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Design of a Psychrometric Coil Testing Facility for Commercial Size Heat Exchanger Coils

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ABSTRACT

The design of a closed-loop psychrometric coil testing facility to enable performance testing of commercial-sized heating, ventilation, air-conditioning, and refrigeration (HVAC&R) coils at Oklahoma State University is presented. This paper discusses the design process, preliminary modeling, and predicted operating envelope of the psychrometric coil testing facility. The closed-loop facility will contain the necessary equipment to condition the air supplied to the coil under test such as, electric heaters, and a steam humidification system. The design phase of the facility is complete with details of select subsystems presented here. The design is compliant with multiple ASHRAE standards, including Standard 41.2 (ASHRAE, 1987) which aided in designing the air flow measurement apparatus. A preliminary model of the facility suggests that, heat exchanger coils of up to 20 nominal tons (70.3 kW) of cooling capacity can be tested in a 7 ft (2.13 m) wide by 8 ft (2.44 m) tall test section that can supply a maximum air flow rate of 8,000 CFM (3.78 m^3/s). The facility will be able to operate over an air temperature range of 0°F (-17.8°C) to 140°F (60°C). Future work includes construction of the facility including physical assembly, selection of instrumentation, and development of a graphical controls interface for facility control.

1. INTRODUCTION

Much of the research in HVAC&R is currently focused on improving overall system performance and efficiency. Large commercial heat exchanger coils are a component which have a significant influence on overall system performance. However, the effect of design changes to the coils is not always intuitive and testing is required to verify and quantify the design improvements. To facilitate the quantification of heat exchanger coil performance a psychrometric coil testing facility is being constructed at Oklahoma State University to test how design changes affect heat exchanger coil performance.

A similar facility was constructed by Wenzel *et al.* (2016), designed for testing cold chain applications. Cold chain equipment is typically used to preserve food during transport. The facility is rated for testing 28.4 tons (100 kW) at -40°F (-40°C) to 99.5 tons (350 kW) at 32°F (0°C) and is designed to test commercial and industrial refrigeration equipment. This facility is a great example of utilizing an ethylene glycol and water mixture to attain cooling temperatures far below the freezing point of water.

Bell and Groll (2011) tested heat exchanger performance in a forced flow wind tunnel. The heat exchangers had a 15.7 in (40 cm) x 19.7 in (50 cm) frontal area. Water was the working fluid for the heat exchangers tested in the tunnel. There are many similarities between the experimental setup of Bell and Groll and the experimental design described below. One key difference is Bell and Groll's facility is an "open loop" design that targets conditions close to room temperature.

2. FACILITY DESIGN

Commercial size heat exchanger coils are the primary type of coil to be tested in this facility. The test section was designed to best accommodate this type of heat exchanger. A target operating envelope was originally developed by

Bach and Sarfraz (2016) which includes the desired ranges of temperature, humidity, and air flow rate for the facility. This operating envelope has since been modified from what Bach and Sarfraz (2016) presented due to heat pump and manufacturing limitations. The final operating envelope, shown in Table 1, served as the basis for the final facility design.

Table 1: Desired facility operating envelope, used as design inputs.

Parameter	Value
Temperature	0°F (-17.78°C) to 140°F (60°C)
Humidity	20 to 90%
Test Coil Capacity	23 tons (80.9 kW) at 67°F (19.4°C)
Maximum Air Flow Rate	8000 CFM (3.78 m^3/s)
Overall Dimensions	42 ft (12.80 m) long by 12 ft (3.66 m) wide by 9 ft (2.74 m) tall
Test Section Dimensions	7 ft (2.13 m) wide by 8 ft (2.44 m) tall
Conditioning Section Dimensions	4 ft (1.44 m) wide by 8 ft (2.44 m) tall

The final design of the facility incorporated the design parameters seen above. The facility consists of two major subsystems: an airside subsystem where coils are tested and a conditioning subsystem which manages the heat within the airside subsystem. A CAD model of the final coil testing facility design can be seen in **Figure 1**. The airside subsystem is a closed-loop design with a smaller conditioning section, on the right side of the isometric view, which adjusts the temperature and humidity of the air in circulation. The conditioning section is adjacent to the larger test section which houses the tested coil. Highlighted in Figure 1 are the insulated door and wall panels, the flow measurement apparatus (code tester), the conditioning coils, and the variable speed fans. Additionally, the piping layout of the conditioning subsystem is highlighted. The following sections expand on the operation of the facility and the analysis used to arrive at the final designs of these key sections/components.

