

IMPROVING REFRIGERATION CYCLE EFFICIENCY BY EVAPORATIVE
PRECOOLING THE CONDENSER INLET AIR

A Thesis

by

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ABSTRACT

The need to improve the energy performance of Heating, Ventilation & Air Conditioning (HVAC) units has always been a primary goal for designers as they implement new concepts and/or modifications to the standard vapor compression cycle. Recently, this primary goal has taken on an added importance due to the phasing out of conventional refrigerants that harm the environment. The newer environmentally safe refrigerants that are being considered as replacements have been shown to hinder the performance of HVAC units. A new method to increase the performance of HVAC units is investigated herein with the approach based on the integration of evaporative precooling to the air side of an HVAC system's condenser.

The goal of this project is two-fold, namely to develop a facility and methodology to test the performance of HVAC units with precooling applied to the HVAC unit's condenser, and then to perform and evaluate the results of testing. The psychometric testing facility at Texas A&M was restored to operation in accordance with AHRI 210/240 standard and then modified in order to accommodate the addition of an evaporative cooler. The testing facility allowed for the complete control of temperature and relative humidity of two separate rooms in order to simulate indoor and outdoor conditions.

The efficiencies of the HVAC unit were shown to improve with the implementation of evaporative precooling. The improvement of the coefficient of performance (COP) revealed to improve by a greater amount when operating at a higher

dry bulb temperature in the outdoor room. The reason for this increased improvement at higher outdoor temperatures was that the evaporative cooler was able to impart a larger reduction in the in dry bulb temperature entering the condenser. For example, at the highest dry bulb temperature testing condition, Test C (110°F), the evaporative cooler was able to provide 18°F of precooling. The effect of reducing the high-side refrigerant condition when precooling was applied to the condenser is a reduction in power consumption by the compressor, especially at higher outdoor dry bulb temperatures. For example, a reduction of 327 watts was observed at Test C, significantly offsetting the evaporative cooler's 25 watt water pump power. The largest improvement in COP, namely 23%, occurred at the C Test conditions, which corresponded to the highest outdoor temperature of 110°F.

DEDICATION

This thesis is dedicated to my family for their love and support as I've journeyed through life. Their unwavering love and guidance has allowed me to pursue my dreams and become the best version of myself.

I would also like to thank my brothers and sisters in the military for the guidance they provided during my time in the Navy. I would encourage all of my military family to pursue their goals in life and to continue their education after completing their service. I wish you guys the best.

CONTRIBUTORS AND FUNDING SOURCES

Contributors

This work was supported by a thesis committee consisting of Dr. Pate and Dr. Pharr of the Department of Mechanical Engineering and Dr. Alvarado of the Department of Engineering Technology and Industrial Distribution.

All work conducted for this thesis was completed the student independently.

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NOMENCLATURE

AHRI	Air-conditioning, Heating, and Refrigeration Institute
AMB	Atmospheric Pressure, in. Hg
ASHAE	American Society of Heating & Air Conditioning Engineers
ASHRAE	American Society of Heating, Refrigerating & Air Conditioning Engineers
ASME	American Society of Mechanical Engineers
ASRE	American Society of Refrigerating Engineers
CFC	Chlorofluorocarbons
COP	Coefficient of Performance
COP _R	Coefficient of Performance for Refrigerant System
COP _{R&E}	Coefficient of Performance for Refrigerant and Evaporative Cooler Combination
DAQ	Data Acquisition
DB	Dry Bulb
DP	Differential Pressure, inches of water
HCFC	Hydrochlorofluorocarbons
HVAC	Heating Ventilation & Air Conditioning
IR	Indoor Room
NI	National Instruments
OR	Outdoor Room
P	Pressure, inches of water
REEL	RELLIS Energy Efficiency Laboratory

RH	Air Relative Humidity, %
RTD	Resistance Temperature Detector
s	Entropy
T	Temperature
VCR	Vapor Compression Refrigeration
VFD	Variable Frequency Drive
WB	Wet Bulb
W	Watts
gpm	Gallons per minute,
cfm	Cubic feet per minute
ϵ_{Evap}	Evaporative Cooler Performance
Q_H	Cooling Rate to Hot Reservoir, Btu/hr
Q_L	Cooling Rate Out of Cold Reservoir, Btu/hr
$q_{sensible}$	Evaporator Sensible Cooling Rate, Btu/hr
q_{latent}	Evaporator Latent Cooling Rate, Btu/hr
$q_{cooling}$	Condenser Sensible Cooling Rate, Btu/hr
W_{in}	HVAC Work Input, Watts
\dot{m}_R	Refrigerant Mass Flowrate, lbm _R /min
h_{1a}	Enthalpy of Air: Inlet Evaporator, Btu/lbm _a
h_{2a}	Enthalpy of Air: Outlet Condenser, Btu/lbm _a
h_{3a}	Enthalpy of Air: Inlet Condenser, Btu/lbm _a
h_{4a}	Enthalpy of Air: Outlet Evaporator, Btu/lbm _a

h_{1R}	Enthalpy of Refrigerant: Outlet Evaporator, Btu/lbm _R
h_{2R}	Enthalpy of Refrigerant: Inlet Condenser, Btu/lbm _R
h_{3R}	Enthalpy of Refrigerant: Outlet Condenser, Btu/lbm _R
h_{4R}	Enthalpy of Refrigerant: Inlet Evaporator, Btu/lbm _R
\dot{V}_a	Volumetric Airflow, cfm
v_a	Specific Volume of the Dry Air Portion of the Mixture, ft ³ /lbm _a
c_{ca}	Air Specific Heat: Outlet Condenser, Btu/lbm _a
c_{pa1}	Air Specific Heat: Inlet Evaporator, Btu/lbm _a
c_{pa2}	Air Specific Heat: Outlet Evaporator, Btu/lbm _a
T_{aceD}	Air Dry Bulb Temperature: Inlet Condenser, °F
T_{acoD}	Air Dry Bulb Temperature: Outlet Condenser, °F
T_{acoW}	Air Wet Bulb Temperature: Outlet Condenser, °F
T_{aeiD}	Air Dry Bulb Temperature: Inlet Evaporator, °F
T_{aeoD}	Air Dry Bulb Temperature: Outlet Evaporator, °F
A_N	Nozzle Area, in ²
P_v	Static Differential Pressure at Nozzle, inches of water
W_{1a}	Air Humidity Ratio: Inlet Evaporator, lbm _v /lbm _a
W_{2a}	Air Humidity Ratio: Outlet Evaporator, lbm _v /lbm _a

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1. INTRODUCTION

In recent years the global market for Heating, Ventilation, and Air Conditioning (HVAC) systems has been altered due to the worldwide banning of certain types of refrigerants. Specifically, conventional refrigerants are being phased out due to potential global warming and ozone depletion concerns. The replacement refrigerants that are being considered may be more environmentally friendly, but their use can have adverse effects such as lowering the coefficient of performance (COP) of the vapor compression refrigeration (VCR) systems. Furthermore, the few refrigerants available that do not reduce COP's have flammability concerns that vary from mild to severe.

One of the methods currently under investigation for recovering some of the lost efficiency by the newer ozone-safe refrigerants is modifications to conventional methods such as the integration of an evaporative cooler to a VCR cycle. The goal of this project is to introduce an evaporative cooling system at the air inlet to the condenser of a VCR cycle and then to determine the effects on the system's COP. By precooling the air entering the condenser, the result can be a lowering of the refrigerant-side pressure for a given refrigerant-to-air differential temperature, which then results in a lower pressure differential across the compressor. This lower differential pressure improves the efficiency of the system because a smaller amount of power is required than that of the compressor in a standard VCR cycle. With the compressor being the largest source of power consumption in an HVAC unit, any reduction in the compressor's power demand would lead to an increase in efficiency or COP for the entire HVAC unit.

2. THEORY

2.1 Vapor Compression Refrigeration Cycle

A vapor compression cycle involves a refrigerant transferring thermal energy (i.e. heat) from a low temperature source, which is often the inside of a building, and then rejecting the thermal energy to a high temperature environment, namely the outdoors. Figure 1 shows a vapor compression cycle and the various states that the refrigerant must undergo when cooling a heat source. Using Figure 1, the various states that the refrigerant must undergo when cooling a heat source are described. Heat from the source (Q_L) is transferred to the refrigerant flowing through an evaporator, from states 4 to 1, which operates at a low temperature and pressure in the VCR cycle. The refrigerant leaving the evaporator is then compressed into a superheated vapor, from states 1 to 2, by the compressor (W_{in}). The high pressure, high temperature vapor exits the compressor, which then transfers its heat (Q_H) to the environment through a condenser, from states 2 to 3. The exiting refrigerant, state 3, from the condenser is now a saturated liquid at a high pressure and low temperature. The expansion valve located downstream of the condenser then drops the pressure, and hence temperature, of the refrigerant, from states 3 to 4, so that it enters the evaporator as a saturated vapor-liquid mixture, which takes us to the beginning of the VCR cycle.

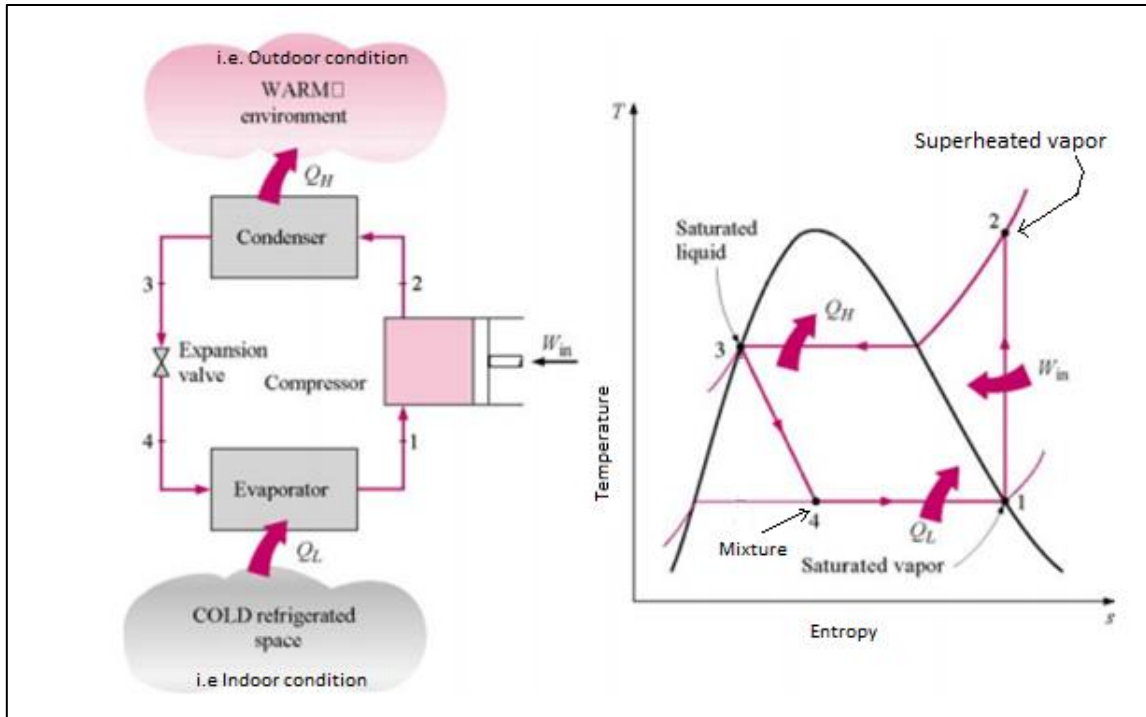


Figure 1: Diagram for a vapor compression refrigeration cycle. [1]

By measuring the heat transfer that occurs within an HVAC unit, the coefficient of performance (COP) can be measured. The heat transfer entering the refrigerant (Q_L) is calculated from an energy balance and measured, while the electrical power needed to power the HVAC unit (W_{in}) is measured by a wattmeter. COP can be found as follows.

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{in}} \quad \text{Eq. (1)}$$

The thermal energy transferred out of the indoor space (Q_L) can be calculated by performing an energy balance on the evaporator with a control volume focusing on the refrigerant side. Specifically, the mass flowrate of the refrigerant (\dot{m}_R) is multiplied by the difference in enthalpy from states 1 and 4 as shown in Figure 1. Additionally, the energy transfer out of indoor space air passing through the evaporator can be calculated by equating it to the energy that goes into the refrigerant. The flowrate of air is calculated by the relation of mass flowrate to volumetric flow (\dot{V}_a) and density of the air (v_a). In summary, these are two methods based on enthalpy changes on the refrigerant and airside of the evaporator for calculating the thermal energy out of the indoor space.

$$\dot{Q}_L = \dot{m}_R (h_{1R} - h_{4R}) = \frac{\dot{V}_a}{v_a} (h_{4a} - h_{1a}) \quad \text{Eq. (2)}$$

Following a procedure similar to that used above for the evaporator, the thermal energy transferred out of the refrigerant onto the outdoor environment (Q_H) is calculated. Specifically, the mass flowrate of the refrigerant (\dot{m}_R) is multiplied by the difference in enthalpy passing through the condenser, from states 3 and 2 as shown in Figure 1. Additionally, the energy transfer onto the outdoor space air can be calculated by equating it to the energy that is removed from the refrigerant. As with the evaporator, there are two methods used for calculating the thermal energy onto the outdoor space.

$$\dot{Q}_H = \dot{m}_R (h_{3R} - h_{2R}) = \frac{\dot{V}_a}{v_a} (h_{2a} - h_{3a}) \quad \text{Eq. (3)}$$

The work input to the VCR cycle is in the form of compression of the refrigerant (W_{in}). This term can be measured by a power meter, or by the power required to compress the refrigerant. It is calculated by using refrigerant enthalpies from state 1 to 2 as shown in Figure 1.

$$W_{in} = \dot{m}_R (h_{2R} - h_{1R}) \quad \text{Eq. (4)}$$

2.2 Evaporative Coolers

The primary method being investigated in this study for recuperating the loss of efficiency from the new environmentally friendly refrigerants is the implementation of an evaporative cooler at the air-side inlet of the HVAC's condenser. The evaporative cooler works on the premise of saturating the air with moisture therefore causing the entering air temperature to decrease. The direct evaporative cooling method was used for this investigational work. This method functions by passing air directly through a wet membrane so as to saturate the air with moisture thus approaching the wet bulb temperature condition as shown on Figure 2. An example of the effects that evaporative cooling can have on an air sample are displayed on the Figure 3 psychometric chart. This example presents an air sample at 90°F and 60% relative humidity being fully saturated to a 100% humidity condition. As this process occurs, the dry bulb temperature is reduced to approximately 78°F, a reduction of 12°F, occurring by simply adding water to the air. This lower air temperature can play a key role in the condensers' performance of the VCR cycle in that the refrigerant-to-air temperature difference is unchanged but the

condenser temperature and hence pressure decreases by a similar amount. With a lower pressure at the high-side outlet of the condenser, the compressor needs less work while transferring a similar amount of heat from the indoor space. The reduction in work required by the compressor results in an increase of VCR cycle COP.

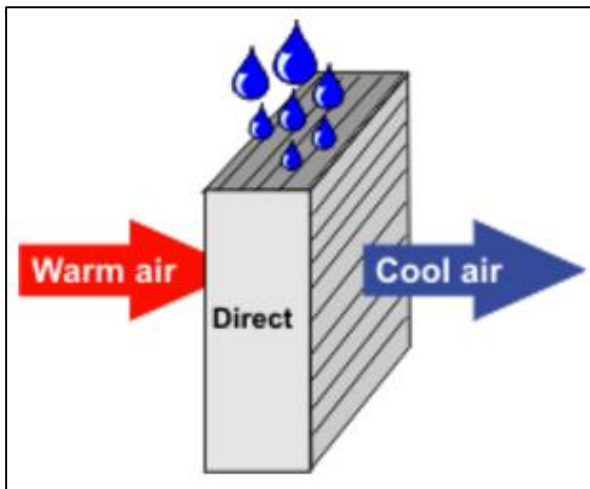


Figure 2: Example of the direct evaporative cooling method.

