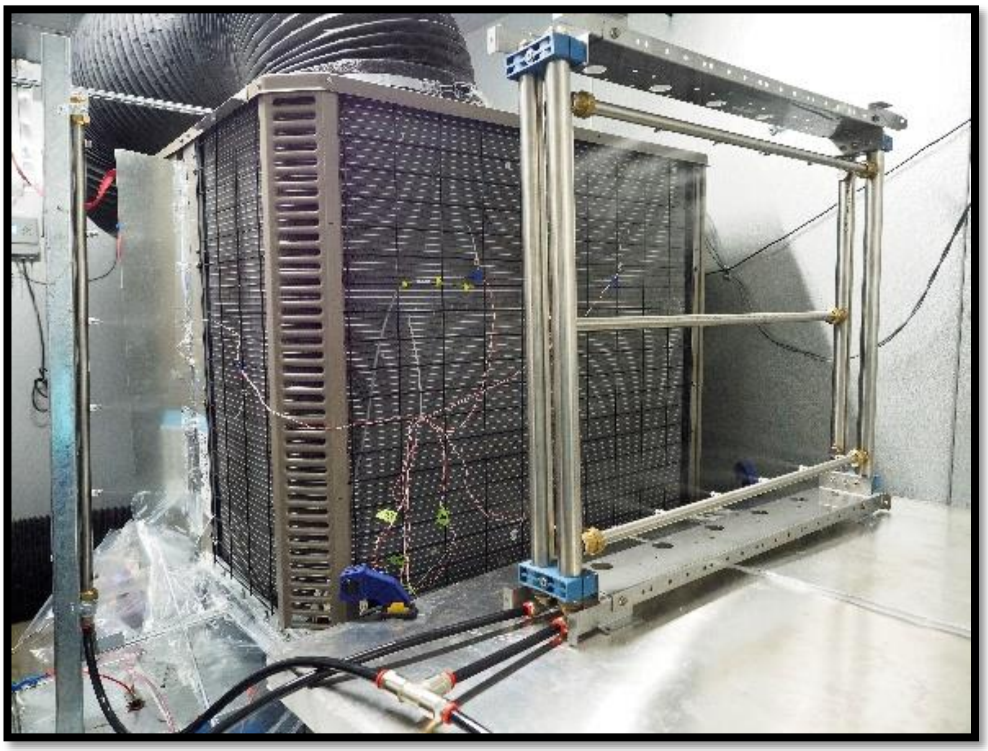
Evaporative Condenser Air Pre-Coolers

ET13SCE1020



Prepared by:

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Acknowledgments

Southern California Edison's Emerging Products (EP) group is responsible for this project. It was developed as part of Southern California Edison's Emerging Technologies Program under internal project number ET13SCE1020. Jay Madden conducted this technology evaluation with overall guidance and management from Jerine Ahmed. Contact Jay.Madden@SCE.com for more information on this project.

EXECUTIVE SUMMARY

In order to understand the energy benefits associated with evaporative pre-coolers, the American Society for Heating, Refrigeration and Air Conditioning Engineers (ASHRAE) formed a Standard Project Committee (SPC-212P) chaired by Mark Modera of the Western Cooling Efficiency Center (WCEC) to develop a Method of Test for “Determining Energy Performance and Water-Use Efficiency of Add-On Evaporative Pre-Coolers for Unitary Air Conditioning Equipment”. This report describes laboratory testing of five evaporative condenser air pre-cooler products on a packaged roof top unit and provided feedback to the ASHRAE SPC-212P.

Evaporative pre-coolers evaporate water into the air stream that cools the condenser coil of an air conditioning system. The evaporated water reduces the sensible temperature of the air stream, increasing the overall air conditioning system efficiency.

There are a large numbers of manufacturers offering evaporative pre-coolers as retrofits to existing RTUs and the methods of pre-cooling air vary. The design of the pre-cooler will impact its performance and the resulting energy and demand savings of the air conditioning system. While various field studies have been conducted, an objective laboratory test protocol is needed to quantify both the energy savings and the associated water use of the pre-cooler.

The objectives of this project as described in the scope of work were to:

1. Laboratory test five evaporative pre-cooler technologies (results to include energy and water impacts). Evaluate the impact of wind speed and direction on one technology.
2. Continue to move the Method of Test through the ASHRAE standards committee.

As part of objective 1, the decision was made by the ASHRAE committee to abort the wind speed tests after the lab determined they were too difficult to conduct and accurately reproduce between laboratories.

Before testing any evaporative condenser air pre-coolers, a set of baseline tests were obtained for a 4-ton RTU. The 4-ton RTU was chosen in order to enable testing in a laboratory scale environmental chamber, however, evaporative condenser air pre-coolers tested in this study are designed for installation on cooling equipment between 3-50 tons. The size of the air conditioning equipment is not expected to affect pre-cooler performance, therefore a smaller RTU was used to simplify laboratory construction and testing.

Each pre-cooler was added to the RTU in the lab by, or in consultation with, the manufacturer of the pre-cooler. If the pre-cooler added airflow resistance to the condenser-air stream due to an evaporative media, a dry test was conducted to determine the performance of the system with the pre-cooler installed with water off, an operating condition that may occur during cooler weather. After dry test was completed, the pre-cooler was tested with the evaporative pre-cooler installed and running. The lab tested the RTU, retrofitted with pre-coolers, at a minimum of four outdoor conditions. Then, for each pre-cooler test, the efficiency of the unit with the pre-cooler installed was compared to the baseline unit performance curve to determine the temperature at which the baseline unit had the equivalent efficiency. This temperature is equivalent to the average temperature supplied by the evaporative pre-cooler and was used to calculate the evaporative effectiveness and water-use effectiveness of the pre-cooler. The evaporative effectiveness, also known as saturation efficiency, is a measurement of how close the temperature of the

air leaving the pre-cooler is to the wet-bulb temperature of the entering air. The theoretical maximum for evaporative effectiveness is 100% (where the exiting dry bulb temperature equals the entering wet bulb temperature). The water-use effectiveness is the percent of the water consumed by the pre-cooler that is used for pre-cooling (and not lost to leaks, unevaporated droplets, overspray, etc.).

A laboratory test protocol that objectively compared evaporative pre-coolers from five manufacturers of differing designs was successfully demonstrated. The main findings of the testing were that:

1. The five products demonstrated evaporative effectiveness in the range of 20 –80% and water-use effectiveness in the 25 – 100%.
2. The three highest performing products demonstrated evaporative effectiveness in the range of 50-80% and a water-use effectiveness greater than 50%. Of these pre-coolers, two recirculated water and had the highest water-use effectiveness (>80%), however, in field applications would require additional water use for “maintenance” or “bleed” water because they are recirculation systems, which was not accounted for in the laboratory test.
3. The two lowest performing products demonstrated evaporative effectiveness consistently below 50%, with water use effectiveness generally below 50%.

Because pre-coolers designs are highly variable and the specific design and control methodology will impact performance, it is critical to complete publication of the ASHRAE test standard, so that end-users and utilities will have objective test data that can be used to compare evaporative pre-cooler products and forecast energy savings.

An analysis tool previously developed shows energy savings and demand reduction vary as a function of evaporative effectiveness and climate. As an example, an evaporative pre-cooler with 70% evaporative effectiveness installed in climate zone 10, one of the most populous in Southern California, is estimated to achieve a total energy savings of ~10% and a peak demand savings of ~20%. In the same climate zone, an evaporative effectiveness of 50% is estimated to achieve a total energy savings of ~8% and a peak demand savings of ~15%.

WCEC recommends inclusion of evaporative pre-cooling products for inclusion in rebate programs. WCEC also recommends that pre-coolers pass laboratory certification based on the test protocol described in this report, with performance requirements set by the utility. Rebate structures could be tiered to qualify more products, setting a higher rebate level for higher performance products.

There are two obvious market barriers prevent widespread adoption of this technology. The first is the lack of infrastructure (published protocol and test facilities) to test and certify pre-cooler products if a rebate program was implemented that required certification. The second, and most serious market barrier, is the staggering drought currently facing California. While evaporative cooling uses a small amount of water in comparison to other building requirements, any additional water burdens will be heavily scrutinized by end-users and regulators. We can address this issue by 1) minimizing the water used in pre-coolers, by optimizing bleed rates to use as little water as possible, 2) evaluating rainwater capture and greywater and potential non-potable water sources, and 3) quantifying the trade-off between electricity saved and water used, and the water and electricity impacts involved in water transportation and electricity generation.

ABBREVIATIONS AND ACRONYMS

AHRI	Air-conditioning, Heating, and Refrigeration Institute
ANSI	American National Standards Institute
ASHRAE	American Society for Heating, Refrigeration, and Air Conditioning Engineers
CDP	Constant Dew Points
COP	Coefficient of Performance
CWBD	Constant Wet Bulb Depression
DB	Dry Bulb
EA	Exhaust Air
EE	Evaporative Effectiveness
IA	Indoor Air
HS	Hot Side
OA	Outside Air
RA	Recirculated or Return Air
PID	Proportional, Integral, and Differential
RTD	Resistance Temperature Device
RTU	Roof Top Unit
SA	Supply Air
SCE	Southern California Edison
WB	Wet Bulb
WCEC	Western Cooling Efficiency Center
WUE	Water-use Effectiveness

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INTRODUCTION

In order to understand the energy benefits associated with evaporative pre-coolers, the American Society for Heating, Refrigeration and Air Conditioning Engineers (ASHRAE) formed a Standard Project Committee (SPC-212P) to develop a Method of Test for "Determining Energy Performance and Water-Use Efficiency of Add-On Evaporative Pre-Coolers for Unitary Air Conditioning Equipment". This report describes laboratory testing of five evaporative condenser air pre-cooler products on a packaged roof top unit and provided feedback to the ASHRAE SPC-212P.

Background

In air conditioning systems, condensing units reject heat from refrigerant directly into the outside air stream. In these systems, higher outside air temperatures result in higher energy use by the compressors. As a result, as the outdoor air temperature rises, the efficiency of the air conditioning system drops and requires more energy to provide the same amount of cooling to the conditioned space. To compound this issue, more space cooling is necessary on days when the outdoor air temperature is higher, due to the increased heat load on the building.

Evaporative cooling takes advantage of the potential of the outside air in dry climates to absorb moisture, which results in a temperature reduction of the air stream. When evaporative cooling is used for pre-cooling condenser inlet air, the condenser operates at a lower temperature than a baseline air-cooled condenser, and needs less power demand and electricity to meet the cooling demand. Evaporative condenser air pre-coolers are of special interest in dry, arid climates such as California. Arid climate zones allow for a larger amount of water to evaporate into the airstream before entering the condenser, which correlates to a higher amount of pre-cooling.

A large numbers of manufacturers offer evaporative pre-coolers as retrofits to existing RTUs. The methods of pre-cooling air vary. The design of the pre-cooler will impact its performance and the resulting energy savings of the air conditioning system. While various field studies have been conducted, an objective laboratory test protocol is needed to quantify both the energy savings and the associated water use of the pre-cooler. While the laboratory test cannot evaluate every facet of the pre-cooler technology, it is much faster and less expensive than field testing. A previous project of the WCEC conducted a similar study for three residential evaporative pre-cooler products and provided the framework for the commercial product testing conducted here [1].

Assessment Objectives

The objectives of this project as described in the scope of work were to:

1. Laboratory test five evaporative pre-cooler technologies (results to include energy and water impacts). Evaluate the impact of wind speed and direction on one technology.
2. Continue to move the Method of Test through the ASHRAE standards committee.

As part of objective 1, the decision was made by the ASHRAE committee to abort the wind speed tests after the lab determined they were too difficult to conduct and accurately reproduce between laboratories.

TECHNOLOGY DESCRIPTION

Five condenser air evaporative pre-coolers were laboratory tested on a 4-ton packaged roof top unit (RTU), using the same test protocol. The pre-cooler evaporates water to lower the dry bulb temperature of the air entering the condenser (Figure 1). Because the condenser air does not interact with the building return and supply air, no humidity is added to the building. The reduced temperature of the condenser inlet air increases the capacity of the RTU and decreases power consumption, resulting in an overall efficiency increase for the unit.

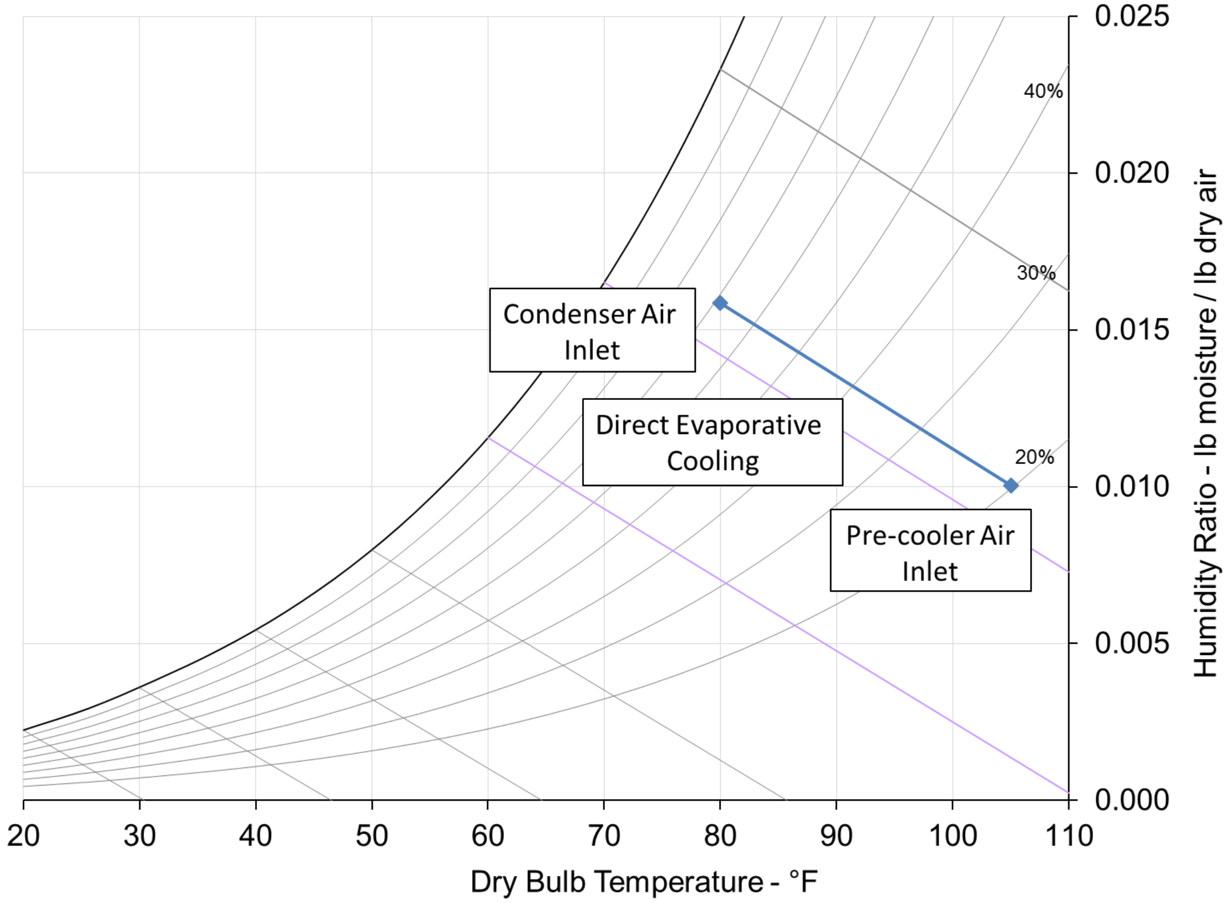

FIGURE 1: EXAMPLE OF DIRECT EVAPORATIVE COOLING OF CONDENSER INLET AIR

In general, an evaporative pre-cooler delivers water through a water distribution system and uses various methods to evaporate the delivered water prior to the condenser coil. The products vary in nozzle type and spray pressure, water flow rate, and the type of evaporative media included (Table 1). In some products, the un-evaporated water is collected and recirculated (2 of 5 tested), while in other products the excess water drains on the ground (3 of 5 tested). In some products, the water is sprayed on an evaporative media (3 of 5 tested), while in others there is no media and the water is sprayed directly on the coil (1 of 5 tested) or is evaporated prior to the coil (1 of 5 tested).

