Emerging Products

Characterizing the Transient and Aggregate Response of Dispatchable Condenser Air Pre-Coolers

DR15.20.00 Report

Prepared by:

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Customer Service
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SOUTHERN CALIFORNIA EDISON

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Disclaimer

EXECUTIVE SUMMARY

INTRODUCTION

Evaporative condenser air pre-coolers, or *"pre-coolers,"* use evaporative cooling methods to pre-cool the inlet air to the condenser of an air conditioning system. This results in a lower temperature for air conditioner heat rejection, reducing power demand and increasing air conditioning process efficiency. While significant work has been done to demonstrate pre-coolers as an energy efficiency measure, this project studied the technology as a dispatchable demand response resource, controlled by the utility for grid management.

APPROACH

In a laboratory setting, the time response impact on a four-ton rooftop unit (RTU) during pre-cooler startup, operation, and shut down was measured at three different outdoor conditions (Table 1). Response time is a key factor in determining whether pre-coolers are practical tools for achieving dispatchable demand reduction.

In field testing, five pre-coolers were installed on a big box retail store in Southern California, in addition to the six that already existed on the building. The research team used the manufacturer's control system to dispatch all 11 pre-coolers 26 times when the outside air temperature was ≥ 95°F between July 1 and October 31, 2017. Dispatch events ranged in length from 60 to 240 minutes, and started at varying times between 11:00 a.m. and 7:00 p.m. Out of the 26 events, 16 were excluded from the analysis, either because they overlapped with actual SCE demand-shedding events, or the dispatch signal failed and none of the RTUs responded. We measured the dispatched RTUs' aggregate power response and water use to analyze the power savings potential of using pre-coolers as demand response assets.

RESULTS

Our laboratory test results show most evaporative pre-cooler power reduction benefits are achieved quickly, within less than two minutes of turning on the pre-cooler (Table 1). This demonstrates that power reductions from pre-coolers could be useful to utilities as a demand response and grid management tool. Capacity increases take longer to achieve and stabilize after approximately 15 minutes. Equipment operating at increased capacity will reduce equipment run times, although equipment cycling behavior is less predictable than the quick drop in RTU power consumption that occurs when a pre-cooler is turned on.

TABLE 1: TRANSIENT RESPONSE TIME FOR "PRE-COOLER ON"

TEST OUTDOOR AIR (DRY-BULB/WET-BULB) °F	Time to achieve 50% of maximum power reduction (min)	Time to achieve 75% of maximum power reduction (min)	Time to achieve 100% of maximum power reduction (min)
95/70	0.6	1.6	13.4
105/73	0.6	0.9	12.5
115/76	0.6	1.1	13.3

Field testing illustrated that dispatched pre-coolers can reduce RTU power usage; however, the aggregate response differs based on the number of dispatched RTUs. Across the 10 analyzed events, the number of RTUs that dispatched varied from 10% to 89%, and the aggregate power reduction was between 0.07kW and 17.56 kW (Table 2). These results illustrate that improvements are needed to increase the reliability of dispatch controls, which will ensure RTUs successfully dispatch during each event and improve overall results.

TABLE 2: SUMMARY OF FIELD TESTING EVENTS WITH LARGEST AND SMALLEST POWER REDUCTION

EVENT STATISTIC	07-08 11:00	07-22 16:00	08-27 15:00	08-30 12:00
DURATION (MINUTES)	60	120	60	120
AVERAGE OUTDOOR AIR TEMPERATURE (°F)	96.5	88.2	97.1	103.1
NUMBER OF RTUS IN COOLING MODE	9	11	10	8
NUMBER OF RTUS THAT DISPATCHED	8	9	1	7
PERCENT OF RTUS DISPATCHED (%)	89	82	10	88
WATER USE (GAL)	196.7	297.6	17.6	391.0
POWER DIFFERENCE BETWEEN EVENT AND BASELINE (KW)	17.56	2.87	0.07	17.05
PERCENT DIFFERENCE FROM BASELINE TO EVENT (%)	12.7	2.5	0.4	14.9
WATER USE / ENERGY SAVED (GAL/KWH)	11.2	51.90	249.05	11.47

RECOMMENDATIONS

The laboratory results illustrate that at steady-state operation, the technology can reduce demand quickly, and by up to 25%. This trend was not as clear in field testing, because the RTUs could change cooling modes during events, potentially delaying cooling until shortly after event start. The field results did not demonstrate the full potential of dispatchable evaporative pre-cooling, because the controls did not reliably dispatch the pre-coolers for every available RTU. Based on these results, we recommend the technology be considered for further study to identify the control changes that would be required to ensure all available units successfully dispatch.

ABBREVIATIONS AND ACRONYMS

AHRI	Air-Conditioning, Heating, and Refrigeration Institute
AMP	Aggregator Managed Portfolio
ANSI	American National Standards Institute
ASHRAE	American Society for Heating, Refrigeration, and Air Conditioning Engineers
BTU	British thermal unit
CFM	Cubic Feet per Minute
COP	Coefficient of Performance
DB	Dry Bulb
ESP	External Static Pressure
GAL	Gallon
Hz	Hertz
In H2O	Inches of Water
KW	Kilowatt
LBA	Pound of Dry Air
LBW	Pound of Water Vapor
NI	National Instruments
OA	Outside Air
PID	Proportional-Integral-Derivative
PSI	Pound per Square Inch
RA	Return Air
RH	Relative Humidity
RMS	Root Mean Squared

RTD	Resistance Temperature Device
RTU	Rooftop Unit
SA	Supply Air
SCE	Southern California Edison
SHR	Sensible Heat Ratio
V	Volts
WB	Wet Bulb
WCEC	Western Cooling Efficiency Center

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EQUATIONS

INTRODUCTION

This report describes a laboratory and field evaluation of an evaporative condenser air pre-cooler, or "pre-cooler," to improve the efficiency of packaged air-conditioning and heating RTUs. The pre-cooler used evaporative cooling methods to pre-cool the inlet air to the air conditioning system's condenser. This resulted in a lower temperature for air conditioner heat rejection, reducing power demand and increasing air conditioning process efficiency. In laboratory testing, a four-ton RTU was retrofitted with a condenser air evaporative pre-cooler, and we characterized the transient response and load reduction achieved by turning the pre-cooler on and off. In field testing, 11 RTUs were retrofitted with condenser-air evaporative pre-coolers. This technology is being considered as a method of saving energy associated with air conditioning at part load conditions, reducing peak electricity demand, and permanently reducing electric load.

BACKGROUND

While significant work has been done to demonstrate pre-coolers as an energy efficiency measure, this project studied the technology as a dispatchable demand response resource, controlled by the utility for grid management purposes[1]. Operating pre-coolers as a dispatchable demand response resource should be explored for the following reasons:

1. Utility peak summer demand is closely tied to air conditioning loads.
2. The demand reduction and efficiency gains for pre-coolers are largest at high outdoor ambient temperatures.
3. The water used for pre-coolers in relationship to the energy saved (known as the water-use efficiency index) is smallest at high outdoor ambient temperatures.
4. The value of the pre-cooler to the utility is higher when it can be operated as a verifiable dispatchable resource, as opposed to an energy efficiency measure.

ASSESSMENT OBJECTIVES

The objective of this project was to conduct laboratory and field testing to evaluate the performance of an evaporative condenser air pre-cooler retrofit package, used by a utility as a dispatchable grid management demand response resource.

For laboratory testing, the combined package was installed on a four-ton commercial York RTU. We evaluated the transient response and load reduction that occurred when the pre-cooler was turned on and off during three different California outdoor air climate conditions.

For field testing, the condenser air pre-cooler package was installed on 11 RTUs that served a big box retail store in Corona, California. The data collection period ran from July 1, 2017 through October 31, 2017, and consisted of 26 remotely-triggered demand response events when the outside air temperature was above 95°F.

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[1] Jonathan Woolley and Thomas Jawin. *The wholesale market value of dispatchable efficiency for commercial air conditioning.* December 15th, 2015. http://wcec.ucdavis.edu/wp-content/uploads/2016/02/The-wholesale-market-value-of-dispatchable-efficeincy-for-commercial-air-conditioning.pdf

TECHNOLOGY DESCRIPTION

BASELINE HVAC EQUIPMENT

LABORATORY TEST

For laboratory testing, we used a commercial YORK RTU (model #D6NZ048N06525NX, refrigerant R-410A) with nominal four-ton cooling capacity.

FIELD TEST

We conducted field testing on 11 Lennox RTUs (10 L-Series models and one Strategos model) serving a big box store in Corona, California. Four of the RTUs had nominal 10-ton cooling capacity, while seven had nominal 20-ton capacity (Table 3).

CONDENSER AIR EVAPORATIVE PRE-COOLER

Evaporative condenser air pre-coolers function by evaporating water to pre-cool inlet air to air conditioning system condensers. This lowers the temperature of air conditioner heat rejection, reduces the amount of power needed, and improves the efficiency of the air conditioning process. Condenser air pre-coolers are most effective in dry climates, and can reduce the condenser air dry-bulb temperature by as much as 20°F. For both phases of the project, we used a pre-cooler with 8″ deep corrugated cellulose media (Figure 2) to evaporate water with a demonstrated evaporative effectiveness of >70% [1].

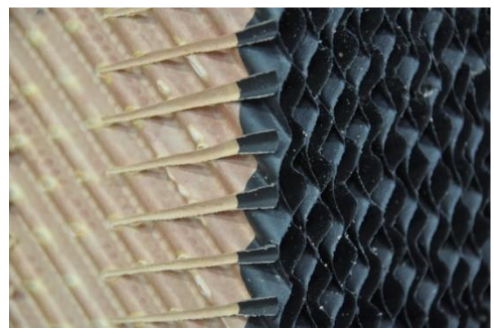

FIGURE 1 – PRE-COOLER CELLULOSE MEDIA

LABORATORY TEST EQUIPMENT

The pre-cooler used for laboratory testing was mounted in front of the laboratory RTU's condenser coil, and was surrounded by a hood made of sheet metal and corrugated plastic, to force all condenser air through the pre-cooler prior to it being pulled through the condenser coil. The pre-cooler had the same surface area as the condenser coil, and consisted of 48″ wide by 46 ½″ tall corrugated cellulose paper media with a thickness of 8″, a sump to store the water, and a pump to circulate the water over the media. The pre-cooler evenly distributed the water over the top of the evaporative media through the use of a perforated top-distribution pipe, which then ran down the media before draining back to the sump. The make-up water was supplied to the pre-cooler at 80°F and 60 psi, and the water level in the pre-cooler was maintained by a copper float valve. The pre-cooler pump was controlled by the research team to be on or off, based on the test matrix.

FIELD TEST EQUIPMENT

Each of the 11 field test condenser air pre-coolers was composed of corrugated cellulose paper media with a thickness of 8″, a sheet metal frame, a sump to store the water, a perforated pipe to evenly distribute water over the cellulose media, and a pump to circulate water over the media. The surface area ratio of the condenser coil to the condenser air pre-cooler varied by RTU type (Table 3). The big box retail store field testing site selected for this project already had 6 of the 11 pre-coolers installed. The existing systems included a ventilation-air pre-cooler and controls capable of coordinating with the store's air handler units, to shift ventilation load to the six pre-coolers based on outdoor air conditions. For systems with ventilation-air pre-coolers, the condenser air pre-cooler sump water was pumped through the ventilation-air pre-cooler before it returned, and was distributed over the cellulose media (Figure 2, left).

All field test RTUs had slanted condenser coil configurations. Seven 20-ton RTUs and one 10-ton RTU had two slanted coils in a "V" formation. All of the pre-coolers were installed based on the manufacturer's standard practice for slanted coils. The process included installing the pre-cooler in a vertical orientation and using an adapter kit of additional sheet metal pieces to block the open area and force the condenser air to pass through the pre-cooler (Figure 2 and Figure 3). Current adapter kits are unable to direct air passing through the pre-cooler to the back side of the "V"-shaped condenser coils on the seven 20-ton RTUs. Therefore, the nominal 20-ton RTUs only had pre-cooling for a nominal 10-tons (stage 1 and 2).

The make-up water was supplied to each pre-cooler by a rooftop connection to the store's domestic water system, and the water level in each sump was maintained by a plastic float valve. Each pre-cooler pump was controlled by its manufacturer-installed control system, and could be accessed remotely via the internet. The control system primarily acted as a pass-through for the store's Building Management System (BMS) and activated various parts of the pre-cooler based on BMS signals and local weather data.

When multiple pre-coolers are installed on the same roof, the manufacturer uses a wireless mesh network to connect the individual systems together. For this project, the manufacturer temporarily reprogrammed the individual systems to activate together when the main coordinating control unit was used to schedule a dispatch event.

