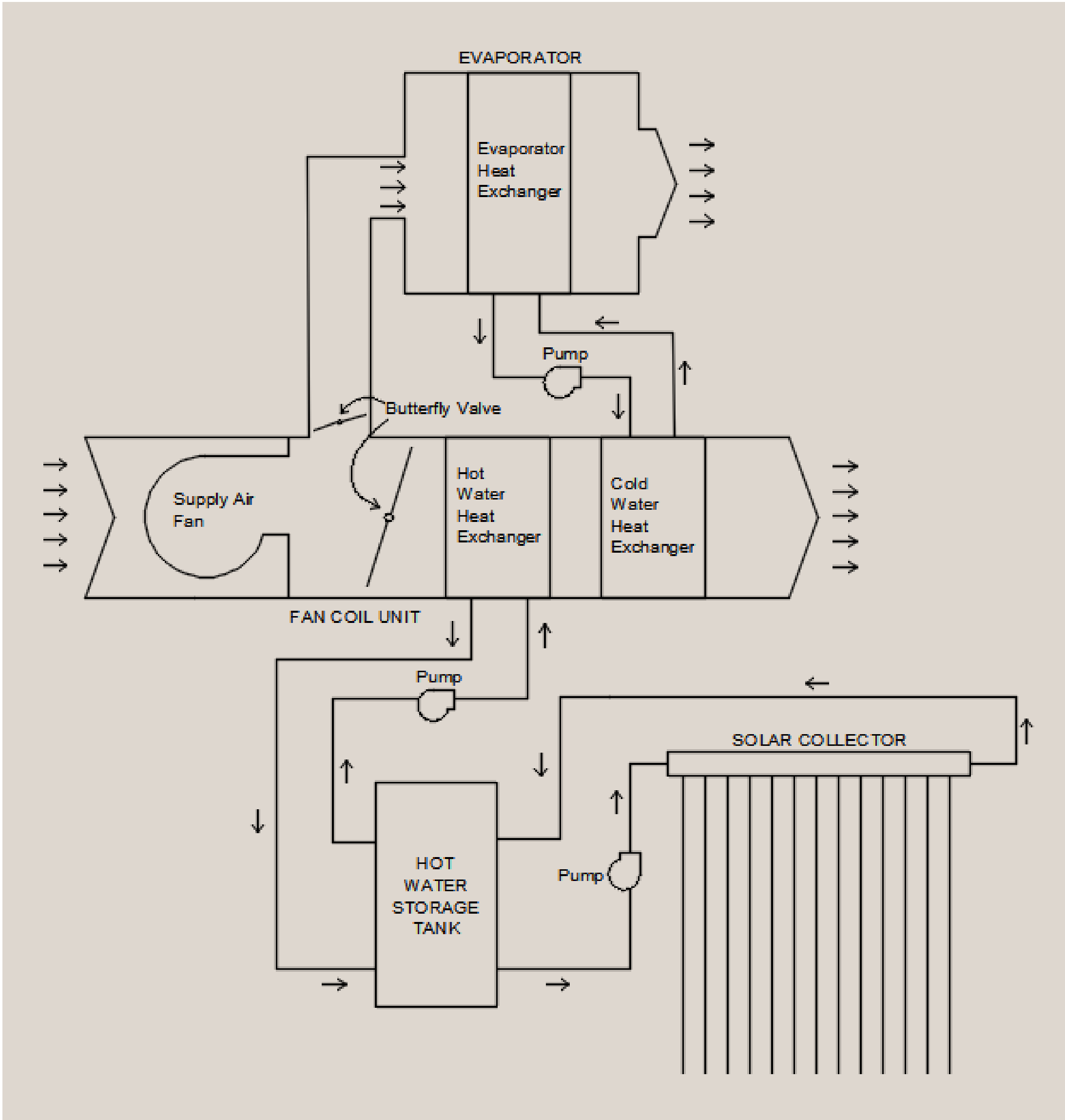
Several other parts will be sourced and assembled together with the box. A water to air heat exchanger will be used in the evaporator. It will be sized according to the analysis in Appendix A. The misting system will consist of a misting nozzle fed by a micro diaphragm pump capable of supplying at least 100 psi for the misting nozzle. The pump will be connected to the evaporator box water reservoir and will pump up to the top of the box to the misting nozzle. Nylon piping will be used to connect the reservoir, pump, and nozzle. The heat exchanger will be connected to the FCU and recirc pump with PEX tubing. Threaded pipe fittings will be installed on the heat exchanger and PEX tubing for easy disassembly. An aluminum foil "accordion" duct will also be supplied to connect the FCU and evaporator.



Drawing Tree – Figure 7.1

Device Operation

The operation of this device starts at the FCU. The recirc fan within the FCU brings air in from the room and splits it to the rest of the FCU and to the evaporator. Butterfly valves control the ratio of air going to the evaporator and the FCU to provide the correct CFM for proper cooling. The air then enters the evaporator where it passes over the heat exchanger taking heat away from the working fluid (water) and exits the back of the evaporator outside. Evaporation is achieved via a misting nozzle located within the evaporator box. The misting nozzle sprays a fine mist onto the heat exchanger and as the air is passed over the heat exchanger it evaporates cooling the air. The working fluid which is cooled by the air is then pumped to a heat exchanger in the FCU. The air passing through the FCU is cooled by the coil and exits the FCU into the room. This air is recirculated by the FCU throughout the day to keep a temperature setpoint. See Figure 8.1 below for a detailed system schematic.



System Diagram - Figure 8.1

TESTING METHOD

The requirements of testing this device must provide a manageable means of acquiring data to demonstrate the level to which the project has met its requirements as stated in the introduction. These requirements are:

- A pipe material, size, and length must be determined that allows the necessary heat transfer from the air going through the evaporator to drive down the temperature of the water in the heat exchanger by 20˚ F.
- The water distribution system (misters) must cause the water to evaporate into the air and drive the air temperature down enough to provide the necessary heat transfer to the water in the heat exchanger.

- The fan coil unit will provide the correct CFM of 55˚ of air to a 1000 cubic foot space to maintain a 75˚ temperature setpoint. This is determined by the heat transfer capability of the evaporator and fan coil unit.
- The evaporator shall be no larger than 3 feet long by 2 feet wide by 2 feet tall.
- The system layout shall be conducive to testing in the provided facilities.

Three main parameters will be monitored directly with measuring equipment in order to acquire the data needed to test the validity of these requirements:

1. Heat exchanger inlet and outlet water temperature
2. Evaporator air inlet and outlet temperature
3. FCU Air inlet and outlet temperature

Several additional secondary parameters will be monitored to provide data for optimization. These parameters can be changed independently and monitored to see how they affect the three main parameters:

1. Pump GPM
2. Evaporator air CFM
3. Evaporator air relative humidity
4. FCU air CFM

The predicted performance of the evaporator heat exchanger is a maximum heat transfer rate of 10,236 BTU/hr. With this predicted heat transfer rate, the unit should be easily able to handle the design heat transfer rate of 2120 BTU/hr. Measuring the air temperature differential and water temperature differential will give values for log mean temperature differential. These values can then be used to calculate the heat transfer rate using the calculated UA value of 851 BTU/hr-˚F.

Data acquisition will be provided by several different measurement instruments. For water and FCU air temperatures 4 Fluke thermocouple thermometers will be used. For evaporator air temp, relative humidity and air CFM 2 anemometers will be used. Data logging will be done by hand because of the slow change in values. Real time data logging is unnecessary for the purposes of this project. The pump GPM is displayed by the pump itself and can be changed with a button on the pump.

Testing was scheduled to start March 9th and run through April 10th with initial testing done for EIC by April 2nd. Testing actually started April 6th and ran through mid-May. Initial testing for EIC was not completed due to complications with priming the pump. Instead testing was done for the solar water heater.

Method/Approach

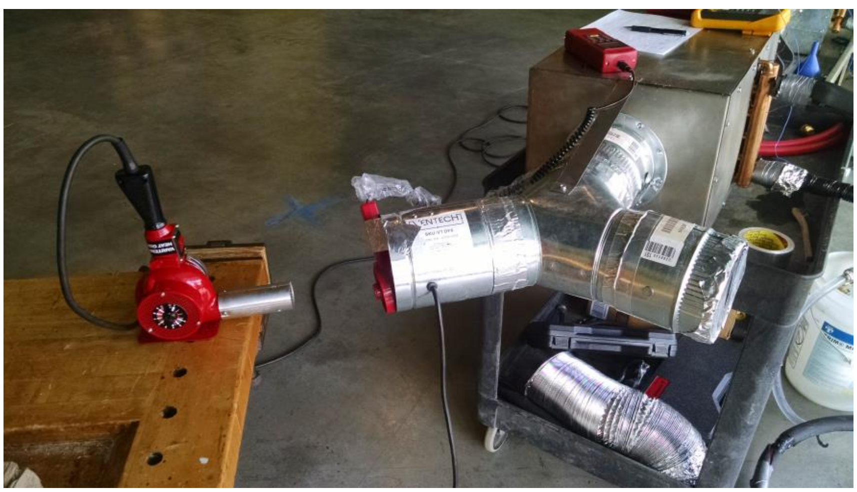
The resources required for this project changed over time as new problems arose and the scope of the project became clear. Many aspects of the project were not know ahead of time and were figured out as the project went along. Initially, when testing started, the setup was done outside to prevent spills from the water circulation system while priming of the pump and leaks were figure out.

Both the evaporator and FCU eventually were put on rolling carts to make transportation easy, help contain leaks and provide a place to store all the required hardware for hooking the system up. A hose and a faucet were required to fill the water circulation system and provide positive pressure for priming the pump. At this point the water circulation system was open and included a 5 gallon bucket in which the water was pumped out of and into the evaporator and pumped back into from the FCU. Additionally, outlets were required for powering the various fans, pump, pump power supply unit, and later on the heat gun. As testing went on it became apparent that closing the water circulation system and bringing the whole setup inside would be required. As a result the whole system was left hooked up and brought inside the Fluke lab instead of taking it apart every time testing was done and stored in the senior project room.

As stated earlier, testing for this project changed over time and happened in stages as the setup changed and new problems presented themselves. The initial testing procedure was to get the system working and gather data to see if the system was working. Once it was determined that the system was working as desired data was acquired as a function of time. This relied on a warm day with temperatures above 70 °F and calm winds. With the initial setup the air from the duct fan was slip into the FCU and evaporator. However, due to the distance required to duct the air into the evaporator and/or leaks, the CFM required to provide adequate cooling in the evaporator was insufficient. As a result a more powerful fan with a VFD was used to provide the air for the evaporator separately from the FCU.

After initial data was gathered it was determined that bringing the system inside where the ambient temperature was constant, and using a closed water circulation loop, would provide the best chance for producing consistence, meaningful data. To provide an easily adjustable constant air temperature entering the FCU a heat gun with an adjustable temperature dial was used. This allowed the temperature of the air entering the FCU to be changed and compared to the change in the heat exchanger water temperature.

After several attempts at gathering data it was determined that some changes would need to be made to optimize the system. It became apparent that the temperature of the air in the evaporator was not dropping enough to provide adequate cooling of the heat exchanger. As a result two more misting nozzles were added to the misting system in order to bring the relative humidity of the air going through the evaporator up further. Once this was done data was gathered to determine to optimal CFM of air through the evaporator. This was done by comparing the change in air temperature as a function of air CFM. To provide a constant value for CFM the VFD powering the fan was set to voltage values at even increments.



Three misting nozzle setup - Figure 12.1

Data Presentation

This first graph represents the very first data logging done. Its purpose was to assess how the unit was working and provide a baseline for which to continue testing. It was decided that data would be taken to see how the unit would perform over time. Analyzing the data it is apparent that the heat exchanger was cooling the water even with an increase in ambient air temperature. However, the water temperature did not decrease as quickly as the air increased. This also made it apparent that the water to air heat exchange was not as efficient as it would be the other way around. Although this heat exchanger is designed to work both ways, it was obvious that the optimal flow rates for air and water needed to be found.