TABLE 1: PROPERTIES OF EVAPORATIVE CONDENSER AIR EVAPORATIVE PRE-COOLERS

PRE-COOLER	WATER DELIVERY	NOZZLE OPERATION	MEDIA
1	Spray nozzles @60psig, once through	Pulse operation, integrated controller	1" thick foamed polyester
2	Spray nozzles @60psig, once through	Pulse operation, integrated controller	Spray directly on coil. No media.
3	Distribution pipe fed by recirculation pump	Continuous operation, on/off control	8" deep cellulose media
4	Spray nozzles fed by recirculation pump	Continuous operation, on/off control	1" thick foamed polyester
5	Spray nozzles @220psig, once through	Continuous operation, on/off control	Evaporate without media prior to coil

TECHNICAL APPROACH/TEST METHODOLOGY

ASHRAE formed a Standard Project Committee (SPC-212P) chaired by Mark Modera of the Western Cooling Efficiency Center (WCEC) to develop an experimental Method of Test for "Determining Energy Performance and Water-Use Efficiency of Add-On Evaporative Pre-Coolers for Unitary Air Conditioning Equipment".

The protocol was designed for retrofit products for packaged roof top cooling systems up to 20 tons cooling capacity, and the protocol was tested using a 4-ton packaged rooftop unit (RTU). Five evaporative condenser pre-cooling products were tested. Because no existing facility was available with adequate dehumidification for testing, WCEC designed and built a facility optimized for testing evaporative cooling equipment as part of this research contract. The capabilities and functionality of the facility will be described as a part of this report in addition to the product test results for the five pre-coolers tested.

Environmental Chamber Design

WCEC built a permanent test facility including two environmental control chambers specifically designed to test evaporative cooling technologies at the West Village on the University of California, Davis campus in Davis, California. The primary focus of the laboratory consists of controlling two conditioned chambers (Table 2). The larger chamber is designed to produce outdoor air conditions and the smaller chamber is designed to produce indoor air conditions.

The humidity and temperature of the air into the outdoor chamber can be fully controlled to any temperature between 60 and 110°F and any humidity ratio between 0.005 and 0.013 lb_w/lb_a, as long as outdoor ambient conditions are within the blue region illustrated in Figure 2. Figure 2 illustrates the numbers of hours per year these weather conditions are expected in Davis, California, where the laboratory is located. The chamber is operational outside of the listed weather conditions but will have some limitations on capabilities (either on humidity, temperature, or airflow).

TABLE 2: DESIGN CONDITIONS FOR ENVIRONMENTAL CHAMBERS

	OUTDOOR CHAMBER	INDOOR CHAMBER
Size	10.5' Wide x 15' Long x 8' Tall	7' wide x 10' Long x 8' Tall
Climate Condition	Dry Bulb – 60-110°F Humidity – 0.005-0.013 lb_w/lb_a	Dry Bulb 70-85°F Humidity – 0.008-0.011 lb_w/lb_a
Airflow	240-8,000 CFM	240-3,000 CFM

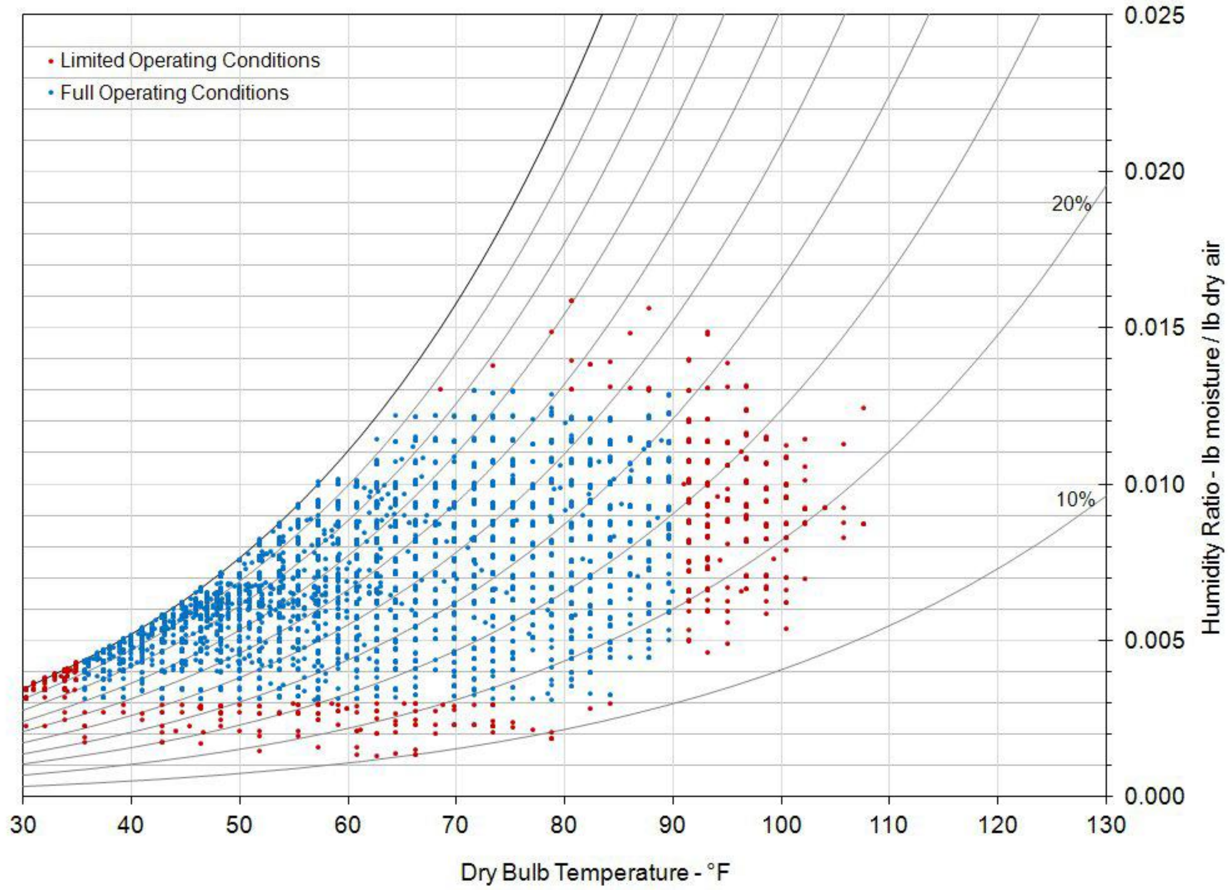

FIGURE 2: TYPICAL METROLOGICAL YEAR DATA - DAVIS CA

The outdoor air chamber has fully conditioned air capacities for flow rates between 240 and 5000cfm. This ensures the capacity to easily test 3 to 5 ton units under the assumption that condensers tend to pull approximately 800cfm of air per ton of cooling. The air cannot be conditioned completely at an airflow rate higher than 5000cfm, but for the purposes of studying wind effects the chamber can supply air at rates up to 8000cfm with limited conditioning capacity.

Temperature and humidity control of the outdoor air chamber is accomplished by two parallel conditioning paths, through which the distribution of airflow is controlled by two computer controlled dampers. One path contains a heating coil supplied by hot water and an evaporative media humidifier, the other path contains a chilled water coil and a gas-fired desiccant dehumidifier. The hot and chilled water coils have computer controlled valves to modulate water flow while the humidifier and dehumidifier have on/off control. Modulating the dampers and valve positions allows for precise control of the chamber humidity. The final temperature of the air is then controlled by additional hot and chilled water coils prior to the chamber inlet.

The indoor air chamber capabilities are limited to heating and humidification. It is designed to re-heat and re-humidify the supply air leaving an evaporator coil. The design flow rate for the indoor air chamber is between 240 and 3000cfm.

The heated and chilled water for the laboratory is supplied by a boiler and chiller located on the roof. They both supply a holding tank of water for use in the load loops that run through the lab. A mixing valve is used to control the temperature supplied by these loops to the coils located in the lab, mixing the return water from the load loop with supply water from the boiler/chiller loop storage tanks.

The outdoor air conditioning loop is designed to run in either an open or recirculation loop, where either Outside Air (OA) or Recirculated Air (RA) is used as a starting condition for the process air. As shown in both the included diagram (Figure 3) and psychometric chart (Figure 4), the OA or RA (OA1) is split into two paths which are heated (OA2a) and humidified (OA3a) and cooled (OA2b) and dried (OA3b). It should be noted that the dehumidifier heats as it dehumidifies, so that the net result of the dehumidification path is hot, dry air. Two modulating dampers determine the percentage of air that travels the heat/dehumidification path and cooling/humidification path. After the two paths are recombined (OA4), they are either chilled or heated to reach a desired set point condition (OA5). A nozzle box is used to monitor the airflow through the loop just prior to the chamber entrance, and blower is used to make up for the losses of this conditioning and measurement. The conditioned and measured air enters the test chamber, where it passes through the test unit. After exiting the test unit, it is either vented to the outside air as Exhaust Air (EA) or recirculated to re-enter the loop as RA.

The control process for the outdoor air chamber is as follows:

1. The differential pressure across the condenser of test unit is measured in free-air (no ducting attached). This pressure is used a proxy measurement for airflow.
2. The exhaust air from the test unit is ducted to the chamber exit. The test unit is turned on.
3. The damper in the humidification path is closed and all air is passed through the drier.
4. The speed on the chamber blower is increased until the differential pressure across the condenser of the test unit matches the measurement in Step 1. The blower speed is fixed for the remainder of the test.
5. The differential pressure across the combined humidity control paths is measured.
6. A control loop (Proportional, Integral, and Differential (PID)) closes the damper for the drying path until the target dew point is reached.
7. A control loop (PID) opens the damper for the humidifying path to maintain the differential pressure measured in Step 5. This maintains the conditioning system at a fixed resistance.
8. A control loop (PID) adjusts the hot and/or chilled water flow until the target dry bulb temperature is reached.

The indoor air conditioning loop can only run as a recirculation loop and lacks the capacity to either chill or dehumidify the air. As shown in Figure 3 and psychometric chart (Figure 5), the air enters this loop as supply from the test unit (IA1) and is then heated (IA2), humidified (IA3), and reheated (IA4) on its path back to the nozzle box and eventually cold side chamber (IA5). A blower is used to make up for losses of this conditioning and measurement. The air from this chamber eventually re-enters the test unit as return air to be cooled and dehumidified, by the test unit, before returning to the conditioning load loop.

The control process for the indoor air chamber is as follows:

1. The supply and return to the test unit are ducted and the test unit is turned on.
2. The bypass damper for the humidifier is closed and all air is passed through the humidifier.
3. The speed on the chamber blower is set by following the procedure described in section 6.1.3.3.1.1 of ANSI/AHRI Standard 210/240-2008.
4. The differential pressure across the humidification path is measured.
5. A control loop (PID) closes the damper for the humidifier until the target dew point is reached.
6. A control loop (PID) opens the damper for the bypass to maintain the differential pressure measured in Step 4. This maintains the conditioning system at a fixed resistance.
7. A control loop (PID) adjusts the hot water flow until the target dry bulb temperature is reached.

In both chambers, the inlet and outlet temperature and dew point of the chambers are measured with resistance temperature devices (RTDs) and chilled mirror hygrometers. Damper actuators and valves are manufactured by Belimo and are fully controllable over a 2-10V range. Data acquisition inputs, PID algorithms, and control outputs are accomplished with National Instruments CompactDAQ hardware and custom LabVIEW software. Detailed tables of the chamber equipment and instrumentation are available in Appendix 1.

