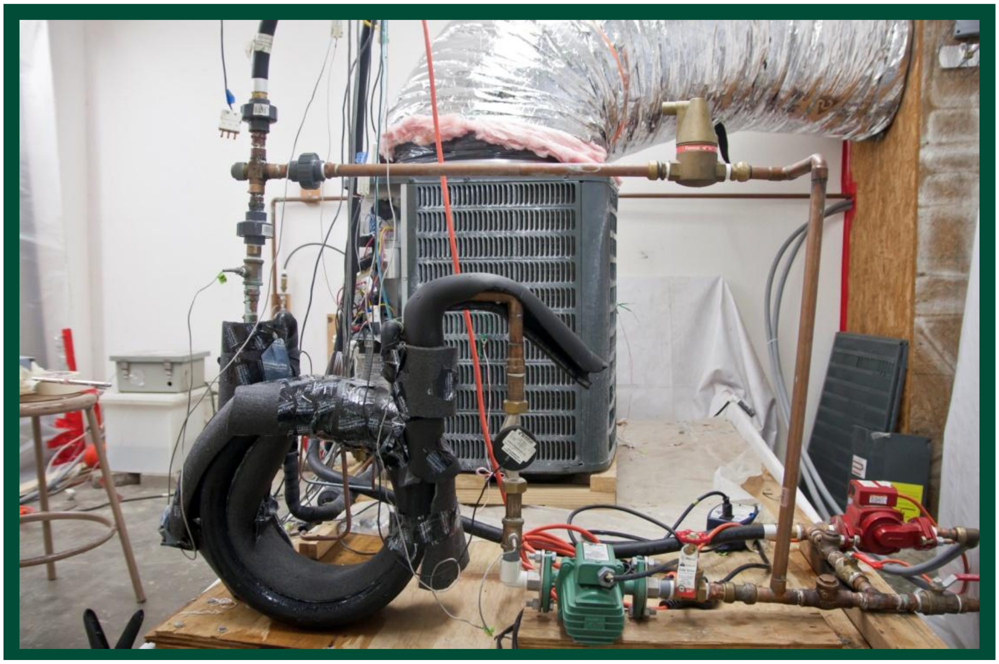
Condenser Air Evaporative Pre-Cooler Test Protocol

HT.11.SCE.019 /HT.11.SCE.021 Report



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EXECUTIVE SUMMARY

Direct evaporative condenser air pre-cooler retrofits for packaged rooftop air conditioners have significant potential to reduce peak electricity demand up to 30% in hot and dry climates. These systems work by pre-cooling the condenser air stream by evaporating water. Theoretically, air can be cooled to its wet bulb temperature. Systems are being introduced to the market that brings the advantages of this evaporative cooling process to small-scale air-cooled air conditioning systems. These systems are designed to be retrofitted onto existing units.

While there are established test standards for rating the performance of air-cooled air conditioning systems, a test standard to objectively compare the performance of condenser air evaporative pre-coolers does not exist. The objectives of this study were:

1. Develop an initial test protocol for condenser air evaporative pre-coolers. This protocol was intended to be used to rate performance of diverse systems under several climate conditions.

2. Evaluate this test protocol by testing pre-cooler designs from three manufacturers.

3. Revise the protocol, based upon experience gained from the tests.

4. Develop an analysis tool to determine the expected energy savings of pre-coolers based on the output of laboratory test data.

To conduct these laboratory assessments, a conditioned chamber and test apparatus were constructed and three pre-coolers were tested at four climate conditions. Two methods of measuring the evaporative effectiveness of each product were compared. An in-depth uncertainty analysis was completed to assist in the evaluation of each method and to characterize the uncertainty of the results. An analysis tool was developed that combined test results with 1) end-use electricity load profiles from the California Commercial End-Use Survey, 2) roof top unit energy consumption profiles, and 3) weather data for Southern California Climate Zones to provide demand and energy savings estimates for pre-coolers.

A typical measurement used in the evaluation of pre-coolers is evaporative effectiveness, which is defined as the percent of the theoretical maximum of pre-cooling that is achieved. For example, if the dry bulb temperature is 100°F, and the wet bulb temperature is 70°F, a pre-cooler that is 50% effective would pre-cool air to 85°F. Measuring the pre-cooler outlet temperatures of evaporative pre-coolers was difficult during actual tests. Water caused the temperature sensors to give inaccurate measurements and air leaving the pre-cooler apparatus was poorly mixed. A possible workaround involved measuring the temperature and relative humidity of the condenser exhaust and pre-cooler inlet air and using psychometric calculations to back out the air temperature at the condenser inlet. This assumed that the absolute humidity ratio was constant between the condenser inlet and the exhaust and that the wet bulb temperature was constant as the air passed through the pre-cooler. These assumptions may not be true if either 1) water evaporates on the condensing coil, or 2) water droplets pass through the condensing coil and evaporate later in the air stream.

To compensate for the deficiencies of using the exhaust air measurements to calculate evaporative effectiveness, another method for calculating the equivalent evaporative effectiveness was developed and included in the test protocol. This method assumed that the performance of the unit (power, capacity, or COP) was 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 was lowered. For example, the condensing unit operates 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 before entering the condenser.

The equivalent dry bulb temperature seen by the condenser with the pre-cooler installed was calculated by using the baseline condenser data with no pre-cooler installed. This method was used to determine the equivalent evaporative effectiveness of the pre-cooler at each test point.

The project developed a pre-cooler test protocol, vetted that protocol using three off-the-shelf-pre-coolers, and developed a new method to analyze the evaporative effectiveness of these pre-coolers that measures the actual power and efficiency changes. For the three pre-coolers tested, the evaporative effectiveness was found to be between 0-40% and the results differed between to the methods. These differences were greater for pre-cooler designs in which water evaporated directly on the condenser coil. While the pre-coolers tested were not particularly effective, they are not to be considered representative of the pre-cooler market, as field test studies have demonstrated 80% effectiveness in other products. An analysis of the uncertainty of the calculated evaporative effectiveness and water use effectiveness showed greater uncertainty than desired. Post-calibration of select sensors and re-calculation of the uncertainty should improve these results. Continued testing and validation of the performance-based methods is the most reliable way to measure the degree of which the evaporation of water is being converted to power and energy savings.

Continued development of the pre-cooler test protocol will likely require additional testing of pre-coolers. Future testing would likely occur on a small RTU with a one-sided condenser face. This would allow easier testing of commercial pre-cooler products. WCEC has identified at least three commercial products for testing, whose designs have significant differences from the residential products tested in this study. WCEC is currently constructing a laboratory to support this testing.

Results from the analysis tool showed a range, depending on climate zone, of 10-30% peak demand savings and 3-25% energy savings for an 80% effective pre-cooler. This equates to an estimated 0.1-0.33 kilowatt/ton demand savings and 30-300 kWh/yr-ton energy savings.

The results from this project and future research will be used by the recently formed ASHRAE Standards Technical Committee 5.7 to develop an ASHRAE test standard for evaporative condenser air pre-coolers.

ABBREVIATIONS AND ACRONYMS

°F	Degrees Fahrenheit
AC	Air conditioning
AHRI	Air-Conditioning, Heating, and Refrigeration Institute
ANSI	American National Standards Institute
ASHRAE	American Society of Heating, Refrigeration, and Air Conditioning Engineers
Btu	British Thermal Unit
CFM	Cubic Feet per Minute
CO_2	Carbon Dioxide
COP	Coefficient of Performance
DB	Dry Bulb
DEG	Davis Energy Group
DX	Direct Expansion
EE	Evaporative Effectiveness
gph	Gallons Per Hour
hr	Hour
HR	Humidity Ratio
kBtu/hr	1000 British Thermal Unit/hour
Ksi	1000

lb/hr	Pounds/hour
mA	Milliamp
MBH	$1.0x10^6$ Btu per hour
Min	Minutes
NI	National Instruments
OAT	Outside Air Temperature
ppm	Parts per million
psi	Pounds per square inch
R-22	Refrigerant-22
R-410A	Refrigerant-410A
RH	Relative Humidity
RTD	Resistance Temperature Device
RTU	Rooftop Unit
USB	Universal Serial Bus
V	Volts
VAC	Volts Alternating Current
VDC	Volts Direct Current
W	Watts
WB	Wet Bulb
WBD	Wet Bulb Depression
WCEC	Western Cooling Efficiency Center

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INTRODUCTION

Approximately 65% of commercial floor area is conditioned by packaged air-cooled air conditioning (AC) units[1]. These units consist of supply fans, direct expansion (DX) cooling systems, heating, and air filters. Most residential AC systems are split-system air-cooled DX units. Both types of systems are inexpensive to install and maintain. Due to their prevalence and their energy performance, air-cooled AC systems are a large part of electrical demand during periods of high outdoor temperature.

Air-cooled DX systems provide cooling by rejecting heat from the indoor conditioned space to the outdoor air. The greater the temperature difference between the roof top unit's (RTU) supply air and outdoor air temperatures, the more mechanical energy is required to remove the heat from the conditioned space. By contrast, air conditioning systems serving larger commercial applications take advantage of the dry climate of the western United States by evaporating water to reject heat into the atmosphere. This evaporative process reduces both energy consumption and peak electrical demand by lowering the DX system's condensing temperature.

Systems are being introduced to the market that bring the advantages of this evaporative cooling process to small scale air-cooled AC systems. These systems are designed to be retrofitted onto existing units. They operate by evaporating water in the condenser air stream, cooling the incoming condenser air.

While there are established test standards for rating the performance of air-cooled AC systems, a test standard to objectively compare the performance of condenser air evaporative pre-coolers doesn't exist. The purpose of this study is to develop a test protocol for these systems, and to evaluate this protocol in a laboratory setting.

BACKGROUND

Commercial and residential air conditioning systems use mechanical energy to remove heat from the conditioned space, rejecting this heat to a higher temperature heat sink such as the outdoor environment. These systems utilize a direct expansion (DX) cycle that works in the following manner:

1. Low-pressure refrigerant boils in the evaporator, removing heat from the airstream. This colder air is then delivered to the conditioned space.
2. The compressor, driven by an electric motor or other means, raises the pressure of the refrigerant gas.
3. In the condenser, heat is transferred from the refrigerant gas to the heat sink, which can be the outdoor air, a water stream, or evaporating water.
4. The thermostatic expansion valve (TxV) reduces the pressure of the refrigerant liquid, repeating the cycle.

Figure 1 illustrates the steps of the DX refrigeration cycle with references to the above steps.

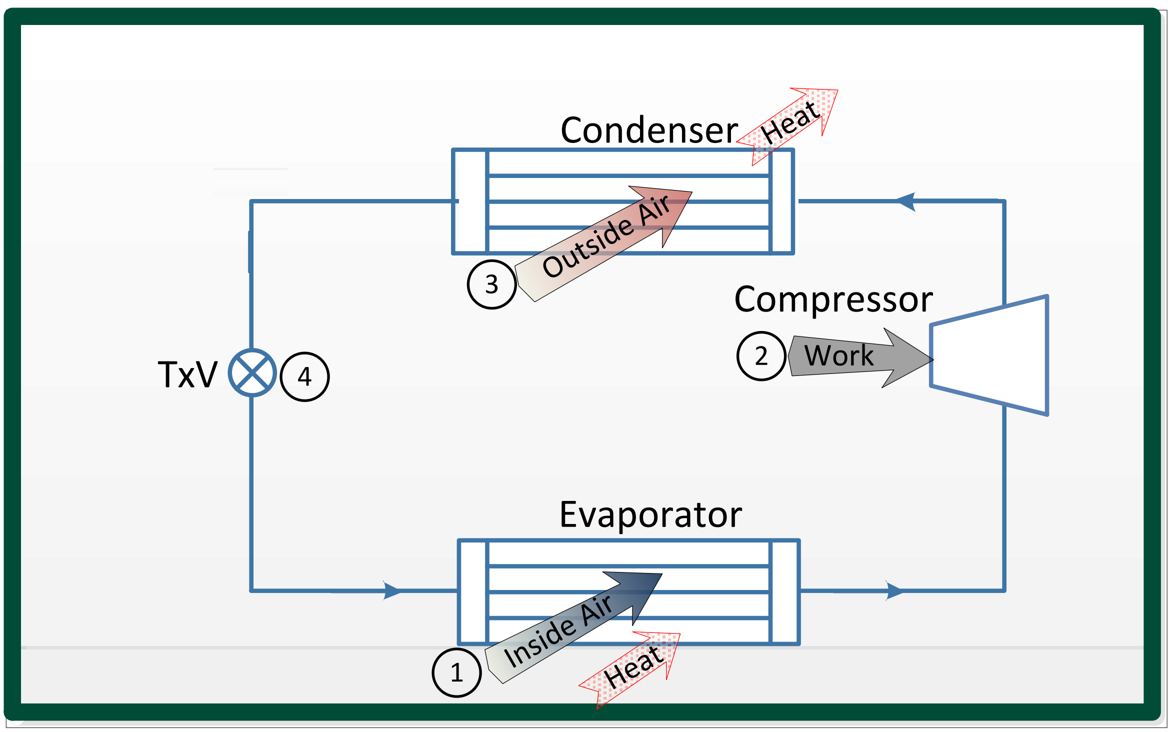

FIGURE 1. DIRECT EXPANSION REFRIGERANT CYCLE

The amount of energy required by the compressor in Step 2 is dependent upon the temperature at which the refrigerant gas condenses. In RTUs and air-cooled chillers, condensers reject heat from refrigerant directly into the outside air stream. In these systems, higher outside air temperatures result in higher energy usage by the compressor. As a result, as the outdoor air temperature rises, the efficiency of the air conditioning system drops quickly and that system requires more energy to provide the same amount of cooling to the conditioned space. To compound this issue, more cooling is necessary on days where the outdoor air temperature is higher.

Evaporative cooling takes advantage of the ability of the outside air to absorb moisture and the heat of vaporization of this process. As water evaporates into the airstream, it absorbs heat from the surrounding air, refrigerant, or remaining water stream. Cooling towers, evaporative condensers, and closed-circuit fluid coolers employ this process. Evaporative condensers operate at a lower temperature than air-cooled condensers, and therefore require less mechanical energy to meet the cooling demand.

In an air-cooled system, air is drawn through a condenser coil, where heat from the refrigerant is rejected into the airstream. The condenser airstream then discharges this heated air into the atmosphere. The airstream is sensibly heated to a higher temperature that is exhausted to the outside air, depicted by the navy blue process line in Figure 2. The red dot in Figure 2 represents the 1% American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) design cooling conditions for Ontario, California (98˚F Dry Bulb (DB), 70˚F Wet Bulb (WB)).

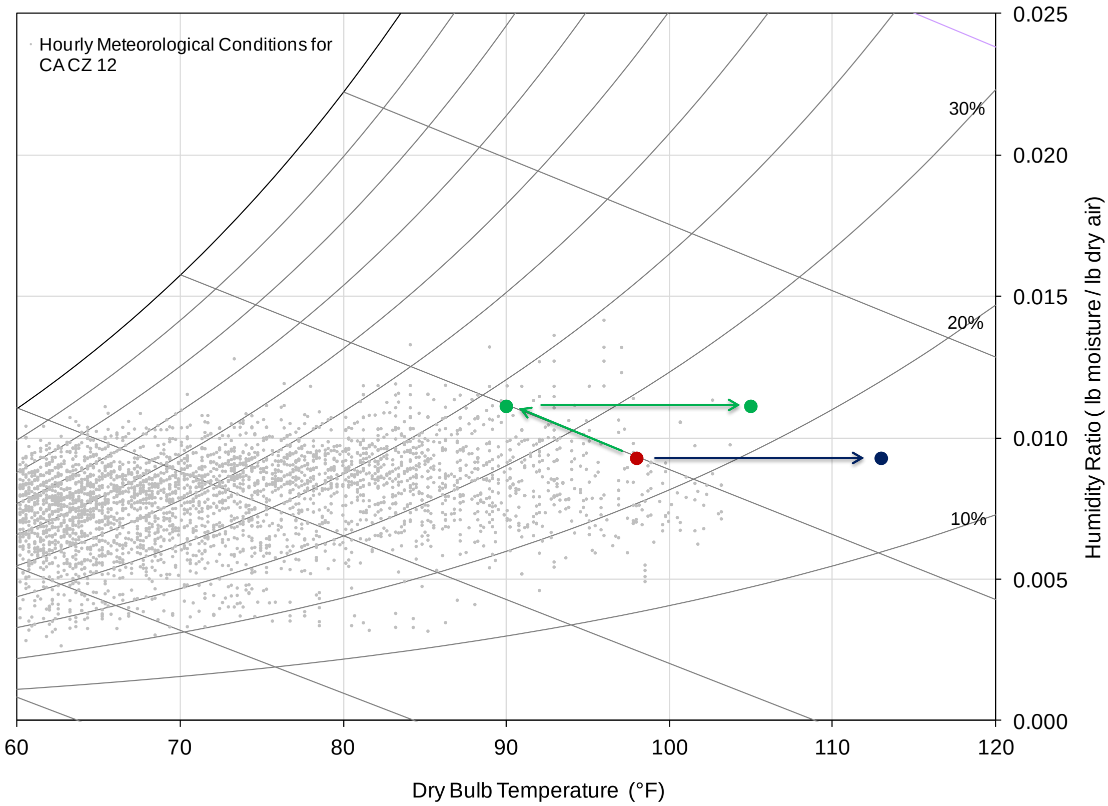

FIGURE 2. PSYCHROMETRIC DIAGRAM OF CONDENSING PROCESS WITH AND WITHOUT EVAPORATIVE PRE-COOLING

A common design for evaporative pre-coolers involves introducing water to an evaporative media surrounding the condenser. The air flows through this evaporative media before reaching the condenser (other designs evaporate water directly onto the condenser coil without using evaporative media). As the air comes in contact with the media, the water within the media evaporates and sensible heat is removed from this airstream in the process, lowering its dry bulb temperature. The air enters the condenser at a lower dry bulb temperature than the ambient temperature, and sensibly heats across the condensing unit. The green process line in Figure 2 shows this process. Lowering the condensing temperature of an air conditioning system's refrigerant affects efficiency in two ways. As stated earlier, the lower condensing pressure corresponding to a lower temperature reduces the necessary work done by the compressor. Additionally, a lower condensing pressure allows a greater proportion of the refrigerant to condense.

Evaporative condenser air pre-coolers are of special interest in dry arid climates such as that in California. Arid climate zones allow a larger amount of water to evaporate into the airstream before entering the condenser, which correlates to a higher amount of pre-cooling. There are a large numbers of manufacturers offering evaporative pre-coolers as retrofits to existing RTUs, and the methods of pre-cooling air vary. Design variables include the method of water delivery (constant spray, pulsing spray, or run down a media), whether or not the water is re-circulated, and whether or not the product has a media protecting the condenser coil from water spray. The design of the pre-cooler will impact the energy savings of the air conditioning system. An objective laboratory test protocol and associated analysis tool is needed to quantify both the energy savings and the associated water use.

ASSESSMENT OBJECTIVES

The objectives of this study were to:

- Develop an initial test protocol for condenser air evaporative pre-coolers. This protocol will be used to rate performance of diverse systems under several climate conditions.
- Evaluate test protocol by testing pre-cooler designs from three manufacturers.
- Make revisions to the protocol, based upon experience gained from the tests.
- Develop an analysis tool to determine the expected energy savings of pre-coolers based on the output of laboratory test data.

TECHNOLOGY/PRODUCT EVALUATION

An experimental protocol was developed to test condenser air evaporative pre-cooler products. While the protocol was designed for retrofit products for RTUs up to 20 tons, the protocol was tested using 3-ton equipment due to the laboratory size and test equipment available. Three commercially available pre-cooler retrofit products were tested using the protocol. An overview and comparison of these three products is shown in Table 1.