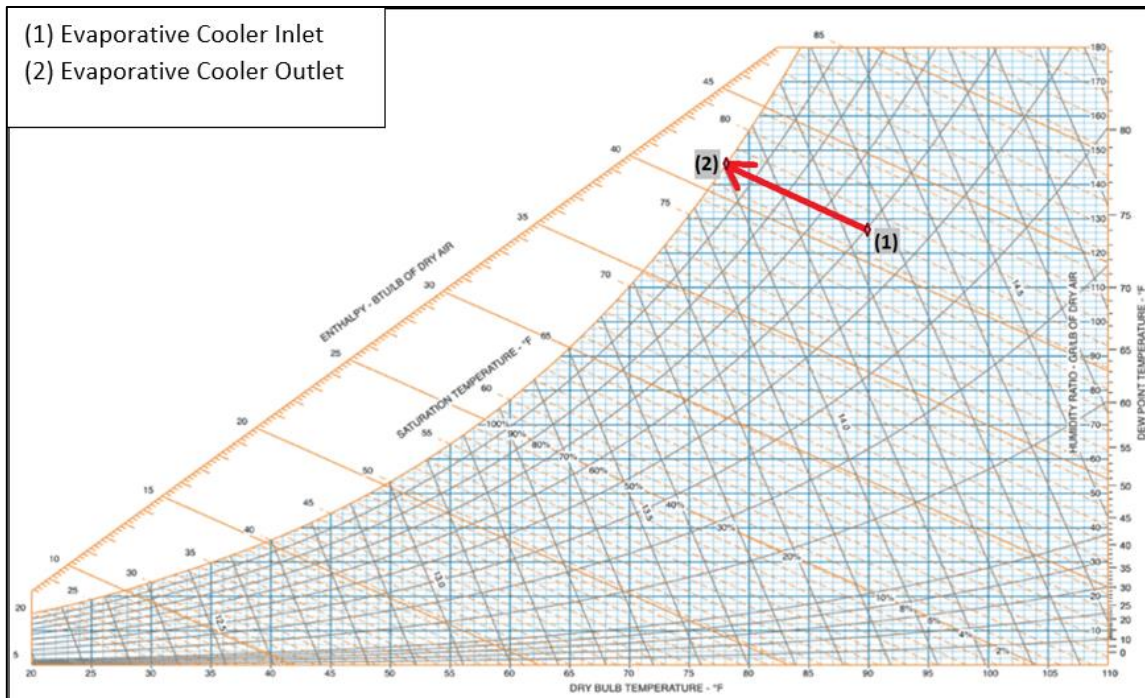


Figure 3: Condenser inlet air path through an evaporative cooler

2.3 Implementing Evaporative Precooling at Condenser Inlet

The evaporative cooler that was installed in this study at the outside air condenser inlet of the VCR cycle allows for the lowering of air temperature by saturating the air with water. The lowered temperature air then acts as the heat sink for the refrigerant rejecting its heat to the environment. As a result, the refrigerant pressure and temperature at the discharge of the condenser, point 3 on Figure 4, will be lower than it is for the non-evaporative cooler VCR case. Since the refrigerant is entering the expansion valve at a lower temperature than before, the refrigerant will have a reduced enthalpy as it enters the evaporator which in turn transfers more heat out of the refrigerated space (Q_L). Even more importantly, the work (W_{in}) going into the VCR

cycle will also be lower due to the compressor having to compress the vapor to a lower pressure. This reduction in compressor work associated with the introduction of evaporative precooling to the condenser can be seen on the Figure 4 temperature entropy diagram.

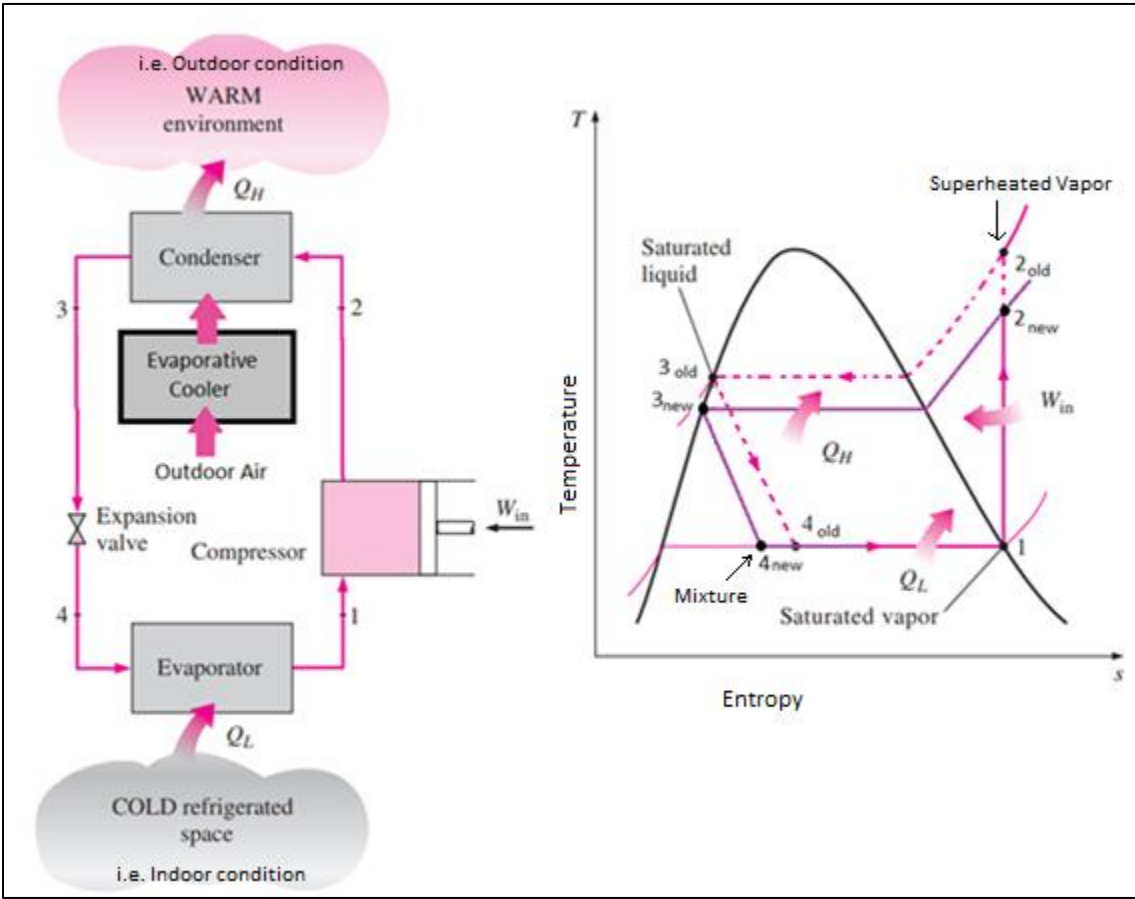


Figure 4: Schematic of a VCR cycle with the introduction of evaporative precooling to the condenser inlet air. [1].

The increase of Q_L and reduction in W_{in} results in an overall increase in the coefficient of performance when compared to that of the VCR cycle alone. The effects on the COP of the VCR cycle and evaporative cooler combination ($COP_{R\&E}$) are as follows.

$$COP_{R\&E} = \frac{\dot{Q}_L \uparrow}{\dot{W}_{in} \downarrow} \quad \text{Eq. (5)}$$

3. TEST METHODOLOGY

3.1 Testing Facility

The psychrometric room at the REELIS Energy Efficiency Laboratory (REEL) was restored to service after being non-operational for several years in accordance with the AHRI 210/240 and ASHRAE 37 standards. Figure 5 displays the testing facility at Texas A&M that was used to conduct experiments. The psychrometric room is composed of two separate rooms that allow for complete control of the environment when testing HVAC units. The rooms were built in order to simulate outdoor and indoor conditions by controlling both temperature and humidity. Additionally, the rooms were designed for testing systems with cooling capacities up to 10 tons.



Figure 5: Psychrometric test chamber at REEL facility.

The environment within the two rooms is controlled by an array of heating, cooling, humidifying, and dehumidifying equipment, plus both rooms have air circulation systems that recirculate air back into the rooms by using blower fans. The arrangement of these two rooms was designed in order to comply with the testing of psychometric conditions by using the room airside enthalpy method in accordance with the ASHRAE 37 standard. A schematic of all the instrumentation and equipment for the room enthalpy testing method can be seen on Figure 6. The air temperatures within the rooms are controlled with heaters and a chill water system. A 75 ton screw chiller was used in order to reduce environmental temperatures within the room, while the temperatures were increased with the usage of three 40 kW electrical heaters. The chill water system distributes 1000 gallons of coolant with the usage of two centrifugal pumps that were controlled with via variable frequency drives (VFD). Humidity within the rooms was increased with the usage of steam from a boiler and decreased with the usage of a MiniPac 100 dehumidifier.

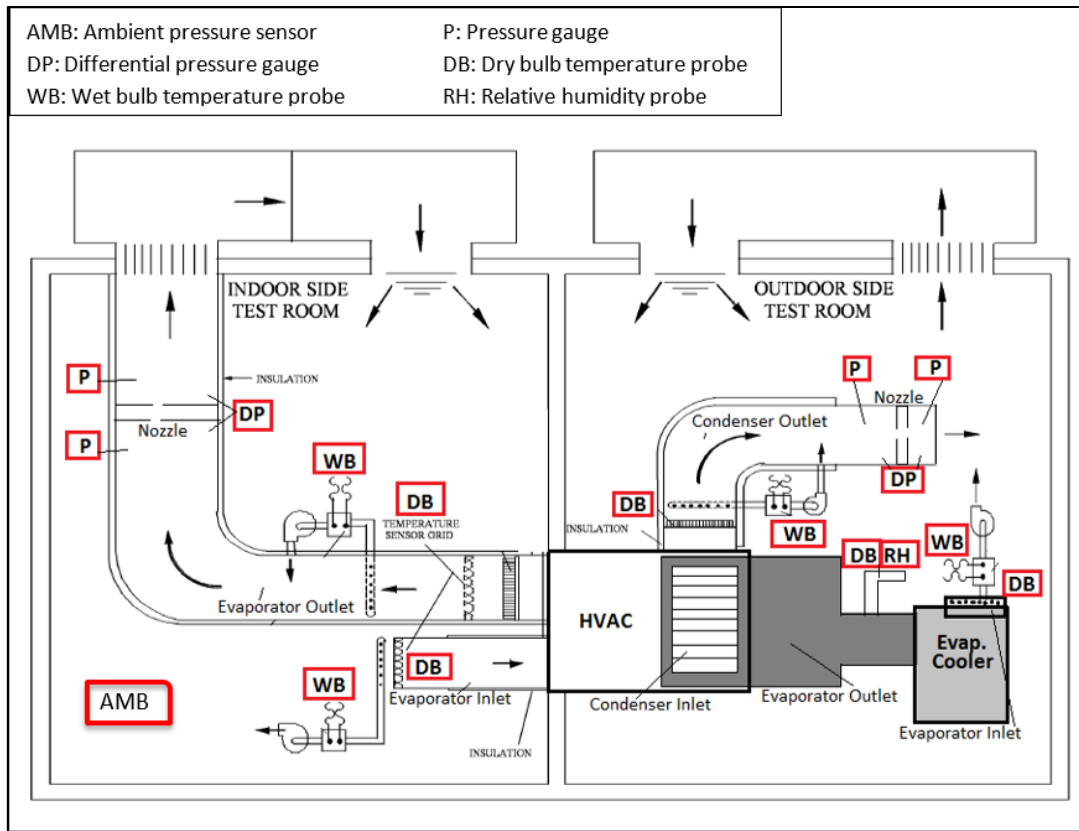


Figure 6: Psychrometric facility layout and instrumentation location.

The HVAC's condenser and evaporator air mass flowrates were measured by using a nozzle airflow apparatus. The nozzle airflow measuring apparatus was built in accordance with the ASHRAE 37 standard. An example of the layout can be seen in Figure 7, and the nozzle configuration sizing capabilities are listed in Table 1.

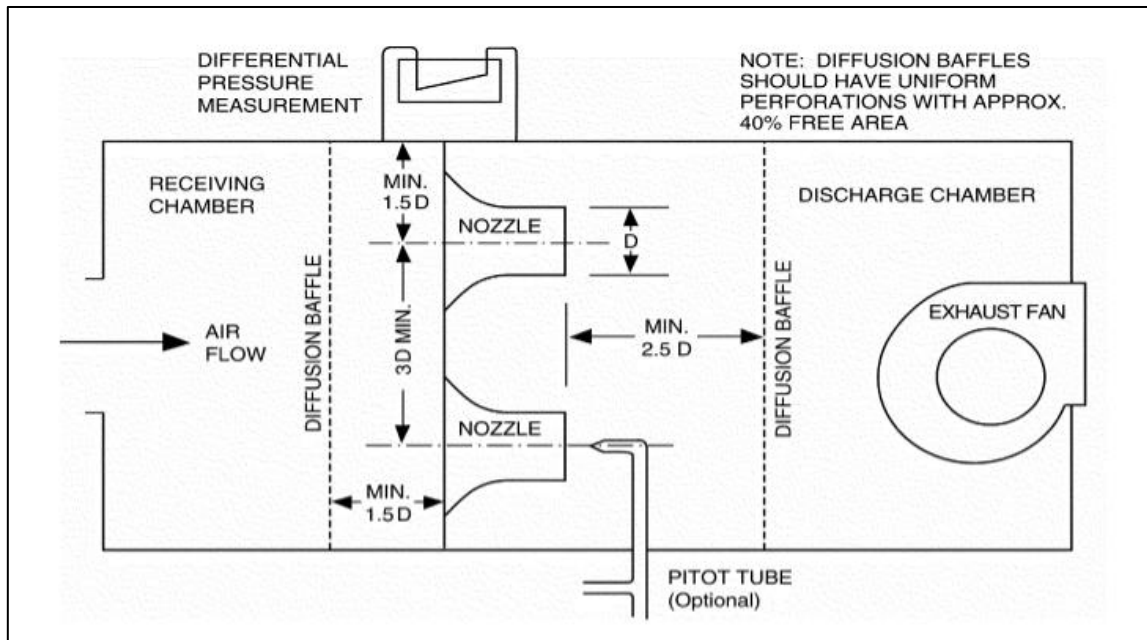


Figure 7: Nozzle airflow measuring apparatus [2].

Table 1: Airflow measuring apparatus nozzle configuration capabilities.

HVAC Component Side	Nozzle Diameters (inches)
Evaporator	Four Nozzles: 4,6,6,7
Condenser	Six Nozzles: 3,3,5,5,5,7

3.2 Evaporator Integration Modifications on the Testing Facility

The standard psychometric testing facility was modified by installing an evaporative cooler at the airside inlet of the HVACs' condenser. A casing unit was built around the condenser's inlet suction ports in order to better control and distribute the airflow from the evaporative cooler to the condenser. This distribution modification also optimizes the ability of the evaporative cooler to fully saturate the air entering the condenser. Figure 8 and Figure 9 display the differences in the testing facility's layout before and after the evaporative cooler modifications were made to the testing room and HVAC unit.



Figure 8: HVAC unit installed in the psychometric station before the integration of an evaporator cooler.



Figure 9: Evaporative cooler being integrated onto an HVAC system.

3.3 HVAC and Evaporative Cooler Equipment Used in the Test Setup

An off the shelf, commercially available 2 ton air-conditioning and heating unit charged with R-410A from Goodman was used as the main HVAC cooling unit for the experimental setup in this study. The Goodman HVAC unit contains both air conditioning and heat pump capabilities with a nominal capacity of 2 tons (24,000 Btu/hr). A list of the model number for the Goodman HVAC unit is shown on Table 2.

