DESIGN AND CONSTRUCTION OF A HYDRONIC CONDITIONING LOOP FOR A PSYCHROMETRIC TESTING FACILITY

By

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DESIGN AND CONSTRUCTION OF A HYDRONIC CONDITIONING LOOP
FOR A PSYCHROMETRIC TESTING FACILITY

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Abstract:

Heat exchanger effectiveness is a major contributor to overall system performance, yet heat exchangers are often the least understood component of the cycle. The major contributing factor to the current gap in heat exchanger knowledge is directly attributed to the difficulty of full scale heat exchanger testing. A novel facility is being constructed at Oklahoma State University to allow the investigation of large commercial heat exchanger coils. Data collected will include baseline performance with different refrigerants, heat exchanger modifications, and frosting scenarios. To collect this data steady-state testing conditions are achieved through the use of a closed loop facility with an integrated air conditioning system.

The role of the conditioning system is ensuring the air passing over the heat exchanger being tested is held at constant temperature and humidity per the heat exchanger testing requirements. This is achieved through the use of heat pumps connected to heat exchangers supplemented by additional downstream conditioning equipment. Currently the conditioning equipment will allow testing from 0°F to 140°F with approximately 30 tons of capacity at 70°F inside air temperature. Utilizing the new facility, coil manufacturers will be able to quickly evaluate new design implementations and refrigerants which will further the advancement of technology in this important field.

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CHAPTER I

Introduction

Improved heat exchangers are crucial to reducing equipment energy consumption. Their impact will be felt in reduced primary energy consumption and minimal ozone depletion from harmful refrigerant leakage. In order to realize these benefits research must be undertaken to drive the heat exchanger industry forward. Heat exchanger testing is an expansive field; however, most research can be divided into two main subcategories: existing component testing and new product development. Heat exchangers are implemented in thermal equipment covering nearly every facet of life but this investigation focuses on heating, ventilation, and air-conditioning (HVAC) applications. To advance the global understanding of heat exchangers, (Kincheloe, 2019) constructed a new psychrometric heat exchanger coil testing facility at Oklahoma State University. This new facility required a psychrometric conditioning system for dynamic heat exchanger testing. At least initially, microchannel heat exchangers will be the primary heat exchanger type explored. The following chapters describe the design and construction of the conditioning loop supporting Kincheloe's testing facility.

1.1 What is a heat exchanger?

Before diving into how we can begin to fill the heat exchanger improvement void one must first understand what a heat exchanger does. According to Merriam-Webster, a heat exchanger is a device for transferring heat from one fluid to another without allowing them to mix (Merriam-Webster, 2016). The author duly notes the obviousness of this definition; however, it allows one to begin visualizing the role of these components in the traditional four component heat pump cycle. The regulation of air temperature and humidity to spaces, hereto referred to as air-conditioning, is the purpose of four component unitary equipment, of which heat exchangers play an integral part. A sample four component air conditioning setup is shown below in Figure 1.1 which shows the evaporator cooling the air to be delivered to a space. In a similar fashion, the condenser is removing heat from the refrigerant to the outside air. The roles are reversed when warm air is desired in the space. There are many different types of heat exchangers even within this narrowly defined role, this facility will initially consider microchannel heat exchangers.

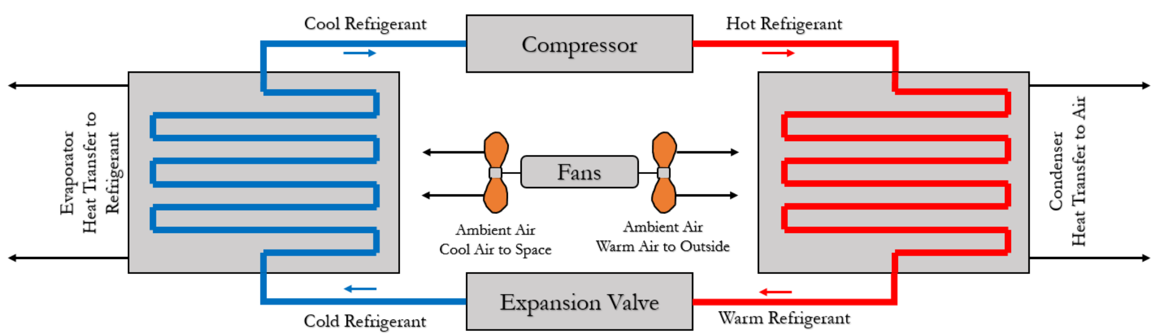

Figure 1.1: Heat exchangers are directly responsible for air-conditioning (Skwiot, 2019).

1.2 Why test heat exchangers?

Like any cycle, the performance of an air-conditioner cycle is more than simply a sum of its constituent parts', different parts affect the overall system in different ways and must be investigated as such. In order to improve overall system efficiency, individual component testing must occur to measure the impact of new design changes. In the realm of heat exchangers these design changes include the implementation of new low-GWP refrigerants, heat exchanger circuitry changes, and the physical footprint. Heat exchanger designers must balance heat exchanger effectiveness and pressure drop to achieve the desired tradeoff between system efficiency and system size (G. Musgrove,

S. Sullivan, D. Shiferaw, P. Fourspring, and L. Chordia, 2017). While a general understanding of heat exchangers has been achieved, new refrigerants and refrigerant blends must now be considered due to Kigali Amendment which has thrust heat exchanger designers into new, previously unexplored territory. The Kigali Amendment to the Montreal Protocol is phasing down the usage of hydrofluorocarbons (HFCs), which were used to replace even more damaging hydrochlorofluorocarbons (HCFCs) and chlorofluorocarbons (CFCs) which were already controlled by the Montreal Protocol (Office of the Inspector General, 2019). These new refrigerants, which must be considered to meet the increased demand of the Kigali Amendment may cause large changes in heat exchanger performance. Much work has already been completed regarding R-22 replacements (Khalid A. Joudi and Qusay R. Al-Amir, 2014) and (Ayyamperumal L. Saravanan, Dhasan M. Lal, and Chandrasekaran Selvam, 2019). Additionally, circuitry changes were researched by Ammar M. Bahman and Eckhard A. Groll (2017) and Madhu S. Emani, Hrishiraj Ranjan, Anand K. Bharti, Josua P. Meyer, and Sujoy K. Saha (2019). These recent studies indicate the need for further research on heat exchangers and the vast support for it.

1.3 How are they tested?

As mentioned previously, there are two primary methods of testing heat exchanger coils. One can test the entire air-conditioning unit or just the heat exchanger in question. When testing just the heat exchanger, research groups have primarily tested just a small section of a heat exchanger coil meant to represent the behavior of the entire coil. But, as Kincheloe (2019) and others have pointed out, this approach limits the ability of the experiment to quantify air and refrigerant maldistribution. In addition, large scale frosting tests similar to the work of Liping (2017) are only possible when the entire coil is tested. The new facility at Oklahoma State is capable of testing entire heat exchanger coils approaching with a 7 ft x 8 ft face area at temperatures ranging from 0°F to 140°F (Kincheloe, 2019). This combination of coil size and operational envelope is unmatched in current published work of similar facilities. Test data is considered valid when the entering dry bulb and wet bulb temperatures do not deviate more than 0.5°F and 0.3°F respectively as per (ASHRAE, 2016). These steady state conditions are achieved in the OSU facility through the use of a conditioning loop subsection which conditions the air back to the testing parameters, this can be seen below in Figure 1.2.

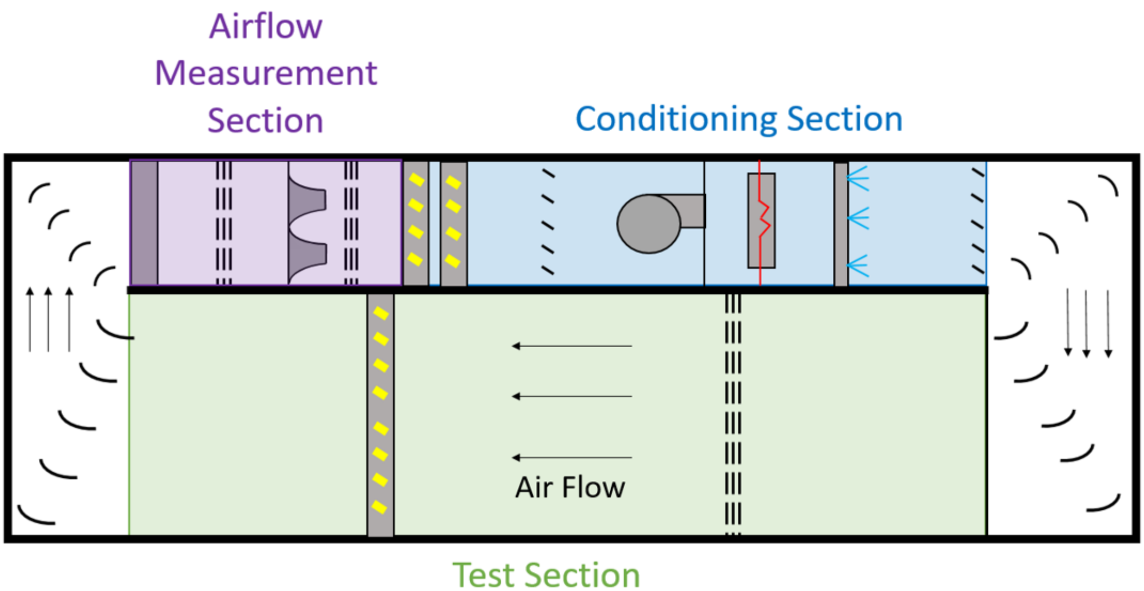

Figure 1.2: The relationship between the testing and conditioning subsections to allow for steady state testing Kincheloe (2019).

The combined facility (airside and conditioning loop) are required to test a heat exchanger. Conditioned refrigerant from a hot-gas bypass chiller enters the test coil and changes the properties of the air at the exit of the test coil. Work on this part of the facility is currently ongoing. Assuming the test coil is acting as a condenser, the air leaving the test coil is now at a higher temperature than at the specific inlet temperature and needs to be cooled in the conditioning section. First, the air passes through the airflow measurement section where the air mass flow rate is calculated by the code tester. The air then passes over the conditioning coils which receive colder than ambient working fluid to cool the air back down below the specified test coil inlet temperature. The airflow stream is then reheated via electric heaters back to the required temperature. Finally, the steam humidification system adds back the humidity lost in the cooling process in the conditioning coils.

1.4 Review of Recent Literature

Currently, there are many studies being published on heat exchanger research, so one might opine the field is well developed and there are many very similar research facilities around the world. This is not necessarily the case due to the broadness of the field and variety of phenomena to be studied. As mentioned previously, heat exchanger testing can vary between the entire air-conditioner unit being tested or just one heat exchanger. When the entire unit is being tested, the most widely utilized approach is a two room psychrometric testing facility similar to the rooftop unit tester at at Oklahoma State (Lifferth, 2009). Current research in these types of facilities include fixed and variable speed testing methodology at Herrick Labs (Andrew L. Hjortland and James E. Braun, 2019), performance using R-22 alternatives at Baghdad University (Khalid A. Joudi and Qusay R. Al-Amir, 2014), and dynamic characteristics of a rotary compressor using R290 at Jiaotong University (Wu et al., 2017) to name a few. These facilities are mentioned because, while they may test heat exchangers in a different manner, the steady state testing conditions are achieved in a similar way to the new closed loop tunnel facility at Oklahoma State. The rooftop testing facility at Herrick Labs uses a secondary conditioning loop using a variable capacity chiller with electric reheat and steam humidification, (Andrew L. Hjortland and James E. Braun, 2019), Lifferth's facility is quite similar as a ground source heat pump (GSHP) takes the place of the variable capacity chiller (Lifferth, 2009). These facilities were used as inspiration for the design of the conditioning loop for the new facility.

A comparison of the new facility at Oklahoma State to other known closed loop wind tunnel type heat exchanger testing facilities was completed by Kincheloe which shows its novelty and is shown below in Table 1.1. This style of testing facility is becoming more common due to the reduced physical footprint and improved airside flow and temperature distribution. Some facilities are open loops on the airside and only utilize water as a working fluid and are thus unable to test conditions under the freezing point of water (Ian Bell and Eckhard A. Groll, 2011). Other test facilities such as the one at POLO Labs, (Christian J.L. Hermes, Valter S. Nascimento Jr., Felipe R. Loyola, Rodrigo P. Cardoso, and Andrew D. Sommers, 2019) are closed loop and employ refrigerant as the working fluid on the conditioning side but are quite small and thus unable to test large commercial heat exchanger coils. The operating envelope and precise capacity control from its conditioning loop is where the new facility really begins to further carve out its niche. These features are the benefit of having independent control of multiple GSHP's acting as chillers in contrast to Lifferth's (2019) facility for example, only having one chiller which then must be modulated in some way. This concept and its implications will be explored further in the following sections. First, the already determined components are examined, then the thermal fluid components, followed by controls and instrumentation, and finally the tests and conclusions. Such a layout depicts not only a fairly correct chronological setup, but also the dependency of each section on the previous one.

Table 1.1: A comparison of similar closed loop wind tunnel type testing facilities Kincheloe (2019).

Author	Testing Area (ft^2)	Air Temp. Range (°F)
ORNL Omega WT4401-D (2019)	0.11	N/A
ORNL High Temperature	N/A	1100°F (maximum)
Markovic et al. (2019)	0.64	N/A
Sun et al. (2019)	0.97	82°F (set point)
Bell & Groll (2011)	2.15	77°F (set point)
DBM Coils & Padoa University (2019)	7.59	5 to 113°F
Oklahoma State University (2019)	56.00	0 to 140°F

CHAPTER II

Predetermined Design Decisions

The main objective of the conditioning loop is to provide constant air properties at the inlet of the heat exchanger being tested. For example, as air passes over that heat exchanger, hereby referred to as the test coil, the temperature will increase and thus the conditioning loop is required to cool the air and vice versa. This quasi-equal and opposite reaction allows for steady state testing conditions for minutes or even hours. A longer test duration is desired as it allows for a better data set to be collected due to outliers falling out and general trends such as the behavior of the test coil to become more apparent. Therefore, the objective of this project is designing and constructing a conditioning loop capable of creating and maintaining steady state air conditions at the inlet of the test coil. To visualize the operation of the conditioning loop, refer to Figure 2.1. Assuming air is heated by the tested coil from 80°F to 95°F, the conditioning coils would cool the air to 75°F, the electric heaters would heat the air back to the set point of 80°F, and finally the steam injection system would replace the humidity removed by the conditioning coils to return the air to the conditions required by the test coil. An example of this relationship is shown below in Figure 2.2.

