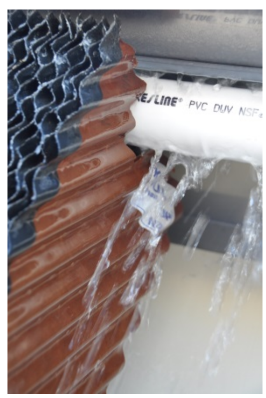
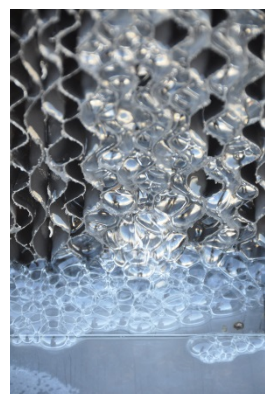
DESIGN & ENGINEERING SERVICES

Performance Evaluation for Dual-Evaporative Pre-Cooling Retrofit in Palmdale, California

ET13SCE1040 Report

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For:

New Program Development & Launch
Customer Programs and Services
Southern California Edison

October 31, 2014

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ACKNOWLEDGEMENTS

Southern California Edison's New Program Development & Launch (NPD&L) group is responsible for this project. It was developed as part of Southern California Edison's Emerging Technologies Program under internal project number ET13SCE1040. The research study was conducted by University of California Davis, Western Cooling Efficiency Center (WCEC) as part of a broader research program "Advancing HVAC Technologies to Support the California Energy Efficiency Strategic Plan". Jay Madden managed the field evaluation for Southern California Edison with guidance and direction from Jerine Ahmed.

Southern California Edison and UC Davis extend thanks to Integrated Comfort for their collaboration in this project and for their leadership in advancing the development of climate appropriate commercial cooling solutions.

DISCLAIMER

ABBREVIATIONS AND ACRONYMS

CEC	California Energy Commission
COP	Coefficient of Performance (*dimensionless*)
c_p	Specific Heat Capacity (e.g. *Btu/lbm-°F*)
CPUC	California Public Utilities Commission
C_X	Concentration (*of constituent X*) (e.g. *ppm*)
DX	Direct Expansion Vapor Compression
ε	Sensible Heat Exchanger Effectiveness (*dimensionless*)
$\dot{E}$	Electric Power, (Rate of Electric Energy Consumption) (e.g. *kW*)
EA	Exhaust Air
EMCS	Energy Management and Control System
$\dot{H}$	Cooling Capacity, (Enthalpy Flow Rate) (e.g. *kBtu/h*)
h	Specific Enthalpy (e.g. *Btu/lbm-dry air*)
HR	Humidity Ratio (e.g. lbm_{water}/lbm_{dryair})
$\dot{m}$	Mass Flow Rate (e.g. *lbm/h*)
OSA	Outside Air
ΔP	Differential Static Pressure (e.g. *in WC*)
RA	Return Air
RH	Relative Humidity (%)
RTU	Rooftop Air Conditioning Unit
SA	Supply Air
SCE	Southern California Edison
T	Temperature (e.g. *°F*)
$\dot{V}$	Volume Flow Rate (e.g. *scfm*)
WBE	Wet Bulb Effectiveness

FIGURES

TABLES

CONTENTS

EXECUTIVE SUMMARY

BACKGROUND

Packaged rooftop air conditioning units (RTUs) are the predominant equipment used for space conditioning of small and medium-size commercial buildings. It is estimated that roughly 70% of space conditioning in commercial buildings is provided by RTUs. Due to the longevity of these RTUs (roughly 15-20 years in California), more companies are entering the market with products that are designed to improve the energy efficiency of existing RTUs. This project was designed to evaluate the performance of one of those products.

OBJECTIVE

The purpose of this project was to gather field data to demonstrate and understand the performance improvement associated with adding an evaporative pre-cooler for condenser-inlet-air and ventilation-air to conventional RTUs. The tested retrofit was expected to save energy two ways: a) by cooling the outdoor air being delivered to the RTU indoor coil, and thereby reducing how much cooling it needs to perform, and b) by reducing the air temperature seen by the RTU condenser coil, thereby decreasing refrigerant pressure and the work that needs to be done by the compressor. The key metrics used to characterize the performance of the retrofit include: a) sensible Coefficient of Performance (COP), b) sensible cooling capacity, and c) electric power draw, all as a function of outdoor weather conditions. In addition, the project was designed to measure the on-site water consumption associated with achieving those performance improvements, as well as to calculate key intermediate parameters, such as the sensible cooling capacity delivered to the ventilation air, and the maximum observable wet-bulb effectiveness of the pre-cooling system. Wet-bulb effectiveness is a measure the ability of the evaporative media to cool the air entering the condenser toward the wet-bulb temperature of the outside air, and is the key parameter used to characterize the performance of evaporative coolers for condenser air in the laboratory. The measured sump-water temperature represents the coolest temperature produced by the pre-cooler, and therefore the highest observable wet bulb effectiveness.

APPROACH

In a field test, the technology was installed on 13 RTUs serving a big-box retail store in Palmdale, California. The field assessment was conducted over a one-year period between October 2012 and October 2013, The testing involved minute-by-minute data collection on four of the RTUs, before and after being retrofitted, and similar monitoring of two RTUs that were not modified. Most of the evaluation is based upon comparing the performance of retrofitted RTUs with and without the retrofit in place, focusing on evaluating the key performance parameters: sensible COP, sensible cooling capacity, and electric power draw. Sensible cooling capacity and sensible COP are based upon the sensible cooling provided by the RTU, not considering latent cooling.

As the energy and peak demand savings produced by the retrofit is expected to depend somewhat upon the particular equipment being retrofitted (e.g. differing equipment sensitivities to changes in condenser-inlet-air temperature, V-coil vs. flat-coil equipment), laboratory test protocols being developed for evaporative pre-coolers use an indirect measurement of the wet-bulb effectiveness of the media as the key performance metric. This field test included measurements of sump-water temperature to indirectly estimate maximum potential wet-bulb effectiveness for the direct and indirect evaporative cooling elements in the system.

In addition to the key performance metrics, the internal workings of the retrofit were also investigated, including isolating the performance of the water-to-air heat exchanger for indirect evaporative cooling of ventilation air, monitoring the performance of the sump-pump control, as well as monitoring total water consumption and estimating the fraction of total water use associated with evaporation.

RESULTS

As noted above, the key metrics used to characterize the performance of the retrofit include: a) sensible Coefficient of Performance (COP), b) sensible cooling capacity, and c) electric power draw. The measured results for these metrics are presented in various scatter plots throughout the report. The Table below presents the performance of the retrofit at two different groupings of outdoor temperature (75-85^{o}F and 85-95^{o}F). In general, between 85-95°F the retrofit was found to increase COP by up to 27%, increase sensible cooling capacity by up to 17%, and decrease power draw by up to 15%. The energy savings was lower at outside air temperatures between 75-85°F. This is not surprising considering that the average outdoor-air wet-bulb depression was roughly 8°F larger at the higher outdoor air temperatures. As illustrated in the Table below, the most robust data set was for RTU-7, and therefore the RTU-7 results are likely the best estimate of retrofit performance (at least for V-coil condensers). However, even though we have the most confidence in the RTU-7 results, it turns out that the 2-compressor savings results for RTU-10 are consistent with the RTU-7 results, as are the 75-85 °F results for RTU-11 operating with either 2 or 3 compressors. The 85-95°F results for RTU-11 are unfortunately based on very few pre-retrofit data points.