A Portacool Cyclone 130 direct evaporative cooler was selected to precool the condenser air during this investigation. The evaporative cooler was capable of operating at two speeds and had a maximum manufacture's air flowrate of 3000 cfm's per specification. The maximum manufacturer setting of 3000cfm's was not achievable during the testing due to flow restrictions that occurred when integrating the evaporative

cooler to the HVAC's condenser. After the installation was completed onto the condenser, the new maximum air flowrate of the evaporative cooler was found to be 1038 cfm's at the free flow condition (zero static pressure). Table 3 and Figure 10 show the new performance capabilities of the Portacool Cyclone 130 evaporative cooler operating at high speed after modifications.

Table 2: Psychrometric equipment summary.

HVAC	Goodman: GPH1424H41
Evaporative Cooler #1	Portacool Cyclone 130

Table 3: Airflow data for the Portacool Cyclone 130 evaporative cooler operating in high speed after modifications.

Fan Tested For:	Evaporator Cooler Flow: High Speed				
Fan Model Tested:	Portacool				
Test Point Number	1	2	3	4	5
Fan Static Pressure	0.001	0.099	0.452	0.762	1.122
Nozzle Differential Pressure	1.109	0.985	0.567	0.222	1.128
Nozzle Key (ID)	8	8	8	8	0
Nozzle Static Pressure	-0.002	0.096	0.450	0.764	1.132
Barometric Pressure (in Hg)	29.99	29.99	29.99	29.99	29.99
Fan Inlet Dry Bulb Temperature (F)	74.3	73.8	74.2	74.0	75.0
Fan Inlet Wet Bulb Temperature (F)	65.8	65.7	65.7	65.6	66.6
Chamber Dry Bulb Temperature (F)	73.7	73.5	73.3	73.3	75.6
Fan Motor Amperage	3.85	3.79	3.57	3.34	3.12
Fan Motor Voltage	120.0	120.0	120.0	120.0	120.0
Fan Motor Wattage	445.0	437.0	407.3	373.5	340.0
Fan Motor RPM	NA	NA	NA	NA	NA
Fan Motor #2 RPM	0	0	0	0	0
Air Density at Fan Inlet (lb/ft3)	0.07393	0.07400	0.07395	0.07397	0.07383
Air Density at Fan Outlet (lb/ft3)	0.07402	0.07407	0.07416	0.07421	0.07395
Air Density at Nozzle Inlet (lb/ft3)	0.07402	0.07407	0.07416	0.07421	0.07395
CFM at Nozzle Inlet	1038.5	978.0	740.5	461.7	0.0
CFM at Fan Outlet	1038.5	978.0	740.5	461.7	0.0
Fan Outlet Static Pressure	0.001	0.099	0.452	0.762	1.122
Fan Outlet Velocity Pressure	0.118	0.105	0.060	0.023	0.000
Fan Outlet Total Pressure	0.119	0.204	0.512	0.785	1.122
Values Corrected for Standard Air Density					
CFM at Nozzle Inlet	1038.5	978.0	740.5	461.7	0.0
Static Pressure	0.001	0.100	0.457	0.770	1.138
Velocity Pressure	0.120	0.106	0.061	0.024	0.000
Total Pressure of Fan	0.120	0.206	0.518	0.793	1.138
Motor Amperage	3.90	3.84	3.61	3.37	3.16
Motor Wattage	450.9	442.5	411.9	377.5	344.8
CFM per Watt	2.30	2.21	1.80	1.22	0.00
Note: All pressures are in inches of water unless otherwise stated.					

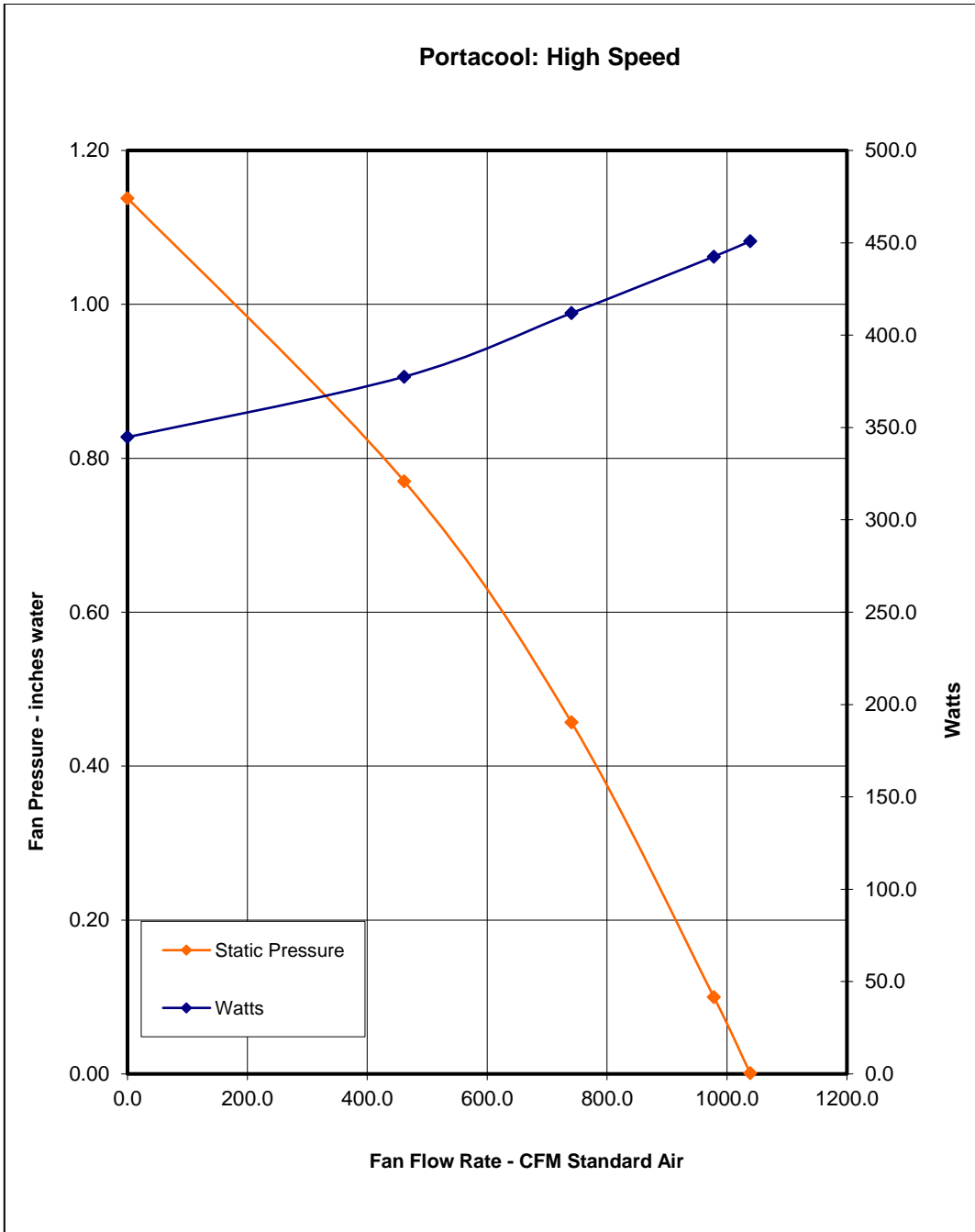


Figure 10: Flowrate to static pressure comparison for the Portacool Cyclone 130 evaporative cooler operating at high speed after modifications.

3.4 Instrumentation

A Visual Basic program in combination with a National Instrument (NI) Compact Data Acquisition (DAQ) system was used to acquire and record test data. The Visual Basic program was coded to average and record 100 input samples every 30 seconds for the duration of a test in accordance with ASHRAE 37. The DAQ system was composed of two separate chassis containing 8 modules each; a list of the NI instruments used are shown on Table 4. Chassis 1 was primarily used for receiving test data while Chassis 2 was used for outputting signals in order to control testing equipment i.e. blower VFD's, heaters, and chill water pumps.

Table 4: List of NI instruments used in the psychometric DAQ system.

Device	Module							
	1	2	3	4	5	6	7	8
Chassis 1	NI 9264	NI 9481	NI 9481	NI 9481	NI 9481	NI 9481	NI 9481	NI 9217
Chassis 2	NI 9205	NI 9211	NI 9211	NI 9211	NI 9217	NI 9211	NI 9217	NI 9205

The temperature readings for this experiment were measured using two different types of measuring devices, T-type thermocouples along with 3-wire RTD's. The two different type of sensors were needed in order to accommodate for the various needs of the experiment such as measuring wet bulb readings and being able to average temperatures over an area. Table 5 displays the specifications for the two different types of temperature sensors used.

Table 5: Temperature sensor specifications.

Sensor Type	Make/Model	Accuracy
Thermocouple Grid (X4)	Type T in Teflon Jacket	± 0.1 °C
RTD (X4)	Omega	± 0.1 °C

Wet bulb temperature measurements were taken by constructing sampling stations in accordance with ASHRAE 41.6. The wet bulb portion of the sampling station works on the premise of collecting air samples as air enters the various stages of the VCR cycle and then subjecting the air to an artificial air velocity. An inline fan was used to impart a velocity to the air in which then an RTD exposed to a wet wick was used to measure the wet bulb temperatures. An example of a sampling station can be seen on Figure 11. In total, four sampling stations located throughout the experimental setup were used to equate dry bulb and wet bulb temperatures into enthalpies for the room enthalpy testing method. The location of the four sampling stations can be seen on Figure 6.

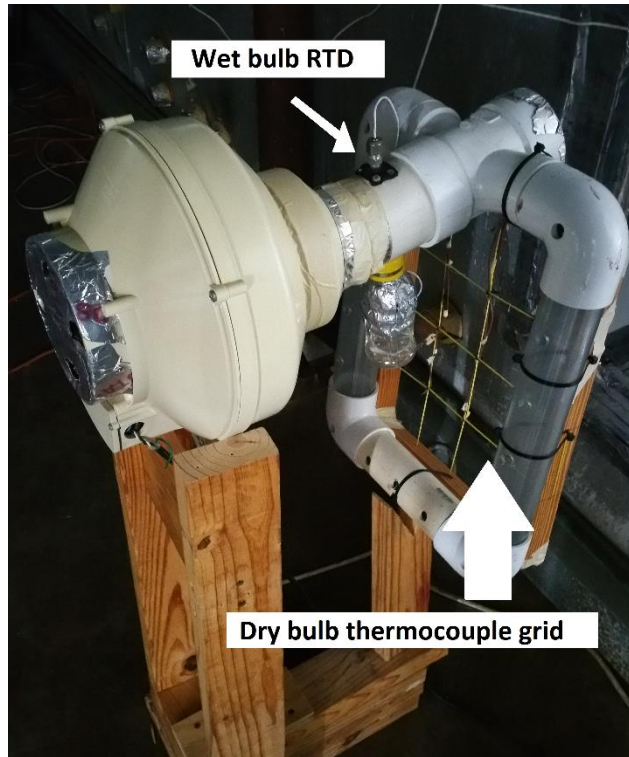


Figure 11: Dry bulb and wet bulb temperature sampling device [3]

Dry bulb temperature measurements were conducted by constructing grids in where an area of air could be sampled. Four 12-point T-wire thermocouple temperature grids were built for the 4 different stages of the VCR cycle. These thermocouple sampling grids allowed for an average temperature to be measured as air passed through the components. An example of a thermocouple grid can be seen on Figure 11.

The pressure readings for this experiment were measured using two different types of pressure sensors. Six Dwyer 616 sensors were used in order to measure static and differential pressure conditions, while a Setra 278 sensor was used in order to measure barometric pressure. The Dwyer sensors tapped into the nozzle air portions of

the experimental setup which would then be used in order to translate static and differential pressures into air flow rates. A list of the pressure sensors used in this experiment are on Table 6.

Table 6: Pressure sensor specifications.

Sensor Type	Make/Model	Accuracy
Pressure (Barometric)	Setra 278	± 0.25% F.S.
Pressure (Differential)	Dwyer 616	± 0.25% F.S.

Power and voltage to the HVAC unit was measured using transmitters. The transmitters converted the measured readings and converted them into a current signal that was read by the DAQ. The instrument specifications are listed on Table 7. Power and voltage for the evaporative cooler was measured using a power quality analyzer (Fluke 435).

Table 7: Electrical measuring sensor specifications.

Sensor Type	Make/Model	Accuracy
Power	Ohio Semitronics Inc. AGW-002E	±0.2% of Rdg., ±0.04% F.S.
Voltage	Ohio Semitronics Inc. AVT 300E2	± 0.25% F.S.

An HMD70 humidity sensor was installed into the casing surrounding the condensers inlet ports. This sensor was capable of measuring the relative humidity and dry bulb temperature that was being discharged from the evaporative cooler. The specifications for the humidity sensor are listed on Table 8.

Table 8: Humidity sensor specifications.

Sensor Type	Make/Model	Accuracy
Humidity and Temperature	HMD70	±2% RH, ± 0.1 °C

3.5 Test Procedure

The psychrometric testing was conducted in accordance ASHRAE 210/240 and ASHRAE 37. The required data collected during the course of these experiments were performed under the instruction of Table 10. Testing conditions were set and stabilized for the indoor room and outdoor rooms. Once steady state conditions were set for both rooms, data was recorded for a 30-minute equilibrium period [2]. Additionally, the evaporator cooler remained in operation during the tests performed; the only difference between the dry membrane and wet membrane test were when the evaporator cooler's water pump was turned on. The two sets of test were identified as being either a dry membrane (DM) or wet membrane test (WM). Another specification that was established during the course of these experiments was that the volumetric flowrate ratio of the evaporator to condenser was to be maintained greater than 2. For all test performed, the indoor room conditions remained the same while the outdoor conditions were varied. Four outdoor testing conditions were set and labeled A through D during the course of this experiment. The A Test and B Test were the only standardized conditions established during these experiments as seen on Table 9. A high temperature and low relative humidity condition (110°F dry bulb; 35%) was also established in order to simulate the best conditions for the performance of the evaporative cooler, Test C.

Table 9: Testing conditions for a standard indoor/outdoor room enthalpy psychrometric test [4].

Test Description	Air Entering Indoor Unit Temperature				Air Entering Outdoor Unit Temperature				Cooling Air Volume Rate
	Dry-Bulb °F °C		Wet-Bulb °F °C		Dry-Bulb °F °C		Wet-Bulb °F °C		
A Test – required (steady, wet coil)	80.0	26.7	67.0	19.4	95.0	35.0	75.0 ⁽¹⁾	23.9 ⁽¹⁾	Cooling Full-load ⁽²⁾
B Test – required (steady, wet coil)	80.0	26.7	67.0	19.4	82.0	27.8	65.0 ⁽¹⁾	18.3 ⁽¹⁾	Cooling Full-load ⁽²⁾
C Test – optional (steady, dry coil)	80.0	26.7	⁽³⁾		82.0	27.8	—		Cooling Full-load ⁽²⁾
D Test – optional (cyclic, dry coil)	80.0	26.7	⁽³⁾		82.0	27.8	—		⁽⁴⁾

Notes:

⁽¹⁾ The specified test condition only applies if the unit rejects condensate to the outdoor coil.

⁽²⁾ Defined in section 6.1.3.3.1.

⁽³⁾ The entering air must have a low enough moisture content so no condensate forms on the indoor coil. (It is recommended that an indoor wet-bulb temperature of 57.0 °F [13.9 °C] or less be used.)

⁽⁴⁾ Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the C Test.

Table 10: Required data when performing a psychometric test [2].