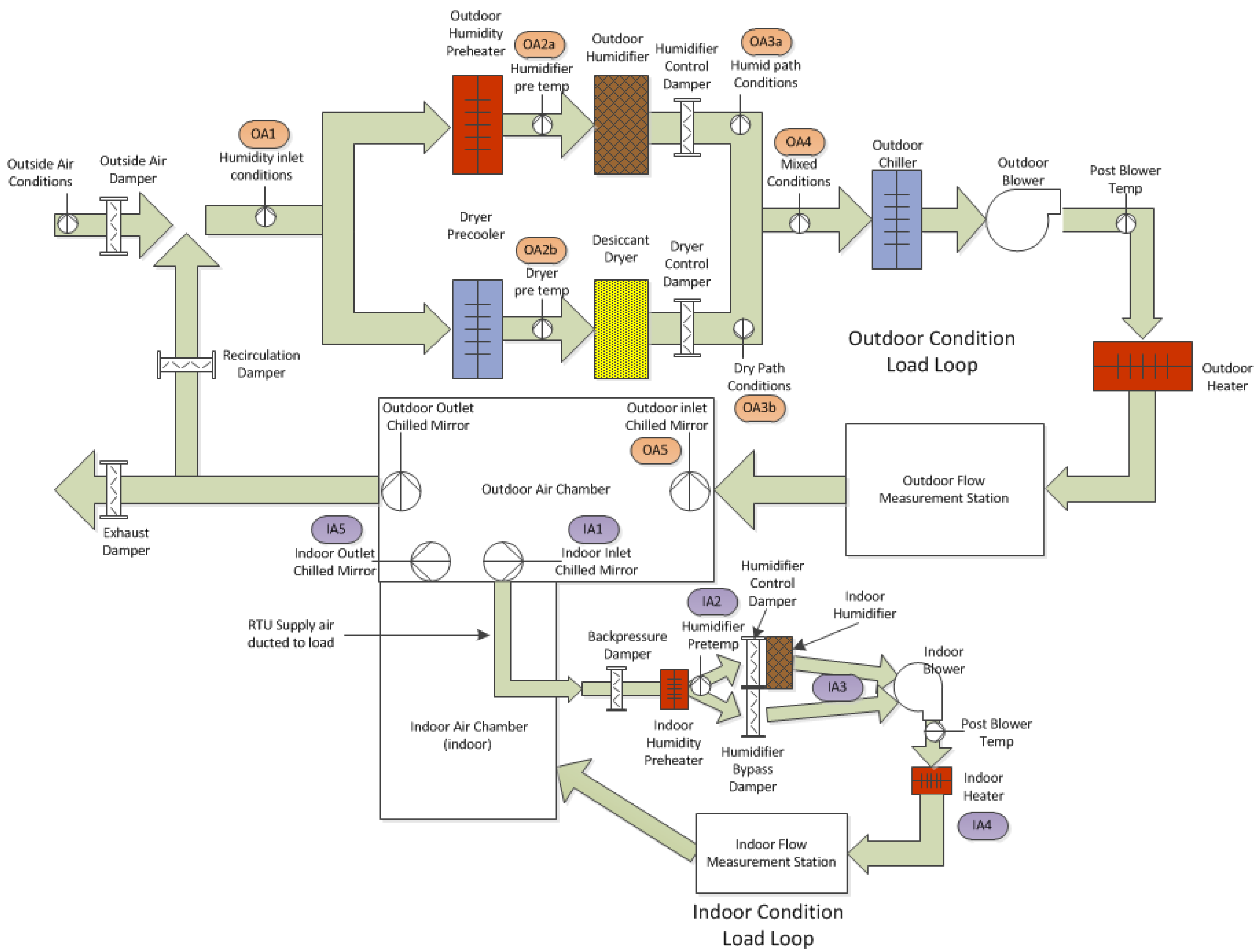

FIGURE 3: SCHEMATIC OF TEST CHAMBERS AND BOTH INDOOR AND OUTDOOR CONDITIONING LOOPS

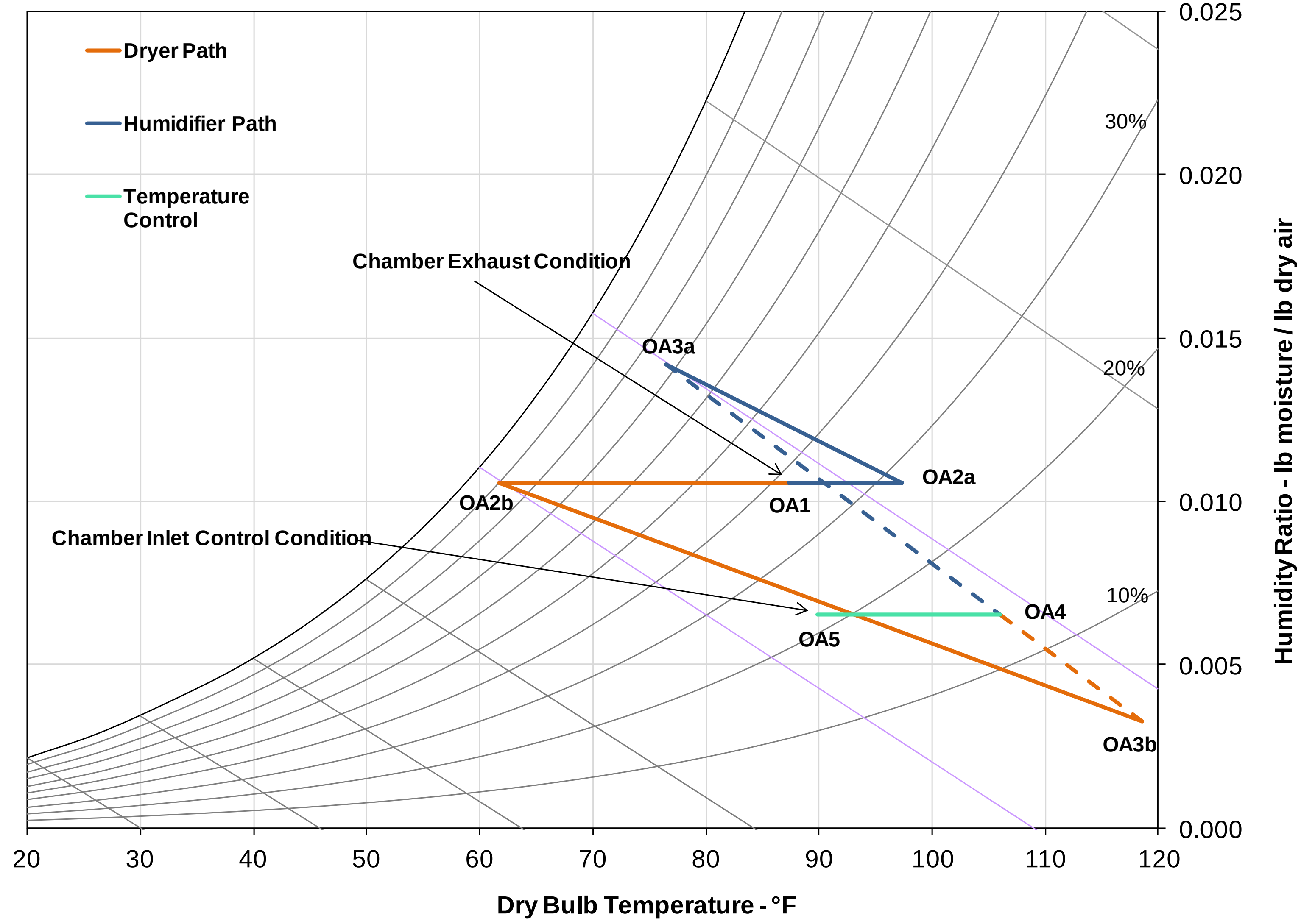

FIGURE 4: EXAMPLE AIR CONDITIONS FOR OUTDOOR AIR CONDITIONING LOOP

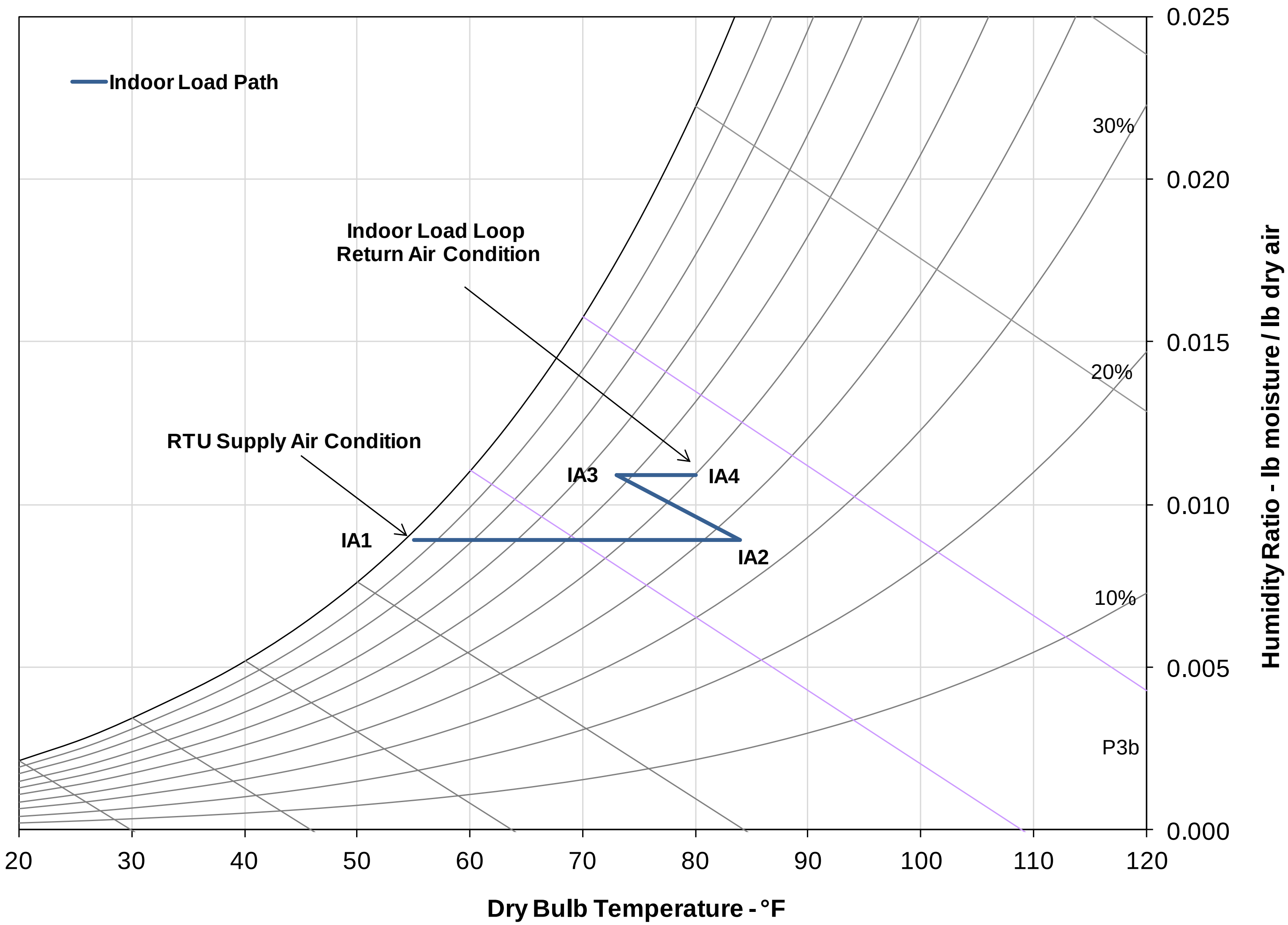

FIGURE 5: EXAMPLE AIR CONDITIONS FOR INDOOR AIR CONDITIONING LOOP

Evaluation of Baseline Technology

Before testing any evaporative condenser air pre-coolers, a set of baseline tests were obtained for a 4-ton York RTU (model #D6NZ048N06525NX). The test team ran baseline tests to record system efficiency and performance for a number of outdoor air dry bulb test points and an indoor air condition of 80°F/67°F dry bulb/wet bulb (DB/WB) (Table 3).

TABLE 3: TEST POINTS FOR COOLING EQUIPMENT WITH NO PRE-COOLER INSTALLED (BASELINE)

TEST	AMBIENT TEMPERATURES (°F DB)	INDOOR LOAD TEMPERATURES (°F DB/°F WB)
B1	115	80/67
B2	105	80/67
B3	95	80/67
B4	90	80/67
B5	82	80/67
B6	75	80/67
B7	73	80/67
B8	64	80/67

Test Plan

Each pre-cooler was installed in the WCEC lab as a retrofit to the RTU by, or in consultation with, the manufacturer of the pre-cooler. If the evaporative pre-cooler added resistance to the condensing coil, the pre-cooler was testing in the “dry” condition with no water running. This measured the penalty of running the RTU with the pre-cooler on and no water running (which may occur in applications when the outdoor air temperature is low). The test point for the dry condition is listed Table 4.

After dry test was completed, the pre-cooler was tested with the evaporative pre-cooler installed and running. WCEC tested the RTU, retrofitted with pre-coolers, at 12 ambient conditions, shown in Table 5. The conditions can be grouped into three categories: Constant Dew Points (CDP) conditions for tests W1, W2, W3, W4, and W5; Constant Wet Bulb Depression (CWBD) conditions for tests W3, W6, W7, W8, and W9; and other comparisons conditions for tests W10, W11 and W12. A few pre-coolers were tested at all of these conditions, but after analysis and some discussion it was decided to limit the scope to just the first four CDP test for the testing of subsequent pre-coolers.

TABLE 4: COOLING EQUIPMENT WITH DRY EVAPORATIVE PRE-COOLER INSTALLED (DRY COOLER)

TEST	AMBIENT TEMPERATURES (°F DB)	INDOOR LOAD TEMPERATURES (°F DB/°F WB)
D1	75	80/67

TABLE 5: COOLING EQUIPMENT WITH WET EVAPORATIVE PRE-COOLER INSTALLED (WET COOLER)

TEST	AMBIENT TEMPERATURES (°F DB/°F WB)	INDOOR LOAD TEMPERATURES (°F DB/°F WB)	TEST GROUP
W1	115/75.7	80/67	CDP
W2	105/73.0	80/67	CDP
W3	95/70.1	80/67	CDP/CWBD
W4	85/67.1	80/67	CDP
W5	75/63.8	80/67	CDP
W6	105/80.1	80/67	CWBD
W7	100/75.1	80/67	CWBD
W8	90/65.1	80/67	CWBD
W9	85/60.1	80/67	CWBD
W10	95/75	80/67	Other
W11	90/64	80/67	Other
W12	82/73	80/67	Other

Instrumentation Plan

The 4 ton RTU with refrigerant 410A (R-410A) was placed inside the conditioned chamber and used for all pre-cooler tests (Figure 6). The measurements are color coded; light blue sensors measure differential pressure, orange sensors measure temperature, green sensors measure pressure, grey sensors measure air properties, purple sensors measure power, and the red sensor measures condensate generation (Figure 7).