It should be noted that the large percentage savings for RTU-8 are based upon limited data points, and therefore have a high level of uncertainty. That said, the percentage savings should be highest for single-compressor operation (in which RTU-8 was operating) due to the layout of the condenser coils in a V for these particular RTUs. One side of the V does not benefit from pre-cooling, and single-compressor operation (which only occurred for RTU-8) uses only the pre-cooled side of the V. Another possible contributing factor is that ventilation cooling is a larger fraction of net system capacity with only one compressor in operation.

RTU#	RTU 7 2-compressor	RTU 8 1-compressor	RTU 10 2-compressor	RTU 11 2-compressor	RTU 11 3-compressor
Total Cooling in all modes (Aug 2013) [KBtu]	38,700	10,700	12,000	49,000	
Change in Sensible Capacity at Outside Air Temperature of 75-85°F [pre/post data points]	8% [7116/28850]	53% [22/66]	10% [182/2993]	7% [756/8047]	11% [624/8162]
Change in Sensible Capacity at Outside Air Temperature of 85-95°F [pre/post data points]	18% [2393/24360]	56% [87/1093]	17% [554/11575]	10% [117/6417]	14% [14/10865]
Change in COP at Outside Air Temperature of 75-85°F [pre/post data points]	14% [7116/28850]	72% [22/66]	18% [182/2993]	12% [756/8047]	20% [624/8162]
Change in COP at Outside Air Temperature of 85-95°F [pre/post data points]	27% [2393/24360]	69% [87/1093]	23% [554/11575]	10% [117/6417]	20% [14/10865]

Sump-water temperature measurements were used to estimate the maximum potential wet-bulb effectiveness for the retrofit. Despite the fact that the dual-evaporative pre-cooler introduces a sensible ventilation-air cooling load into the evaporative process, the sump water temperature remained very close to the outside air wet-bulb temperature, as long as the condenser fans were operating. Therefore, the maximum potential effectiveness was found to be between 80-100% when the condenser fans were operating.

Due to the fact that compressors ran very little for RTU-8 and RTU-10, observation of these units provided a large quantity of data on the performance of the retrofit as an indirect evaporative cooler for ventilation air while the condenser fans were off. In this mode, water is circulated through the ventilation cooling coil and across the evaporative media, but evaporation relies on free convection air flow as there is no mechanically driven air flow. A large amount of ventilation pre-cooling was observed in this mode of operation, which reduces the needs for compressor cooling. During the month of August 2013, ventilation air cooling by RTU-8 was calculated to save 530 kWh, which is roughly a third of the estimated 1600 kWh savings associated with COP improvements for RTU-7. However, a comparison of the ventilation-air cooling capacity with and without the condenser fans in operation indicates that running the condenser fan to enhance indirect evaporative cooling of ventilation air while compressors are off can increase the sensible cooling capacity generated in this mode by a factor of 2.5 to 4.5. This type of integrated control strategy entails a trade-off between added ventilation-air cooling and condenser fan power.

The retrofit's estimated water consumption for evaporation was relatively modest, corresponding to a cost penalty of roughly 5% of the electricity savings for the unit with the best pre/post retrofit data (RTU-7). However, maintenance water use was not adequately controlled for the monitored RTUs, such that it dramatically exceeded water evaporation in some cases, and was essentially zero in others. Using RTU-7 as our example once again, at the observed excessive bleed rate the estimated water cost would represent 14% of the electricity savings. On the other hand, going back to evaporated water use only, we calculated that the evaporated water was 2.9 gal/kWh, which is comparable to the average1.3 gal/kWh water consumed when generating electricity in California. Using appropriate maintenance water (15% of evaporated water) would result in a total water cost of 6% of electricity savings, or 3.4 gal/kWh.

CONCLUSIONS AND RECOMMENDATIONS

There are four key conclusions that can be drawn from this field study about the dual-evaporative pre-cooling system tested: a) considerable energy savings, capacity improvement, and peak demand reduction can be achieved with the technology, b) the water evaporated to achieve these savings is of the same magnitude as the water used to generate the saved kWh, c) the actual savings realized depends on the usage and operating mode of the RTU to which the system is applied, and d) the savings potential for this technology might benefit from improved controls and integration strategies. Moreover the results of this field test suggest that the building owner could disconnect one or more RTUS, due to the increase in cooling capacity of other units, thereby reducing their connected electrical load.

We would also recommend a detailed analysis of the interface and interactions between the use of the ventilation coil and existing RTU controls. The idea would be to produce a simple decision tool that uses condenser fan power, equipment geometry, climatic conditions, and outdoor air fraction to produce a schedule of set points for pump operation, condenser-fan operation, and even to recommend using the condenser-air pre-cooler retrofit without the ventilation coil in some instances.

Finally, based upon a large observed variability in water consumption as a result of the water management strategy used by the systems observed, the authors also recommend that the manufacturer develop more robust hardware and complementary installation procedures to manage maintenance water use. The conclusion here is that the evaporative water use of this retrofit is reasonable as compared to the water used to generate electricity in California, but that maintenance water use is an important factor that needs to be set up properly at the outset, and maintained on a regular basis (e.g. on the media maintenance schedule), or perhaps be controlled in more reliable manner.

INTRODUCTION

Conventional air conditioning systems are responsible for a majority share of the summer-time peak electrical demand in California. On average, cooling accounts for more than 50% of the peak electrical demand by commercial buildings. These systems (cooling plus ventilation) represent a smaller but still significant fraction of annual electricity consumption; accounting for approximately 25% of the annual electricity use in commercial buildings (EIA 2014, CEC 2006).

Cooling for commercial facilities is provided predominantly by rooftop packaged air conditioning units (RTUs) and other unitary vapor-compression systems. The nature of this technology is such that, everything being kept equal, both cooling capacity and energy efficiency diminish as the ambient temperature increases. With more than one million RTUs across California (CEC 2006), it is obvious that periods with high cooling needs result in greater stress on California's electrical grid.

Projected population growth will introduce additional cooling needs and corresponding peak electrical demand. Many factors, including the imperative for global environmental stewardship and the economic and societal costs of additional electrical generation capacity, point toward the need for significant efficiency improvements. The California Public Utilities Commission has emphasized the need for a rapid industry–wide shift toward dramatically more efficient cooling technologies. Among those strategies, the California Energy Efficiency Strategic Plan calls for the market uptake of "climate-appropriate" cooling technologies that utilize various techniques designed to produce cooling in California's hot-dry climate with far less energy consumption.