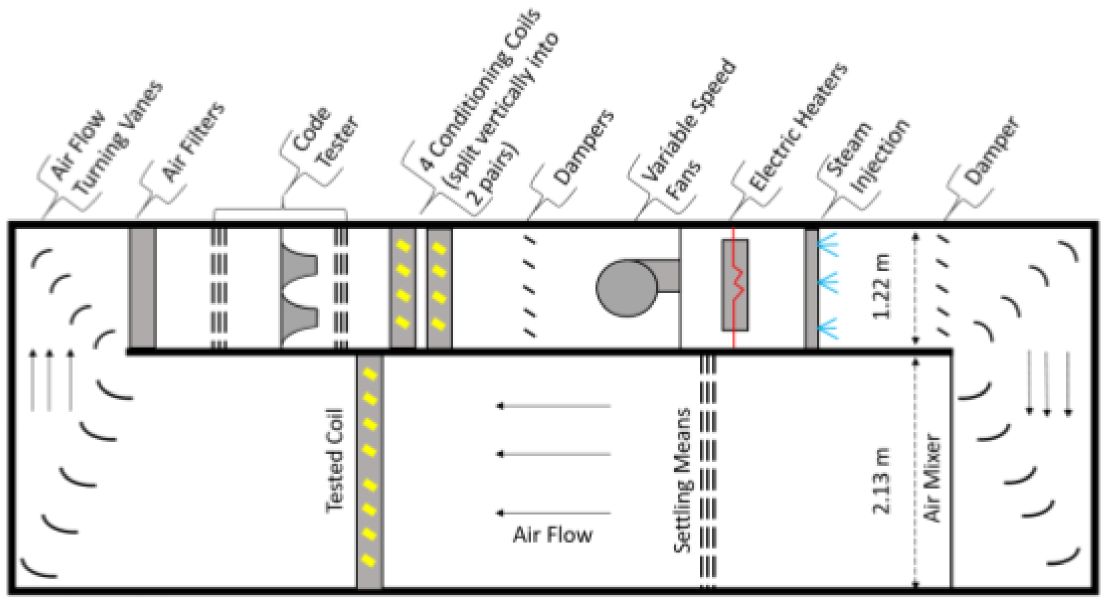

Figure 2.1: This is a top view of the major components of the coil testing facility Kincheloe (2019).

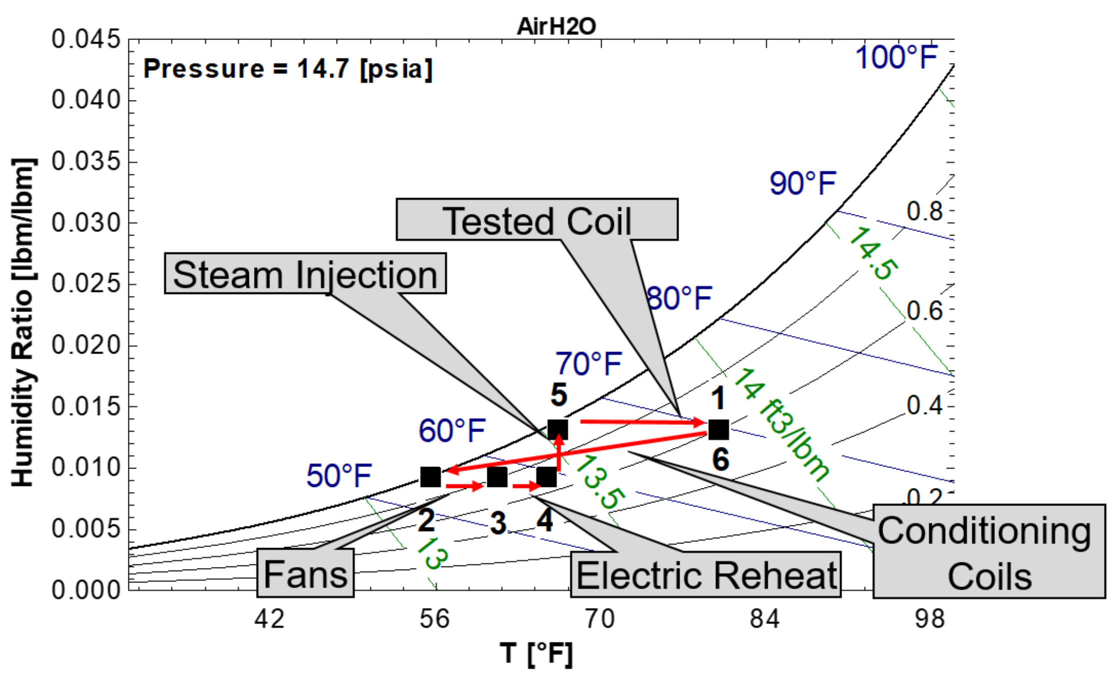

Figure 2.2: An example of the change in humidity and temperature expected inside the testing facility Kincheloe (2019).

The project statement listed above would generally be the starting point of the design phase of a project; however, in this case there were a few other matters to attend to first. This project design did not begin with a clean slate, as there were two major inherent design constraints. These being the composition and location of the conditioning equipment.

2.1 Donated conditioning equipment

The aforementioned major components which were preselected for this project are heat pumps and heat exchangers. During the airside portion of the facility design, it was determined a pair of stacked heat exchangers would best suit the facility. This arrangement can be seen below in Figure 2.3.

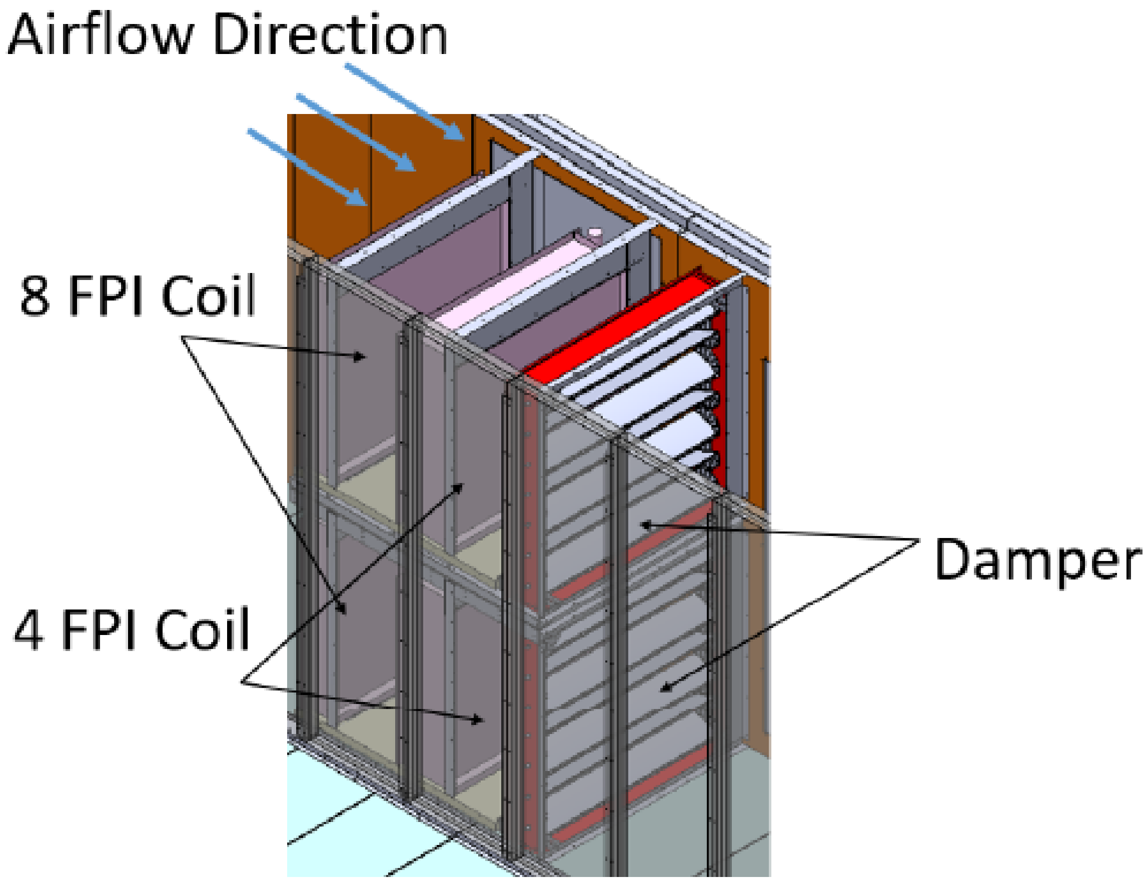

Figure 2.3: The arrangement of the conditioning coils within the airside facility Kincheloe (2019).

These heat exchangers, referred to as the conditioning coils, are also fluid-to-air heat exchangers like the test coil, but not of the microchannel variety. Instead, the conditioning coils, generously donated by RAE Corporation, are designed for utilizing water or a brine mixture as the working fluid. For reasons of construction and control simplicity, it was determined to connect one heat pump to one conditioning coil as seen below in 2.4.

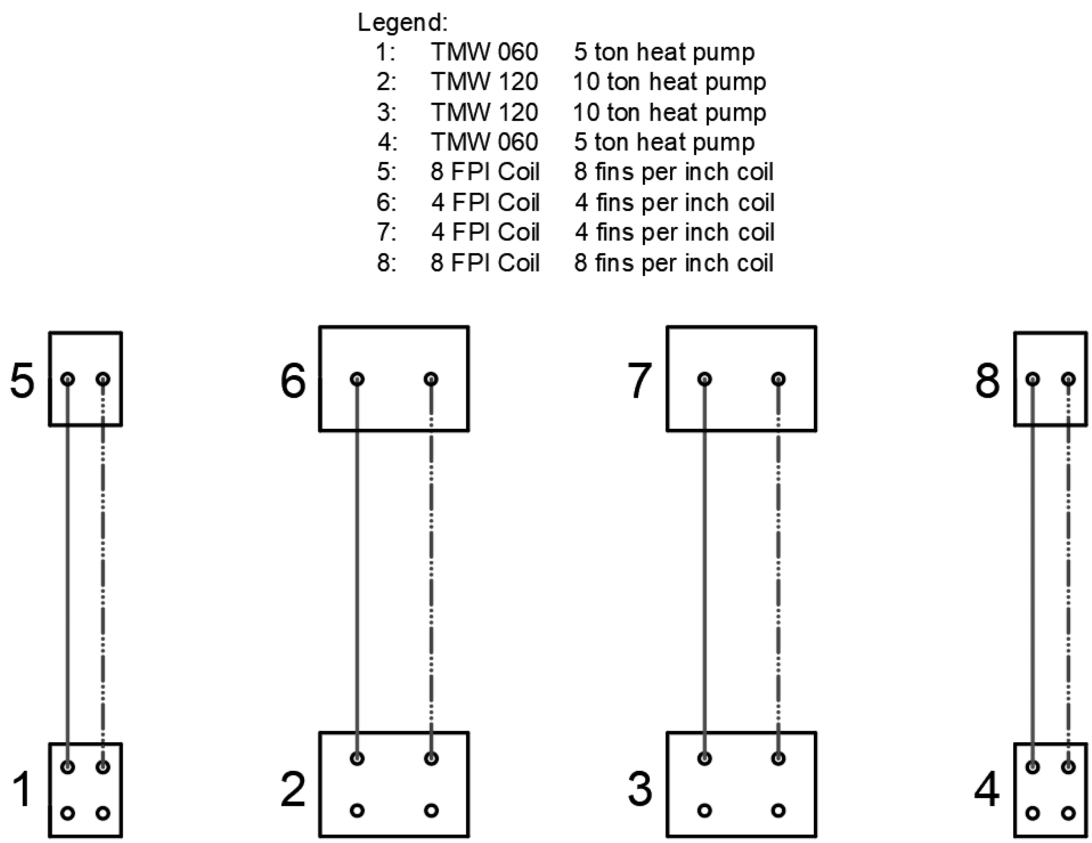

Figure 2.4: The connection scheme for the heat pumps and coils.

ClimateMaster had previously donated many heat pumps of varying types, but due to the design of the conditioning coils already implemented, water-to-water heat pumps were necessary. Frosting on coils can cause large air pressure drops which could cause instabilities due to changing requirements of the blowers as the air flowrate changes. Due to the 8 FPI coils being upstream of the 4 FPI coils when the conditioning coils are in cooling mode at low temperatures, frosting is very likely on the 4 FPI coils. For this reason, the 4 FPI coils are located downstream of the 8 FPI coils, as there will be less air pressure drop through a frosted 4 FPI coil than 8 FPI coil due to the increased spacing between fins. The pressure drop is lower because there is less flow restriction from the frosted fins i.e. the air has more open space to move through the coil. Whereas, the 8 FPI coil could become fully blocked due to frost accumulation if downstream. Ultimately, to maximize the available equipment two TMW060 heat pumps would connect to the 8 FPI coils and likewise two TMW120's to the 4 FPI coils. As previously mentioned, the conditioning coils were part of the airside design;therefore, their size, location, and arrangement were already determined. To allow for future expansion, the heat pumps were placed above the tunnel on a mezzanine. Location and arrangement of the heat pumps can be seen below in Figure 2.5, note the TMW120 heat pumps are the larger heat pumps as they are simply two TMW060 heat pumps in one shell.

Figure 2.5: The placement and location of the heat pumps.

CHAPTER III

Thermal Fluid Design

The majority of technical design decisions in this project revolve around thermodynamic and fluid dynamic considerations. Knowing four heat pumps and four conditioning coils must be connected causes many questions to emerge. What pipe material will be used? What working fluid works best? Or most importantly, which pump can move the fluid in the most efficient manner? All of these questions and more will be answered in chronological order as the decisions build on each other.

3.1 Buffer tank selection?

Hydronic loops often use buffer tanks as a way to distribute the working fluid to the heat exchanger and also increase temperature stability within the loop. A buffer tank is essentially a large insulated holding vessel which is several times larger than the rest of the system volume. Buffer tanks provide many advantages to a hydronic loop such as the ones employed in this project; however, there are some drawbacks in certain applications. The major drawbacks are the large increase in working fluid volume required, as well as, a much larger physical footprint for the entire system. Due to the large fluid volume inside the buffer tank, relative to the rest of the system, the thermal response of the system is slowed. This allows the increase in system stability, but comes at the cost of the ability to quickly move between different testing points in a dynamic system. These drawbacks ultimately led to the decision to not include buffer tanks in the conditioning loop. The project needed to reserve budget for other components, retain fast dynamic response time, and the buffer tanks were simply too large to be easily integrated into the facility location.

3.2 Pipe material selection

Selecting a pipe material is an important decision in piping design as it dictates many future design aspects of the project. A requirement of the material in this project is the ability to operate in a temperature envelope of 0°F to 140°F. An ideal candidate supports a large variety of working fluids. Due to the ever changing nature of research facilities, a material which is widely available was desired. This will prove useful in the event of design changes or assembly mistakes. Finally, in order to avoid downtime associated with material shortages due to leaks or other assembly mistakes, the pipe material needed to require little experience or expertise to assemble. These criteria narrowed the search to copper, stainless steel, and high density polyethylene (HDPE). Polyvinyl chloride (PVC) pipe was not considered due to the large thermal changes required and due to the operating envelope being outside of PVC's long-term capabilities. Stainless steel is a good option except for the expertise and equipment required to assemble it. HDPE suffered a similar complaint while also not being widely accessible.