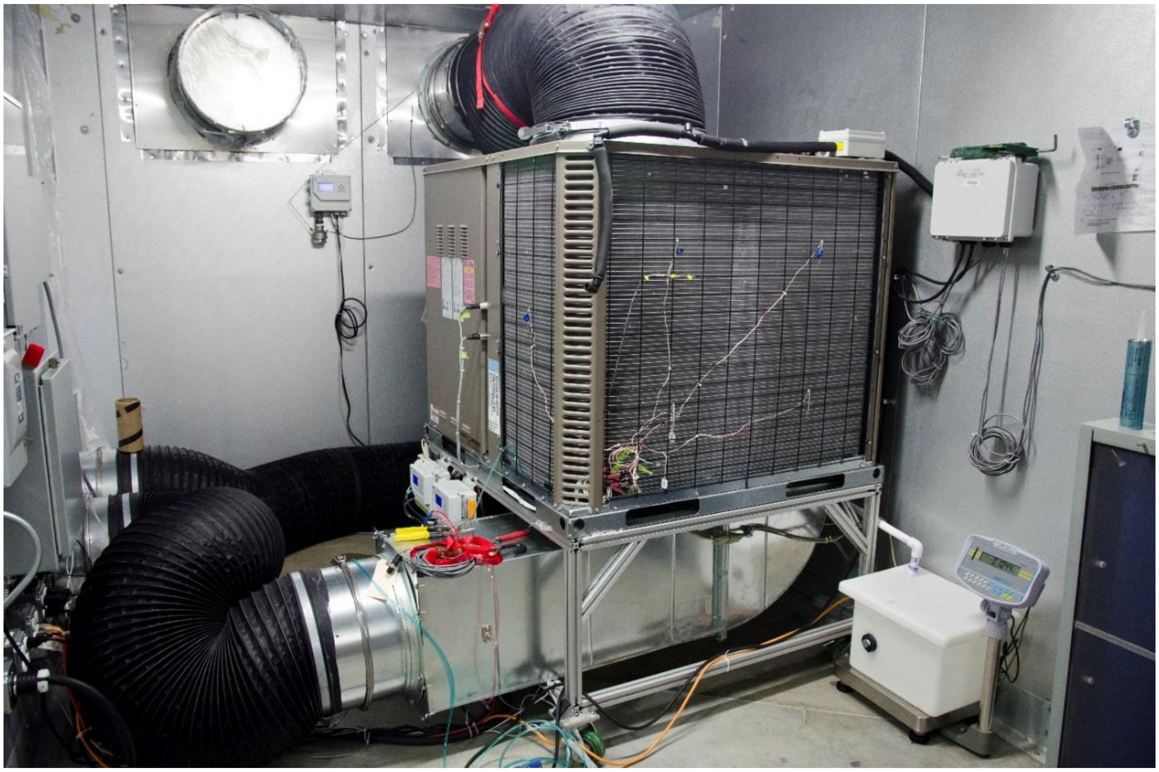

FIGURE 6: TEST UNIT INSTALLED IN THE ENVIRONMENTAL CHAMBER

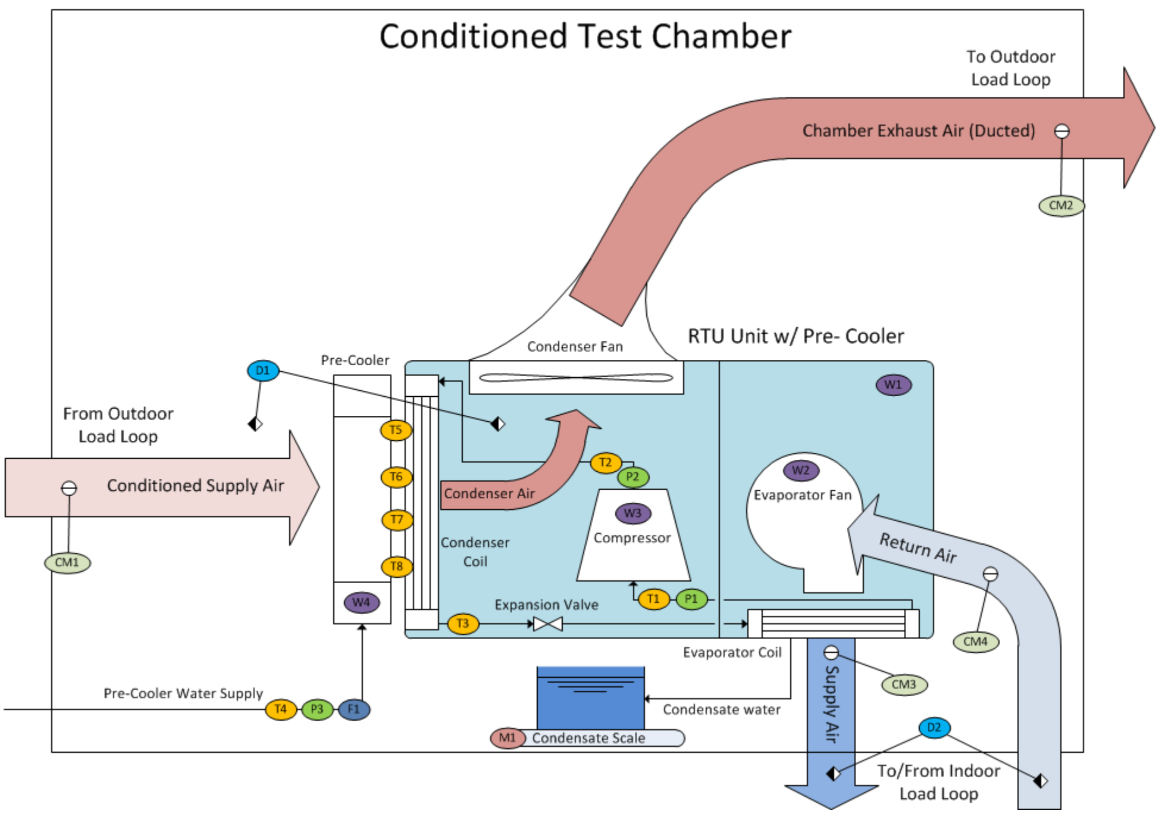

FIGURE 7: MEASUREMENTS FOR PRE-COOLER TESTING APPARATUS

TABLE 6: TABLE OF INSTRUMENTS

MEASUREMENT TYPE	MANUFACTURER AND MODEL #	ACCURACY	SIGNAL TYPE	DAQ CHANNEL	CALIBRATION DATE
Inlet Outdoor Air Temp	GE Optisonde	±0.3°F	RS-232	Serial	03/13/2012 Serial #:0670312
Inlet Outdoor Air Dew Point Temp	GE Optisonde	±0.4°F	RS-232	Serial	03/13/2012 Serial #:0670312
Exhaust Outdoor Air Temp	GE Optisonde	±0.3°F	RS-232	Serial	1/13/2014 Serial #:0051213
Exhaust Outdoor Air Dew Point Temp	GE Optisonde	±0.4°F	RS-232	Serial	1/13/2014 Serial #:0051213
Return Indoor Air Temp	GE Optisonde	±0.3°F	RS-232	Serial	01/30/2014 Serial #:0291113
Return Indoor Air Dew Point Temp	GE Optisonde	±0.4°F	RS-232	Serial	01/30/2014 Serial #:0291113
Supply Indoor Air Temp	GE Optisonde	±0.3°F	RS-232	Serial	3/18/2013 Serial #:0690113
Supply Indoor Air Dew Point Temp	GE Optisonde	±0.4°F	RS-232	Serial	3/18/2013 Serial #:0690113
Delta P Static (Condenser)	Energy Conservatory DG-500	1% of reading	RS-232	Serial	7/23/2013 Serial #CR6547
Delta P Static (RTU Fan)	Energy Conservatory DG-500	1% of reading	RS-232	Serial	
Upstream Flow Nozzle Pressure (Indoor Side)	Energy Conservatory APT	1% of reading	RS-232	Serial	
Flow Nozzle Differential Pressure (Indoor Side)	Energy Conservatory APT	1% of reading	RS-232	Serial	
Upstream Flow Nozzle Pressure (Outdoor Side)	Energy Conservatory APT	1% of reading	RS-232	Serial	
Flow Nozzle Differential Pressure (Outdoor Side)	Energy Conservatory APT	1% of reading	RS-232	Serial	
Indoor Chamber Static Pressure	Energy Conservatory APT	1% of reading	RS-232	Serial	

Measurement Type	Manufacturer and Model #	Accuracy	Signal Type	DAQ Channel	Calibration Date
Outdoor Chamber Static Pressure	Energy Conservatory APT	1% of reading	RS-232	Serial	
Atmospheric Pressure	OMEGADYNE PX409-26BI	±0.08% BSL	4-20mA	NI Compact DAQ Model #9203	3/19/2010
RTU Compressor, Blower, and Total Power	Dent PowerScout 18™	±0.5% kW reading	RS-485	Serial	7/24/2013 Serial# PS18909134
Condensate Generation	Adam Equipment-GBK 16A – Bench Scale	±0.3 g ±0.006 lb	RS-232	Serial	
Pre-cooler Water Temperature	OMEGA RTD	±0.3°F	RTD	NI Compact DAQ Model #9217	7/24/2013
Pre-cooler Water Pressure	Omega PX209-100AI	0.25% of reading	4-20mA	NI Compact DAQ Model #9203	7/24/2013 Serial #83070
Pre-cooler Water Flow Rate	Omega FTB-4705	1% of reading 0.2-10 GPM	Pulse	NI PCI-6321	7/24/2013 Serial #8117297

Pre-Cooler Water Supply Measurements

The flow rate, pressure, and temperature of the water flow to the pre-cooler were measured. The water source available at the laboratory was purified using a reverse osmosis system to prevent scale accumulation on the condensing unit. The pre-cooler water, which was stored in a five gallon buffer tank, was controlled to be 90±1 °F as required by the test protocol. The temperature was measured by an RTD in a circulated flow stream. The temperature was controlled using two relays that turned a small chiller and electrical resistance heater on and off. After filtration and temperature adjustment, the water from the storage tank was pressurized and regulated to 60±10 psi (gauge) which is consistent with typical municipal service water pressure. For pre-cooler devices with intermittent spray patterns, a 5 gallon pressurized buffer tank was added in-line between the pump and pre-cooler to stabilize the operation of the pump. The flow rate of the water was measured, but not controlled, as it was a function of the pre-cooler operation. The water flow meters used were a paddle wheel, pulse output design, where the flow rate is proportional to the frequency of the pulsed signal. Pulses were counted, converted to flow rate using the manufacturer-reported conversion factor, and recorded.

Refrigerant Measurements

Properties of the refrigerant were determined by measuring the temperature and pressure of the refrigerant before and after the compressor, as well as measuring the temperature after the condenser. The refrigerant properties were recorded for information only; they were not used to calculate system capacity. The RTDs used to measure the refrigerant temperatures were placed in contact with the refrigerant pipes and insulated.

EVAPORATOR MEASUREMENTS

The evaporative load was supplied to the unit using a separate load conditioned air chamber, where the indoor load air conditions were controlled similarly to the outdoor conditions described above. Dry bulb temperature, wet bulb temperature, and flow rate were controlled to provide return air at 80/67 (DB°F/WB°F) at the manufacturer specified flow rate for the test unit. The external static pressure for the test unit was maintained at a minimum of 0.20 in H2O, as specified in Table 11 of AHRI/ASHRAE 210/240 [2]. Weight of condensate generated was measured and recorded using a high accuracy bench scale.

CONDENSING AIR MEASUREMENTS

The dry bulb temperature of the air entering the condenser was measured using four RTDs spaced equally over the surface of the condenser. Measurements were taken during baseline tests to ensure uniform temperature distribution of the inlet condenser air. The sensors values were averaged, and maximum and minimum readings were assured to be within 1°F of the average. The average value was compared to the inlet air temperature measured by the GE Optisonde. For the tests of evaporative cooling equipment, only the inlet air temperature measured by the GE Optisonde was used because the additional RTDs become wet during testing yielding incorrect readings of the dry bulb temperature.

DIFFERENTIAL PRESSURE AND AIRFLOW MEASUREMENTS

The differential and static pressures for the environmental chambers were recorded using an Energy Conservatory APT-8 pressure transducer with 8 differential pressure channels. For each chamber, the following values were measured and recorded: the static pressure upstream of the flow nozzle with respect to the laboratory, the differential pressure across the flow nozzle, and the static pressure of the chamber with respect to the laboratory.

Differential pressures for the RTU were measured with an Energy Conservatory DG-500 pressure transducer with two differential pressure channels. These two channels were used to measure differential pressure across just condenser coil and evaporator fan with evaporator (total external static pressure). A baseline measurement across the condenser coil with no ducting attached was performed for the baseline test unit and with each of the pre-coolers tested. This measurement was matching during testing after the ductwork had been reattached to set the condenser air flow rate.

CHAMBER CONDITIONS MEASUREMENTS

During all tests the inlet and exit conditions of both chambers were monitored with four GE Optisonde chilled mirror hygrometers. These sensors use an RTD to measure dry bulb temperature and air from a sampling grid to measure the dew point. Wet bulb temperature is then calculated from the dry bulb temperature and dew point. Data is digitally output via serial interface every second.

POWER MEASUREMENTS

Measurements for the total power, compressor power, and fan power were recorded using a PowerScout 18 with a serial interface and Modbus protocol. It digitally outputs data every three seconds.

DATA ACQUISITION SYSTEM

All signals were acquired using National Instruments hardware at 0.3 Hz or greater, averaged every 30 seconds using LabVIEW software, and logged to a text file.

Tolerances

The goal for all tests was to adhere to the relevant tolerances specified in ANSI/AHRI Standard 210/240-2008 [2], ANSI/AHRI Standard 340/360-2007 [3], and ASHRAE 37-2009 [4]. Tolerances for both indoor and outdoor dry bulb and wet bulb tolerances specified in these standards were adhered to.

The tolerances are listed in Table 8. There are two types of tolerances; the "range tolerance" and the "mean tolerance." The range tolerance specifies the maximum and minimum limits that the controlled variable was allowed, and the mean tolerance specifies the range that the average value of all recorded test points must fall within. The range and mean tolerance had to be met for a 30 minute period to allow the test equipment to reach steady state and for the immediately following 30 minute test period.

TABLE 8: TEST TOLERANCES

TEST CONDITION	RANGE TOLERANCE	MEAN TOLERANCE
Dry Bulb Temp. (indoor and outdoor)	±2°F	±0.5°F
Wet Bulb Temp. (indoor and outdoor)	±1°F	±0.3°F
Pre-Cooler Water Temp.	±2°F	±1°F
Pre-Cooler Water Pressure	60±10 psi	
Condenser Coil Pressure Drop	±7% of setpoint	

In order to operate the condensing unit inside the conditioned chamber, external ducting and fans are needed to replicate the free air condition that the system normally operates in.

The pressure drop across the condenser coil was measured during operation in free air and was replicated with the external ducting attached. Since no information was found for tolerances for this measurement, a sensitivity analysis from a previous experiment was used to approximate the sensitivity of the condensing unit performance with respect to changes in the pressure drop across the condenser coil. In these tests a condensing unit was tested at a range of pressure drops from -15 pascals to -28 pascals where the airflow through the condensing unit was changed by using external resistance and fans while all other variables were held constant. For each test, the unit was allowed to run for 10 minutes to obtain steady state for each pressure drop, and then data was obtained for another 10 minutes after steady state. System coefficient of performance (COP) was calculated for each pressure drop and the results from this previous test are plotted and shown in Figure 8. A tolerance on pressure drop was set to ±7% of the free-air condenser pressure drop. The sensitivity results show this has a less than ±1% impact on COP.