One of the key impediments to introducing climate appropriate RTUs is the longevity of this equipment in the California market. As this equipment generally has a useful lifetime of 15-20 years, retrofitting RTUs to make them more efficient and "climate appropriate" is of interest to electric utilities and policy makers in California. The purpose of this project was to gather field data to demonstrate and understand the performance improvement associated with adding a particular dual-evaporative pre-cooler technology to conventional RTU systems.

There are variety of new 'high efficiency' HVAC strategies, including variable-fan-speed, variable-compressor-capacity rooftop air conditioners. However, while these improvements have been shown to provide upwards of 50% annual electricity savings, they generally do not provide much savings under peak cooling demand conditions. This study emerges from a variety of efforts and innovation surrounding climate-appropriate cooling strategies that are especially appropriate for reducing peak demand. The dual-evaporative pre-cooling technology tested in this project was laboratory tested in 2012 by UC Davis, and that testing measured a 43% demand reduction at an outdoor temperature of 105^{o}F (73^{o}F wet-bulb) (Woolley, 2012). This particular product was used to help develop the Western Cooling Challenge, which focuses on advancing the commercialization of climate appropriate rooftop air conditioners, and recognizes the technology tested here as a central opportunity for improving efficiency for rooftop air conditioners.

Conventional DX systems have generally been optimized to meet national energy performance standards, which are focused on performance at generally cooler and more humid conditions relative to cooling conditions in California. This optimization is particularly problematic under hot dry conditions, as occur during peak electricity demand periods in California. There are two issues: a) this equipment makes up a significant portion of peak power demand on the grid, and b) they provide lower cooling capacity under hot conditions, resulting either in user discomfort, or more often, in significant equipment oversizing.

Some studies have demonstrated that evaporative cooling technologies can be an effective method to reduce electricity consumption and peak power demand from conventional DX systems (Wang 2014). Field testing is limited, especially for new emerging strategies such as the evaporative cooling retrofit tested herein.

Another goal of this study was to learn about real-world operating characteristics, as well as any constraints and challenges related to using this retrofit on existing HVAC systems. This is information that cannot be uncovered with laboratory testing in a controlled environment. For example, this study learned that the actual energy savings depends significantly on the initial (and post-retrofit) operating characteristics of the existing DX systems. For example, the compressors on one retrofitted unit operated almost continuously throughout the test period, whereas another unit operated almost exclusively in ventilation-only mode. In general, no two units operated in the same operating modes under the same weather conditions, and the water consumption of the different units was similarly all over the map. Similarly, the building's existing EMCS never requested economizer operation for these systems, so we were unable to investigate interactions between the dual-evaporative pre-cooler and economizer operation. Similarly, we also discovered that the pump did not turn on and off at a consistent outdoor air temperature, which can impact the energy-savings performance of the retrofit.

This report describes the application tested, our experimental design, the measured performance results, and lessons learned through this process.

BACKGROUND

OVERVIEW OF THE DUAL-EVAPORATIVE PRE-COOLER TECHNOLOGY

The dual-evaporative pre-cooler product tested in this project takes advantage of indirect evaporative cooling to pre-cool the ventilation air on a conventional rooftop unit, and uses direct evaporative cooling for condenser-air pre-cooling. It thereby does not add or subtract any moisture to/from the conditioned space. The system consists of evaporative media installed at the intake to the condenser coil, a water sump below the media, a finned tube heat exchanger installed at the outdoor air intake, and a circulating pump. Water evaporates in the media, thereby cooling itself and the air entering the condenser. This cooled water is collected in the sump, and the cool sump water is pumped to the finned-tube coil where ventilation air is sensibly cooled by the water. This water warms up, and is then returned to the top of the media.

The tested retrofit saves energy two ways: a) by cooling the outdoor air being delivered to the RTU indoor coil, and thereby reducing how much cooling it needs to perform, and b) by reducing the air temperature seen by the RTU condenser coil, thereby decreasing refrigerant pressure and the work that needs to be done by the compressor. Looking at the second effect thermodynamically, if the heat sink temperature is lower, refrigeration cycle efficiency is increased. This can be observed in manufacturer performance tables, which generally show an efficiency increase of 1-2% for every ^{o}F drop in the temperature of the air entering the condenser coil. Looked at yet another way, a reduced condenser inlet temperature results in a reduced refrigerant temperature, and therefore reduced pressure at the compressor, which in turn translates to reduced compressor electricity consumption. Further physical explanation of the savings potential from this measure is provided in the appendices for the Phase I report from this project (Madden, 2012).

The dual-evaporative pre-cooler tested was designed for retrofit of conventional RTUs, in principle requiring relatively modest integration efforts. It also uses a relatively small amount of materials, which should result in reasonable retrofit costs. However, the retrofit does have some impacts on the RTU. Specifically, it increases the flow resistance associated with bringing in outdoor air during economizer operation, and in principle it adds some additional resistance to air flow across the condenser coil. Concerning the condenser air flow, laboratory testing indicated that the flow resistance added by this particular evaporative pre-cooler was almost unmeasurable (Woolley, 2012). Finally, the outdoor-air pre-cooling process reduces the temperature of the mixed air entering the RTU evaporator coil, which would tend to reduce sensible efficiency of the vapor compression cooling cycle.

Another factor that has to be considered with this retrofit is that it evaporates water, and therefore the first cost of installing the water supply needs to be considered, as does the cost of water being supplied during normal operation.