A few additional factors favored the selection of copper piping. The research group receives a university discount from a local copper pipe distributor. Early costing efforts unveiled copper fittings were on average the cheapest fittings. Choosing the cheaper fittings versus cheaper pipe is because the number, type, or amount of fittings is much more likely to change due to design changes than a large amount of pipe. Finally, the ability to quickly train undergraduate research assistants to a satisfactory performance level was a major selling point. For those reasons copper piping is the best choice for this project.

3.3 Working fluid selection

Choosing the correct working fluid was instrumental in ensuring both the physical and economic success of this project. A working fluid is the fluid or mixture of fluids which acts as the heat transfer medium in the piping system. In this case, the working fluid described is located within the pipes connecting the heat pumps and conditioning coils. Naturally, the working fluid is under the same temperature envelope requirement as the piping, this eliminates water as a working fluid. Another continuing theme is the cost versus performance aspect of component selection. This theme is perhaps best illustrated in this design decision as there are numerous fluids which satisfy the temperature envelope and are compatible with copper pipe, but may not necessarily be the best fit for this project. It is worth noting the volume of the system was estimated to be on the order of tens of gallons, hence the emphasis on cost. Having established the requirements of the working fluid, three fluids were considered on the basis of performance in terms of pressure drop and capacity, toxicity and corrosiveness, and cost. Mineral oil, ethylene glycol (EG), and propylene glycol (PG) were the fluids considered. Mineral oil yields great performance with minimal toxicity; however, it is extraordinarily expensive for the required number of gallons. EG holds a slight edge in performance over PG, is more toxic than PG, but is also less expensive than PG. Ultimately, EG was selected due to its lower cost, better performance, and the toxicity and corrosiveness concerns are only applicable in the case of a leak. There is always a chance for a leak or exposure; however, these situations were mitigated through an initial shakedown leak test with water and usage of proper personal protective equipment (PPE). The initial shakedown test highlighted structural locations which could become vulnerable over time and care was taken to provide extra leakage protection in these areas. A 50/50 EG to water by weight mixture is able to prevent freezing down to -40°F which is the ultimate goal of the conditioning loop. This mixture was used for sizing to enable all selected components to not need replaced in the future when such temperatures become attainable.

3.4 Pipe size selection

The ultimate goal of the thermal fluid design phase is to select a pump for each loop. A system curve, i.e. the variation of pressure drop versus flow rate, is required for accurate pump sizing. Pressure drop is a function of the working fluid physical properties and its velocity. But, to calculate the velocity, one must know the diameter of the pipe used, hence the need to select a pipe diameter at this step in the design process. Selecting the proper pipe diameter requires calculating the estimated pressure drop and fluid velocity in a range of different pipe sizes. These pressure drop and fluid velocities values are then compared to recommended parameters found in the ASHRAE Handbook of Fundamentals (ASHRAE, 2017). The Handbook recommends a maximum fluid velocity of 4 ft/s in pipes with diameters smaller than 2 in to minimize noise due to flow. Additionally, 4 ft of water per 100 ft of pipe is a general rule of thumb for sizing closed loop hydronic systems.

Obtaining pressure drop and fluid velocity in the loop requires a satisfactorily precise estimation of total system length and number of each type of fittings expected. This estimation was possible through the creation and use of the properly scaled model shown below in Figure 3.1.

The pressure drop and velocity calculations were performed using Engineering Equation Solver (EES) due to its built-in library of thermophysical properties which includes the EG mixture of interest. Pressure drop calculations can be separated into major and minor losses.

Major losses are frictional losses encountered in straight pipe runs and are calculated by

$$hl_{major} = f(\frac{L}{D})(\frac{V^2}{2g}). \tag{3.1}$$

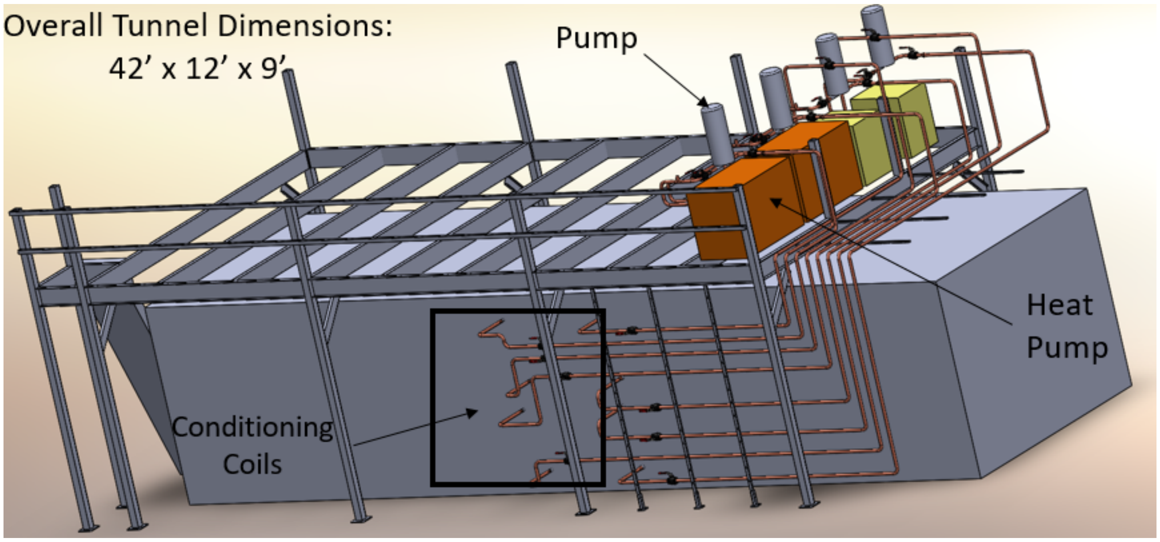

Figure 3.1: The layout of piping and fittings used for pressure drop estimation.

S.E. Haaland's correlation is used to compute the friction factor denoted as f (Neutrium, 2019). The head loss in ft of water is represented by hl, L is the total loop length, D is the pipe inner diameter, V is the fluid velocity, and g is the gravitational constant.

Minor losses are all of the frictional losses not associated with straight pipe runs. Thus, in this project this refers to fittings and losses through the heat pumps and conditioning coils. The two fittings considered are reducers and elbows, each with their own equation. Losses through reducers are due to an increase in velocity associated with the smaller diameter as one can see in the equation below. This equation is from ANSI/ISA (ANSI, 2007) and is calculated from

$$hl_{reducers} = n_{reducers} \cdot 0.5 \cdot (1 - \frac{D_{small}}{D_{large}} \cdot \frac{V_{small}^2}{2g}). \tag{3.2}$$

Elbow losses occur due to flow disruption as the fluid turns the corner. The correction factor of 0.9 is due to large radius elbows assumed for the project. This leads to an elbow loss equation of

$$hl_{elbows} = n_{elbows} \cdot 0.9 \cdot \frac{V^2}{2g}. \tag{3.3}$$

The expected losses through the heat pumps were located in the heat pump performance datasheets. Losses for the conditioning coils were computed by the manufacturer's software, (RAE Corporation, 2017). The losses for both components were based on pure water, but an EG mixture is much more viscous fluid at low temperatures and thus a pressure drop correction factor is needed. The pressure drop calculated for water is multiplied by 1.86 for a 50/50 EG mixture .

The major and minor losses were calculated and then added to the heat pump and conditioning coil losses to estimate total pressure drop in each loop

$$hl_{total} = hl_{major} + hl_{reducers} + hl_{elbows} + hl_{coil} + hl_{heatpump}. \tag{3.4}$$

This was repeated for each of the four loops. A flowrate of 15 gpm and 30 gpm was used in the TMW060 and TMW120 loops respectively as this is the maximum flowrate for which each heat pump's performance is rated. The greater of the two pressure drops for each heat pump loop size was retained for pump sizing. Using a fluid temperature of 0°F, the predicted head losses in the large and small heat pump loops is 40 ft of water and 31 ft of water respectively.

3.5 Pump selection

Selecting a pump for each loop was the pinnacle of the fluid design phase. While being possibly the most important design decision, the pump selection phase was quite brief and straightforward due to the work completed in the pipe sizing phase. Choosing a pump requires matching a pump curve created by the pump manufacturer/distributor to a system curve developed by the end user. The point at which these two curves meet is known as the operating point. The system curve is created by determining the pressure drop at different flow rates which were calculated from the pipe sizing code.

Generally, the operating point is the most extreme condition the pump will face in operation, as this guarantees performance over the operational envelope. This project required a slightly different approach due to the very large temperature envelope and high EG concentration. This combination leads to vastly different operating points at different fluid temperatures. The author acknowledges the existence of large multi-stage pumps, but these were above and beyond the budget of the entire conditioning loop. The most extreme operating point for each loop is considered as 0°F and the max flowrate of the heat pump. This creates two untenable situations. One scenario is the pumps are not able to provide the necessary pressure rise at the low temperature conditions, but are very efficient in normal operating conditions. Or, conversely the pumps are sufficient in the low temperature region, but are grossly over sized and prone to overheating or overdrawing electricity in all other conditions. To avoid such a drastic decision being made, a compromise was reached. By reducing the flowrate expected of the pumps by 1/4, the required pressure rise was reduced significantly, allowing a smaller pump to be applicable. The reduction in flow rate comes from the knowledge that the heat pumps will not likely need to be operating at full capacity in cooling mode at such low fluid temperatures. This is based on the small amount of capacity available from the heat pumps at low temperatures, for this reason increasing the heat in the loop from excess pump energy serves no benefit. The capacity of the test coil to heat the air in such low ambient air temperatures would be much lower than the total cooling capacity available from the heat pumps.

Having established the operating points, all of the necessary sizing parameters were allocated and are as follows. The pumps for the small heat pump loops need to supply 30 ft of water at 11.25 gpm, while the large heat pump loops require 40 ft of water at 22.5 gpm. Additional requirements are the pumps need to be compatible with EG mixtures and capable of 0°F fluid temperatures. Grundfos CRN 3-2 and CRN 5-2 centrifugal pumps were selected for the small and large heat pump loops respectively. Their selection was based on the ability of the supplier to guarantee

EG mixture compatibility and provide pump curves at the EG mixture concentration level. An overview of the pump specifications for the CRN 3-2 and CRN 5-2 in Figure 3.2 and Figure 3.3 respectively.

Conditions of Service	
Flow:	13.9 US GPM
Head:	45.43 ft
Efficiency:	
Liquid:	Ethylene Glycol
Temperature:	-40 °F
NPSH required:	6.49 ft
Viscosity:	3.67 cSt
Specific Gravity:	1.075

Pump Data	
Max pressure at stated temp:	363 psi / 250 °F
Liquid temperature range:	-40 .. 248 °F
Maximum ambient temperature:	104 °F
Approvals:	ANSI/NSF61
Shaft seal:	HQQE
Flange standard:	PJE (Victaulic)
Pipe connection:	1 11/16"
Product number:	96083740

Motor Data	
Rated power - P2:	0.33 HP
Rated voltage:	208-230YY/460Y V
Mains frequency:	60 Hz
Enclosure class:	55 Dust/Jetting
Insulation class:	F
Motor protection:	NONE
Motor type:	71AA
Motor_efficiency:	78.5 %

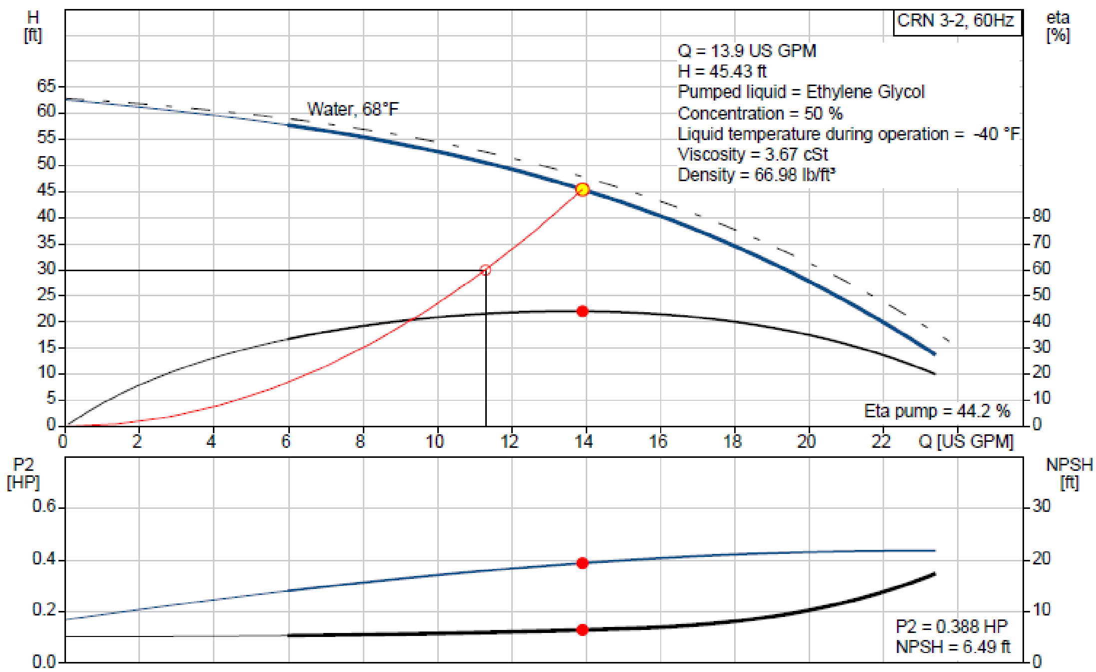

Figure 3.2: An overview of the general specifications of the CRN 3-2 pump are presented here. With permission from Grundfos (2017).