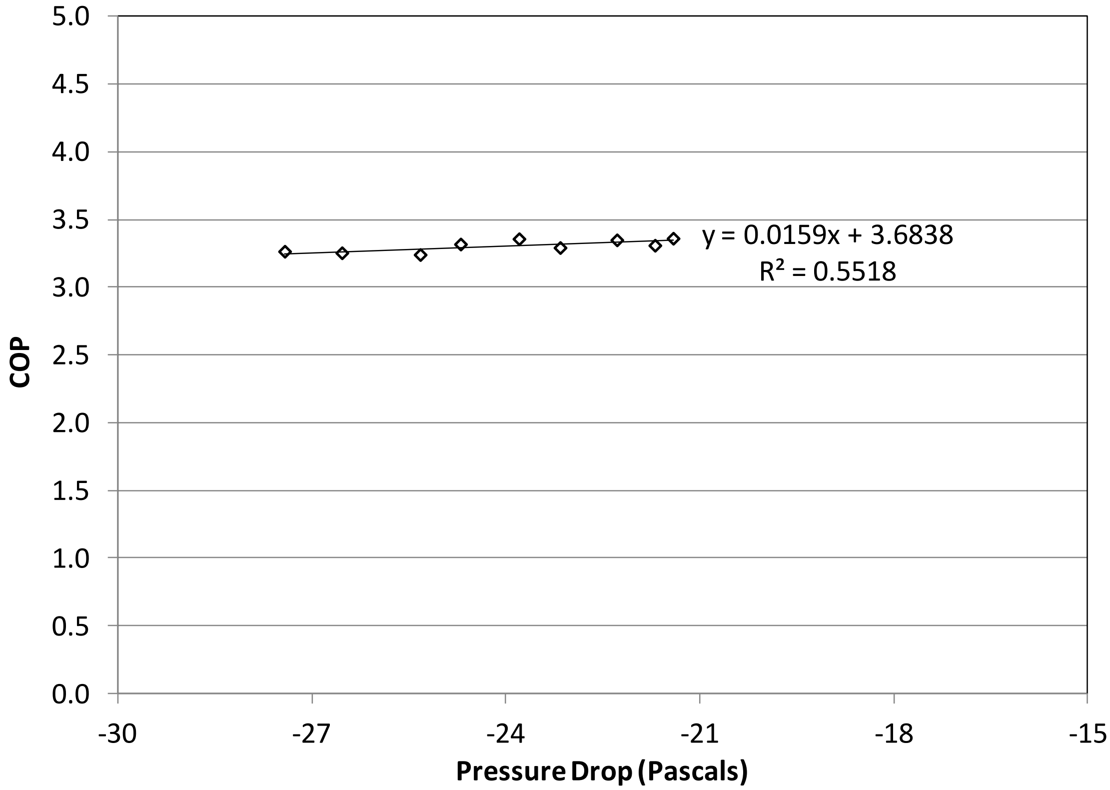

FIGURE 8: SENSITIVITY ANALYSIS OF CONDENSER COIL PRESSURE DROP TOLERANCES

BASELINE TEST RESULTS

The performance of the baseline performance for the RTU is documented in Table 7. The coefficient of performance (COP) versus outdoor air dry bulb temperature is plotted in Figure 9. For comparison, the COP at 95°F as measured by an Air-Conditioning, Heating, and Refrigeration Institute (AHRI) certified lab is shown in the table and plot, and the agreement is within 2% of power, 5% of capacity, and 3% of COP.

TABLE 7 : BASELINE TEST DATA FOR YORK 4-TON RTU

LAB	TARGET OA TEMP °F	ACTUAL OA TEMP °F	ACTUAL RA DB TEMP °F	ACTUAL RA WB TEMP °F	POWER (KW)	CAPACITY (BTUH)	COP
WCEC	64	64.3	80.0	67.0	3.22	53813	4.89
WCEC	73	73.2	80.0	66.8	3.44	51388	4.38
WCEC	75	75.1	80.0	66.8	3.49	50782	4.27
WCEC	82	82.1	80.0	66.6	3.69	47801	3.79
WCEC	90	90.0	80.0	66.8	3.94	45746	3.40
WCEC	95	95.0	80.0	66.7	4.12	43656	3.11
WCEC	105	105.0	80.4	67.0	4.52	39255	2.55
WCEC	115	115.0	80.1	66.8	4.98	35186	2.07
AHRI	95	95	80	67	4.2	46000	3.21

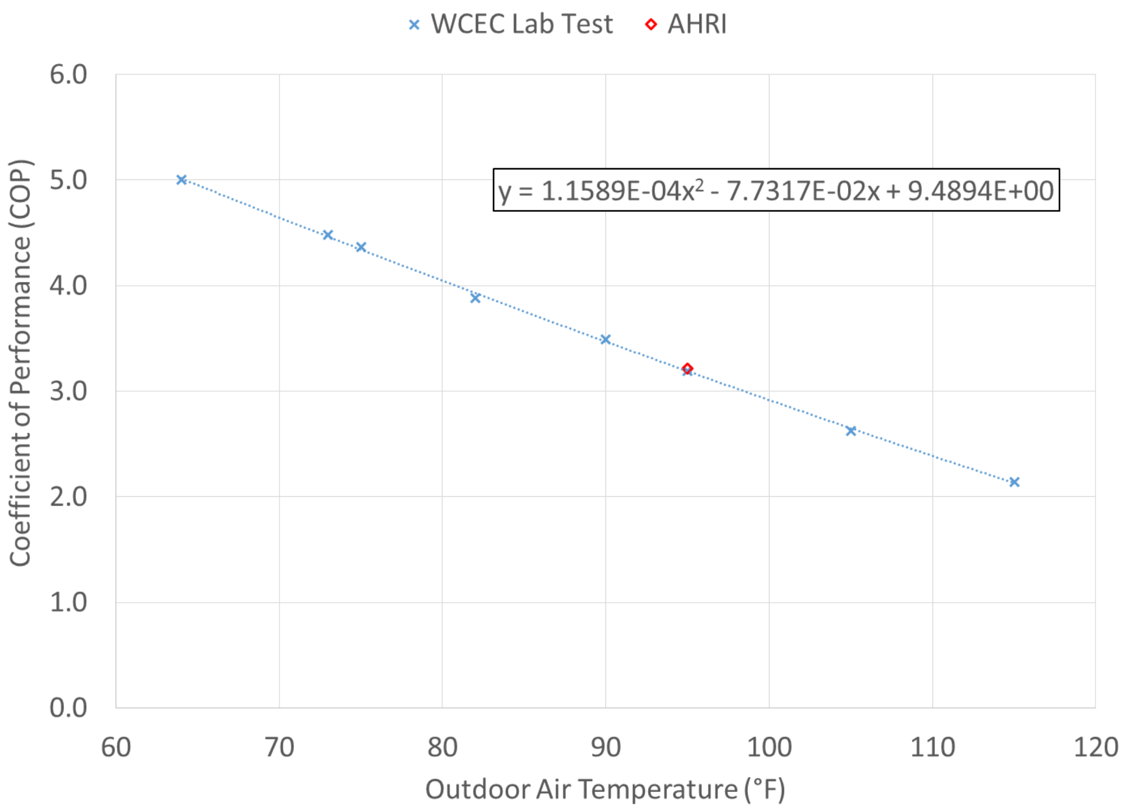

FIGURE 9: COEFFICIENT OF PERFORMANCE FOR BASELINE RTU VERSUS OUTDOOR AIR TEMPERATURE

Data Analysis

For each pre-cooler test, the following calculations were made to determine evaporative effectiveness at each test point. First the capacity of the test unit with pre-cooler installed, power of the test unit with pre-cooler installed, and coefficient of performance with the pre-cooler installed were measured and calculated as described in the following sections. The resulting coefficient of performance was compared to the least squares polynomial curve for baseline coefficient of performance trend in order to calculate the equivalent evaporative effectiveness, the equivalent evaporated water, and the water-use efficiency.

CAPACITY

The capacity of the test unit with the pre-cooler installed was determined for each test from Equation 2 [4]:

EQUATION 1: CAPACITY

$$q = \frac{Q_e \times (h_1 - h_2)}{v_{e,n} \times (1 + W_{e,n})}$$

where Q_e is the measured flow rate of the evaporator air in ft^3/min as described by ANSI/ASHRAE Standard 41.2-1987 [5], h_1 and h_2 are the enthalpy of the return and supply air, respectively, in btu/lb, $v_{e,n}$ is the specific volume of dry air at the evaporator side nozzle, measured in ft^3/lb, and $W_{e,n}$ is the humidity ratio of the air at the evaporator side nozzle in lbw/lba.

Coefficient of Performance

The coefficient of performance (COP) of the test unit with the pre-cooler installed was determined for each test from Equation 2:

Equation 2: Coefficient of Performance

$$COP = \frac{q}{P}$$

where q is the capacity of the test unit as calculated in equation 2 and P is the power of the unit, including the compressor, condenser fan, and blower.

Evaporative Effectiveness

The evaporative effectiveness (EE) of an evaporative pre-cooler apparatus is defined as how closely the dry bulb temperature leaving the pre-cooler approaches saturation along the wet bulb temperature line (Equation 3).

Equation 3: Evaporative Effectiveness

$$EE = \frac{T_{dB,in} - T_{dB,out}}{T_{dB,in} - T_{wB,in}}$$

where $T_{dB,in}$ and $T_{wB,in}$are the dry bulb and wet bulb temperatures entering the pre-cooler and $T_{dB,out}$ is the dry bulb temperature leaving the pre-cooler.

Measuring the temperature at the pre-cooler outlet of evaporative pre-coolers is difficult for several reasons. It is difficult to measure directly because water droplets on the temperature sensors give inaccurate measurements. In addition, the air leaving the pre-cooler apparatus may be poorly mixed, which causes difficulty in determining where or how to take the measurement. A possible workaround involves measuring the temperature and humidity of the condenser exhaust, using psychometric calculations to back out the air temperature at the condenser inlet. This involves assuming that the absolute humidity ratio is constant between the condenser inlet and the exhaust, and that the wet bulb temperature is constant as the air passes through the pre-cooler Figure 10. This is potentially unreliable for several reasons: 1) poorly mixed exhaust air contributes to measurement inaccuracy, 2) pre-coolers that have pulsing sprays have fluctuating exhaust data that is difficult to measure, and 3) the method ignores heat transfer benefits from water directly contacting the condensing coil.

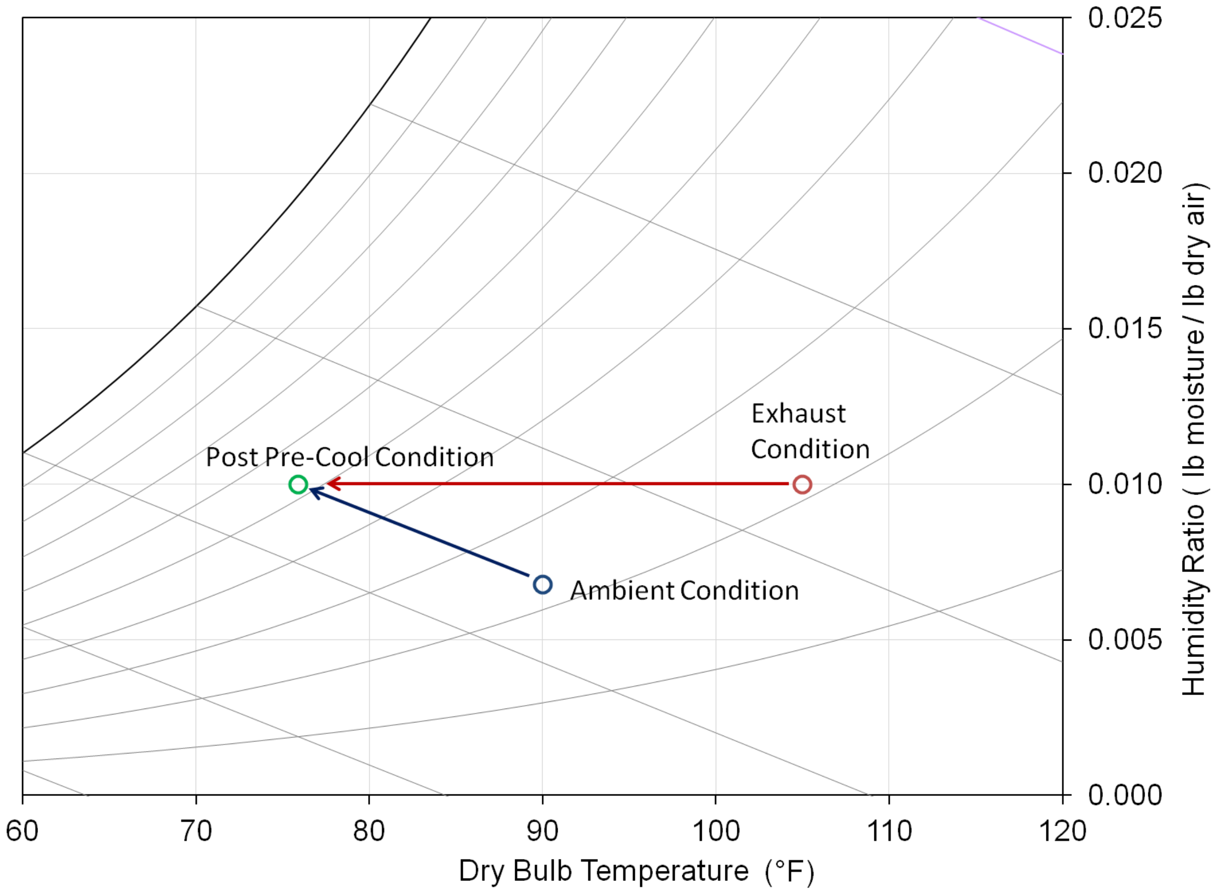

FIGURE 10: USING EXHAUST AND AMBIENT CONDITIONS TO CALCULATE POST PRE-COOL CONDITION

To compensate for the deficiencies of using the exhaust measurements to calculate evaporative effectiveness, another method for calculating the equivalent EE was developed. This method assumes that the performance of the unit (COP) is only a function of the outside air dry bulb temperature when evaporator conditions are held constant; with the installation of an evaporative pre-cooler on a condensing unit, the equivalent air temperature seen by the condenser is changed. For example, the condensing unit will operate the same for both of the following scenarios:

1. The outside air temperature is 90°F and there is no evaporative pre-cooler installed; or
2. The outside air temperature is 105°F and an evaporative pre-cooler is installed that cools the air to an average of 90°F and supplies this air to the condenser coil.