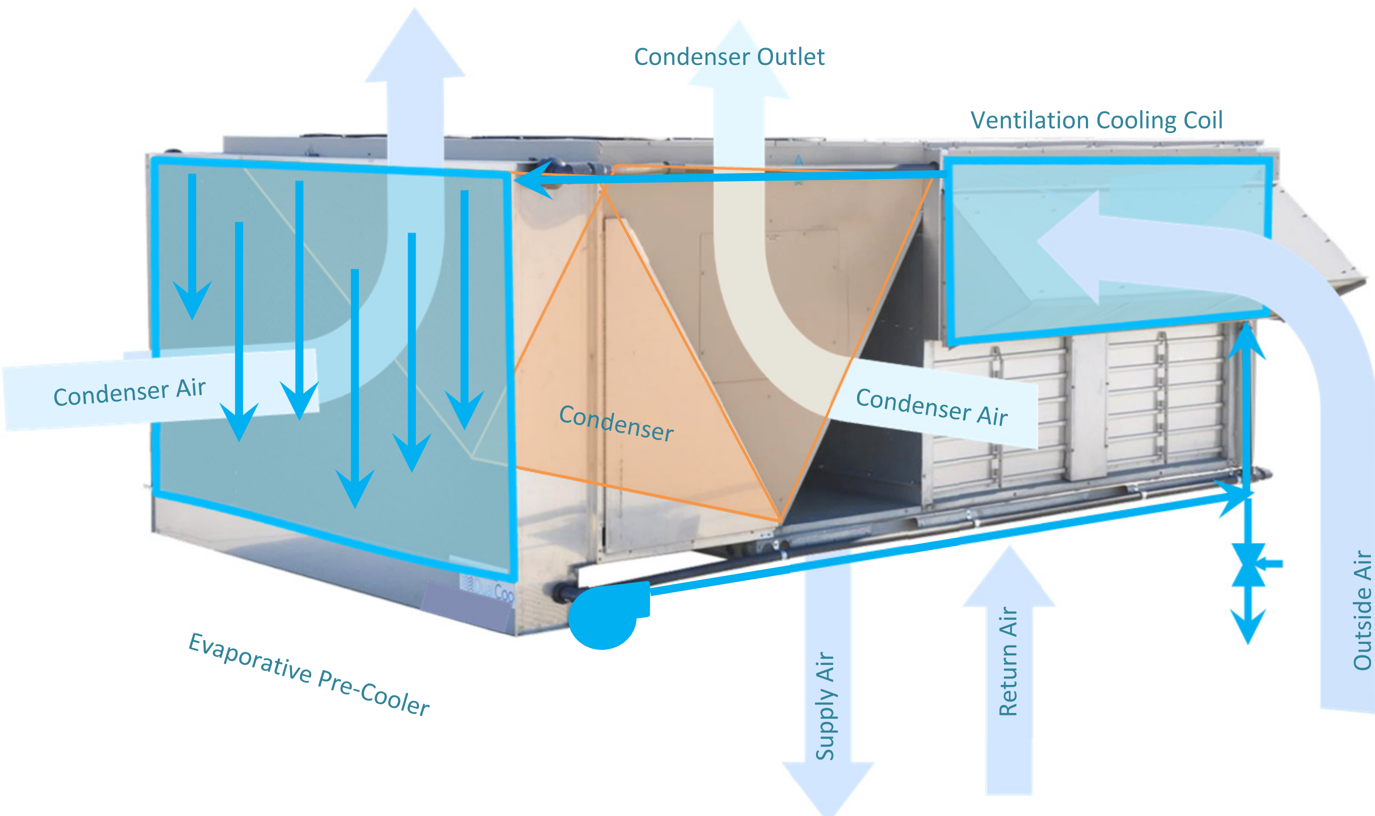


FIGURE 1: CONCEPTUAL SCHEMATIC FOR DUAL EVAPORATIVE PRE-COOLER RETROFIT

OVERVIEW OF FIELD TEST SITE

The field test site chosen for this study was a big-box retail store located in Palmdale, California. This building is conditioned by 21 RTUs produced by the same manufacturer, all of which utilize V-shaped condenser coils. The retrofit was applied to 13 of the 21 RTUs, and was monitored on four RTUs serving the sales floor. In addition to monitoring the four retrofitted RTUs, two un-retrofitted RTUs were also monitored. The nominal capacity of the monitored units ranged between 13-ton and 20-tons, with each RTU having three or four compressors. The two units that were used as the baseline for this test serve the warehouse portion of the building, but had their outdoor-air intakes modified so as to operate more like the units being retrofitted. The design specifications for all of the RTUs included in the test are summarized in Table 1, and the layout of the RTUs on the roof is shown in Figure 2.

TABLE 1: *ROOF-TOP UNIT (RTU) EQUIPMENT SCHEDULE*

RTU	*Manufacturer*	*Model*	*Design Supply Airflow [cfm]*	*Design Ventilation Airflow [cfm]*	*Nominal Cooling Capacity [tons]*	*Number of Compressors*	*Status*
RTU 7	Lennox	LGC180H2B	5,470	1,570	15	3	Retrofit
RTU 8	Lennox	LGC156H2B	5,390	1,210	13	3	Retrofit
RTU 10	Lennox	LGA240H2B	7,070	2,080	20	4	Retrofit
RTU 11	Lennox	LGC210H2B	6,070	1,870	17.5	4	Retrofit
RTU 20	Lennox	LGC180H2B	4,630	1,120	15	3	Baseline
RTU 21	Lennox	LGC210H2B	6,060	N/A	17.5	4	Baseline

SITE PLAN

FIGURE 2: LAYOUT OF RTUS ON THE BIG-BOX BUILDING TESTED IN PALMDALE, CALIFORNIA.
RED = BASELINE UNITS, GREEN = RETROFIT UNITS MONITORED, YELLOW = RETROFIT UNITS NOT MONITORED

OPERATING MODES

The RTUs with dual-evaporative pre-cooling investigated in this study can operate in many different modes. These modes are summarized in Table 2.

TABLE 2: DEFINITION OF EACH OPERATING MODE

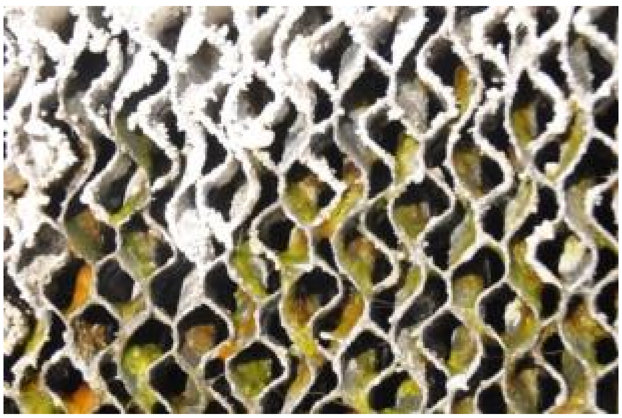
	Indoor Blower	OSA Fraction	Condenser Fans1 (of 4)	Compressor(s)	Water Pump
Off	OFF	0%	OFF	OFF	OFF
Ventilation Only	ON	MIN	OFF	OFF	OFF
DX1	ON	MIN	0-2	1	OFF
DX2	ON	MIN	0-4	2	OFF
DX3	ON	MIN	0-4	3	OFF
DX4	ON	MIN	0-4	4	OFF
Pump+DX1	ON	MIN	0-2	1	ON
Pump+DX2	ON	MIN	0-4	2	ON
Pump+DX3	ON	MIN	0-4	3	ON
Pump+DX4	ON	MIN	0-4	4	ON
Pump+Ventilation	ON	MIN	0	0	ON
Pump While Off	OFF	0%	0	0	ON
Heating	ON	MIN	0	0	OFF
Integrated Economizer Modes[2]	ON	100%	0-4	0-4	See Note

[1] Condenser fan operation is controlled in part by head pressure measurement on each refrigerant circuit. Nominally, the number of condenser fans corresponds to the number of compressors operating, except that when compressor discharge pressure is below a threshold some condenser fans will cycle off. These RTUs all have a V-shaped condenser. For the 4-compressor units, compressor circuits 1 and 2 have condenser coils on the outside of the V, while the condensers for circuits 3 and 4 are on the inside of the V. For the smaller 3-circuit units, circuit 1 is on the outside of the V, circuit 2 is on the inside of the V, and condenser coil 3 is split, with ½ on the outboard side and ½ on the inboard side of the V.