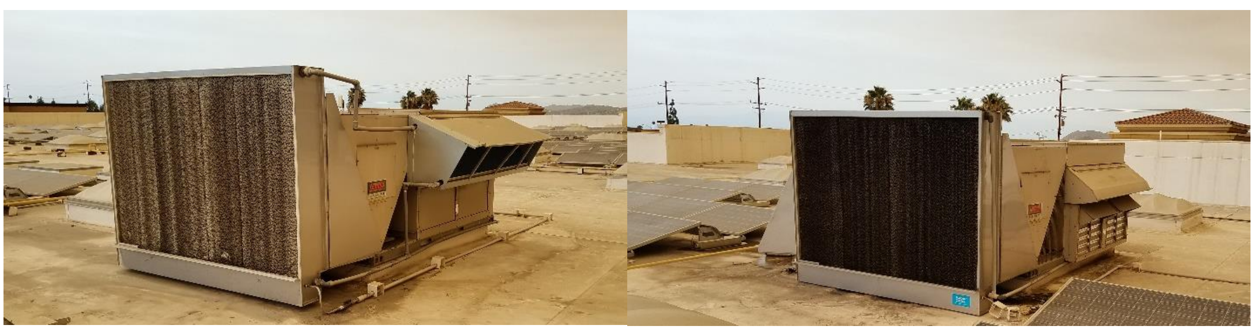

FIGURE 2: CONDENSER AIR PRE-COOLER INSTALLED ON 20-TON RTUS WITH "V" SHAPED CONDENSER COILS, (LEFT), WITH VENTILATION COOLING COIL, (RIGHT), CONDENSER AIR COOLING ONLY

FIGURE 3: CONDENSER AIR PRE-COOLER INSTALLED ON 10-TON RTUS, (LEFT), SLANTED CONDENSER COIL, (RIGHT), "V" SHAPED CONDENSER COIL.

TABLE 3: FIELD TEST EQUIPMENT SUMMARY

UNIT #	BRAND	MODEL	NUMBER OF STAGES	COOLING CAPACITY	SURFACE AREA RATIO[1]	PRE-COOLING RETROFIT
R1	Lennox	Strategos	2	10 ton	0.65[2]	Condenser Air
R8	Lennox	L-Series	4	20 ton	1.05	Condenser Air + Ventilation Air
R9	Lennox	L-Series	2	10 ton	1.07	Condenser Air
R10	Lennox	L-Series	4	20 ton	1.05	Condenser Air + Ventilation Air
R11	Lennox	L-Series	4	20 ton	1.05	Condenser Air + Ventilation Air
R12	Lennox	L-Series	2	10 ton	1.07	Condenser Air
R13	Lennox	L-Series	4	20 ton	1.05	Condenser Air
R14	Lennox	L-Series	4	20 ton	1.05	Condenser Air + Ventilation Air
R15	Lennox	L-Series	4	20 ton	1.05	Condenser Air + Ventilation Air
R16	Lennox	L-Series	4	20 ton	1.05	Condenser Air + Ventilation Air
R23	Lennox	L-Series	2	10 ton	1.07	Condenser Air

[1]Surface Area Ratio = (condenser air pre-cooler area) / (condenser coil(s) area)

[2]One condenser air pre-cooler used to cool air for both sides of the "V"-shaped condenser coils (Figure 2, right).

LABORATORY TESTING: TECHNICAL APPROACH

ENVIRONMENTAL CHAMBER DESIGN AND CONTROL

WCEC built an environmental control chamber in Davis, California, specifically designed to test unitary air conditioners. The primary focus of the laboratory consisted of controlling two conditioned chambers, one 10.5 feet wide, 15 feet long, and 8 feet tall; and another 7 feet wide, 10 feet long, and 8 feet tall. The larger chamber was designed to produce outdoor air conditions, and the smaller one for indoor air conditions.

This equipment could be used to fully control the humidity and temperature of the outdoor chamber air to any temperature between 60°F and 110°F, and to any humidity ratio between 0.005 and 0.013 lbw/lba. The equipment under test could further increase or decrease the temperature of the chamber, if desired. For example, when testing a heat pump in cooling mode, the outdoor air temperature could exceed 110°F, as the test unit was a heat source for the outdoor loop. When testing a heat pump in heating mode, the outdoor air temperature could be lower than 60°F, because the heat pump was a heat sink for the outdoor loop. The limitation on the temperature minimum and maximum was a function of the test equipment, as well as the leakage and heat transfer through the ductwork. The design airflow rate for the outdoor chamber was 240-5,000 cfm, although flow rates up to 8,000 cfm were possible with reduced thermal conditioning capabilities.

We controlled the outdoor air chamber’s temperature and humidity two parallel conditioning paths, with airflow distribution managed by two computer-controlled dampers (Figure 4). One path contained a heating coil supplied by hot water and an evaporative media humidifier, and the other path contained a chilled water coil and a gas-fired desiccant dehumidifier. The hot and chilled water coils had computer-controlled valves to modulate water flow, while the humidifier and dehumidifier had an on/off switch. Modulating the dampers and valve positions allowed for precise chamber humidity adjustment. The final air temperature was controlled by additional hot and chilled water coils prior to the chamber inlet.

The indoor air chamber capabilities were designed to provide heating and humidification when testing cooling systems, and limited cooling and dehumidification capabilities for testing heat pump systems. Modulating dampers and valves allowed for precise chamber temperature and humidity control. The indoor chamber’s designed airflow rate was 240-2500 cfm.

We used Resistance Temperature Devices (RTDs) and chilled mirror hygrometers to measure both chambers’ inlet and outlet temperatures and dew points. Damper actuators and valves were manufactured by Belimo, and were fully controllable over a 2-10V range. Data acquisition inputs, PID algorithms, and control outputs were accomplished using National Instruments CompactDAQ hardware and custom LabVIEW software.

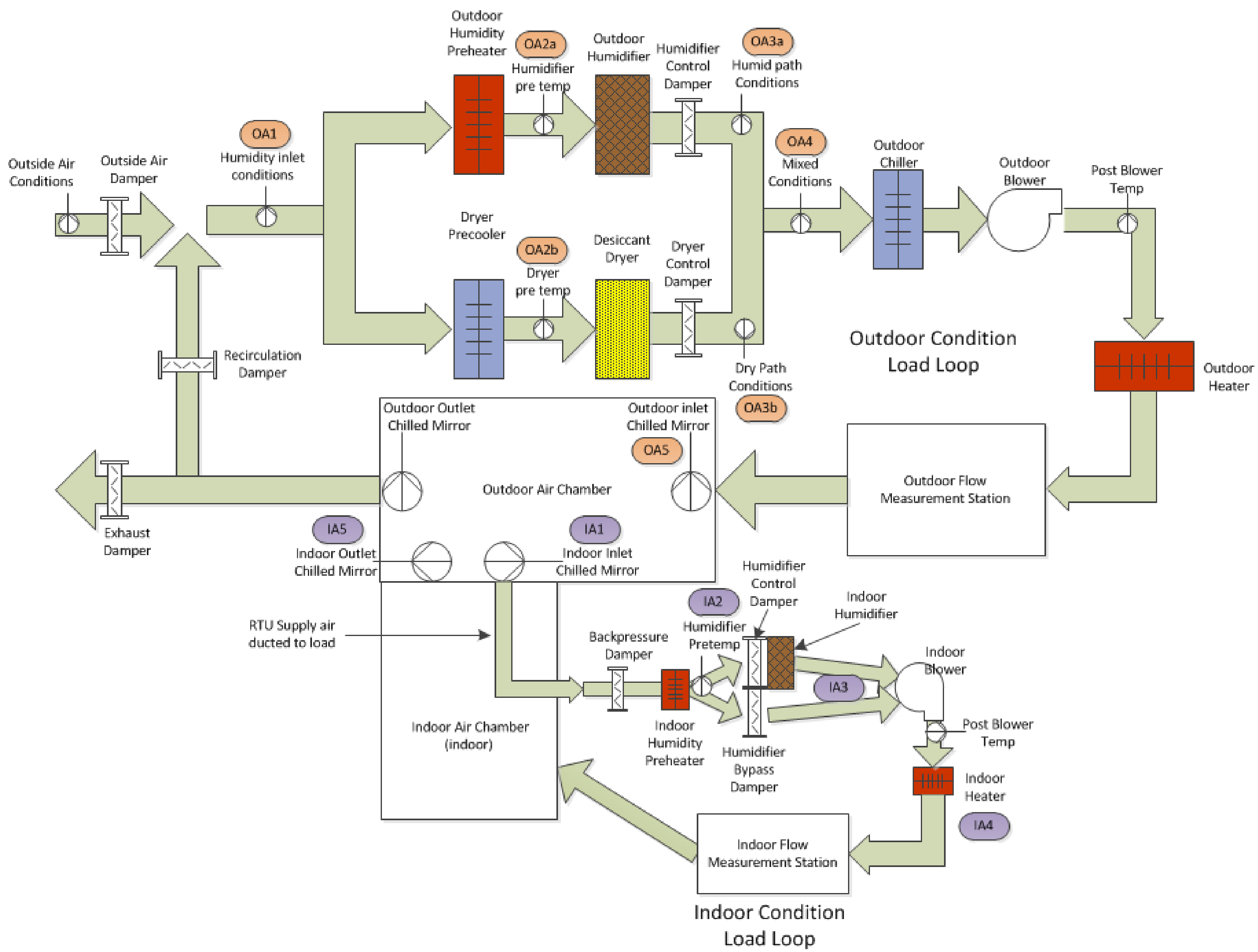

FIGURE 4: SCHEMATIC OF TEST CHAMBERS AND BOTH INDOOR AND OUTDOOR CONDITIONING LOOPS

LABORATORY TESTING: TEST PROTOCOLS

EVALUATION OF BASELINE TECHNOLOGY

In the first phase of testing, a set of baseline tests was obtained for the York four-ton RTU. These tests recorded steady-state system power, capacity, and efficiency for a number of outdoor air dry-bulb test points. The indoor return air condition was set to 80°F/67°F Dry Bulb/Wet Bulb (DB/WB) (Table 4).

TABLE 4: TEST POINTS FOR BASELINE EQUIPMENT

TEST	PRE-COOLER	OUTDOOR CONDITION (°F DB)	INDOOR CONDITIONS (°F DB/°F WB)
B1	None	75	80/67
B2	None	85	80/67
B3	None	95	80/67
B4	None	105	80/67
B5	None	115	80/67

EVALUATION OF PRE-COOLER IN "DRY" STATE

In the second phase of testing, a set of tests was performed for the RTU with the pre-cooler installed, but with the water off (the RTU's operating condition if the utility did not call for a dispatchable resource). The "pre-cooler off" tests recorded steady-state system power, capacity, and efficiency for a number of outdoor air dry-bulb test points. The indoor return air condition was set to 80°F/67°F DB/WB (Table 5). The purpose of these tests was to quantify any negative impacts on air conditioner efficiency (resulting from the increase in resistance to the condenser airflow) produced by the pre-cooler in the "off" state.

TABLE 5: TEST POINTS FOR PRECOOLER IN "DRY" STATE

TEST	PRE-COOLER	OUTDOOR CONDITION (°F DB)	INDOOR CONDITIONS (°F DB/°F WB)
D1	Installed/Dry	75	80/67
D2	Installed/Dry	85	80/67
D3	Installed/Dry	95	80/67
D4	Installed/Dry	105	80/67
D5	Installed/Dry	115	80/67

EVALUATION OF DISPATCHABLE DEMAND RESPONSE

In the third phase of testing, we tested the unit with a pre-cooler installed, to determine the transient response under a few representative weather conditions (Table 6). An example is shown in Figure 5, as an illustration of the test protocol only. We started the test by running the unit with a pre-cooler in a dry condition until a steady state condition (State A) was reached and held for 30 minutes. At this point, the pre-cooler was turned on and run until a new steady state condition was reached (State B) and then held for 30 minutes. Finally, the pre-cooler was turned off, and the unit continued running until the initial steady state condition was seen again (State C) and held for a final 30-minute period.

The complete data set was sampled and saved at five-second intervals to characterize the "pre-cooler on" time response, and five-to-30-second intervals to capture the slower "pre-cooler off" time response. After reviewing the results, the team decided to repeat the "pre-cooler on" test on a different day, to gather higher-resolution RTU current draw data at half-second intervals. The goal was to better characterize the drop in total RTU power usage when the pre-cooler was switched on.

Once the transient time periods (t_{on}) and (t_{off}) were characterized, the research team ran one additional test at the 115/76 outdoor air condition, to determine how the transient response was affected if another dispatch call for the pre-cooler was made before the "transient off" period was satisfied.