TABLE 1. PRE-COOLER PRODUCTS

SPECIFICATION	PRE-COOLER A	PRE-COOLER B	PRE-COOLER C
Size Range of Condenser	--	1 to 5 tons	1 to 5 tons
Water Pressure	~ 60 pounds per square inch (psi)	City Water	30-120 psi
Electrical Requirements	230 Volts Alternating Current (VAC)	3 AAA or 24v from thermostat	None
Water Consumption	~4 gallons per hour (gph)/nozzle continuous (pulses based on outdoor air temperature). 1-2 nozzles for 3-ton condenser	0.8 gph/nozzle (continuous usage). 4 nozzles for 3-ton condenser	0.4 gph/nozzle (continuous usage) 3 nozzles for 3-ton condenser
Water Filtration	None	Fiberglass filters placed around unit that collects minerals and sediment before it reaches the coil	3 or 6 month water filtration cartridges available with polyacralate chemical designed to inhibit ability of minerals to adhere to metal surfaces
Duty Cycle Control	Measures temperature; duty cycle is a function of temperature	Measures temperature; operates continuously when temperature is above a programmed threshold	Activates via mechanical flapper valve when condensing unit exhaust is on; always on if condensing unit is on

The three evaporative pre-cooler models investigated in this experiment were designed to be installed on a wide range of residential and light commercial units; they were not custom designed. Each model can be purchased as a package and installed on an existing air conditioning unit. Each unit was tested in a laboratory environment within a conditioned chamber developed by Western Cooling Efficiency Center (WCEC) in order to control environmental conditions and provide a platform where the pre-cooling products can be directly compared.

PRE-COOLER A

This product is an evaporative condenser air pre-cooler that uses nozzles that spray directly onto the condenser coils. It was initially designed for split-systems where the condensing unit has a single side and is typically not applied to box-style condensing units where an exhaust fan is located at the top, pulling air from four sides of a heat exchanger. The top-mounted spray nozzle has a controller that energizes a solenoid valve with a duty cycle that is determined according to the temperature of the outdoor air; the higher the temperature, the more often the water turns on. A 230VAC power source is required to power the solenoid valve. When the valve is open, the system draws approximately 5W of power. The pre-cooler works with water pressures typical of city water and does not incorporate any water filtration or treatment.

PRE-COOLER B

Pre-Cooler B is an evaporative pre-cooler designed for residential and light commercial applications. It is typically installed on 3-5 ton condensing units. This design can be used with condensing units as small as 1-2 tons; however the cost of the system is relatively constant. Pre-cooler B has also been installed commercially on a few rooftop units ranging in size from 4 to 7.5 tons. The design of the pre-cooler utilizes a fiberglass filter media that is wrapped around a small condensing unit. Nozzles create a mist of water droplets surrounding the condensing unit. The product claims to evaporate the mist and cool the air entering the condenser coil. The unit is designed to shut off when the outside air is too cold for condenser air pre-cooling to be of benefit and this threshold is adjustable. In addition, Pre-Cooler B is designed with varying nozzle quantities according to the climate and condensing unit size.

The unit can be powered using either three AAA batteries, which typically last one cooling season, or by the preferred method of connecting the pre-cooling unit to the 24V control power from the house thermostat. According to the manufacturer, no water treatment is necessary since the fiberglass filters placed around the condensing unit will catch the minerals and sediment deposited from the water. As a result of not treating the water, there are some maintenance needs. The nozzles need to be brushed lightly in order to keep them clean from minerals. Calcium, lime, and rust clog nozzles. These nozzles are easily removed and replaced. The nozzles can also be soaked in vinegar or another cleaning solution, which can be purchased at hardware stores. The fiberglass filters need to be replaced once or twice a year.

PRE-COOLER C

Pre-cooler C is a condenser air evaporative pre-cooler that can be used with condensing units ranging in size from 1 to 5 tons. It utilizes a mechanical vane valve placed on top of the condensing unit. When the condenser is exhausting air, this valve opens, allowing water from a hose to flow to the misting nozzles. The design is therefore always running when the condensing unit is running and off when the condensing unit is off. No additional electricity source is needed for this unit and the added resistance of the valve on the exit of the condensing unit is expected to be negligible. The water provided to the nozzles can be at a pressure range of 30 to 100 psi. The water is treated with a 90 day or 6 month sized filtration cartridge that utilizes a polyacralate chemical designed to inhibit the ability for mineral scale to adhere to metal surfaces. A standard unit comes with three misting nozzles, but an expansion is available that allows for a total of five nozzles in the system. Each nozzle is advertised to use a maximum of 0.4 gallons per hour of water in continuous use.

TECHNICAL APPROACH/TEST METHODOLOGY

WCEC built an interim laboratory in Davis, California while construction of a permanent laboratory was underway. The permanent facility was not expected to be completed in time for the deadlines required for this test program. The interim laboratory was designed to accommodate evaporative pre-cooling tests on condensing units up to 3 tons capacity and that draw up to 3000 cubic feet per minute (CFM) condenser air. The test chamber operated on a closed-loop air circuit, incorporating 18 tons cooling and dehumidification capacity, 180 MBH heating, and 60 lb/hr humidification capacity. Variable control of temperature and humidity level was provided. A Goodman GSC13-0361 3-ton condensing unit with refrigerant 22 (R-22) was placed inside the conditioned chamber and used for all pre-cooler tests (Figure 3).

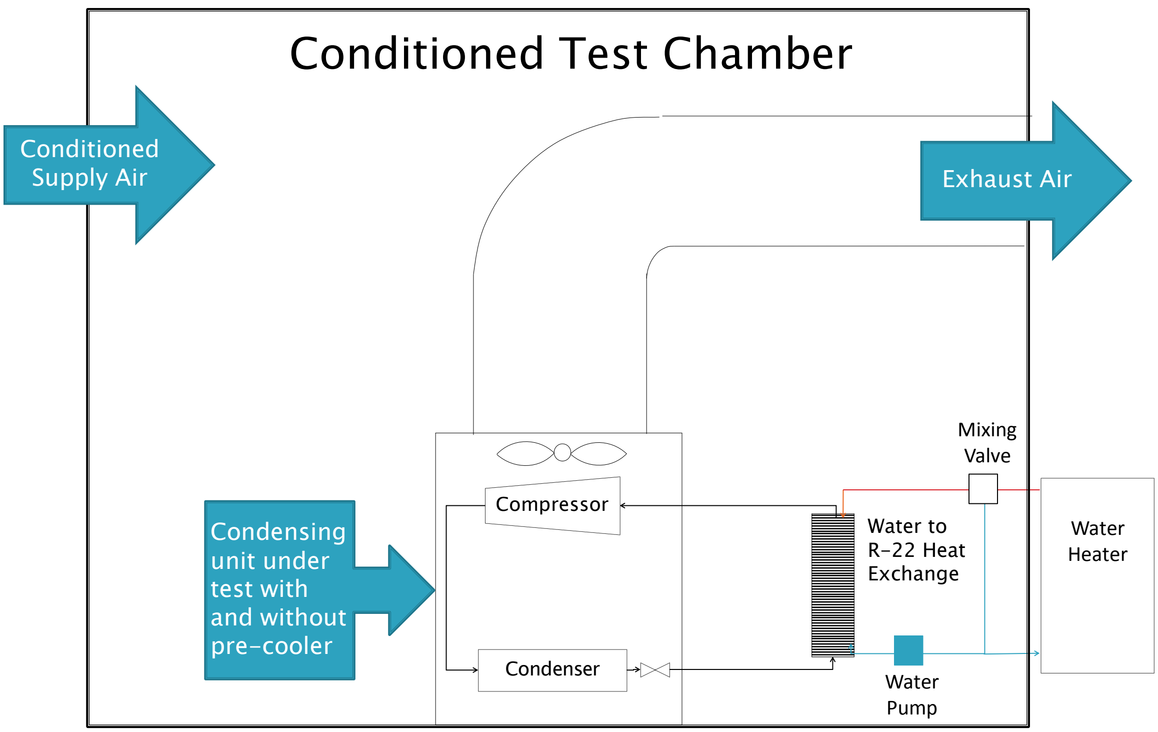

FIGURE 3. BASIC DIAGRAM OF TEST EQUIPMENT ARRANGED INSIDE CONDITIONED TEST CHAMBER.

The chamber was designed to test a baseline condenser at ambient conditions between 64-105°F DB. The same condensing unit, retrofitted with pre-coolers, was tested at four ambient conditions, shown in Table 2. The objective of the test was to maintain the measured chamber dry bulb temperature within ±2°F and the chamber wet bulb temperature within ±1°F over a one-hour test period. In addition, the average of the dry bulb temperature was required to be within ±0.5°F of the target, and the average of the wet bulb temperature was required to be within ±0.3°F of the target. The data were sampled each second, averaged over 30 seconds, and logged.

TABLE 2. EVAPORATIVE PRE-COOLING TEST CONDITIONS

TEST CONDITION	AMBIENT TEMPERATURES (°F DB/°F WB)	CLIMATE CONDITION
A	105/73	Hot Dry
B	95/75	Hot Humid
C	90/64	Warm Dry
D	82/73	Warm Humid

CONDITIONING PROCESS

Testing of the baseline-condensing unit required testing seven ambient dry bulb conditions for which the low was 64°F and the high was 105°F. Testing of each pre-cooler required testing four climate conditions where both dry bulb and wet bulb temperatures were controlled. These conditions are listed in Table 2 (row 1 and 2).

Figure 4 illustrates the test chamber designed at the interim laboratory. Letters in green circles refer to conditioning process sub-section headers. Numbers in orange circles refer to instrumentation and controls listed in Table 3. The laboratory was designed using existing conditioning equipment under the following assumptions:

1. Maximum condensing unit capacity was 3 tons with a Coefficient of Performance (COP) of 3 at rated conditions, for maximum heat rejection of 4 tons to chamber air stream.
2. Maximum condensing unit airflow was 3,000 cfm (typical condenser airflows are 800 cfm/ton)[2]
3. The pre-cooler operated with a maximum evaporative effectiveness of 80%.
4. The system re-circulated 100% of the air.

The conditioned air entered the test chamber at (Figure 4, A), where the condensing unit added 3-4 tons of heat, increasing the temperature of the air. When a pre-cooler was operational, the air also increased in humidity and decreased in temperature. The chamber exhaust air stream (Figure 4,B) entered a small split system for cooling (nominally 2.5 tons), and then a Hypak Rooftop Unit (RTU) for cooling (Figure 4,C), with a nominal capacity of either 7.8 or 15.7 tons of cooling, depending on the number or compressors operating (1 or 2). The number of compressors was controlled by manually switching relays to ensure the absolute humidify the air exiting the cooling system was lower than the final chamber target. The air was then re-heated with a hot water coil (also located inside the Hypak RTU, (Figure 4, D)). The hot water flow rate was electronically controlled with a variable speed pump so that the hot water temperature was approximately five degrees below the final chamber target temperature. The air was then humidified with an evaporative humidifier system. The fraction of air going through the humidifier and the remaining fraction going through a bypass was electronically controlled by two actuated dampers to achieve the target absolute humidity (Figure 4, E). The air was finally re-heated to the target dry bulb temperature by passing through a final hot water coil (Figure 4, F) for which the water temperature was controlled with an electronically controlled mixing valve. The flow rate of the air was controlled by the variable speed drive fan in the Hypak and an electronically-controlled actuated damper prior to the chamber entrance.

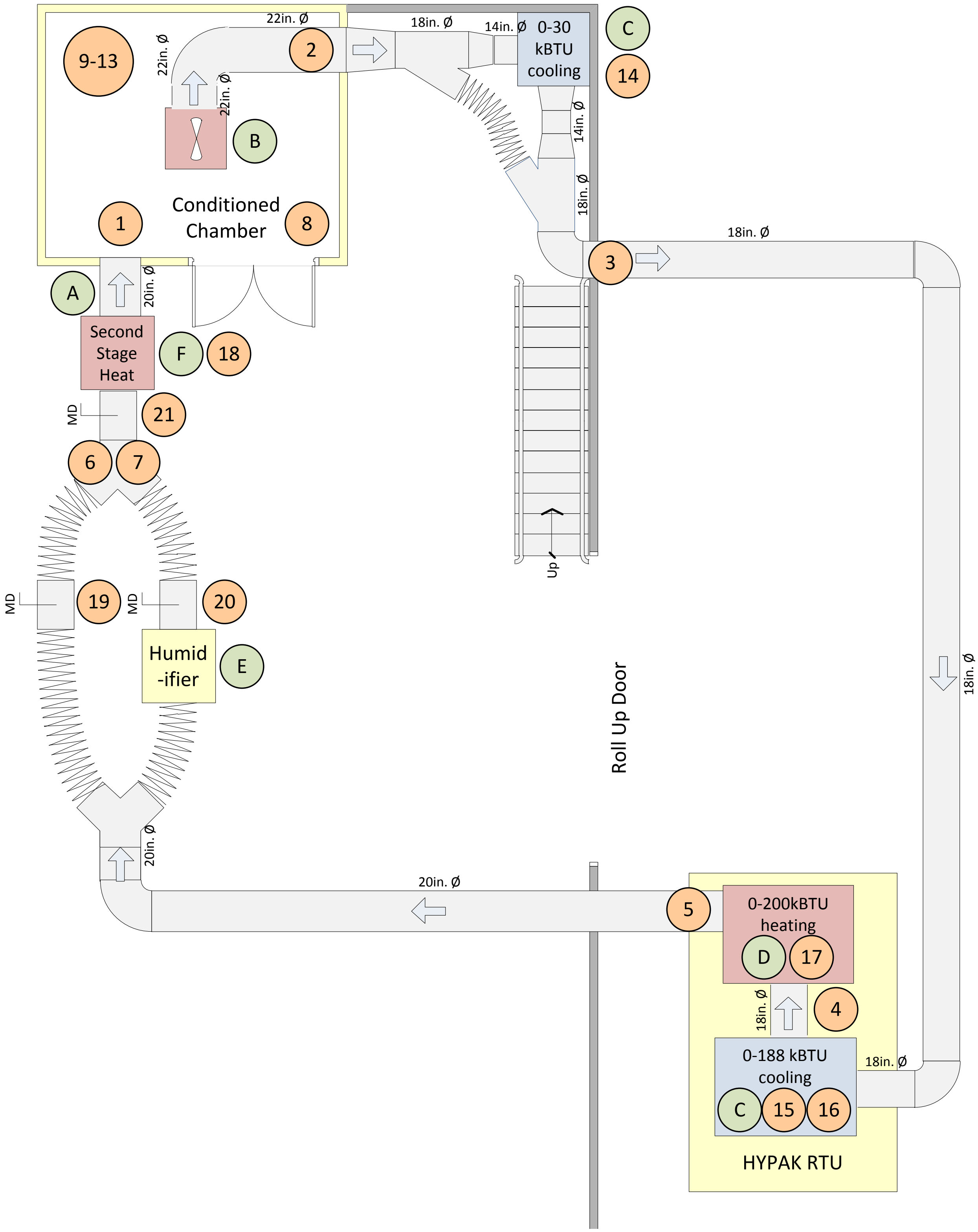

FIGURE 4. SCHEMATIC OF TEST LABORATORY

TABLE 3. CONTROLS INSTRUMENTATION TABLE

ITEM #	MEASUREMENT TYPE	MANUFACTURER MODEL #	ACCURACY	INPUT	SIGNAL	DAQ CHANNEL
1	Chamber Air Temp	GE Optisonde	±0.27°F	100-240 VAC	RS-232	Serial
	Chamber Dew Point Temp		±0.36°F		RS-232	Serial
2	Post Chamber Air Temp	Vaisala	0.1°F	10-35 VDC	4-20mA	NI 9203
	Post Chamber RH		2% RH		4-20mA	NI 9203
3	Post Cooling Air Temp 1	Vaisala	0.1°F	10-35 VDC	4-20mA	NI 9203
	Post Cooling RH 1		2% RH		4-20mA	NI 9203
4	Post Cooling Air Temp 2	Vaisala	0.1°F	10-35 VDC	4-20mA	NI 9203
	Post Cooling RH 2		2% RH		4-20mA	NI 9203
5	Post Heating Temp	Omega RTD-806	0.4°F	RTD Excit	RTD	NI 9217
6	Post Humidifier Temp	Vaisala	0.1°F	10-35 VDC	4-20mA	NI 9203
	Post Humidifier RH		2% RH		4-20mA	NI 9203
7	(a) Delta P Humidifier	Energy Conservatory DG-500	1% of reading	120 VAC	Serial	Serial Port 1
	(b) Delta P - Extra					
8	(a) Delta P Static (Condenser)	Energy Conservatory DG-500	1% of reading	120 VAC	Serial	Serial Port 2
	(b) Delta P Chamber					
9	Air Temp RTD 1	Omega RTD-806	0.4°F	RTD Excit	RTD	NI 9217
10	Air Temp RTD 2	Omega RTD-806	0.4°F	RTD Excit	RTD	NI 9217
11	Air Temp RTD 3	Omega RTD-806	0.4°F	RTD Excit	RTD	NI 9217
12	Air Temp RTD 4	Omega RTD-806	0.4°F	RTD Excit	RTD	NI 9217
13	Air Temp RTD 5	Omega RTD-806	0.4°F	RTD Excit	RTD	NI 9217
14	Relay - 2.5 ton Cooling				On/Off	NI PCI-6321
15	Relay - 7.8 ton Cooling				On/Off	NI PCI-6321
16	Relay - 7.8 ton Cooling				On/Off	NI PCI-6321
17	Pump Control				0-10 VDC	NI 9264
18	Mixing Valve Control				0-10 VDC	NI 9264
19	Damper 1 Control				0-10 VDC	NI 9264
20	Damper 2 Control				0-10 VDC	NI 9264
21	Damper 3 Control				0-10 VDC	NI 9264

TEST PLAN

Before testing any evaporative condenser air pre-coolers, a set of seven baseline tests were obtained for the Goodman GSC13-0361* condensing unit without installing additional equipment. These baseline points are called the "no-cooler" tests and are listed in Table 4. This baseline data was used to provide a comparable basis for all pre-cooler apparatus tests.

TABLE 4. COOLING EQUIPMENT WITH NO PRE-COOLER INSTALLED (BASELINE)

TEST	T_{OUT} (°F)
1A	105
1B	95
1C	90
1D	82
1E	75
1F	73
1G	64

Some evaporative condenser air pre-coolers have evaporative media in front of the condenser coil or other airflow obstruction. For these types of pre-coolers, additional data were taken after the evaporative pre-cooler equipment was installed but before the water was turned on. This set of tests is called the "dry-cooler" tests and they are shown in Table 5. This data describes performance of the condensing unit when the pre-cooler is installed but not operational.

TABLE 5. COOLING EQUIPMENT WITH DRY EVAPORATIVE PRE-COOLER INSTALLED (DRY COOLER)

TEST	T_{OUT} (°F)
2A	105
2B	95
2C	90
2D	82
2E	75
2F	73
2G	64

For the measurement of the air conditioning system performance change due to the evaporative condenser air pre-cooler, the following tests were conducted with the evaporative pre-cooler installed and running. These tests are shown in Table 6.

TABLE 6. COOLING EQUIPMENT WITH WET EVAPORATIVE PRE-COOLER INSTALLED (WET COOLER)

TEST	$T_{DB,OUT}$ (°F)	$T_{WB,OUT}$ (°F)
3A	105	73
3B	95	75
3C	90	64
3D	82	73

INSTRUMENTATION PLAN

The test apparatus and measurements taken during the experiment are illustrated in Figure 5. The measurements are color coded:

- Red sensors measure flow-rate,
- Green sensors measure temperature,
- Yellow sensors measure pressure,
- Blue sensors measure power.

Detailed instrumentation specifications are listed in Table 7 and a photograph of the experimental apparatus within the conditioned chamber is shown in FFigure 6.