[2] The economizers function independently of other systems, as is typical for RTUs. In this way, the unit may run in full economizer mode (outside air only), or as an integrated economizer (outside air plus compressor cooling), with or without the dual-evaporative pre-cooling function. Since the retrofit provides some temperature reduction to the ventilation air stream, there is good reason to expect that equipment with the retrofit should benefit from operating with an elevated economizer changeover set point. However, it appears that for the units studied here, the economizer feature was disabled for the entire store through the building's EMCS.

EXPLANATION OF OPERATING MODES AND SEQUENCE OF OPERATIONS

Controls for the rooftop unit were not augmented in any way for the retrofit, the controls for the retrofit product operating wholly independently. This makes the system simpler to install, and more generalizable for addition to various rooftop air conditioners. Controls for the retrofit consist of a single outside air temperature switch that enables pump operation. In brief, the pump will run anytime the measured outside-air temperature is greater than the set point. The appropriate set point may vary some by application, but 70°F is generally suggested by the manufacturer as the changeover point. Below 70°F the cooling effect for ventilation air is expected to be small, and they expect very little efficiency benefit from condenser-air cooling. Further, it has been observed that if the pump is allowed to circulate at temperatures well below 70°F, the media may not dry out each day, which might allow biological growth on the media. Finally, the electricity consumption of the pump has to be weighed against any compressor savings achieved. The photo in Figure 3A was taken at a different installation where the customer had programmed the changeover set point so low that the pump never shut off.



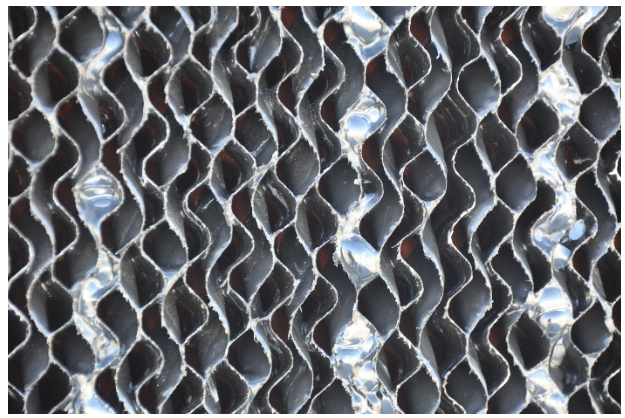


FIGURE 3: (A) MEDIA WITH ALGAE GROWTH AND INCREASED SCALE ACCUMULATION AS A RESULT OF IMPROPER SET-POINT
(B) MEDIA WITH PROPER SET-POINT THAT ALLOWS MEDIA TO DRY-OUT DAILY

ASSESSMENT OBJECTIVES

The overall objective of this assessment was to collect accurate field measurements of performance for evaluating the energy efficiency of RTUs that utilize a dual-evaporative pre-cooler to cool both condenser inlet air and ventilation air. The project was designed to identify the energy savings potential of the retrofit under field operating conditions, as well as to evaluate the feasibility of broadening its implementation in arid climates. Specifically, RTU electricity consumption, cooling capacities, and resulting coefficient of performance (COP) were calculated from field measurements on RTUs with and without the retrofit installed. Comparisons were made for specific units before and after retrofit, as well as based upon side-by-side comparisons of RTUs with and without retrofits under the same weather and operating conditions.

In addition to simply measuring the performance of this retrofit in the field, another objective of this project was to understand the variability in performance of the retrofit, and the factors that contribute to that variability. Factors investigated include pump operation strategies, operating mode control, integration with existing RTUs, and water loop functionalities.

TECHNICAL APPROACH

OVERVIEW OF ASSESSMENT METHODOLOGY

The basic methodology for this project was to conduct approximately one year of monitoring of six RTUs with minute-by-minute data recording, utilizing data storage on an off-site server via a wireless communication on the EDGE network.

Six pre-installed Lennox RTUs on the roof of a big-box building located in Palmdale, CA were selected for the field measurements. Among the six RTUs, four had the evaporative cooling retrofit installed and the other two did not. The untouched RTUs were meant to serve as a baseline for comparison.

The minute-by-minute monitored data for each RTU unit were combined into monthly data sets for post-process performance evaluation of electricity usage, cooling capacity, COP, wet-bulb effectiveness, and the status of the pump on/off switch under various operating modes. The cooling-performance metric chosen for cooling capacity and COP was sensible system cooling capacity, described by equations 3–7.

A statistics package (R) (R Core Team) was chosen for the merge, clean-up, reduction and analysis of raw data, as well as for plotting the results.

INSTRUMENTATION SCHEME

The instrumentation scheme utilized for this study is illustrated schematically in Figure 4, and the performance specifications for the sensors utilized is summarized in Table 3.