The next graph represents an attempt to see how the performance of the evaporator affected the temperature of the air in the FCU. This would provide the data necessary to see where optimization could be made. The data shows that as the heat exchanger water temperatures rise, that the air temperature differential increases. This seems to suggest that the heat exchangers are

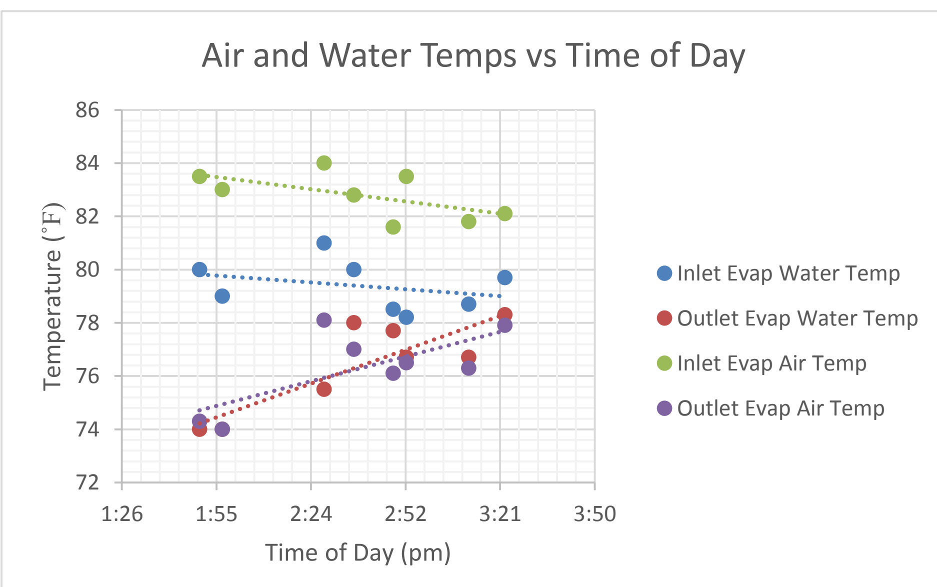


Figure 12.2

more effective at higher temperatures. Also apparent from this data is that the water temperature did not change significantly from the inlet to the outlet. This could be due to inefficiency within the heat exchanger or not having the optimal flow rates for air or water. The last thing that the data shows is that the evaporator was not cooling the water sufficiently. After this was determined the extra two misting nozzles were added.

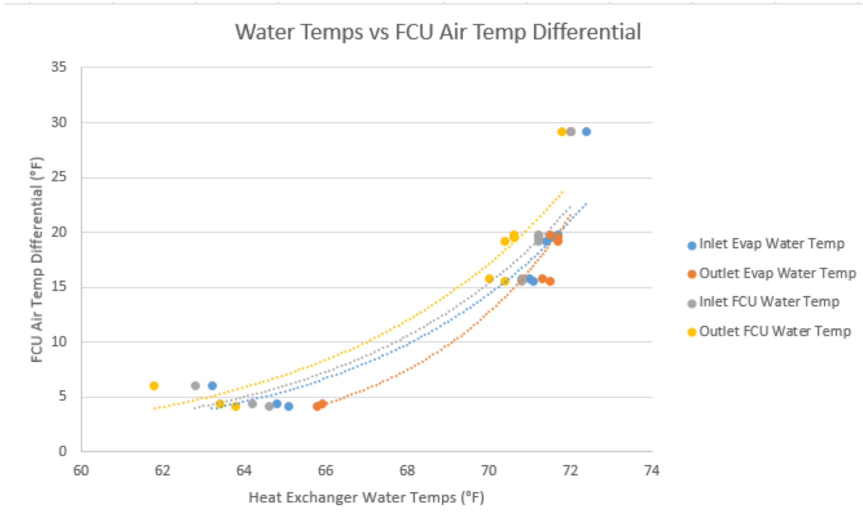


Figure 13.1

Once the additional misting nozzles were added a test was done to optimize the relative humidity of the air passing through the evaporator. The next graph shows how the evaporator air temperature differential changes with relative humidity. To change the relative humidity the inlet fan was set at different flow rates. This changed the volume of air the mist had to evaporate into therefore changing the relative humidity of the air. The data seems to suggest that about 65% relative humidity is where the air temperature differential peaks. After that point it drops off.

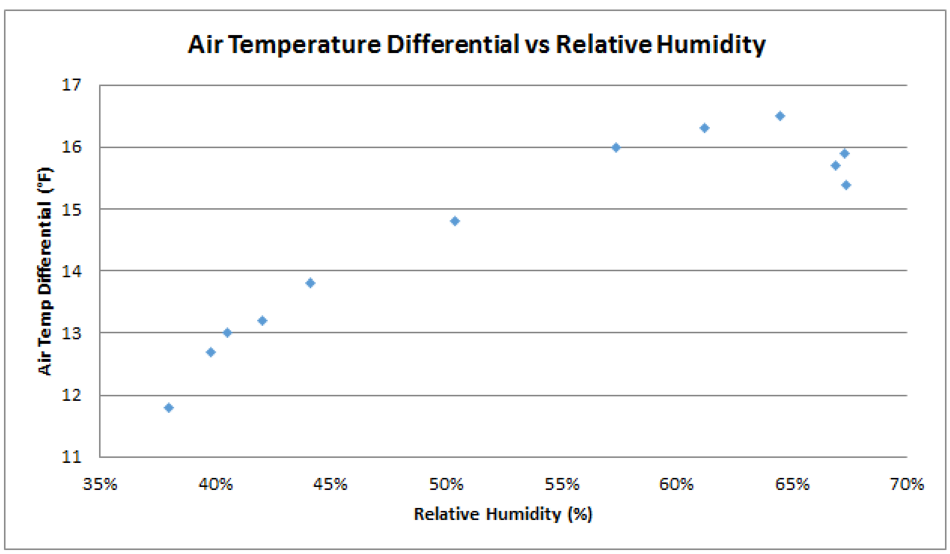


Figure 13.2

Now that the air flow rate had been optimized for maximum relative humidity a test was done to determine how the FCU performed at different inlet temperatures. From the second test it was theorized that the heat exchanger was more efficient at higher inlet temperatures. For this test the FCU inlet air temperature was changed while all other parameters were kept constant. The result is a 2:3 ratio of FCU air temperature differential to inlet FCU air temperature. This means that as the inlet air temperature increased the heat exchanger was better able to transfer heat away from the air. The reason for this is most likely that the water provided by the evaporator was not cold enough to provide a large temperature differential between the water and the air. The performance of the heat exchanger diminishes as the temperature differential between the water and the air gets closer.

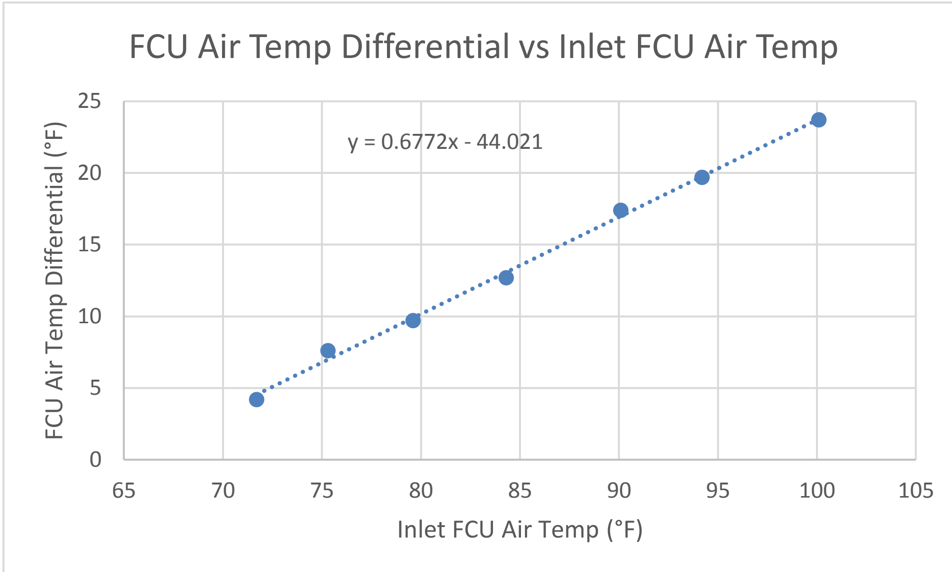


Figure 14.1

The last test done was to compare the inlet FCU air temperature to the water differential temperature. This was done in order to see how the heat exchangers were performing now that it was determined that higher temperatures were needed to get sufficient cooling properties. As the Figure 15.2 shows, there is on average a 3°F difference in the water temperature differential in the evaporator heat exchanger vs the FCU heat exchanger. This would seem to suggest that the evaporator was more efficient in its heat transfer capabilities that the FCU.

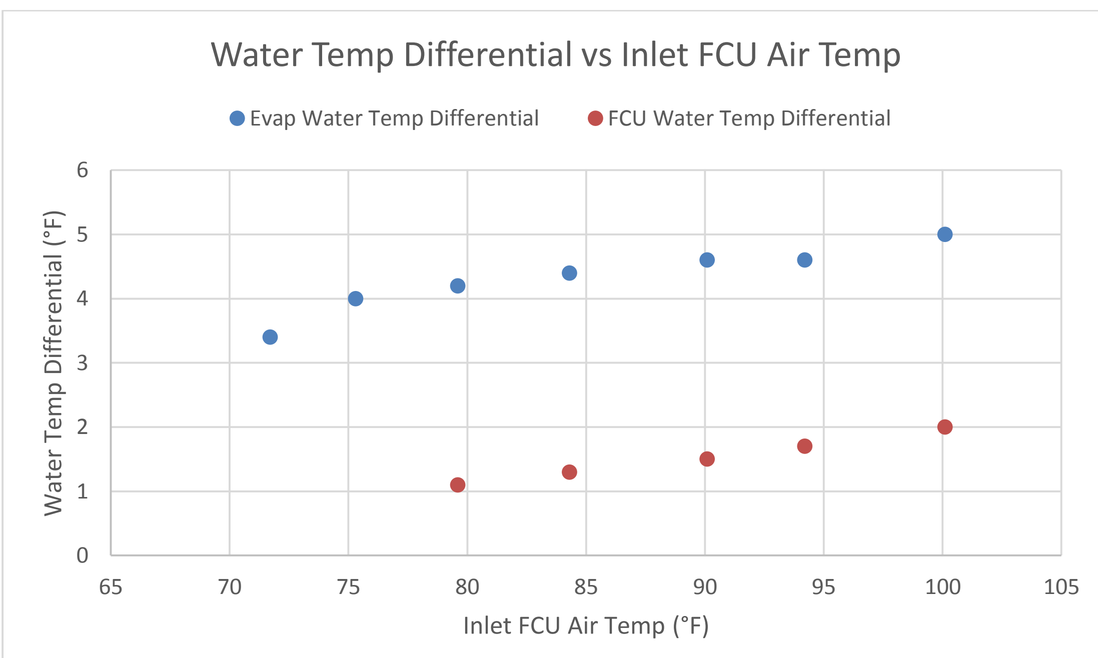


Figure 14.2

Deliverables

Using the values in appendix F-5 we can determine if the heat exchanger met the predicted design heat transfer rate of 2120 BTU/hr. By calculating the log mean temperature difference using the values for FCU air temperature and FCU water temperature and then multiplying by the UA value of 851 BTU/hr-˚F we can determine if the system met the predicted heat transfer rate. Figure 15.1 shows that for inlet FCU air temperature values above 79.6°F the heat transfer rate met or exceeded the predicted value. Not shown is data below the inlet FCU air temperature of 79.6°F because the FCU water temperature differential was negative. This means no heat transfer was taking place and the heat exchanger did not perform as predicted.

Inlet FCU Air Temp	FCU Water Temp Differential	FCU Air Temp Differential	Log Mean Temp Difference	Heat Transfer Rate
90.1	1.5	17.4	6.49	5520.55
94.2	1.7	19.7	7.35	6252.27
100.1	2	23.7	8.78	7469.36
84.3	1.3	12.7	5.00	4256.42
79.6	1.1	9.7	3.95	3362.07

Figure 15.1

Additionally the requirement of a 20°F air temperature differential was met but only at FCU values above 100°F. This does not satisfy the requirement that 55°F air be supplied to the room to maintain a 75°F temperature setpoint.

In conclusion it was determined through testing that the heat exchangers did perform at the predicted level. However, they only performed at values above what was needed for this application. The evaporator was unable to provide chilled water at a low enough temperature to provide a large temperature differential necessary to cool the air to the temperature stated in the requirements.

BUDGET/SCHEDULE/PROJECT MANAGEMENT

Proposed Budget

The main supplier for this project will be Amazon.com. Most of the materials needed for this project can be bought from Amazon with free two day shipping. This will be important for keeping on schedule and makes it easy if replacement or new parts need to be obtained quickly. The other reason Amazon will be used is that if sometime in the future we would like to patent or sell our idea we are not allowed to used University funds or get discounts through the University. This ended up not being an issue because we decided not to go ahead with the patent process.

A potential source of funding was the Washington State Clean Energy Institute which provides student teams focusing on tech innovation in solar energy, electrical energy storage, and/or the software or hardware for renewable integration with the grid, with prototype funding through the University of Washington Environmental Innovation Challenge. We asked for $2000 of funding for this project. As it turns out we did not receive this funding. As a backup plan we offered to give our projects to the engineering department to use in exchange for buying the parts up front.