TABLE 6: TEST POINTS FOR TRANSIENT RESPONSE

TEST	PRE-COOLER OPERATION	OUTDOOR CONDITION (°F DB/°F WB)	INDOOR CONDITIONS (°F DB/°F WB)
DR1	As shown in Figure 7	95/70	80/67
DR2	As shown in Figure 7	105/73	80/67
DR3	As shown in Figure 7	115/76	80/67

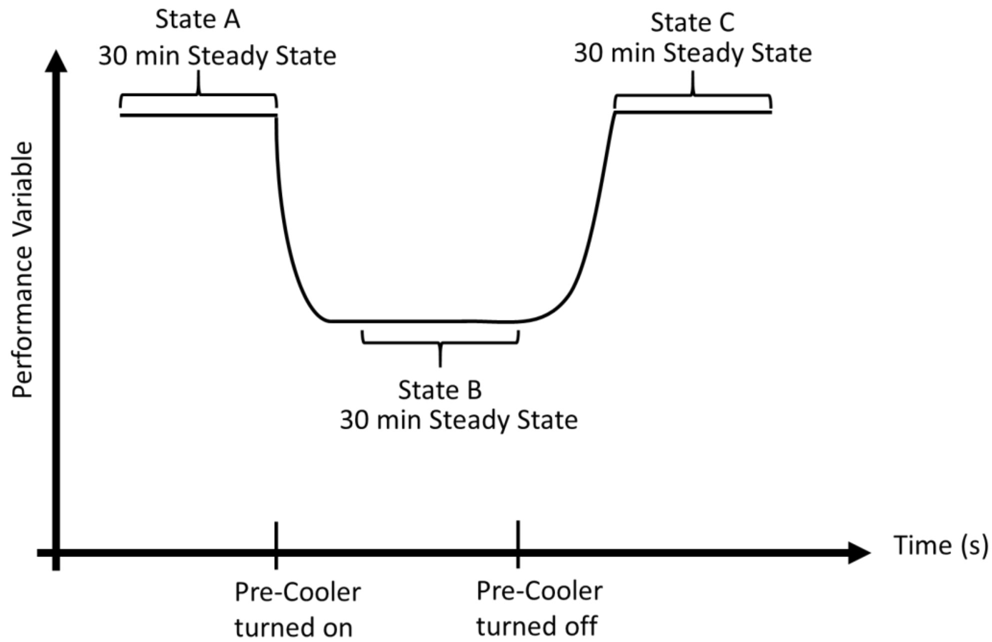

FIGURE 5: EXAMPLE OF A HOW PRE-COOLER RESPONSE WAS CHARACTERIZED.

INSTRUMENTATION PLAN

A York four-ton RTU with refrigerant R-410A was placed inside the conditioned chamber and used for all baseline and dispatchable pre-cooler tests (Figure 6). The measurement locations for each instrument are shown in Figure 7, and details of the instrumentation are summarized in Table 7.

FIGURE 6: TEST UNIT INSTALLED IN THE ENVIRONMENTAL CHAMBER

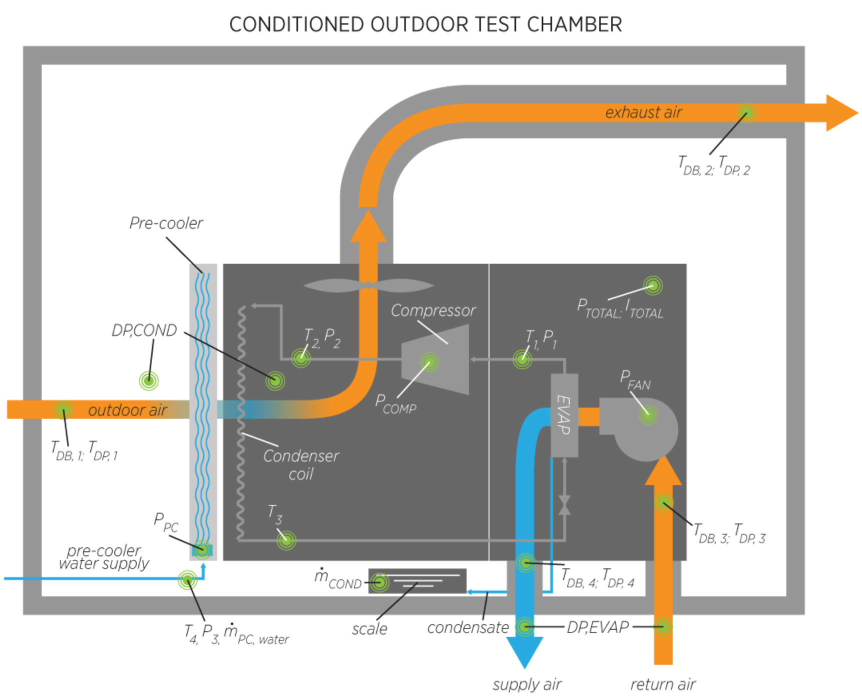

FIGURE 7: MEASUREMENTS FOR RTU AND PRE-COOLER TESTING

TABLE 7: MEASUREMENTS AND INSTRUMENTATION

SYMBOL (FIGURE 2)	MEASUREMENT TYPE	MANUFACTURER AND MODEL #	ACCURACY	SIGNAL TYPE
$T_{DB,1}$	Inlet Outdoor Air Temp	GE Optisonde	±0.3°F	RS-232
$T_{DP,1}$	Inlet Outdoor Air Dew Point Temp	GE Optisonde	±0.4°F	RS-232
$T_{DB,2}$	Exhaust Outdoor Air Temp	GE Optisonde	±0.3°F	RS-232
$T_{DP,2}$	Exhaust Outdoor Air Dew Point Temp	GE Optisonde	±0.4°F	RS-232
$T_{DB,3}$	Return Indoor Air Temp	GE Optisonde	±0.3°F	RS-232
$T_{DP,3}$	Return Indoor Air Dew Point Temp	GE Optisonde	±0.4°F	RS-232
$T_{DB,4}$	Supply Indoor Air Temp	GE Optisonde	±0.3°F	RS-232
$T_{DP,4}$	Supply Indoor Air Dew Point Temp	GE Optisonde	±0.4°F	RS-232
DP_{COND}	Delta P Static (Condenser)	Energy Conservatory APT	1% of reading	RS-232
DP_{EVAP}	Delta P Static (RTU Fan)	Energy Conservatory APT	1% of reading	RS-232
-	Upstream Flow Nozzle Pressure (Indoor Side)	Energy Conservatory APT	1% of reading	RS-232
-	Flow Nozzle Differential Pressure (Indoor Side)	Energy Conservatory APT	1% of reading	RS-232
-	Upstream Flow Nozzle Pressure (Outdoor Side)	Energy Conservatory APT	1% of reading	RS-232
-	Flow Nozzle Differential Pressure (Outdoor Side)	Energy Conservatory APT	1% of reading	RS-232
-	Indoor Chamber Static Pressure	Energy Conservatory APT	1% of reading	RS-232
-	Outdoor Chamber Static Pressure	Energy Conservatory APT	1% of reading	RS-232
P_1	Compressor In Pressure	ClimaCheck 35 Bar	<1% of F.S.	4-20mA
P_2	Compressor Out Pressure	ClimaCheck 35 Bar	<1% of F.S	4-20mA
T_1	Compressor In Temperature	Omega PR-20-2-100	±0.2°F	Ohms
T_2	Compressor Out Temperature	Omega PR-20-2-100	±0.2°F	Ohms
T_3	Condenser Coil Out Temperature	Omega PR-20-2-100	±0.2°F	Ohms

Symbol (Figure 2)	Measurement Type	Manufacturer and Model #	Accuracy	Signal Type
-	Atmospheric Pressure	OMEGADYNE PX409-26BI	±0.08% BSL	4-20mA
P_{TOTAL}, P_{FAN}, P_{COMP}, P_{PC}	RTU Total Power, RTU Indoor Fan Power, RTU Compressor Power, Pre-Cooler Pump Power	Dent PowerScout 18™/Dent PowerScout 3™	±0.5% kW reading	RS-485
I_{TOTAL}	RTU Total Current ("Pre-cooler on" tests only)	Fluke 1735 Power Logger	1% of reading	N/A
$\dot{m}_{COND}$	Condensate Generation	Adam Equipment-GBK 16A –Bench Scale	±0.3 g ±0.006 lb	RS-232
T_3	Pre-cooler Water Temperature	OMEGA RTD	±0.3°F	Ohms
P_2	Pre-cooler Water Pressure	Omega PX209-100AI	0.25% of rdg	4-20mA
$\dot{m}_{PC,water}$	Pre-cooler Water Flow Rate	Omega FTB-4705	1% of reading 0.2-10 GPM	Pulse

Refrigerant Measurements

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

Indoor Coil Temperature and Condensate Generation Rate Measurements

The air supplied to the indoor coil was conditioned by the indoor air chamber. Dry-bulb temperature, wet-bulb temperature, and flow rate were controlled to provide return air at 80/67 (DB°F/WB°F) for all tests, at the manufacturer-specified flow rate for the test unit (1550 CFM).

GE Optisonde chilled mirror hygrometer systems were used to measure the dry-bulb and dew-point temperatures of both the return and supply air. For the dry-bulb measurement, a single temperature was read using an RTD sensor. Well-mixed return air entered the RTU from the indoor chamber. To ensure accuracy, a mixing device was added prior to the supply air RTD measurement. A vacuum pump with a manifold and stainless steel tubing drew air from each stream, to obtain the sample air used for dew-point measurements. The wet-bulb temperature was calculated from the dry-bulb temperature and dew-point.

Generated condensate weight was measured and recorded using a high-accuracy bench scale. The condensate generation rate was used to verify the accuracy of the indoor coil temperature measurements.

Outdoor Coil Temperature Measurements

GE Optisonde chilled mirror hygrometer systems were also used to measure the dry-bulb and dew-point temperatures of the outdoor chamber inlet and exhaust air. For the dry-bulb temperature measurement, a single temperature was read using an RTD sensor. The inlet air temperature and exhaust air stream were well-mixed. The air sampled for the dew-point measurements was obtained using a small vacuum pump that drew from each air stream using a manifold and stainless steel tubing. The wet-bulb temperature was calculated from the dry-bulb temperature and dew-point.

Differential Pressure and Airflow Measurements

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

Differential pressures for the RTU were measured with a second APT-8 pressure transducer. Two channels were used to measure differential pressure across just the condenser coil and evaporator fan with evaporator (total external static pressure). For the differential pressure measurement across the condenser coil, a "free air" measurement was taken, with no ducting attached for the baseline test unit, and with the pre-coolers installed. This measurement was then matched during testing, after the ductwork had been re-attached. Ducting the exhaust air out of the chamber during testing allowed for tighter control of chamber environmental conditions, since the WCEC chamber has a low (8-ft.) ceiling height.

The differential air pressure across the indoor side of the test unit was maintained at a minimum of 0.20 in H_2O, as specified in Table 11 of AHRI/ASHRAE 210/240. The flow rate was measured using the flow nozzle apparatus, and was maintained at the manufacturer's specified flow rate (1550 CFM).

Power Measurements

Measurements for the total power, compressor power, fan power, and pre-cooler pump were recorded using a Dent PowerScout 18 and PowerScout 3, with a serial interface and Modbus protocol. It digitally output data every three seconds. Because this rate was too slow to capture the "pre-cooler on" power time response, a separate current meter (Fluke 1735 Power Logger) was used to sample the total RTU current at a rate of two Hz for the "pre-cooler on" tests only.

Data Acquisition System

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

Tolerances

The goal for all tests was to adhere to the relevant tolerances specified in ANSI/AHRI Standard 210/240-2008 [2], and ASHRAE 37-2009 [3]. These tolerances were used for all baseline tests, and steady-state portions of the transient tests. During transient testing, we were able to keep the indoor and outdoor dry-bulb temperatures within the tolerance. However, turning on the pre-cooler resulted in a temporary drop in the indoor wet-bulb temperature, with the largest drop being 1.2°F for the 115/76 OA test condition. The wet-bulb temperature would then recover as the chamber conditioning system increased humidity generation rates. This drop in the wet-bulb temperature was avoided for the outside air chamber, because outdoor air was conditioned for testing and exhausted back outdoors; therefore, turning on the pre-cooler did not affect the chamber conditioning system.

Two types of tolerances ("range" and "mean") are listed in Table 8. The range tolerance specifies the maximum and minimum limits the controlled variable is allowed, and the mean tolerance specifies the range within which the average value of all recorded test points must fall.

TABLE 8: TEST TOLERANCES

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

EQUATIONS AND ERROR ANALYSIS

From the direct measurements taken capacity, coefficient of performance (total and sensible), and sensible heat ratio were calculated.