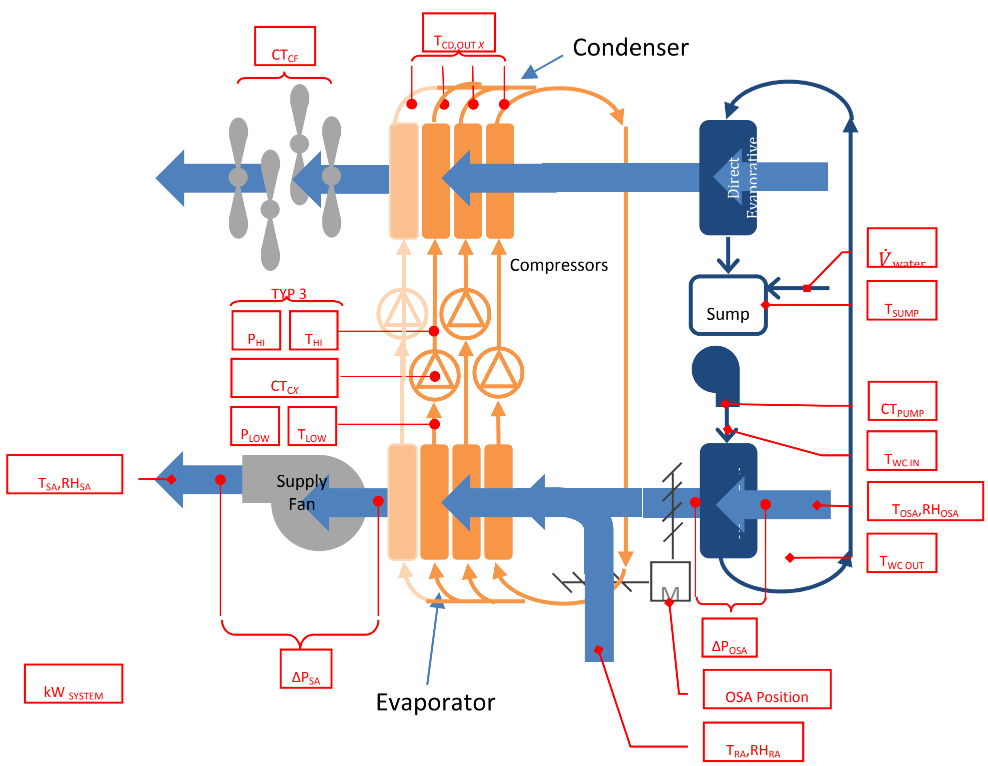


FIGURE 4: INSTRUMENTATION SCHEMATIC

TABLE 3: INSTRUMENTATION TABLE

Name	*Measurement*	*Sensor Model*	*Uncertainty*	*RTUs*
T_{OSA}	Outside Air Temperature	Vaisala HUMICAP HMP110	+/- 0.2 °C	7,8,20
RH_{OSA}	Outside Air Relative Humidity	Vaisala HUMICAP HMP110	+/- 1.1% RH	7,8,20
T_{RA}	Return Air Temperature	Vaisala HUMICAP HMP110	+/- 0.2 °C	7,8,10,11,20,21
RH_{RA}	Return Air Relative Humidity	Vaisala HUMICAP HMP110	+/- 1.1% RH	7,8,10,11,20,21
T_{SA}	Supply Air Temperature	Vaisala HUMICAP HMP110	+/- 0.2 °C	7,8,10,11,20,21
RH_{SA}	Supply Air Relative Humidity	Vaisala HUMICAP HMP110	+/- 1.1% RH	7,8,10,11,20,21
ΔP_{SA}	Supply Fan Differential Static Pressure	Dwyer	+/- 2% FS	7,8,10,11,20,21
ΔP_{OSA}	OSA Differential Static Pressure across Water Coil	Dwyer	+/- 2% FS	7,8,10,11
$\dot{V}_{WATER}$	Water Supply Flow Rate (Total RTU Consumption)	OMEGA FTB 4105 A P 1 pulse per gallon	+/- 1.5% FS	7,8,10,11
OSA Position	RA/OSA Damper Actuator	RA/OSA Damper Actuator	+/- 2 % FS	7,8,10,11,20,21
CT_{C1}	AC Current-Compressor 1	NK AT1-005-000-SP	1.0% FS ACT005-42L-F	7,8,20
CT_{C2}	AC Current-Compressor 2	NK AT1-005-000-SP	1.0% FS	7,8,20
CT_{C3}	AC Current-Compressor 3	NK AT1-005-000-SP	1.0% FS	7,8,20
CT_{C4}	AC Current-Compressor 4	NK AT1-005-000-SP	1.0% FS	NA
$T_{LOW, C1}$	Suction Temperature-Compressor 1	Omega 100 Ω SA1-RT-B	+/- 0.12% at 0°C	7,8,20
$T_{LOW, C2}$	Suction Temperature-Compressor 2	Omega 100 Ω SA1-RT-B	+/- 0.12% at 0°C	7,8,20
$T_{LOW, C3}$	Suction Temperature-Compressor 3	Omega 100 Ω SA1-RT-B	+/- 0.12% at 0°C	7,8,20
$T_{CD\ OUT\ 3\ W}$	Liquid line temperature on Condenser 3 outlet with DC	Omega 100 Ω SA1-RT-B	+/- 0.12% at 0°C	8,20
$T_{HI, C1}$	Discharge Temperature-Compressor 1	Omega 100 Ω SA1-RT-B	+/- 0.12% at 0°C	7,8,20
$T_{HI, C2}$	Discharge Temperature-Compressor 2	Omega 100 Ω SA1-RT-B	+/- 0.12% at 0°C	7,8,20
$T_{HI, C3}$	Discharge Temperature-Compressor 3	Omega 100 Ω SA1-RT-B	+/- 0.12% at 0°C	7,8,20
$T_{CD\ OUT\ 3\ WO}$	Liquid line temperature on Condenser 3 outlet w/o DC	Omega 100 Ω SA1-RT-B	+/- 0.12% at 0°C	8,20
$P_{LOW, C1}$	Suction Pressure-Compressor 1	ClimaCheck 200200, 10*bar*	<1% FS	7,8,20
$P_{LOW, C2}$	Suction Pressure-Compressor 2	ClimaCheck 200200, 10*bar*	<1% FS	7,8,20
$P_{LOW, C3}$	Suction Pressure-Compressor 3	ClimaCheck 200200, 10*bar*	<1% FS	7,8,20
$P_{LOW, C4}$	Suction Pressure-Compressor 4	ClimaCheck 200200, 10*bar*	<1% FS	NA
$P_{HI, C1}$	Discharge Pressure-Compressor 1	ClimaCheck 200100, 35*bar*	<1% FS	7,8,20
$P_{HI, C2}$	Discharge Pressure-Compressor 2	ClimaCheck 200100, 35*bar*	<1% FS	7,8,20
$P_{HI, C3}$	Discharge Pressure-Compressor 3	ClimaCheck 200100, 35*bar*	<1% FS	7,8,20
$P_{HI, C4}$	Discharge Pressure-Compressor 4	ClimaCheck 200100, 35*bar*	<1% FS	NA
CT_{PUMP}	AC Current-Pump	NK AT1-005-000-SP	1.0% FS	7,8,10,11,
$T_{CD\ OUT\ 1}$	Outlet Temperature-Condenser 1	Omega 100 Ω SA1-RT-B	+/- 0.12% at 0°C	7,8,20
$T_{CD\ OUT\ 2}$	Outlet Temperature-Condenser 2	Omega 100 Ω SA1-RT-B	+/- 0.12% at 0°C	7,8,20
$T_{CD\ OUT\ 3}$	Outlet Temperature-Condenser 3	Omega 100 Ω SA1-RT-B	+/- 0.12% at 0°C	7,8,20
$T_{CD\ OUT\ 4}$	Outlet Temperature-Condenser 4	Omega 100 Ω SA1-RT-B	+/- 0.12% at 0°C	NA
T_{SUMP}	Sump Water Temperature	Thermocouple Type T	0.5%	7,8,10,11
$T_{WC\ IN}$	Water Coil Inlet Water Temperature	Thermocouple Type T	0.5%	7,8,10,11
$T_{WC\ OUT}$	Water Coil Outlet Water Temperature	Thermocouple Type T	0.5%	7,8,10,11
KW_{SYSTEM}	System Power Draw	Dent Powerscout 3	0.5%	7,8,10,11,20,21
$CT_{CF\ 1\&2}$	AC Current-Compressor 1&2	NK AT1-005-000-SP	1.0% FS	NA
$CT_{CF\ 3\&4}$	AC Current-Compressor 3&4	NK AT1-005-000-SP	1.0% FS	NA

TRACER GAS AIRFLOW MEASUREMENTS

The mass flow rate of supply air is required to calculate cooling capacity and COP. Since the cooling capacity and mass flow rate are directly related, it is necessary to measure the mass flow rate accurately. To achieve this, we employed a tracer-gas airflow measurement system. The process involved injecting a known mass flow rate of carbon dioxide gas (CO_2) into the return air stream of the unit and measuring the corresponding change in the carbon dioxide concentration in the supply air stream. The injection is controlled and measured by a mass flow controller and the CO_2 concentration is measured by a portable gas analyzer. By subtracting out the ambient-air CO_2 concentration we calculate the change in concentration caused by the injection of CO_2. Combining the time-averaged CO_2 concentration change and CO_2 mass flow rate, the mass flow rate (or standard volumetric flow rate) of air can be calculated. This procedure was used to measure the supply fan flow (across the evaporator).