The single most expensive part of this project is the heat exchanger at $69.99. It is however the most important part of this project and therefore made the most sense to buy one premade from a reputable manufacturer. The next most expensive material is the galvanized sheet metal at $41.10 for a 4'X2' sheet and 4.5'X1.5' sheet. It will be sourced from Haskins Steel. The last material not being sourced from Amazon will be PEX tubing which is going to be donated by Jeff Greear, a local business man who took interest in our project. The total cost of this project was estimated to be $347.21 (Appendix D-1). After the parts had all been sourced the total cost was $222.29, coming in at $129.54 under budget.

Schedule

The schedule for this project starting in January has been broken up into 6 categories: build prep, sheet metal fabrication, tubing assembly, part assembly, testing, and evaluation. An additional time constraint has been set by the UW Environmental Innovation Challenge in that the project must be built, working, and some testing done by April 2nd. This is when the competition takes place in Seattle.

Build prep consists of ordering parts, taking delivery of parts, and preparing the sheet metal drawings for CNC plasma cutting. The estimated total hours for this category is 8. Sheet metal fabrication consists of actually cutting the sheet metal and then bending the sheet metal to the final shape. The total estimated hours for this category is 18. Tube assembly consists of cutting the tubing to length and attaching the fittings. This is estimated to take 8 hours. These three categories are scheduled to take place from January to the beginning of February.

After sheet metal fabrication and tube assembly is finished the parts assembly can take place. Part assembly consist of assembling the sheet metal parts, assembling the tubing and pumps, finalizing the assembly and sealing the evaporator, and lastly, attaching it to the FCU. This is estimated to take 36 hours and scheduled to take place from the beginning of February to the middle of March.

After final assembly the testing will take place. Testing is scheduled to start as soon as the final assembly is complete and run through mid-April. It is estimated to take 26 hours. The total estimated time for this project is 93.5 hours. The final time for this project turned out to be 51.08 hours. The main the reason for this was overestimation for every activity. The closest estimation was for testing. It was estimated to take 23 hours. It actually took 20.24 hours.

ACTIVITY	PLAN DURATION	PLAN HOURS	ACTUAL HOURS	PERCENT COMPLETE	PERIODS Jan	Feb	Mar	Apr	May	June
Build Prep	Jan 6-23	14	6.25	100%						
Sheet Metal Fab	Jan 19-Feb 13	7.5	3.78	100%						
Tubing Assembly	Jan 19-30	8	2.5	100%						
Part Assembly	Feb 6-Mar 13	21	10.16	100%						
Testing	Mar 9-Apr 10	23	20.24	100%						
Evaluation	Apr 13- Jun 5	20	8.15	100%						
	TOTAL HRS	93.5	51.08							

High Level Gant Chart - Figure 8.1

Some additional deliverables are also due within this timeline. For the UW Environmental Innovation Challenge we must have our 5-7pg business summary submitted by February 22rd. This summary is our official entry into the Challenge. A 1pg business summary which will be distributed to the judges at the competition is due March 29th. Lastly, the competition is to be held on April 2nd in Seattle. If we are chosen to compete we will need to have our project completed and ready to display by this date. Additionally, a fully functioning unit must be completed by the end of winter quarter on March 13th to satisfy the MET495B class requirement. Another requirement is that our project be submitted to SOURCE. The submission period for SOURCE is March 2nd to April 10th and the presentation for SOURCE is on March 21st. Lastly, the testing, evaluation, and final report is then due at the end of spring quarter by June 5th. After testing and evaluation is complete a presentation will be prepared for our groups scheduled time during the last few weeks of spring quarter.

DISCUSSION

The idea for this project originally was to use ammonia as the working fluid. Because of the unique properties of ammonia and its close chemical composition to water it would have been a perfect fluid for heat transfer. By using the heat of the sun we could have created a thermosiphon effect that would have negated using a pump, therefore saving electricity. However, after doing a feasibility study of using anhydrous ammonia is was determined that it would be too expensive and unsafe to use and test with. The next design iteration was to use evaporation. This design was ultimately chosen for its simplicity. Doing analysis would be easier, and cost would be greatly reduced because of the simplicity of the design.

Next, the integration into the rest of system needed to be determined. It was decided that to solve the problem of evaporators only working in dry and hot environments that we could use the already dry air inside the room to be cooled. The FCU then was designed to split the air between the evaporator and the rest of the FCU. This way dry dehumidified air is always available. For the heat exchanger it was determined that an air to water fin tube heat exchanger could be used effectively. It was decided that analysis should be done to specify a suitable fin tube heat exchanger. Once this analysis was done and a suitable heat exchanger was chosen the final dimensions for the evaporator could be determined.

Another design decision that needed to be made was how to build a water reservoir for the evaporator misting system. The first design use a tray that would direct water into the bottom of the reservoir. However this design would mean the water would be shallow and any movement or uneven surface might deprive the pump. A new tray was then designed that would direct the water to one point where the outlet to the pump would be located.

The build portion of the project had a few issues, however none of them caused problems that affected the overall performance of the system. The first issue during construction was with the CNC plasma table. The dimensions of the part were just outside the travel distance of the table and resulted in the dimensions of one side of the evaporator base being out of spec. When the sheet metal was then bent the affected area left a sizable gap. This was solved by filling the gap with enough silicone caulking to keep it from leaking. Another problem with the bending of the small side tabs of the base. None of the brakes available had the correct geometry to fit the side tabs without the larger already bent sides interfering. This was solved by bend the side tabs as far as possible (about 45°) and the using a hammer and anvil to bend them the rest of the way. Lastly, when the sheet metal for the tray was delivered it turned out to be hot rolled sheet instead of galvanized sheet. There was not enough time to order a new sheet, so in order to make the tray rust resistant it was painted.

During the assembly portion of the build one main issue cropped up. The tubing that was purchased for the misting system turned out to be too small for the inlet and outlet of the misting pump. In order to make the tubing fit it had to be heated up with a torch just enough to soften it so it would expand and fit over the inlet and outlet. The nylon tubing used was very stiff and when heated would melt quickly and bend easily. This created a huge hassle when trying to expand the ends to fit the pump. Significant time was spent trying to get the tubing on the pump and stop it from leaking. The nylon tubing was chosen because it could withstand a significant amount of pressure. However, after running the misting system it was apparent that a different material could have been chosen that would have been easier to attach to the pump.

After the build portion of the project was complete it was time to integrate all the parts together. This proved to be the most complicated, time consuming, and difficult part of the whole project. Most of the problems were related to the chilled water loop. The biggest and most reoccurring problem with the chilled water loop was leaks. Since the hoses had to be removable, the clamps used were adjustable. Unfortunately these clamps did not provide the necessary clamping force on the Pex hoses to stop them from leaking. One of the first problems that occurred with the chilled water system was trying to figure out how to prime the circulation pump. The fast and easy way that was used for the first test was to use a 5 gallon bucket as a reservoir. This however was not as ideal as having a closed loop. Eventually, for the third and fourth tests, a valve was added that allowed the system to be primed with a garden hose and then shutoff creating a closed loop.

Another problem that occurred when testing the evaporator was that the amount of air being supplied was inadequate due to a long lead on the inlet ducting from the splitter attached to the FCU. Significant pressure and therefore CFM loss resulted. This was solved by directly adding a separate fan to the evaporator. As a result the original design to split the inlet air between the FCU and evaporator was not used. However, if the system could have been setup differently with equal length ducting for the FCU and evaporator the design may have worked better.

Finally, the last design changed that happened as a result of testing was the misting system. It became obvious through testing that the evaporator was not performing at a level expected by the design. It was eventually determined that the relative humidity of the air going through the evaporator was not reaching a level in line with its design. The addition of two more misting nozzles helped to raise the relative humidity of the air and resulted in a greater drop of chilled water temperature.

CONCLUSION

The success of this project was based on the requirement of determining an optimal length of 3/8" copper pipe for adequate heat transfer through analysis of the heat transfer from water to air through a copper pipe. It was determined that a 3/8" copper pipe of 21ft long is required and a suitable heat exchanger was chosen that meets this requirement. Analysis was done to determine that if air enters the evaporator at 75° F and 10% humidity, that if the air is saturated to 70% humidity that it will drop 20° F and provide the necessary chilling to the FCU. The misting system was ultimately successful in raising the relative humidity of the air enough to bring the temperature down 20°F. The additional requirement of a suitable water distribution system for evaporation to occur was also successfully determined. Lastly, the data gathered through testing proved that for temperatures above 79°F the system provided the required heat transfer rate as outlined in the performance prediction. However, the system was unsuccessful in providing cooling at the level specified in the requirements (75°F to 55°F).

REFERENCES

[1] "Condensation on Windows," North Georgia Replacement Windows, 02 July 2012. [Online]. Available: http://ngwindows.com/blog/tag/psychrometric-chart/.

[2] Y. A. Cengel and J. M. Cimbala, Fundamentals of Thermo-fluid Mechanics, New York: McGraw Hill.

[3] Honeywell, "CS071AE," [Online]. Available: http://www.honeywellaircoolers.com/products/CS071AE.

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Matt Burvee – MET Instructional Support Tech

Donations
Jeff Greear – Owner of Ellensburg Solar

APPENDIX A – Analysis

A-1 Heat Transfer Rate

Jeremy Dickson | Working Fluid Heat Transfer Rate Analysis | 10/22/14

GIVEN: $\Delta T = 20°F$ Temp drop needed for fluid in coil

$C_p = .9893\ Btu/lb_m \cdot R$ Specific heat of water @ 75°F
Thermo-fluid Sciences
Cengel/Cimbala
Table A-3E

$\dot{m} = .214\ gpm$ calculated value based on demand for fan coil unit

FIND: $\dot{Q}$ Heat transfer rate of working fluid

SOLUTION:

$$\dot{Q} = \dot{m} C_p \Delta T$$

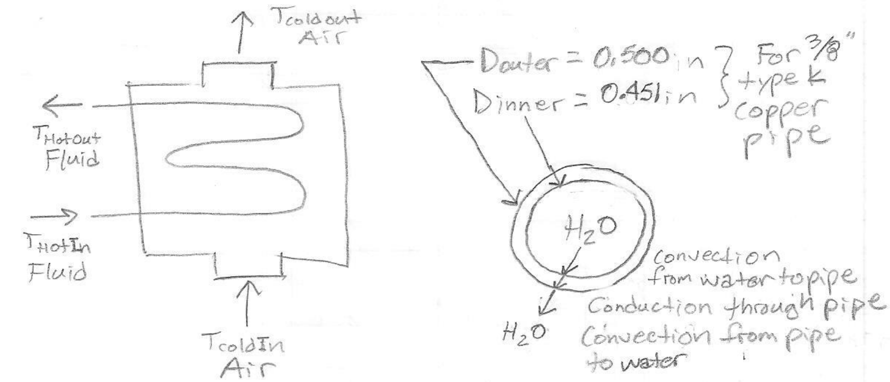
$$= \left(.214 \frac{gal}{min}\right)\left(.9893\ Btu/lb_m \cdot R\right)\left(20°F\right)\left(\frac{8.345\ lb_m}{1\ gal}\right) \times \left(\frac{60\ min}{1\ hr}\right)$$

$$\boxed{\dot{Q} = 2120\ Btu/hr}$$

A-2 Pipe Area

Jeremy Dickson | Heat-exchanger Sizing Analysis | 10/22/14 | 1/2

GIVEN: $T_{HotIn} = 75°F$ (Fluid), $T_{HotOut} = 55°F$; $T_{cold\,In} = 50°F$ (Air), $T_{cold\,Out} = 50°F$ } Assumed ideal Values



$\dot{Q} = 2120$ Btu/hr $\qquad U = 60$ Btu/ft$^2\cdot$hr$\cdot$°F Engineering Toolbox

For water to water through copper

FIND: Area of pipe needed to drop temp of working fluid 20°F by 50°F air

SOLUTION:

Equations needed:

$$\dot{Q} = UA\Delta T_{lm} \Rightarrow A = \frac{\dot{Q}}{U\Delta T_{lm}}$$ Eq. 22-15 Thermo-Fluid Sciences pg 941

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$$ Eq. 22-25 Thermo-Fluid Sciences p941

$$\Delta T_1 = T_{HotIn} - T_{coldout} = 75°F - 50°F = 25°F$$

$$\Delta T_2 = T_{Hotout} - T_{coldin} = 55°F - 50°F = 5°F$$

$$\Delta T_{lm} = \frac{25°F - 5°F}{\ln(25°F/5°F)} = \underline{12.4\,°F}$$

$$A = \frac{2120 \text{ Btu/hr}}{(60 \text{ Btu/ft}^2\cdot\text{hr}\cdot°F)(12.4°F)} = \underline{2.85 \text{ ft}^2}$$