Item	Indoor Air Enthalpy Method	Outdoor Air Enthalpy Method
Date	X	X
Observer(s)	X	X
Barometric pressure, kPa [in. Hg]	X	X
Equipment nameplate data	X	X
Test interval times	X	X
Total power/energy input to equipment, W / Wh [W / Wh]		X
Power input, indoor side, W [W]		
Applied voltage(s), V [V]	X	X
Frequency, [Hz]	X	X
External resistance to airflow, Pa [in. H ₂ O]	X	X
fan speed(s), setting	X	X
Dry-bulb temperature of air entering equipment, indoor side, °C [°F]	X	X
Wet-bulb temperature of air entering equipment, indoor side, °C [°F]	X	X
Dry-bulb temperature of air leaving equipment, indoor side, °C [°F]	X	
Wet-bulb temperature of air leaving equipment, indoor side °C [°F]	X	
Dry-bulb temperature of air entering equipment, outdoor side, °C [°F]	X	X
Wet-bulb temperature of air entering equipment, outdoor side, °C [°F]	X	X
Dry-bulb temperature of air leaving equipment, outdoor side, °C [°F]		X
Wet-bulb temperature of air leaving equipment, outdoor side °C [°F]		X
Throat diameter of nozzle(s), mm [in.]	X	X
Velocity pressure at nozzle throat or static pressure difference across nozzle(s), Pa [in. H ₂ O]	X	X
Temperature at nozzle throat, °C [°F]	X	X
Pressure at nozzle throat, Pa [in. Hg]	X	X
Condensing pressure or temperature, kPa or °C [psig or °F]		
Evaporator pressure or temperature, kPa or °C [psig or °F]		
Temp. of refrigerant vapor entering compressor, °C [°F]		
Temp. of refrigerant vapor leaving compressor, °C [°F]		
Temperature of high side refrigerant vapor leaving reversing valve, °C [°F]		
Refrigerant or surface temperature used for leakage coefficient determination, °C [°F]		
Refrigerant-oil flow rate, kg/s [lbm/h]		
Refrigerant volume in refrigerant-oil mixture, m ³ /m ³ [ft ³ /ft ³]		
Outdoor coil water flow rate, kg/s [lbm/h]		
Temperature of outdoor water entering equipment, °C [°F]		
Temperature of outdoor water leaving equipment, °C [°F]		
Rate of condensate collection, kg/s [lbm/h]		
Refrigerant liquid temperature, indoor side, °C [°F]		Note ^b
Refrigerant liquid temperature, outdoor side, °C [°F]		Note ^b
Refrigerant vapor temperature, indoor side, °C [°F]		Note ^b
Refrigerant vapor temperature, outdoor side, °C [°F]		Note ^b
Refrigerant vapor pressure, indoor side, kPa [psig]		

a Required only during cooling capacity tests.
b Required only for line loss adjustment.

4. DATA ANALYSIS METHODOLOGY

4.1 Air Volumetric Flowrate

Air volumetric flowrates are calculated for both the indoor and outdoor rooms by using the differential pressure across the nozzle apparatuses. Additionally, the dry bulb and wet bulb temperatures are used in conjunction with a psychometric program in order to calculate the specific volume of the air-water vapor mixture [2].

$$\dot{V}_a = 1097 C_{ca} A_n \sqrt{P_V v_a} \quad \text{Eq. (6)}$$

4.2 Outdoor Room Enthalpy Method

Heat is transferred from the refrigerant onto the air-vapor mixture in the condenser portion of the VCR cycle. During this process, a phase change does not occur in the air-vapor mixtures therefore, cooling capacities can be calculated by accounting for only the sensible heat. Additionally, the power input to the HVAC unit is subtracted from the cooling capacity calculation [2].

$$q_{cooling} = \frac{60 \dot{V}_a (h_{4a} - h_{3a})}{v_a} - 3.41 W_{in} \quad \text{Eq. (7)}$$

4.3 Indoor Room Enthalpy Method

Heat that is transferred from the air-vapor mixture to the refrigerant as the air-vapor flows through the VCR's evaporator. During this heat transfer process, phase change occurs in the air-vapor mixture as water vapor condenses so that the cooling capacities must take into account both latent and sensible heat changes. Therefore, the total cooling capacity for the evaporator side of the VCR cycle is the sum of the sensible and latent cooling [2].

$$q_{sensible} = \frac{60\dot{V}_a(c_{p_{a1}}T_{aeiD} - c_{p_{a2}}T_{aeoD})}{v_a} \quad \text{Eq. (8)}$$

$$q_{latent} = (1061)(60) \frac{\dot{V}_a(W_{1a} - W_{2a})}{v_a} \quad \text{Eq. (9)}$$

4.4 Evaporative Cooler Performance

Evaporator cooler performance in terms of an efficiency was calculated for wet membrane testing conditions. The performance calculations use the outdoor wet bulb and dry bulb temperatures to determine how effective the evaporative cooler performs.

$$\epsilon_{Evap} = \frac{(T_{aceD} - T_{acoD})}{(T_{aceD} - T_{acoW})} \quad \text{Eq. (10)}$$

4.5 Data Presentation

The data that was collected over a 30-minute steady-state testing segment was averaged and presented in a summary table as seen on Table 11. This table presents the atmospheric conditions set for both the outdoor and indoor rooms in terms of dry bulb temperature, wet bulb temperature, and their resulting relative humidity. The table also provides two sets of nozzle airflows for the given condenser and evaporator unit, one being the measured standard airflow (SCFM) and the other when air density compensation was implemented (CFM). The table section labeled “Electrical Information” provides the voltage and wattage used by the HVAC unit and the evaporative cooler. The table section labeled “Evaporative Cooler” provides the dry bulb temperature and relative humidity entering and leaving the evaporative cooler. The table section labeled “Capacity Data” provides the coolant rates for the HVAC’s evaporator and condenser components. Additionally, this section provides the COP calculated from measured parameters for the given test performed.

Table 11: Summary report example- tabulations of testing conditions and results.

	Indoor Entering		Indoor Leaving		Outdoor Entering		Outdoor Leaving	
	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)
DB Temperature								
WB Temperature								
	RH: %		RH: %		RH: %		RH: %	
Indoor Condition				Outdoor Condition				
Nozzle Diameter (inch) 7+4				Nozzle Diameter (inch) 7+4				
		AVG	STDEV			AVG	STDEV	
Barometric (in. Hg)				Barometric (in. Hg)				
External SP (in. wg)				External SP (in. wg)				
Static B4 Nozzle (in. wg)				Static B4 Nozzle (in. wg)				
Nozzle Diff. (in. wg)				Nozzle Diff. (in. wg)				
Nozzle Temp. (°F)				Nozzle Temp. (°F)				
STD Airflow (SCFM)				STD Airflow (SCFM)				
Nozzle Airflow (CFM)				Nozzle Airflow (CFM)				
Electrical Information				Evaporative Cooler				
		HVAC	Evap Cooler			Entering	Exiting	
Power (W)				DB Temperature (°F)				
Voltage (V)				RH (%)				
Capacity Data								
Evap Air Sensible (Btu/hr)				Cond Air Total (Btu/hr)				
Evap Air Latent (Btu/hr)				Sensible Heat Ratio				
Evap Air Total (Btu/hr)								
COP								

5. RESULTS

5.1 HVAC Standalone: Dry Membrane Test

The results of the four conditions that were tested and evaluated while the evaporative cooler membrane remained dry are presented in Figure 12 through Figure 15. These plots display the various stages of air as it flows through the evaporator and condenser portions of the HVAC unit. For all of the dry membrane testing conditions, the evaporator cooler fan remained in operation thereby imparting its volumetric flow rate through the condenser which is important for performance comparisons to the wet membrane cases. The HVAC evaporator points, A to B, on these figures show the full saturation of air that was occurring near the evaporator outlet, and its subsequent drop in dry bulb temperature along the 100% relative humidity slope which represents latent heat. Additionally, all four of these plots show horizontal trend lines as heat was imparted from the refrigerant onto the air passing through the HVAC's condenser. The horizontal line from points C to D indicate that only sensible heat was added to the air.

The summary report of the test condition labeled C-DM can be seen in Table 12 while the airside data is presented in Figure 12. This test provided the highest dry bulb temperature (110°F) and lowest relative humidity (35%) for the outdoor room testing conditions. Additionally, the table presents the indoor and outdoor rooms conditions that were established, and the subsequent coolant rates that were obtained for the given air volumetric flowrates and atmospheric pressure conditions.

Table 12: Summary report for test sample C-DM.

	Indoor Entering		Indoor Leaving		Outdoor Entering		Outdoor Leaving	
	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)
DB Temperature	80.68	0.055	65.66	0.768	111.24	1.168	132.80	1.687
WB Temperature	72.19	0.112	63.10	0.494	85.39	0.163	89.73	0.353
	RH: 67%		RH: 87%		RH: 35%		RH: 19%	
Indoor Condition				Outdoor Condition				
Nozzle Diameter (inch) 7+4				Nozzle Diameter (inch) 7+4				
	AVG	STDEV			AVG	STDEV		
Barometric (in. Hg)	29.51	0.014			Barometric (in. Hg)	29.51	0.014	
External SP (in. wg)	0.328	0.007			External SP (in. wg)	0.062	0.124	
Static B4 Nozzle (in. wg)	0.320	0.007			Static B4 Nozzle (in. wg)	0.063	0.137	
Nozzle Diff. (in. wg)	0.098	0.002			Nozzle Diff. (in. wg)	0.131	0.006	
Nozzle Temp. (°F)	68.20	0.292			Nozzle Temp. (°F)	123.41	1.156	
STD Airflow (SCFM)	429	4			STD Airflow (SCFM)	898	21	
Nozzle Airflow (CFM)	438	4			Nozzle Airflow (CFM)	1016	25	
Electrical Information: HVAC Unit			Evaporative Cooler					
	HVAC	Evap Cooler		Entering	Exiting			
Power (W)	2942	439		DB Temperature (°F)	NA	NA		
Voltage (V)	225	120		RH (%)	NA	NA		
Capacity Data								
Evap Air Sensible (Btu/hr)	7254	Cond Air Total (Btu/hr)	12922					
Evap Air Latent (Btu/hr)	6738	Sensible Heat Ratio	0.523					
Evap Air Total (Btu/hr)	13867							
COP	1.20							

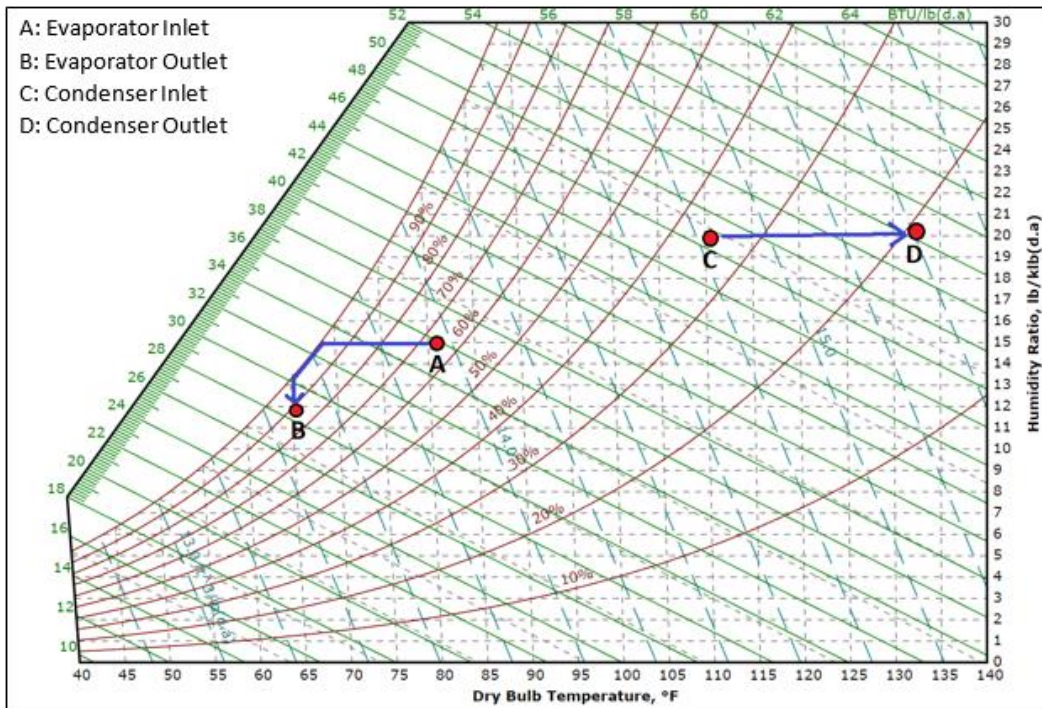


Figure 12: Test results for C-DM showing air-side psychrometric changes in the HVAC units' evaporator and condenser.

The test sample labeled A-DM was performed at a dry bulb temperature (95°F) and relative humidity (39%). A summary of the test conditions set for the indoor and outdoor room and its subsequent cooling rate results are presented in Table 13. Additionally, the psychrometric effects of air as it flows through the evaporator and condenser of the HVAC unit can be seen on Figure 13.

Table 13: Summary report for test sample A-DM.

	Indoor Entering		Indoor Leaving		Outdoor Entering		Outdoor Leaving		
	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	
DB Temperature	79.90	0.029	58.90	0.191	95.42	0.081	117.73	0.087	
WB Temperature	69.55	0.074	58.32	0.374	74.86	0.110	80.88	0.376	
	RH: 59%		RH: 96%		RH: 39%		RH: 21%		
Indoor Condition					Outdoor Condition				
Nozzle Diameter (inch) 7+4					Nozzle Diameter (inch) 7+4				
	AVG		STDEV			AVG		STDEV	
Barometric (in. Hg)	29.61		0.009		Barometric (in. Hg)	29.61		0.009	
External SP (in. wg)	0.458		0.006		External SP (in. wg)	0.018		0.059	
Static B4 Nozzle (in. wg)	0.445		0.006		Static B4 Nozzle (in. wg)	0.026		0.057	
Nozzle Diff. (in. wg)	0.114		0.002		Nozzle Diff. (in. wg)	0.148		0.005	
Nozzle Temp. (°F)	63.90		0.040		Nozzle Temp. (°F)	108.58		0.290	
STD Airflow (SCFM)	466		3		STD Airflow (SCFM)	971		15	
Nozzle Airflow (CFM)	469		3		Nozzle Airflow (CFM)	1065		16	
Electrical Information				Evaporative Cooler					
	HVAC		Evap Cooler			Entering		Exiting	
Power (W)	2478		429		DB Temperature (°F)	NA		NA	
Voltage (V)	223		120		RH (%)	NA		NA	
Capacity Data									
Evap Air Sensible (Btu/hr)	10876		Cond Air Total (Btu/hr)		16526				
Evap Air Latent (Btu/hr)	5211		Sensible Heat Ratio		0.671				
Evap Air Total (Btu/hr)	16197								
COP	1.63								

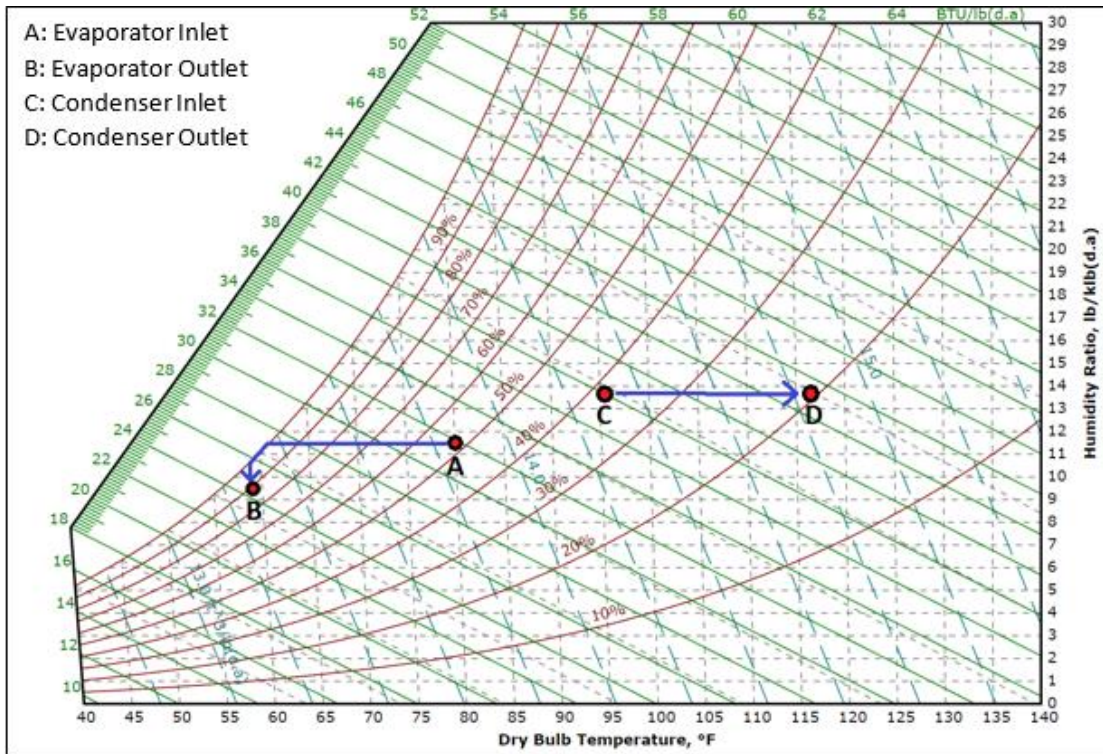


Figure 13: Test results for A-DM showing air-side psychrometric changes in the HVAC units' evaporator and condenser.