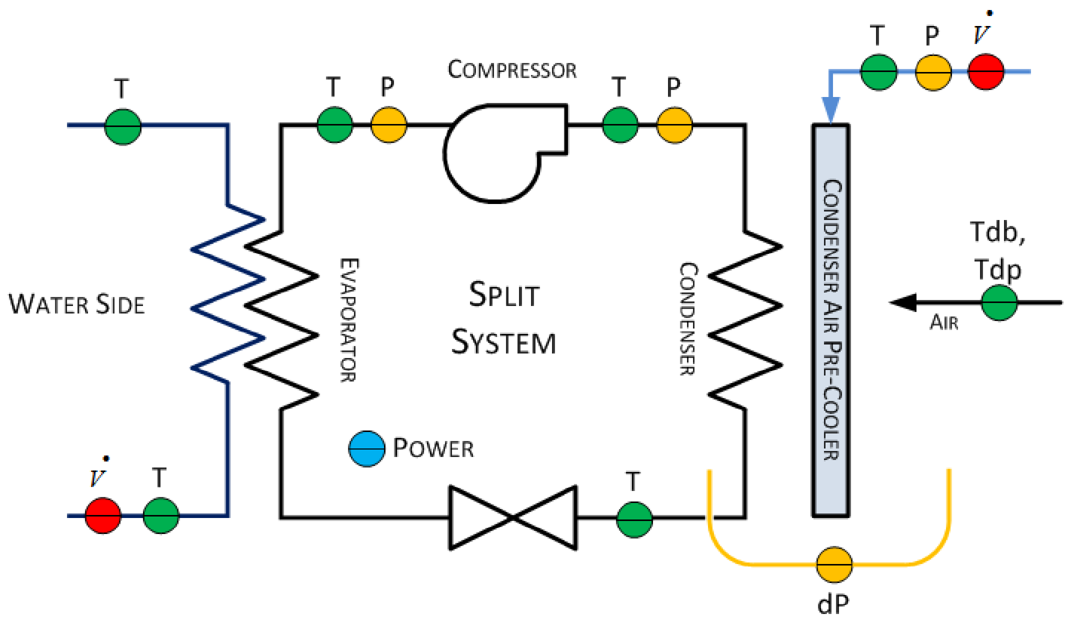

FIGURE 5. MEASUREMENTS FOR PRE-COOLER TESTING APPARATUS

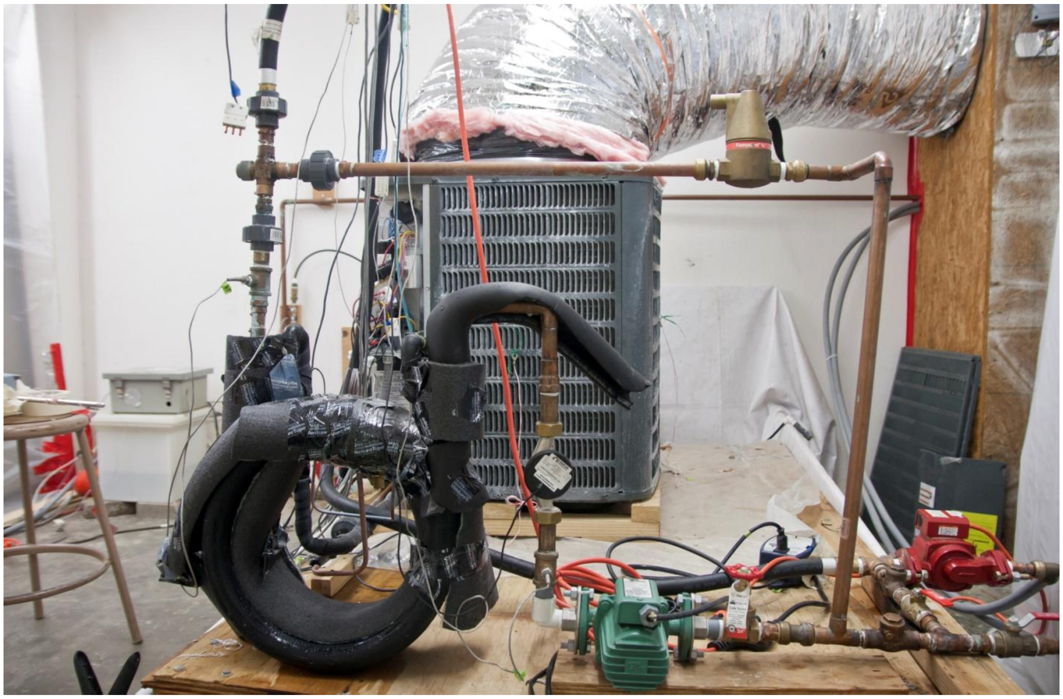

FIGURE 6. EXPERIMENTAL APPARATUS WITHIN CONDITIONED CHAMBER WITH EXHAUST DUCTED OUT OF THE ROOM

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 non-potable well water; therefore the water was run through a reverse osmosis and filtration system before use, as this set of experiments did not attempt to quantify the effects of hard water on pre-coolers. The pre-cooler water, stored in a five gallon buffer tank, was controlled to be 85±2.5 °F as measured by a resistance temperature device (RTD) submerged in the tank. 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 55±10 psi (gauge) that is consistent with typical municipal service water pressure. The flow rate of the water was measured, but not controlled, as flow 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. The flow meter for the pre-cooler water flow measurement was selected based on the pre-cooler design's water flow rate to provide the required accuracy over the expected range; one meter was used for the higher flow rate in pre-cooler A and a second meter was used for the lower flow rate in pre-coolers B and C.

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 measured to ensure proper refrigerant charge; 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. Thermal paste was applied between the RTD and the pipe.

EVAPORATOR MEASUREMENTS

The evaporative load was supplied to the condensing unit with a water-to-refrigerant heat exchanger, where the water temperature and flow rate were controlled to provide a load equivalent to an evaporator cooling incoming air at 80/67 (DB°F/WB°F). The normal operation of the condensing unit was determined from the manufacturer cooling data that corresponds to Test A rating conditions per the Air-Conditioning, Heating, and Refrigeration Institute (AHRI 210/240) standards. The temperature of the water exiting the mixing valve was set manually to 80°F. The target for the supply water temperature was set to be equivalent to the supply air temperature in AHRI tests, and the water flow rate required to achieve this temperature was calculated from the rated capacity. The water flow rate was set to this target by manually adjusting a needle valve downstream of the constant speed pump. The water temperatures before and after the heat exchanger were measured using RTDs inserted directly into the water flow using threaded fittings. The water flow meter was 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.

DIFFERENTIAL PRESSURE AND AIRFLOW MEASUREMENTS

Differential pressures across the condenser fan, between the conditioned chamber and the warehouse, and across the condensing coil and pre-cooler combined (when pre-cooler introduced condensing coil resistance) were measured using an Energy Conservatory APT-8 pressure transducer with 8 differential pressure channels. For the baseline condenser and for pre-cooler B, that affected condenser airflow, the pressure drop across the condenser coil was determined in free-air conditions before the exhaust ducting was installed. To ensure that the airflow through the condensing unit during testing matched the airflow of the unit without obstructions, the pressure drop across the condenser coil was kept at the value measured in the free-air condition by adjusting the flow into and out of the chamber using dampers and a secondary fan placed in the exhaust ducting.

With the exhaust ducting connected, the actual airflow rate was measured using a custom-built tracer gas airflow measurement system. During normal chamber operation at 95°F, industrial grade CO_2 was injected at a known mass flow rate using an Alicat scientific MC series 250 slpm mass flow controller (accuracy ±0.4% of reading +0.2% of full scale) into the entrance of the test chamber. The CO_2 concentration was then measured at the exit of the chamber using a PP Systems EGM4 CO_2 gas analyzer calibrated to 5,000 ppm range and with an accuracy of 50ppm. The flow rate of air was then calculated from the resulting concentration of CO_2 and the known injection rate, after subtracting out the baseline environmental CO_2. The uncertainty of the airflow measurement was 2.3%.

CHAMBER CONDITIONS MEASUREMENTS

During baseline tests, four resistive temperature devices (RTD) were placed around the condensing unit, one on each side. These RTDs were used to verify that the chamber had a uniform temperature and that there was no temperature gradient across the room. These RTDs could not be used while the pre-cooler was running because the sensors would become wet and report incorrect values. They were only used for the baseline tests with no pre-cooler installed. During the testing, it was observed that if the pressure in the chamber was slightly higher than the pressure in the residing warehouse, the temperature was very uniform in the chamber. If the pressure in the chamber was lower than that of the warehouse, the temperature distribution could vary by a few degrees throughout the room. This was due to the leaks trending outwards when the chamber was pressurized. Therefore, the chamber pressure was kept slightly higher (~5×10^{-4} psi) than the pressure in the warehouse, and the temperature distribution in the chamber was assumed to be uniform. A 30-minute test showed less than a 1°F variation among all sides of the condensing unit (Figure 7).

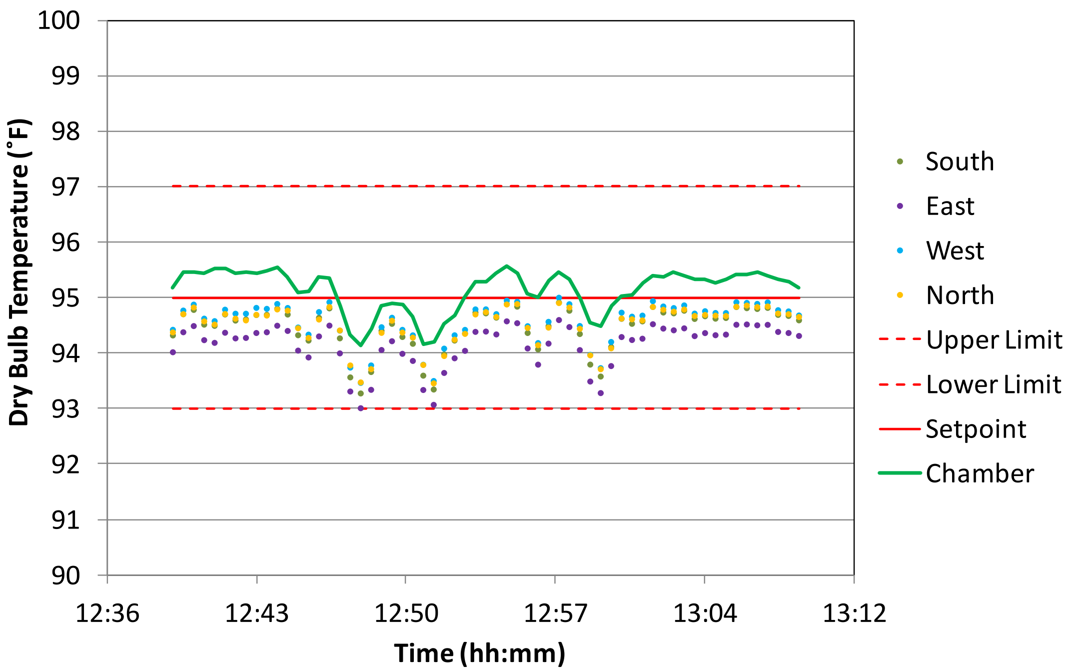

FIGURE 7. TEMPERATURE DISTRIBUTION IN TEST CHAMBER

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 output data every three seconds.

DATA ACQUISITION SYSTEM

All signals were acquired using National Instruments (NI) hardware at 0.3 Hz or greater, averaged over every 30 seconds using LabVIEW software, and logged to a text file. All RTDs were calibrated using an ice bath and boiling water within LabVIEW.

TABLE 7. INSTRUMENTATION FOR EXPERIMENTS

ITEM #	MEASUREMENT	MODEL INFO	ACCURACY	INPUT REQ	SIGNAL	DAQ CHANNEL
1	Total Power	PowerScout 18	1 % of reading	120VAC	RS-485	Serial Port #3
2	Compressor Power					
3	Fan Power					
4	Ref. Pressure: Compressor Inlet	ClimaCheck 200 105	1% of full scale	8-28 VDC	4-20 mA	NI 9203
5	Ref. Pressure: Compressor Outlet	ClimaCheck 200 105	1% of full scale	8-28 VDC	4-20 mA	NI 9203
6	Ref. Temp: Compressor Inlet	Omega PR-20	0.4°F	RTD Excit.	RTD	NI 9217
7	Ref. Temp: Compressor Outlet	Omega PR-20	0.4°F	RTD Excit.	RTD	NI 9217
8	Ref. Temp: Condenser Outlet	Omega PR-20	0.4°F	RTD Excit.	RTD	NI 9217
9	Evap. Load: Supply Water Temp	Omega NPT-72-E	0.4°F	RTD Excit.	RTD	NI 9217
10	Evap. Load: Return Water Temp	Omega NPT-72-E	0.4°F	RTD Excit.	RTD	NI 9217
11	Evap. Load: Water Flow rate	Omega FTB4607	2% of reading	6-16 VDC	Pulse	NI PCI-6321
12a	Pre-Cooler Water Flow rate	Omega FTB4705	1% full scale	5-30 VDC	Pulse	NI PCI-6321
12b		Omega FTB601	3% of reading	5-18 VDC	Pulse	NI PCI-6321
13	Pre-Cooler Water Temp	Omega PR-20	0.4°F	RTD Excit.	RTD	NI 9217
14	Pre-Cooler Water Pressure	Omega PX209	0.25% of reading	7-35 VDC	4-20 mA	NI 9203
15	Chamber Dry Bulb Temp	Vaisala HDM60Y	±0.9°F		4-20 mA	NI 9203
16	Chamber Dew Point Temp	GE Optisonde	±0.36°F	RS-232	Serial	Serial Port #4

TOLERANCES

The goal for all tests was to adhere to the relevant tolerances specified in American National Standards Institute (ANSI)/AHRI Standard 210/240-2008, ANSI/AHRI Standard 340/360-2007, and ASHRAE 37-2009. Tolerances for the outdoor dry bulb and wet bulb tolerances are specified in these standards and upheld during these tests. Tolerances for the indoor dry bulb and wet bulb temperature are also specified, but since a water-refrigerant heat exchanger was being used for the evaporative load, a new set of tolerances was developed. The water inlet temperature was set to 80ºF to match the inlet air temperature for the evaporator under Test A rating conditions per AHRI 210/240. Test A rating conditions are 95ºF dry bulb inlet condenser air temperature and 80/67ºF (Dry Bulb/Wet Bulb) inlet evaporator air temperature. 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 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.	±2°F	±0.5°F
Wet Bulb Temp.	±1°F	±0.3°F
Evaporative Load Water Temp.	±2°F	±0.5°F
Evaporative Load Water Flow rate	±7% of setpoint	±3% of setpoint
Pre-Cooler Water Temp.	±2.5°F	±1°F
Pre-Cooler Water Pressure	55±10 psi	
Condenser Coil Pressure Drop	±7% of setpoint	
Room RTD Average (baseline only)	±1°F variation	
Vaisala to RTD Average (baseline only)	±1°F variation	

No tolerances were specified for an experimental apparatus using a water-to-refrigerant heat exchanger for the evaporative load, so the tolerance was set by WCEC. The tolerances for the evaporative load water inlet temperature were kept the same as those specified for indoor air temperatures specified in Standards 210/240-2008 and 340/360-2007. The range tolerance for the evaporative water flow rate was set to ±7% and the mean tolerance was set at ±3%. The pre-cooler water temperature was maintained at 85±2.5°F, trying to stay as close in adherence to ASHRAE 37-2009 as possible.

In order to operate the condensing unit inside the conditioned chamber, external ducting and fans were 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 was conducted to determine the sensitivity of the condensing unit performance with respect to changes in the pressure drop across the condenser coil. The condensing unit was tested for 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, 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 were 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.

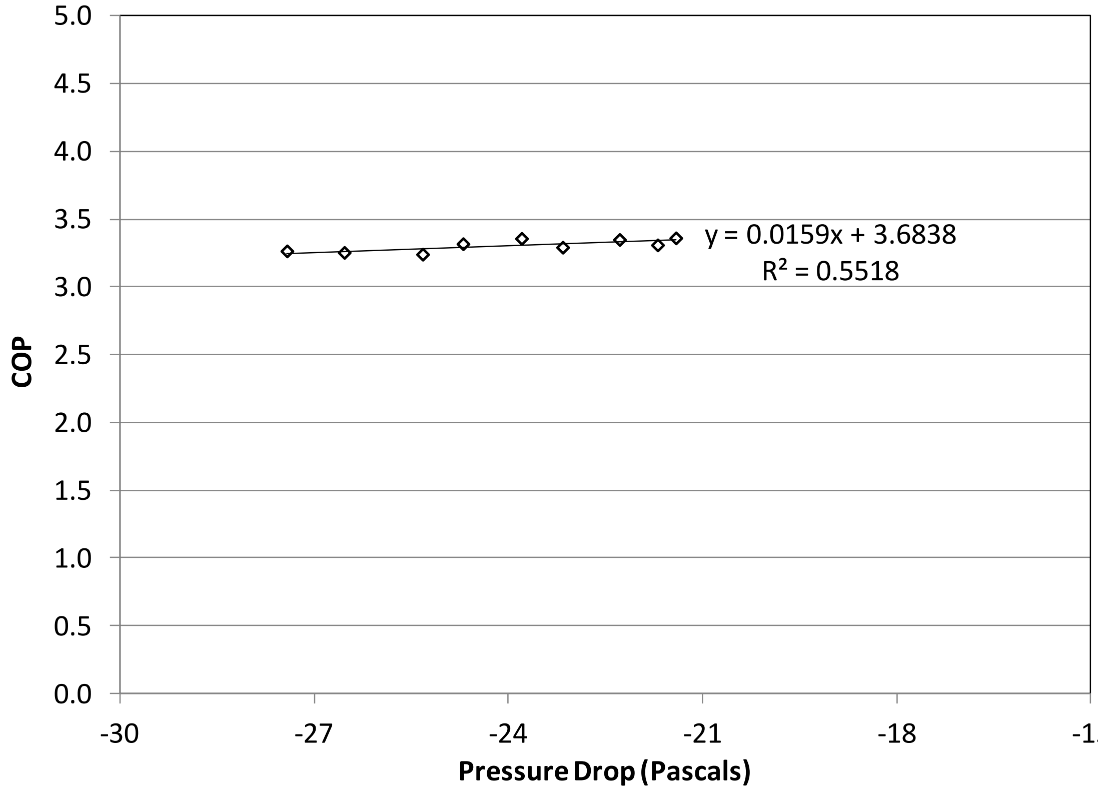

FIGURE 8. SENSITIVITY ANALYSIS OF CONDENSER COIL PRESSURE DROP TOLERANCES

RESULTS

EXPERIMENTAL CALCULATIONS

POWER

The refrigerant line was accidentally drilled into when installing pre-cooler A, which was the second pre-cooler tested. The condensing unit had to be evacuated, repaired, and re-charged. While the refrigerant was charged as closely as possible to the original charge, a new baseline curve was obtained to ensure consistent results. The power measurements for the two baselines were compared and found to be very close, with the largest discrepancy between the baselines occurring at 105°F with a percent difference of 3% between the two baselines. These baselines are shown in Figure 9; the initial baseline corresponds to pre-cooler B, and the second baseline corresponds to pre-coolers A and C.

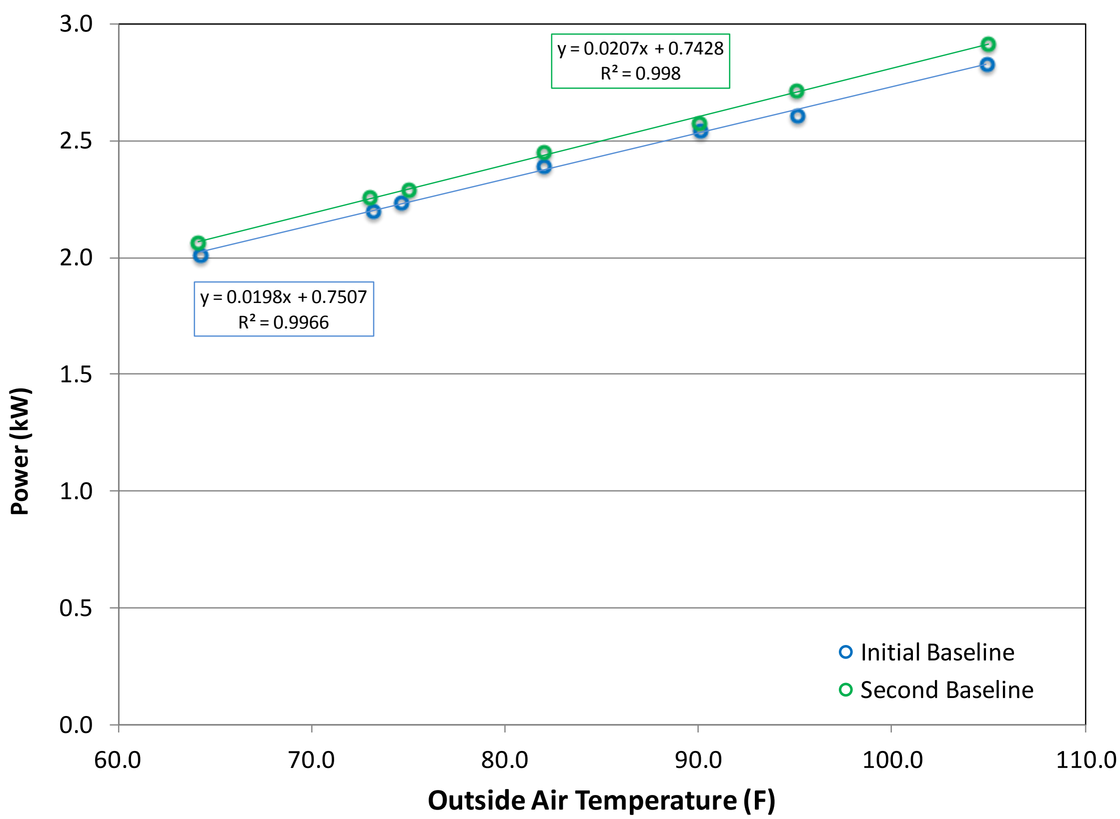

FIGURE 9. POWER BASELINE COMPARISON

Pre-cooler B was plotted against the initial baseline and the results are shown in Figure 10. The power usage of the condensing unit with pre-cooler B attached was lower than the power consumption of the condensing unit alone. The power savings increased as the dry bulb temperature increased, with power savings of 5.7% at test point A (105/73 °F dB/°F wB). Pre-coolers A and C were plotted against the second baseline (Figure 11) and both products showed a trend of increasing power savings with increasing dry bulb temperature. For test condition A (105/73 °F dB/°F wB), pre-cooler A had power savings of 5.8% and pre-cooler C had power savings of 1.9%.