TOTAL CAPACITY

The total capacity of the RTU was computed using the change in enthalpy of the air entering and exiting the evaporator coil, as shown in Equation 1.

EQUATION 1: TOTAL CAPACITY

$$Total\ Capacity\ \left[\frac{btu}{hr}\right] = (h_{CI}\ [\frac{btu}{lb}] - h_{CO}\ [\frac{btu}{lb}]) * \frac{Q_C\ \left[\frac{ft^3}{min}\right] * 60[\frac{min}{hr}]}{v_{CI}\ [\frac{ft^3}{lb}] * (1 + W_{CI}\ [\frac{lbw}{lba}])}$$

Where:

h_{CI} is the cold inlet enthalpy calculated from the measured cold inlet dry-bulb temperature and dew point using the ASHRAE handbook of fundamentals

h_{CO} is the cold outlet enthalpy calculated from the measured cold outlet dry-bulb temperature and dew point using the ASHRAE handbook of fundamentals

Q_C is the measured flow rate across the evaporator coil described by ANSI/ASHRAE Standard 41.2-1987 and calculated from the pressures measured in the flow nozzle measurement apparatus.

v_{CI} is the cold inlet specific volume calculated from the measured cold inlet dry-bulb temperature and dew point using the ASHRAE handbook of fundamentals

W_{CI} is the cold inlet humidity ratio calculated from the measured cold outlet dry-bulb temperature and dew point using the ASHRAE handbook of fundamentals

COEFFICIENT OF PERFORMANCE

To determine the COP of the RTU Equation 2 was used.

EQUATION 2: COEFFICIENT OF PERFORMANCE

$$COP = \frac{Total\ Capacity[\frac{btu}{hr}]}{Total\ Power\ [KW] * 3414.42\ [\frac{btu}{hr * KW}]}$$

Where the total power was calculated as the sum of the measured power of the RTU and pre-cooler.

LABORATORY TESTING: RESULTS

BASELINE AND DRY PRE-COOLER PERFORMANCE

Installing the pre-cooler reduced the condenser airflow by less than 2%. We determined the change in the coefficient of performance across a range of outdoor ambient. The decrease in the coefficient of performance was found to be negligible when the dry pre-cooler was added (Figure 8), meaning installing the pre-cooler would not appreciably affect the performance of the air conditioner when the pre-cooler was off. This result only applies to the particular pre-cooler studied; other pre-cooler designs may have different results, depending on the degree of airflow restriction the pre-cooler caused. In addition, the pre-cooler pump consumed 128 W of power that is not included in the results shown.

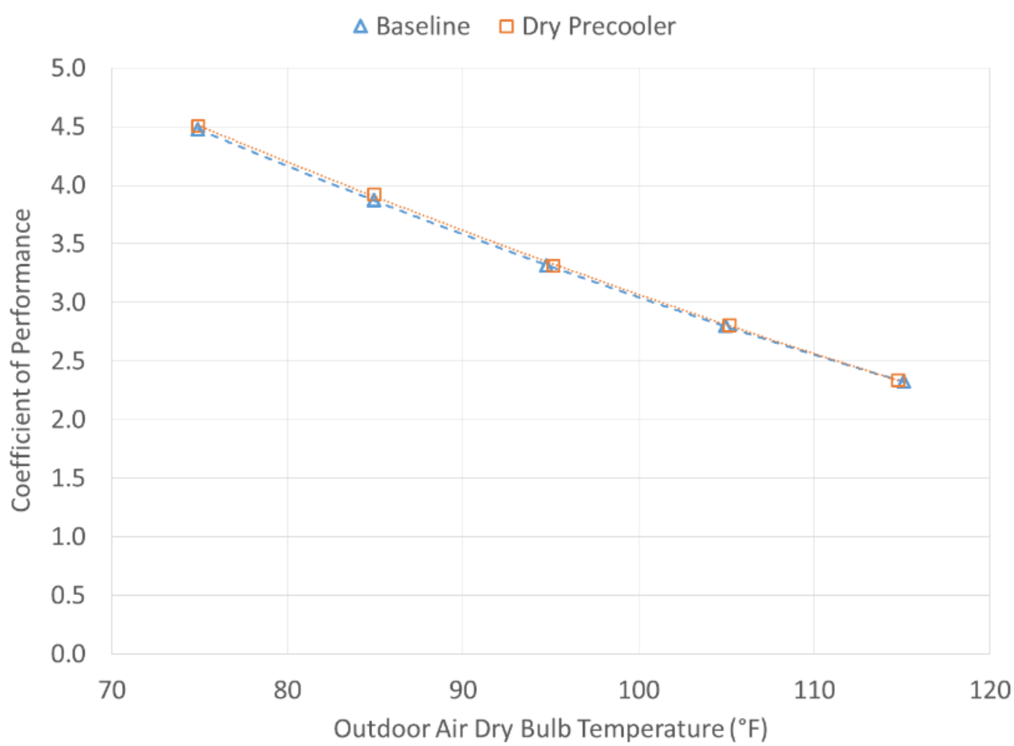

FIGURE 8: COEFFICIENT OF PERFORMANCE FOR BASELINE AND DRY PRE-COOLER

"PRE-COOLER ON" TRANSIENT RESPONSE

The decrease in total power consumption, increase in total capacity, and coefficient of performance were determined when the pre-cooler was turned on for the 95/70, 105/73, and 115/76 outdoor conditions (Figure 9, Figure 10, and Figure 11). We observed a decrease in total power between 16%-25%, and an increase in capacity and coefficient of performance between 11%-21% and 32%-62%, respectively. Lastly, we determined the transient times necessary for the reduction in power, which demonstrate 75% of the maximum power reduction for all test conditions occurred in less than two minutes (Table 9).

"Pre-Cooler Off" Transient Response

The time necessary for the total power, total capacity, and coefficient of performance to return to the dry pre-cooler state ranged between 37 and 43 minutes (Figure 12, Figure 13, Figure 14, and Table 10). Calling the pre-cooler again during the drying process had a negligible effect on performance improvement. The power consumption decrease rate was the same as when the pre-cooler began in the dry state (Figure 15). The time it took for performance improvement to take effect was so small that the further reduction in time (due to the pre-cooler already being partially wet) made it inconsequential.

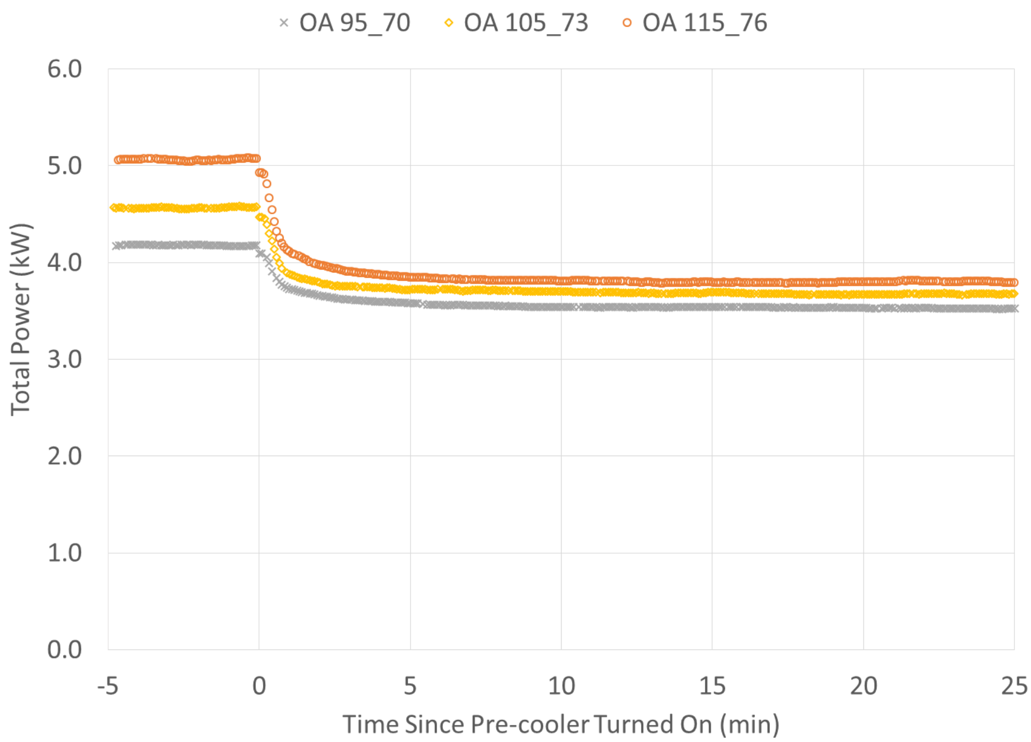

Figure 9: Transient "Pre-Cooler On" power response

Table 9: Transient response time for "Pre-cooler On"

Test Outdoor Air Dry-Bulb/Wet-Bulb °F	Time to achieve 50% of maximum power reduction (min)	Time to achieve 75% of maximum power reduction (min)	Time to achieve 100% of maximum power reduction (min)
OA 95/70	0.6	1.6	13.4
OA 105/73	0.6	0.9	12.5
OA 115/76	0.6	1.1	13.3