Conditions of Service	
Flow:	22.6 US GPM
Head:	40.37 ft
Efficiency:	
Liquid:	Ethylene Glycol
Temperature:	-40 °F
NPSH required:	6.96 ft
Viscosity:	20.3 cSt
Specific Gravity:	1.089

Pump Data	
Max pressure at stated temp:	363 psi / 250 °F
Liquid temperature range:	-40 .. 248 °F
Maximum ambient temperature:	104 °F
Approvals:	ANSI/NSF61
Shaft seal:	HQQE
Flange standard:	PJE (Victaulic)
Pipe connection:	1 11/16"
Product number:	96084821

Motor Data	
Rated power - P2:	0.75 HP
Rated voltage:	208-230YY/460Y V
Mains frequency:	60 Hz
Enclosure class:	55 Dust/Jetting
Insulation class:	F
Motor protection:	NONE
Motor type:	71BA
Motor_efficiency:	79.0-80.0 %

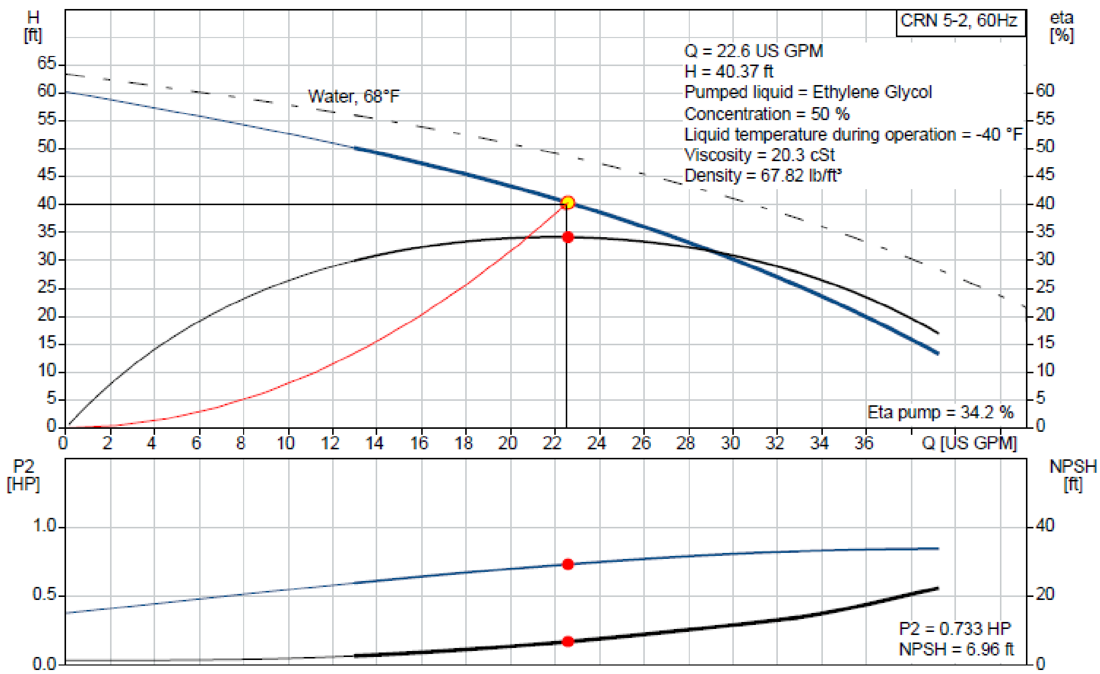

Figure 3.3: An overview of the general specifications of the CRN 5-2 pump are presented here. With permission from Grundfos (2017).

3.6 Thermal expansion tank selection

The last major component to select is the thermal expansion tank (TET). A TET provides additional volume in the piping system to account for expansion or contraction of the working fluid due to large temperature changes. An additional benefit of the TET is the ability to provide the net positive suction head (NPSH) required by the pump to the system. Selection of a TET was based on material compatibility, which can be a cause for concern with some TET types being exposed to EG mixtures, and the required acceptance volume. To find the acceptance volume, the total amount of fluid in each loop was estimated in addition to the change in density of the EG mixture from the highest to lowest operating temperatures. The formula for the acceptance volume is

$$\mathrm{Vol}_{\mathrm{TET}} = \mathrm{Vol}_{\mathrm{total}} - (\mathrm{Vol}_{\mathrm{total}} \cdot \frac{\rho_{\mathrm{cold}}}{\rho_{\mathrm{hot}}}). \tag{3.5}$$

Vol refers to volume and ρ is fluid density. Over-sizing a TET is not a problem, in fact it is generally recommended. This logic was employed in selecting a TET one size larger than necessary to accommodate the required acceptance volume. A membrane style Wilmet ZEP-12 TET was selected for each of the conditioning loops. For those not familiar, the cylinder is split into two separate parts via the tank membrane. The area below the membrane is air charged into the tank which exerts an upward force on the membrane while the area above is the working fluid. When charging the system, the tank was pre-charged with half of the final working pressure with air, then the working fluid was added to the system until it pressed against the membrane up to the required system pressure. The final tank pressure is the net positive suction head (NPSH) of the pump with a couple of extra psi to be safe. A visual description of the discussion above is shown below in Figure 3.4.

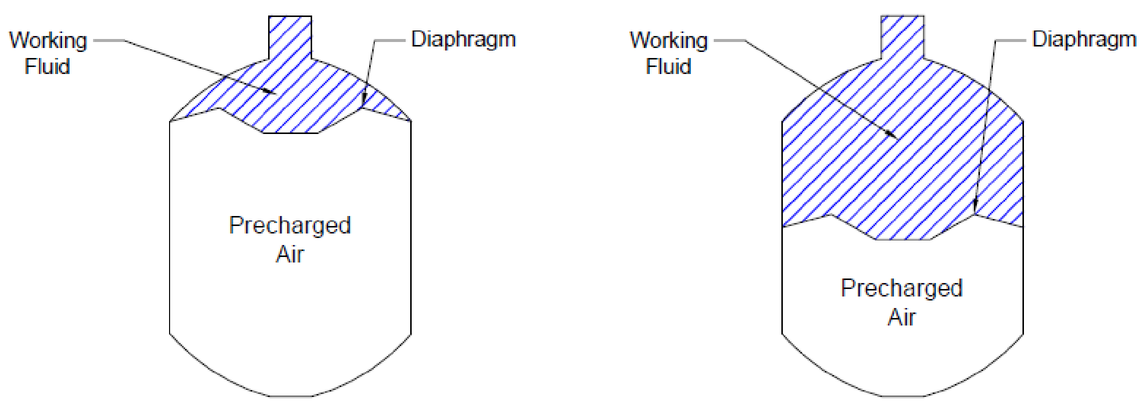

Figure 3.4: A graphical demonstration of the behavior of the TET when the temperature of the working fluid is increased. From Watts (2019).

3.7 Miscellaneous component selection

There were numerous other components added as the project was constructed, the most important of which will be discussed here. One necessary component for any conditioning system is pipe insulation. The role of pipe insulation is primarily protecting against capacity loss to the ambient; however, it also keeps users safe from freeze or burn injuries and can help prevent vapor condensation. Pipe insulation requirements are outlined in ASHRAE Standard 90.1 (ASHRAE, 2010), which recommend between 1" and 1-1/2" insulation for the application of this project. Due to the piping being in a conditioned space 1" insulation was deemed acceptable for this project as per a discussion within the standard. A flexible closed-cell elastomeric insulation was chosen which eliminates the need for an additional vapor retarder (*URI Product Catalog*, 2019).

Filters play an important role in eliminating foreign contaminants from the loops. Each filter utilizes a stainless steel 100-mesh rating screen to accomplish their purposes. The filters are of a y-shaped design allowing them to be cleaned without having to be removed. There is one filter per loop. Due to the setup being built by amateurs and much in situ installation, there is an elevated risk of debris from the soldering process. This necessitates the heavy duty filters being implemented rather than simple wound cotton or other residential cartridge filters. Pressure drop did not prove to be a concern; however, drop in replacements of larger mesh sizes are available if deemed necessary in the future.

Air vents are needed to purge air from the loops to minimize the risk of pump cavitation. The vents are strategically placed at the highest points in the system and at the outlet of the conditioning coils to ensure the maximum possibility of removing air from the system.

Ball valves are located upstream and downstream of the heat pumps, conditioning coils, and pumps. This is done to isolate the major components. Isolation allows for much easier maintenance or replacement of these components. The arrangement of ball valves also paid dividends when leak checking and fixing leaks as it split the loops into smaller and more manageable sections. Each ball valve is equipped with a Schrader port which allows a gauge set to be connected at important system locations. An additional benefit of the Schrader ports is the ability to connect an air compressor to the loop to push the working fluid out when draining the loop is necessary.

CHAPTER IV

Electrical Components

With respect to the overall aim of the testing facility, there is not any test data generated by the conditioning loop; therefore, all electrical equipment is either for controls or safety. Often the two go hand-in-hand as the controlling equipment generally possesses safeties of its own. Due to the inherent difficulty of achieving and maintaining steady state conditions in a large thermal system, there is an inherent need for a automatic controls scheme in this project. The primary goal of this portion of the design phase is to design and construct an inherently safe controls system capable of finding and holding steady state operating conditions with minimal user oversight.

4.1 General safety considerations

The first step in creating an inherently safe system is properly identifying all of the possible critical failure scenarios. The conditioning loop creates multiple critical failure cases which will be analyzed in no particular order. As one would expect, all potentially dangerous situations relate to the physical condition of the working fluid threatening the integrity and longevity of a major component.

Fluid cooled centrifugal pumps most often fail due to insufficient fluid flowing through the pump causing the motor to overheat. This situation can arise due to air bubbles forming at the pump inlet, known as cavitation. Additionally, unexpected large pressure drops in the system such as a valve being closed, fluid freezing in the pipes, or even a clogged filter can also attribute to low/no flow conditions. This is one failure scenario.

The heat pumps are susceptible to the aforementioned flow interruptions as well, but also have maximum flowrates they are rated for. Flowrates in excess of the heat pump maximum rating are not a concern as such a scenario would not cause any adverse effects to any components, rather it would simply increase the capacity slightly. Due to its role as the source of heating or cooling, the heat pump can also create unsafe high/low fluid temperatures.

The conditioning coils and TET's are passive devices and thus their only foreseeable critical failure would be the fluid pressure exceeding their maximum pressure rating. This is a concern for every component in the system and thus needed to be addressed. A pressure relief valves is used to ensure the fluid pressure does not exceed the critical pressure of any component. There is one located in each system downstream of the pump. This location was chosen as the most likely point for the highest fluid pressure to be found due to losses throughout the system. The safety relief valve is set to 50 psi, well below any component's maximum rated pressure.

An additional situation to avoid, which does not rise to the standard of critical failure, is the operation of the heat pump without the pump running. Also, providing heating to the airside facility without any means of dissipating the heat and generating excess moisture should be avoided. These situations differ in their severity and in the way they are measured and thus are treated differently.

To summarize, there are four critical failure cases which must be avoided and a few non-critical situations which should be avoided. The four critical failures arise from no fluid flow (fluid pressure too low), fluid temperature too high, fluid temperature too low, and fluid pressure too high. These critical situations are handled by a physical safety circuit, while the situations of nuisance are negated by user control input limitations within the controlling software code.

4.2 Physical safety circuit

The purpose of the physical safety circuit is to ensure the critical failure situations are never reached. This circuit is composed of only mechanical switches which act automatically. This is a huge benefit as it enables the circuit to become fully independent from user inputs or communications with an outside information source. For example, imagine a train hurtling down a track and a landslide is reported ahead. The engineer begins braking procedures but the controlling software crashes. Now, the passengers would hope there is a mechanical backup system which could still be used to stop the train. This mechanical redundancy is the behavior of the physical safety circuit, it will always function to prevent unsafe operation. The facility operator should not let the unsafe conditions come in to play; however, the physical safety circuit will always activate to prevent critical failures.

Below are the circuit diagrams for the large and small heat pump loops. The only difference between the two is the addition of a second compressor contactor in the TMW120 loops. This second contactor must be energized to allow the second stage of the heat pump to activate, as one remembers it is simply two TMW060 heat pumps in one. Due to the similarity, the TMW120 safety loop will be examined in detail. The switches labeled LV 1, LV 2, and LV 3 are discussed in greater detail below in the controls section; however, for the purpose of the safety circuit they are simply on/off boolean commands sent from the LabVIEW user interface to the heat pumps via a relay module.

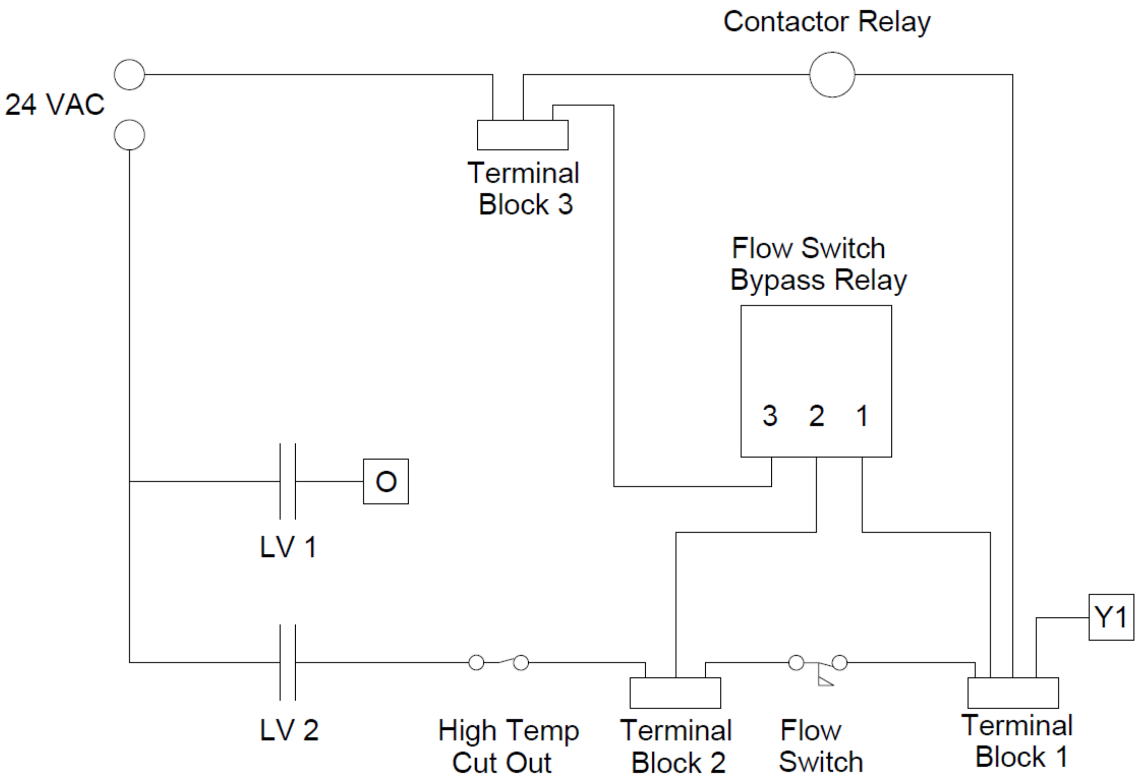

Figure 4.1: The safety circuit for the TMW060 heat pump discussed above.