A-3 Pipe Length

Jeremy Dickson	Heat Exchanger Sizing Analysis	10/22/14	2/2

Area of 1 ft_L of 3/8 inch type k copper pipe

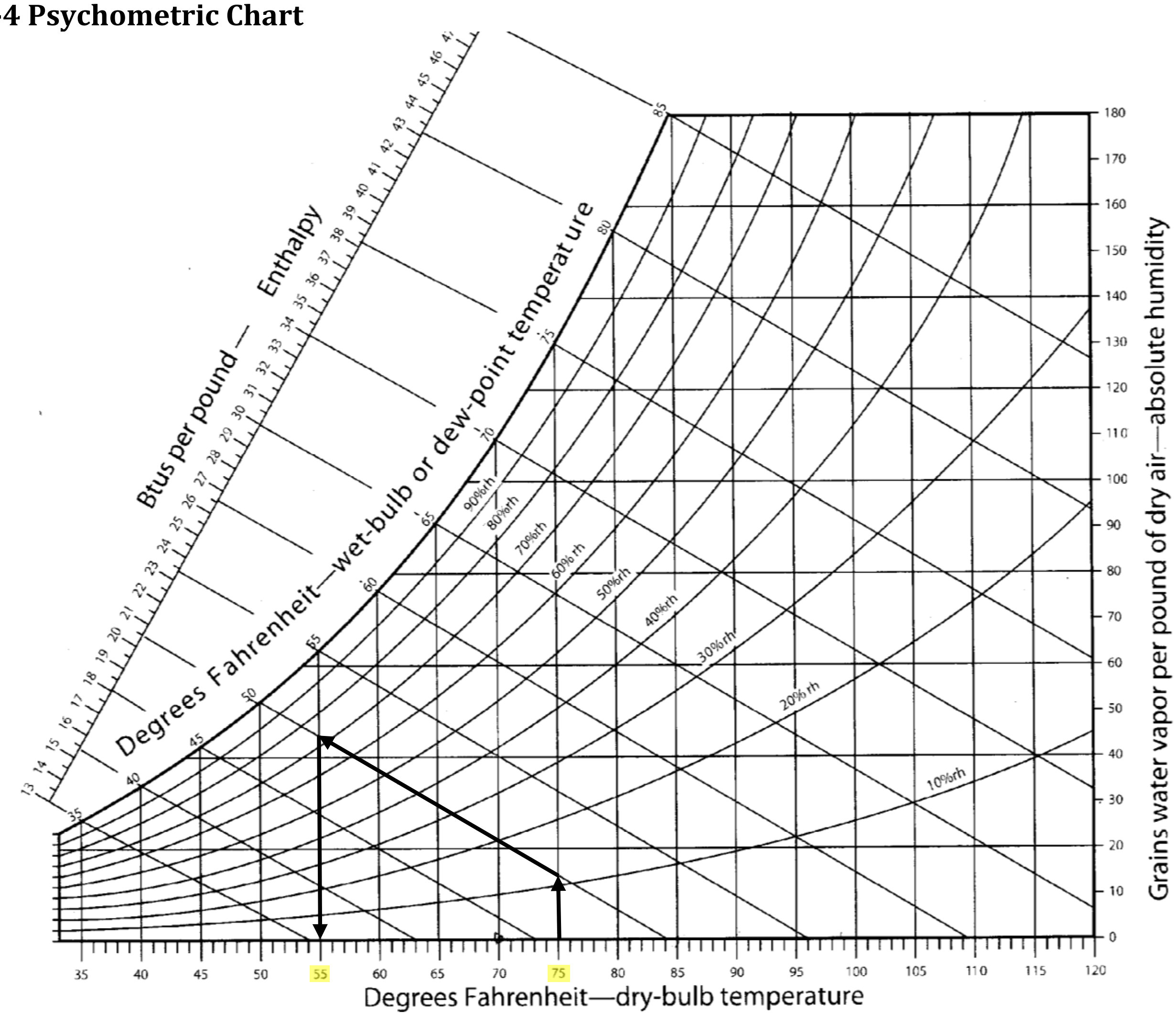
$A = (1ft_L)\pi d = (1ft_L)(\pi)(0.500\ in)\left(\frac{1\ ft}{12\ in}\right)$

$A = 0.131\ ft^2$ (per 1 ft_L of pipe)

Length of pipe needed for heat transfer demand:

$L = \frac{2.849\ ft^2}{0.131\ ft^2/ft_L} = \underline{21.75\ ft_L}$

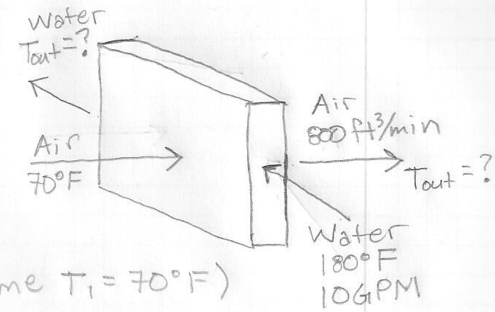
A-4 Psychometric Chart



A-5 Heat Exchanger Performance Part 1

JEREMY DICKSON	MET495	2/7/15

GIVEN: $\dot{Q} = 54589$ BTU/hr
Inlet water temp = 180°F
water flow rate = 10GPM
Air flow rate = 800 ft³/min



FIND: UA value for heat exchanger

SOLUTION:

Air outlet temperature (Assume $T_1 = 70°F$)

$$\dot{Q}_{AIR} = \dot{m}_{AIR} C_p (T_f - T_i) = \rho \dot{vol} C_p (T_f - T_i)$$

$$\Delta T_{AIR} = \frac{\dot{Q}}{\rho \dot{vol} C_p} = \frac{54589 \frac{BTU}{hr}\left(\frac{1 hr}{60 min}\right)}{\left(.67489 \frac{lbm}{ft^3}\right)\left(800 \frac{ft^3}{min}\right)\left(.2404 \frac{BTU}{lbm\text{-}R}\right)}$$

$$= 63.1°R \Rightarrow T_{out} = 70°F + 63.1 = \underline{\underline{133.1°F}}$$

Water outlet temperature

$$\dot{Q} = \dot{m}_{water} C_p (\Delta T) = \rho \dot{vol} C_p (\Delta T)$$

$$\Delta T = \frac{\dot{Q}}{\rho \dot{vol} C_p} = \frac{54589 \frac{BTU}{hr}\left(\frac{1 hr}{60 min}\right)}{\left(61.4 \frac{lbm}{ft^3}\right)\left(10 \frac{gal}{min}\right)\left(.999 \frac{BTU}{lbm\text{-}R}\right)\left(.13368 \frac{ft^3}{gal}\right)}$$

$$= 11.1°R \Rightarrow T_{out} = 180°F - 11.1 = \underline{\underline{168.9°F}}$$

Log mean temperature difference

$$\Delta T_{LM} = \frac{\Delta T_{in} - \Delta T_{out}}{\ln\left(\frac{\Delta T_{in}}{\Delta T_{out}}\right)}$$

$$\Delta T_{in} = (180 - 70)°F = 110°F$$

$$\Delta T_{out} = (168.9 - 133.1)°F = 35.8°F$$

$$= \frac{110°F - 35.8°F}{\ln\left(\frac{110°F}{35.8°F}\right)}$$

$$\Delta T_{LM} = \underline{\underline{66.1°F}}$$

A-6 Heat Exchanger Performance Part 2

JEREMY DICKSON | META95 | 2/7/15

Air temp correction

$$R = \frac{180-168.9}{133-70} \quad \begin{matrix} \text{Water} \\ \text{Air} \end{matrix}$$

$$= .176$$

$$P = \frac{133-70}{180-70}$$

$$= .573$$

Fig 22-18
pg 943
Thermo-fluids

$$F = .97$$

UA Value

$$\dot{Q}_{Air} = [UA](\Delta T_{LM} F)$$

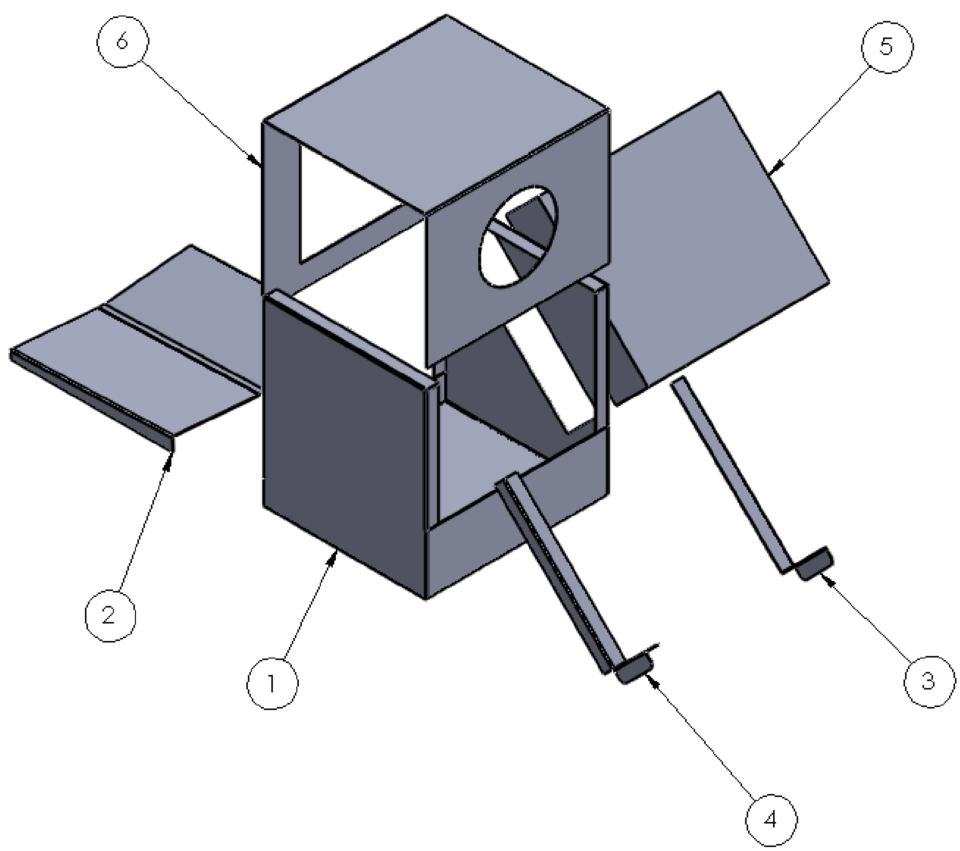
$$[UA] = \frac{\dot{Q}_{Air}}{(\Delta T_{LM} F)}$$

$$= \frac{54589 \text{ BTU/hr}}{(66.1°F)(.97)}$$

$$\boxed{[UA] = 851 \frac{BTU}{hr\text{-}°F}}$$

Appendix B – Drawings
B-1 Drawing No. C-ASM-01

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	C-01	Evaporator Box Base	1
2	C-04	Evaporator Base Tray	1
3	C-05	Heat Exchanger Bracket Right	1
4	C-06	Heat Exchanger Bracket Left	1
5	C-03	12X12 Heat Exchanger	1
6	C-02	Evaporator Box Top	1