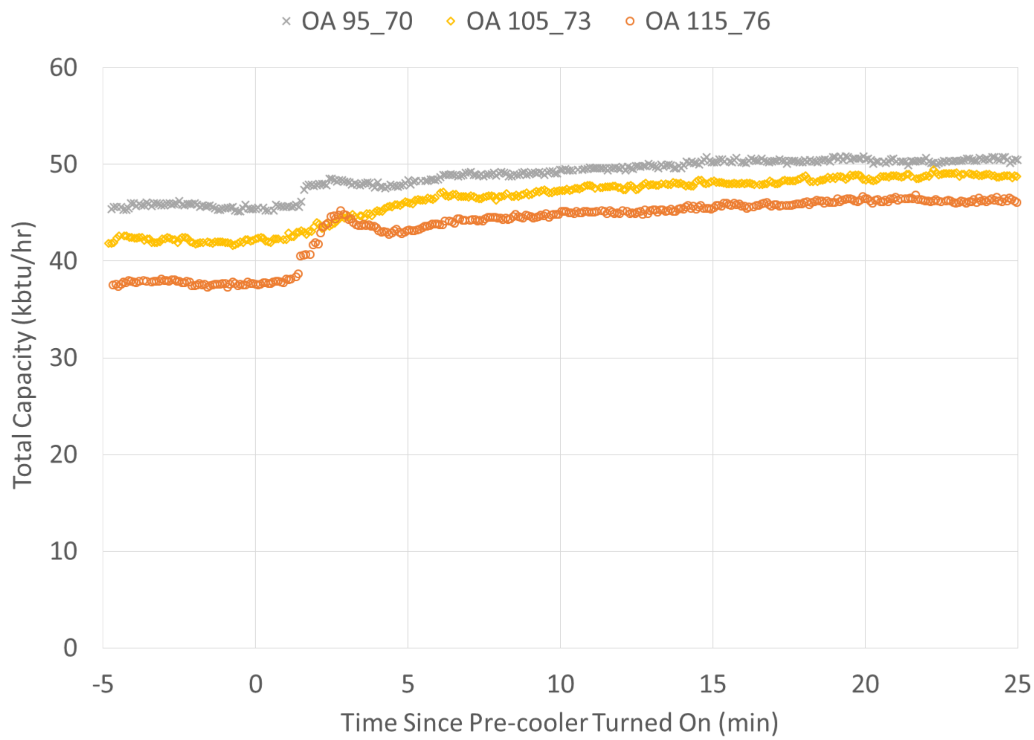

FIGURE 10: TRANSIENT "PRE-COOLER ON" CAPACITY RESPONSE

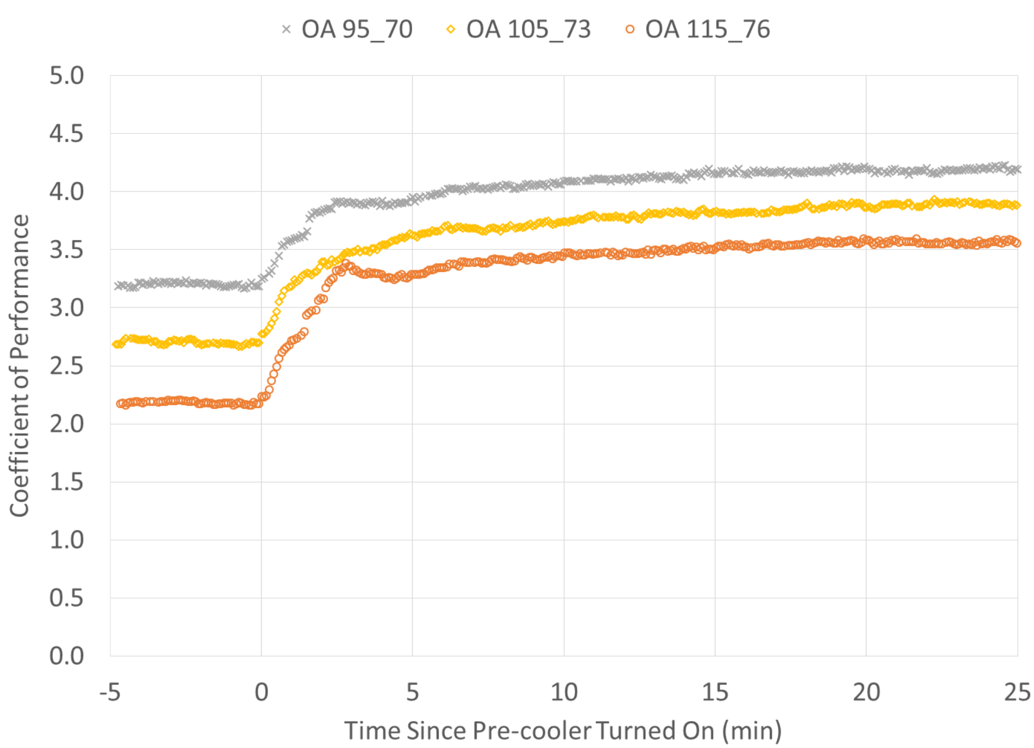

FIGURE 11: TRANSIENT "PRE-COOLER ON" COEFFICIENT OF PERFORMANCE RESPONSE

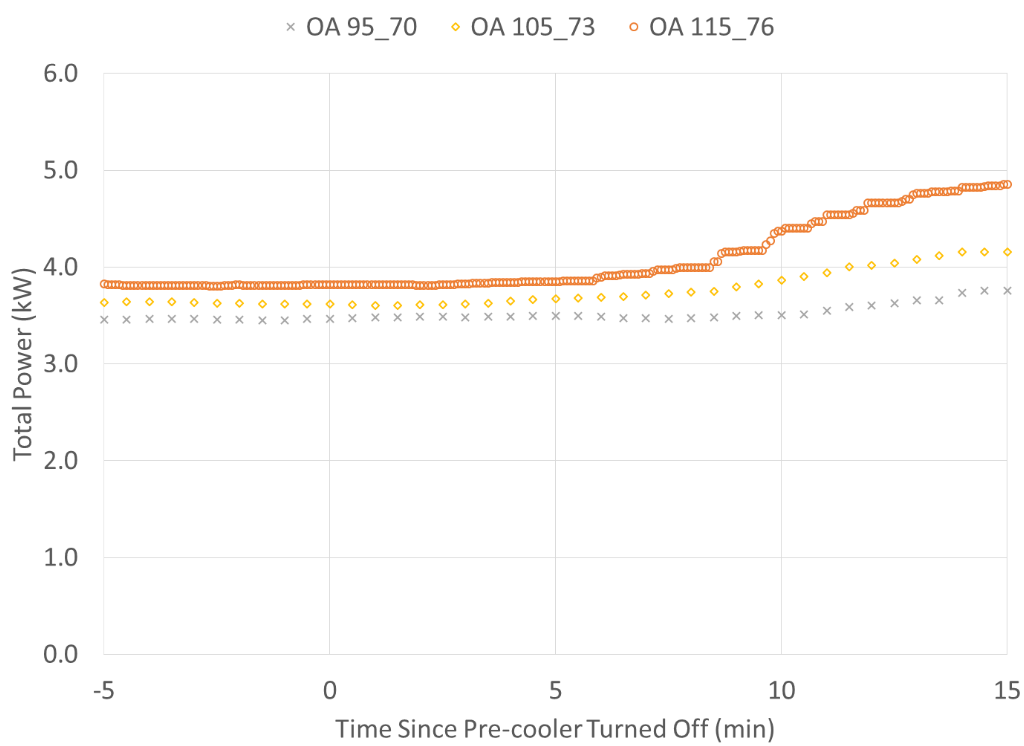

FIGURE 12: TRANSIENT "PRE-COOLER OFF" POWER RESPONSE

TABLE 10: TRANSIENT RESPONSE TIME OF "PRE-COOLER OFF"

TEST OUTDOOR AIR DRY BULB/WET BULB °F	TIME UNTIL POWER RETURNS TO 50% OF OFF STEADY STATE VALUE (MIN)	TIME UNTIL POWER RETURNS TO 75% OF OFF STEAD STATE VALUE (MIN)	TIME UNTIL POWER RETURNS TO 100% OF OFF STEAD STATE VALUE (MIN)
OA 95/70	15.0	19.0	42.5
OA 105/73	16.0	17.0	36.5
OA 115/76	10.7	12.9	40.8

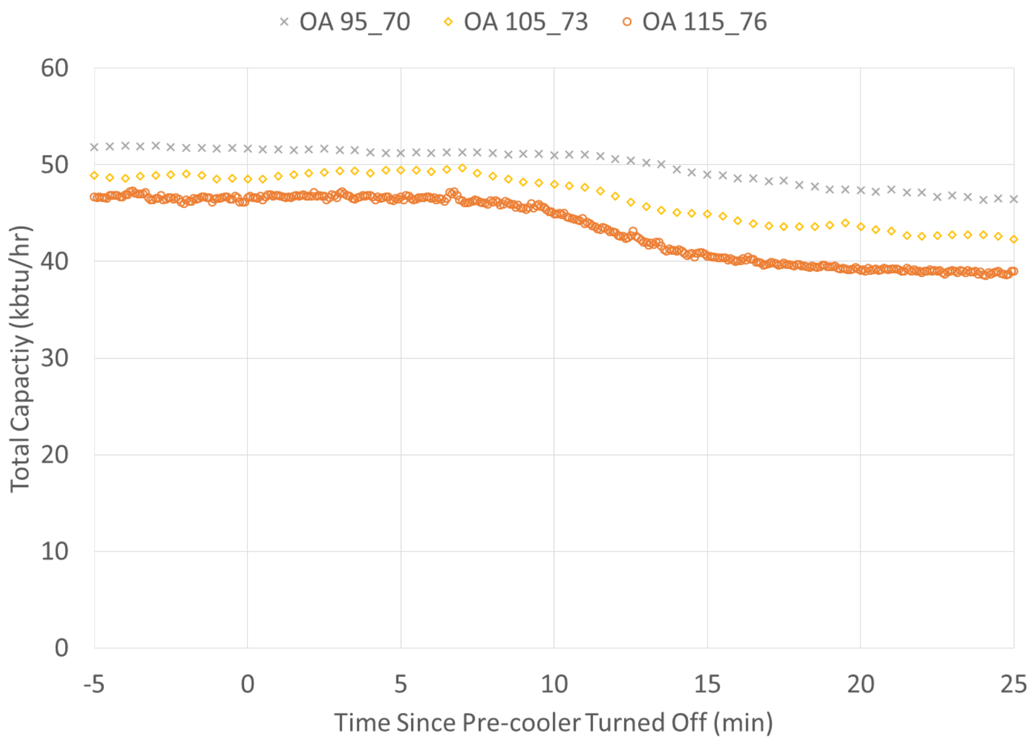

FIGURE 13: TRANSIENT "PRE-COOLER OFF" CAPACITY RESPONSE

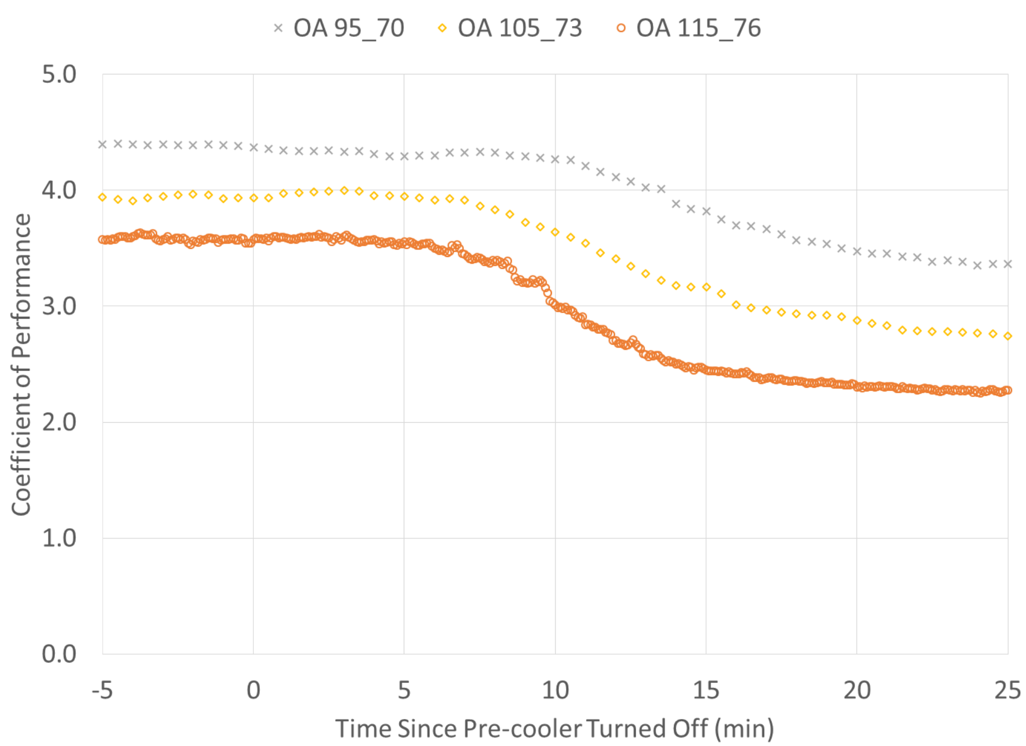

FIGURE 14: TRANSIENT "PRE-COOLER OFF" COEFFICIENT OF PERFORMANCE RESPONSE

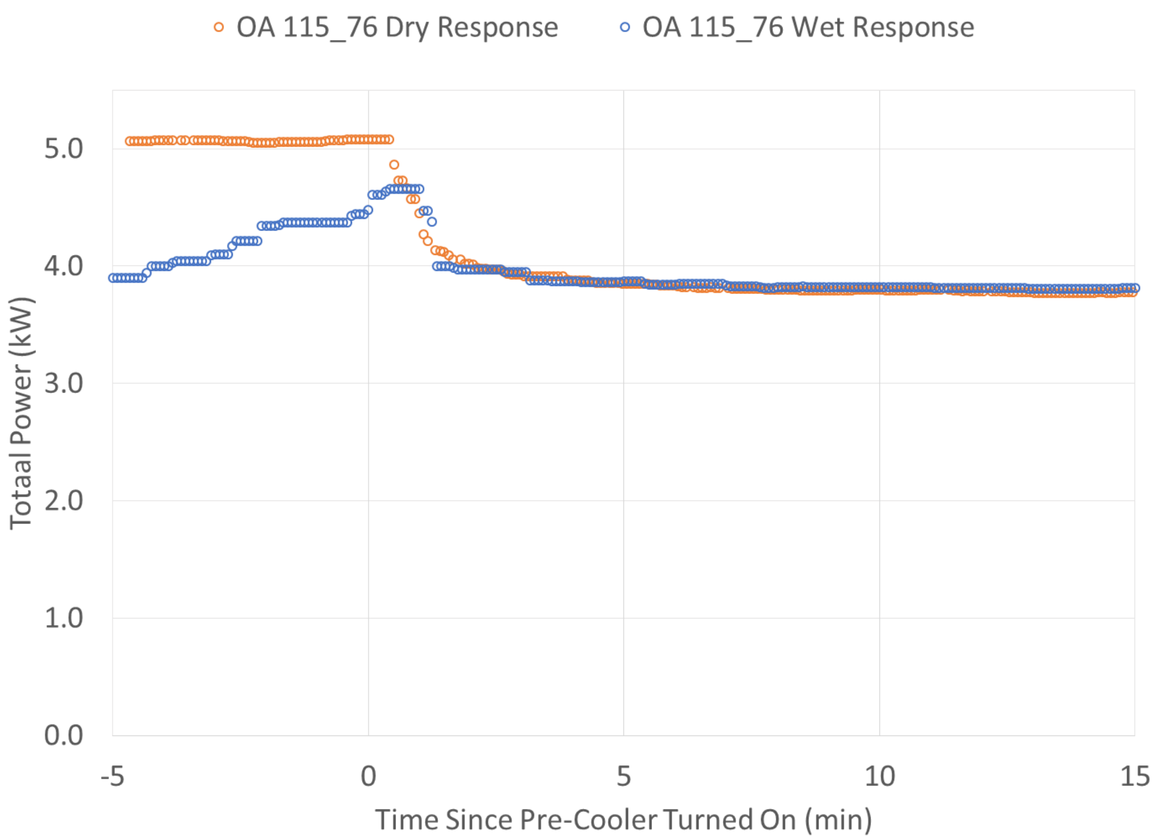

FIGURE 15: TRANSIENT "PRE-COOLER ON" POWER RESPONSE COMPARING A COMPLETELY DRY PRE-COOLER TO A PARTIALLY WET PRE-COOLER

LABORATORY TESTING: CONCLUSIONS

Lab study results show the majority power reduction benefits obtained by turning on an evaporative pre-cooler are achieved quickly, within less than two minutes. This demonstrates power reduction from pre-coolers may be useful to utilities as a demand response and grid management tool. Capacity increases take longer to achieve and stabilize after approximately 15 minutes. Equipment operating at increased capacity will reduce equipment run times, although the behavior of equipment cycling is less predictable than the quick drop in RTU power consumption achieved when the pre-cooler is turned on.