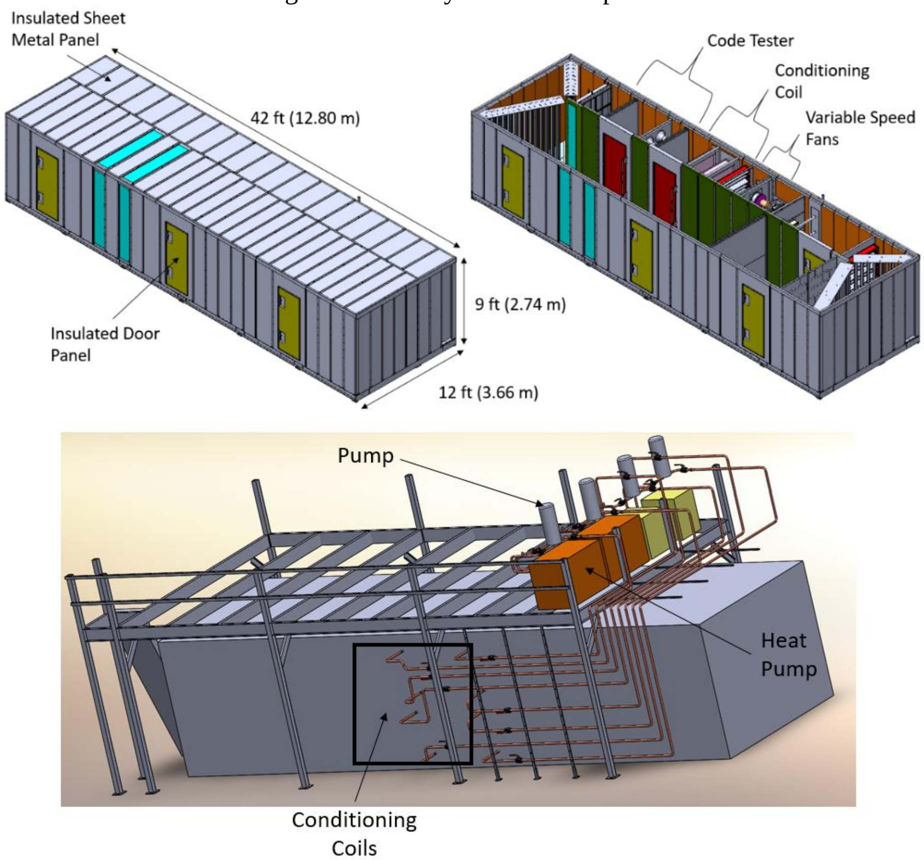

Figure 1: Global views of final coil testing facility design airside subsystem (top left) with roof panels removed and key sections highlighted (top right) and conditioning subsystem design relative to airside subsystem (bottom).

2.1 Airside Subsystem Design

The closed-loop airside subsystem contains the tested heat exchanger coil, along with the necessary conditioning equipment to recirculate air and achieve the desired steady-state conditions at the inlet of the tested coil. As air flows over the tested coil in the test section, the thermodynamic properties of air are modified through heat addition or rejection. Using the conditioning section, the air properties are then returned to the desired condition at the inlet of the tested coil prior to recirculation. A schematic of the airside subsystem can be seen in **Figure 2**. Air first travels through the tested coil, where it heated or cooled and/or dehumidified depending on the experiment. From the exit of the tested coil the air flows through two sets of turning vanes and enters the conditioning section of the facility. The conditioning section contains a series combination of air filters, an airflow measurement section (*i.e.* code tester), conditioning coils, variable speed fans, electric reheat, steam humidification, and dampers. The air filters prevent any large debris or contaminants from entering subsequent conditioning equipment. The code tester measures air flow rate. It contains a nozzle plane with upstream and downstream air settling means, placed according to ASHRAE Standard 41.2 specifications. Four conditioning coils, arranged into two pairs, provide heating or cooling. A pair of dampers, downstream of the conditioning coils, determine which conditioning coils will be active. Each coil can independently operate in heating or cooling mode. Variable speed fans provide the pressure rise needed to pass the air through all other components. Electric heaters provide reheat and are intended for precise air temperature control. Humidity control is achieved by a steam humidifier and injection manifold, allowing moisture to be reintroduced to the airstream. The damper located after the steam injection allows operation at reduced air flow rates by increasing static pressure on the fans. Upon leaving the conditioning section, air is returned to the test section via turning vanes. Air is then mixed to prevent temperature and humidity stratification throughout the cross section of the facility. Finally, a set of settling means creates a more uniform air flow distribution before again arriving at the inlet of the tested coil.