The test sample labeled D-DM was performed at a dry bulb temperature (87°F) and relative humidity (41%). A summary of the test conditions set for the indoor and outdoor room and its subsequent cooling rate results are presented in Table 14. Additionally, the psychrometric effects of air as it flows through the evaporator and condenser of the HVAC unit can be seen on Figure 14.

Table 14: Summary report for test sample D-DM.

	Indoor Entering		Indoor Leaving		Outdoor Entering		Outdoor Leaving		
	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	
DB Temperature	80.66	0.224	56.12	0.213	86.83	0.705	108.78	0.802	
WB Temperature	68.55	0.213	55.83	0.134	69.26	0.420	75.80	0.422	
	RH: 54%		RH: 98%		RH: 41%		RH: 22%		
Indoor Condition					Outdoor Condition				
Nozzle Diameter (inch) 7+4					Nozzle Diameter (inch) 7+4				
	AVG		STDEV			AVG		STDEV	
Barometric (in. Hg)	29.78		0.012		Barometric (in. Hg)	29.78		0.012	
External SP (in. wg)	0.470		0.007		External SP (in. wg)	0.033		0.003	
Static B4 Nozzle (in. wg)	0.456		0.007		Static B4 Nozzle (in. wg)	0.036		0.004	
Nozzle Diff. (in. wg)	0.100		0.002		Nozzle Diff. (in. wg)	0.128		0.003	
Nozzle Temp. (°F)	61.82		0.072		Nozzle Temp. (°F)	99.38		0.493	
STD Airflow (SCFM)	439		5		STD Airflow (SCFM)	914		9	
Nozzle Airflow (CFM)	438		5		Nozzle Airflow (CFM)	978		10	
Electrical Information: HVAC Unit				Evaporative Cooler					
	HVAC		Evap Cooler			Entering		Exiting	
Power (W)	2212		431		DB Temperature (°F)	NA		NA	
Voltage (V)	222		120		RH (%)	NA		NA	
Capacity Data									
Evap Air Sensible (Btu/hr)	11948		Cond Air Total (Btu/hr)		15865				
Evap Air Latent (Btu/hr)	5882		Sensible Heat Ratio		0.667				
Evap Air Total (Btu/hr)	17910								
COP	1.99								

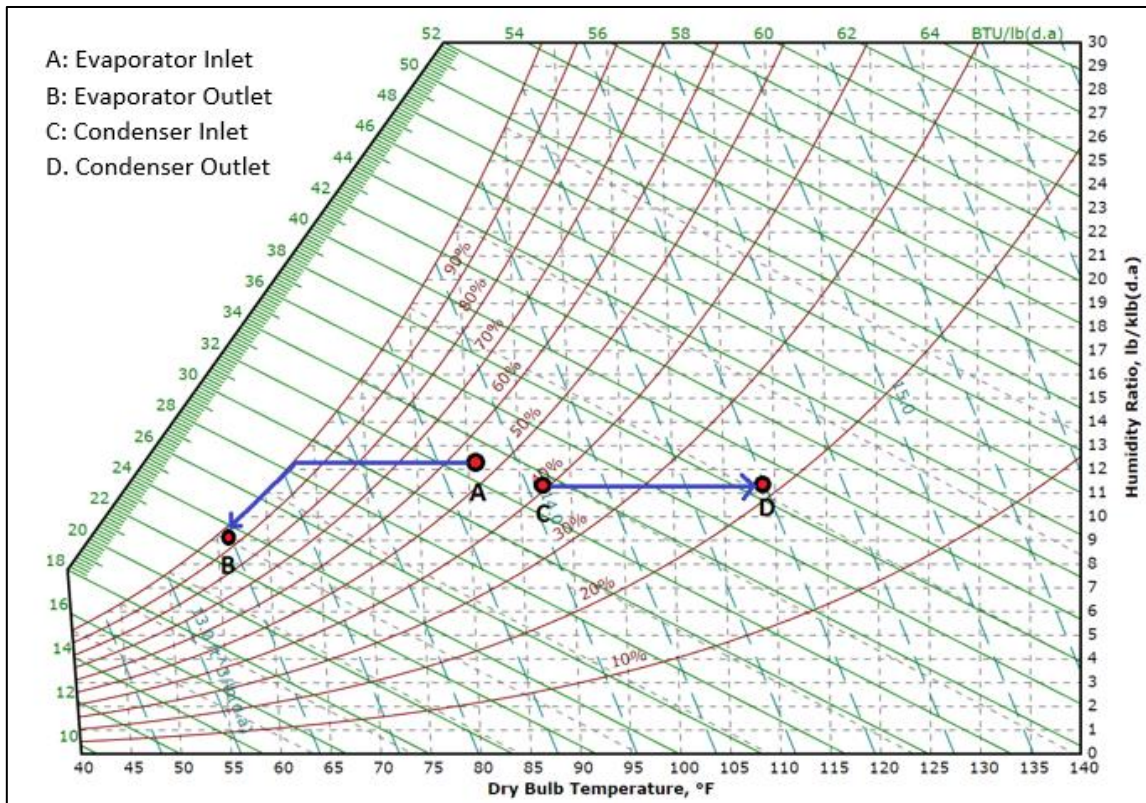


Figure 14: Test results for D-DM showing air-side psychrometric changes in the HVAC units' evaporator and condenser.

The test sample labeled B-DM was performed at the lowest dry bulb temperature (82°F) and highest relative humidity (43%). A summary of the test conditions set for the indoor and outdoor room along the subsequent cooling rate results are presented in Table 15. Additionally, the psychrometric effects of air as it flows through the evaporator and condenser of the HVAC unit can be seen in Figure 15.

Table 15: Summary report for test sample B-DM.

	Indoor Entering		Indoor Leaving		Outdoor Entering		Outdoor Leaving	
	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)
DB Temperature	79.61	0.051	56.20	0.078	81.73	0.209	105.91	0.205
WB Temperature	66.93	0.058	53.56	0.057	65.59	0.099	71.81	0.044
	RH: 52%		RH: 85%		RH: 43%		RH: 18%	
Indoor Condition				Outdoor Condition				
Nozzle Diameter (inch) 7+4				Nozzle Diameter (inch) 7+4				
	AVG	STDEV			AVG	STDEV		
Barometric (in. Hg)	29.61	0.015			Barometric (in. Hg)	29.61	0.015	
External SP (in. wg)	0.488	0.005			External SP (in. wg)	0.008	0.003	
Static B4 Nozzle (in. wg)	0.468	0.005			Static B4 Nozzle (in. wg)	0.018	0.005	
Nozzle Diff. (in. wg)	0.090	0.002			Nozzle Diff. (in. wg)	0.154	0.003	
Nozzle Temp. (°F)	61.73	0.056			Nozzle Temp. (°F)	96.59	0.168	
STD Airflow (SCFM)	414	4			STD Airflow (SCFM)	1003	10	
Nozzle Airflow (CFM)	415	4			Nozzle Airflow (CFM)	1072	10	
Electrical Information: HVAC Unit			Evaporative Cooler					
	HVAC	Evap Cooler			Entering	Exiting		
Power (W)	2110	432			DB Temperature (°F)	NA	NA	
Voltage (V)	221	120			RH (%)	NA	NA	
Capacity Data								
Evap Air Sensible (Btu/hr)	10789	Cond Air Total (Btu/hr)	15277					
Evap Air Latent (Btu/hr)	6309	Sensible Heat Ratio	0.629					
Evap Air Total (Btu/hr)	17156							
COP	1.98							

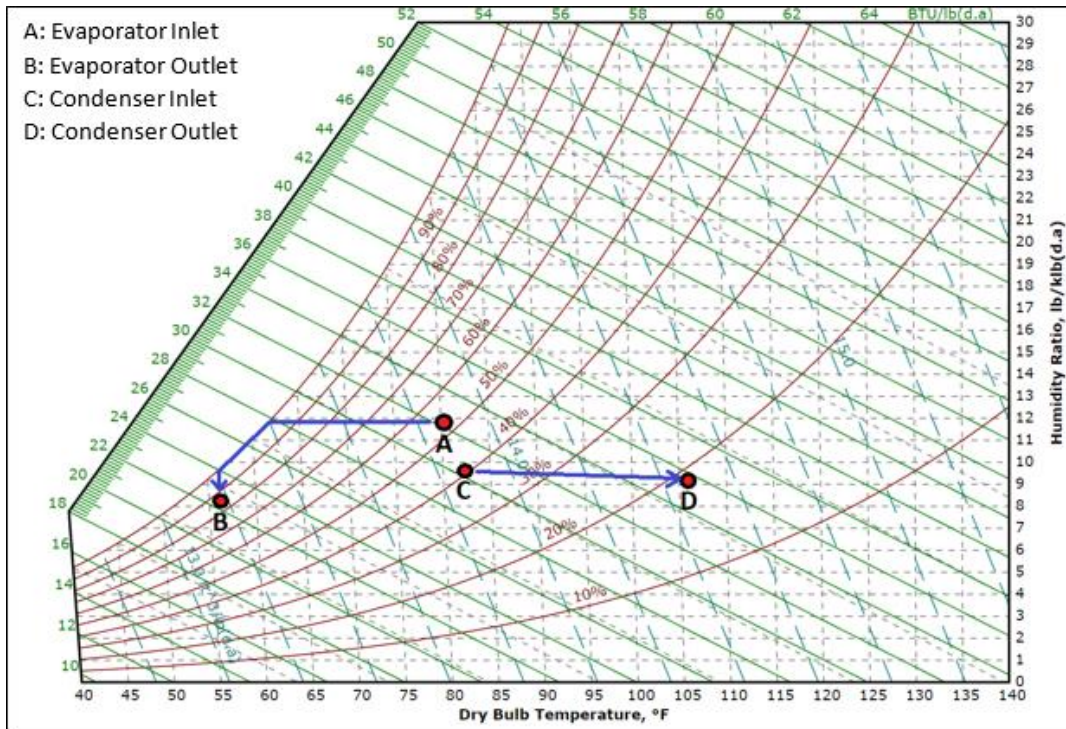


Figure 15: Test results for B-DM showing air-side psychrometric changes in the HVAC units' evaporator and condenser.

5.2 Evaporative Cooler Integration: Wet Membrane Test

The testing of the conditions ending with the acronym WM (wet membrane) differed from the DM (dry membrane) due to the water pump of the evaporator cooler being on so as to supply water to the membrane element in contact with the condenser inlet air. Furthermore, the indoor and outdoor conditions remained the same for the wet membrane and dry membrane samples in the A Test, B Test, C Test, and D Test performed. The evaporative cooler fan remained on for all testing performed for both dry and wet membranes. The points labeled C to C* on Figure 16 through Figure 19 display the constant enthalpy process that occurs when the evaporative cooler pump is turned on.

The moist air that flows through the evaporative cooler essentially follows a constant enthalpy process while also providing a cooling effect to its dry bulb temperature.

With regards to a specific test, the test labeled C-WM was performed under the same atmospheric conditions as the test labeled C-DM. As noted before, the water pump was turned on for the duration of these tests causing the membrane to become moisturized therefore saturating the air entering the evaporative cooler. A summary of the test conditions for C-WM in the indoor and outdoor rooms along with the cooling rate results are presented on Table 16. Additionally, Figure 16 displays the various stages of air as it flows through the evaporative cooler and the HVAC's evaporator and condenser.

Table 16: Summary report for test sample C-WM.

	Indoor Entering		Indoor Leaving		Outdoor Entering		Outdoor Leaving		
	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	
DB Temperature	80.44	0.139	62.48	0.387	109.48	1.378	123.41	1.116	
WB Temperature	70.99	0.334	60.66	0.464	85.05	1.217	91.14	1.318	
	RH: 63%		RH: 90%		RH: 37%		RH: 30%		
Indoor Condition					Outdoor Condition				
Nozzle Diameter (inch) 7+4					Nozzle Diameter (inch) 7+4				
	AVG		STDEV			AVG		STDEV	
Barometric (in. Hg)	29.55		0.009		Barometric (in. Hg)	29.55		0.009	
External SP (in. wg)	0.356		0.007		External SP (in. wg)	0.015		0.006	
Static B4 Nozzle (in. wg)	0.350		0.007		Static B4 Nozzle (in. wg)	0.014		0.006	
Nozzle Diff. (in. wg)	0.097		0.002		Nozzle Diff. (in. wg)	0.131		0.003	
Nozzle Temp. (°F)	66.27		0.225		Nozzle Temp. (°F)	118.38		0.972	
STD Airflow (SCFM)	427		4		STD Airflow (SCFM)	903		108	
Nozzle Airflow (CFM)	433		4		Nozzle Airflow (CFM)	1012		120	
Electrical Information: HVAC Unit				Evaporative Cooler					
	HVAC		Evap Cooler			Entering		Exiting	
Power (W)	2615		463		DB Temperature (°F)	109.48		91.4	
Voltage (V)	227		120		RH (%)	37.4		77	
Capacity Data									
Evap Air Sensible (Btu/hr)	8623		Cond Air Total (Btu/hr)		23053				
Evap Air Latent (Btu/hr)	6781		Sensible Heat Ratio		0.553				
Evap Air Total (Btu/hr)	15580								
COP	1.48								

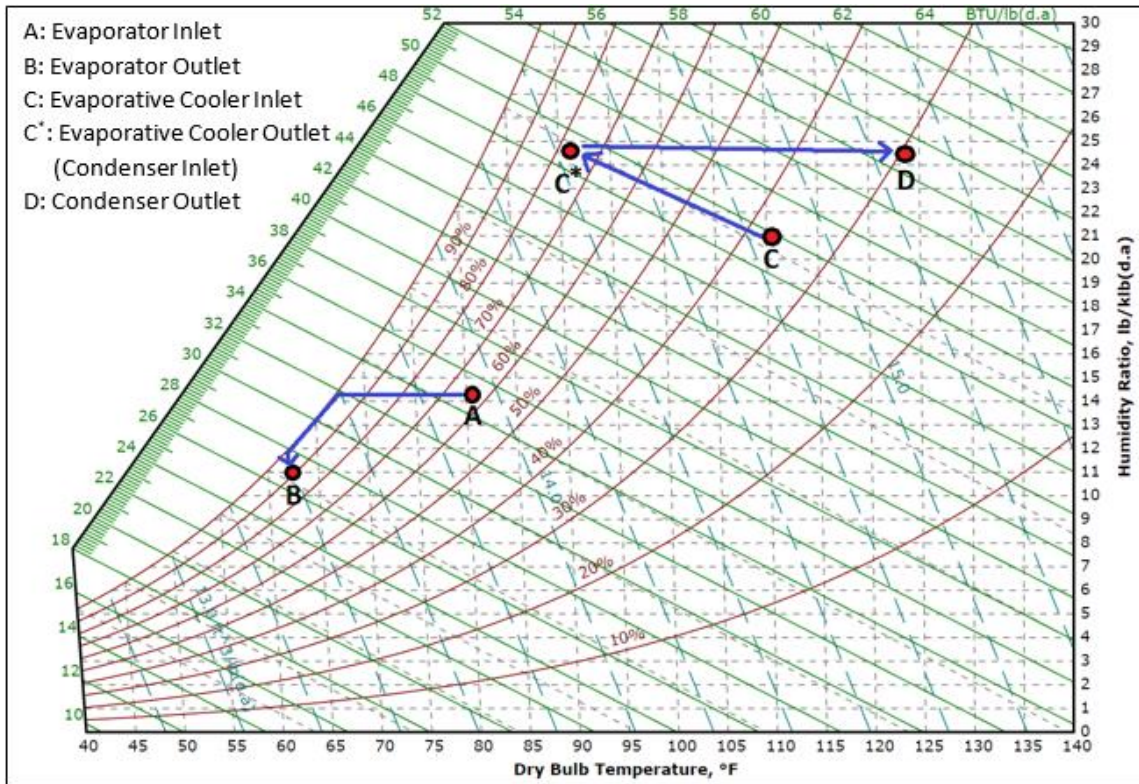


Figure 16: Test results for C-WM showing air-side psychrometric changes in the evaporative cooler and HVAC units' evaporator and condenser.