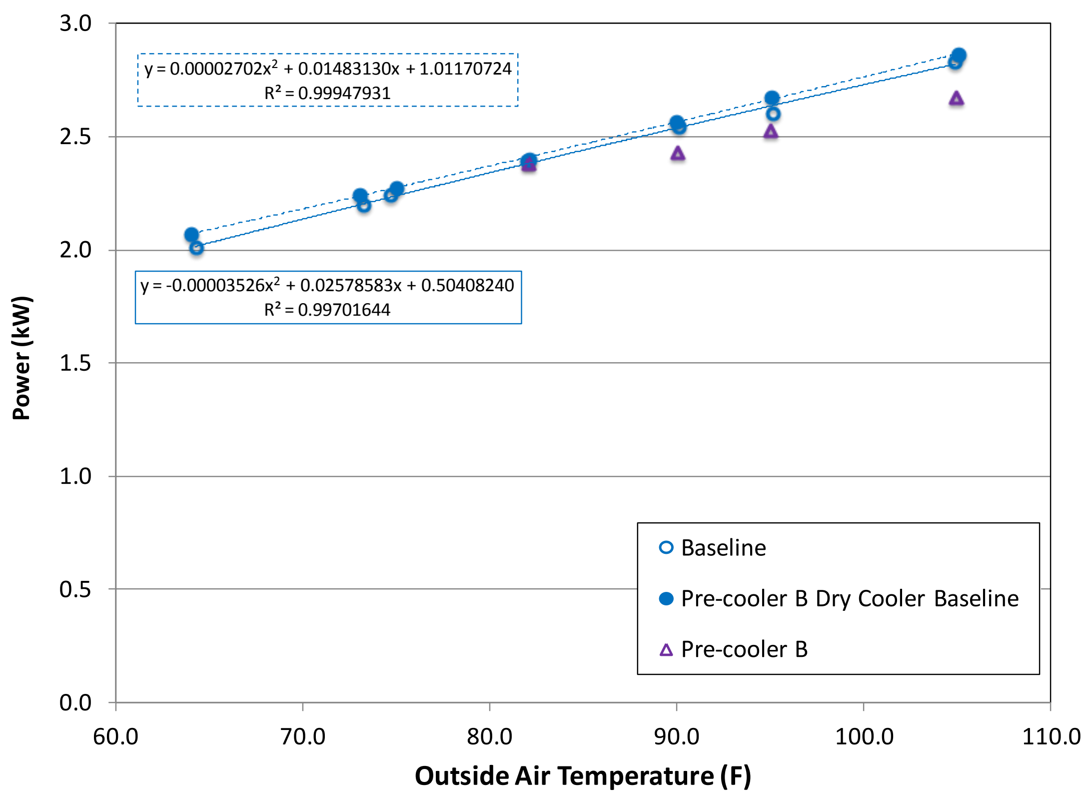

FIGURE 10. POWER BASELINE AND TEST DATA FOR PRE-COOLER B

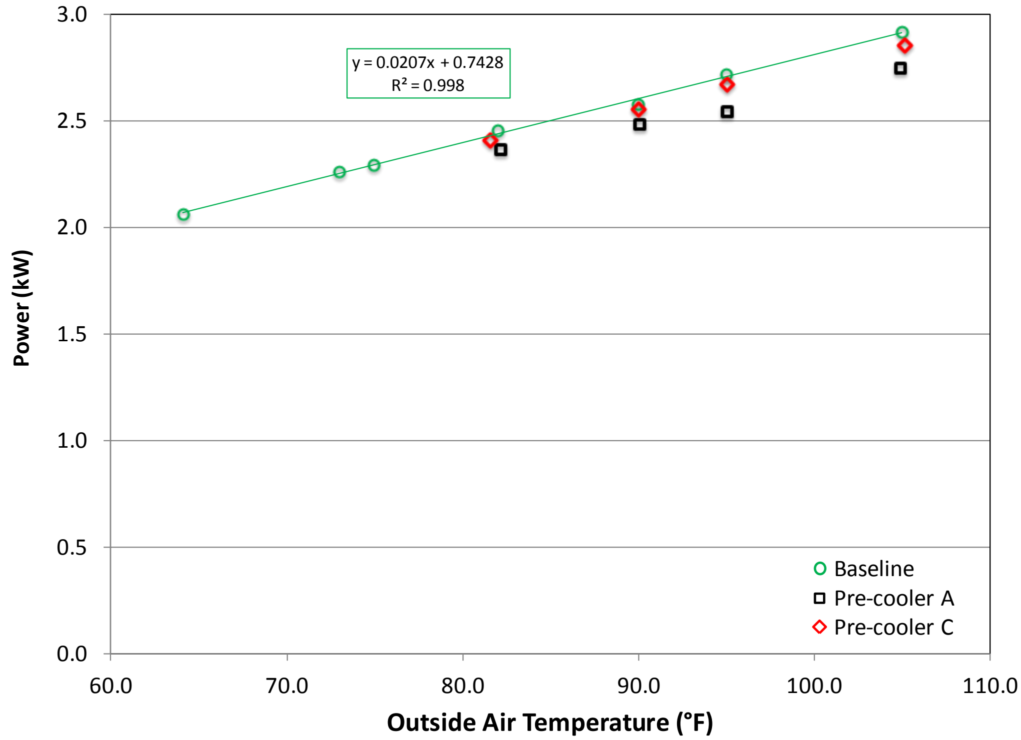

FIGURE 11. POWER BASELINE AND DATA FOR PRE-COOLERS A AND C

COEFFICIENT OF PERFORMANCE

The coefficient of performance (COP) of the condensing unit is calculated from the ratio of the cooling delivered across the evaporator coil and the power consumed by the condensing unit. It was calculated using Equation 1 where m was the mass flow rate of the water in the water-to-refrigerant heat exchanger, C_p was the specific heat of water, ΔT was the temperature drop of the water across the heat exchanger, and P was the power consumed by the condensing unit.

EQUATION 1. COEFFICIENT OF PERFORMANCE

$$COP = \frac{mC_P\Delta T\ _{Water}}{P}$$

Two sets of baseline data were gathered. The initial baseline was used as the comparison for pre-cooler B, the first pre-cooler that was tested. Due to the change in refrigerant charge between baselines, a comparison of the two baselines is shown in Figure 12. The initial baseline was used for the Pre-cooler B tests, and the second baseline was used for pre-cooler A and C tests, conducted after the refrigerant leak and repair. The two baselines are very similar; the largest discrepancy occurs at 64 degrees where there is a 3.8% difference between the baseline values. Despite the similarity, all test calculations for the products will be done using the corresponding baseline. For all tests, the COP was calculated as shown in Equation 1. These results for Pre-cooler B are shown in Figure 13 and the results for Pre-cooler A and C are shown in Figure 16. The error bars show the uncertainty analysis for the COP measurements for which the methodology is described in the uncertainty analysis section.

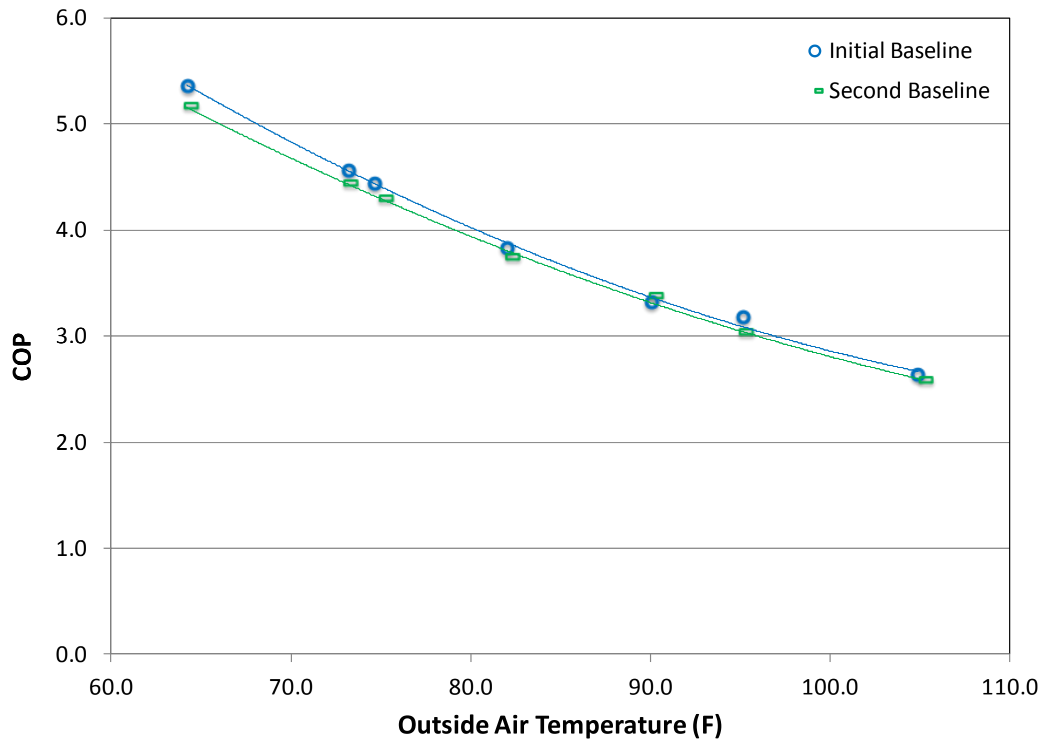

FIGURE 12. BASELINE COMPARISON BEFORE AND AFTER REFRIGERANT RE-CHARGE

In Figure 13, the blue line with hollow circles is the baseline data of the test points listed in Table 4. The dashed blue line with filled circles is the dry-cooler baseline from Table 5 (pre-cooler B has a fiberglass media surrounding the condensing unit). The purple triangles are the results for the test points with an operational pre-cooler from Table 6. Since pre-coolers A and C do not have anything that adds to the system resistance of the condensing unit, it was not necessary to obtain the dry-cooler baseline from Table 5 for these products. The results for these two pre-coolers are shown in a similar format in Figure 14.

The pre-cooler test data shown in Figure 13 and Figure 14 do not follow a similar trend as the baseline because the efficiency is a function of two variables, dry bulb and wet bulb temperatures. The COP increase was calculated for each test point by finding the difference between the performance during pre-cooler operation and the baseline data (Figure 15). The baseline performance was calculated from the best fit quadratic curve of the seven baseline test points. All three pre-cooler products had the poorest performance for test 3D (82/73 °F dB/wB) due to the small wet bulb depression (WBD) of 9°F. The pre-coolers had a slightly better performance when a larger WBD was observed, with some tapering off occurring for test point 3A (105/73 °F dB/wB), that had the highest temperature and the largest WBD of 32°F.

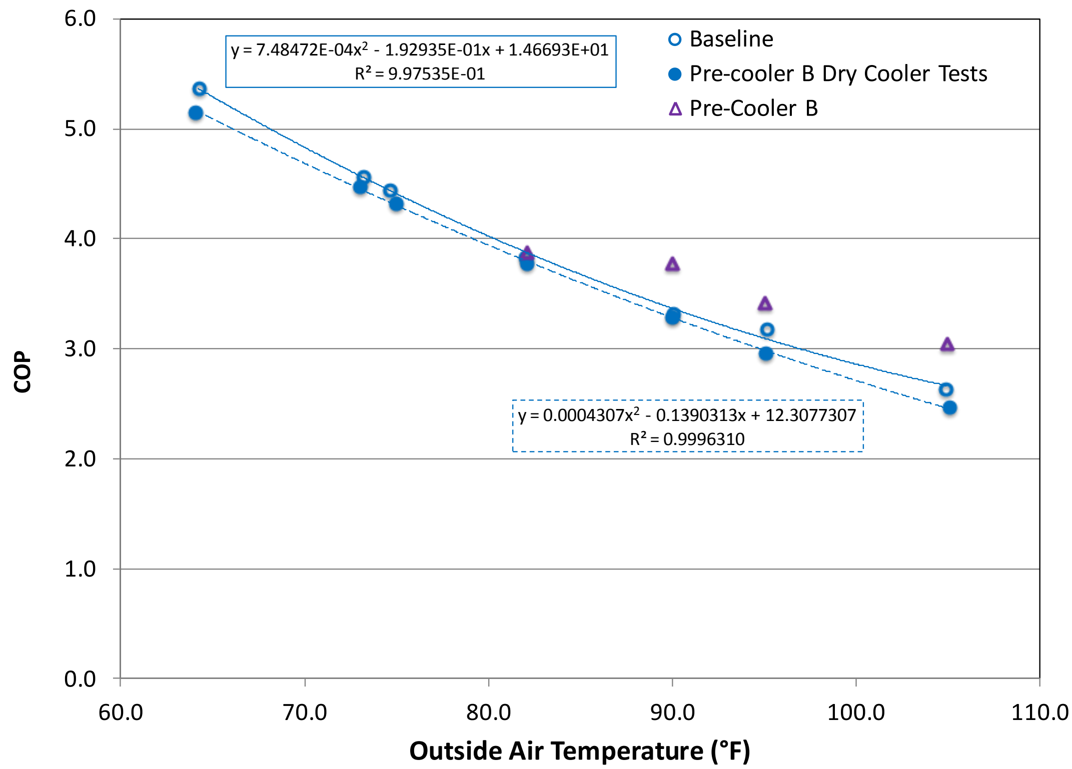

FIGURE 13. COP VS OUTSIDE AIR TEMPERATURE FOR PRE-COOLER B

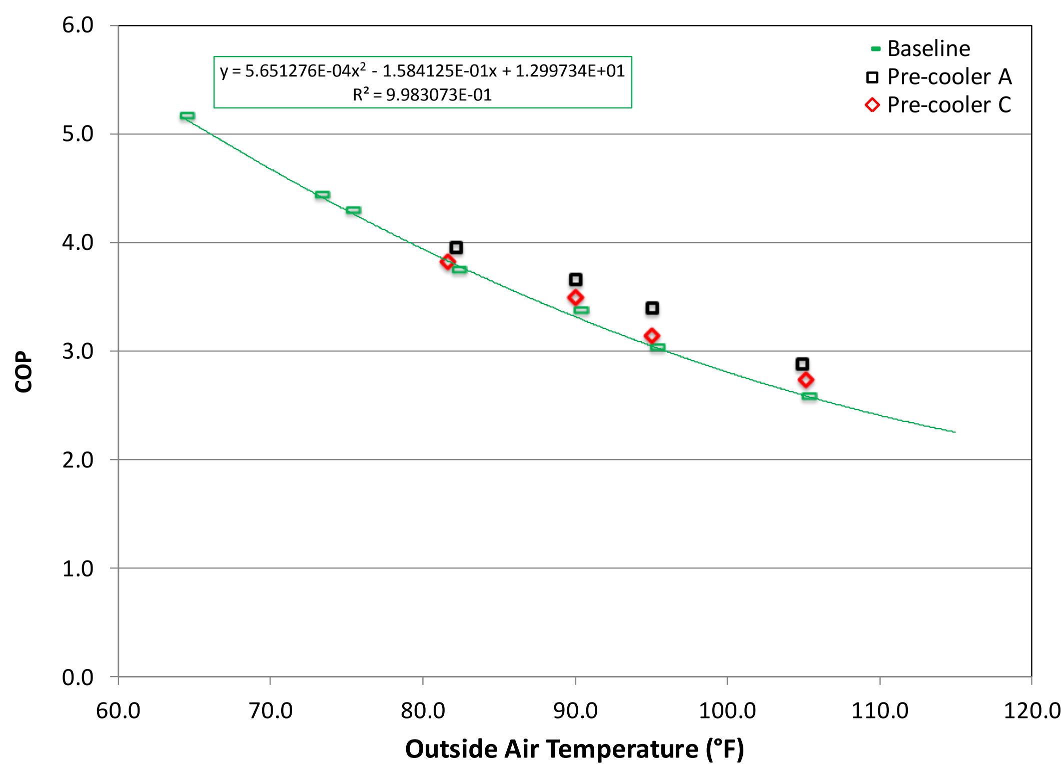

FIGURE 14. COP VS OUTSIDE AIR TEMPERATURE, PRE-COOLERS A AND C

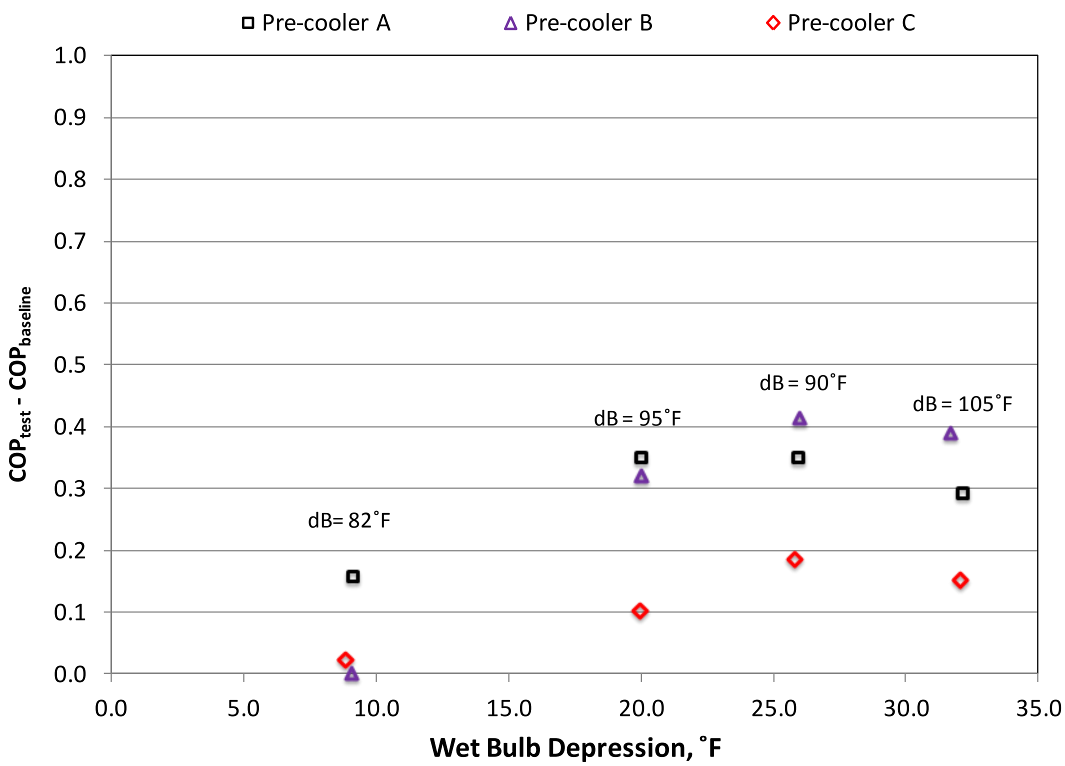

FIGURE 15. COP INCREASE VERSUS WET BULB DEPRESSION

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. This applies to evaporative pre-coolers that use wetted media to introduce water to the air for evaporation and can be calculated using Equation 2. Evaporative Effectiveness, where $T_{dB,in}$ and $T_{wB,in}$are the dry bulb and wet bulb temperatures entering the condenser pre-cooler and $T_{dB,out}$ is the dry bulb temperature leaving the condenser pre-cooler.