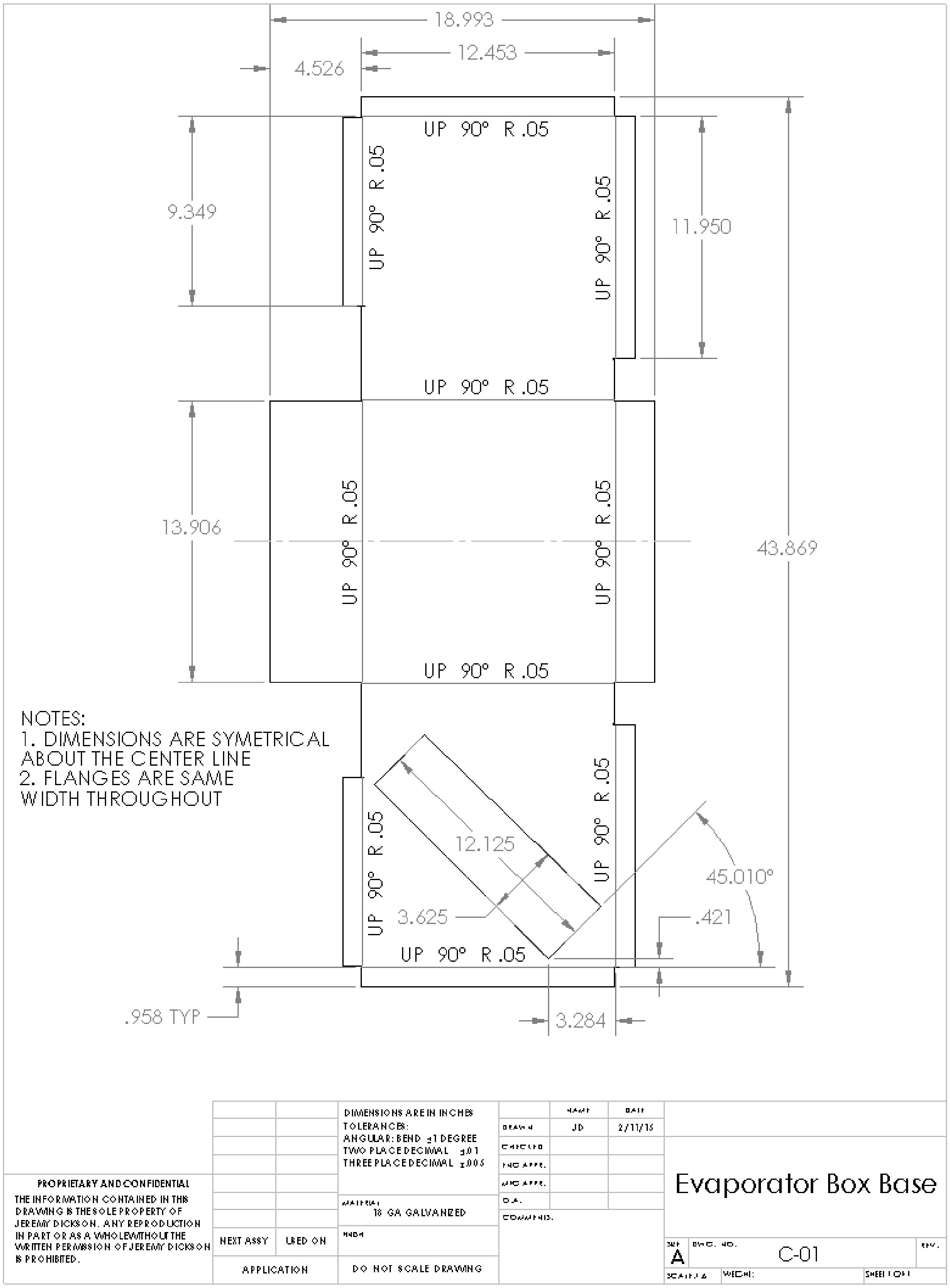


Evaporator Assembly

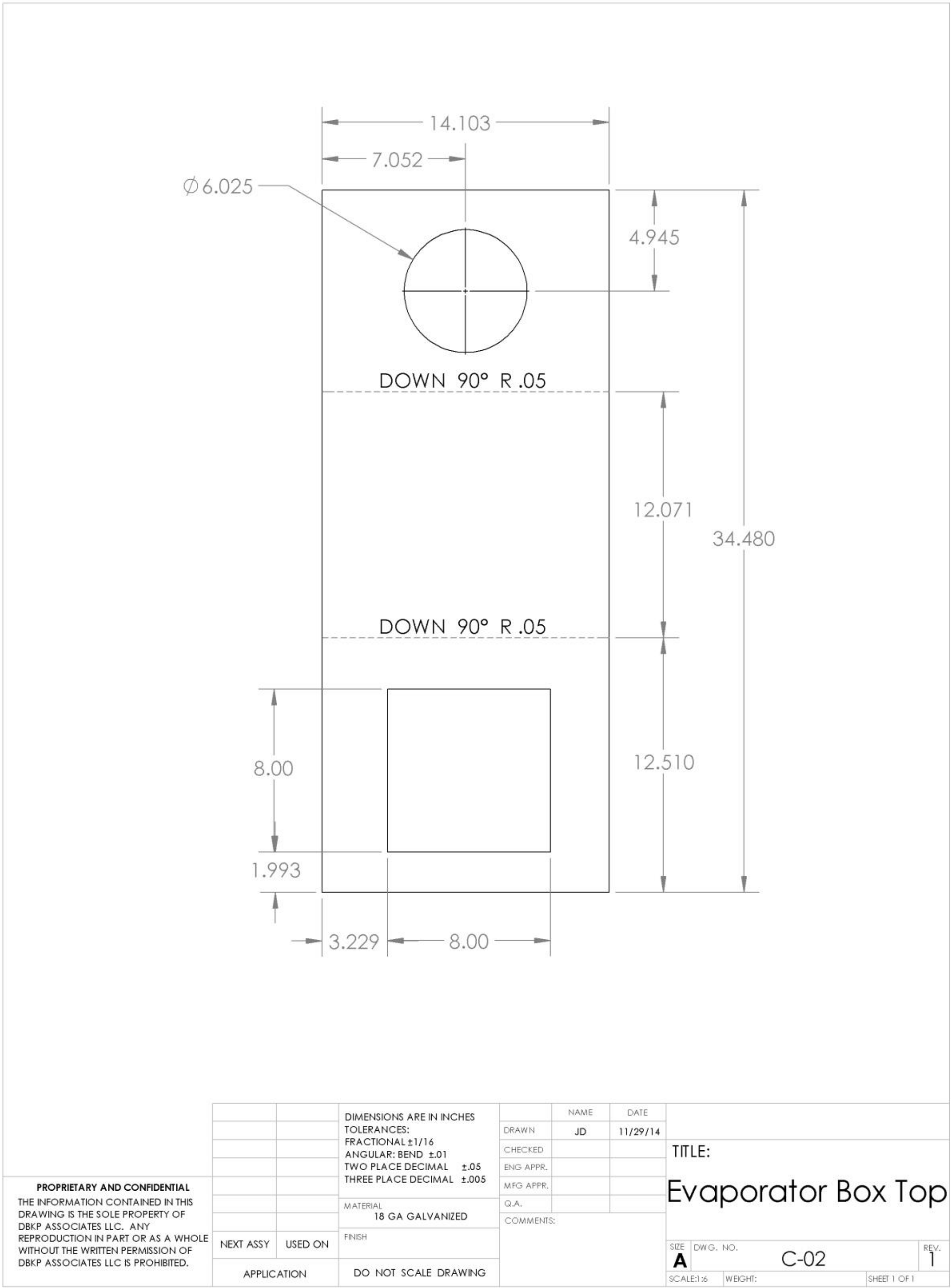
SIZE A | DWG. NO. C-ASM-01 | REV.