A similar process was used to measure the outdoor air fraction. In this case the process utilized was similar to the temperature split method used to calculate air flow ratios. In other words, we measured the CO_2 concentration in the outside air, the return air, and the supply air. Performing a mass flow balance, the outside air fraction (OSAF) is calculated as:

$$OSAF = \frac{\dot{m}_{outside}}{\dot{m}_{supply}} = \frac{C_{supply} - C_{return}}{C_{outside} - C_{return}} \quad (1)$$

DETERMINATION OF OPERATING MODE

The operating mode during each one-minute time interval was determined by an examination of various component-operating-status indicators. For instance, the pump on/off status was determined from the measured pump current, CT_PUMP, by choosing a threshold value of 0.1 amps. Similarly, the compressor cycling status was determined from the compressor current readings by choosing appropriate threshold values.

The key control variables for determining the operating mode included:

- *Pump Operation:* CT_PUMP : if CT_PUMP > 0.1V, Pump is On; else, Pump is Off;
- RTU Operation: KW_{system}: if (KW_{system} - $KW_{standby}$)>0.5KW, unit is On; else, unit is Off;
- *Outside Air Damper Position:* OSA_{pos}: if OSA_{pos}>8, damper is fully open; if 2<OSA_{pos} <8 ,damper is in Minimum position; OSA_{pos} <2 damper is closed;
- *Heating On or Off:* $\Delta T = T_{SA} - T_{OSA}$: Temperature difference between supply air and outside air. If (ΔT<15 °F), Heating is Off, otherwise Heating is On;
- *Compressors 1/2 Status:* CT_C1: if CT_C1<0.1V, both compressors 1 and 2 are off; if 0.1<CT_C1<2V, one compressor is On; if CT_C1>2V both compressors 1 and 2 are On;
- *Compressors 3/4 Status:* CT_C3: if (CT_C3<0.1V), both compressors 3 and 4 are off; if (0.1<CT_C3<2V) one compressor is On; if CT_C3>2V both compressors 3 and 4 are On.

Table 4 summarizes the control logic used for determining operating mode (described in Table 2). The threshold values were determined by pre-examining the patterns of time series data for each control variable. Fortunately, the time series data was banded distinctively, such that the threshold values could be easily chosen.

TABLE 4: CONTROL LOGIC FOR DETERMINING OPERATION MODES

	System Power	OSA Fraction	Heating	Compressor(s)		Water Pump
Control Variables	kW_{system}	OSA_{pos}	T_{SA}-T_{OSA}	CT_C1	CT_C3	CT_PUMP
Status	ON | Off	0% | MIN | 100%	On | OFF	0 | 1 | 2	0 | 1 | 2	On | OFF
Thresholds	0.1	2, 8	15^{o}F	0.1, 2V	0.1,2V	0.1V

DATA CONFIDENCE

Table 5 summarizes the uncertainties in the calculated metrics (sensible cooling capacity, coefficient of performance, and wet-bulb effectiveness of the water coil) at peak capacity. These uncertainties were based upon propagating the uncertainties in the primary measurements (Table 3) used for each of the calculated metrics.

TABLE 5: UNCERTAINTY OF CALCULATED METRICS

Calculated Metrics	Coefficient of Performance (COP)	Sensible Cooling Capacity	Wet Bulb Effectiveness
Uncertainly (%)	3.7%	3.6%	5.7%

METHODS & CALCULATIONS

CALCULATION OF PERFORMANCE METRICS:

COEFFICIENT OF PERFORMANCE

The COP for the RTUs was determined at all operating modes and conditions. COP is generally calculated as the ratio of total cooling capacity to electric power consumption:

$$COP \quad [-] = \frac{\dot{H}}{P_{total}} X\, 0.293 \frac{W}{Btu/h} \qquad 2$$

Where: $\dot{H}$ is defined by Equation 4

P_{total} is the total power consumption of the RTU (including all compressors, fans, pump, electronics, etc.) [W]

Note that a COP can be calculated based upon different types of cooling capacity (e.g., total capacity, sensible capacity, space-cooling capacity); however, in this report, the principal cooling capacity that is utilized is the system sensible cooling capacity, calculated based upon Equation 6.

System cooling capacity (total)

The total system cooling capacity was determined as a function of the supply airflow rate and the specific enthalpy difference between a 'virtual-mixed-air' and the supply air, where the virtual-mixed-air enthalpy is the mixed-air enthalpy that would have been seen by the RTU evaporator coil in the absence of the water coil. There is no air that actually has this enthalpy. In reality, the actual mixed air seen by the RTU evaporator coil is a mixture of the return air and outside air that has been cooled by the water coil, however the use of virtual mixed air allows us to take credit for the cooling being provided by the water coil. The virtual enthalpy of the 'virtual mixed air' as defined as:

$$h^*_{MA} = \alpha h_{OA} + (1-\alpha)h_{RA} \qquad 3$$

where α is the outside air fraction, determined by field CO_2 tracer-gas airflow measurement for each RTU (see Table 7)

Correspondingly, the net cooling produced by the system, including what is lost due to fan heat, is:

$$\dot{H}_{system} = \dot{m}_{\mathrm{SA}} \cdot (h^*_{MA} - h_{SA}) \qquad 4$$

Sensible cooling capacity

The principal capacity used in this report is the sensible cooling capacity, which does not include any enthalpy change that goes toward dehumidification. The reason for choosing sensible cooling as the metric is that in many western climates the absolute humidity of outside air is well within the acceptable envelope for human comfort, and any dehumidification can be considered unnecessary. In this case, a comparison of the efficiency for alternative cooling equipment should discount the latent cooling provided by a system that dehumidifies unintentionally. The sensible cooling capacity is calculated based upon the virtual-mixed-air temperature, which is calculated, similar to virtual mixed-air enthalpy, as:

$$T^*_{MA} = \alpha T_{OA} + (1-\alpha)T_{RA} \qquad 5$$

The sensible cooling capacity is calculated according to Equation 6, which is the net sensible cooling produced by the system, including losses due to fan heat.

$$\dot{H}_{sensible} = \dot{m}_{\mathrm{SA}} \cdot C_p \cdot (T^*_{MA} - T_{SA}) \qquad 6$$

The sensible cooling COP therefore, is calculated as:

$$COP_{sensible} = \frac{\dot{H}_{sensible}}{P_{total}} X\ 0.293 \frac{W}{Btu/h} \qquad 7$$