EQUATION 2. EVAPORATIVE EFFECTIVENESS

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

Measuring the temperature at the pre-cooler outlet of evaporative pre-coolers is difficult for several reasons. It is difficult to measure directly because the water can cause the temperature sensors to give inaccurate measurements. In addition, the air leaving the pre-cooler apparatus may be poorly mixed, causing difficulty in determining where to take the measurement. A possible workaround involves measuring the temperature and relative humidity of the condenser exhaust and inlet air to the pre-cooler and 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 16). These assumptions may not be true if either 1) water evaporates on the condensing coil, or 2) water droplets pass through the condensing coil and evaporate later in the air stream. In addition the exhaust air stream may not be well mixed at the sensing location. With these caveats in mind, the exhaust data was obtained during tests using a Vaisala HDM60Y that measured temperature and relative humidity. The absolute humidity of the exhaust was calculated, and the dry bulb temperature at the inlet wet bulb temperature was calculated to determine the post pre-cooler dry bulb temperature. The EE was then calculated using Equation 2 and the results are shown in Table 9.

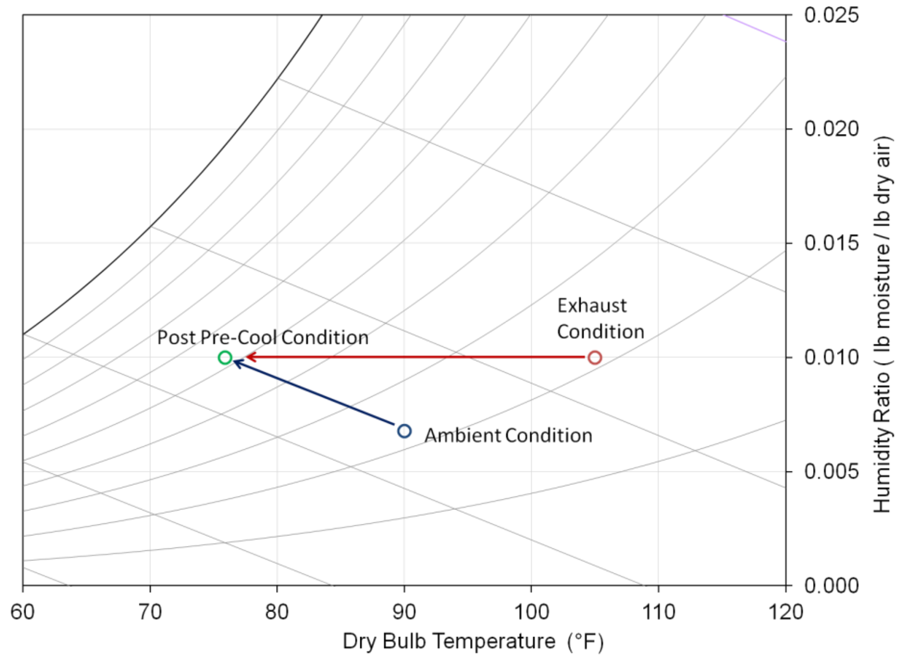

FIGURE 16. USING EXHAUST AND AMBIENT CONDITIONS TO CALCULATE POST PRE-COOL CONDITION

TABLE 9. EVAPORATIVE EFFECTIVENESS WITH RESPECT TO EXHAUST HUMIDITY MEASUREMENTS

TEST CONDITIONS (DB/WB °F)	PRE-COOLER A	PRE-COOLER B	PRE-COOLER C
105/73	3%	28%	-6%
95/75	13%	27%	-5%
90/64	13%	41%	1%
82/73	14%	43%	-17%

For most tests of Pre-Cooler C, the exhaust measurements recorded a lower humidity ratio than the inlet air conditions, resulting in negative evaporative effectiveness. Clearly, the evaporative effectiveness cannot be less than zero (the product is not de-humidifying). The negative result is from a measurement error that occurred when trying to measure a small (or non-existent) change in humidity ratio. This error could be reduced by measuring the exit humidity ratio using a second chilled mirror sensor.

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 (power, capacity, or 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. This method will be 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 pre-cooler test point, as shown in Figure 17, 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 equivalent to the COP obtained during the test period.

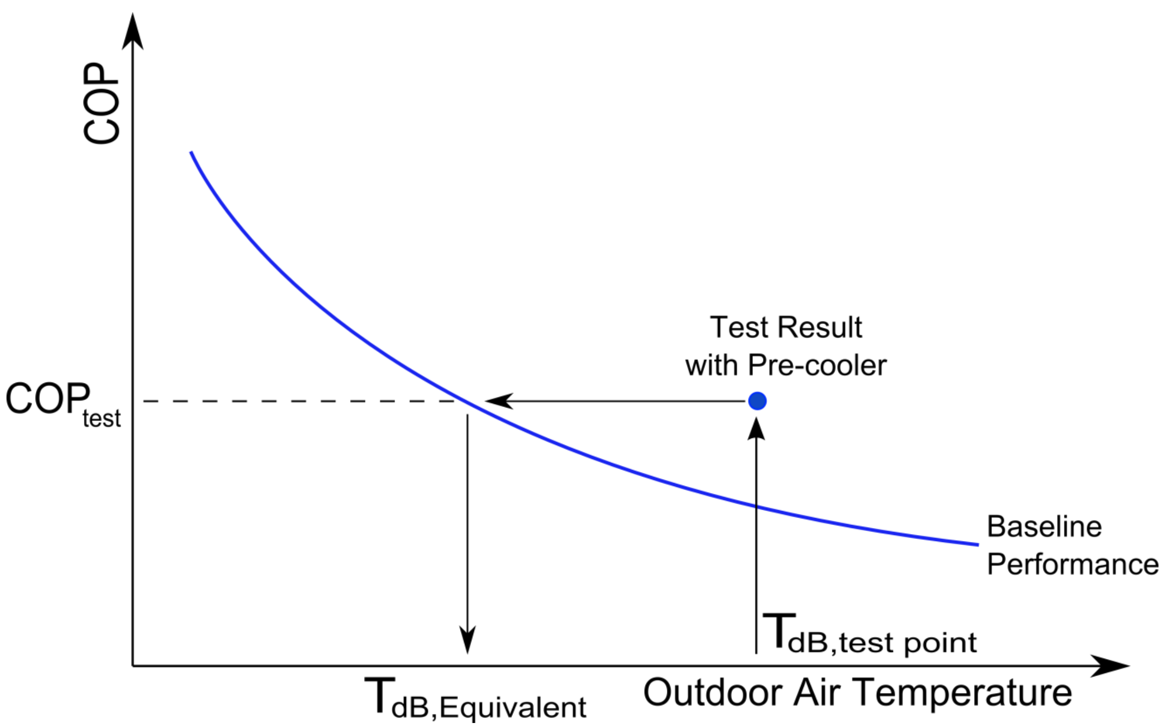

FIGURE 17. PROCESS FOR CALCULATING $T_{DB,EQUIVALENT}$

The baseline curves obtained in this experiment were all second order polynomials. The general equation for a second order polynomial is shown in Equation 3.

EQUATION 3. GENERAL SECOND ORDER POLYNOMIAL

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

Constants a, b, and c are solved from a least squares fit of the baseline test data from the condensing unit, T_{db} is the condenser inlet dry bulb temperature, and PP is the condensing unit performance parameter (power, capacity, or COP). To determine the equivalent dry bulb temperature entering the condenser during a pre-cooler test, the quadratic equation was solved as shown in Equation 4.

EQUATION 4. GENERAL EQUATION TO DETERMINE T_{DB} EQUIVALENT

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

Constants a, b, and c are equal to the constants of the second order baseline equation in Equation 3, PP_{test} is the performance parameter measured during the pre-cooler test (such as COP), and $T_{dB,eq}$ is the equivalent dry bulb temperature of the test. These constants were determined from a least squares fit from the test data for each baseline of the three products for the COP, the power draw, and the capacity of the unit (Table 10). For pre-cooler B, the coefficients are also defined for the three dry-cooler baselines; this was not needed for pre-coolers A and C since they do not add system resistance and no dry-cooler baselines were necessary.

TABLE 10. COEFFICIENTS A, B, AND C FOR EQUATION 4 TO DETERMINE T_{DB} EQUIVALENT

Performance Parameter	Pre-cooler B			Pre-cooler A and C		
	Constant A	Constant B	Constant C	Constant A	Constant B	Constant C
COP	7.485E-04	-1.929E-01	1.467E+01	5.651E-04	-1.584E-01	1.300E+01
Capacity	1.025E-03	-4.530E-01	6.189E+01	3.231E-04	-3.194E-01	5.566E+01
Power	-3.526E-05	2.579E-02	5.041E-01	-8.149E-07	2.080E-02	7.371E-01
COP – dry	4.307E-04	-1.390E-01	1.231E+01	--	--	--
Capacity – dry	-5.165E-04	2.201E-01	5.289E+01	--	--	--
Power - dry	2.702E-05	1.483E-02	1.012E+00	--	--	--

The equivalent dry bulb temperature was calculated from Equation 4 and the relevant coefficients from Table 10. The results are shown in Table 11. The dry bulb temperatures calculated from the exhaust method (Equation 2) are shown for comparison.

TABLE 11. EQUIVALENT DRY BULB TEMPERATURES FOR ALL THREE PRE-COOLERS AND ALL FOUR MEASUREMENT METHODS

	Pre-cooler A				Pre-cooler B				Pre-cooler C			
Test Condition (dB/wB °F)	COP	Power	Capacity	Exhaust	COP	Power	Capacity	Exhaust	COP	Power	Capacity	Exhaust
105/73	98.5	97.1	99.4	104.1	96.4	97.2	95.8	96.2	101.4	102.4	100.6	107.1
95/75	88.9	87.3	90.3	92.5	89.7	89.7	90.2	89.5	93.4	93.6	92.8	96.0
90/64	84.2	84.3	83.9	86.6	83.6	84.8	83.0	79.5	87.1	87.6	86.1	89.8
82/73	79.5	78.6	80.7	80.8	82.0	82.5	82.2	78.2	81.6	80.7	82.7	83.0

Using the equivalent dry bulb temperatures from Table 11, the evaporative effectiveness of each pre-cooler was solved using the equation:

EQUATION 5. 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 4. The evaporative effectiveness for all three products was solved using data from all three performance parameters (COP baseline, capacity baseline, and total power baseline). The results are shown in Table 12. The evaporative effectiveness calculated using the exhaust humidity measurements that were presented in Table 9 are also shown in Table 12 for comparison.

The relatively small differences in temperature calculated in Table 11 result in a significant difference in evaporative effectiveness, particularly when wet bulb depression is low. For example, the tests of pre-cooler A at conditions 95°F/75°F have equivalent dry bulb temperatures between 87.3°F and 92.5°F resulting in a calculated evaporative effectiveness between 13-39%. Measuring and/or calculating this temperature is the biggest challenge in characterizing pre-cooler performance; this topic will be addressed further in the uncertainty analysis and discussion.

In Table 12, pre-cooler B data was calculated using the no-cooler baselines so that the data is comparable to the other pre-coolers. However, pre-cooler B evaporative effectiveness was also calculated using the dry-cooler baseline, and the data is compared and presented with the no-cooler baseline evaporative effectiveness in Table 13. The evaporative effectiveness calculated using the baseline data with the pre-cooler installed is the most accurate measurement of the actual pre-cooling delivered. However, it does not take into account the penalty of the additional resistance. Calculating the evaporative effectiveness using the baseline (without blanket) data is a better model for actual performance and takes into account efficiency losses from the installation of the fiberglass blanket. On average the calculated evaporative effectiveness was eight percentage points less when the blanket was installed.

TABLE 12. EVAPORATIVE EFFECTIVENESS CALCULATED USING T_{DB}, EQUIVALENT FROM BASELINE ANALYSIS

		PRE-COOLER A				PRE-COOLER B				PRE-COOLER C			
TEST CONDITION (DB/WB °F)	WBD °F	COP	POWER	CAPACITY	EXHAUST	COP	POWER	CAPACITY	EXHAUST	COP	POWER	CAPACITY	EXHAUST
105/73	32	21%	24%	17%	3%	28%	24%	29%	28%	12%	8%	14%	-6%
95/75	20	32%	39%	23%	13%	28%	28%	24%	27%	10%	7%	11%	-5%
90/64	26	23%	22%	23%	14%	25%	21%	27%	41%	12%	9%	15%	1%
82/73	9	28%	39%	16%	16%	0%	-2%	-1%	43%	-1%	9%	-12%	-17%

TABLE 13. EVAPORATIVE EFFECTIVENESS FOR PRE-COOLER B CALCULATED WITH AND WITHOUT FIBERGLASS BLANKET

		WITH BLANKET			WITHOUT BLANKET		
TEST CONDITION (DB/WB °F)	WBD °F	COP	POWER	CAPACITY	COP	POWER	CAPACITY
105/73	32	35%	29%	37%	28%	24%	29%
95/75	20	36%	34%	33%	28%	28%	24%
90/64	26	30%	27%	31%	25%	21%	27%
82/73	9	13%	15%	9%	0%	-2%	-1%

The data in Table 12 was plotted with respect to wet bulb depression for each pre-cooler (Figure 20, Figure 21, and Figure 22). For pre-cooler A, the highest equivalent evaporative effectiveness was calculated based on the power data, followed by the COP data, and then the capacity data (Figure 18). The data had the closest agreement at the highest wet bulb depression, which is most likely due to reduction of the error relative to the magnitude of the signal being measured. The exhaust data measured a lower evaporative effectiveness in comparison to the equivalent evaporative effectiveness method. Recall that pre-cooler A is not a traditional air pre-cooler; it sprays water directly on the coil that is expected to increase the heat transfer effectiveness of the condenser coil. The energy savings are expected to be greater than would be achieved strictly by the air pre-cooling. The equivalent evaporative effectiveness method accounts for this energy savings while the exhaust measurement method does not.

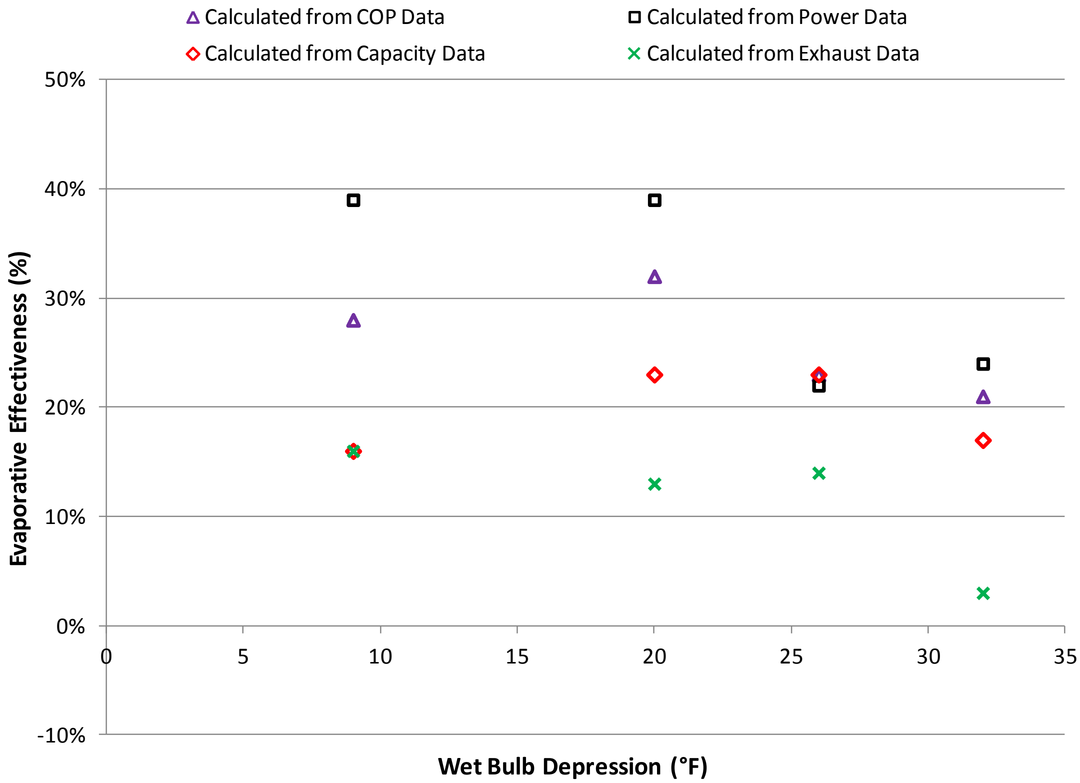

FIGURE 18. EVAPORATIVE EFFECTIVENESS CALCULATED WITH FOUR METHODS VS. WET BULB DEPRESSION FOR PRE-COOLER A

The equivalent evaporative effectiveness method had excellent agreement using power, capacity, and COP data for pre-cooler B (Figure 19). For data points at 20°F wet bulb depression and greater, the equivalent evaporative effectiveness was between 20-30%. At the high humidity test point, the equivalent evaporative effectiveness was near zero. The evaporative effectiveness measured using the exhaust data had reasonable agreement with the equivalent calculation method, except at the low wet bulb depression test point where both methods have the highest magnitude of error. Recall that pre-cooler B is a traditional pre-cooler with fiberglass blanket several misting nozzles, so agreement between the two methods is more likely than with pre-cooler A. However, as shown, the exhaust method resulted in a higher evaporative effectiveness. One potential explanation is that the exhaust method does not account for losses due to the flow resistance of the pre-cooler.

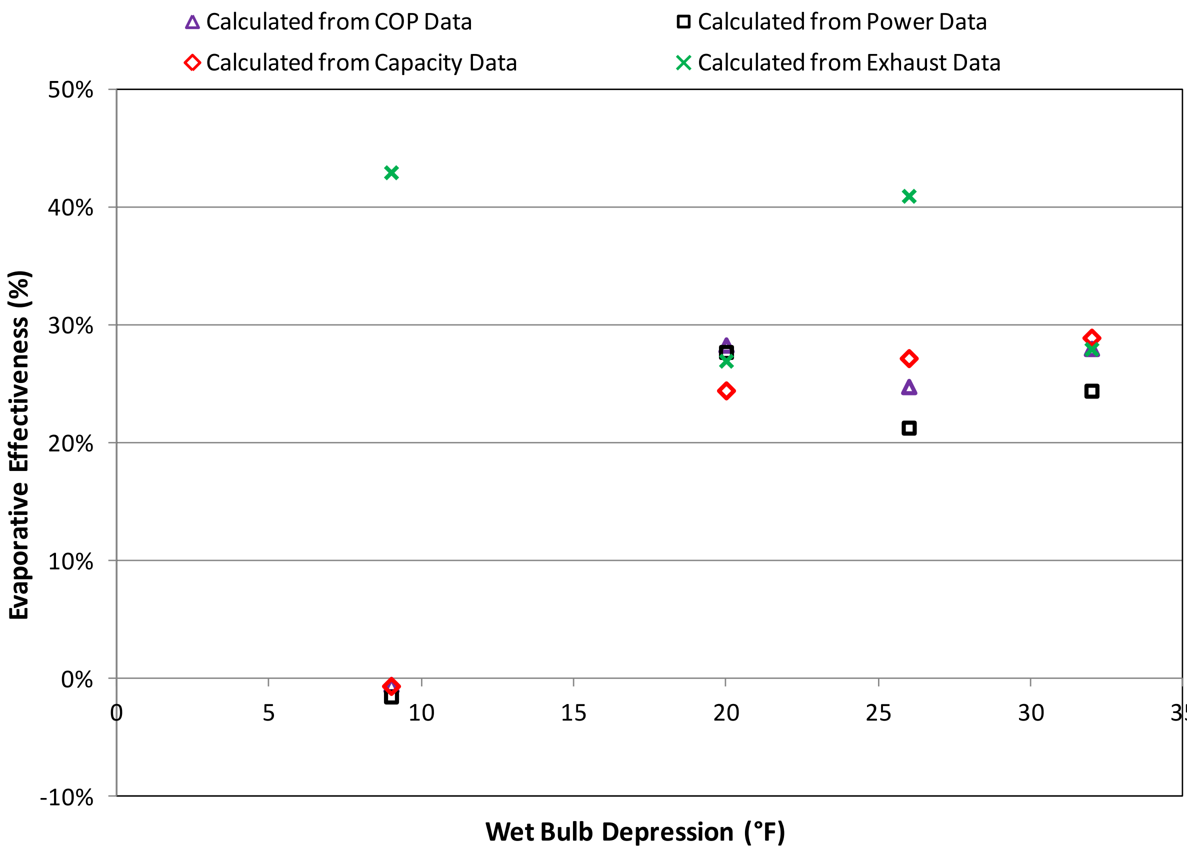

FIGURE 19. EVAPORATIVE EFFECTIVENESS CALCULATED WITH FOUR METHODS VS. WET BULB DEPRESSION FOR PRE-COOLER B

The equivalent evaporative effectiveness method had excellent agreement using power, capacity, and COP data for pre-cooler C for tests where the wet bulb depression was 20°F or greater (Figure 20). However, the equivalent evaporative effectiveness was very low, 15% or less in all tests using both methods. Upon visual inspection, the pre-cooler appears to mist a very small amount of water, although all three misting nozzles appeared to be working properly. The exhaust measurement calculation for evaporative effectiveness generally showed a negative result, most likely due to the inability of the two humidity sensors to measure a small humidity change accurately.