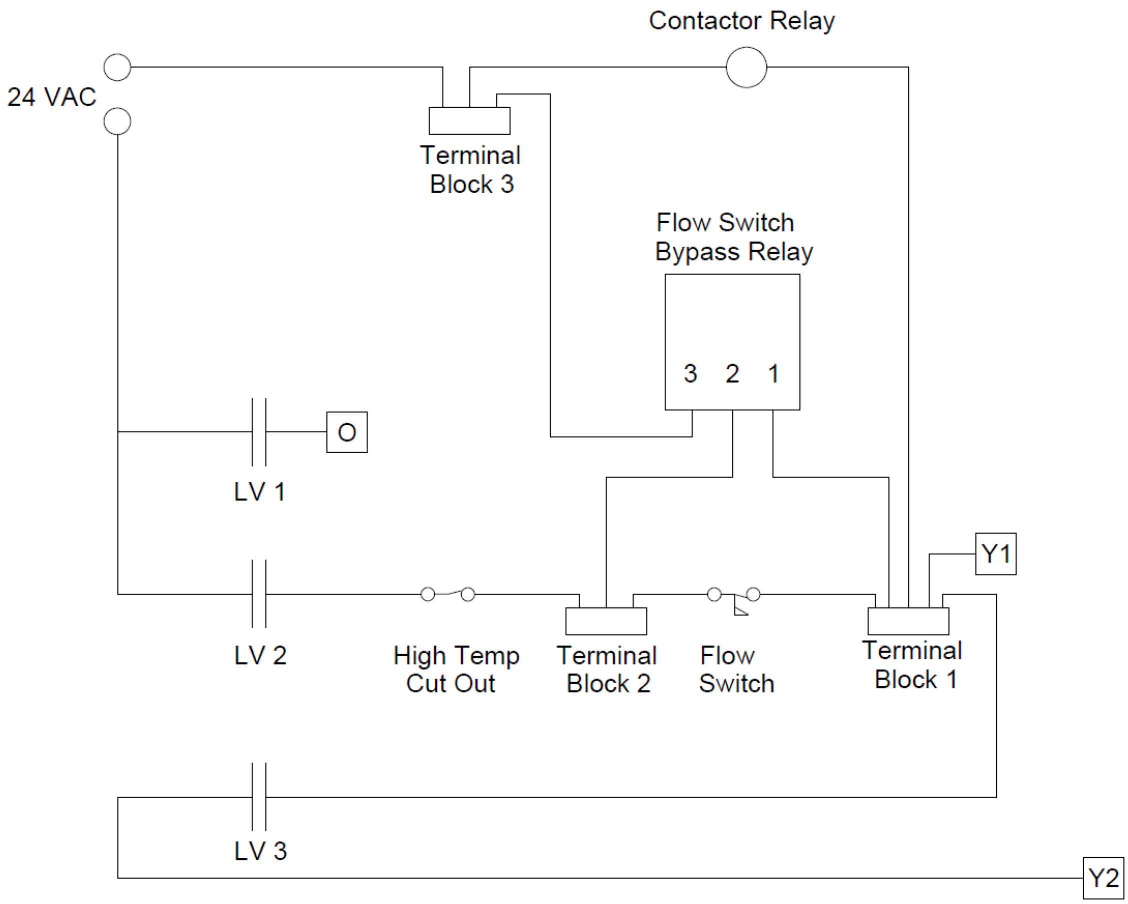

Figure 4.2: The safety circuit for the TMW120 heat pump discussed above.

Stepping through the circuit, first encountered is LV 1 which controls the reversing valves on the heat pumps. A reversing valve allows the user to select the flow direction of the refrigerant inside the heat pump. This direction dictates where the heat pump is in heating or cooling mode. LV 2 turns the compressor on to begin generating heating or cooling. Next, is the high temperature cutoff switch. This is a mechanical switch with a temperature sensing probe attached to a pipe on each conditioning loop. The switch is normally open, meaning it will remain closed and allow current to flow through the circuit until its coil is energized due to a temperature above the set point of the device being registered. More succinctly, it allows the compressor to run unless the fluid in the pipe gets too hot in which case the compressor is shut off to allow the fluid to return to a safe temperature. The following item, a terminal block, does not have a mechanical action. Instead, it is simply used for organizing wires. Next is another major safety component, the flow switch preventing the critical "no or low flow" failure. This switch is also normally open, meaning as long as sufficient flow is present the circuit is active, but deactivates if flow is interrupted. Then there is another terminal block used to organize the second half of the circuit. Y1 is the compressor contactor which allows the compressor to run when Y1 receives current. As mentioned previously, on the large heat pump circuits there is another compressor contactor, Y2, for second stage operation. The time delay relay shown connected to the terminal blocks, allows the pump to run for a specified amount of time before checking the flow switch. This bypass is necessary because there is a time lag between when a pump is turned on and when it generates sufficient pressure rise to move fluid through the entire loop. Choosing the proper amount of time to bypass the flow switch is more art than science. In this project, the bypass time was determined by measuring the time required for a user to select the pump in LabVIEW, choose the minimum flowrate required to activate the flow switch, and then the time required for the pump to achieve that flowrate from a "no flow" starting point. This procedure was completed five times and the average time is used as the bypass time. Once the bypass timer is no longer active, if sufficient flowrate has not been achieved then the flow switch does not activate and everything is turned off and debugging can begin. Low temperature sensing and circuit deactivation is controlled via an in-built safety located within the heat pump circuity. If this safety device is triggered, or any other in-built heat pump safety, the 24 VAC supply of the safety circuit will be interrupted. This is because the 24 VAC is fed by the heat pump and will deactivate all components if the heat pump enters unsafe operating conditions. To aid in understanding how the safety circuits function, a flowchart is shown below in Figure 4.3.

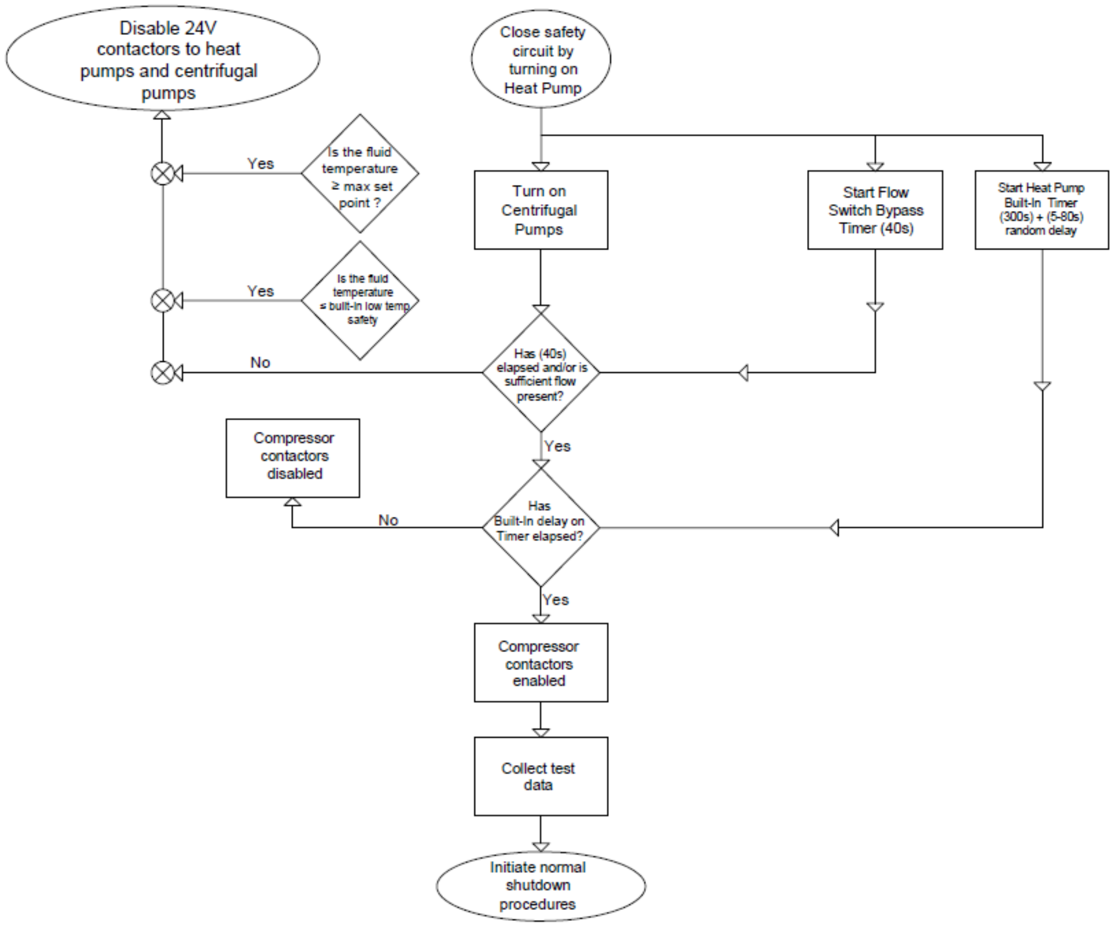

Figure 4.3: A flowchart detailing the operation of the heat pump safety circuits.

4.2.1 Software safety circuit

There are a few safety features built into the LabVIEW code with more planned as the project matures. One such feature is the code must detect at least one pump and one fan operating before the electric heaters are able to turn on. This ensures there is a means of dissipating the heat generated to prevent the heater from melting itself or other components. Additionally, the pump can not be turned on separate from the heat pump through LabVIEW. This guarantees the physical safety loop is active and protecting the conditioning loop. Once the heating capacity of the loops is known, a limitation will be placed on the pump flowrate preventing the conditioning loop from generating more heat than the airside equipment can safely dissipate. A similar control is already present on the heaters in the airside controls that checks the flowrate from the blowers and calculates the maximum heat input allowed. Additional information will be added as new user control limitations are implemented.

4.3 Controls

For a test point to be considered valid, precise control of the test coil inlet air properties is required. Precise as defined by ASHRAE Standard 33 (ASHRAE, 2016) states, the average entering dry bulb and wet bulb temperatures must remain within 0.5°F and 0.3°F respectively. In addition, no single entering value of the same metrics can vary more than 1.0 °F and 0.5 °F respectively. These steep requirements necessitate an automatic controls system capable of sensing minute system changes and making the necessary adjustments to maintain steady-state conditions. The conditioning loop must act in concert with the steam re-humidification and electric reheat. Thus an integrated controls scheme was implemented in LabVIEW with all of the instrumentation and controls available on the same screen. One of the greatest and currently ongoing design challenges of the entire facility is coordinating the response of all of the integrated systems to different stimuli and test conditions. With so many different individual systems with their own control schemes care must be taken to ensure the controls work in harmony rather than fighting against one another.

4.3.1 Control options

Within the conditioning loop there are two components whose operation can be adjusted. These adjustments are necessary to match the capacity of the test coil. It is highly unlikely the test coil's capacity will be a multiple of the heat pumps, and also small transient changes need to be handled as well. The heat pumps are controlled via digital inputs whereas the pumps have analog controls. Digital controls can be thought of as switching between on or off, or the values 0 and 100. In contrast, analog controls are a range between 0 and 100, not simply one or the other. For this reason, the two components are controlled via different devices; however, both are still adjusted through user inputs to LabVIEW.

The heat pumps are controlled via a 0-24 VDC signal sent from a National Instruments (NI) 9476 digital source module to a ZIPLink ZL-RRL16F-24-2 relay module. A 0 V signal corresponds to not active while a 24 V signal energizes the coil switching the relay. The ZIPLink relay module removes the need for mechanical relays in each heat pump circuit. To better illustrate the importance of this, below in Figure 4.4 and Figure 4.5 one can see the lack of space for two or three additional relays. An added benefit of the ZIPLink module is the ease of debugging most operational issues with the safety circuit as the status of each relay is displayed on the ZIPLink module located next to the user control station. Alternatively, if the relays were in the heat pumps, the user would need to climb up to the mezzanine, find the relay of interest via the wiring diagram, and finally use the multimeter to acquire the same information as the LED provides on the module.

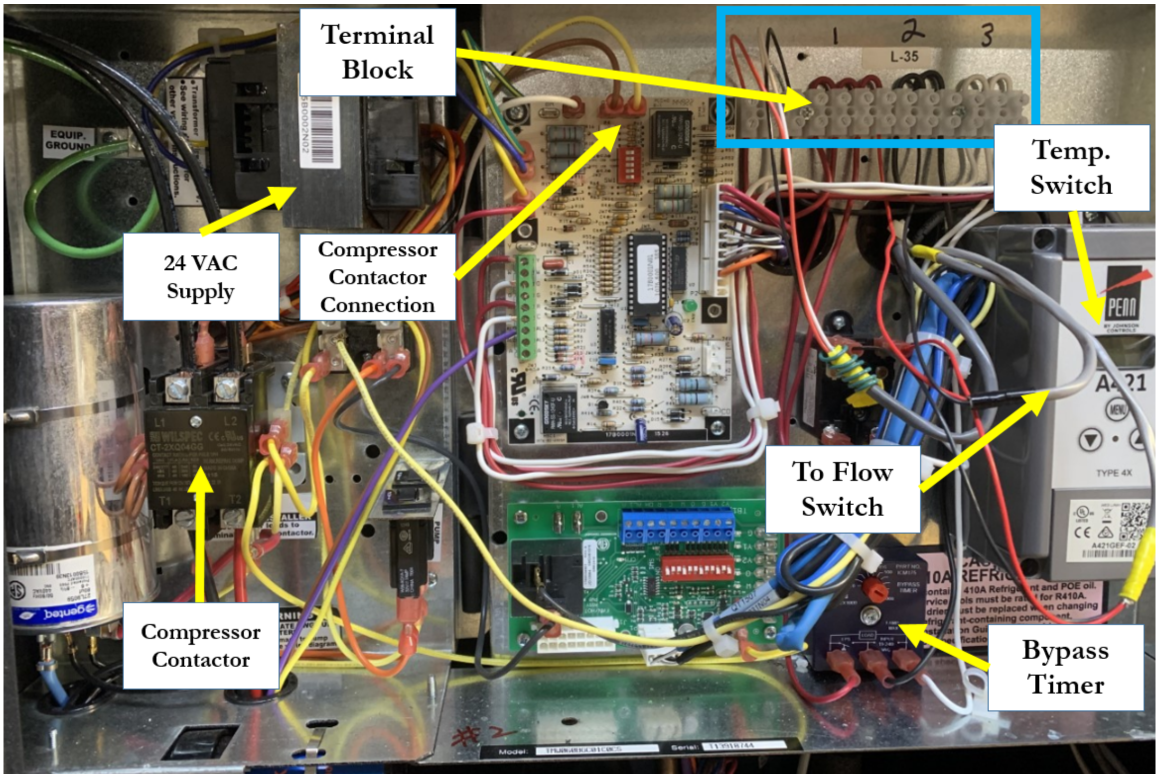

Figure 4.4: This is the wiring for the TMW060 heat pumps, a combination of factory and aftermarket wiring.