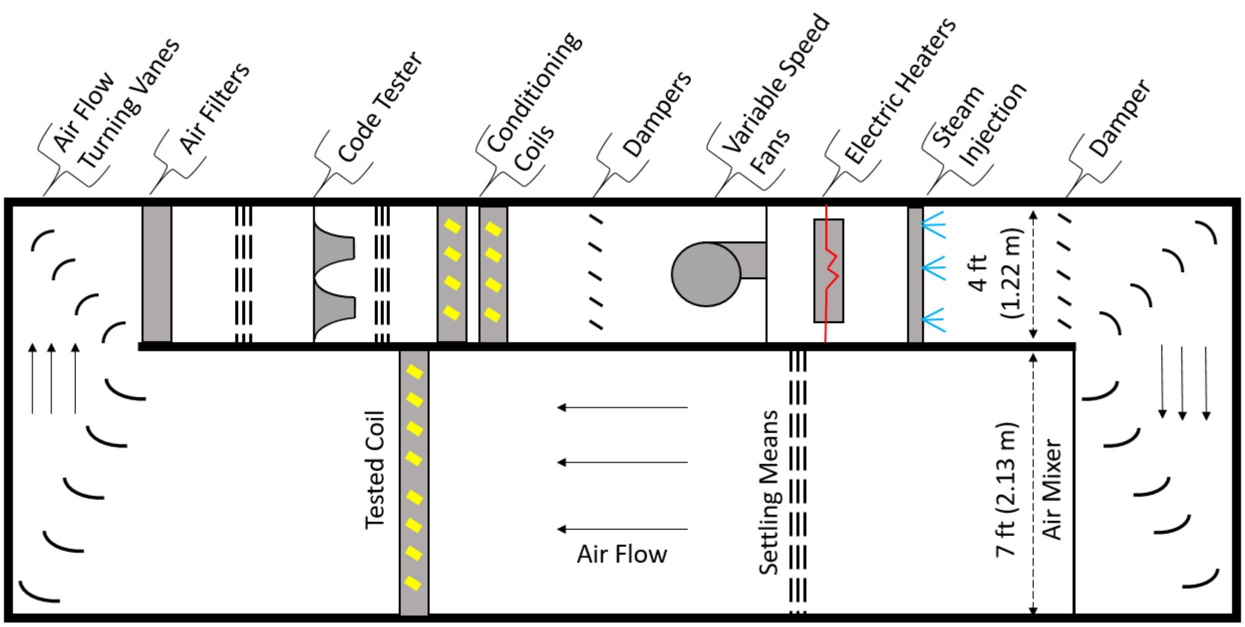

Figure 2: Simplified schematic of airside subsystem with major components identified.

2.2 Conditioning Subsystem Design

To manage the heat added or removed from the airside subsystem of the facility through the conditioning coils, a separate conditioning subsystem has been developed. The objective of this subsystem is to deliver the heating or cooling to the conditioning coils required by the airside subsystem. The conditioning subsystem consists of the following: water-to-water heat pumps, centrifugal pumps, thermal expansion tanks, and hydronic fin and tube conditioning coils. Each conditioning coil is connected to one water-to-water heat pump as seen in the schematic presented in Figure 3. The heat pumps collect heat managed by the conditioning subsystem, and either reject or collect it from the campus chilled water supply via an intermediate heat exchanger. The one-to-one connection of each coil and heat pump simplifies the operation of the conditioning subsystem. Also, additional capacity is present due to the fluid always entering the conditioning coils at the highest or lowest possible temperature available from the heat pumps.