The summary report for the A-WM sample is presented in Table 17. The table provides the atmospheric conditions and resulting coolant rates that were achievable when evaporative precooling was provided to the condenser inlet air. Additionally, the psychrometric effects of air as it flows through the evaporative cooler and HVAC's evaporator and condenser can be seen on Figure 17.

Table 17: Summary report for test sample A-WM.

	Indoor Entering		Indoor Leaving		Outdoor Entering		Outdoor Leaving	
	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)
DB Temperature	79.84	0.020	57.93	0.147	94.98	0.441	111.39	0.431
WB Temperature	69.30	0.087	57.34	0.143	76.64	0.348	84.04	0.413
	RH: 58%		RH: 97%		RH: 43%		RH: 32%	
Indoor Condition				Outdoor Condition				
Nozzle Diameter (inch) 7+4				Nozzle Diameter (inch) 7+4				
	AVG	STDEV			AVG	STDEV		
Barometric (in. Hg)	29.56	0.010			Barometric (in. Hg)	29.56	0.010	
External SP (in. wg)	0.402	0.006			External SP (in. wg)	0.009	0.003	
Static B4 Nozzle (in. wg)	0.394	0.006			Static B4 Nozzle (in. wg)	0.005	0.003	
Nozzle Diff. (in. wg)	0.089	0.001			Nozzle Diff. (in. wg)	0.139	0.002	
Nozzle Temp. (°F)	62.97	0.089			Nozzle Temp. (°F)	105.44	0.406	
STD Airflow (SCFM)	412	3			STD Airflow (SCFM)	943	9	
Nozzle Airflow (CFM)	415	3			Nozzle Airflow (CFM)	1030	9	
Electrical Information: HVAC Unit			Evaporative Cooler					
	HVAC	Evap Cooler			Entering	Exiting		
Power (W)	2270	454			DB Temperature (°F)	94.98	84	
Voltage (V)	228	120			RH (%)	48.4	75	
Capacity Data								
Evap Air Sensible (Btu/hr)	10058	Cond Air Total (Btu/hr)	26151					
Evap Air Latent (Btu/hr)	6154	Sensible Heat Ratio	0.616					
Evap Air Total (Btu/hr)	16330							
COP	1.76							

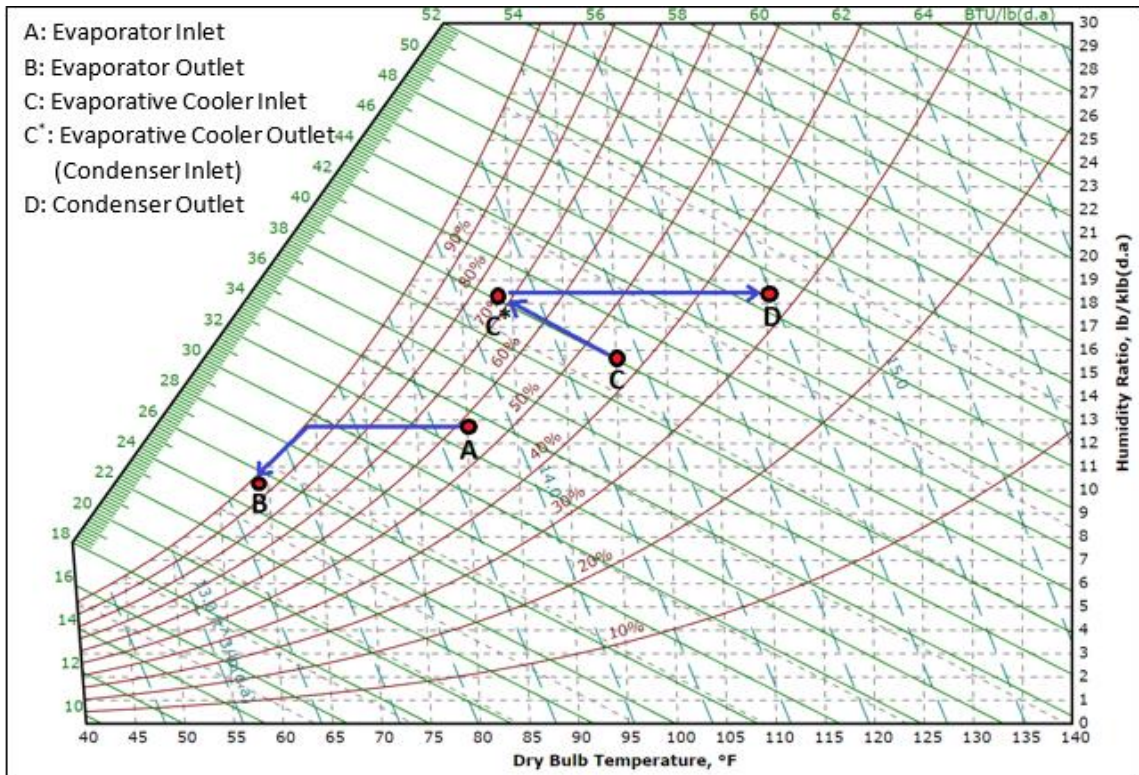


Figure 17: Test results for A-WM showing air-side psychrometric changes in the evaporative cooler and HVAC units' evaporator and condenser.

The summary report for the D-WM sample is presented in Table 18. The table provides the atmospheric conditions and the resulting coolant rates that were achievable when evaporative precooling was provided to the condenser inlet air. Additionally, the psychrometric effects of air as it flows through the evaporative cooler and HVAC's evaporator and condenser can be seen on Figure 18.

Table 18: Summary report for test sample D-WM.

	Indoor Entering		Indoor Leaving		Outdoor Entering		Outdoor Leaving	
	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)
DB Temperature	80.02	0.163	55.84	0.118	87.36	0.403	104.28	0.408
WB Temperature	68.08	0.169	54.63	0.187	69.39	0.255	76.41	0.285
	RH: 54%		RH: 93%		RH: 40%		RH: 28%	
Indoor Condition				Outdoor Condition				
Nozzle Diameter (inch) 7+4				Nozzle Diameter (inch) 7+4				
	AVG	STDEV		AVG	STDEV		AVG	STDEV
Barometric (in. Hg)		29.81	0.010	Barometric (in. Hg)		29.81	0.010	
External SP (in. wg)		0.472	0.008	External SP (in. wg)		-0.033	0.006	
Static B4 Nozzle (in. wg)		0.460	0.008	Static B4 Nozzle (in. wg)		-0.038	0.006	
Nozzle Diff. (in. wg)		0.097	0.001	Nozzle Diff. (in. wg)		0.132	0.005	
Nozzle Temp. (°F)		61.09	0.073	Nozzle Temp. (°F)		97.27	0.418	
STD Airflow (SCFM)		433	3	STD Airflow (SCFM)		928	16	
Nozzle Airflow (CFM)		430	3	Nozzle Airflow (CFM)		987	18	
Electrical Information: HVAC Unit			Evaporative Cooler					
	HVAC	Evap Cooler		Entering	Exiting			
Power (W)	2078	455		DB Temperature (°F)	87.36	77		
Voltage (V)	223	120		RH (%)	48.4	70		
Capacity Data								
Evap Air Sensible (Btu/hr)	11627	Cond Air Total (Btu/hr)	19034					
Evap Air Latent (Btu/hr)	6647	Sensible Heat Ratio	0.633					
Evap Air Total (Btu/hr)	18358							
COP	2.12							

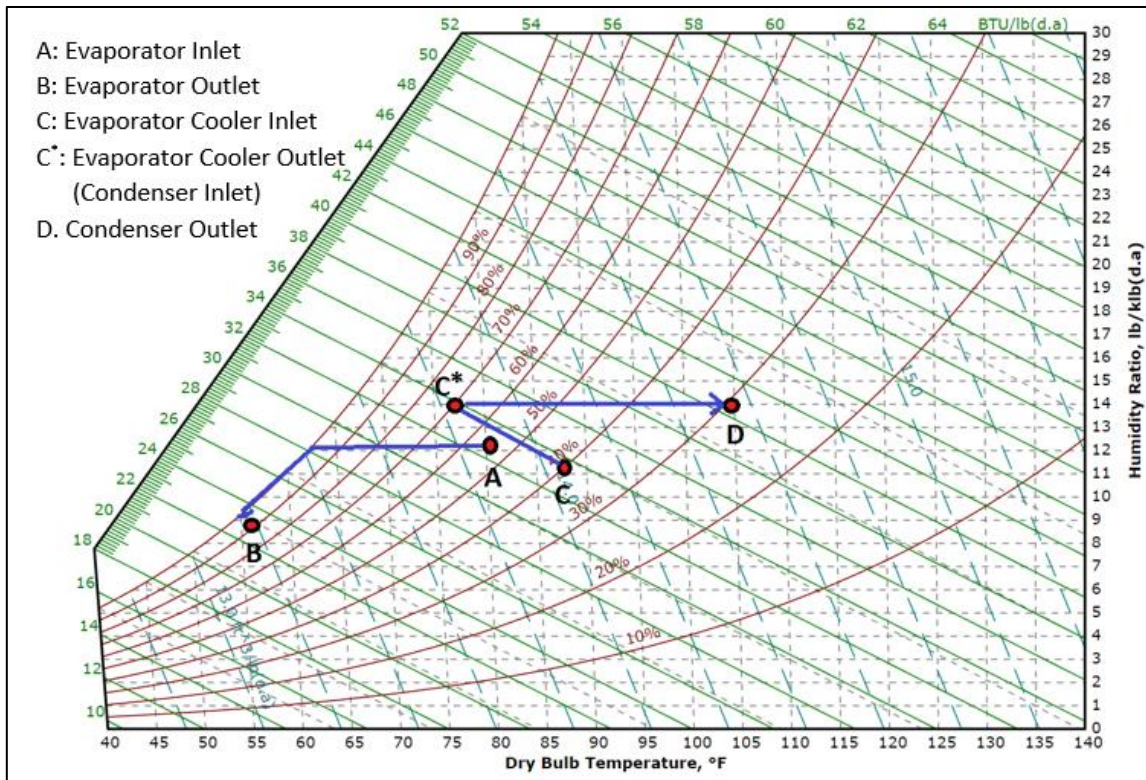


Figure 18: Test results for D-WM showing air-side psychrometric changes in the evaporative cooler and HVAC units' evaporator and condenser.

Similar to the B-DM sample, the B-WM test provided the lowest dry bulb temperature for the outdoor room. The summary report for the B-WM sample is presented in Table 19. The table provides the atmospheric conditions and resulting coolant rates that were achievable when evaporative precooling was provided to the condenser inlet air. Additionally, the psychrometric effects of air as it flowing through the evaporative cooler and HVAC's evaporator and condenser can be seen on Figure 19.

Table 19: Summary report for test sample B-WM.

	Indoor Entering		Indoor Leaving		Outdoor Entering		Outdoor Leaving		
	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	AVG (°F)	STDEV (°F)	
DB Temperature	80.00	0.046	56.04	0.047	82.25	0.157	102.27	0.107	
WB Temperature	67.91	0.040	55.94	0.278	68.02	0.129	77.88	0.142	
	RH: 54 %		RH: 99%		RH: 48%		RH: 34%		
Indoor Condition					Outdoor Condition				
Nozzle Diameter (inch) 7+4					Nozzle Diameter (inch) 7+4				
	AVG		STDEV			AVG		STDEV	
Barometric (in. Hg)	29.60		0.012		Barometric (in. Hg)	29.60		0.012	
External SP (in. wg)	0.465		0.007		External SP (in. wg)	0.063		0.004	
Static B4 Nozzle (in. wg)	0.453		0.007		Static B4 Nozzle (in. wg)	0.064		0.006	
Nozzle Diff. (in. wg)	0.113		0.002		Nozzle Diff. (in. wg)	0.126		0.003	
Nozzle Temp. (°F)	61.49		0.052		Nozzle Temp. (°F)	96.10		0.086	
STD Airflow (SCFM)	465		3		STD Airflow (SCFM)	906		10	
Nozzle Airflow (CFM)	466		3		Nozzle Airflow (CFM)	969		11	
Electrical Information: HVAC Unit				Evaporative Cooler					
	HVAC	Evap Cooler			Entering	Exiting			
Power (W)	2085	457			DB Temperature (°F)	82.25	74.9		
Voltage (V)	229	120			RH (%)	48.4	74		
Capacity Data									
Evap Air Sensible (Btu/hr)	12351		Cond Air Total (Btu/hr)	29179					
Evap Air Latent (Btu/hr)	5310		Sensible Heat Ratio	0.697					
Evap Air Total (Btu/hr)	17721								
COP	2.04								

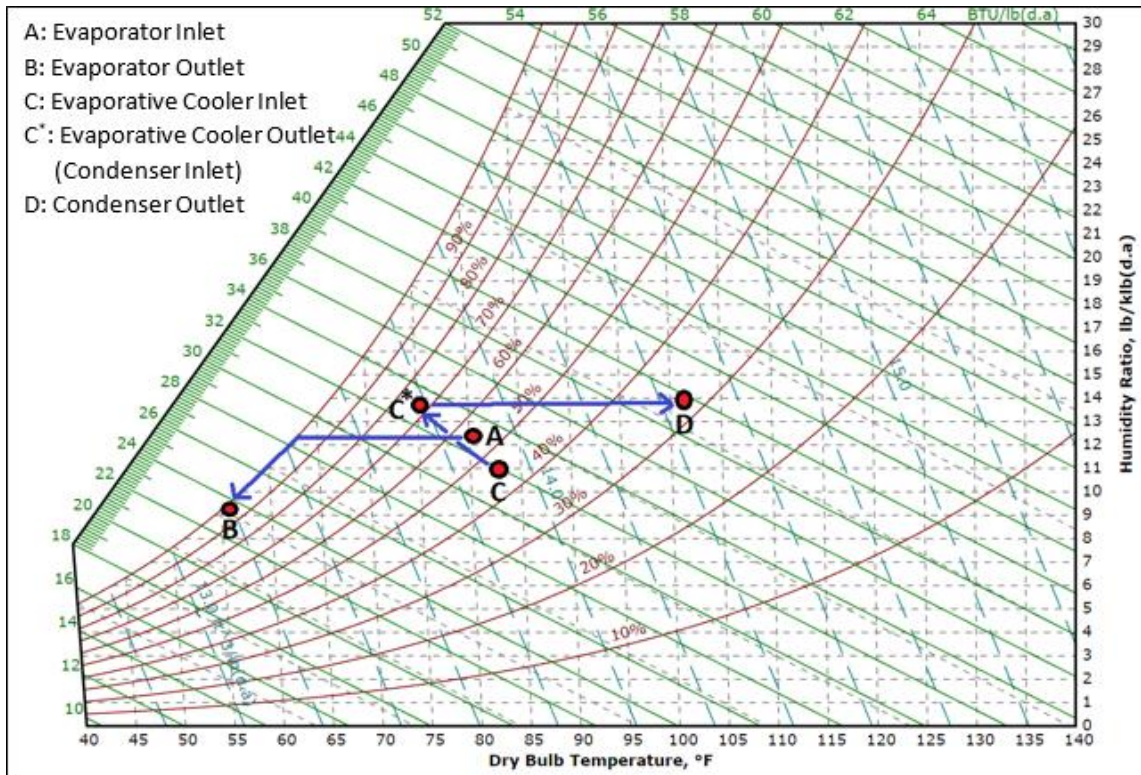


Figure 19: Test results for B-WM showing air-side psychrometric changes in the evaporative cooler and HVAC units' evaporator and condenser.