Since the condensing unit will perform comparably for the same condenser inlet temperatures, the equivalent dry bulb temperature seen by the condenser with the pre-cooler installed can be calculated by using the baseline condenser data with no pre-cooler installed. For the remainder of the results this method is used to determine the evaporative effectiveness of the pre-cooler at each test point.

Using this theory, the equivalent dry bulb temperature was calculated by solving for the point on the baseline curve where the condensing unit performs comparably to the test point, as shown in Figure 11, which is an example calculation using COP data as the performance metric. The equivalent dry bulb temperature leaving the pre-cooler apparatus is calculated by determining the temperature on the baseline curve where the COP is equal to the COP obtained during the test period.

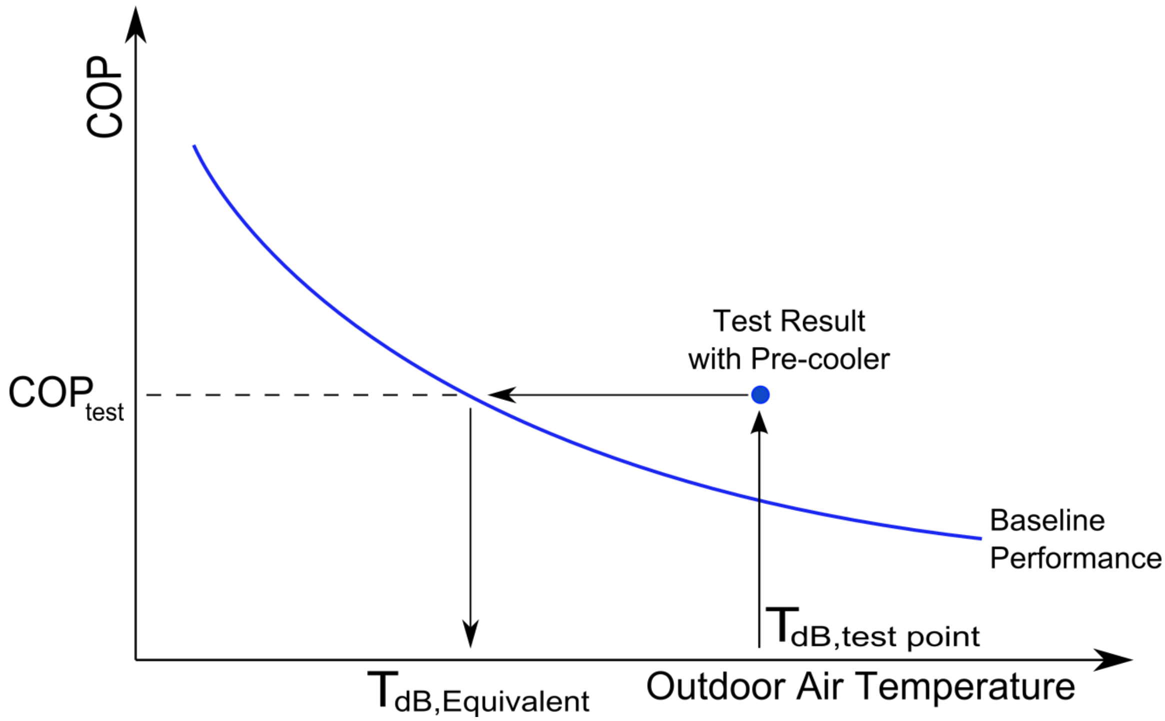

FIGURE 11: PROCESS FOR CALCULATING TDB,EQUIVALENT

The baseline curve for COP obtained in this experiment is a second order polynomial. The general equation for a second order polynomial is shown in Equation 4.

EQUATION 4: GENERAL SECOND ORDER POLYNOMIAL

$$COP = a \cdot {T_{dB}}^2 + b \cdot T_{dB} + c$$

Constants a, b, and c are solved from a least squares fit of the baseline test data from the condensing unit (Figure 9), T_{dB} is the condenser inlet dry bulb temperature, and COP is the coefficient of performance of the unit. To determine the equivalent dry bulb temperature entering the condenser during a pre-cooler test, the quadratic equation was solved as shown in Equation 5.

EQUATION 5: GENERAL EQUATION TO DETERMINE TDB,EQUIVALENT

$$T_{dB,eq} = \frac{-b + \sqrt{b^2 - 4a(c - COP_{test})}}{2a}$$

Constants a, b, and c are equal to the constants of the second order baseline equation in Equation 4, COP_{test} is the COP of the test unit measured during the pre-cooler test, and $T_{dB,eq}$ is the equivalent dry bulb temperature of the test. Using the equivalent dry bulb temperature, the evaporative effectiveness of each pre-cooler was solved using Equation 6:

EQUATION 6: EVAPORATIVE EFFECTIVENESS

$$EE = \frac{T_{dB,in} - T_{dB,\mathrm{eq}}}{T_{dB,in} - T_{wB,in}}$$

where $T_{dB,in}$ and $T_{wB,in}$ are the dry bulb and wet bulb temperatures entering the pre-cooler and $T_{dB,eq}$ is the equivalent dry bulb temperature from Equation 5. The evaporative effectiveness for all five products calculated and the results are shown in Figure 13.

WATER-USE EFFECTIVENESS

Water-use effectiveness (WUE) is defined as the percentage of water that is used for pre-cooling divided by the total water supplied to the pre-cooler. In order to calculate the water-use effectiveness, it is necessary to calculate the rate at which water is evaporated into the air before passing through the condensing unit. This can be calculated using Equation 7.

EQUATION 7: WATER EVAPORATION RATE

$$\dot{m}_{water,evap} = {}^{(W_{out} - W_{in}) \times Q_{c,n}}/_{v_{c,n}}$$

where $\dot{m}_{water,evap}$ is the rate at which water evaporates into the air in lb/min and W_{out} and W_{in} are the humidity ratio exiting and entering the pre-cooler apparatus in lbw/lbda, respectively. The volumetric flow rate of the air, across the condenser, in ft3/min, $Q_{c,n}$ was measured as described by ANSI/ASHRAE Standard 41.2-1987 [5]. The specific volume of the dry air at the condenser side nozzle, $v_{c,n}$, was measured in ft3/lb. The exiting humidity ratio, W_{out}, was calculated using the equivalent dry bulb temperature (Equation 5) and a psychometric calculator to determine the humidity ratio at that dry bulb temperature, which assumes the pre-cooling process has a constant wet bulb temperature. With this, the water-use effectiveness was calculated as shown in Equation 8.

EQUATION 8: WATER-USE EFFECTIVENESS

$$WUE = \frac{\dot{m}_{water,evap}}{\dot{m}_{water,supplied}}$$

where the volumetric flow rate of supplied water was converted to units of lb/min, assuming a density of 8.33 lb/gal, and recorded as $\dot{m}_{water,supplied}$. The results are plotted in Figure 14.

MEASUREMENT UNCERTAINTY

The uncertainty of the evaporative effectiveness and the water-use effectiveness calculations were conducted using the sequential perturbation method, which is a numerical approach that utilizes a finite difference method to approximate the derivatives representing the sensitivity of the calculated value to the variables used within the calculation [6]. This method is well accepted and used when the partial differentiation method of the propagation of error is complex, or the amount of variables used is very large. The process used for sequential perturbation involves calculating a result, R_o, based on measured values. After R_o has been calculated, an independent variable within the equation for R_o is increased by its respective uncertainty, and a new value, R_i^+ is calculated. Next, the same independent variable within R_o is decreased by its respective uncertainty, and a new value, R_i^- is calculated. The differences between R_i^+ and R_o, and R_i^- and R_o are calculated and the absolute values are averaged. The result is defined as δR_i. This process is repeated for every independent variable within R_o, and the final uncertainty is calculated as shown in Equation 9.

EQUATION 9: UNCERTAINTY USING SEQUENTIAL PERTURBATION

$$U_R = \pm\left[\sum_{i=1}^{L}\left(\delta R_i{}^2\right)\right]^{1/2}$$

An example of the error propagation for the test of pre-cooler 3 and outdoor air conditions of 115°F DB and 75°F WB is shown in Figure 12. The process starts with measured values, and their uncertainties shown in the blue boxes of the figure. As intermediate values and their uncertainties are calculated by the method described above, using the appropriate equations. Uncertainties are propagated until those related to the final desired values are reached. The uncertainty of the evaporative effectiveness and water-use effectiveness were calculated using this method for all five pre-cooler products for all tests conducted and the results are shown as error bars in Figure 13 and Figure 14. Of note in these calculations, as the wet bulb depression of the outdoor air increases, the accuracy of the signal relative to the noise significantly improves.

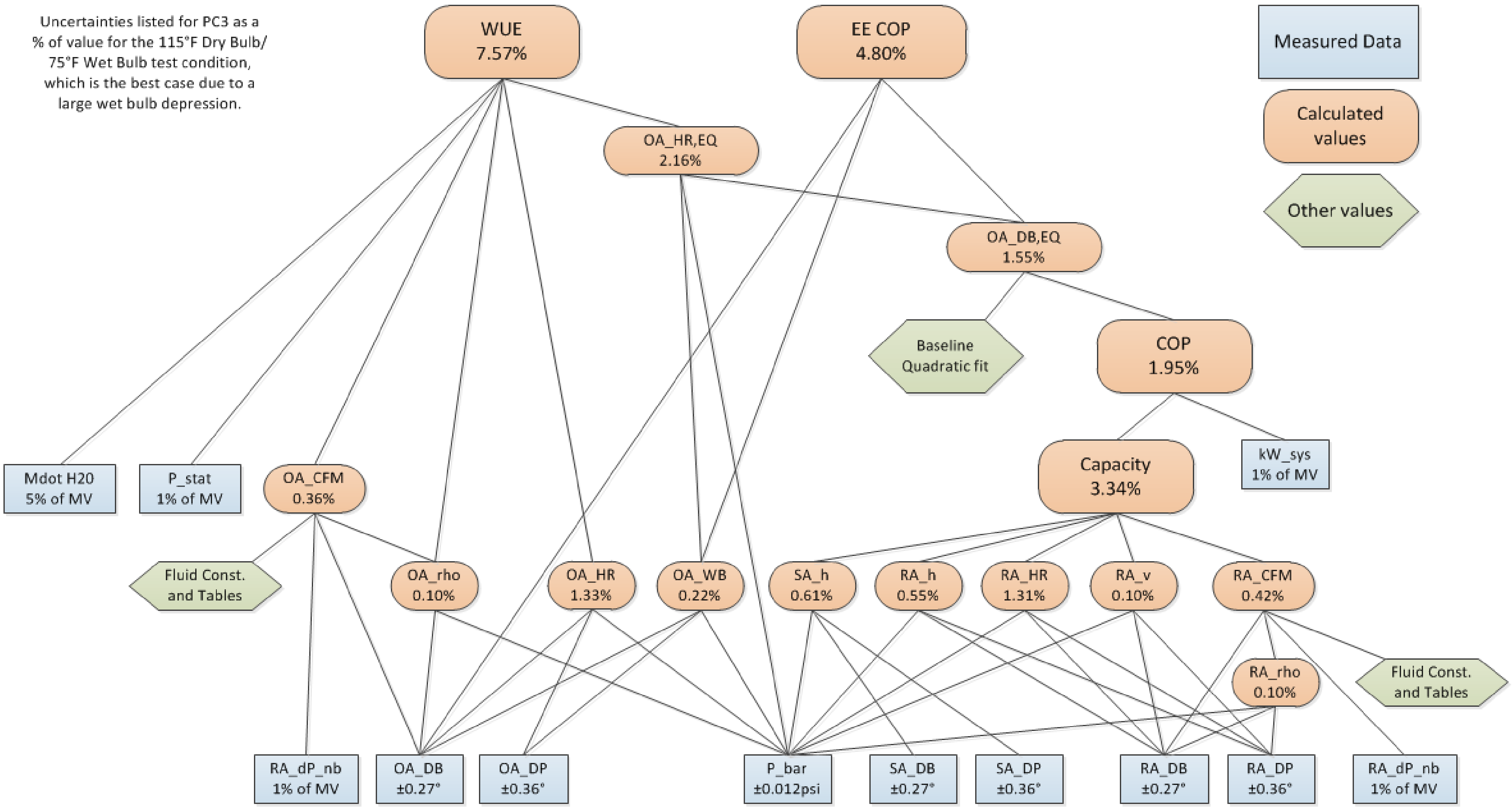

FIGURE 12: EXAMPLE UNCERTAINTY ANALYSIS PROPAGATION

Evaporative Effectiveness

Evaporative effectiveness varied for each pre-cooler technology, with results ranging between 20-80% for the four constant dew point tests at outdoor air temperatures of 85-115°F (Figure 13). Evaporative effectiveness of pre-coolers 3, 4, and 5 were similar and, in most cases, the results clustered together within the uncertainty limits and between 60-75% evaporative effectiveness. The exceptions were that the performance of pre-cooler 4 was reduced at 115° and the performance of pre-cooler 5 was reduced at 85°F. Generally speaking, the results show that pre-coolers with significant design differences are able to achieve similar results for evaporative effectiveness. The limit for designs tested to date in all tests was 75% evaporative effectiveness (Appendix: Table 11).

Evaporative effectiveness of pre-coolers 1 and 2 was lacking in comparison to pre-coolers 3, 4, and 5, with all evaporative effectiveness measurements between 20-50%. Pre-cooler 1 showed increased performance with increasing outdoor air temperature.

In general, the magnitude of the uncertainty relative to the magnitude of the result is concerning for the 85°F test results. For the purposes of setting test protocols and performance requirements for utility rebate programs, the results at 95°F and higher may be more useful from the stand-point of product comparisons.