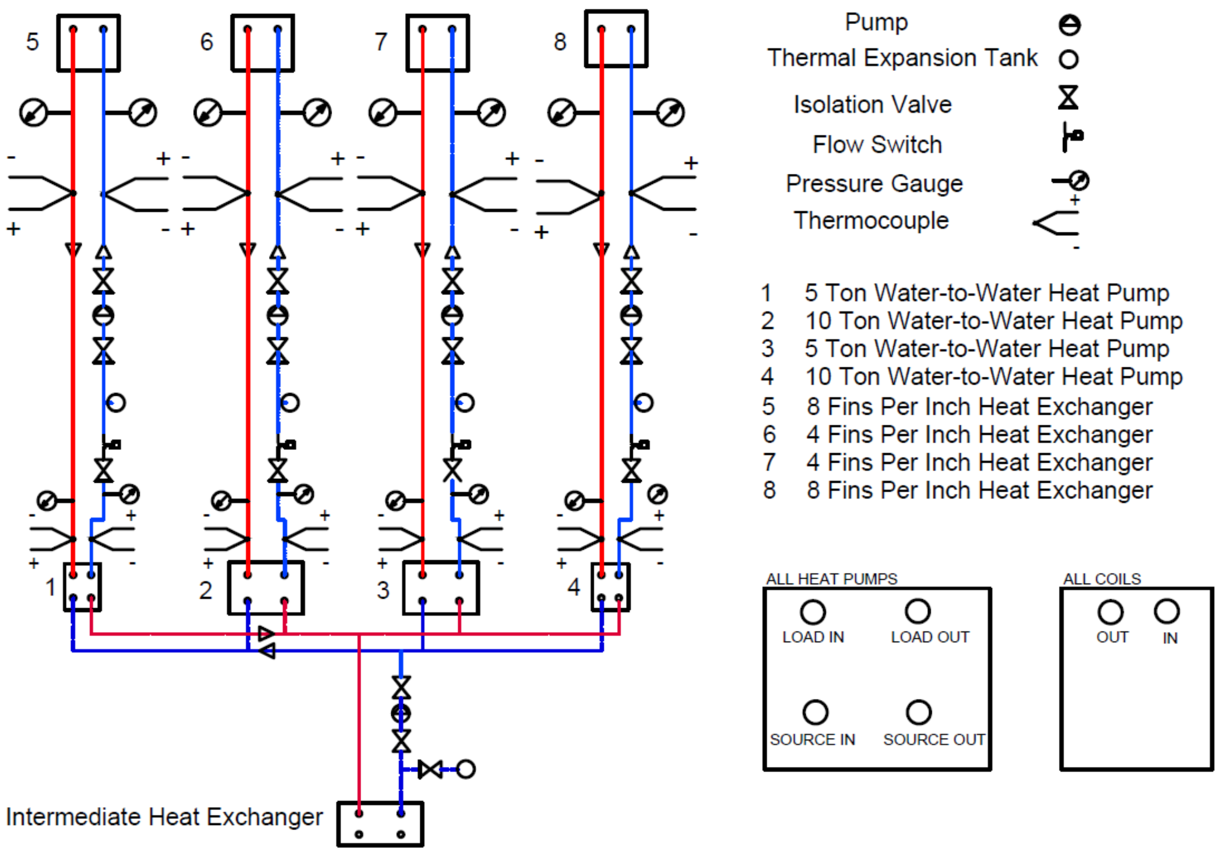

Figure 3: Piping schematic showing the connection of important individual components.

To support this equipment, a mezzanine was constructed on top of the airside facility with space to house the array of heat pumps and accompanying equipment. Figure 1 shows the physical layout of the piping in relation to the facility and mezzanine. The heat pumps were placed at the far end of the mezzanine to allow as much room for future development as possible. The piping was arranged in such a way as to not block access panels and doors of the facility, and obstruct as little of the mezzanine as possible. The development of the conditioning subsystem includes a few key design activities including piping layout, working fluid selection, pipe sizing and pump selection as presented in Section 3.

3. FACILITY COMPONENT AND SUBSYSTEM DESIGN

The psychrometric coil testing facility includes a variety of components and subsystems, with a few that warrant more detailed description. The critical components from the airside subsystem include structural materials and airflow measurement apparatus (code tester). The conditioning subsystem components of interest are piping layout, pipe sizing, fluid selection, and pump selection. Below is a detailed discussion of these aforementioned subsystems.

3.1 Facility Structure and Materials

Structural materials of the airside subsystem must be capable of creating an airtight seal for the closed loop to operate efficiently. The panels must also be able to withstand a large range of air conditions. Additionally, the facility must be able to support its own weight, be resistant to rusting, and minimize external condensation at low temperature testing. Galvanized sheet metal panels, filled with a polyurethane spray foam, used to construct rooftop air-conditioning units were sourced for this task from a local unitary equipment manufacturer. Cremaschi and Lee (2008) completed a heat transfer analysis on these panels when constructing a similar facility. Their study found

4 in (10.16 cm) thick panels provide adequate insulation to the surrounding room air temperature and prevent external surface condensation down to -20°F (-28.9°C). Additionally, the study results suggest the insulating value of 4 in (10.16 cm) thick panels is sufficient for continuous operation. The space between panels is occupied by a neoprene foam tape to create the required airtight seal. Figure 4 shows the process of connecting two panels together.

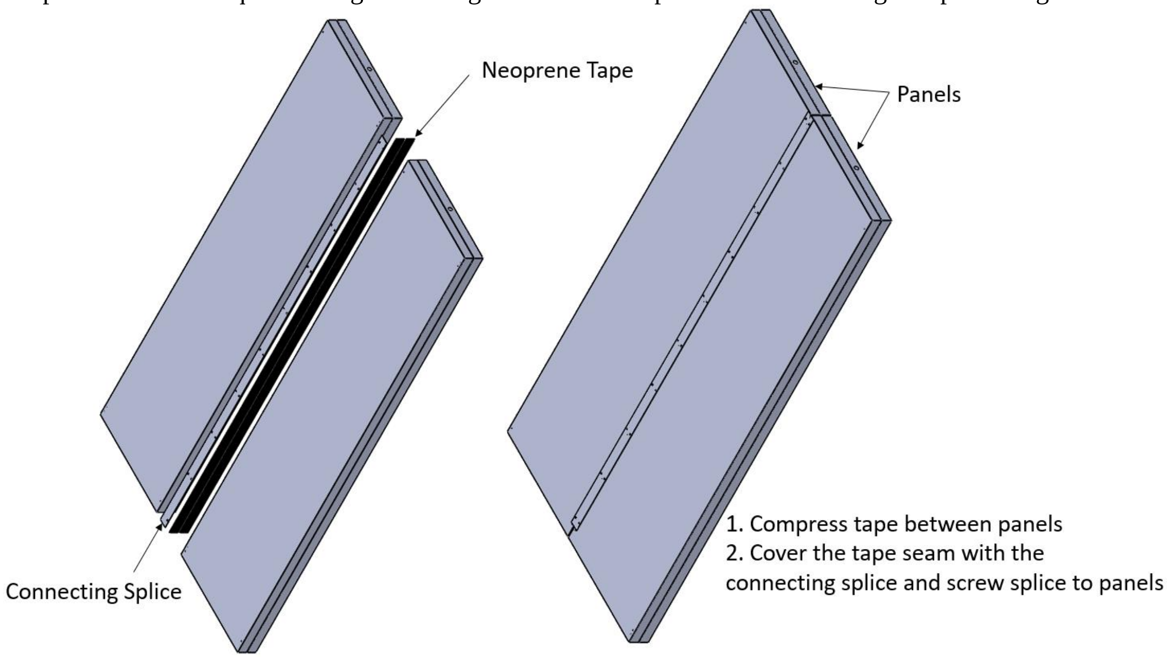

Figure 4: Process of assembling two panels.