5.3 Steady State Testing Condition Example

An example of the steady state conditions that were established for each test are shown in Figure 20 through Figure 23. These examples are for B-WM test conditions and they show the independent variables over the 30-minute segment in which data was collected. The indoor and outdoor temperatures were required to be maintained within a 2°F temperature band during the data-collecting time period, but only varied by less than 1°F from the established condition.

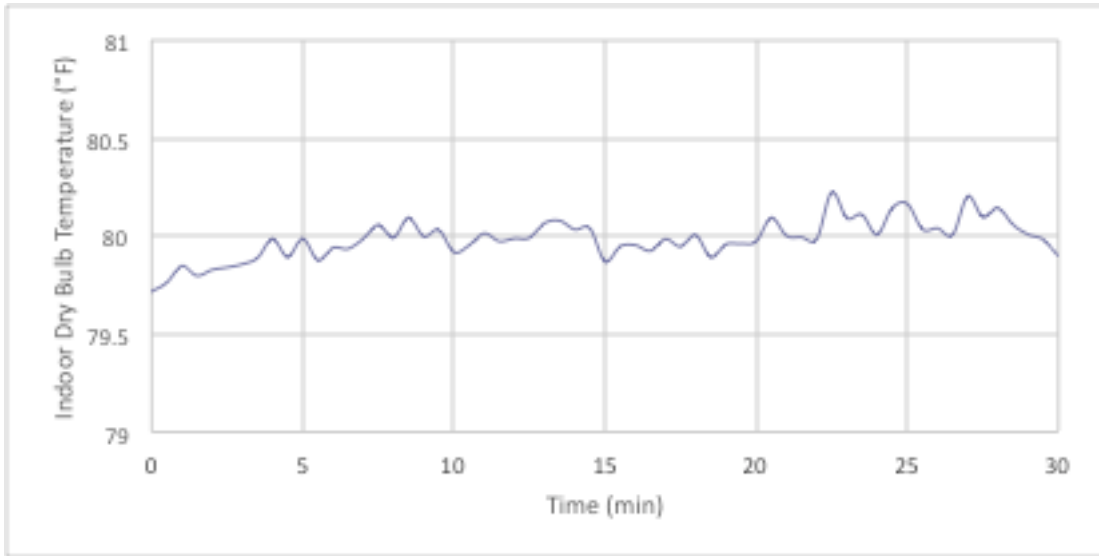


Figure 20: Indoor dry bulb temperature for the B-WM testing condition.

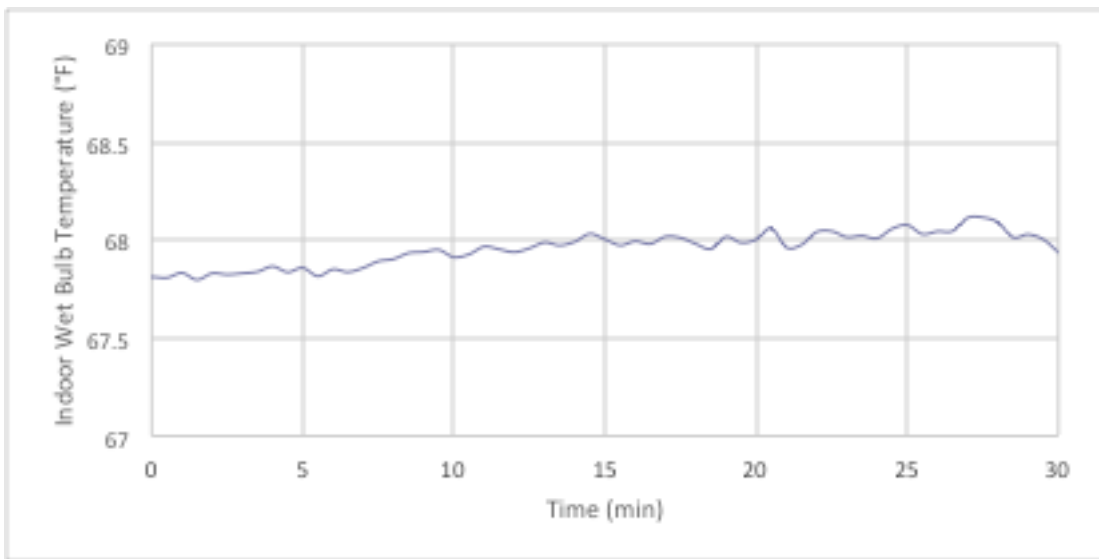


Figure 21: Indoor wet bulb temperature for the B-WM testing condition.

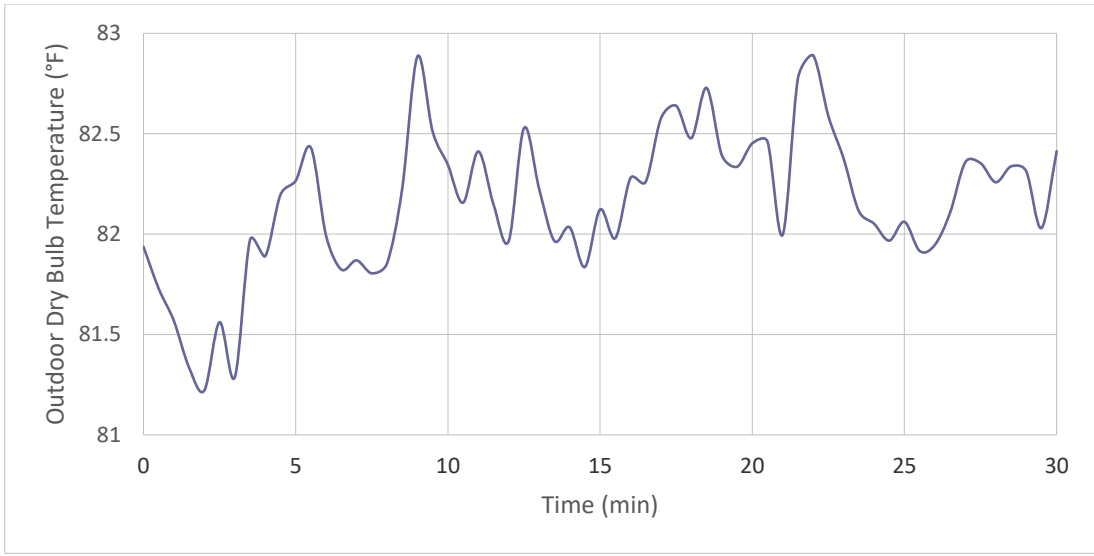


Figure 22: Outdoor dry bulb temperature for the B-WM testing condition.

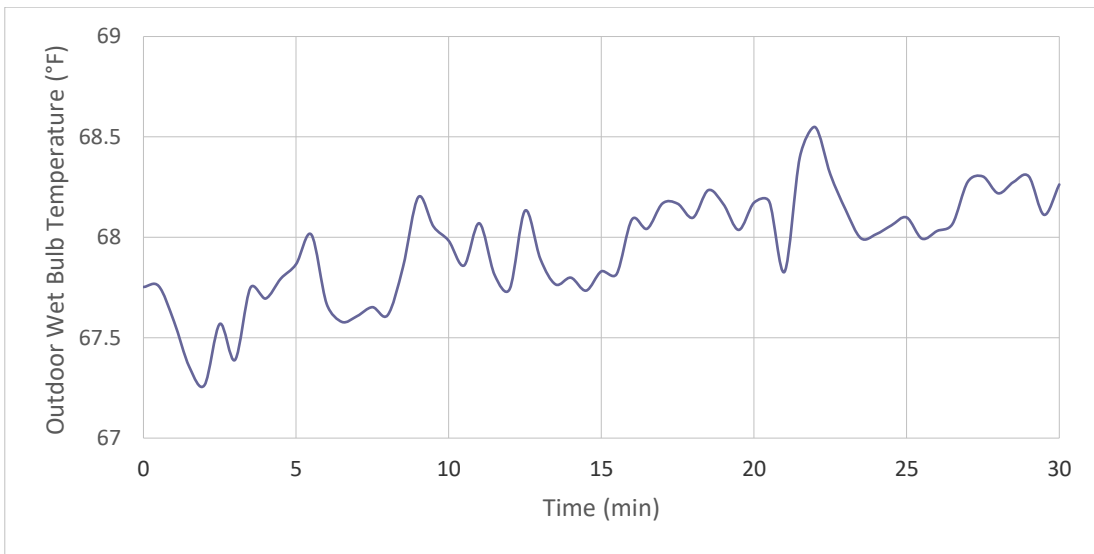


Figure 23: Outdoor wet bulb temperature for the B-WM testing condition.

5.4 Test Analysis

A summary of the coolant rates and power consumption for all of the test performed can be found in Table 20. Additionally, this table presents the COP's that were achieved for each given dry and wet membrane test, which in turns provides the base for evaluating and determining the effects of implementing the evaporative cooling to the condensers air-side inlet.

Table 20: Summary of the resulting coolant rates and power usage for the testing conditions.

	Dry Membrane				Wet Membrane			
	C	A	D	B	C	A	D	B
Evaporator Cooling Capacity (Btu/hr)	13867	16197	17910	17156	15580	16330	18358	17721
Condenser Cooling Capacity (Btu/hr)	12922	16526	15865	15277	23053	26151	19034	29179
HVAC Power (W)	2942	2478	2212	2110	2615	2270	2078	2085
Evaporative Cooler Power (W)	439	429	431	432	463	454	455	457
COP	1.20	1.63	1.99	1.98	1.48	1.76	2.12	2.04

The largest improvement in COP resulting from the use of evaporative precooling occurred at the highest dry bulb temperature and the lowest starting relative humidity, which coincided with Test C. Specifically, the trends in the table show that the higher the dry bulb temperature, then the more effective the evaporative cooler is for use with the HVAC unit. The evaporative cooler was able to saturate the condenser inlet air to approximately 70% for all four testing conditions; however, at the higher temperatures the evaporative cooler was able to affect the dry bulb temperature more significantly. At the condition C Test (110°F), the dry bulb temperature was reduced by 18°F. Table 21 and Figure 24 display the impact the evaporative cooler had on the condenser inlet air when the wet membrane condition was introduced. As C-WM had the highest dry bulb

temperature and the lowest starting relative humidity, it provided a 23% increase in COP. Table 22 and Figure 25 provide the COP increase for the various testing conditions starting from the highest dry bulb temperature to the lowest.

Table 21: Impact of dry bulb temperatures when implementing the wet membrane condition.

Reduction of Dry Bulb Temperature (°F)			
C (110°F)	A (95°F)	D (87°F)	B (82°F)
18	11	10	7

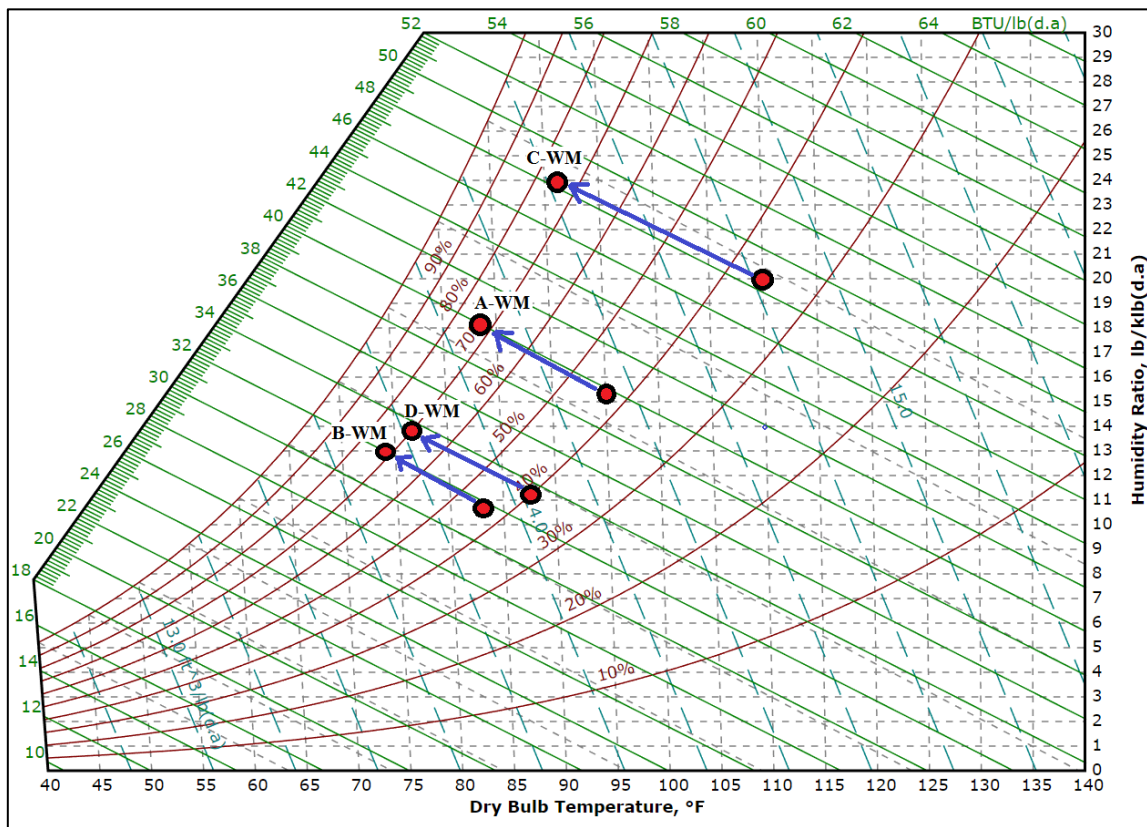


Figure 24: Evaporative cooler saturation effects on the outdoor air.

Table 22: Percent change in COP for the testing conditions.

Increase in COP (%)	
(C-WM) / (C-DM)	23.4
(A-WM) / (A-DM)	7.6
(D-WM) / (D-DM)	6.9
(B-WM) / (B-DM)	3.3

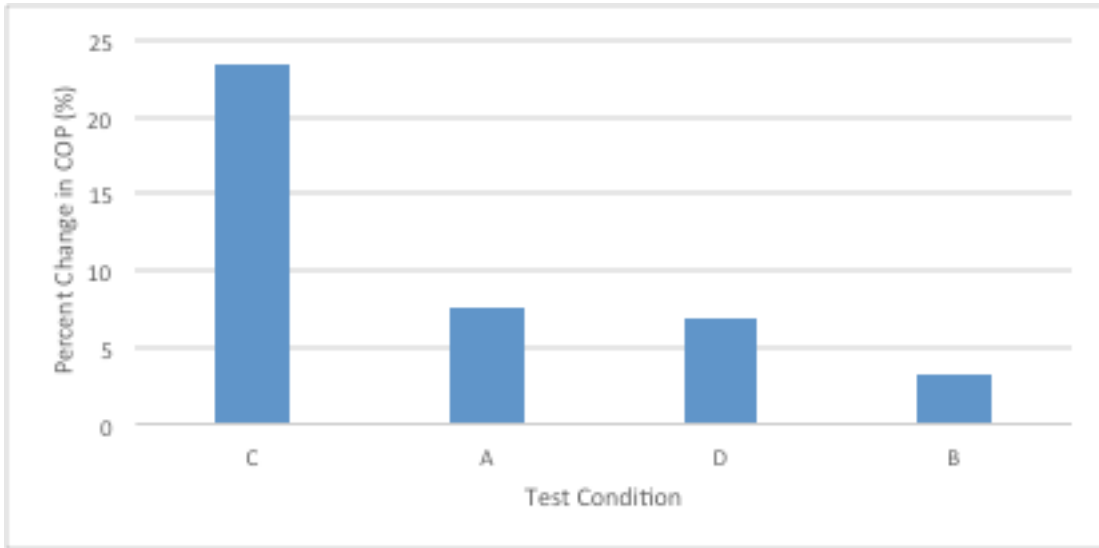


Figure 25: Improvement of COP when applying the wet membrane condition.