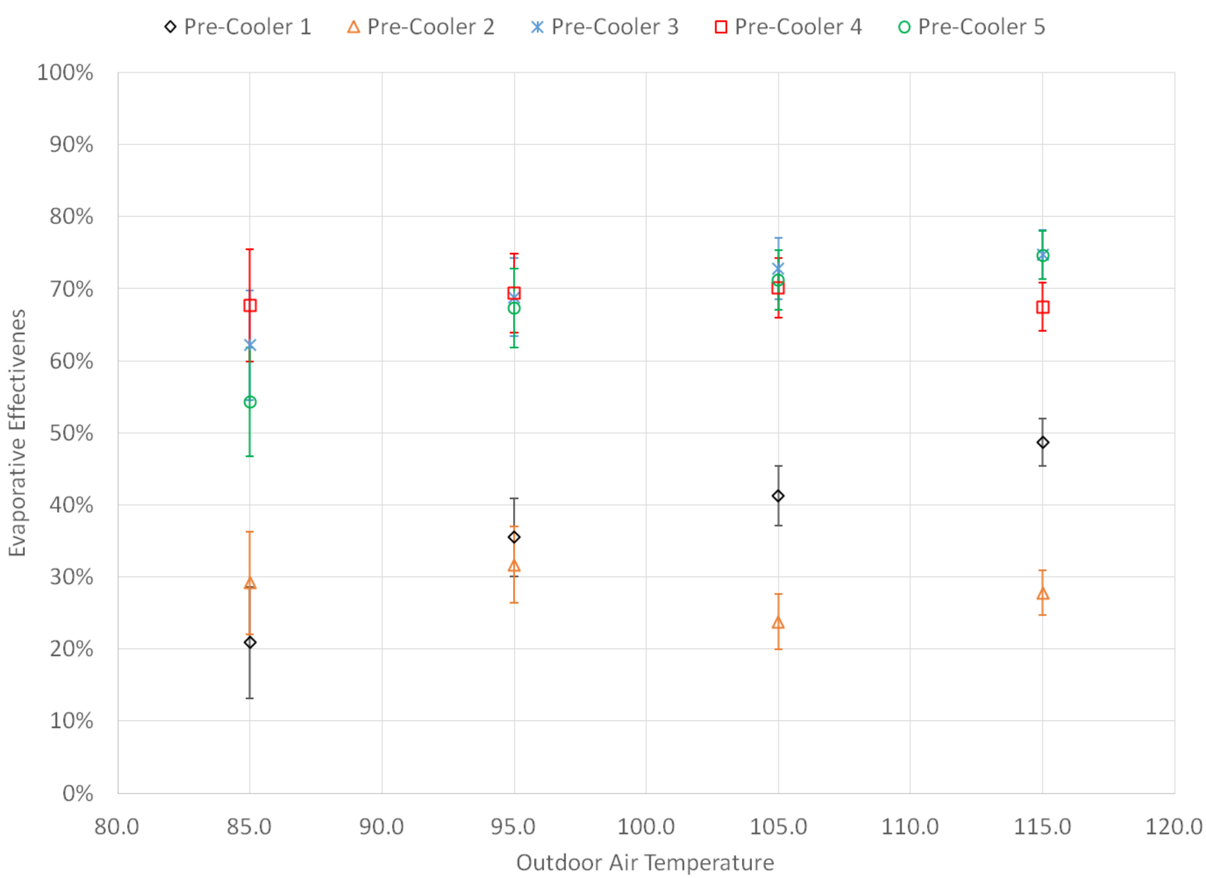

FIGURE 13: COMPARISON OF EVAPORATIVE EFFECTIVENESS OF FIVE PRE-COOLERS TESTED. OUTDOOR AIR DEWPOINT IS CONSTANT FOR ALL TESTS (56°F).

Water-Use Effectiveness

Water-use effectiveness was highest for pre-coolers 3 and 4, measuring between 80-100% in the four constant dew point tests at outdoor air temperatures of 85-115°F (Figure 14). However, pre-coolers 4 and 5 are re-circulation technologies and require a constant bleed of sump water to prevent scaling of the pre-cooler (Table 8). This maintenance water is not included here and may increase water use 10-50% based on manufacturer recommendations and the hardness of the water supply (which may be reduced by softening, which has its own water burden). It should also be noted that both of these systems are configured so that the condensate generated by the air conditioner can be routed to the pre-cooler sump, which would reduce the water burden slightly (this was not considered in the laboratory test).

Water-use effectiveness for pre-cooler 5 was in the range of 55-75%, increasing with outdoor air temperature. Pre-cooler 5 does not re-circulate water and requires no maintenance water. This pre-cooler creates a mist upstream of the coil. The system is designed to evaporative the water prior to the condenser coil surface. However, the manufacturer recommends water treatment such as softening or possibly reverse osmosis to protect the condenser coil from stray droplets. In addition, the water treatment is needed to prevent small orifice nozzles from clogging. Water treatment methods such as softening and reverse osmosis consume additional resources including water, electricity, and salt, which were not considered in this analysis.

Water-use effectiveness measured for pre-coolers 1 and 2 was generally less than 50% and was lacking in comparison to the performance of pre-coolers 3-5. Pre-coolers 1 and 2 do not re-circulate water so do not require maintenance water. Pre-cooler 2 sprays directly on the coil and therefore requires water treatment such as softening or osmosis for continuous use. The manufacturer also markets the device for occasional peak demand load shedding, in which case occasional wetting of the condenser coil may be considered acceptable by the customer.

TABLE 8: WATER MANAGEMENT METHODS OF TESTED PRE-COOLERS

	Re-circulation?	Coil-Wetted?
PC1	NO	NO
PC2	NO	YES
PC3	YES	NO
PC4	YES	NO
PC5	NO	Not intentionally, some droplets may reach coil

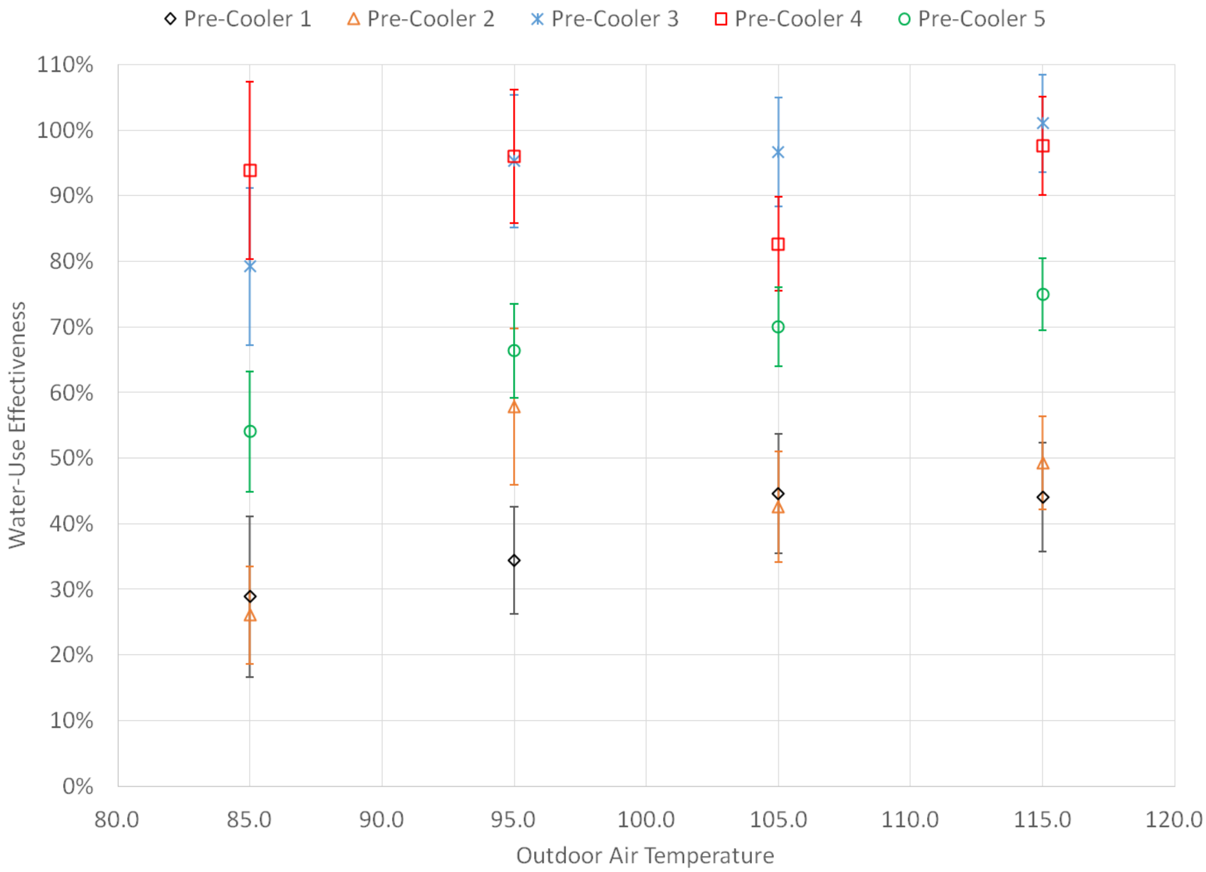

Figure 14: Comparison of Water-Use Effectiveness of Five Pre-coolers Tested. Outdoor Air Dewpoint is Constant For all Tests (56°F).

Pre-Cooler Power Consumption

The pre-cooler power consumption is not included in the evaporative effectiveness calculation and must be accounted for separately. Pre-coolers 1 and 2 consumed minimal power to accomplish the delivered pre-cooling, averaging 51 Watts and 20 Watts, respectively. Pre-cooler 3 consumed an average of 117 Watts. Pre-coolers 4 and 5 used high pressure pumping systems and consumed significantly more power than other pre-coolers at 287 and 506 Watts, respectively. However, manufacturers of these pre-coolers only market pre-coolers to cooling systems sized 20 tons or greater, and did not have a pump sized appropriately for the laboratory experiment on the 4 ton cooling system. The pump for pre-cooler 4 is sized for a cooling system of approximately 20 tons and the pump for pre-cooler 5 is sized for a cooling system of approximately 25 tons. For the purposes of laboratory testing over-sized pumps the extra water was pumped in a re-circulation loop.

TABLE 9: PRE-COOLER POWER CONSUMPTION

		PRE-COOLER POWER CONSUMPTION (AVERAGE WATTS DURING TEST)				
PRE-COOLER	NOTES	85°F TEST	95°F TEST	105°F TEST	115°F TEST	AVERAGE
PC1	Controller	50	50	51	53	51
PC2	Controller	28	10	8	32	20
PC3	Re-circulation pump	118	117	117	116	117
PC4	Re-circulation pump (sized for 20-ton cooling system)	302	271	290	284	287
PC5	Re-circulation pump (sized for 25-ton cooling system)	474	510	516	525	506

Impact of Dry Media

Pre-coolers 1, 3, and 4 have media in front of the condenser coil that causes some resistance to airflow. This may reduce performance of the air conditioning system during times when the air conditioner is on but the pre-cooler is not running. In some cases, the manufacturer may configure the controller to always run the pre-cooler when the air conditioner is on. In other cases, the manufacturer may set a minimum outdoor air temperature threshold that has to be met for the pre-cooler to turn on, in which case the air conditioner may run without the pre-cooler. Specific control schemes were not evaluated during this laboratory test. However, a single test was run at an outdoor air temperature of 75°F to determine the impact of a "dry" pre-cooler on the air conditioner performance.

Pre-coolers 3 and 4 had minimal performance impacts in the presence of a "dry" pre-cooler (Table 10). The power consumption increased less than 1% and the efficiency decreased approximately 1%. Pre-cooler 1 had a significant impact. The power increased 1.7% and the efficiency decreased more than 10%. Pre-coolers 2 and 5 were not tested because the pre-cooler did not contain a media.

TABLE 10: IMPACT OF DRY MEDIA ON RTU PERFORMANCE AT 75°F OUTDOOR AIR TEMPERATURE

	% Power Impact	% Capacity Impact	% COP Impact
PC1	1.70%	-9.21%	-10.50%
PC2	N/A	N/A	N/A
PC3	0.55%	-0.77%	-1.06%
PC4	0.73%	-0.45%	-0.93%
PC5	N/A	N/A	N/A

Expected Energy Savings and Demand Reduction

A model for energy savings and demand reduction in climate zones in SCE territory was developed as part of a previous project on pre-cooling technology [1]. The results from that analysis are summarized here, and the complete methodology is available in the referenced report. The absolute and percent savings for both energy and peak demand were determined by calculating the difference between a baseline RTU and the same RTU with pre-cooler installed. The results show that increasing evaporative effectiveness of the pre-cooler increases energy savings and that inland climate zones are expected to have higher savings than coastal climate zones (Figure 15 - Figure 18).

The energy savings described here are for an average RTU based on aggregate load data. An RTU with increased run time would have a greater total energy savings. If the baseline energy use of a particular RTU or building is known, pre-coolers can be strategically installed on units with high run times to increase annual energy savings. Peak demand savings are not a function of the load data and are strictly a function of modeled RTU efficiency and pre-cooler effectiveness. Since all RTUs are assumed to run during a peak event, the savings are expected for any RTU regardless of the load profile of the building.