3.2 Airflow Measurement Apparatus (Code Tester)

The multi-nozzle airflow measurement apparatus (code tester) allows for the calculation of air flow rate, a critical condition to consider when analyzing heat exchanger performance. Air flow rate is calculated by measuring the pressure drop across the nozzle plane. Nozzle size selection, positioning of nozzles within the nozzle plane, and positioning of air settling means relative to the nozzle plane are designed per ASHRAE Standard 41.2. The nozzle bank consists of six 5.5 in (13.97 cm) diameter, two 4 in (10.16 cm) diameter, and one 3 in (7.62 cm) diameter nozzles positioned within the nozzle plane according to ASHRAE Standard 41.2. This particular combination was selected to support the specified facility operation range of 1000 to 8000 CFM (0.47 to 3.78 m^3/s) air flow rate, while also adhering to the allowable nozzle throat velocities found in ASHRAE Standard 41.2.

Collins *et al.* (2016) suggested an alternative arrangement of upstream and downstream chamber components to improve overall accuracy of air flow rate measurement. They present a method to modify existing chambers by changing inlet and exit airflow measurement apparatus (code tester) lengths. Figure 5 shows diagrams comparing the current facility airflow measurement apparatus (code tester) to that seen in the ASHRAE Standard 41.2 and in Collins *et al* (2016). To improve air flow rate accuracy, an increased entrance length before the upstream air settling means is implemented according to the results of Collins *et al.* (2016), while maintaining the recommended exit length from ASHRAE Standard 41.2. These considerations increase the overall airflow measurement apparatus (code tester) length by approximately 10% and the facility remains ASHRAE Standard 41.2 compliant.

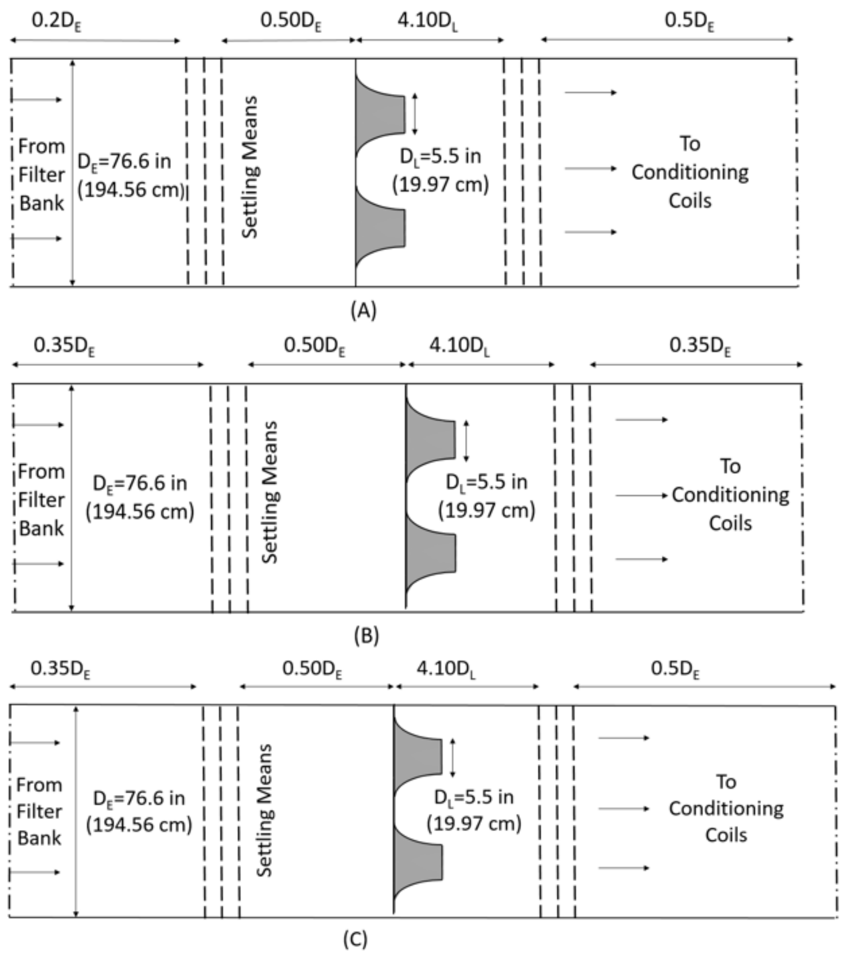

Figure 5: Comparison of (A) ASHRAE Standard 41.2, (B) Collins et al. and (C) facility air flow rate measurement.