FIELD TESTING: TECHNICAL APPROACH

FIELD SITE SELECTION

We used TMY3 data to calculate the typical hour average wet-bulb depression for various cities in Southern California. The number of hours for which the wet-bulb depression was above 15°F between 7:00 a.m. and 9:00 p.m. from May to October was used to identify areas where condenser air pre-coolers could operate with a ≥ 10°F temperature drop during the field test. Corona was one of the top candidates, because 1,300 hours out of the possible 2,562 hours between 7:00 a.m. and 9:00 p.m., May to October, typically have a wet-bulb depression of ≥15°F (~50%). Additionally, approximately half of those 1,300 hours occurs when the average outside air temperature is ≥ 85°F (Figure 16). Based on the wet-bulb depression distribution, a Corona big box retail store was identified and selected as the field test site for this project. The site was cooled by packaged RTUs ranging in nominal tonnage from three to 20 tons, for a total of 262 tons.

Prior to this study, the retail store had condenser and ventilation air pre-coolers installed on six 20-ton RTUs (shown as black squares in Figure 17). These units served the main retail space, checkout area, front entry, and produce section. The research team retrofitted the remaining 20-ton RTU, as well as four 10-ton RTUs, with condenser air pre-coolers (Figure 17). The four 10-ton units were selected based on short-term monitoring data collected from the seven available 10-ton RTUs over four days, during which the outdoor air temperature ranged between 58°F – 90°F. The additional 20-ton and four 10-ton units served the electronic and automotive sections, the bakery, the restaurant, the indoor portion of the garden shop, and the stock room receiving area.

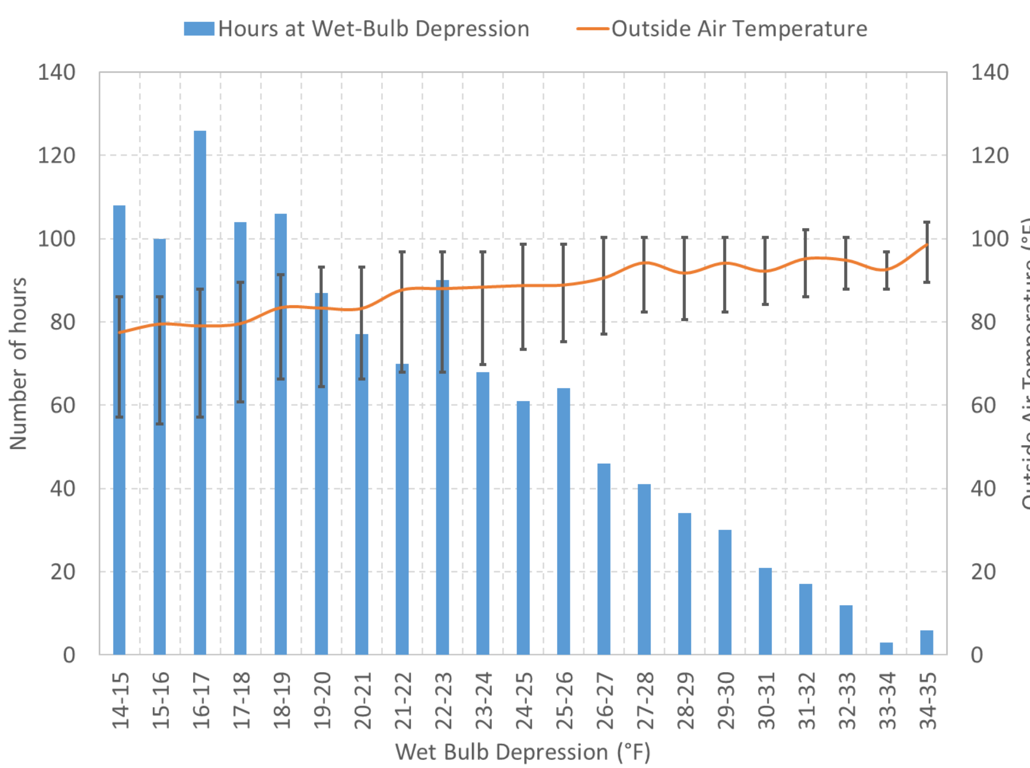

FIGURE 16: WET-BULB DEPRESSION BETWEEN THE HOURS OF 7AM - 9PM FOR MAY TO OCTOBER. CALCULATED FROM TMY3 DATA. THE ERROR BARS REPRESENT THE MINIMUM AND MAXIMUM OUTSIDE AIR TEMPERATURE FOR EACH BIN.

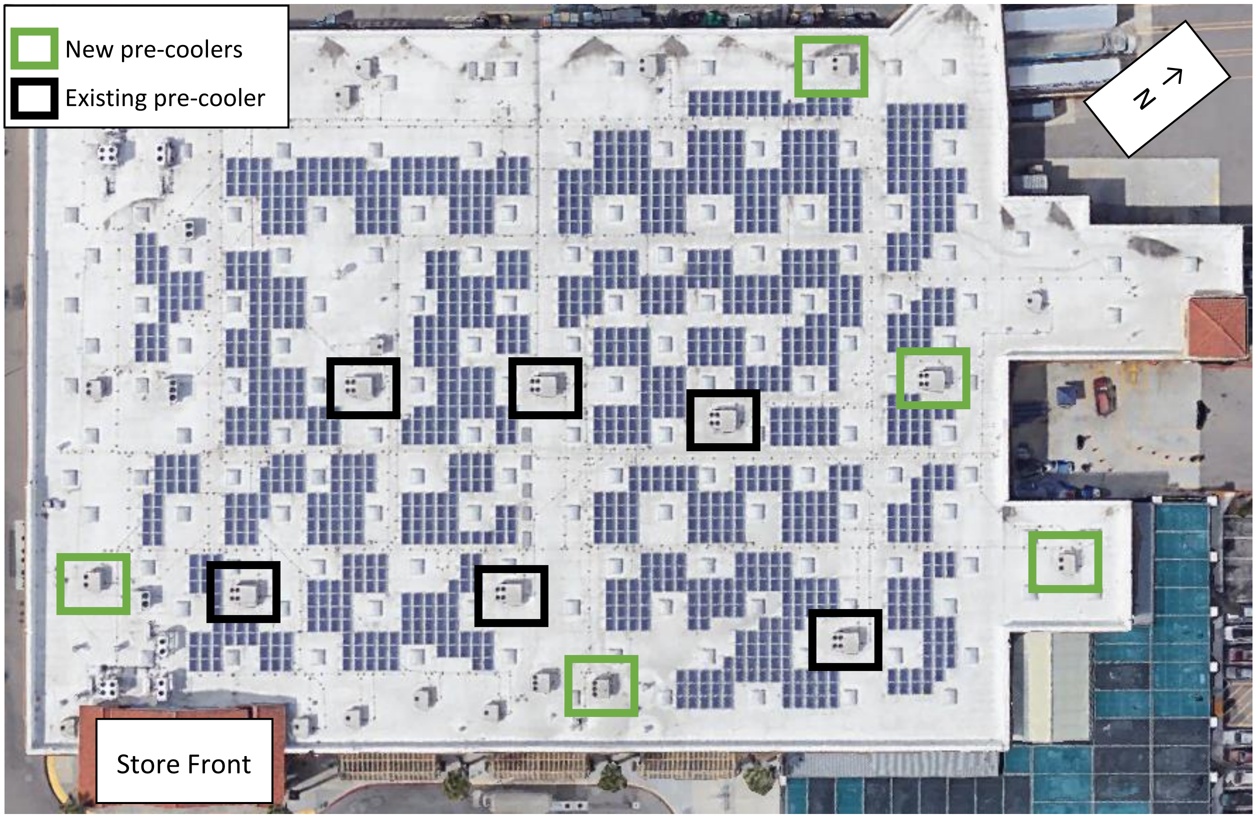

FIGURE 17: BIRD'S EYE VIEW OF FIELD TEST SITE. BLACK ILLUSTRATES EQUIPMENT WITH EXISTING PRE-COOLERS AND GREEN ILLUSTRATES THE EQUIPMENT WHERE PRE-COOLERS WERE ADDED

INSTRUMENTATION

Table 11 and Figure 18 provide a description of the sensors used to monitor each RTU, as well as a general outline of where the sensors were installed. Each RTU was monitored with a three-phase power meter, a water flow meter, and a return air enthalpy sensor. An outdoor enthalpy sensor was installed on the roof, to measure local outdoor air conditions. All sensors were wired to a DataTaker 85M data acquisition system and sampled once per minute. During the field testing period, a daily data file was transferred to an offsite FTP server each night.

TABLE 11: FIELD TEST MEASUREMENTS

SYMBOL (FIGURE 18)	**MEASUREMENT TYPE**	**MANUFACTURER AND MODEL #**	**ACCURACY**	**SIGNAL TYPE**
T_{OSA}	Outside Air Temperature	Visala HMP110	±0.1°F	Analog
RH_{OSA}	Outside Air Relative Humidity	Visala HMP110	±1.6%	Analog
T_{RA}	Return Air/Indoor Temperature	Visala HMP110	±0.1°F	Analog
RH_{RA}	Return Air/Indoor Relative Humidity	Visala HMP110	±1.6%	Analog

Symbol (Figure 18)	Measurement Type	Manufacturer and Model #	Accuracy	Signal Type
V_{WATER}	Make-up Water Flow into Pre-Cooler	Omega	±1.5% of reading	Analog
$Power_{SYSTEM}$	RTU System Power	Dent PowerScout	1% of reading	RS-485

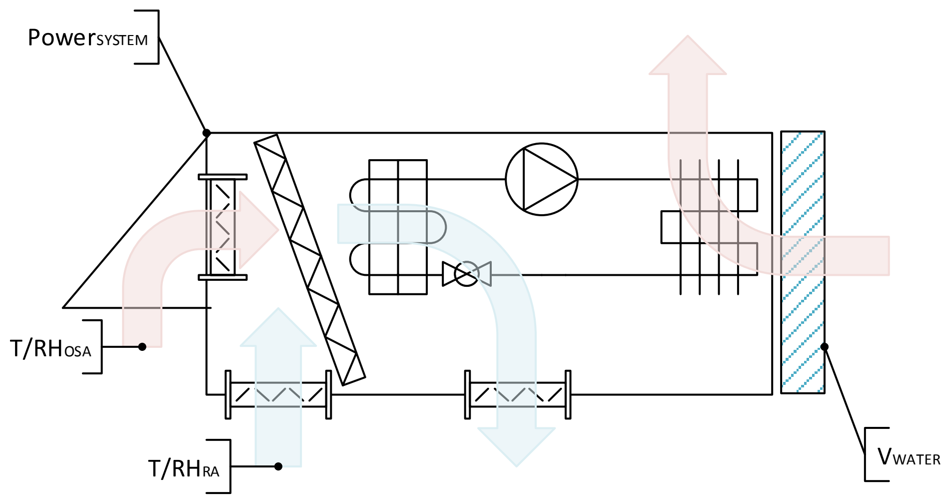

Figure 18: Schematic of instrumentation

Air Temperature and Relative Humidity Measurements

We used single temperature and Relative Humidity (RH) sensors, placed near the center of each airstream (to minimize any wall effects from the ducting) to measure air temperature and RH.

Power Measurements

RTU electrical power was measured using a three-phase Root Mean Squared (RMS) power meter installed on the RTU's main power lines. The power meter measured true power, reactive power, apparent power, power factor, voltage, and current for all three power phases. The three-phase power meter did not measure the power used by the pre-cooler water pump, as it was powered from the RTU service outlet. Therefore, a one-time power measurement was taken while the pump operated, and the result was added to all of the power data, during the periods when the dispatchable event was called.