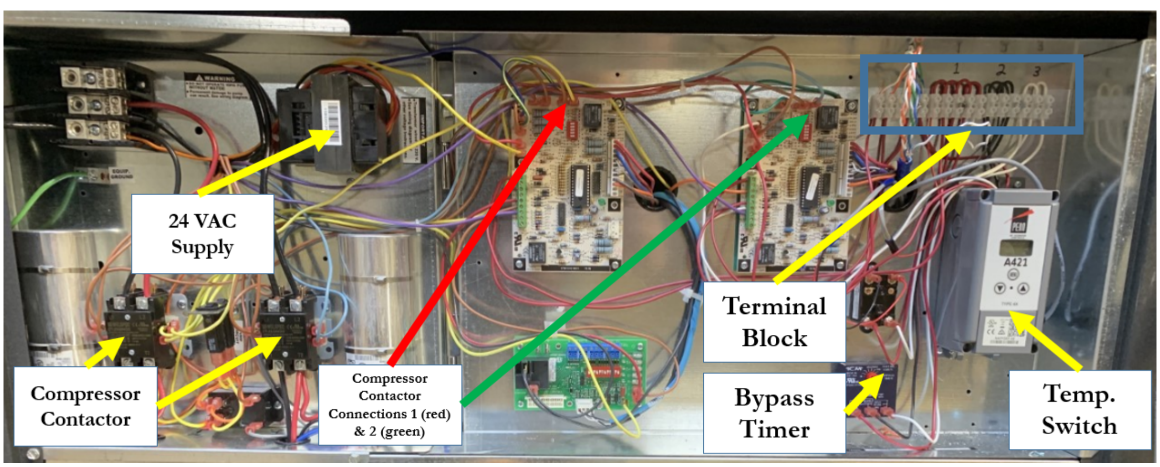

Figure 4.5: This is the wiring for the TMW120 heat pumps, a combination of factory and aftermarket wiring.

As mentioned previously in the discussion of the physical safety circuit, the control of the heat pumps is limited to turning the compressor on and switching the reversing valve. This does not allow for much in the way of capacity control. Often, heat pumps will have variable speed compressors to help with this; however, the compressors for this project are all single speed. Capacity will be grossly controlled in this heat pump by staggering how many heat pump stages are in operation, and finely controlled by varying pump speed for each loop. The pumps are controlled via a variable frequency drive, (VFD), which allow the pump to operate as a variable speed device. This is accomplished via an analog signal sent by an NI 9266 current output module which sends a 0-20 mA signal to the VFD, the VFD then modulates the line voltage received by the pump with 0 Hz representing 0 mA and 20mA as 60 Hz. This arrangement allows each pump to operate between 0 and 100% of the maximum RPM's the pump is rated for. So capacity is controlled by determining the number of stages of heating or cooling needed and then adjusting the flowrates of one or more loops to achieve the capacity required for the test.

4.3.2 Miscellaneous Facility Upgrades

There were a number of assorted upgrades which were necessary to ensure the facility would operate as intended. First, additional wiring was added to the blowers to enable control and feedback to be sent to the LabVIEW software. Next, the airside physical safety circuit was upgraded to ensure all failure scenarios were handled appropriately and the heaters and steam humidification system only operated when safe conditions were present. Finally, changes were made to the LabVIEW interface to allow for automatic control of the major components of the entire facility.

The airside physical safety circuit is shown below in Figure 4.6. To activate the circuit, the emergency stop button is deactivated. This method of closing the circuit is used as it restarts all of the timers. The fans are allowed to run for 40 seconds before the air flow switch checks to ensure the required 4000 cfm flowrate has been achieved. Again, the delay is necessary as the fans do not immediately create the necessary pressure rise. At any time during operation if the adjustable temperature switch is tripped, or the air temperature is above 175°F, a door leading to dangerous equipment, or the necessary flowrate is not present the circuit will deactivate and all equipment is turned off. The heaters and steam humidification equipment is allowed to turn on after 270 seconds. The circuit can be deactivated at any time by the user depressing the emergency stop button.

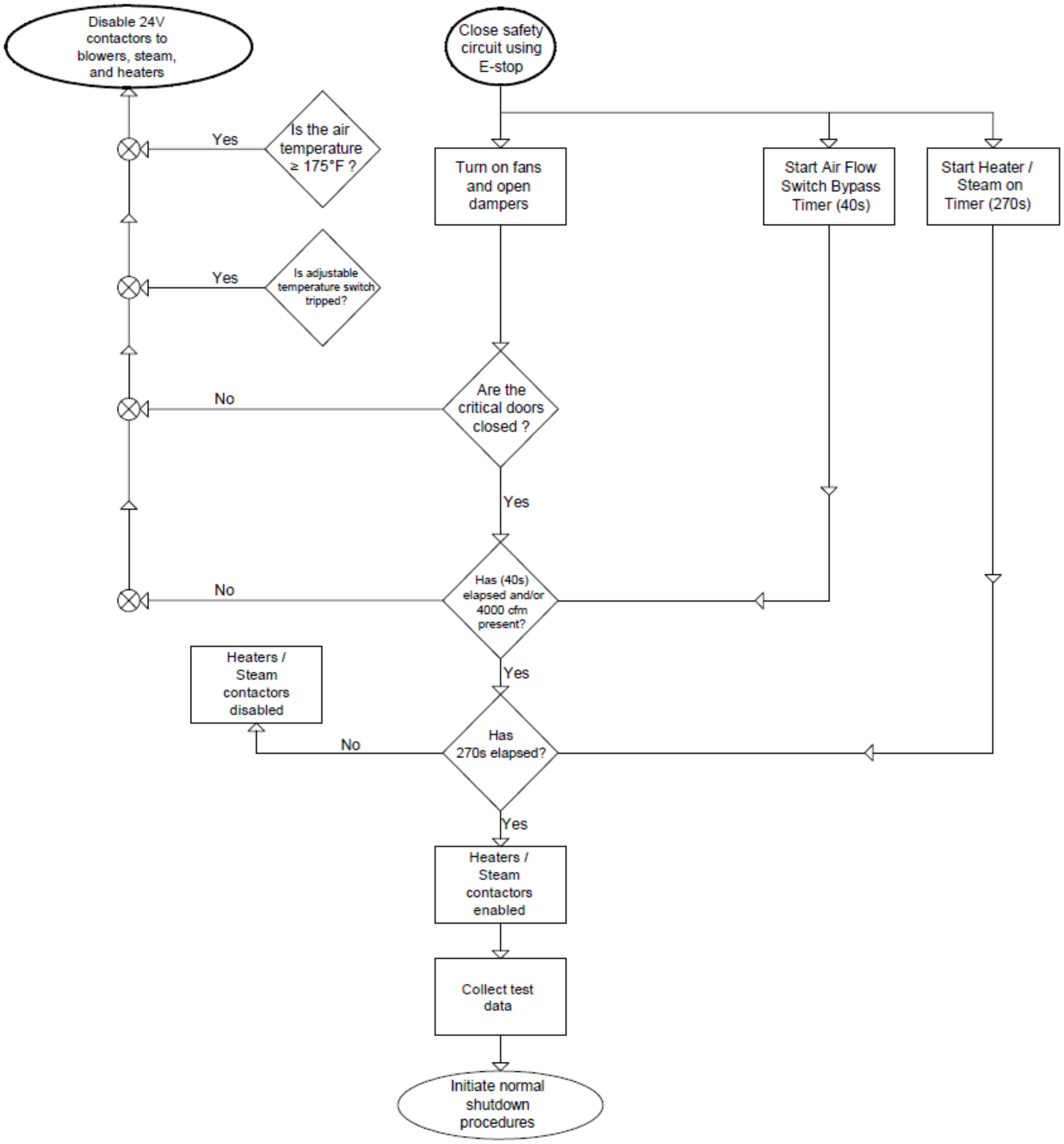

Figure 4.6: This is a flowchart representation of the airside physical safety circuit.

The LabVIEW software was adjusted to allow for automatic control of the fans and heaters. The entire facility is operated from the LabVIEW user interface seen below. Starting from the left, the user selects which nozzles are currently in operation, i.e. not plugged, based on the testing flowrate. Active boolean buttons are illuminated. Next, one finds the centrifugal pump and heat pump controls. As mentioned previously, the heat pumps are controlled via the "Stage Switch" or "Stage

1" and "Stage 2" buttons for the TMW060 and TMW120 respectively, and reversing valve toggle. The centrifugal pumps are controlled via the 0-10 dial which is scaled to output a corresponding 0-20 mA signal to the pump VFD. Next are the dampers which are directly downstream of the conditioning coils. The 0-100% dials reflect 0 being fully closed to 100 being fully open. The values on the dial are scaled to output a corresponding 2-10 VDC signal. There are three fans, two 10 hp and one 3 hp. The fans are able to be toggeled on/off and to be in manual or automatic control. The 0-10 dials correspond to a 4-20 mA signal on the fan drives. Currently for automatic mode, the two 10 hp fans run at a constant speed while only the smaller top fan is adjusted via the in-built PID LabVIEW controls with the set flow rate integer being the value of interest. There are two 20 kW heaters which are coupled together and act in concert according to the in-built PID LabVIEW controls with the test coil inlet average temperature the value of interest. The PID parameters for the fans and heaters shown are the most efficient values determined by the "guess and check" method. The user interface is shown below in Figure 4.7. Future work includes adjusting the operation of all three fans simultaneously. Additionally, investigating moving from PI to PID control, as well as, calculation of more efficient gains and gain scheduling will be explored.

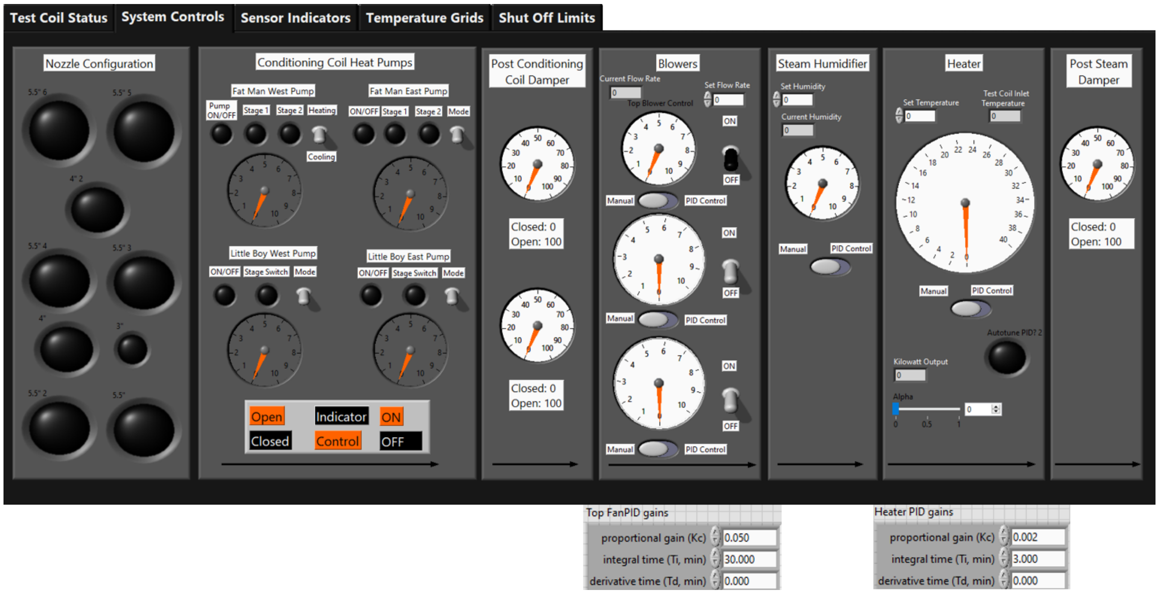

Figure 4.7: LabVIEW user interface to control the facility; system controls tab shown.

CHAPTER V

Mechanical System

As is often the case with large designs using new equipment, the implementation of said design is often the most challenging portion of the project. This project is not exempt. The criteria for mounting and connecting equipment in this project had to follow the following mantra. The equipment must be mounted safely, in an easily maintainable location, and posses the minimum footprint possible. Certain design constraints contributed to some of the design decisions, but overall almost of all of the mechanical design decisions were made with project longevity in mind, not constraints. Or in other words, the author's attitude towards the matter was "What is possible given the circumstances", not "what cannot be done in this situation?" The following sections highlight many steps necessary to properly integrate and operate the conditioning loop.

5.1 Heat pump layout and support structure

The layout of the heat pumps and their support structure is an excellent starting point for exploring this project's emphasis on safety, ease of maintenance, and minimal footprint. Seen below in Figure 5.1, is the finished layout and support structure. Please note the wooden supports are a temporary measure. The heat pumps are placed atop unistrut pallets to distribute the weight in a manner which does not exceed the 125 pounds per square foot (psf) limit of the mezzanine grating. All of the heat pumps face forward so that the access panels are easily reached. Also note, the lack of piping obscuring access to the front panels. Plywood acts as a moisture barrier between the heat pumps and airside facility while also allowing the heat pumps to be moved via pallet jacks across the mezzanine grating. The heat pumps are placed at the end of the mezzanine to allow maximum space for future project expansion. Finally, the pumps and TET's are mounted on the support structure to be maximally accessible while not adding to the project's overall footprint. Pipe hangers tie into the unistrut to help support the piping.

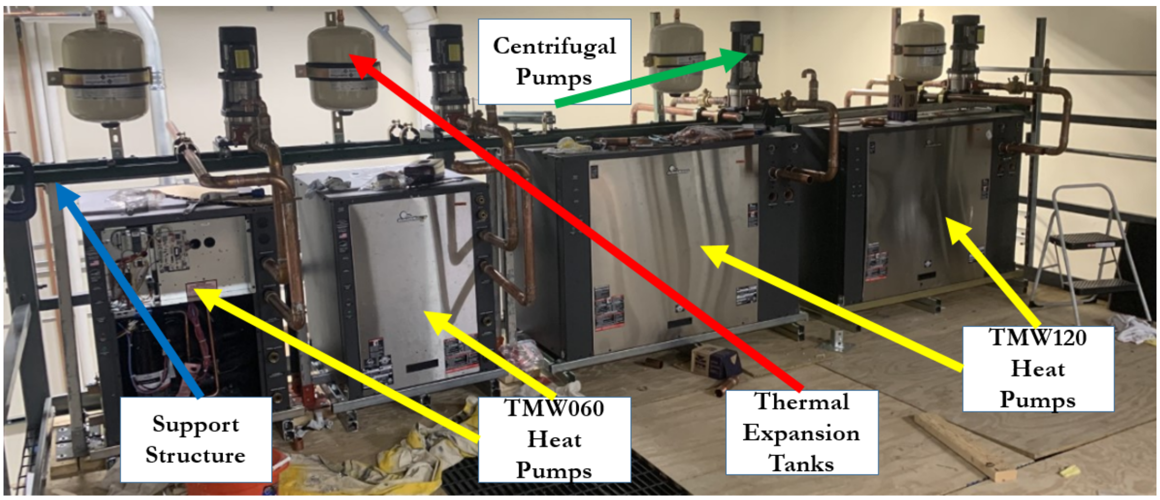

Figure 5.1: The placement of the heat pump and support structure shown for help visualizing the setup as it evolved.

5.1.1 Verification of mezzanine safety

The total combined weight of the heat pumps, pumps, and TET's is 2172 lbs, 209.2 lbs, and 110.1 lbs respectively. This would be a concern for the mezzanine deck rating except for the fact the plywood and support structure help to distribute the weight over a much larger area than the footprint of each individual component. It is difficult to precisely estimate the loading at specific points; however, care was taken to include a large factor of safety by distributing the loading over as many points of contact as possible. Assuming the 70 sq ft array of plywood supports the load equally, the loading on the mezzanine grating is only about 35.6 psf, well below the 125 psf rating.