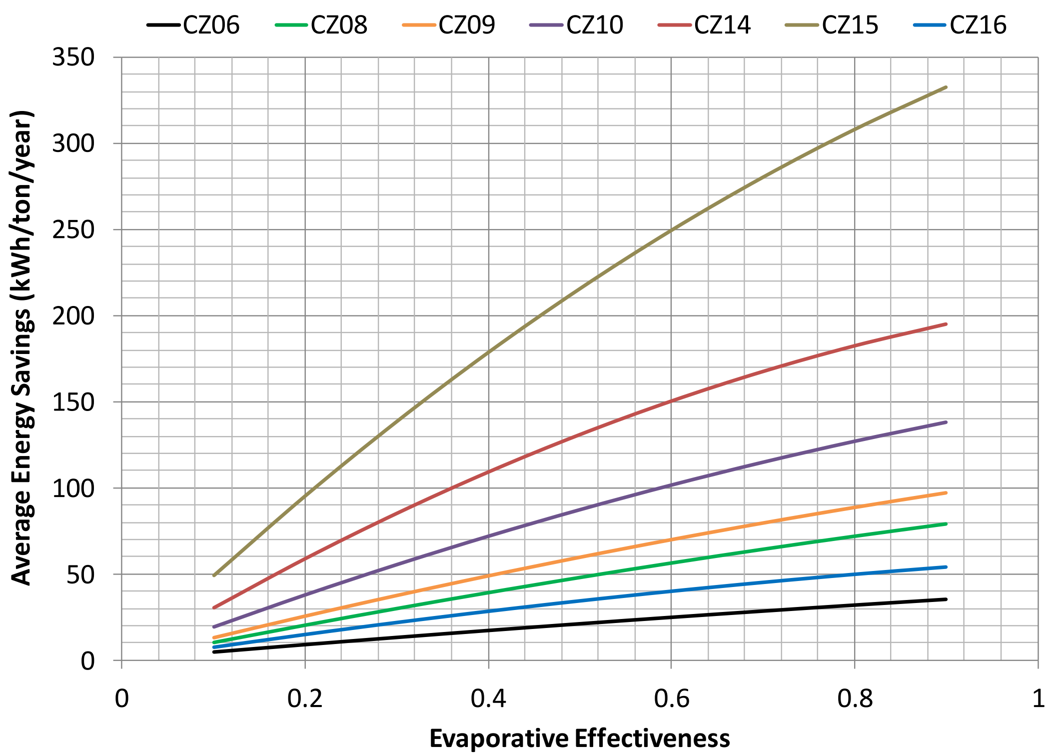

FIGURE 15: MODELED AVERAGE ENERGY SAVINGS OF AN EVAPORATIVE CONDENSER AIR PRE-COOLER

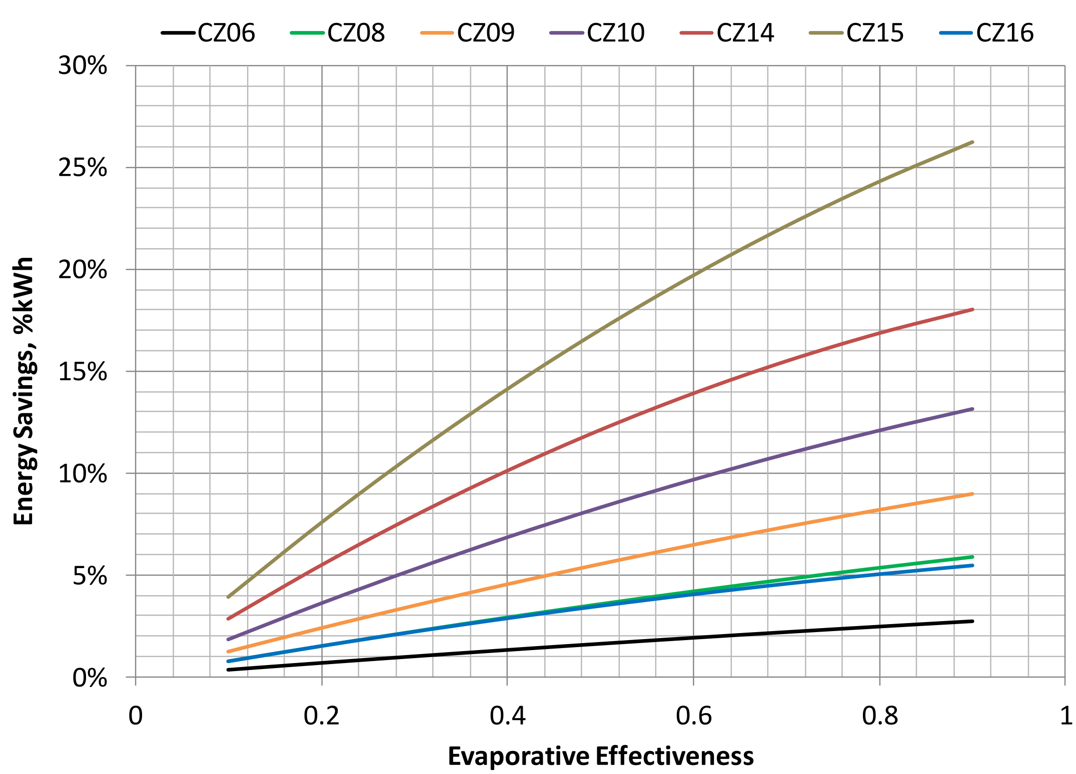

FIGURE 16: MODELED AVERAGE PERCENT ENERGY SAVINGS OF AN EVAPORATIVE CONDENSER AIR PRE-COOLER

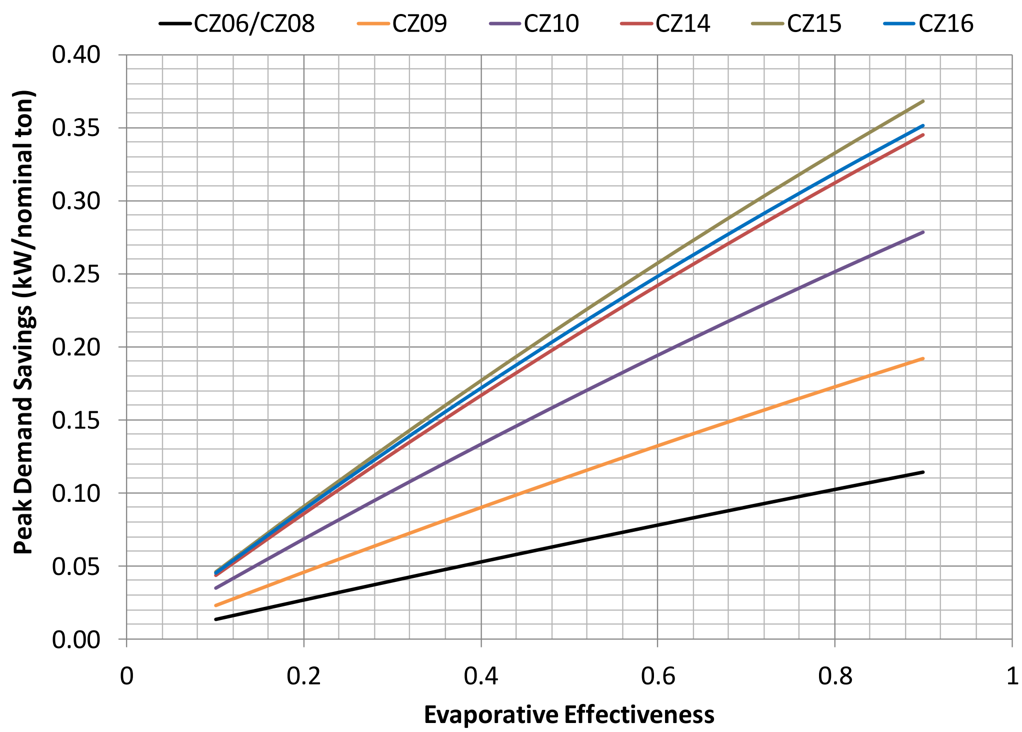

FIGURE 17: MODELED POWER SAVINGS OF AN EVAPORATIVE CONDENSER AIR PRE-COOLER

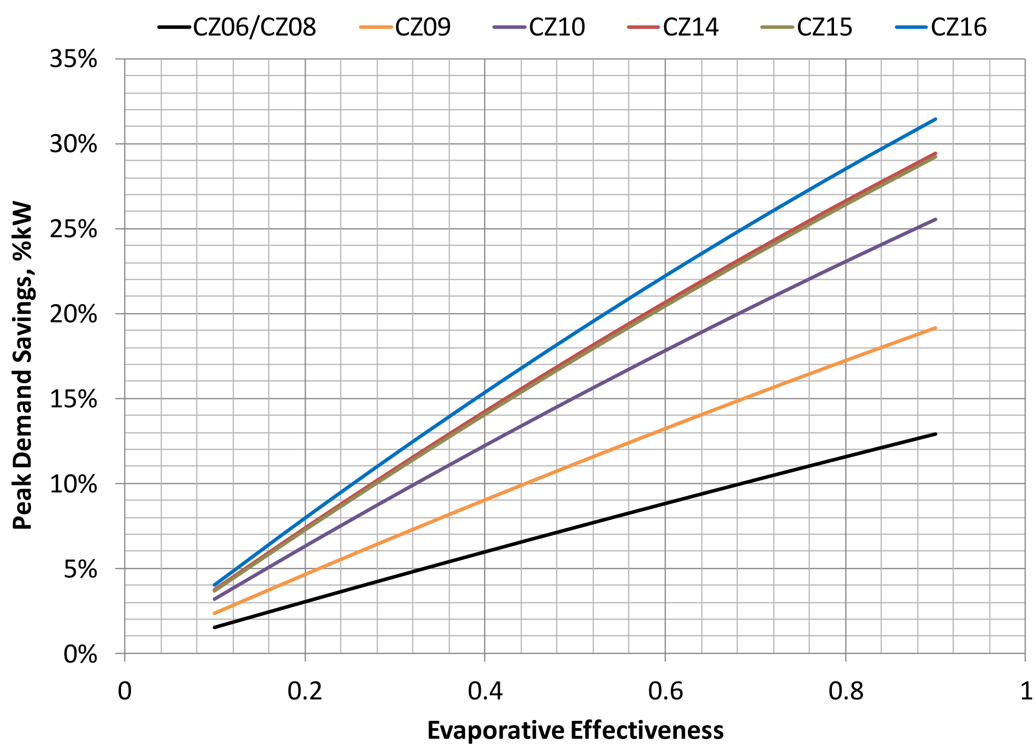

FIGURE 18: MODELED PERCENT POWER SAVINGS OF AN EVAPORATIVE CONDENSER AIR PRE-COOLER

Cost of Technology

The cost of evaporative pre-coolers varies by manufacturer and the size of the installation, but an estimated cost range is $250-$500 per ton of cooling equipment retrofitted.

Life expectancy of Technology

Generally speaking, pre-cooler components may consist of pumps, sprayers, evaporative media, and electronic controllers. The evaporative media is expected to need periodic replacement, similar to air filter replacement in air handlers. The period of replacement is a function of the type of media and the water quality at the location. Generally speaking, the media should last at least one cooling season and possibly several cooling seasons. Several evaporative pre-cooling manufacturers include a maintenance agreement in the sale of the pre-cooler, and this is recommended to ensure persistence in performance. Several pre-cooling manufacturers also include a 10 year warranty in the sale of their pre-cooler, providing reasonable assurance that the life expectancy of the technology is at least 10 years.

Furthermore, pre-coolers allow compressors to operate at reduced head pressures. It is reasonable that this would extend the life of compressors and reduce failures. Pre-coolers with evaporative media, while requiring periodic replacement, protect the condenser coils and fins from dirt and debris, potentially extending the lifetime of the condenser coil.

CONCLUSIONS AND RECOMMENDATIONS

A laboratory test protocol that objectively compared evaporative pre-coolers from five manufacturers of differing designs was successfully demonstrated. The main findings of the testing were that:

1. The five products demonstrated evaporative effectiveness in the range of 20 – 80% and water-use effectiveness in the 25 – 100%.
2. The three highest performing products demonstrated evaporative effectiveness in the range of 50-80% and a water-use effectiveness greater than 50%. Of these pre-coolers, two recirculated water and had the highest water-use effectiveness (>80%), however, in field applications would require additional water use for "maintenance" or "bleed" water because they are recirculation systems, which was not accounted for in the laboratory test.
3. The two lowest performing products demonstrated evaporative effectiveness consistently below 50%, with water use effectiveness generally below 50%.

Because pre-coolers designs are highly variable and the specific design and control methodology will impact performance, it is critical to complete publication of the ASHRAE test standard, so that end-users and utilities will have objective test data that can be used to compare evaporative pre-cooler products and forecast energy savings.

An analysis tool previously developed shows energy savings and demand reduction vary as a function of evaporative effectiveness and climate. As an example, an evaporative pre-cooler with 70% evaporative effectiveness installed in climate zone 10, one of the most populous in Southern California, is estimated to achieve a total energy savings of ~10% and a peak demand savings of ~20%. In the same climate zone, an evaporative effectiveness of 50% is estimated to achieve a total energy savings of ~8% and a peak demand savings of ~15%.

WCEC recommends inclusion of evaporative pre-cooling products for inclusion in rebate programs. WCEC also recommends that pre-coolers pass laboratory certification based on the test protocol described in this report, with performance requirements set by the utility. Rebate structures could be tiered to qualify more products, setting a higher rebate level for higher performance products.

There are two obvious market barriers prevent widespread adoption of this technology. The first is the lack of infrastructure (published protocol and test facilities) to test and certify pre-cooler products if a rebate program was implemented that required certification. The second, and most serious market barrier, is the staggering drought currently facing California. While evaporative cooling uses a small amount of water in comparison to other building requirements, any additional water burdens will be heavily scrutinized by end-users and regulators. We can address this issue by 1) minimizing the water used in pre-coolers, by optimizing bleed rates to use as little water as possible, 2) evaluating rainwater capture and greywater and potential non-potable water sources, and 3) quantifying the trade-off between electricity saved and water used, and the water and electricity impacts involved in water transportation and electricity generation.

APPENDICES

TABLE 11: TABLE OF ALL RESULTS FOR FIVE PRE-COOLERS TESTED

Pre-Cooler	Outdoor Air Dry Bulb Temp °F	Outdoor Air Dew Point Temp °F	Evaporative Effectiveness	Evaporative Effectiveness Absolute Uncertainty	Water-Use Effectiveness	Water-Use Effectiveness Absolute Uncertainty
1	85.0	57.0	21%	8%	29%	12%
1	95.0	56.9	36%	5%	34%	8%
1	104.9	57.0	41%	4%	45%	9%
1	115.0	56.8	49%	3%	44%	8%
2	86.0	55.9	29%	7%	26%	7%
2	90.1	46.7	29%	9%	59%	19%
2	94.9	57.4	32%	5%	58%	12%
2	95.1	66.8	27%	7%	79%	25%
2	105.1	56.1	24%	4%	43%	8%
2	115.0	56.1	28%	3%	49%	7%
3	75.0	57.1	54%	21%	70%	29%
3	82.0	68.9	44%	25%	66%	42%
3	85.0	56.4	62%	8%	79%	12%
3	85.1	40.4	63%	10%	91%	15%
3	90.0	48.9	67%	6%	94%	10%
3	90.0	46.3	65%	5%	88%	9%
3	95.0	66.1	71%	7%	97%	13%
3	95.1	56.6	69%	5%	95%	10%
3	100.1	64.2	72%	5%	71%	8%
3	104.9	57.2	73%	4%	97%	8%
3	105.0	71.1	68%	5%	95%	11%
3	115.1	56.0	75%	3%	101%	7%
4	75.1	57.1	47%	20%	66%	30%
4	82.0	69.0	45%	25%	71%	44%
4	85.0	57.4	68%	8%	94%	14%
4	85.1	40.2	71%	9%	98%	14%
4	89.9	48.2	71%	5%	97%	10%
4	89.9	45.7	72%	5%	101%	10%
4	95.0	57.1	69%	5%	96%	10%
4	95.0	66.2	65%	7%	92%	13%
4	100.0	64.4	67%	5%	99%	11%
4	104.9	57.0	70%	4%	83%	7%
4	104.9	70.8	63%	5%	92%	12%
4	114.9	56.3	67%	3%	98%	8%
5	85.0	56.6	54%	8%	54%	9%
5	95.0	56.8	67%	5%	66%	7%
5	105.0	56.5	71%	4%	70%	6%
5	115.0	56.3	75%	3%	75%	6%

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