SCALE: | WEIGHT: | SHEET 1 OF 1

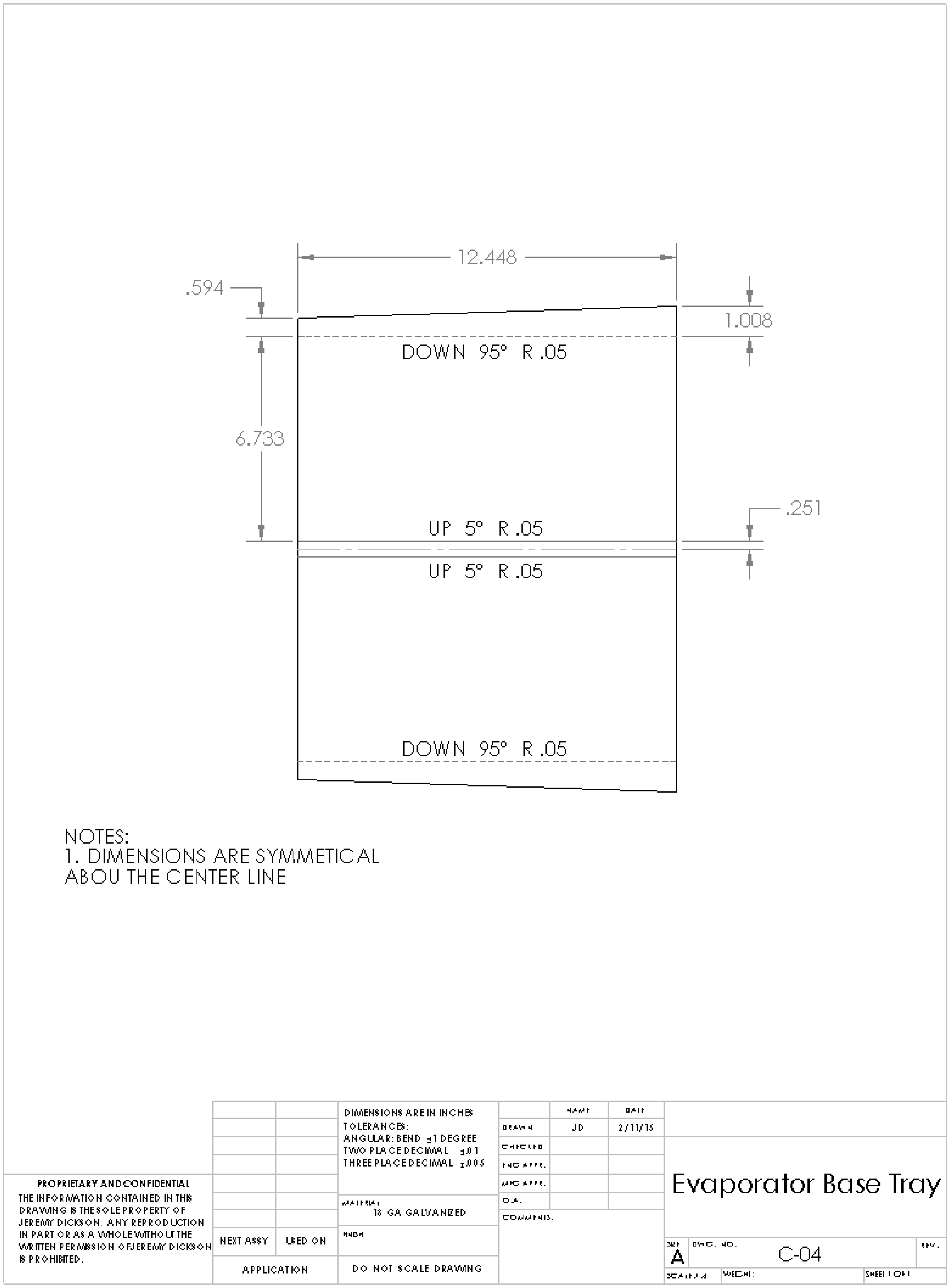
B-2 Drawing No. C-01



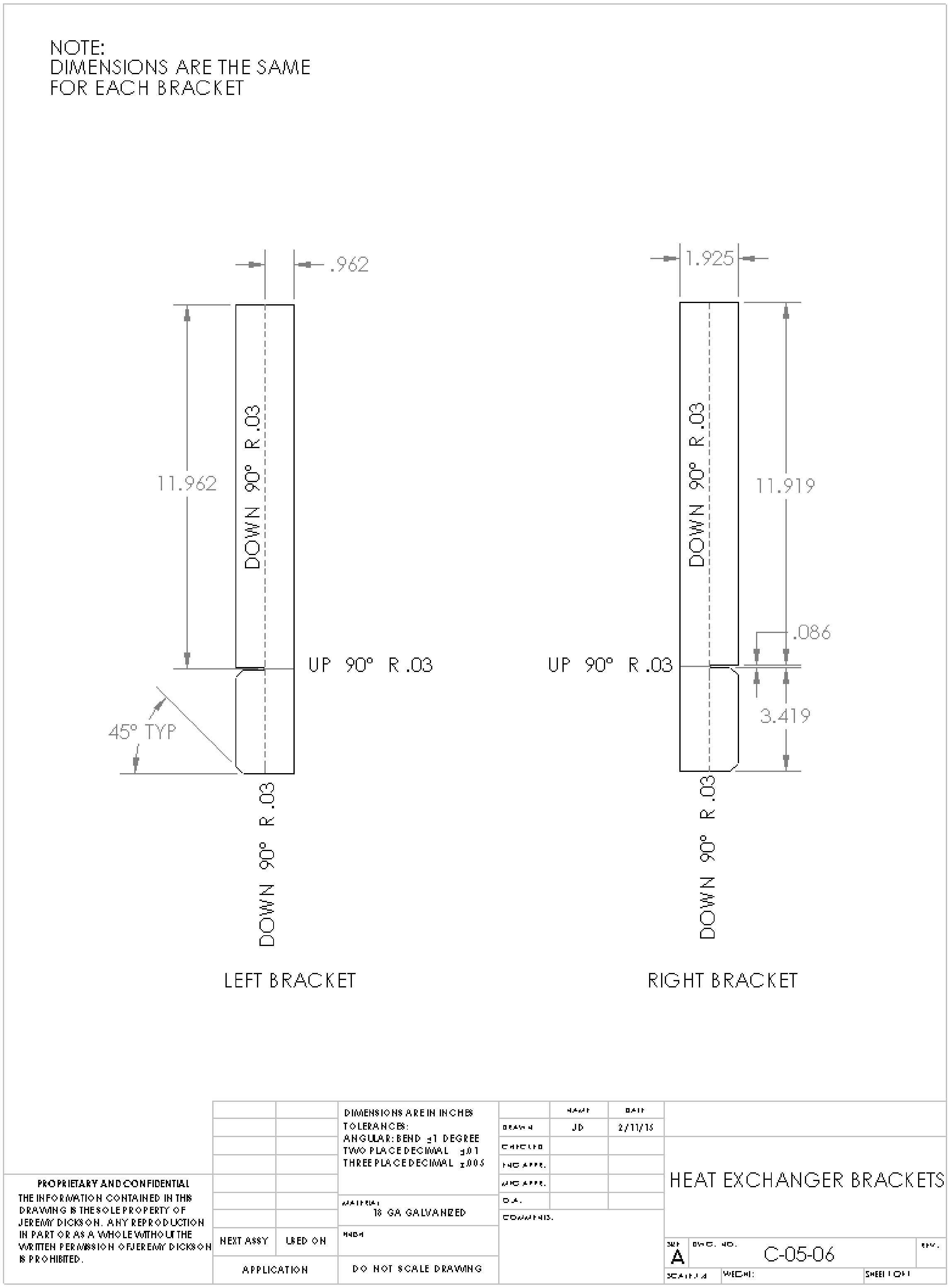
B-3 Drawing No. C-02



B-4 Drawing No. C-04



B-5 Drawing No. C-05-06



Appendix C – Numerical Code
C-1 – Part C-01

```
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(Import File:  Flat pattern - C-01 - For Cut.dxf)
(Import Date:  12/2/2014)

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N20 G01 X5.0767 Y42.4014 F80.00
N30 G01 X2.6018 Y39.9265
N40 G01 X11.4407 Y31.0877
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N60 G01 X13.9671 Y33.5356
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C-2 – Part C-02

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N1790 G01 X-0.0748 Y-0.1116
N1800 G01 X-0.0599 Y-0.0992
N1810 G01 X-0.0440 Y-0.0788
N1820 G01 X-0.0337 Y-0.0551
N1830 G01 X-0.0295 Y-0.0295
N1840 G01 X-0.0295 Y34.5098
N1850 G01 X14.1325 Y34.5098
N1860 G01 X14.1325 Y-0.0295
N1870 G01 X-0.0295 Y-0.0295
N1880 G01 X-0.0485 Y-0.0326
N1890 G01 X-0.0667 Y-0.0390
N1900 G01 X-0.0834 Y-0.0485
N1910 G01 X-0.0982 Y-0.0609
N1920 G01 X-0.1140 Y-0.0810
N1930 G01 X-0.1246 Y-0.1044
N1940 G01 X-0.1295 Y-0.1295
N1950 G01 X-0.1103 Y-0.1270
N1960 G01 X-0.0918 Y-0.1210
N1970 G00 X0.0000 Y0.0000
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C-3 – Part C-03

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(File created using Torchmate DXF Import)
(Import File:  Flat pattern - C-03 - For Cut.dxf)
(Import Date:  12/2/2014)

N10 G00 X7.3628 Y9.3082
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N30 G01 X7.3410 Y9.3401
N40 G01 X7.3206 Y9.3560
N50 G01 X7.2969 Y9.3663
N60 G01 X7.2714 Y9.3705
N70 G01 X7.2714 Y7.4295
N80 G01 X7.7142 Y7.4295
N90 G01 X7.7142 Y9.3705
N100 G01 X7.2714 Y9.3705
N110 G01 X7.2745 Y9.3515
N120 G01 X7.2809 Y9.3333
N130 G01 X7.2904 Y9.3166
N140 G01 X7.3027 Y9.3018
N150 G01 X7.3229 Y9.2860
N160 G01 X7.3462 Y9.2753
N170 G01 X7.3714 Y9.2705
N180 G01 X7.3689 Y9.2897
N190 G01 X7.3628 Y9.3082
N200 G00 X7.6519 Y11.6790
N210 G01 X7.6689 Y11.6884
N220 G01 X7.6839 Y11.7008
N230 G01 X7.6997 Y11.7212
N240 G01 X7.7100 Y11.7449
N250 G01 X7.7142 Y11.7705
N260 G01 X7.2714 Y11.7705
N270 G01 X7.2714 Y9.8295
N280 G01 X7.7142 Y9.8295
N290 G01 X7.7142 Y11.7705
N300 G01 X7.6952 Y11.7674
N310 G01 X7.6770 Y11.7610
N320 G01 X7.6603 Y11.7515
N330 G01 X7.6455 Y11.7391
N340 G01 X7.6297 Y11.7190
N350 G01 X7.6191 Y11.6956
N360 G01 X7.6142 Y11.6705
N370 G01 X7.6334 Y11.6730
N380 G01 X7.6519 Y11.6790
N390 G00 X7.3628 Y6.9082
N400 G01 X7.3534 Y6.9252
N410 G01 X7.3410 Y6.9401
N420 G01 X7.3206 Y6.9560
N430 G01 X7.2969 Y6.9663
N440 G01 X7.2714 Y6.9705
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N460 G01 X7.7142 Y5.0295
N470 G01 X7.7142 Y6.9705
N480 G01 X7.2714 Y6.9705
N490 G01 X7.2745 Y6.9515
N500 G01 X7.2809 Y6.9333
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N600 G01 X7.3410 Y4.5401
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N620 G01 X7.2969 Y4.5663
N630 G01 X7.2714 Y4.5705
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N650 G01 X7.7142 Y2.6295
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N670 G01 X7.2714 Y4.5705
N680 G01 X7.2745 Y4.5515
N690 G01 X7.2809 Y4.5333
N700 G01 X7.2904 Y4.5166
N710 G01 X7.3027 Y4.5018
N720 G01 X7.3229 Y4.4860
N730 G01 X7.3462 Y4.4753
N740 G01 X7.3714 Y4.4705
N750 G01 X7.3689 Y4.4897
N760 G01 X7.3628 Y4.5082
N770 G00 X7.3628 Y2.1082
N780 G01 X7.3534 Y2.1252
N790 G01 X7.3410 Y2.1401
N800 G01 X7.3206 Y2.1560
N810 G01 X7.2969 Y2.1663
N820 G01 X7.2714 Y2.1705
N830 G01 X7.2714 Y0.2295
N840 G01 X7.7142 Y0.2295
N850 G01 X7.7142 Y2.1705
N860 G01 X7.2714 Y2.1705
N870 G01 X7.2745 Y2.1515
N880 G01 X7.2809 Y2.1333
N890 G01 X7.2904 Y2.1166
N900 G01 X7.3027 Y2.1018
N910 G01 X7.3229 Y2.0860
N920 G01 X7.3462 Y2.0753
N930 G01 X7.3714 Y2.0705
N940 G01 X7.3689 Y2.0897
N950 G01 X7.3628 Y2.1082
N960 G00 X-0.0918 Y-0.1210
N970 G01 X-0.0748 Y-0.1116
N980 G01 X-0.0599 Y-0.0992
N990 G01 X-0.0440 Y-0.0788
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N1100 G01 X-0.1140 Y-0.0810
N1110 G01 X-0.1247 Y-0.1044
N1120 G01 X-0.1295 Y-0.1295
N1130 G01 X-0.1103 Y-0.1270
N1140 G01 X-0.0918 Y-0.1210
N1150 G00 X0.0000 Y0.0000
```

Appendix D – Parts List and Budget
D-1

Parts List and Budget

Part #	Item Description	Source	Manufacturer	Model #	Estimated Price/Cost (US Dollars) ($/hr)	Quantity (or hrs)	Actual Costs: Subtotals
C-01	18 GA Galvanized Sheet Metal	Haskins Steel	N/A	N/A	$115.00	24" X 48"	$48.75
C-05							
C-06							
C-02	18 GA Galvanized Sheet Metal					18" X 48"	
C-04	18 GA Galvanized Sheet Metal					18" X 18"	
C-03	12"X12" Water to Air Heat Exchanger	Amazon	Brent Industries	HWC-12X12	$113.99	1	$69.00
C-05	12V DC 5L/min 60W Micro Diaphragm High Pressure Water Pump	Amazon	AUBIG	EAN: 0520345227233	$29.99	1	$29.99
C-06	Orbit 3/8" Slip Lok Coupling w/Nozzle	Amazon	Orbit	B000YAFTLG	$7.98	1	$7.98
C-07	SharkBite Threaded Male Adapter, 3/4" by 1"	Amazon	SharkBite	UC139LFA	$23.40	3	$23.40
C-08	3/4" PEX Tubing	Jeff Greear			$0.00	20 ft	$0.00
C-09	7/8" Stainless Steel Cinch Clamp (10 Pack)	Amazon	Precision Brand	B007Q4YDOC	$10.26	1	$3.85
C-10	1" Copper Female Adapter	Amazon	Elkhart	30160	$10.74	2	$14.28
C-11	10mm Nylon 12 Flexible Metric Tubing	Amazon	Small Parts	B000FOV05A	$9.79	10 ft	$11.16
C-12	6" Sheet Metal Ductboard Collar	Amazon	Sheet Metal Co.	ASIN: B00BG0VVYM	$11.99	1	$11.99
C-13	Self Drilling Sheet Metal Screws	CWU ETSC			$9.67	1	$0.00
C-14	Silicone Sealant	CWU ETSC			$7.13	1	$0.00
C-15	Arizona Mist 3/8" Brass Slip Lok End Plug	Amazon	Orbit	B0044FUMGY	$1.89		$1.89
				Total	$351.83		$222.29
					Difference of $129.54		

D-2 Heat Exchanger Spec Sheet

BRAZETEK HEAT EXCHANGERS

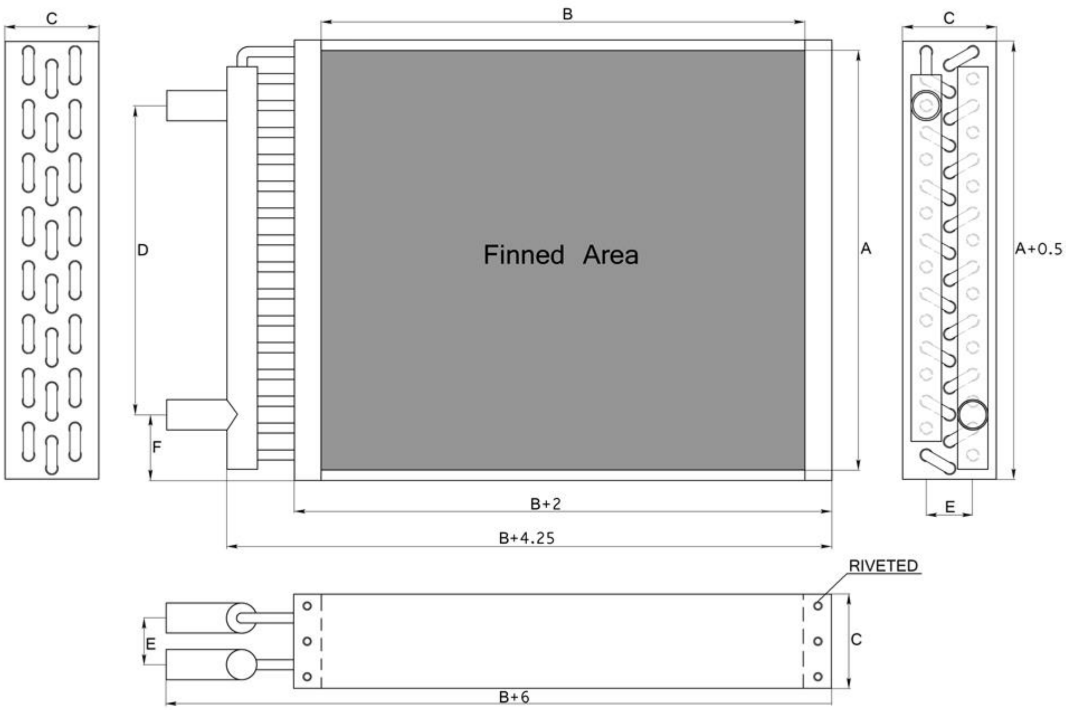
FINNED COIL HEAT EXCHANGERS TECHNICAL SPECIFICATIONS SHEET

Materials of Construction:
Manifold & Inlet/Outlet Connections: 1" copper (1-1/8"OD)
Coil Tube: 3/8" seamless copper
Casing: 18 gauge galvanized steel
Fins: 0.004" thick aluminum, 12 fins per inch

Performance Data:
Max Operating Temperature: 350F
Max Test Pressure: 435psi
Max Operating Pressure: 360psi

Applications:
- Hot Water Heating
- Chilled Water Cooling
- Other Liquid to Air Heat Transfer Applications

Features:
- Standard high capacity output
- Maintenance free
- Compact & sturdy design