3.3 Working Fluid Selection

The desired operational envelope 0°F (-17.78°C) to 140°F (60°C) immediately excludes water as a working fluid. A fluid with similar properties to water was desired to be able to use the existing heat exchangers in the heat pumps. Below is a table of attributes which shows the comparison of three fluids which were considered for the conditioning subsystem. While silicone oil has very attractive attributes it was eliminated due to high cost. Potassium formate had reasonable attributes across the board. However, inhibited ethylene glycol has both the lowest price and comparable properties to the potassium formate. A 33% ethylene glycol to water mixture was chosen to lower the freezing point of the mixture to a value which would allow facility operation at air temperatures of 0°F (-17.78°C).

Table 2: Potential working fluid attributes.

Parameter of Interest	Inhibited Ethylene Glycol	Low-Temperature Heat Transfer Fluid (Potassium Formate)	Silicone Oil
Toxicity	Average	Good	Very Good
Corrosion Protection	Average	Average	Excellent
Performance	Good	Very Good	Excellent
Material Compatibility	Very Good	Very Good	Good
Price	Very Good	Average	Very Poor

3.4 Pipe Sizing

Using the piping layout previously presented, the predicted frictional losses were determined for different diameters of pipe. The piping was sized to limit pressure drop to 4 ft of water (11.96 kPa) per 100 ft (30.48 m) in accordance with the flow rate limitations described by Steward and Dona (1987) in the ASHRAE Handbook (ASHRAE, 2009). Frictional losses were calculated using water frictional loss correlations. Appropriate ethylene glycol and water mixture correction factors were incorporated to ensure the accuracy of the predicted frictional losses. Determining the diameter of pipe required was a balance between conditioning coil performance and cost of piping. Larger diameters lead to higher capacity and lower frictional losses, but at the expense of space and increased cost. A diameter of 1.5 in (3.81 cm) was selected as a reasonable balance between cost and capacity.

3.5 Pump Selection

A pump was selected for each of the four circuits using the selected pipe size and calculated frictional losses. The major contributors of losses were the heat exchanger circuits of the heat pumps and conditioning coils. The larger heat pumps require 30 GPM (1.89E-3 m^3/s) while the smaller heat pumps 15 GPM (9.46E-4 m^3/s), a system curve of the conditioning subsystem was calculated to match against a pump curve. Similar to the pipe sizing, an ethylene glycol and water mixture correction factor was used in scaling the pump curve to meet the corrected system curve and finally one pump was selected for each loop in the conditioning subsystem.

3.6 Conditioning Coil Arrangement

Special consideration was given to the arrangement of the conditioning coils for maximum experimental flexibility. The arrangement of coils can be used to mitigate frosting in low temperature tests. Frosting on coil fins lowers the total capacity the coil is able to provide. Additionally, experiments may need to halt to allow coils to defrost. To avoid downtime during testing, the four conditioning coils are positioned in two pairs as shown in Figure 6, with 8 fins per inch (FPI) coils located upstream and 4 FPI coils located downstream. Air entering the 4 FPI coil will be colder than that seen by the 8 FPI coil when the conditioning coils are operating in cooling mode. Frost will be less likely to completely block airflow through the 4 FPI coil since the space between fins is larger than the fin spacing in the 8 FPI coil. Since the coils are arranged in pairs, it is possible to run cooling fluid in either the upper or lower pair of coils while the second set is able to defrost with warmer working fluid. Airflow will be directed through the appropriate coils via electronically controlled dampers. Total cooling capacity will be lowered when only using one pair of conditioning coils.

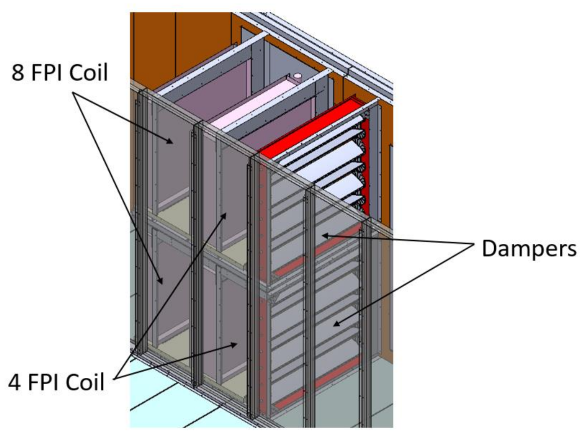

Figure 6: Conditioning coil arrangement in the conditioning section.