Water Use

We used turbine flow meters, installed in the makeup water lines, to measure each pre-cooler's water use. Each flow meter was installed with a minimum straight pipe length of 5" upstream and 2.5" downstream of the turbine.

DISPATCH EVENT STRATEGY

During the field test period, the research team monitored the weather forecast for Corona, California and scheduled dispatch events for times when the outdoor air would be at or above 95°F at the start of the event. Each event was scheduled through the pre-cooler manufacturer's control system. Table 12 provides a summary of the 26 dispatch events, which were not called by or coordinated with SCE. The event durations varied between 60 and 240 minutes to test event response. If multiple dispatch events were scheduled on the same day, they were separated by at least three hours, to ensure the evaporative media would dry out before the start of the next event.

TABLE 12: LIST OF DISPATCH EVENTS FOR FIELD TESTING PERIOD.

EVENT START TIME	EVENT DURATION (MINUTES)	NOTES	EVENT START TIME	EVENT DURATION (MINUTES)	NOTES
07-08 11:00:00	60		**08-27 12:00:00**	60	
07-08 16:00:00	60		**08-27 15:00:00**	60	
07-22 12:00:00	60	Dispatch Signal Failed	**08-28 12:00:00**	240	Overlapped SCE Event
07-22 16:00:00	120		**08-28 18:00:00**	60	Overlapped SCE Event
08-01 12:00:00	60	Overlapped SCE Event	**08-30 12:00:00**	120	
08-01 16:00:00	120	Overlapped SCE Event	**08-30 16:00:00**	60	Overlapped SCE Event
08-03 11:00:00	120	Overlapped SCE Event	**08-30 19:00:00**	120	Overlapped SCE Event
08-03 15:00:00	120	Overlapped SCE Event	**08-31 12:00:00**	120	Overlapped SCE Event
08-11 13:20:00	60		**08-31 16:00:00**	120	Overlapped SCE Event
08-12 12:00:00	120	Dispatch Signal Failed	**09-01 12:00:00**	60	Overlapped SCE Event
08-12 16:00:00	60	Dispatch Signal Failed	**09-01 15:00:00**	180	Overlapped SCE Event
08-26 13:00:00	120		**09-28 12:00:00**	120	
08-26 17:00:00	60	Overlapped SCE Event	**09-29 12:00:00**	120	

DISPATCH RESPONSE ANALYSIS

RTU data was visually inspected for every event, to manually record if the RTUs were in one of the three following states:

1. Cooling, and the pre-cooler successfully dispatched.
2. Cooling, and the pre-cooler failed to dispatch.
3. Not cooling, or off.

A successful dispatch was confirmed if the RTU was cooling and the power data illustrated the expected "U" shape during the event. RTUs that stopped cooling (or started cooling after the event started) were considered to have successfully dispatched, if the leading or trailing half of the "U" shape was observed, respectively.

Visual inspection identified that three events (one on 7/22 at 12:00:00, and both events on 8/12) did not experience successful pre-cooler dispatches. Additionally, the field test site participated in a third-party Aggregator Managed Portfolio (AMP) demand shedding program. Since the dispatch events called by the research team were not coordinated with SCE, several dispatch events overlapped with actual SCE demand-shedding events. Because the exact AMP demand-shedding methodology was proprietary, the potential effect on RTU operation was unknown. The events that overlapped the SCE event, as well as the three failed events, were excluded from detailed analysis. This reduced the number of analyzed events to 10.

EQUATIONS AND ERROR ANALYSIS

AGGREGATE POWER

We used the power draw for each dispatched RTU, together with Equation 3, to calculate the aggregate power draw for each event.

EQUATION 3: AGGREGATE POWER

$$Aggregate\ Power\ (kW) = \sum_i \left(\frac{1}{T} \cdot \sum_{t=1}^{T} P_{RTU,i} \right)$$

Where:

i is the set of RTUs that had a pre-cooler dispatch during the event

t is each minute from the start to the end of an event (T)

$P_{RTU,i}$ is the one minute average power draw of RTU number i in kilowatts

WATER USE

The water use for each event was calculated using Equation 4.

EQUATION 4: WATER USE

$$Water\ Use\ (Gal) = \sum_i \sum_{t=1}^{T} \dot{V}_{RTU,i} \left(\frac{Gal}{min} \right)$$

Where:

i is the set of RTUs that had a pre-cooler dispatch during the event

t is each minute from the start to the end of an event (T)

$\dot{V}_{RTU,i}$ is the volumetric flow rate of water during each minute of an event

MEASUREMENT UNCERTAINTY

We used the sequential perturbation method to determine the uncertainty of all the calculations. This numerical approach used a finite difference as a way of approximating derivatives to represent the sensitivity of the calculated value to the variables [4]. This well-accepted tactic is used when the partial differentiation method of the propagation of error is complex, or when there are many variables. The process used for sequential perturbation involved calculating a result (R_o) based on measured values. After R_o was calculated, an independent variable within the equation for R_o was perturbed by its respective uncertainty, and a new value (R_i^+) was calculated. Next, the same independent variable within R_o was decreased by its respective uncertainty, and a new value (R_i^-) was calculated. The differences between R_i^+ and R_o, and R_i^- and R_o, were calculated, and the absolute values averaged. The result was defined as δR_i. This process was repeated for every independent variable within R_o, and the final uncertainty was calculated as shown in Equation 5:

EQUATION 5: UNCERTAINTY USING SEQUENTIAL PERTURBATION

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

Where:

δR_i is the calculated uncertainty for an independent variable

L is the total number of independent variables in a calculation

U_R is the total calculated uncertainty

The uncertainty for all tests was calculated using this method, and listed in Table 15.

FIELD TESTING: RESULTS

OVERVIEW OF DISPATCH EVENT RESPONSE

For each of the 10 analyzed events, the histogram in Figure 19 shows the response of each RTU with pre-cooler. The figure breaks down the number of RTUs that were 1) not cooling at the time of dispatch; 2) cooling and successfully dispatched; and 3) cooling and failed to dispatch. The average outside air temperature during each event is also shown. For each event, eight to 11 RTUs (73-100%) were cooling. However, the number of RTUs in cooling mode did not trend with the outside air temperature. The number of RTUs that successfully dispatched was highly variable, ranging from 10% to 89%. The reason available RTUs did not successfully dispatch is unknown; however, this represents one area in which the technology could be improved.

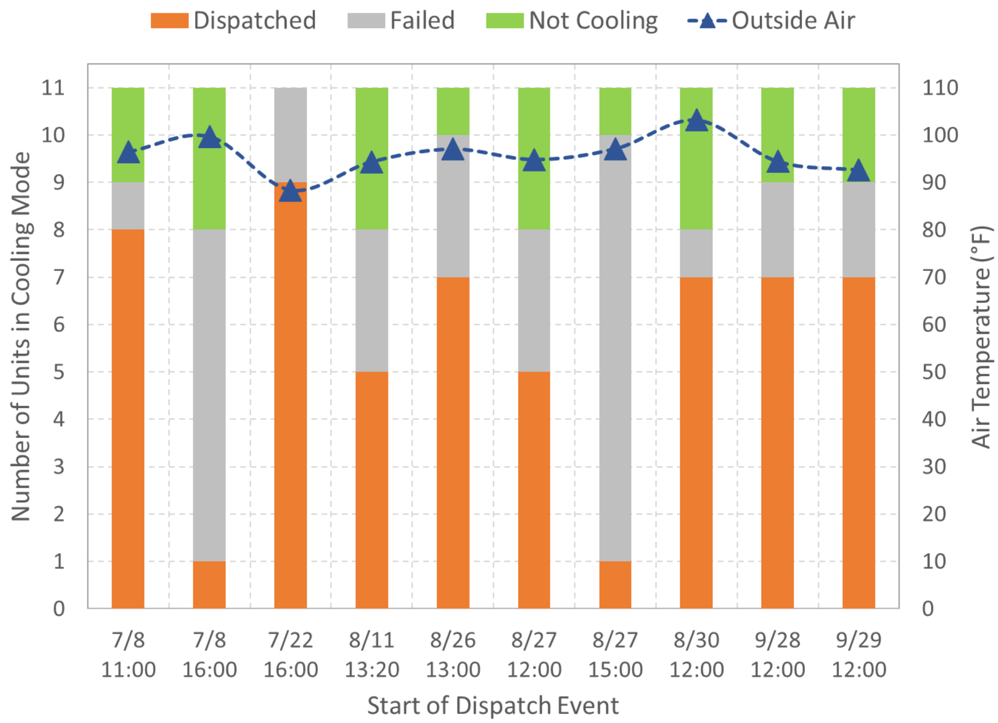

FIGURE 19: NUMBER OF RTUS IN COOLING MODE AND WHAT DISPATCHED DURING EVENT WITH THE CORRESPONDING AVERAGE OUTSIDE AIR TEMPERATURE.

AGGREGATE POWER

Figure 20 illustrates the total power draw for dispatched RTUs for the six events in which at least 70% of the RTUs that were cooling responded to the dispatch signal. Outdoor air temperature and average return air temperature to the RTUs is also shown. The events on July 8 and August 30 had an aggregate power response similar to what was measured in laboratory testing. At the start of the events, the power draw was rapidly reduced and remained at the lower power draw for the duration of the events. When the events ended, the power draw slowly increased to levels similar to those from before the start of the events.

The remaining four events had an aggregate response that was not as well defined. In these four events, all the dispatched RTUs did not remain in the same mode as at the start of the events, which affected the aggregate power response results. Key event metrics are tabulated in Table 13. Across the six events, the lowest percentage of units to dispatch was 70% (7/10) and the highest was 89% (8/9). Compared to the average power draw in the 15 minutes before and 60-75 minutes after the events, the aggregate power draw for the dispatched RTUs was reduced by 2.5% to 14.9% (2.87 kW and 17.05 kW). The post-event power draw was compared 60-75 minutes after the events ended, because the RTUs still benefitted from the pre-cooler while the media dried. The media was expected to be dry one hour after the events. To compare water usage, we used the total gallons of pre-cooler water (including all evaporated water plus water drained to maintain water quality) per kilowatt-hour saved for each event. For the six events, water usage ranged between 11.07 Gal/kWh and 51.90 Gal/kWh, with a median of 11.33 Gal/kWh.

Figure 21 shows the total dispatched RTU power draw for the four events in which fewer than 70% of the cooling RTUs responded to dispatch signals. Outdoor air temperatures and average RTU return air temperatures are also shown. Three of the four events had aggregate responses similar to those in the laboratory testing. However, the July 8 event response was caused by the RTU switching from stage two to stage one cooling, and the pre-cooler activating. On August 28, immediately before the pre-cooler successfully dispatched, the RTU increased by one cooling stage – so the pre-event power was lower than it was when the event started. Key event metrics are tabulated in Table 14. Across the four events, the lowest percentage of dispatched units was 10% (1/10), and the highest was 63% (5/8). Compared to the average power draw in the 15 minutes before and 60-75 minutes after, the four dispatch events reduced the aggregate RTU power draw by 0.07 kW to 7.65 kW (0.4% to 44.4%). The percent reduction values were influenced by the two events for which only a single RTU dispatched. During the event with a 44% power reduction, the one dispatched RTU's cooling stage change significantly impacted the result. The four events' water usage varied between 4.78 Gal/kWh and 249.05 Gal/kWh, with a median of 14.51 Gal/kWh. The water usage value spread was heavily influenced by the single event that only achieved 0.07 kWh energy reduction.