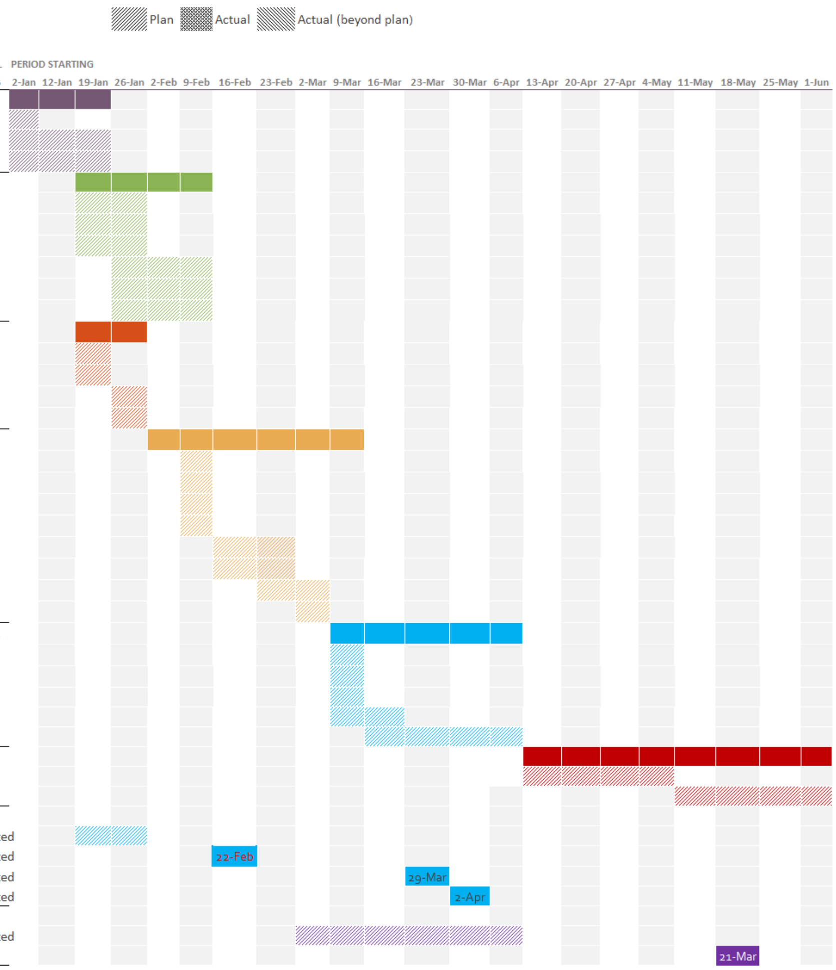
Model	A (inches)	B (inches)	C (inches)	D (inches)	E (inches)	F (inches)	System connection	Weight (lbs)	Box Dimensions (L x W x H)
BT-HTL12x12	12"	12"	3.50"	7.75"	1.75"	2.40"	1" Sweat	10.5	13"x21"x4"
BT-HTL16x16	16"	16"	3.50"	11.50"	1.75"	2.50"	1" Sweat	15.5	17"x25"x4"
BT-HTL16x18	16"	18"	3.50"	11.50"	1.75"	2.50"	1" Sweat	16.5	17"x27"x4"
BT-HTL18x18	18"	18"	3.50"	13"	1.75"	2.75"	1" Sweat	18.5	19"x27"x4"
BT-HTL18x20	18"	20"	3.50"	13"	1.75"	2.75"	1" Sweat	19.5	19"x29"x4"
BT-HTL19x20	19"	20"	3.50"	14.50"	1.75"	2.50"	1" Sweat	20	20"x29"x4"
BT-HTL20x20	20"	20"	3.50"	14.50"	1.75"	3"	1" Sweat	22	21"x29"x4"
BT-HTL22x22	22"	22"	3.50"	17.50"	1.75"	2.50"	1" Sweat	25	23"x31"x4"

Brazetek® Heat Exchangers
Phone: 718-874-0197 Fax: 718-874-0198
www.brazetek.com

Appendix E – Schedule
E-1

Schedule

TASK ID	TASK	PLAN DURATION	PLAN HOURS	ACTUAL HOURS
	Build Prep	**Jan 6-23**	**14**	**6.25**
1a	Order Parts	Jan 6-9	2	1.35
1b	Parts Delivery	Jan 6-23	2	0.3
1c	Prepare drawings	Jan 6-23	10	4.6
2a	**Sheet Metal Fab**	**Jan 19-Feb 13**	**7.5**	**3.78**
2b	Plasma Cut Part C-01	Jan 19-30	1.5	0.46
2c	Plasma Cut Part C-02	Jan 19-30	1	0.39
2d	Plasma Cut Part C-03	Jan 19-30	1	0.4
2e	Bend Part C-01	Jan 26-Feb 13	2	1.23
2f	Bend Part C-02	Jan 26-Feb 13	1	0.8
2g	Bend Part C-03	Jan 26-Feb 13	1	0.5
	Tubing Assembly	**Jan 19-30**	**8**	**2.5**
3a	Cut Pex Tubing to Length	Jan 19-23	2	0.3
3b	Cut Nylon Tubing to Length	Jan 19-23	2	1.2
3c	Attach Pex Tubing Fittings	Jan 26-30	2	0.8
3d	Attach Nylon Tubing Fittings	Jan 26-30	2	0.2
	Part Assembly	**Feb 6-Mar 13**	**21**	**10.16**
4a	Check Fitment of Sheet Metal	Feb 9-13	2	0.1
4b	Spot Weld Part C-01	Feb 9-13	1	0.1
4c	Spot Weld Part C-02	Feb 9-13	1	0.1
4d	Seal Part C-01	Feb 9-13	1	0.2
4e	Assemble Sheet Metal Parts	Feb 16-30	3	1.86
4f	Assemble pump/hoses	Feb 16-30	2	2
4g	Finalize Assembly	Feb 23-Mar 6	3	2.2
4h	Attach to FCU	Mar 2-13	8	3.6
	Testing	**Mar 9-Apr 10**	**23**	**20.24**
5a	Attach Measurement Devices	Mar 9-13	2	0.56
5b	Reconnect Hoses to FCU	Mar 9-13	0.5	0.28
5c	Add Water to Evaporator	Mar 9-13	0.5	0.1
5d	Preliminary Operational Test	Mar 9-20	8	3.3
5e	Data Logging Testing	Mar 16-Apr-10	12	16
	Evaluation	**Apr 13- Jun 5**	**20**	**8.15**
6a	Data Evaluation	Apr 13-May 8	8	2.25
6b	Finalize Project Report	May 11-June 5	12	5.9
	UW EIC			
7a	Registration	Jan 22-Feb 1		Completed
7b	5-7pg Summary Due	22-Feb		Completed
7c	1pg Summary Due	29-Mar		Completed
7d	Presentation	2-Apr		Completed
	SOURCE			
8a	Submission Period	Mar 2-Apr 10		Completed
8b	Presentation	21-Mar		
		Total Hours	93.5	51.08



Appendix F – Testing Data
F-1 - 4/20/2015 Test Data

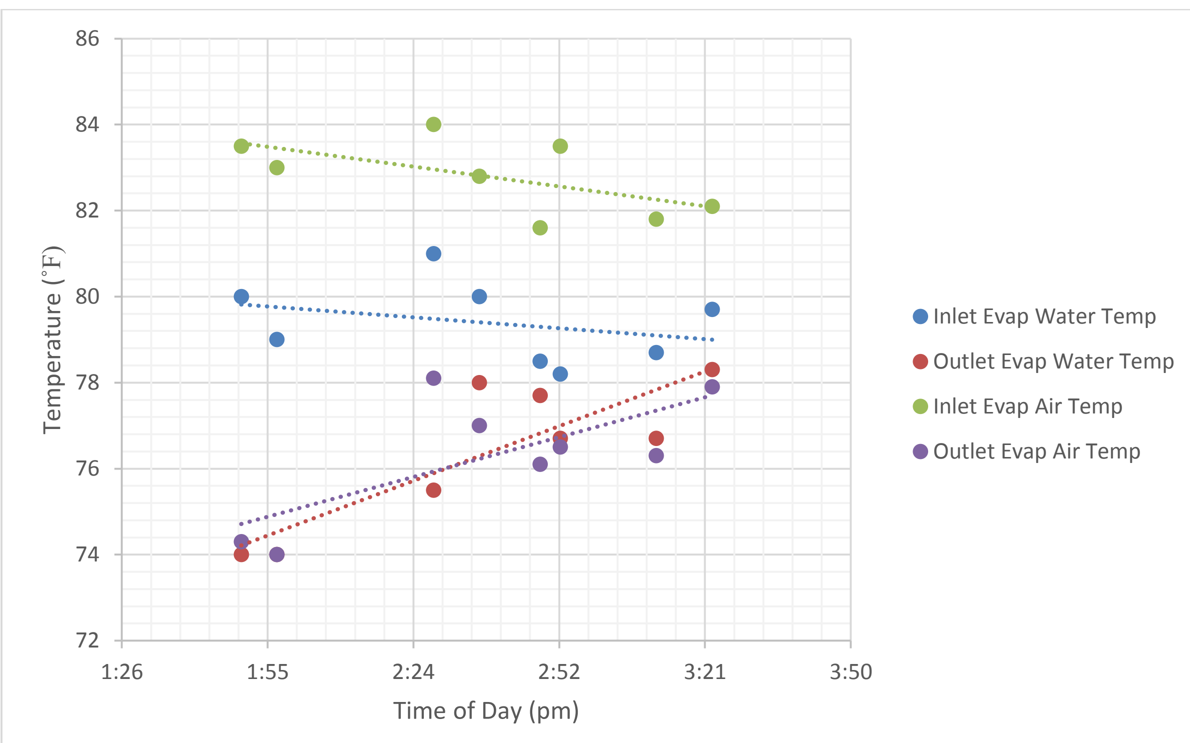
Jeremy Dickson

Evaporative Chiller Data

Date: 4/20/2015

Notes:
VFD Voltage: 230V
Flow Rate: 3 GPM
Used bucket as a reservoir
Fan input direct, rather than through Y duct
All temps in Fahrenheit

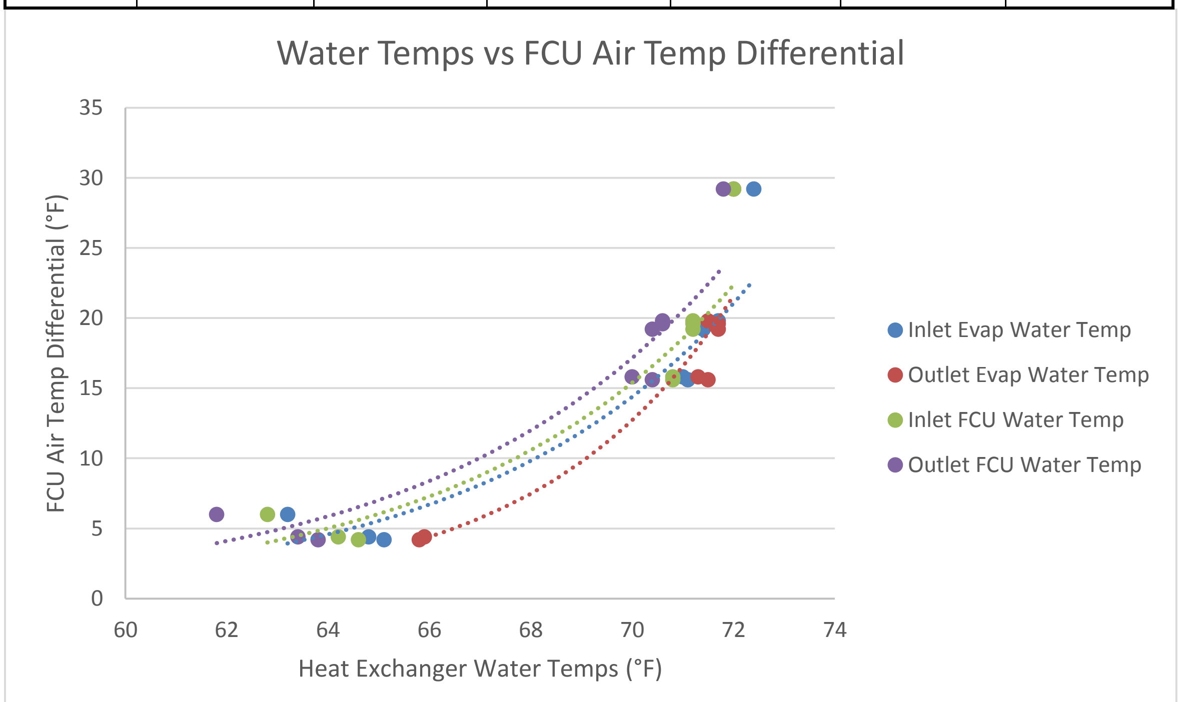
Time	Inlet Evap Water Temp	Outlet Evap Water Temp	Inlet Evap Air Temp	Outlet Evap Air Temp
1:50	80	74	83.5	74.3
1:57	79	74	83	74
2:28	81	75.5	84	78.1
2:37	80	78	82.8	77
2:49	78.5	77.7	81.6	76.1
2:53	78.2	76.7	83.5	76.5
3:12	78.7	76.7	81.8	76.3
3:23	79.7	78.3	82.1	77.9