4. Preliminary Capacity Analysis

A critical test metric of the facility will be the total capacity of a coil that is able to be tested. The operating envelope set an objective of 23 tons (80.9 kW) of capacity during the design process. A preliminary capacity analysis is presented using data/models from the equipment selected, to evaluate how close tested coil capacity is to the objective at various conditions. The preliminary capacity analysis assumes a tested coil in heating mode with cooling provided by the conditioning coils, heat added by the blower motors, electric reheat, and adiabatic mixing from steam addition. An energy balance showing the maximum heating capacity a tested coil can achieve is given as,

$$\dot{m}_{air}\left(h_{air,in} - h_{air,out}\right) = -\dot{m}_{water}\left(h_{water,in} - h_{water,out}\right) - \dot{W}_{blowers} - \dot{W}_{elec,reheat} - \dot{m}_s h_s \,. \qquad (1)$$

Figure 7 shows the control volume of the airside subsystem showing the energy transfer considered.

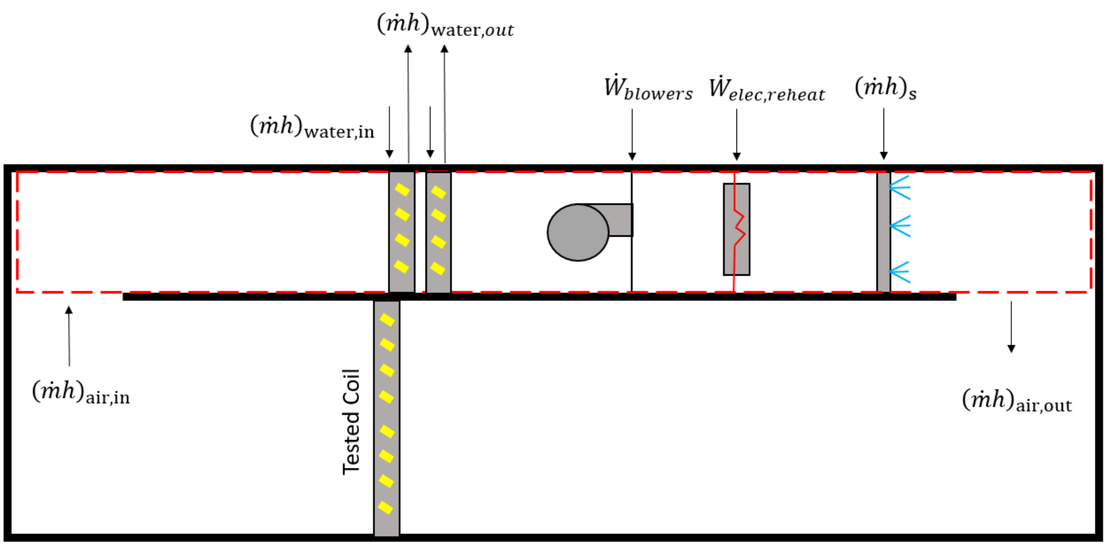

Figure 7: Airside subsystem conditioning section energy balance.

To determine the capacity of the tested coil, it is necessary to determine each of the other energy transfers from Equation 1. First, the energy fluxes in and out of the conditioning coils,$\dot{m}h_{water}$, are found by considering an energy balance about the conditioning subsystem. By assuming there is not significant heat losses in the piping, due to insulation, between the heat pumps and the conditioning coils, the capacity delivered to the airside subsystem is equal to that delivered by the heat pumps.

Therefore, by equating the heat pump capacity predicted by the manufacturer's datasheet and the capacity predicted by the conditioning coil manufacturer's design software, the appropriate steady state energy flux is predicted. This balances energy for a given inlet air condition in the airside subsystem. The outcome of this analysis is the total capacity delivered to the airside subsystem, as well as, the water inlet and exit conditions to the conditioning coils. The total conditioning coil capacity predicted can be seen in

Figure 8 for various air temperatures and a range of relative humidity entering the conditioning coils. Only a small number of capacity points could be determined due to limited heat pump submittal data.

Next, $\dot{W}_{blowers}$ is then taken as the maximum motor horsepower, 20 HP (14.9 kW), used by the blowers. Steam re-humidification is considered to return air absolute humidity to that of the inlet of the conditioning coils. Electric reheat was not considered unless the steam re-humidification resulted in a relative humidity ≥ 100%. Relative humidity ≥ 100% creates adverse testing conditions, such as fog or rain. For these cases, electric reheat was added to lower relative humidity to 95%. These additional energy transfers functionally reduce the capacity available to the tested coil.

With all energy transfers considered, the tested coil capacity, as a function of entering air temperature, was determined and can be seen in

Figure **9**. It is apparent from the figure that electric reheat addition reduces capacity. Also, as relative humidity decreases, tested coil capacity increases. This increase in capacity can be attributed to steam re-humidification, because to achieve higher relative humidity conditions, more steam re-humidification is needed compared to lower relative humidity conditions. With larger amounts of steam mixed into the airflow, the overall tested coil capacity is reduced.

A capacity rating expected from the tested coil can be determined at an AHRI specified rating condition. For air-cooled condensers, AHRI 340/360 (AHRI, 2015) states that the wet-bulb temperature condition is not required for the Low Temperature Operating Cooling condition of 67°F (19.4°C) dry-bulb temperature. This will be the operating condition of interest for this analysis. With air entering the tested coil at 67°F (19.4°C) and 30% relative humidity, the tested coil capacity is approximately 23 tons (80.9 kW). This value will serve as the rating for this facility.