The performance of the evaporative cooler increased as the higher outdoor temperature conditions were established as shown in Table 23 and Figure 26. At the highest temperature setting, C Test, the evaporative cooler was able to achieve a 74% performance in saturating the outdoor air before subsequently entering the HVAC's condenser, while the increase was 52% at the lowest outdoor temperature, B Test.

Table 23: Performance of the evaporative cooler during the wet membrane test.

	C (110°F)	A (95°F)	D (87°F)	B (82°F)
Performance (%)	74	60	58	52

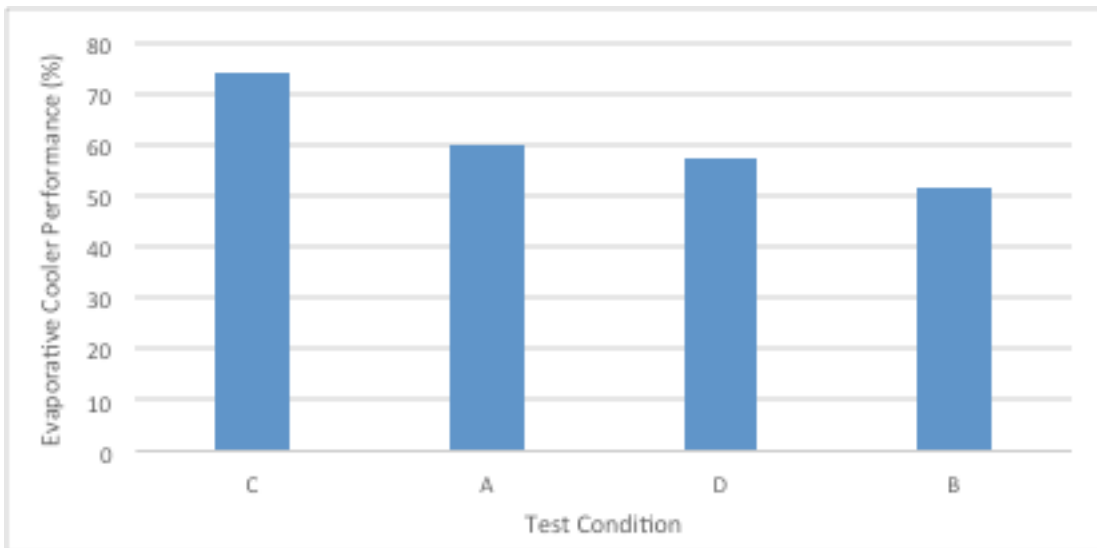


Figure 26: Performance of the evaporative cooler during the wet membrane test.

For all test conditions, the HVAC power consumption dropped with the addition of the wet membrane condition. The power consumption drop in conjunction with a higher evaporator coolant rate resulted in COP increase when comparing the wet membrane to those of the dry membrane condition. Even with the addition of approximately 25 watts from running the evaporative cooler’s water pump, a decrease in net power consumption could be achieved. A summary and comparison of the power consumption changes is presented in Table 24 and Figure 27.

Table 24: Power consumption savings when implementing a wet membrane.

	C (110°F)	A (95°F)	D (87°F)	B (82°F)
HVAC Power Change (W)	327	207	134	25
Evaporative Cooler Power Change	-24	-25	-24	-25

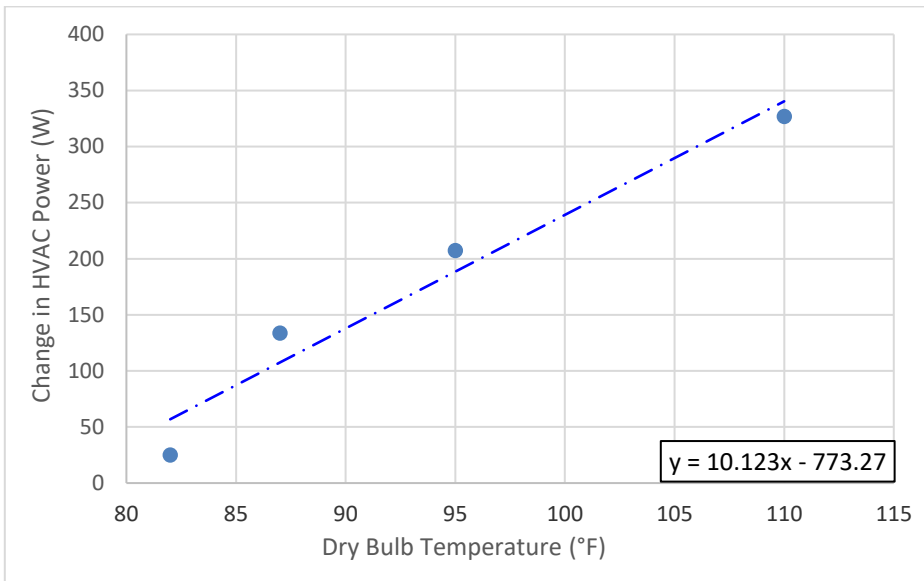


Figure 27: HVAC power savings based on utilizing the wet membrane condition for different dry bulb temperatures.

The higher the dry bulb temperature, then the higher savings that were achievable for the HVAC’s compressor power. Specifically, Table 24 shows a compressor power reduction of 327 watts, which was achieved at the highest dry bulb temperature condition when the evaporative cooler was used.

Economically, this reduction in power consumption would provide savings for the HVAC unit by simply introducing an evaporative cooler. If an assumption of 0.2 \$/kW·hr was established for a summer, the 327 watt reduction would save \$188 for a 4 month duration. Table 25 displays the summer savings that were achieved by implementing evaporative precooling to the condenser inlet air during the hottest portion of the year.

Table 25: Achievable savings when implementing evaporative precooling.

	Summer Savings (\$) [4 months]			
	C (110°F)	A (95°F)	D (87°F)	B (82°F)
Assume 0.2 \$/kwh	188	119	77	14

6. EXPERIMENTAL CONCERNS ADDRESSED IN THE PROJECT

6.1 High Temperature Complications

During attempts to collect data at high temperatures and low relative humidity, frequent malfunctions would occur with the operating equipment. The evaporative cooler and HVAC unit both ceased to operate due to overheating. Specifically, operation of equipment at temperatures above 100°F would hinder the equipment performance, therefore the time testing at these high temperatures had to be minimized in order to collect data.

6.2 Dehydrator Improvements

The permanently installed dehumidifier that was located in the outdoor room was not as effective as was needed during test when the evaporative cooler operated with a wet membrane. The evaporative cooler was adding more moisture into the air than was capable of being removed by the dehumidifier; therefore, two mobile dehumidifiers had to be used in order to control humidity.

6.3 Volumetric Flowrate Verification

During the restoration of the testing facility, volumetric flowrates throughout the evaporator and condenser portion of the HVAC unit had to be verified. It was observed that the volumetric air flowrates did not meet the capacities that were established by the Goodman manufacturer, which meant further flowrate checks and verifications were required. Specifically, the pressure gauges used to calculate the air flowrates were compared to those of a calibrated hand held pressure transducer. Pressures matched for

the installed pressure gauges and the hand held therefore verifying the readings shown through the program were correct. Additionally, a calibrated hand held hot wire anemometer was used at both the condenser and evaporator side ducting to verify the amount of flow through the ducting.

6.4 Evaporator Cooler Distribution

The integration of the evaporative cooler to the condensers airside inlet ports presented a problem due to need to achieve an equal distribution of the moisture-saturated supply air. In order to correct this problem, a casing was constructed around the condenser inlet ports. Once the casing was constructed, three circular holes were made at the top of centerline portions of the casing. Three collapsible ducting hoses were attached to the discharge of the evaporative cooler and then routed to the casing which aligned with the three condenser inlet ports.

7. FUTURE WORK

7.1 Emissions

The refrigerants used in the VCR cycle were once thought to be miracle fluids due to the ability to reach some of the highest efficiencies while still being nonflammable and nontoxic. However, as time progressed these refrigerants that were composed of chlorofluorocarbons (CFC) and hydrochlorofluorocarbons (HCFC) came under scrutiny due to environmental and ozone depletion concerns. As the type of refrigerants being used switched to being more environmentally friendly, the efficiencies of the VCR cycle suffered.

An investigation needs to be performed on the net damage caused to the environment by switching to the newer ozone friendly refrigerants. Due to the lowered operational efficiencies for the same constant demand of cooling, more emissions will occur on the power generation side in order to power HVAC units. So even though the newer refrigerants are ozone safe, the emissions released could be more detrimental to the environment.

7.2 HVAC Volumetric Flow Rate Ratios

The tests in this study were conducted for condenser-to-evaporator flow rate ratios of approximately 2. Further testing needs to be conducted in order to find the optimal condenser-to-evaporator flow ratio in order to improve overall efficiency of the HVAC units, and then evaluate if the variance in effects when the evaporative cooler is integrated to the HVAC unit.

7.3 Optimum Evaporative Cooler Conditions

The optimum conditions for the operation of an evaporative cooler are when the atmospheric conditions are low in relative humidity and high in temperature. A more effective dehumidifier needs to be purchased and installed in the testing rooms in order to perform further experiments at these conditions or even lower humidity. Currently the dehumidifier installed does not have the capabilities to remove the humidity within the outdoor testing room while also removing the humidity added by the evaporative cooler. With the implementation of a more effective dehumidifier, further experiments can be conducted in order to estimate the best conditions for which the evaporator can improve the HVAC efficiencies.

7.4 Water Conservation Methods

The evaporative unit within the HVAC system removes moisture from the air while in operation. The recycling of this moisture into the evaporator cooler could be used in order to save on water resources while the unit is in operation. In some areas that are hot and dry, the cost and demand of water can be extremely high. Consequently, methods for recycling water while not adversely affecting the net power usage of the system need to be investigated.

8. CONCLUSIONS

The major finding of this investigation is that the HVAC unit performed better with the addition of evaporative cooling from the wet membrane condition at the airside inlet of the condenser. Specifically, the wet membrane increased the COP because the precooling of the condenser inlet air increased the cooling capacity of the evaporator while also simultaneously reducing the compressor's power consumption. The largest reduction in compressor power, 327 watts, occurred at the highest dry bulb temperature condition, C Test (110°F). The higher the outdoor dry bulb temperature, then the higher the increase in COP. This increase in COP occurred primarily because of the increase in the ability to provide precooling to the condenser inlet air. Specifically, the evaporative cooler was able to saturate moist air to approximately 70% humidity for all of the tests, but was more effective at reducing the condenser inlet air at the higher dry bulb temperatures. An evaporative cooler performance efficiency of 74%, the highest of all the test, was achieved for the evaporative cooler at the C Test condition (110°F). As previously identified, the more significant the precooling applied, the larger increase in COP.

To summarize, the addition of evaporative cooling resulted in an increase in the HVAC unit's COP that ranged from 23.4% at the highest temperature of 110°F, to a low of 3.3% at the lowest temperature of 82°F.

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APPENDIX A

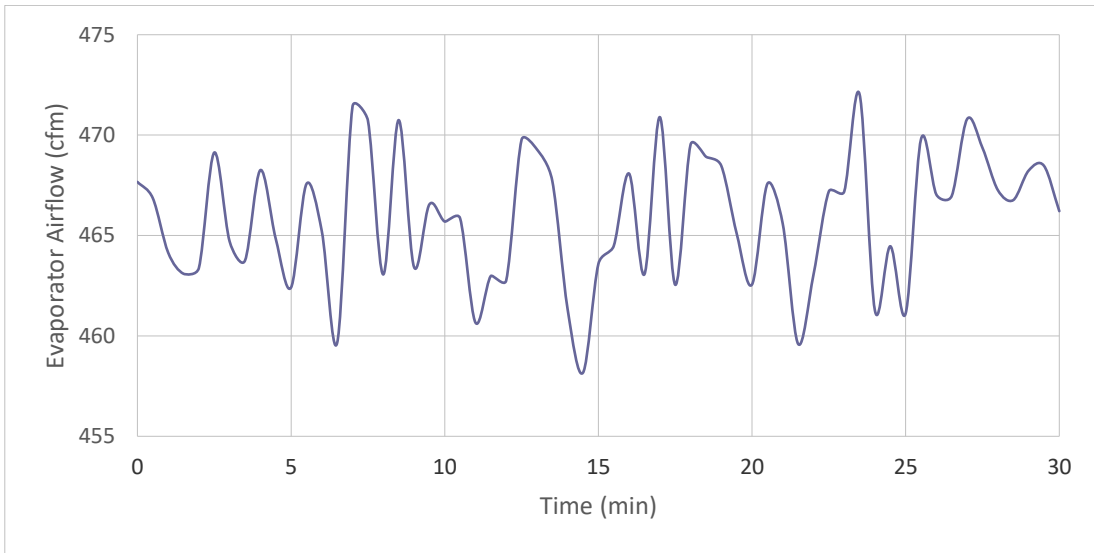


Figure 28: Evaporator airflow-dependent variable for the B-WM testing condition.

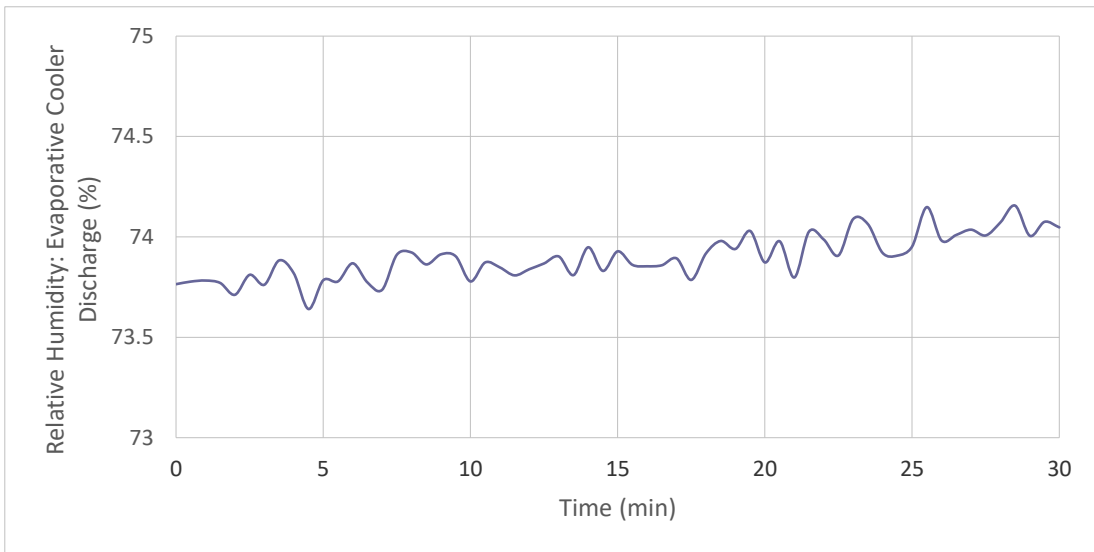


Figure 29: Relative humidity-dependent variable for the B-WM testing condition.