F-2 – 4/24/2015 Test Data

Jeremy Dickson
Evaporative Chiller Data
Date: 4/24/2015
Notes: Pump Flow Rate: 3 GPM
Evap Fan VFD Voltage: 230V
FCU Fan Flow Rate: 196 CFM
Inside, closed loop, heat gun used for higher temps
All temps in Fahrenheit

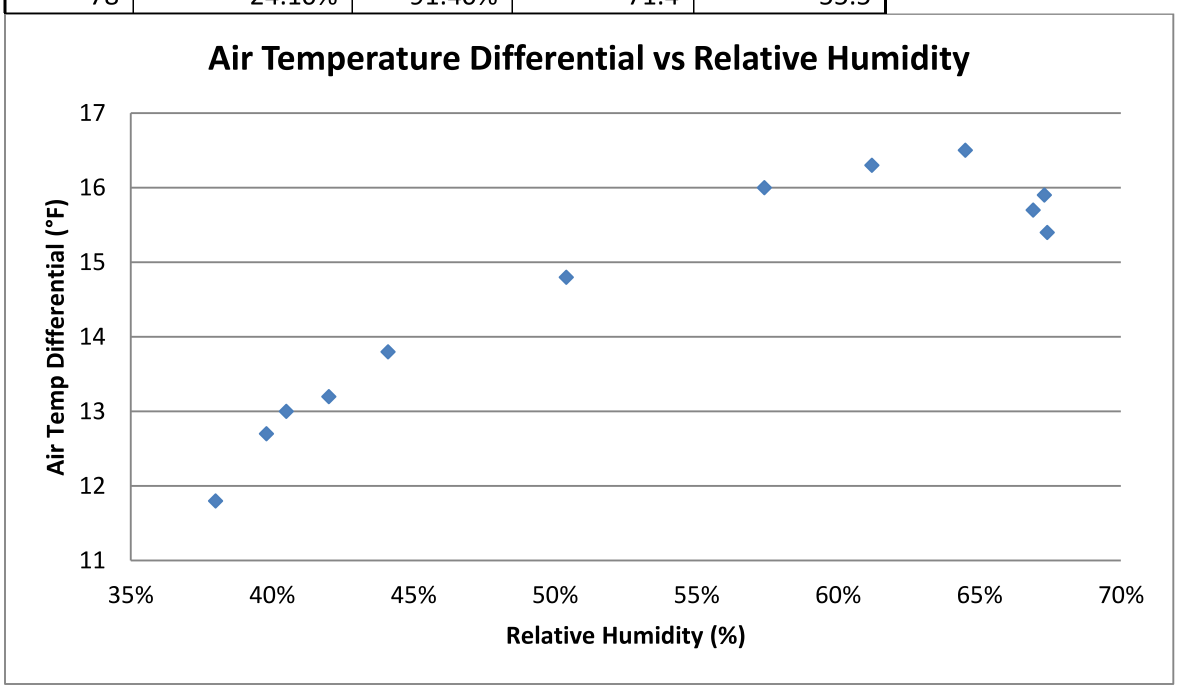
Time	Inlet Evap Water Temp	Outlet Evap Water Temp	Inlet FCU Water Temp	Outlet FCU Water Temp	Inlet FCU Air Temp	Outlet FCU Air Temp
4:15	63.2	54.1	62.8	61.8	69.3	63.3
4:22	65.1	65.8	64.6	63.8	69.6	65.4
4:30	64.8	65.9	64.2	63.4	70	65.6
5:30	71.1	71.5	70.8	70.4	89	73.4
5:37	71.4	71.7	71.2	70.4	91.8	72.6
5:41	71.5	71.7	71.2	70.6	92.2	72.6
5:49	71.7	71.5	71.2	70.6	92.4	72.6
	71	71.3	70.8	70	87.2	71.4
	72.4	72	72	71.8	104.8	75.6



F-4 – 5/6/2015 Test Data

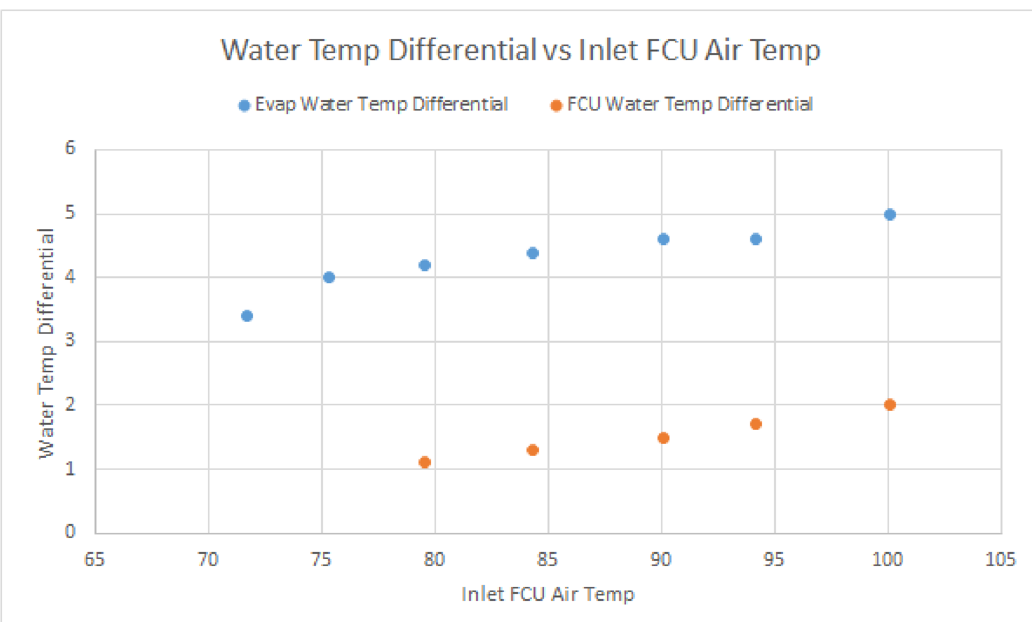
Jeremy Dickson
Evaporative Chiller Data
Date: 5/6/2015
Notes: 3 misting nozzles
No water flow, only evaporator
Misting pump settings: 6.25V 5A

VFD Voltage	Relative Humidity In	Relative Humidity Out	Air Temperature In	Air Temperature Out
230	23%	61%	70.4	58.6
180	23%	65%	70.8	57.6
200	24%	64.50%	70.8	57.8
212	23%	62.80%	71	58.3
159	24%	68.10%	71	57.2
131	23.60%	74%	71.4	56.6
110	23.60%	81%	71.6	55.6
100	22.80%	84%	71.7	55.4
89	23.10%	87.60%	71.8	55.3
68	24.30%	91.20%	71.2	55.5
58	23.60%	91%	71.2	55.8
78	24.10%	91.40%	71.4	55.5



F-5 – 5/13/2015 Test Data

Jeremy Dickson					
Evaporative Chiller Data					
Date:	13-May				
Notes:	VFD: 230V				
	Pump flow: 1 GPM				
Inlet Evap Water Temp	Outlet Evap Water Temp	Inlet FCU Water Temp	Outlet FCU Water Temp	Inlet FCU Air Temp	Outlet FCU Air Temp
68.8	64.2	67.4	68.9	90.1	72.7
70	65.4	68.5	70.2	94.2	74.5
71.2	66.2	69.4	71.4	100.1	76.4
69.6	65.2	67.9	69.2	84.3	71.6
68.2	64	66.7	67.8	79.6	69.9
66.8	62.8	66.2	65.3	75.3	67.7
67	63.6	66.3	65.8	71.7	67.5



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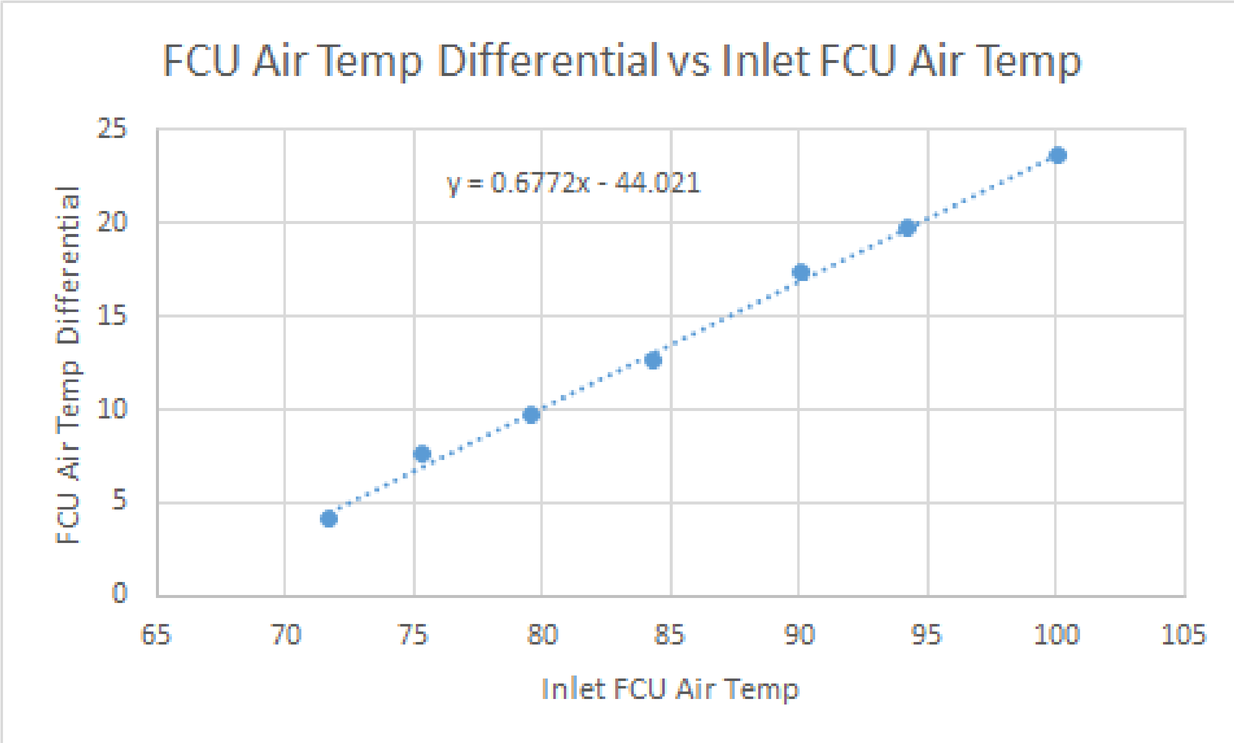

Appendix G – Resume

JEREMY DICKSON
Ellensburg, WA 98926 | (206) 849-6215
jcdickson@gmail.com | www.linkedin.com/in/jeremycdickson

EDUCATION

Bachelor of Science, Mechanical Engineering Technology (ABET Accredited) Expected June 2015
Central Washington University – Ellensburg, WA

- Walter R. Kaminski Memorial Scholarship recipient
- Symposium on Undergraduate Research and Creative Expression:
  2014 Best Presentation Award – Constructed Object Category
  2015 People's Choice Award – Best Poster Presentation

Associate of Arts, Direct Transfer Agreement Graduated 2009
North Seattle Community College – Seattle, WA

EXPERIENCE

Engineering Lab Technician 11/2012 to Present
Central Washington University – Ellensburg, WA
Department of Engineering Technologies, Safety, and Construction

- Maintain machinery, tools, and instruments in labs and shops
- Design and/or fabricate tooling, fixtures, jigs, and other hardware
- Supervise and provide technical instruction for students in machine shop during open lab

Controls Technician Intern 4/2014 to 9/2014
ATS Automation – Ellensburg, WA

- Performed start-up functions based on project plans, specifications, and other contract documents
- Developed CAD drawings and compiled project management documentation
- Provided analysis and troubleshooting of building control and HVAC system performance

Volunteer Various Events 11/2012 to 4/2015
FIRST Robotics and VEX Robotics Competitions
Central Washington University – Ellensburg, WA

- Coordinated with regional planning committee to create marketing material and informational website
- Assisted with technical operation of several events throughout the school year
- Served as scorekeeper, judge, and DJ for multi-day regional competition held in March of each year

TECHNICAL SKILLS

- SolidWorks (Certified SolidWorks Associate), AutoCAD, MasterCAM, MS Office
- Machining, fabricating, rapid prototyping
- Familiarity with lean manufacturing, 5S, and six sigma practices

LEADERSHIP

Society of Manufacturing Engineers (SME) 11/2013 to Present

- Student chapter founding member and President from Nov 2012 to June 2014
- Project Manager of an interdisciplinary team of 4 students awarded $1000 in prototype funding and selected to compete in the UW Environmental Innovation Challenge 2 years in a row
- Fundraised $500 for club activities selling custom designed, CNC machined bottle openers

American Society of Mechanical Engineers (ASME) 9/2012 to Present

- Student Chapter Secretary from September 2013 to June 2014
- Took role, meeting notes, and kept all club paperwork filed and organized
- Worked with other club officers to plan and organize club activities and meetings