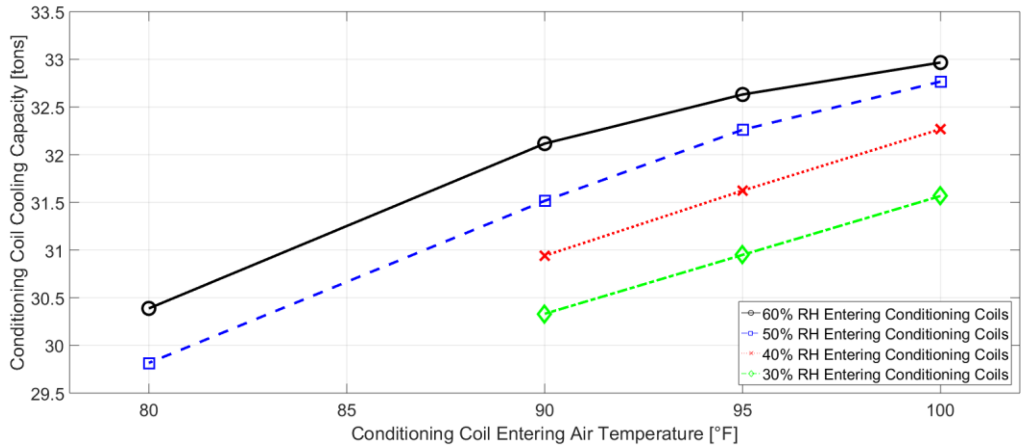

Figure 8: Conditioning coil cooling capacity versus conditioning coil entering air temperature.

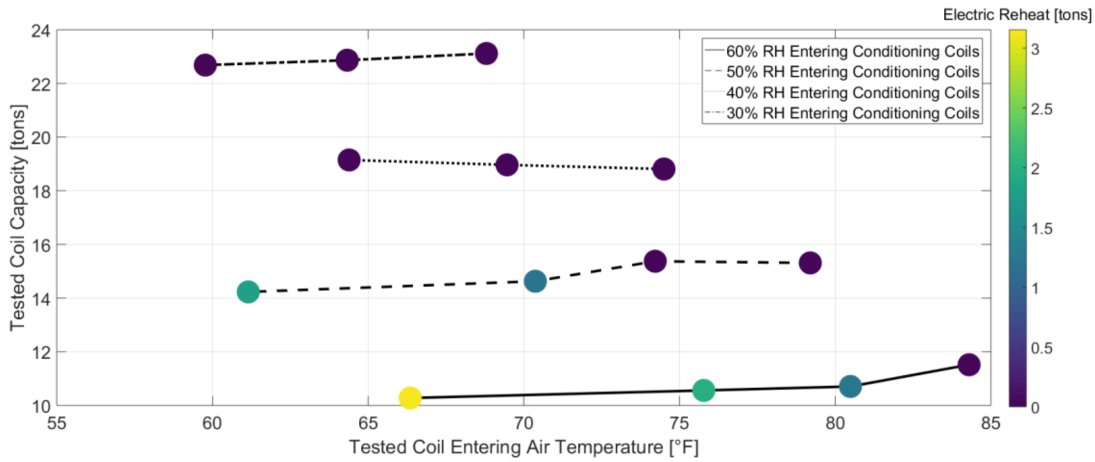

Figure 9: Tested coil capacity versus tested coil entering air temperature.

5. CONCLUSION AND FUTURE WORK

The design of the airside and conditioning subsystems for a new psychrometric coil testing facility for the evaluation of commercial sized heat exchanger coil performance is presented. Design considerations are presented which give greater detail into design decisions made. Finally, preliminary capacity calculations were completed to estimate the capacity available for the tested coil. The facility is currently under construction. Upon completion, heat exchanger testing can occur. Experimental data from these tests can then be used to create empirical models or tune gray box models of heat exchangers and their components. Such models will allow manufacturers to reduce the number of design, build, and test iterations needed during equipment development.

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NOMENCLATURE

D_E	Equivalent duct diameter	in (cm)
D_L	Largest nozzle throat diameter	in (cm)
$\dot{m}_{air}$	Mass flow rate of air	lbm/hr (kg/s)
h_{air}	Enthalpy of air	BTU/lbm (kJ/kg)
$\dot{m}_{water}$	Mass flow rate of water	lbm/hr (kg/s)
h_{water}	Enthalpy of water	BTU/lbm (kJ/kg)
$\dot{W}_{blowers}$	Electric heat addition from blowers	BTU/hr (W)
$\dot{W}_{elec,reheat}$	Electric heat addition from electric heaters	BTU/hr (W)
$\dot{m}_s$	Mass flow rate of steam	lbm/hr (kg/s)
h_s	Enthalpy of saturated water vapor	BTU/lbm (kJ/kg)

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