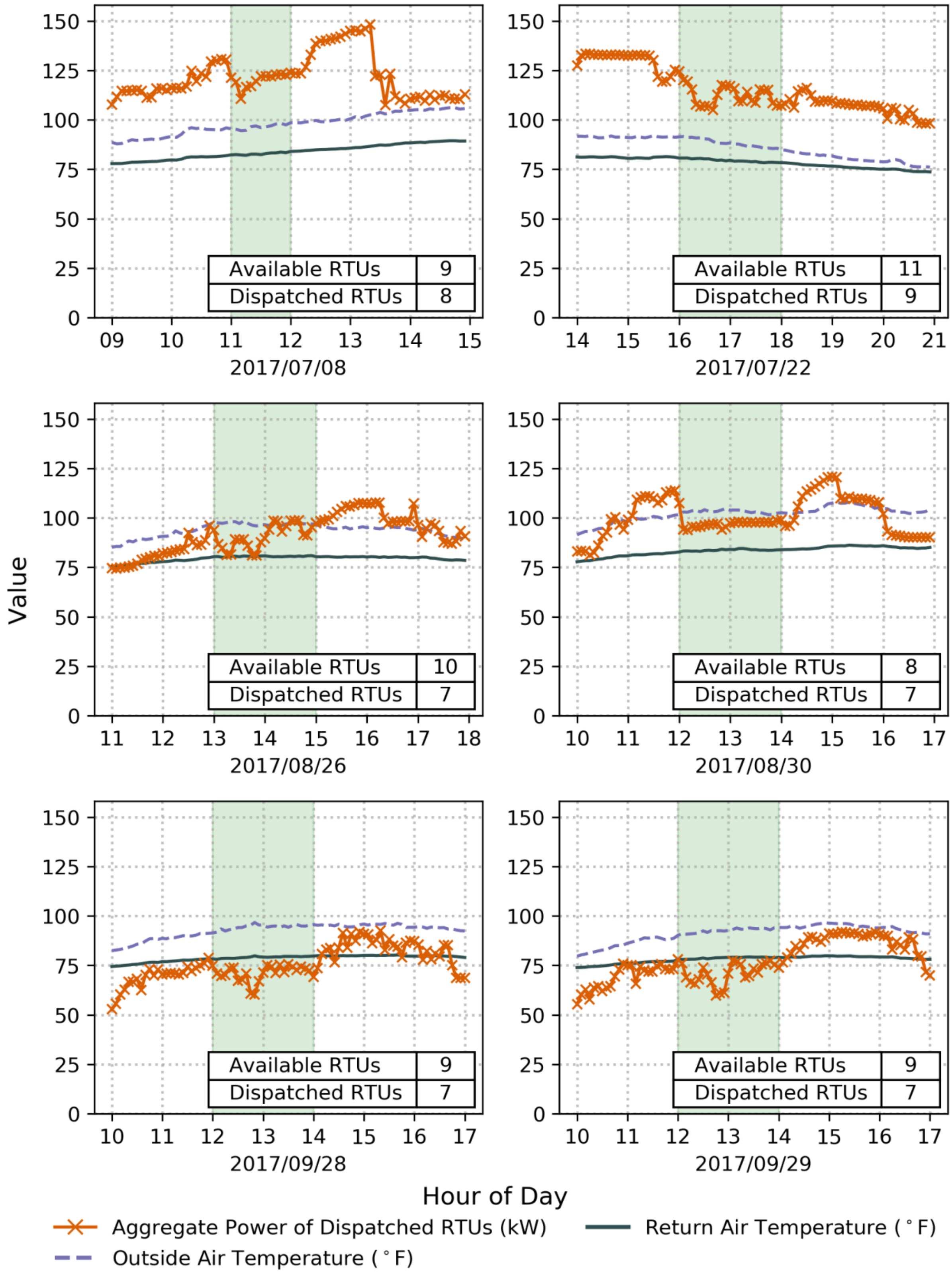

FIGURE 20: AGGREGATE POWER OF DISPATCHED RTUS WHEN AT LEAST 70% OF RTUS DISPATCHED

Table 13: Summary for events where >75% of RTUs dispatched

Event Statistic	07-08 11:00	07-22 16:00	08-26 13:00	08-30 12:00	09-28 12:00	09-29 12:00
Duration (minutes)	60	120	120	120	120	120
Average Outdoor Air Temperature (°F)	96.5	88.2	97.0	103.1	94.4	92.6
Number of RTUs in Cooling Mode	9	11	10	8	9	9
Number of RTUs that Dispatched	8	9	7	7	7	7
Percent of RTUs Dispatched (%)	89	82	70	88	78	78
Water Use (GAL)	196.7	196.7	196.7	196.7	196.7	196.7
Average Power 15 min Before Event (kW)	130.37	122.16	90.38	113.43	76.72	73.06
Average Power During Event (kW)	120.24	112.48	90.89	97.26	71.11	71.12
Average Power 60-75 min After Event (kW)	145.23	108.55	107.51	115.19	88.96	91.33
Baseline (Average of Before/After Power) (kW)	137.80	115.35	98.94	114.31	82.84	82.20
Power Difference between Event and Baseline (kW)	17.56	2.87	8.06	17.05	11.73	11.08
Percent Difference from Baseline to Event (%)	12.7	2.5	8.1	14.9	14.2	13.5
Water Use / Energy Saved (Gal/kWh)	11.2	51.90	14.91	11.47	11.19	11.07

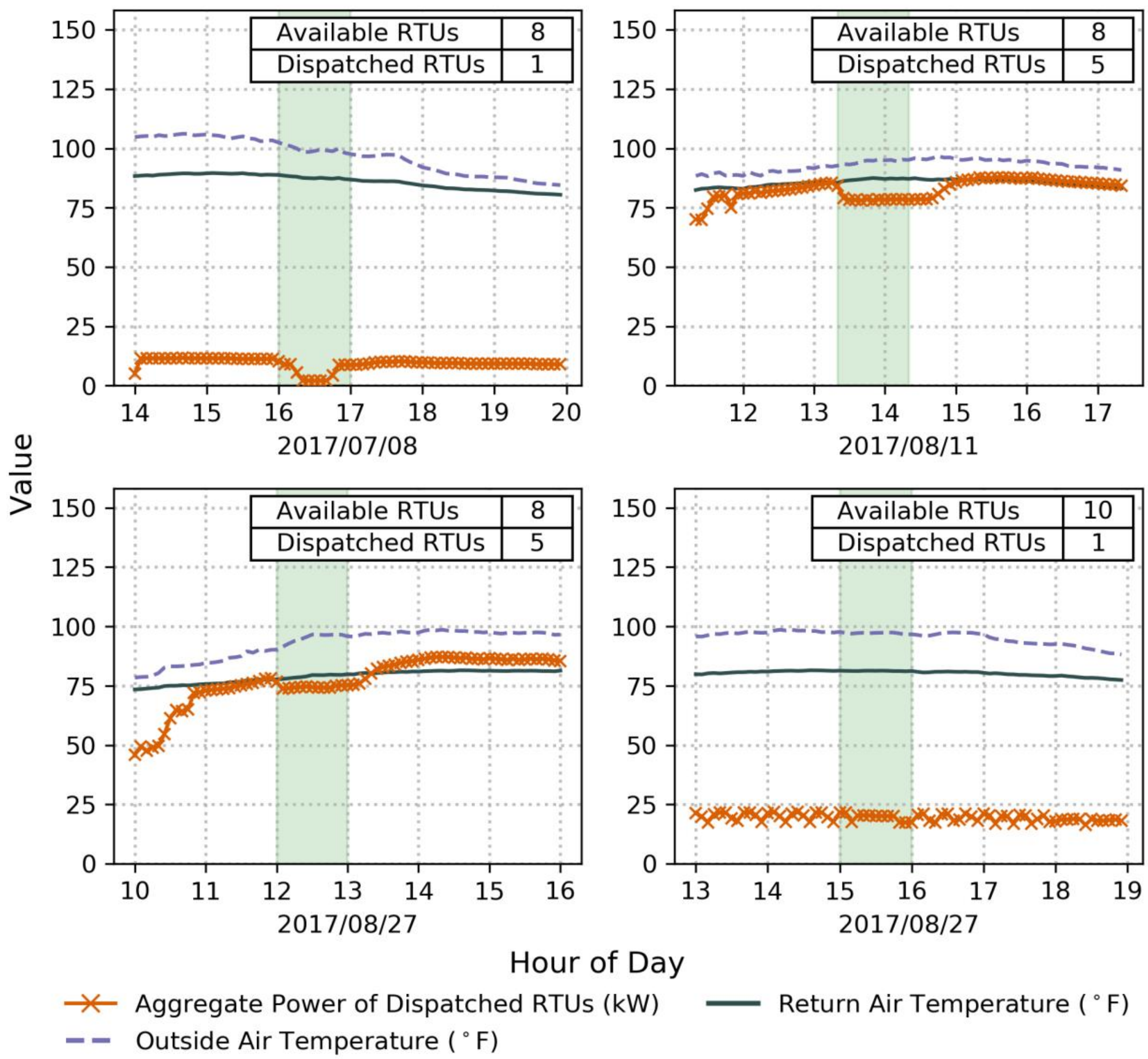

FIGURE 21: AGGREGATE POWER OF DISPATCHED RTUS WHEN LESS THAN 70% OF RTUS DISPATCHED

Table 14: Summary for events where <75% of RTUs dispatched

Event Statistic	07-08 16:00	08-11 13:20	08-27 12:00	08-27 15:00
Duration (minutes)	60	60	60	60
Average Outdoor Air Temperature (°F)	99.5	94.4	94.9	97.1
Number of RTUs in Cooling Mode	8	8	8	10
Number of RTUs that Dispatched	1	5	5	1
Percent of RTUs Dispatched (%)	13	63	63	10
Water Use (GAL)	22.2	22.2	22.2	22.2
Average Power 15 min Before Event (kW)	11.25	85.19	77.76	19.71
Average Power During Event (kW)	5.82	78.88	74.61	19.63
Average Power 60-75 min After Event (kW)	9.66	87.67	86.57	19.70
Baseline (Average of Before/After Power) (kW)	10.46	86.43	82.16	19.14
Power Difference between Event and Baseline (kW)	4.64	7.55	7.55	0.07
Percent Difference from Baseline to Event (%)	44.4	8.7	9.2	0.4
Water Use / Energy Saved (Gal/kWh)	4.78	15.86	13.16	249.05

UNCERTAINTY

Table 15 provides an example of the typical uncertainty of the calculated parameters.

TABLE 15: UNCERTAINTY RESULT FOR CALCULATED METRICS

EVENT STATISTIC	TYPICAL VALUE	UNCERTAINTY
AVERAGE POWER 15 MIN BEFORE EVENT (KW)	113.43	± 1.58
AVERAGE POWER DURING EVENT (KW)	97.26	± 1.36
AVERAGE POWER 60-75 MIN AFTER EVENT (KW)	115.19	± 1.61
WATER USE / ENERGY SAVED (GAL/KWH)	11.47	± 1.22

FIELD TESTING: CONCLUSIONS

The research team called 26 dispatch events between July 1 and October 31, 2017. Out of the 26 events, 16 were excluded from the final analysis, because they either overlapped with actual SCE demand-shedding events, or the dispatch signals failed and no RTUs responded. The 10 analyzed dispatch events show results differ based on the number of dispatched RTUs.

In six of 10 events, more than 70% of the available RTUs successfully dispatched, and the aggregate power reduction was between 2.87 kW and 17.05 kW. For these six events, water usage varied from 11.07 Gal/kWh to 51.90 Gal/kWh, with a median of 11.33 Gal/kWh.

In four of 10 events, 10%-63% of the available RTUs successfully dispatched, and the aggregate power reduction was between 0.07 kW to 7.65 kW. For these four events, water usage varied from 4.78 Gal/kWh to 249.05 Gal/kWh, with a median of 14.51 Gal/kWh.

These results show improvements are needed to increase dispatch control reliability, which will ensure RTUs successfully dispatch during each event and improve overall results.

CONCLUSIONS AND RECOMMENDATIONS

The laboratory study showed most power reduction benefits obtained from turning on evaporative pre-coolers are achieved quickly, within less than two minutes. Capacity increases take longer to achieve and stabilize after approximately 15 minutes. Equipment operating at increased capacity will reduce equipment run times, although equipment cycling behavior is less predictable than the quick drop in RTU power consumption that occurs when pre-coolers are turned on.

Aggregate power draw field study results differed based on the number of dispatched RTUs. In three events, 0% of RTUs dispatched; in four events, fewer than 70% of RTUs dispatched; and in six events, more than 70% of RTUs dispatched. When more than 70% of the available RTUs successfully dispatched, the aggregate power draw was reduced by up to 17.05 kW. These results indicate improvements are needed to increase dispatch control reliability, to ensure RTUs successfully dispatch during each event, which will improve the overall results. Additionally, the RTU's operating mode during the event can significantly affect the result.

RECOMMENDATIONS

Laboratory results illustrate that at steady-state operation, the technology can reduce demand quickly, by up to 25%. In field testing, this trend was not as clear, because the RTUs could change cooling modes during events, and potentially did not start cooling until shortly after event start. The field results did not demonstrate the full potential of evaporative pre-cooling, because controls did not reliably dispatch pre-coolers for every available RTU. Based on these results, it is recommend the technology be considered for further study to determine the control changes required to ensure all available units will dispatch.

REFERENCES

[1] Southern California Edison, "ET13SCE1020 Evaporative Condenser Air Pre-Coolers," Emerging Technologies Coordinating Council, 2015.

[2] ANSI/AHRI, "Standard 210/240 "2008 Standard Rating of Unitary Air-Conditioning and Air-Source Heat Pump Equipment"," 2013.

[3] ASHRAE, "ASHRAE 37-2009. Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment," ASHRAE, 2009.

[4] R. S. Figliola and D. E. Beasley, Theory and Design for Mechanical Measurements, 3rd ed., John Wiley & Sons, Inc, 2000.