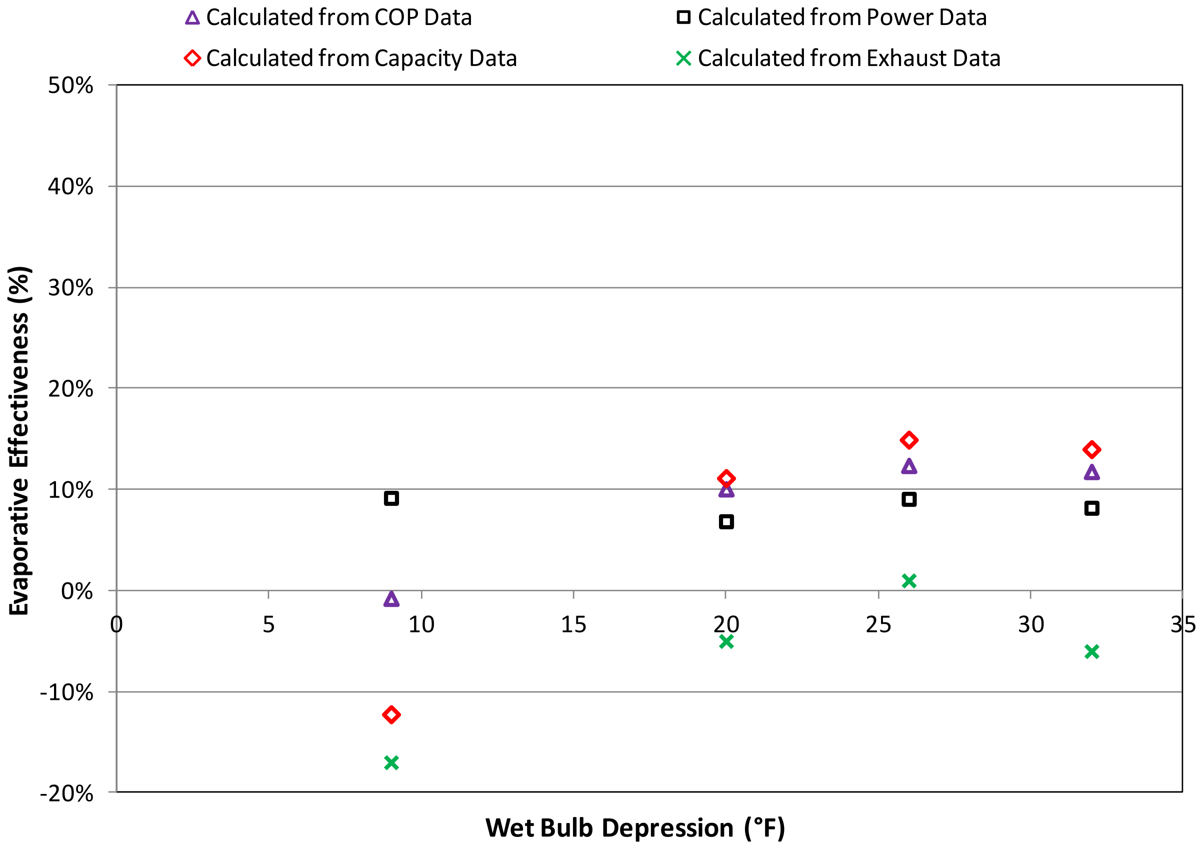

FIGURE 20. EVAPORATIVE EFFECTIVENESS CALCULATED WITH FOUR METHODS VS. WET BULB DEPRESSION FOR PRE-COOLER C

While it appears the performance of the pre-coolers have some weather dependence, the relationship is not clear based on the four tests conducted. For modeling purposes, the evaporative effectiveness values calculated using the COP baseline for all four test points were averaged together to create a single evaporative effectiveness value for each pre-cooler.

TABLE 14. AVERAGE EVAPORATIVE EFFECTIVENESS FOR TESTED PRODUCTS BASED ON COP MEASUREMENTS

PRODUCT	AVERAGE EQUIVALENT EVAPORATIVE EFFECTIVENESS
Pre-cooler A	25%
Pre-cooler B	20%
Pre-cooler C	8%

WATER USE EFFECTIVENESS

Water use effectiveness 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 6.

EQUATION 6. WATER EVAPORATION RATE

$$m_{water,evap} = (HR_{outlet} - HR_{inlet}) \cdot m_{air}$$

Where: $m_{water,evap}$ is the rate at which water evaporates into the air in slugs/s, HR_{outlet} and HR_{inlet} refer to the humidity ratio exiting and entering the pre-cooler apparatus in lb_w/lb_{da}, respectively, and m_{air} is the mass flow rate of the air across the pre-cooler in slugs/s. Using the tracer gas system described previously, the airflow rate through the condensing unit was determined to be 2465 scfm. With pre-cooler B installed, the airflow was reduced to 2388 scfm due to the fiberglass blanket around the condensing unit. HR_{inlet} was measured from the chilled mirror sensor at the inlet to the chamber. HR_{outlet} was calculated using the equivalent dry bulb temperature (Equation 4), inlet wet bulb temperature, and psychometric calculator. The process assumes the pre-cooling process has a constant wet bulb temperature. With this, the water use effectiveness was calculated as shown in Equation 7.

EQUATION 7. WATER USE EFFECTIVENESS

$$WUE = \frac{m_{water,evap}}{m_{water,supplied}}$$

The average water use effectiveness is shown for each pre-cooler product, with the range being between 20-51% (Table 15). In addition, the absolute average use in gal/hr is shown. Pre-cooler A had the best performance for energy efficiency but did not turn the majority of the water-use into energy savings. The water use effectiveness was only 20% and the water use was 7.89 gal/hr. Pre-cooler B had slightly better water use effectiveness of 31% and consumed water at the average rate of 4.94 gal/hr. Pre-cooler C used minimal water (1.21 gal/hr), and used half of that water to generate a modest evaporative effectiveness of 8%.

TABLE 15. AVERAGE EVAPORATIVE EFFECTIVENESS AND AVERAGE WATER USE EFFECTIVENESS FOR TESTED PRODUCTS BASED ON COP CALCULATIONS

PRODUCT	AVERAGE EQUIVALENT EVAPORATIVE EFFECTIVENESS	AVERAGE WATER USE EFFECTIVENESS	WATER USE (GAL/HR)
Pre-cooler A	25%	20%	7.89
Pre-cooler B	20%	31%	4.94
Pre-cooler C	8%	51%	1.21

UNCERTAINTY ANALYSIS

The uncertainty of the evaporative effectiveness and the water use effectiveness calculations were conducted using the sequential perturbation method[3],a numerical approach that utilizes a finite difference method to approximate the derivatives that represent the sensitivity of the calculated value to the variables used within the calculation. 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, Ro, based on measured values. After Ro has been calculated, an independent variable within the equation for Ro is increased by its respective uncertainty, and a new value, Ri+ is calculated. Next, the same independent variable within Ro is decreased by its respective uncertainty, and a new value, Ri- is calculated. The differences between Ri+ and Ro, and Ri- and Ro are calculated and the absolute values are averaged. The result is defined as δRi. This process is repeated for every independent variable within Ro, and the final uncertainty is calculated as shown in Equation 8. An example calculation is shown in Appendix A.

EQUATION 8. UNCERTAINTY USING SEQUENTIAL PERTURBATION

$$U_R = \pm \left[\sum_{i=1}^{L} \delta R_i^{\,2} \right]^{\frac{1}{2}}$$

The uncertainty of the evaporative effectiveness and water use effectiveness was calculated using this method for all three pre-cooler products and the results are shown in Table 16 for pre-cooler A, Table 17 for pre-cooler B and Table 18 for pre-cooler C.

TABLE 16. UNCERTAINTY RESULTS FOR PRE-COOLER A

Psych. Conditions	Exhaust		COP		Power		Capacity	
	EE	WUE	EE	WUE	EE	WUE	EE	WUE
105/73	0.025 ± 0.17	0.032 ± 0.21	0.198 ± 0.08	0.25 ± 0.12	0.244 ± 0.05	0.308 ± 0.09	0.17 ± .05	0.215 ± 0.09
95/75	0.126 ± 0.22	0.094 ± 0.16	0.307 ± 0.13	0.23 ± 0.12	0.389 ± 0.07	0.292 ± 0.08	0.234 ± .08	0.175 ± 0.08
90/64	0.134 ± 0.16	0.128 ± 0.14	0.223 ± 0.1	0.214 ± 0.1	0.221 ± 0.05	0.211 ± 0.06	0.235 ± .06	0.225 ± 0.06
82/73	0.145 ± 0.38	0.049 ± 0.12	0.29 ± 0.17	0.098 ± 0.1	0.387 ± 0.150	0.13 ± .07	0.165 ± .16	0.055 ± 0.07

TABLE 17. UNCERTAINTY RESULTS FOR PRE-COOLER B WITH NO-COOLER BASELINE

Psych. Conditions	Exhaust		COP		Power		Capacity	
	EE	WUE	EE	WUE	EE	WUE	EE	WUE
105/73	0.275 ± 0.16	0.498 ± 0.29	0.271 ± 0.09	0.492 ± 0.17	0.245 ± 0.05	0.444 ± 0.11	0.29 ± 0.06	0.526 ± 0.12
95/75	0.274 ± 0.22	0.382 ± 0.3	0.267 ± 0.14	0.372 ± 0.21	0.266 ± 0.08	0.37 ± 0.14	0.242 ± 0.1	0.338 ± 0.15
90/64	0.405 ± 0.14	0.591 ± 0.2	0.248 ± 0.1	0.361 ± 0.15	0.202 ± 0.06	0.294 ± 0.09	0.27 ± 0.07	0.393 ± 0.11
82/73	0.425 ± 0.39	0.228 ± 0.2	0.006 ± 0.36	0.003 ± 0.17	-0.047 ± 0.17	-0.025 ± 0.12	-0.012 ± 0.35	-0.006 ± 0.13

TABLE 18. UNCERTAINTY RESULTS FOR PRE-COOLER C

Psych. Conditions	Exhaust		COP		Power		Capacity	
	EE	WUE	EE	EE	WUE	WUE	EE	WUE
105/73	-0.06 ± 0.2	-0.564 ± 1.81	0.117 ± 0.08	1.094 ± 0.83	0.085 ± 0.05	0.796 ± 0.52	0.141 ± 0.05	1.094 ± .53
95/75	-0.05 ± 0.25	-0.259 ± 1.27	0.081 ± 0.14	0.425 ± 0.77	0.072 ± 0.08	0.377 ± 0.49	0.113 ± 0.08	0.425 ± 0.51
90/64	0.01 ± 0.16	0.047 ± 0.75	0.115 ± 0.1	0.547 ± 0.5	0.093 ± 0.06	0.441 ± 0.29	0.15 ± 0.06	0.547 ± 0.31
82/73	-0.165 ± 0.42	-0.396 ± 0.97	-0.005 ± 0.28	-0.012 ± 0.78	0.096 ± 0.16	0.229 ± 0.5	-0.129 ± 0.33	-0.012 ± 0.54

The uncertainties that were calculated for pre-cooler A could be improved by retesting with a higher accuracy flow meter. The pre-cooler flow meter had a full scale error of 1%, but the pre-cooler operated on the low side of the flow range, incurring large uncertainties for the pre-cooler water flowrate. WCEC is sending the flow meter out for a post calibration to reduce this uncertainty. Another instrument that introduced large uncertainty was the Vaisala humidistat measurements used for exhaust dry bulb and relative humidity measurements to calculate the humidity ratio. The impact of installing a chilled mirror for a dew point measurement rather than using the Vaisala relative humidity measurement would reduce the uncertainty of the EE and WUE by a factor of three to five for the exhaust calculations only. In addition, the relative uncertainty decreases significantly as the evaporative effectiveness increases. This is because many variables measured in the experiment have fixed uncertainties; a small change will have a larger incurred uncertainty because of the fixed instrumentation uncertainties. For example, using the uncertainty calculations detailed in this section, and artificially raising the COP to model a higher evaporative effectiveness around 0.78 will result in an uncertainty of 0.07 (9%). The same calculations for an evaporative effectiveness of 0.27 have an uncertainty of 0.09 (33%).

ANALYSIS TOOL

WCEC developed a model to analyze the expected energy and demand savings resulting from installation of an evaporative condenser air pre-cooler. The four basic steps of the analysis were to:

1. Define "average" RTU performance with temperature.
2. Define weather data.
3. Define load data.
4. Combine RTU performance, pre-cooler effectiveness, weather, and load data to produce result.

Pre-cooler savings were calculated for an "average" RTU, where the average RTU was defined by analyzing manufacturer data for 26 RTUs obtained from a database file available within EnergyPlus whole building energy simulation software. The RTUs were a diverse sample of four brands (Lennox, Carrier, Goodman, and York) and multiple tonnages, (5-20 tons). Approximately half of the RTUs had R-22 refrigerant and the other half had R-410a refrigerant. Six of the RTUs had two stages of cooling; data was available for both part capacity and full capacity operation. Since the data set used was an un-weighted average of RTUs available in the EnergyPlus database, the model was not necessarily representative of the population of RTUs in Southern California.

For modeling the performance of pre-coolers, the most critical variable is the slope of the RTU performance as outdoor air temperature (OAT) changes. Two performance metrics were analyzed; the Energy Intensity Ratio (EIR), that is the power in kilowatts consumed per ton (inverse of COP), and power in kilowatts, that was normalized by the rated capacity in tons at 95°F. The slope of the EIR versus OAT was analyzed with respect to various RTU properties (size, brand, refrigerant type, and rated efficiency) to evaluate whether or not there was a statistically significant difference between groups. Refrigerant type and rated efficiency were the only variables that showed possible statistical significance, and these variables correlated to each other. For the 26 RTUs sampled, the average EIR at 95°F was 0.885 kW/ton for units with refrigerant 410-A (R-410A) and 0.925 kW/ton for units with R-22. This is a 5% increase in the average EIR for 410-A over R-22, that is likely due to recent laws banning the use of R-22 coinciding with federal laws requiring increased efficiency in air conditioning units. The data set of 26 RTUs was segregated by refrigerant type. Analyzing the slope of the EIR showed that, on average, RTUs with R-410A have a slightly larger percent change in EIR per °F than RTUs with R-22 (Figure 23). The 95% precision interval for the average is shown in Figure 21 (black error bars), and was calculated from the equation [3]:

EQUATION 9. CONFIDENCE INTERVAL

$$\mathrm{CI}_{95\%} = \mathrm{x} \pm 1.96 \times \frac{\mathrm{s}}{\overline{\mathrm{n}}}$$

Where: x is the average of the sample, s is the standard deviation of the sample, and n is the sample size.

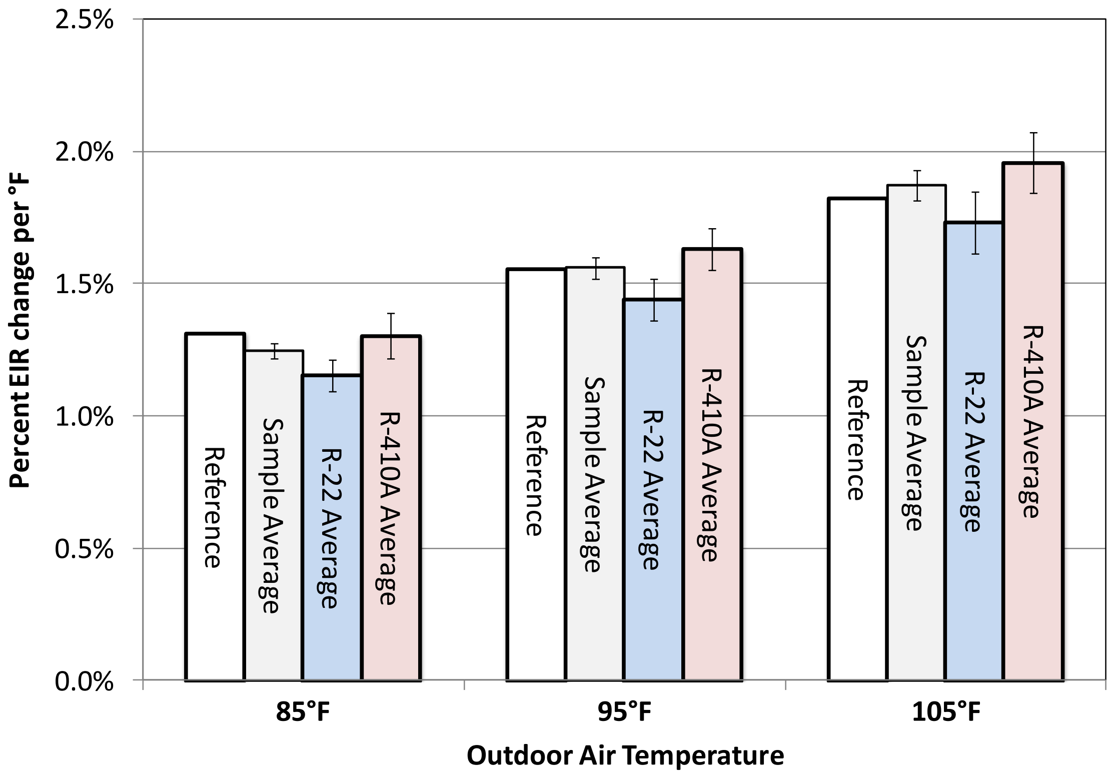

FIGURE 21. REFRIGERANT TYPE AND SLOPE OF RTU ENERGY INTENSITY RATIO

The 95% precision interval shows that the average for R-22 samples and the average for the R-410A are not statistically different based on the precision interval applied. Furthermore, the difference between the two averages is only 12%. The complete sample average (gray bars, Figure 23) was also compared against reference data (white bars, Figure 23) available in the Nonresidential Alternative Calculation Manual published by the California Energy Commission. This is a manual for building energy modeling software to ensure building design complies with California's Title 24 energy code[4]. The agreement between the two data sets is excellent and within 3%.