5.2 Connections to heat pumps and coils

There are numerous different types and sizes of connections required by the heat pumps and conditioning coils. Care was taken when selecting fittings to avoid the use of reducers which mitigates additional flow disruption and reduces the amount of soldering required. An example of the selection process is the 4 FPI coil which has a 2" male pipe thread (MPT) connection, so a 2" FPT x 1-1/2" C (female pipe thread to cup) fitting was employed to connect the coil directly to the pipe leading to it.

5.2.1 Threaded fittings

One major point of learning in this project is the treatment of threaded fittings. At one point or another at least one of each type of threaded fitting leaked. Learning the proper amount of thread sealant and torque to apply to the fitting was crucial in ensuring the conditioning loop is and will continue to be leak tight.

5.3 Piping support structures

There are three main piping support structures for the conditioning loop. One of them is part of the heat pump support structure which will be ignored as it has already been discussed. That leaves the supports on the top and side of the airside facility. The aforementioned remaining support structures are shown below in Figure 5.2 and Figure 5.3. All of the pipe hangers are self-insulating to prevent condensation formation which can lead to future degradation. The ASHRAE Handbook of Fundamentals recommends a pipe hanger spacing of 8 ft for 1-1/2" copper tube with water as the working fluid (ASHRAE, 2017). This requirement was fulfilled in all spans of the piping. Both support structures easily carry the weight and prevent significant movement during full flow conditions without over-constraining the piping. Note again how both structures minimize the overall footprint of the project.

Figure 5.2: This is a view of the piping supports located on top of the airside facility.

Figure 5.3: This is a view of the piping supports located on the side of the airside facility.

5.4 Airside facility modifications

Physical modifications to the airside facility were necessary to fully integrate the conditioning loop. The majority of modifications revolved around the conditioning coils. To prevent water damage from condensation or water due to a defrost cycle, drain pans are located beneath both the top and bottom sets of conditioning coils. These drain pans need to empty periodically, thus holes were drilled into the airside facility to allow PVC pipe to connect the drain pans to drains located beneath an access panel further downstream in the facility. Additionally, the access panels covering the conditioning coils required alterations to allow the pipes and air vents to connect to the coils. Sections were cut to allow the panels to still be removable to allow for cleaning and maintenance of the conditioning coils.

Figure 5.4: Holes were cut into the access panels to allow them to still be easily removable.

5.5 Connection to lab chilled water

As mentioned previously in Chapter II, the heat pumps must reject heat to an external water source. In this project, the external source is the lab chilled water system. A contractor was hired to freeze the lines and insert all of the copper piping and brass ball vales seen below in Figure 5.5. The heat pumps are connected to the lab chilled water using PVC pipe because it is inexpensive and quickly assembled. In the future, copper pipe will replace the PVC to alleviate concerns over the brass and PVC having different thermal expansion rates which can lead to small slow leaks at threaded fitting locations.

Figure 5.5: The heat pumps are connected to the lab chilled water via PVC as seen here, please note the wood supports are temporary.

CHAPTER VI

Integration to airside facility and shakedown testing

The conditioning loop is fully integrated and in the beginning stages of shakedown testing and validation. All of the pumps and heat pumps are operating normally and fully controllable through the LabVIEW interface. Potentially unsafe operating conditions are handled appropriately by the safety circuit. Having accomplished the above, one would assume full facility testing could begin; however, the test coil is not yet connected to a heat source and thus unavailable. There are a few tests which can still yield valuable results. One test is estimating the thermal mass of the tunnel, which is an excellent opportunity to verify the operation of the automatic controls. Also, the cooling capacity of each heat pump loop at different operating conditions can be determined up to the limit of the heaters ability to counter the heat pumps. These tests were completed with water as the working fluid to prevent possible damage, in the case a leak occurred.

6.1 Thermal mass test

Estimating the thermal mass of the tunnel allows users to anticipate the amount of losses to current leakage paths in the airside facility. Also, the amount of time to move between different testing conditions is calculable based on these preliminary steady-state tests. To calculate the thermal mass, a transient response test was attempted. The test involved running the blowers at a constant power input and allowing the air temperature in the airside facility to reach a steady temperature a predetermined amount above the ambient air temperature, as measured by a thermocouple located outside the airside facility. Such a test is designed to solve,

$$\dot{W}_{Fans} + \dot{Q}_{Elec.Heaters} = (mcp\frac{dT}{dt}) + (UA\Delta T). \tag{6.1}$$

The heat transfer, Q, is the power input of the blowers (measured from the VFD's) which is held constant for the duration of the test The temperature change inside the facility is plotted against time. When t=0, ΔT=0 so one can solve for δt/dt assuming *mcp* as one term with known heat input. The inverse is true when a steady temperature is achieved inside the facility relative to ambient. Via a curve fit, the UA or thermal mass of the system can be solved. This allows one to calculate the time it should take to move between different set points, and allow the second shakedown test.

One thermal mass test was attempted with inconclusive results. The ambient temperature was 71°F and the air inside the facility was raised to 75.5°F over the span of 40 minutes. The expected trends are for the mcp portion (transient system response) of the equation to draw near zero as a steady temperature difference is approached and for the total heat transfer to be equal to the heat input of the test. As one can see in the results below in Figure 6.1, there was simply a bit too much noise in the data and the initial transient response is not easily fit. An improvement to this test might require creating a larger change in temperature before the steady-state condition is reached. This should improve the results as it minimizes the error introduced by the thermocouple's uncertainty.

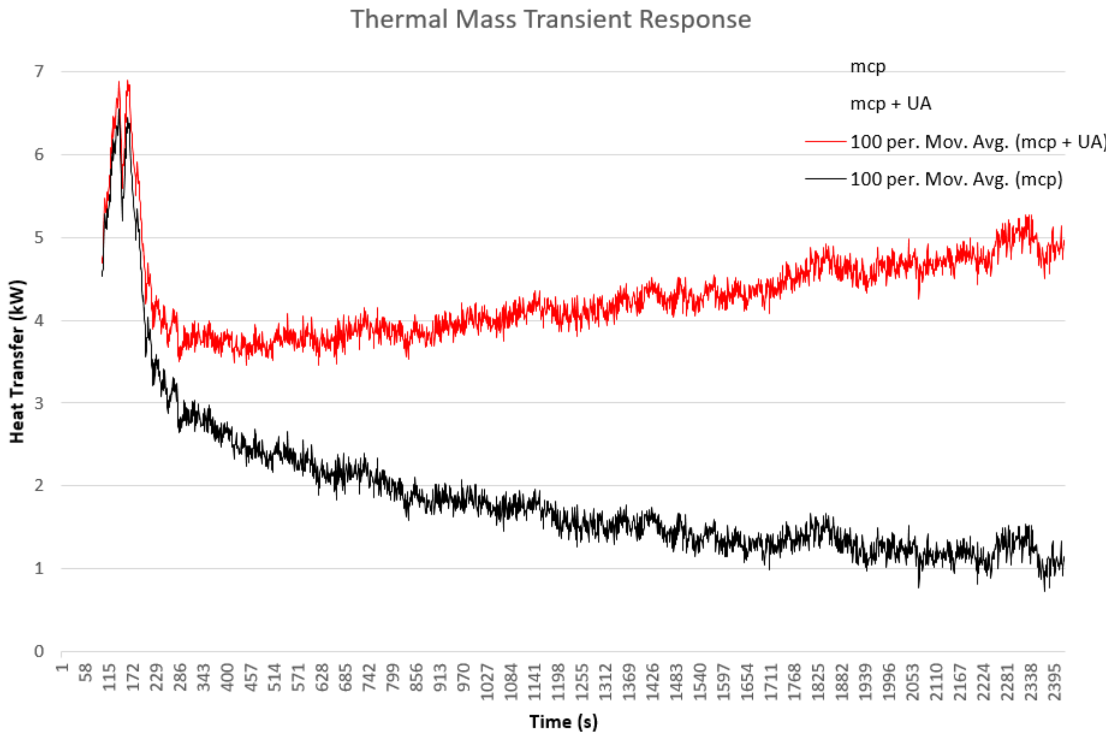

Figure 6.1: This is the initial attempt to estimate the thermal mass of the airside facility.

Taking the lessons learned from the initial test, a new test matrix has been created. Due to a possible change in ambient temperature in which the facility is located, ΔT is used to refer to the difference between air inside and outside the facility. A higher or lower ambient temperature may slightly affect the results even if the ΔT remains the same, but sufficient test points should negate these effects. An additional source of possible error is the flowrate of air which could increase or decrease heat losses due to changing the infiltration rate. For this reason, the same ΔT's will be tested at two different flowrates to see if there is any impact on the results. Rather than only using the blowers to create the temperature difference, the ΔT will be controlled via PID control of the electric heaters when needed, whose output will be read from LabVIEW. The calculations will remain the same except Q is now the total input of the blowers and heaters. The same ΔT's are approached from a lower and higher temperature to investigate capacitive effects.

Table 6.1: Revised Thermal Mass Test

Case	Test Description	ΔT	Flowrate (cfm)
1	increase to 5	5	6000
2	increase to 10	10	6000
3	increase to 15	15	6000
4	decrease to 10	10	6000
5	decrease to 5	5	6000
6	increase to 5	5	8000
7	increase to 10	10	8000
8	increase to 15	15	8000
9	decrease to 10	10	8000
10	decrease to 5	5	8000

The new thermal mass study more accurately determined the response of the system at different temperatures. The test coil inlet temperature is the average at the test coil inlet which represents the different set points used. The ambient temperature is the average ambient temperature directly outside the airside facility and total capacity in terms of the heaters and fans are also shown. This data is required to calculate the thermal mass (mcp) and effective resistance (UA). Thermal mass is the slope of the test coil inlet temperature graph when the test coil inlet and ambient temperature intercept, i.e there is no ΔT. The effective resistance (UA) on the other hand is the division of the total capacity by the difference in temperature between the test coil inlet average and the ambient temperature average. At roughly 1-1/2 minutes into the test, the thermal mass is estimated to equal approximately 500 kJ/K. The effective resistance is approximated at 2-3/4 hours and 6-1/4 hours as 0.54 kW/K and 5.13 kW/K. Ideally more test points will be taken to further explore the change in effective resistance and total capacity with test coil inlet temperature. After analyzing the results, there are some concerns that the system was not truly steady when the previous results were computed. While the test coil inlet temperature was steady, the heat input was not fully steady and thus lends to the inconclusive results. The results are shown below in Figure 6.2.

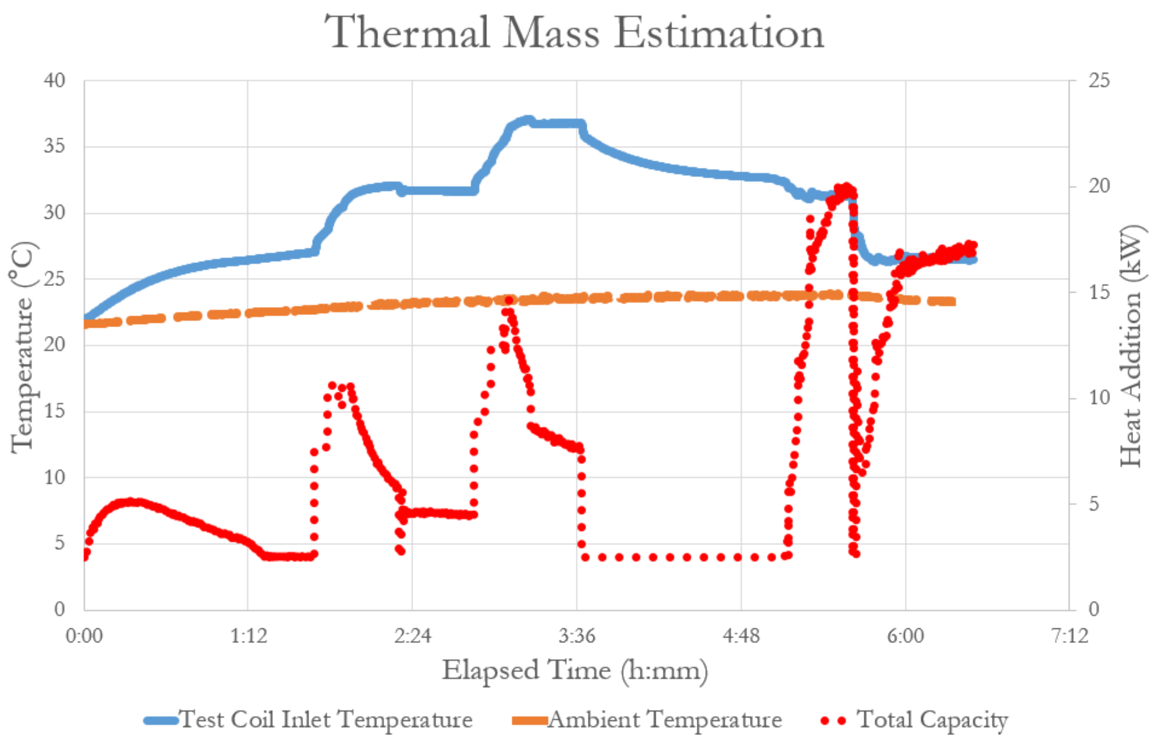

Figure 6.2: This is the second attempt to estimate the thermal mass of the airside facility.

6.1.1 Adherence to ASHRAE Standards 33 & 41.2

The airside facility was designed and constructed to meet the requirements set forth by ASHRAE Standards 33 & 41.2. During this initial testing phase, care was taken to measure the ability to adhere to the standards. Preliminary steadiness results were available from the aforementioned test as well. The average test coil inlet temperature and flow rate during the time Case 3 were considered steady and are plotted below in Figure 6.3 and Figure 6.4. This shows the ability of the facility to gain and maintain a steady condition for approximately 20 minutes. The small instabilities might be a result of noise or due to the lack of calibration on the instrumentation.