Since no strong correlation between RTU properties and percent change in EIR per °F were found, the average of all 32 samples (two samples for each of the six RTUs with two stages of operation) was used for the analysis. The data was averaged and the 95% confidence interval was calculated for three rating temperatures for which manufacturer data was available (85, 95, and 105°F OAT). The average EIR confidence interval and best fit polynomial curve for the three data points based on the least squares method are shown in Figure 22. The EIR based on ambient dry bulb temperature was reduced to the following equation:

EQUATION 10. EIR BASELINE

$$P_{baseline} = 0.0001391 \times T_{DB}^2 - 0.01244 \times T_{DB} + 0.8303$$

Using similar methods, the power demand of the average RTU in kW per nominal ton versus OAT was reduced to the following equation, where T_{DB} is the outdoor air dry bulb temperature:

EQUATION 11. POWER DEMAND

$$P_{baseline} = 0.00015044 \times T_{DB}^2 + 0.0061684 \times T_{DB} + 0.50183$$

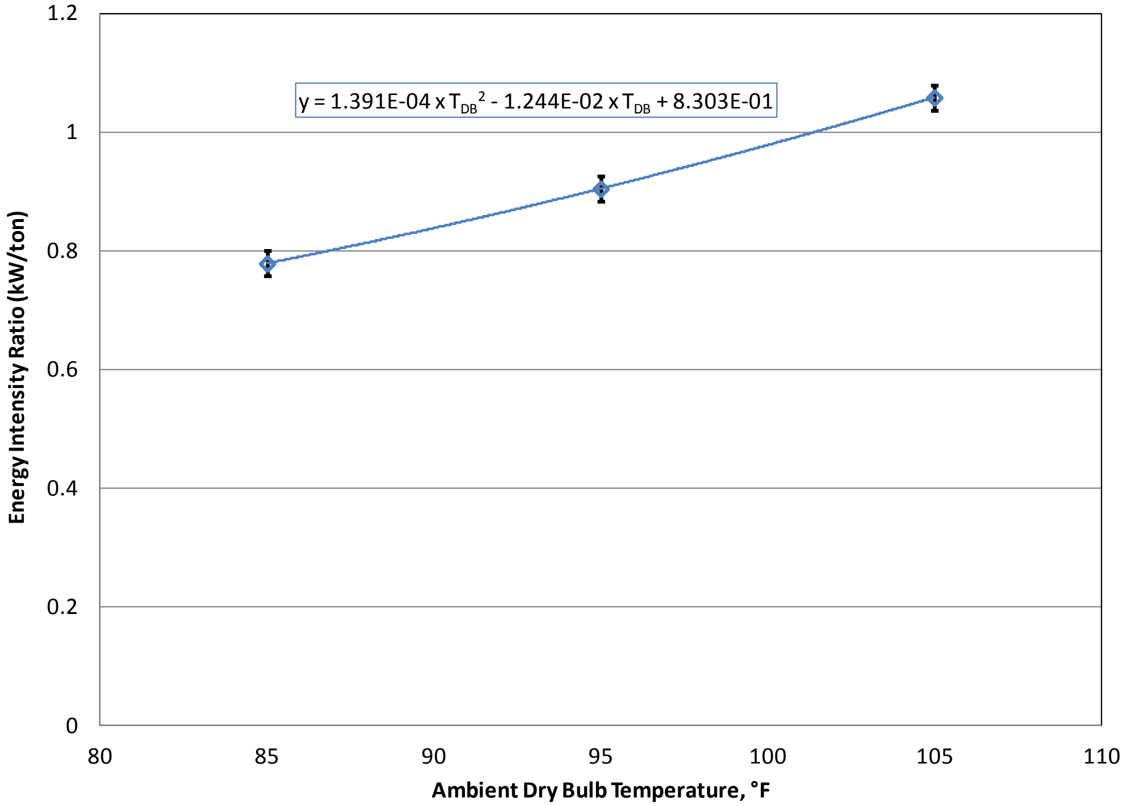

FIGURE 22. ENERGY INTENSITY RATIO FOR AVERAGE RTU

Commercial end-use data for cooling in SCE territory was downloaded from the 2003 California End-Use Survey (CEUS) [1]. The CEUS contains a commercial end-use load data set, developed using surveying, utility billing data, and modeling. Hourly energy end-use estimates were downloaded for forecasting climate zones 8, 9, 10, that are the forecasting climate zones in SCE territory and that have different boundaries than the climates zones defined by California's Title 24 (Figure 23, left). Weather data relevant to the performance of air conditioning systems and evaporative pre-coolers (ambient dry bulb and wet bulb temperatures) were downloaded for the California Title 24 climate zones in SCE Territory, 6, 8, 9, 10, 14, 15, 16 (Figure 23, right). The weather data were matched to the fractional on-time data by choosing the forecasting climate zone that most closely matched the geographical boundaries of the California Title 24 climate zone.

For each forecasting climate zone, the commercial cooling end-use data was analyzed to determine the maximum hourly energy use; this number was defined as the maximum air conditioning load in the forecast zone. The hourly data was then divided by the maximum and the result was defined as the fractional on time for the average air conditioning system. The fractional on time was averaged on a monthly basis based on the hour of the day; for example, the 6am hour for each day of August was averaged.

Using typical weather data for Title 24 climate zones, the hourly wet bulb depression was calculated by taking the difference between the dry bulb and wet bulb temperatures. The weather data were then averaged using the same scheme as the fractional on-time data. For peak demand savings calculations, the 1% design-day dry bulb and wet bulb temperatures were used for each climate zone analyzed.

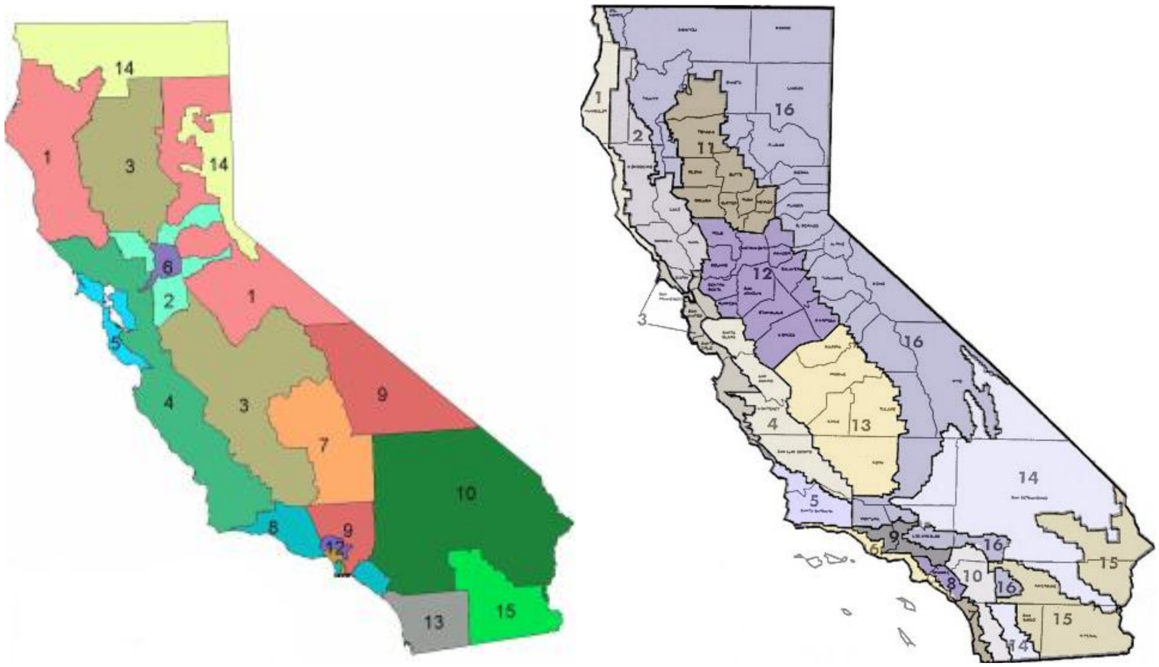

FIGURE 23. CALIFORNIA FORECASTING CLIMATE ZONES (LEFT) AND CALIFORNIA TITLE 24 CLIMATE ZONES (RIGHT)

The baseline energy use per ton of cooling was then calculated from the equation:

EQUATION 12. ANNUAL ENERGY-USE PER TON, "AVERAGE" BASELINE SYSTEM

$$E_{baseline} = \sum_{i=1}^{12} \sum_{i=1}^{24} EIR_{i,h\,(DB)} \times f_{i,h} \times D_i$$

Where: $EIR_{i,h\,(DB)}$ is the energy intensity ratio, or kW/ton, for an average air conditioner for each hour at the ambient dry bulb temperature, $f_{i,h}$ is the fractional on-time at each hour, and D_i is the number of days in the month. The equation is summed over 24 hours and 12 months.

With the pre-cooler installed, the retrofit energy use per ton of cooling was calculated from the equation:

EQUATION 13. ANNUAL ENERGY-USE PER TON, PRE-COOLER INSTALLED

$$E_{pre-cooler} = \sum_{i=1}^{12} \sum_{i=1}^{24} EIR_{i,h\,(DB,eq)} \times f_{i,h} \times D_i$$

Where: $EIR_{i,h\,(DB,eq)}$ is the energy intensity ratio, or kW/ton, for an average air conditioner for each hour at the equivalent ambient air temperature of the condenser inlet when the pre-cooler is installed, $f_{i,h}$ is the fractional on-time at each hour, and D_i is the number of days in the month. The equivalent dry bulb temperature is calculated from the evaporative effectiveness of the pre-cooler, which is measured using the laboratory protocol:

EQUATION 14. EQUIVALENT DRY BULB TEMPERATURE

$$T_{dB,\mathrm{eq}} = T_{dB,in} - EE \times \left(T_{dB,in} - T_{wB,in}\right)$$

The pre-cooler was modeled only to operate when the OAT was above 70°F and the equivalent temperature was only applied when this condition was met; otherwise, the ambient OAT was used.

The baseline peak demand in kW per nominal ton was calculated using Equation 11 and the design day dry bulb condition. The peak demand with the pre-cooler was then calculated from Equation 11 modified with the equivalent dry bulb temperature calculated from Equation 14 using 1% design day weather conditions for the climate zone analyzed. The modified equation for the system with the pre-cooler is:

EQUATION 15. POWER DEMAND

$$P_{baseline} = 0.00015044 \times T_{DB}^2 + 0.0061684 * T_{DB} + 0.50183$$

The absolute and percent savings for both energy and peak demand were then determined by calculating the difference between the baseline system and the system with the pre-cooler. 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 24 to Figure 27). The greatest savings are expected in Climate Zone 15, with a savings of 330 kWh/ton/year for a pre-cooler with evaporative effectiveness of 0.9, or 26% energy savings. Peak demand savings of 30% and 0.35 kW per nominal ton are calculated. Note that the energy savings 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, the expected energy savings can be calculated using the appropriate percentage from Figure 27. 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.

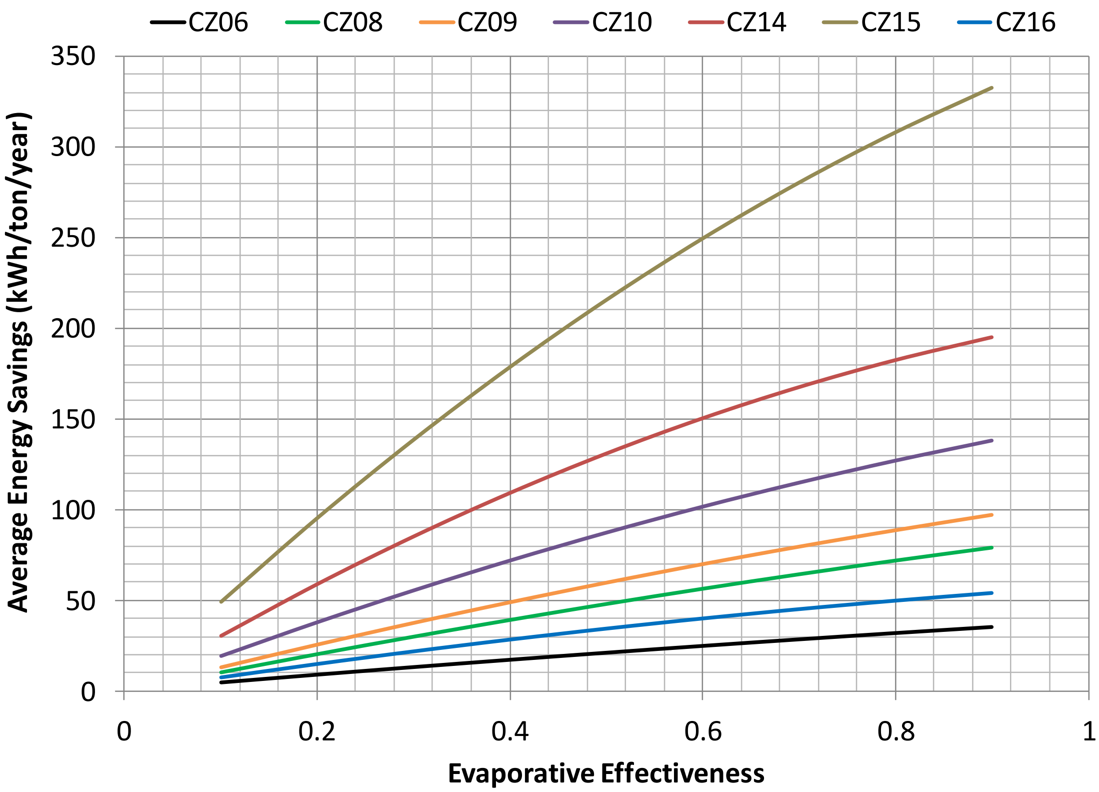

FIGURE 24. MODELED AVERAGE ENERGY SAVINGS OF AN EVAPORATIVE CONDENSER AIR PRE-COOLER

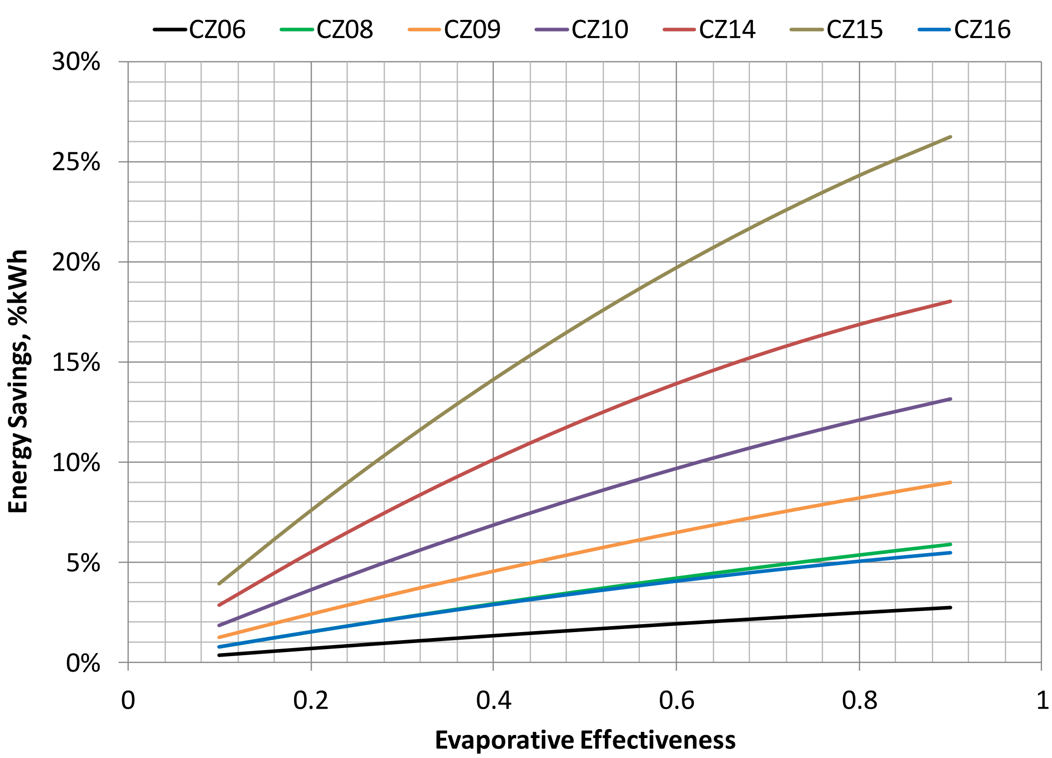

FIGURE 25. MODELED AVERAGE PERCENT ENERGY SAVINGS OF AN EVAPORATIVE CONDENSER AIR PRE-COOLER

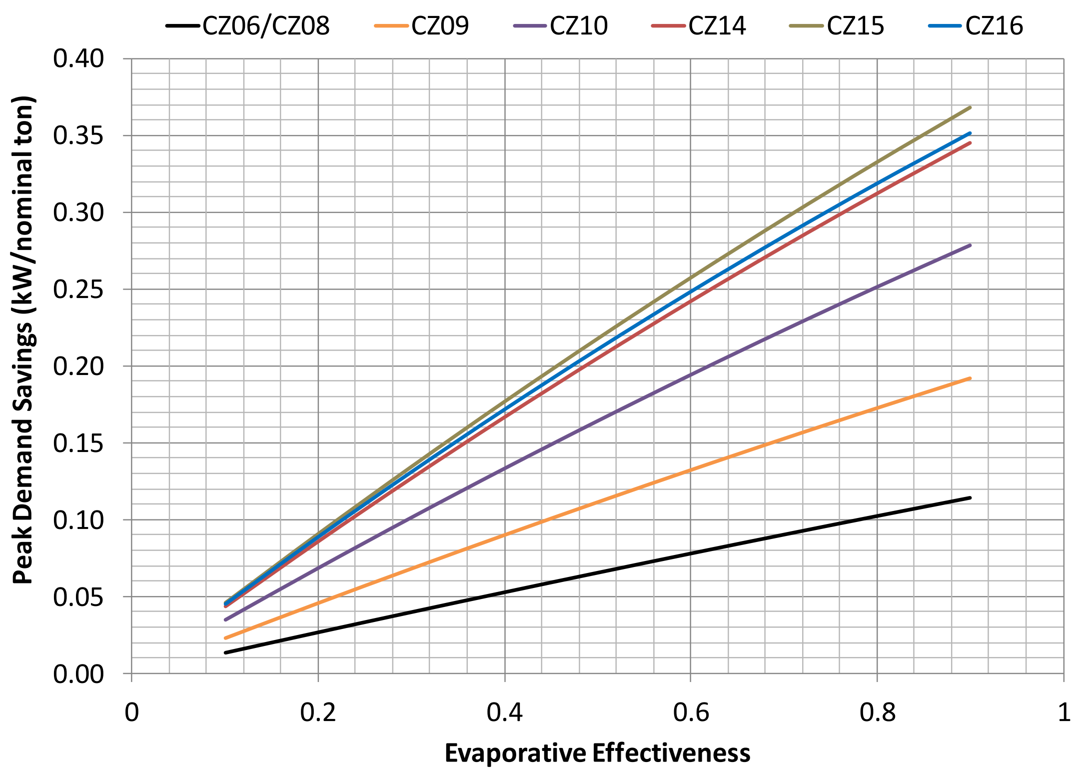

FIGURE 26. MODELED POWER SAVINGS OF AN EVAPORATIVE CONDENSER AIR PRE-COOLER

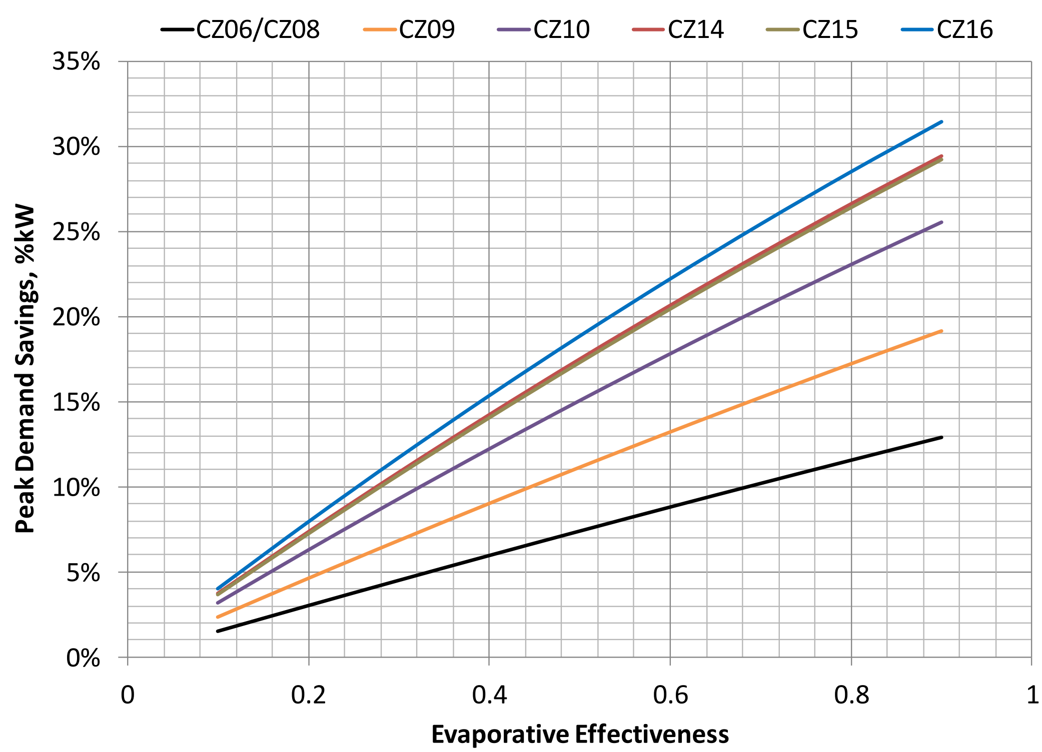

FIGURE 27. MODELED PERCENT POWER SAVINGS OF AN EVAPORATIVE CONDENSER AIR PRE-COOLER

DISCUSSION

The purpose of this experiment was to design and test a protocol for evaluating condenser air pre-coolers. To this end, the project was successful. The reader should not consider the products that were tested to be representative of the industry. The three evaporative condenser air pre-cooler products that were assessed in this report are designed to retrofit on existing air conditioners for the residential or light commercial market. They are easily purchased and installed on small condensing units and are not custom designed for each individual application. Some modifications to these products can be made in the form of adding or subtracting water delivery devices to increase water delivery to the air surrounding the condensing unit. Each of the pre-cooler products tested was able to achieve the intended goal of improving condensing unit performance. They were not as effective as commercial pre-coolers, which are generally advertised as having a much greater evaporative effectiveness of around 70-80%. Commercial condenser air pre-coolers are generally custom designed for each application, ensuring that all or a specific area of the condenser intake receives consistent amounts of water. These designs often use evaporative media where water is delivered to a manifold, trickles down through the media, and returns to a sump. The water is then either flushed out or re-circulated through the evaporative media. This method was not employed by any of the designs tested. It would be interesting to test a small pre-cooler design utilizing evaporative media on a residential air conditioner to see where the evaporative effectiveness of this design would fall in comparison to the three pre-coolers tested herein.

The majority of instrumentation utilized for measurements were of high accuracy, but some improvements could be made in future research. One such improvement would be to use a chilled mirror on the condenser exhaust in place of the Vaisala HMD60Y that was used to record dry bulb temperature and the relative humidity of the exhaust airstream. As stated earlier, this would reduce the uncertainty by a factor of 3 to 5 for the exhaust calculations. In addition, uncertainty can be further reduced by increasing the level of calibration for critical measurements. Conducting a seven-point calibration in the test measurements range on the capacity-related measurements (evaporator in temperature, evaporator out temperature, and water flow rate) would reduce the uncertainty of the results. In addition, the water flow meter for the pre-cooler can be calibrated to the range of interest. WCEC plans to re-calibrate these sensors post-experiment and update the data set and uncertainty analysis accordingly. Furthermore, measuring pre-coolers of higher evaporative effectiveness would reduce the relative uncertainty (for example, an effectiveness of 0.8 ± 0.1 is perhaps a more useful result than 0.2 ± 0.1).

A reduction of measurement uncertainty in both the exhaust-based and the performance-based methods would allow for better comparison between the two methods. With a pre-cooler installation, the water droplets are expected to evaporate in one of three places, with each place impacting the performance and the resulting measurements differently:

1) Before the coil, this pre-cools the air only. Both the exhaust method and performance-based method should calculate similar EE. This was observed in several of pre-cooler B tests.

2) On the coil, this changes the heat transfer mechanism and more heat is rejected due to the water evaporation than would be in a process where the air is simply pre-cooled. In this case, the EE measured by the exhaust measurements underestimated

the potential energy and demand savings. This was seen in testing of pre-coolers A and C.

3) After the coil, this does not benefit the condensing unit performance. It is possible that some water droplets pass through the condenser coil but evaporate before reaching the exhaust sensor. In this case, the exhaust method would report a higher EE than is reflected in the performance. It does not appear that this was an issue during testing of these pre-coolers.

Continued testing and validation of the performance-based methods is the most reliable way to measure the degree of which the evaporation of water is being converted to power and energy savings.

Since the water available at the temporary laboratory was non-potable well water, it was decided to use water that had been purified by reverse osmosis filtration for testing, so the effects of poor water quality were not able to be assessed on a formal basis. However, during the chamber construction, some initial tests were conducted with pre-coolers A and B where they were exposed to the poor water quality from the well. Pre-cooler A did not seem to have any issues with clogging in short durations of testing (~10-15 non-consecutive hours), but scale became evident on the exterior of the condensing unit, which is a significant and expected concern for a design that sprays directly onto the coil. Pre-cooler B was also exposed to the well water for a similar duration of time, and was found to have clogging issues with the small orifice nozzles. These nozzles constantly required brushing with a metal brush, soaking in anti-scaling solutions such as vinegar, or replacement with new nozzles or they would not spray water properly. Pre-cooler C was not exposed to this same well water, but it utilizes very similar nozzles to pre-cooler B, so this pre-cooler product is expected to also have difficulties with clogging when exposed to poor water quality. These nozzles had the most issues after being shut off overnight and being restarted the following day. Since this a typical use profile, this could lead to problems in hard water areas.

CONCLUSIONS

The project was successful in developing a pre-cooler test protocol, vetting that protocol using three off-the-shelf-pre-coolers, and developing a new method to analyze the "evaporative effectiveness" of these pre-coolers that measures the actual power and efficiency changes. The pre-coolers tested were not particularly effective and are not to be considered representative of the pre-cooler market. An analysis of the uncertainty of the calculated evaporative effectiveness and water use effectiveness showed greater uncertainty than desired. Post-calibration of select sensors and re-calculation of the uncertainty should improve these results.

Continued development of the pre-cooler test protocol will likely require additional testing of pre-coolers. Future testing would likely occur on a small RTU with a one-sided condenser face. This would allow easier testing of commercial pre-cooler products. WCEC has identified at least three commercial products for testing whose designs have significant differences from the residential products tested here. A test chamber is currently being constructed to test up to 5-ton capacity systems. The chamber is being designed to operate in two modes. The first mode is full recirculation mode, where heating and cooling coils are used to offset the thermal and moisture load provided by the equipment under test. Under most conditions, this mode will require the least amounts of thermal energy, and loading will be less than the capacity of the installed chiller and boiler. In recirculation mode, the chamber has been designed to maintain temperatures between 60°F and 120°F and humidity levels in the .005 - .012 $lb_{Water}/lb_{Dry\ Air}$ range. The second mode is a pass-through mode. In this mode, ambient air is taken from the outside and conditioned to the desired conditions, passed over the equipment and discharged again to the outside. This mode will allow testing at higher airflow rates (up to 8,000 CFM) under temperature and humidity constraints that are dependent on the outdoor air conditions.

The results from this project and future research will be used by the recently formed ASHRAE Standards Technical Committee 5.7 to develop an ASHRAE test standard for evaporative condenser air pre-coolers.

Communication and coordination with California utilities is needed to ensure that the results of the test standard and future testing can be used to develop specifications and qualify pre-coolers for incentive programs. WCEC has been participating in, and plans to continue, a pre-cooler specification working group led by Pacific Gas and Electric. A standard laboratory test protocol and test results should reduce the amount of field verification that is needed for incentivizing pre-cooler installations.

APPENDIX A

UNCERTAINTY ANALYSIS: SAMPLE CALCULATION

The uncertainty of all calculations used in this report were conducted using the sequential perturbation method[3], which is a numerical approach that utilizes a finite difference method to approximate the derivatives which represent the sensitivity of the calculated value to the variables used within the calculation. 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. Example uncertainty calculations for product B at the psychometric conditions of test point 3A (105/73 °F DB/WB) are shown for two methods of calculations: the COP baseline method, and the exhaust sensor method.

COP BASELINE METHOD FOR PRODUCT B, TEST POINT 3A (105 ˚F DB/WB)

Sequential perturbation is used to determine the error in all of the calculations. This process involves calculating an initial result R_o, based on measured values. Next, an independent variable within the equation for R_o is increased by its respective uncertainty, and a new value, R_i^+ is calculated (Equation 16).

EQUATION 16: R_i^+ [3]

$$R_1^+ = f\ x_1 + u_{x1}, x_2, \dots \dots .., x_L$$
$$R_2^+ = f\ x_1, x_2 + u_{x2}, \dots \dots .., x_L$$
$$R_L^+ = f(x_1, x_2, \dots \dots .., x_L + u_{xL})$$

Where u_{xi} is the uncertainty of variable *i*. After this, the same independent variable within R_o is decreased by its respective uncertainty, and a new value, R_i^- is calculated (Equation 17).

EQUATION 17: R_i^- [3]

$$R_1^- = f\ x_1 - u_{x1}, x_2, \dots \dots .., x_L$$
$$R_2^- = f\ x_1, x_2 - u_{x2}, \dots \dots .., x_L$$
$$R_L^- = f(x_1, x_2, \dots \dots .., x_L - u_{xL})$$

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 8. The uncertainty of the equation for COP was calculated using Equation 1 and is shown in Table 19. The COP calculation has four independent variables: evaporative load water flow rate, evaporative water temperatures before and after the heat exchanger, and the power consumed by the condensing unit. Each of these independent variables has an instrumentation uncertainty that was found in the manufacturer published data for each sensor used. The resulting uncertainty of the COP for Pre-Cooler B at test conditions 3A was ±0.13˚F.

TABLE 19: UNCERTAINTY CALCULATIONS FOR COP (PRE-COOLER B FOR TEST CONDITIONS 3A)

PROPAGATION VARIABLE	NOMINAL VALUE	UNCERTAINTY	R_O (COP)	R_i^+	R_i^-	δR_i^+	δR_i^-	δR_i
Evap. Flow Rate (gpm)	0.21	0.0041	3.05	3.12	2.99	0.06	(0.06)	0.06
Hot Water Temp (˚F)	80.0	0.4	3.05	3.13	2.97	0.08	(0.08)	0.08
Cold Water Temp (˚F)	62.92	0.4	3.05	2.97	3.13	(0.08)	0.08	0.08
Power (kW)	2.68	0.03	3.05	3.02	3.08	(0.03)	0.03	0.03
							u_{COP} 105/73	**0.13**

In order to calculate the uncertainty in the evaporative effectiveness calculations (Equation 2), it is necessary to know the uncertainty of the wet bulb temperature entering the pre-cooler apparatus as well as the uncertainty of using a linear regression model for the COP baseline in Excel to calculate $T_{DB,eq}$. The uncertainty in calculating the wet bulb temperature from the measured dry bulb and dew point temperatures in the chamber are shown in Table 20. The sensors used for measuring these two temperatures both have an instrument uncertainty of 0.36°F. The wet bulb temperature is calculated using the psychometric calculator function in Excel, and the inputs are the dry bulb and dew point temperatures and their associated instrument uncertainties. The resulting uncertainty of the wet bulb temperature for Pre-Cooler B at test conditions 3A was ±0.19°F.

TABLE 20: UNCERTAINTY CALCULATIONS FOR WET BULB TEMPERATURE (PRE-COOLER B FOR TEST CONDITIONS 3A)

PROPAGATION VARIABLE	NOMINAL VALUE	UNCERTAINTY	R_O (TWB)	R_i^+	R_i^-	δR_i^+	δR_i^-	δR_i
Dry Bulb Temp (˚F)	104.95	0.36	73.30	73.40	73.20	0.10	(0.10)	0.10
Dew Point Temp (˚F)	58.10	0.36	73.30	73.46	73.14	0.16	(0.16)	0.16
							uT_{WB} 105/73	**0.19**

The uncertainty behind calculating $T_{DB,eq}$ was determined by propagating by two independent variables: the chamber dry bulb temperature and the uncertainty in using the linear regression model in Excel. The model and uncertainty in using the linear regression was determined by using Excel's LINEST function using the COP as the x-values and the chamber dry bulb temperature as the y-values. This outputs the coefficients of the linear regression curve as well as the standard error of the y-estimate. This standard error of the y-estimate was determined for Pre-Cooler B to be 1.0141°F. This error was used as the uncertainty value in the model. The uncertainty in the COP calculation was tabulated in

Table 19 and determined to be 0.13. The results of the uncertainty calculations for $T_{DB,eq}$ are shown in Table 21.

TABLE 21: UNCERTAINTY CALCULATIONS FOR $T_{DB,EQ}$ (PRE-COOLER B FOR TEST CONDITIONS 3A)

PROPAGATION VARIABLE	NOMINAL VALUE	UNCERTAINTY	R_O ($T_{DB,EQUIV}$)	R_i^+	R_i^-	δR_i^+	δR_i^-	δR_i
COP	3.05	0.13	96.36	98.96	93.84	2.6	(2.52)	2.56
y estimate		1.0141	96.36	97.37	95.35	1.01	(1.01)	1.01
							$uT_{DB,equiv}$ 105/73	**2.76**

With these uncertainties calculated, the uncertainty in calculating the evaporative effectiveness (Equation 2) was determined with independent variables of T_{DB}, $T_{DB,eq}$, and T_{WB}. These results are shown in Table 22 and the EE uncertainty was found to be ±0.09 for pre-cooler B in test conditions 3A.

TABLE 22: UNCERTAINTY CALCULATIONS FOR EE (PRE-COOLER B FOR TEST CONDITIONS 3A)

PROPAGATION VARIABLE	NOMINAL VALUE	INSTRUMENT UNCERTAINTY	R_O (EE)	R_i^+	R_i^-	δR_i^+	δR_i^-	δR_i
T_{DB} (°F)	104.95	0.36	0.271	0.280	0.263	0.008	(0.008)	0.008
$T_{DB,eq}$ (°F)	96.36	2.76	0.271	0.184	0.358	0.087	(0.087)	0.087
T_{WB} (°F)	73.30	0.19	0.271	0.273	0.270	0.002	(0.002)	0.002
							uEE 105/73	**0.09**

For the uncertainty calculations for the WUE, it is first necessary to determine the uncertainty in calculating the humidity ratio entering and leaving the pre-cooler apparatus. The humidity ratio entering the pre-cooler was determined using the Psych function in Excel with the dry bulb and dew point temperatures. Both of these measurements had instrument uncertainties of 0.36˚F. The resulting uncertainty of the humidity ratio entering the pre-cooler at test conditions 3A was ±0.000136 (Table 23).

Table 23: Uncertainty Calculations for HRin (Pre-Cooler B for Test Conditions 3A)

Propagation Variable	Nominal Value	Uncertainty	R_O (EE)	R_i^+	R_i^-	δR_i^+	δR_i^-	δR_i
T_{DB} (°F)	104.95	0.36	1.031E-02	1.031E-02	1.031E-02	5.000E-08	(5.000E-08)	5.000E08
T_{DP} (°F)	58.1	0.36	1.031E-02	1.045E-02	1.018E-02	1.363E-04	(1.347E-04)	1.355E04
							uHRin 105/73	**0.000136**

The humidity ratio of the exhaust was calculated using the wet bulb temperature in the chamber (which is assumed to be constant across the evaporative pre-cooler) and the equivalent dry bulb temperature that was calculated using the COP baseline model. The results are shown in Table 24 and the uncertainty was found to be ±0.00066 lbw/lba.

Table 24: Uncertainty Calculations for HRout (Pre-Cooler B for Test Conditions 3A)

Propagation Variable	Nominal Value	Uncertainty	R_O (HRout)	R_i^+	R_i^-	δR_i^+	δR_i^-	δR_i
$T_{DB,eq}$ (°F)	96.36	2.76	0.01229	0.01166	0.01293	(0.00064)	0.00064	0.00064
T_{WB} (°F)	73.30	0.19	0.01229	0.01245	0.01213	0.00016	(0.00016)	0.00016
							uHRout 105/73	**0.00066**

Finally, the uncertainty of the WUE (Equation 7) can be determined. The WUE has four independent variables that are: the humidity ratio entering and leaving the pre-cooler, the mass flowrate of the air passing through the pre-cooler, and the flowrate of water that was consumed by the pre-cooler. The results are shown in Table 25 and the uncertainty of the WUE was calculated as ±0.17 for pre-cooler B at test conditions 3A.

Table 25: Uncertainty Calculations for WUE (Pre-Cooler B for Test Conditions 3A)

Propagation Variable	Nominal Value	Uncertainty	R_O (WUE)	R_i^+	R_i^-	δR_i^+	δR_i^-	δR_i
HRin (LBw/LBa)	0.010312	0.0014	0.492	0.458	0.525	(0.03)	0.03	0.03
HRout (LBw/LBa)	0.012292	0.0066	0.492	0.655	0.329	0.16	(0.16)	0.16
Mair (SCFM)	2465	51.41	0.492	0.502	0.481	0.01	(0.01)	0.01
Mwater,supplied (gpm)	0.0086	0.026	0.492	0.477	0.507	(0.01)	0.02	0.01
							uWUE 105/73	**0.17**

EXHAUST METHOD FOR PRODUCT B, TEST POINT 3A (105 ˚F DB/WB)

The method for calculating the uncertainties using the exhaust dry bulb temperature and relative humidity measurements also use sequential perturbation but do not follow the same procedure as the method for COP baseline model calculations. The two main differences in the uncertainty calculations are:

The humidity ratio leaving the pre-cooler is calculated using the exhaust air dry bulb temperature and relative humidity measurements.

The equivalent dry bulb temperature is calculated using the Psych function in Excel with the above humidity ratio and the wet bulb temperature entering the pre-cooler apparatus.

The first uncertainty value calculated is for the chamber wet bulb temperature, which is the same as for the COP baseline method and is outlined in Table 20. The humidity ratio after the pre-cooler apparatus is calculated using the exhaust air dry bulb temperature and relative humidity measurements. The results are shown in Table 26.

TABLE 26: UNCERTAINTY CALCULATIONS FOR HROUT USING EXHAUST METHOD (PRE-COOLER B FOR TEST CONDITIONS 3A)

PROPAGATION VARIABLE	NOMINAL VALUE	UNCERTAINTY	R_O (HROUT)	R_i^+	R_i^-	δR_i^+	δR_i^-	δR_i
$T_{DB,ex}$ (˚F)	110.68	0.36	0.012318	0.012449	0.012188	0.000131	(0.000130)	0.000130
RH	0.22	0.02	0.012318	0.013465	0.011174	0.001148	(0.001144)	0.001146
							uHRout 105/73	**0.001153**

The next uncertainty calculation was for the equivalent dry bulb temperature, which has two independent variables in the calculation: wet bulb temperature entering the pre-cooler and the humidity ratio of the exhaust. These results are shown in
Table 27.

TABLE 27: UNCERTAINTY CALCULATIONS FOR $T_{DB,EQ}$ USING EXHAUST METHOD (PRE-COOLER B FOR TEST CONDITIONS 3A)

PROPAGATION VARIABLE	NOMINAL VALUE	UNCERTAINTY	R_O ($T_{DB,EQ}$)	R_i^+	R_i^-	δR_i^+	δR_i^-	δR_i
T_{WB} (˚F)	73.30	0.19	96.24	97.32	95.17	1.08	(1.07)	1.08
HRout (LBw/LBa)	0.012232	0.001153	96.24	91.27	101.24	(4.97)	4.99	4.98
							uDB,eq 105/73	**5.10**

With these calculated, the uncertainty of the EE using the exhaust method was calculated and the results display in Table 28 and found to be ±0.16.

TABLE 28: UNCERTAINTY CALCULATIONS FOR EE USING EXHAUST METHOD (PRE-COOLER B TEST CONDITIONS 3A)

PROPAGATION VARIABLE	NOMINAL VALUE	UNCERTAINTY	R_O (EE)	R_i^+	R_i^-	δR_i^+	δR_i^-	δR_i
T_{DB} (˚F)	104.95	0.36	0.275	0.283	0.267	0.01	(0.01)	0.01
$T_{DB,eq}$uiv (˚F)	96.24	5.10	0.275	0.114	0.436	(0.16)	0.16	0.16
T_{WB} (˚F)	73.3	0.19	0.275	0.277	0.273	0.002	(0.002)	0.002
							uEE 105/73	**0.16**

The uncertainty for WUE requires knowing the uncertainty of four variables: HRin, HRout, the mass flowrate of the air across the pre-cooler, and the water flowrate supplied. HRin is the same as for the previous method since it is determined based on the chamber dry bulb and dew point temperatures (Table 23). With these uncertainties all previously calculated, the uncertainty of the WUE was calculated and is shown in Table 29 and determined to be ±0.29.

TABLE 29: UNCERTAINTY CALCULATIONS FOR WUE USING EXHAUST METHOD (PRE-COOLER B TEST CONDITIONS 3A)

PROPAGATION VARIABLE	NOMINAL VALUE	UNCERTAINTY	R_O (WUE)	R_i^+	R_i^-	δR_i^+	δR_i^-	δR_i
HRin (lbw/lba)	0.010312	0.0014	0.498	0.465	0.532	(0.03)	0.03	0.03
HRout (lbw/lba)	0.012318	0.00115	0.498	0.785	0.212	0.29	(0.29)	0.29
Mair (SCFM)	2465	51.41	0.498	0.509	0.488	0.01	(0.01)	0.01
Mwater,supplied (gpm)	0.00544	0.00163	0.492	0.477	0.507	(0.01)	0.02	0.01
							uWUE 105/73	**0.29**

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