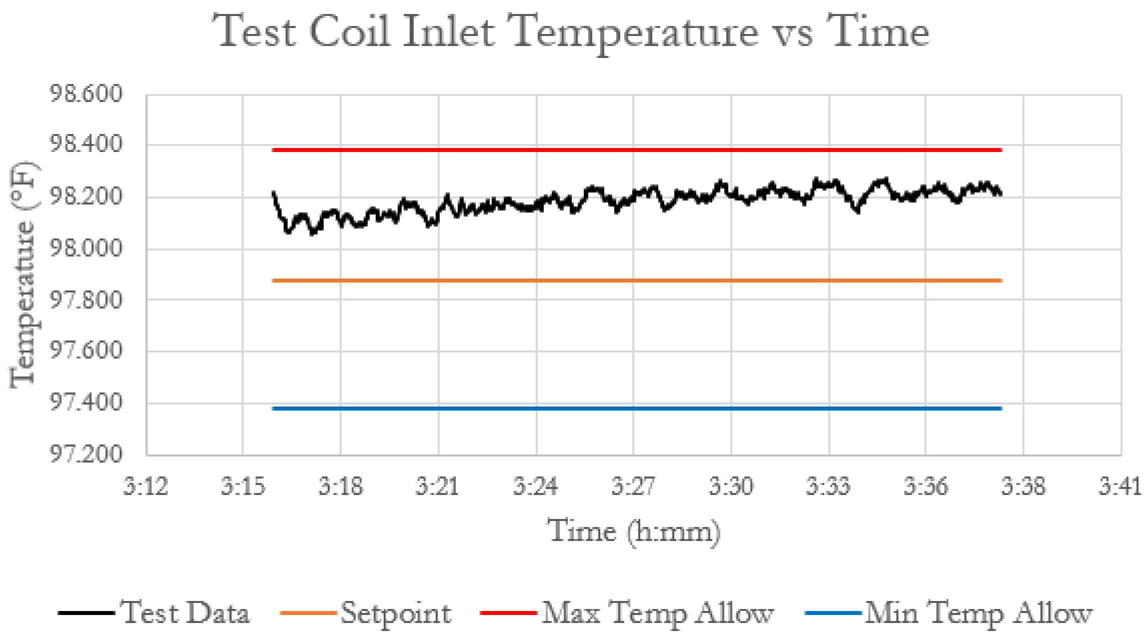

Figure 6.3: The data shown is the average test coil inlet temperature at the 15°F above starting temperature set point

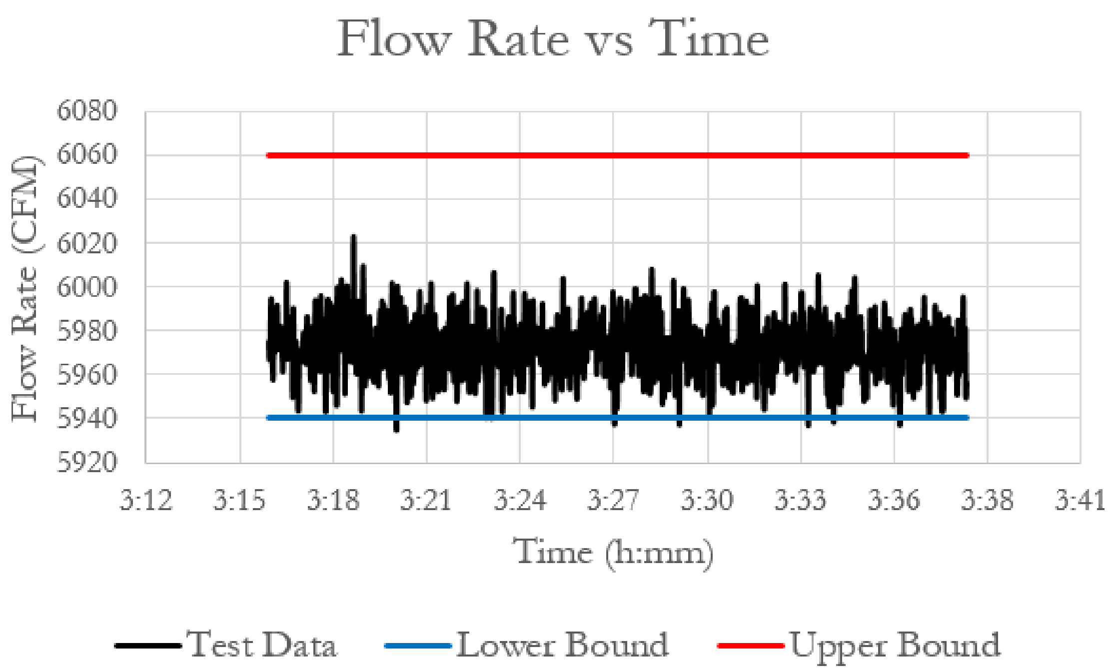

Figure 6.4: The data shown is the air flowrate at the 15°F above starting temperature set point.

6.2 Capacity test

Once the UA value of the facility has been calculated, the capacity of each heat pump loop can be estimated. The blowers will be running at a steady power input while one heat pump loop is active in cooling mode. Electric reheat will be added until the air inside the facility is held steady at ambient temperature. Knowing the fan input power and the electric reheat power (measuring the amps and multiplying by rated voltage) allows the cooling capacity of the heat pump loop to be estimated.

To begin the test, the blowers will be switched on to activate the flow switch and allow the heaters to be turned on. At this point, the heaters will be activated at a small output value and a TMW060 loop will be turned on in cooling mode. The amount of electric reheat will be adjusted until the temperature inside the facility is maintained at roughly ambient temperature, eliminating the need to calculate losses due to infiltration. After each heat pump loop has been rated at ambient conditions, additional capacities at different air temperatures can be tested, but in these cases including heat losses in the form of UAΔT. A test matrix for estimating loop capacity is shown below.

Table 6.2: Heat Pump Loop Capacity Test

Case	Test Description	TC_{inlet}(°F)	Flowrate (cfm)
1	TMW060 (1)	80	6000
2	TMW060 (1)	70	6000
3	TMW060 (1)	60	6000
4	TMW120 (3)	80	6000
5	TMW120 (3)	70	6000
6	TMW120 (3)	60	6000

This test matrix is designed to provide enough data points to estimate the capacity of each heat pump loop at any testing condition within the heat pump and electric heater operational limits. The heat pumps are denoted by number's 1-4 according to their arrangement on the mezzanine. Additionally, data will be provided showing the amount of time required by the different sized loops to affect the temperature and achieve set points. This is important information for deciding how to stage the heat pumps to match a test coil. If there is enough capacity from electric reheat available, different combinations of heat pump loops can be tested.

The data defined as total system heat input is the electric heater output power and fan input power both in kW. The total loop capacity can be estimated when the test coil inlet temperature and total system heat input are both constant to balance the performance of the heat pumps. Estimated loop capacity for the TMW060 is shown below in Figure 6.5. Unfortunately, the capacity for the TMW120 loop was not able to be determined as there is not currently enough system input heat to keep the working fluid temperature above the minimum temperature allowed by the heat pump in-built safeties.

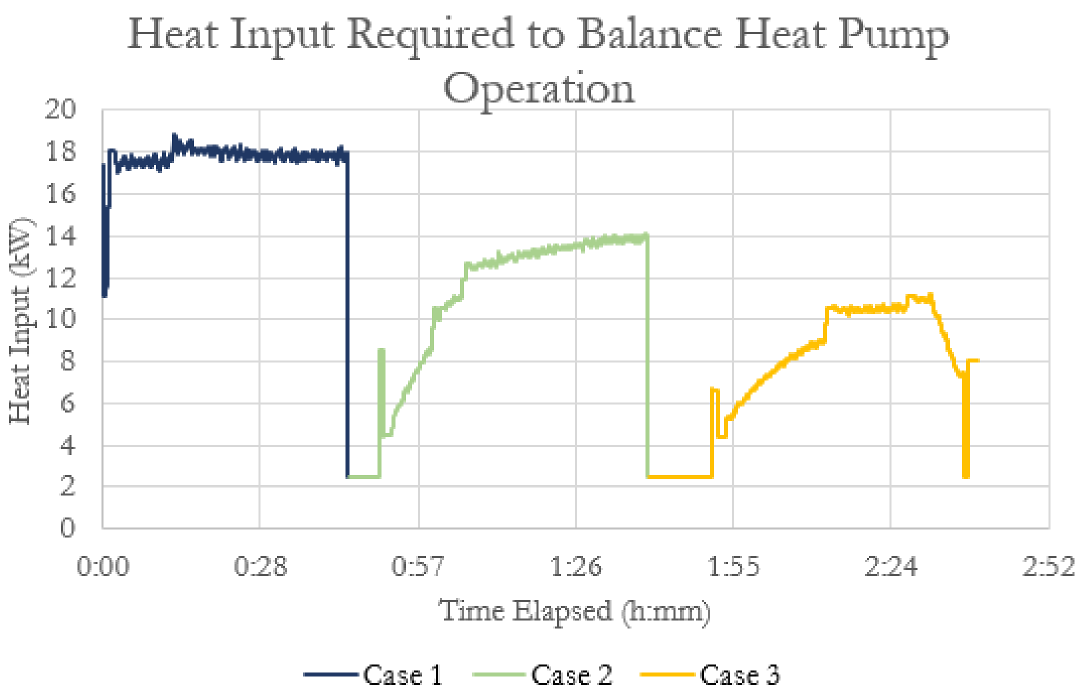

Figure 6.5: The total loop capacity for the TMW060 is shown at the three different cases.

6.2.1 Thermal stratification concerns

A recommendation for future improvement of the testing facility is the identified need for better air mixing in the airside facility. This can be seen below in Figure 6.6, as there is large thermal stratification at the test coil inlet thermocouple grid which is located downstream of the air mixers. This data is from the TMW060 capacity test and thus reflects the conditioning coil which in upstream and on the bottom being in cooling mode.

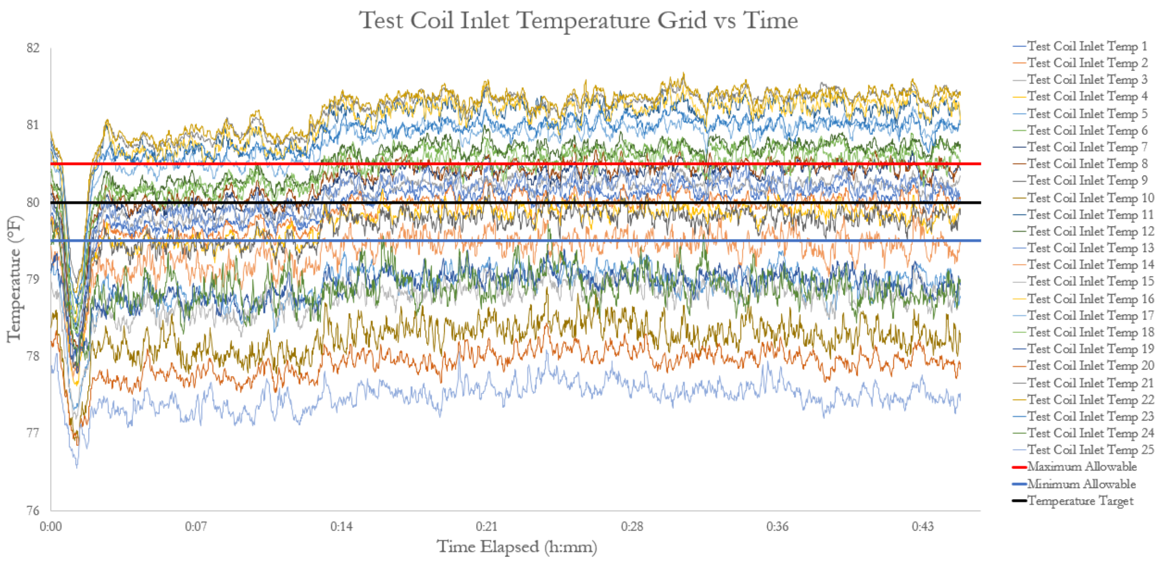

Figure 6.6: Thermal stratification seen at the test coil inlet.

CHAPTER VII

Conclusion and future work

A conditioning loop to support a new psychrometric coil testing facility for commercial size heat exchangers was designed and constructed at Oklahoma State University. The goal of the integrated facility is to advance research regarding the effects of implementing low-GWP refrigerants and heat exchanger improvements on a scale not previously seen. The conditioning loop design is flexible, affordable, easily maintained, and most importantly safe. Plumbing, pumps, and associated components were sized and connected to provide the conditioning required for continuous testing. An inherently safe and simple controls scheme was implemented, and necessary facility modifications were made to fully integrate the testing facility. Tests are ongoing to verify the cooling capacity of the conditioning loop in preparation for testing a heat exchanger in the facility.

The conditioning loop is fully constructed and some shakedown testing has been completed highlighting the functionality of the design and construction. Future work primarily exists on the airside portion of the facility in the matters of making the facility fully air tight and calibrating necessary instrumentation. Testing of a heat exchanger in the form of a commissioning experiment is the next logical step for the project. Small details remain such as finishing the insulation of piping which will ensure the capacity tests are an accurate representation of actual performance of the heat pumps. Testing of different staging techniques of the heat pumps would provide useful information to help the user approach steady-state conditions more quickly. Such investigations could also include the effect of running only both top conditioning coils and vice versa, the de-humidification effect of the large and small conditioning coils, the de-humidification effect of using one small coil upstream and one large downstream etc. There are many different phenomena which could arise and will only become known once testing has begun. Calculating all of the thermal gains of the combined system and their effective on total system cooling capacity would provide further useful knowledge. Also, calculating the optimal pump speed for maximum cooling capacity would give the next user of the facility some useful knowledge.

The conditioning loop still has a few upgrades which need to be made. One issue which needs to be addressed is the aforementioned sporadic leakage from the connection of the PVC components to the brass ball valves in the intermediate water loop. On the topic of the chilled water lines, the temporary supports of the TMW060 chilled water lines need replaced with a permanent support structure. As mentioned during the testing explanation, water is currently the working fluid in the conditioning loop; however, with shakedown testing complete the water needs to be replaced with the EG mixture to allow for low temperature testing. There are currently electrical panels which need installed to protect users from high voltage exposure. The heat pump panels need installed to protect safety circuit and components. Another necessary addition to allow low temperature testing is drain pans on the heat exchangers lacking them and drain pan heaters on all drain pans. This is necessary to allow the coils to be repeatedly defrosted. Hydronic pressure gauges installed at the inlet to the centrifugal pumps will provide the user with instantaneous head pressure readings to provide warning of and help prevent the pumps running at low inlet pressure. These gauges will also provide an immediate indication if the conditioning loop needs recharged with fluid. Standard operating procedures (SOP's) and maintenance programs need to be developed and implemented.

Similarly, the airside facility has a few remaining upgrades before it is considered fully operational. There are a few remaining essential and multiple non-essential measurement instruments which need to be installed. All measurement instruments need to be calibrated. One metric in particular worthy of further investigation is the airflow mass flow rate. This can be achieved through the project which Samantha Davis, an undergraduate student, began. This project is the installation and instrumentation of the tested coil. Added to this is a pair of psychrometers and sampling trees which in tandem with Samantha's project would yield a means of validating airflow rate through a capacity balance analysis. Uniformity of temperature, humidity, and velocity will also be investigated once these special projects are completed. Installing a "delay on make" timer relay on the airside physical safety circuit will help resolve the resonance issues which have been encountered and have the potential to damage and destroy contactors in the safety circuit. Finally, revisiting the drain from the steam generator will ensure safe evacuation of excess steam and water to the proper drain.

Overall, the project is beginning to take shape and the author looks forward with great excitement to what the future of this project holds.

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