Pre-Cooling of Ventilation-Air

The product air temperature, T_p, is the temperature of the ventilation air being delivered to the RTU after being cooled sensibly by the sump water through a fin coil. It is determined indirectly through an air-water energy balance in the water coil (fin-coil), as is presented schematically in Figure 5. The equation for calculating the product air temperature is:

$$T_p = T_{OSA,in} - \frac{\dot{m}_w c_{pw}}{\dot{m}_a c_{pa}}(T_{w,in} - T_{w,out}) \qquad 8$$

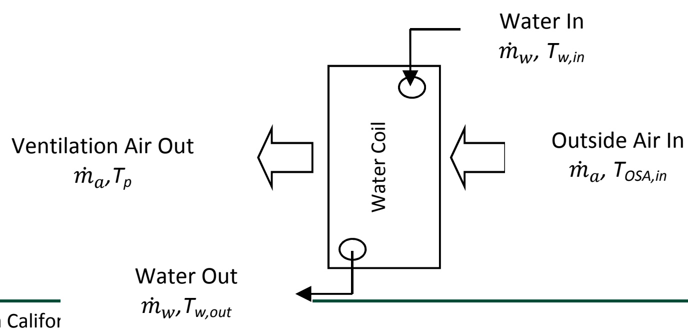


FIGURE 5: ENERGY BALANCE USED TO CALCULATE OUTSIDE AIR DELIVERY TEMPERATURE

To analyze the circuit in Figure 5 we need the measured outside-air and water flow rates. The flow rate of the sump water to the fin-coil was measured for all retrofits. It was measured 5-10 times on each RTU by measuring the time required to produce a measured mass of water in a bucket. The measured flow rates in Table 6 show that the sump-water flow rate to the Water Coil was very consistent between units.

WET-BULB EFFECTIVENESS

The wet-bulb effectiveness for indirect evaporative cooling of the ventilation air (by the water coil) can be calculated according Equation 9. This is the ratio of the change in ventilation-air dry-bulb temperature across the water coil, to the wet-bulb depression of the entering outside air:

$$\mathrm{WBE}_{ind.\ vent.\ cooling} = \frac{T_{OA,in} - T_P}{T_{OA,in} - T_{OA,in}^{wb}} \qquad 9$$

The wet-bulb effectiveness for indirect evaporative cooling of ventilation air is affected by the sump water temperature and the sensible heat exchange effectiveness for the coil utilized.

Separately, wet-bulb effectiveness for the condenser air pre-cooler represents the degree to which evaporation from the direct evaporative media is able to cool condenser inlet air toward the wet-bulb temperature of the outside air. This metric is calculated in a similar way:

$$\mathrm{WBE}_{cond.\ pre-cooling} = \frac{T_{OA,in} - T_{condenser,in}}{T_{OA,in} - T_{OA,in}^{wb}} \qquad 10$$

Physical constraints limit the ability to accurately measure the condenser inlet temperature, and product air temperature. The product air temperature can be calculated according to equation 8, but the condenser inlet temperature is more difficult to estimate because water is introduced to the top of the media at several degrees warmer than the wet –bulb temperature, and must cool by evaporation while flowing down toward the sump. For the results presented in this study, the condenser inlet temperature in equation 10 is estimated to be equal to the measured sump water temperature. In reality, the condenser inlet temperature will be somewhat warmer than the sump water temperature, but this estimate sets a limit for the maximum possible wet-bulb effectiveness that could be achieved for condenser air pre-cooling. The air cannot be cooled below the sump water temperature. This limit for the maximum possible wet-bulb effectiveness can be applied for both the direct evaporative condenser air pre-cooler, and the indirect evaporative cooling of ventilation air.

CUMULATIVE VERSUS STEADY-STATE PERFORMANCE ANALYSIS

Two types of performance results are presented in this report: 1) cumulative performance values, and 2) steady-state performance values calculated whenever the equipment had been running in a particular mode for more than ten minutes. The cumulative values include every minute of the test period, whereas the steady-state values only correspond to periods when the system operated in a particular mode for more than 10 minutes. For the steady-state values, the only data included in the calculation is data acquired after 10 minutes of operation in a particular mode.

In addition to examining RTU performance as a function of outdoor air temperature, the steady state performance data is aggregated into two outdoor temperature bins (75-85^{o}F and 85-95^{o}F) to analyze the performance of the retrofit. This analysis is based upon comparing RTU efficiency and sensible capacity within each of these bins: a) with the retrofit operating during July and August of 2013, and b) with the retrofit removed in October of 2012.

RESULTS

The performance-metric results presented below are based principally on monitoring performed during the month of August in 2013 in Palmdale, California. The key performance metrics (sensible cooling capacity, COP, and water consumption) depend upon one-time measurements of mass flow rates. These results include the mass flow rates of supply air (single speed fans), and the mass flow rates of water from the sump to the water coil (single speed pumps). The water measurements, presented in Table 6, show remarkable consistency between RTUs.

Table 7 presents the total supply-air flow rate (evaporator coil), and outside air fraction, for each of the RTUs, including the baseline RTUs in this case. In addition, the normalized supply air flow rate (cfm/ton) is calculated for each RTU. It can be seen in Table 7 that the outside air fraction varies by a factor of two between RTUs, and that the normalized supply air flow also varies by almost that much. Condenser-coil flow measurements were not conducted, so condenser flow was estimated from equipment manufacturer data. This flow is only utilized to estimate the fraction of water consumption associated with evaporation (as opposed to associated with water bleed used to control the concentration of dissolved solids).

TABLE 6: MEASURED WATER FLOW RATES FROM SUMP TO WATER COIL

RTU#	RTU 7	RTU 8	RTU 10	RTU 11
Measured Water Flow Rate (Gal/Min)	1.55	1.55	1.64	1.58

TABLE 7: MEASURED SUPPLY AIRFLOW RATES AND OUTSIDE AIR FRACTION FOR ALL RTUS

RTU#	RTU 7	RTU 8	RTU 10	RTU 11	RTU 20	RTU21
Measured Supply Air Flow Rate (CFM)	3540	4370	3770	4710	2630	4080
Measured Supply Air Flow Rate (CFM/ton)	236	336	189	269	175	233
Outside Air Fraction	28%	24%	15%	30%	24%	21%
Ventilation Air Flow Rate (CFM)	990	1050	570	1410	630	860

OPERATING MODE

In looking at the monitored data, it quickly became apparent that these RTUs operated in very different modes under the same outdoor air temperature conditions, both between units, and for the same unit. Therefore, it seems appropriate to present the operating mode distributions for each unit, and to compare the distributions of operating modes between units. The results for all units are presented in Figure 6 – Figure 8.

Figure 6 clearly shows that the two baseline RTUs do not show the same behavior with respect to operating modes. It is clear that there is more load on RTU-21, as under the same weather conditions, it spends far more time in multiple-compressor modes, whereas RTU-20 is off much of the time (its thermostat was set in auto mode) and operates mostly with a single compressor.

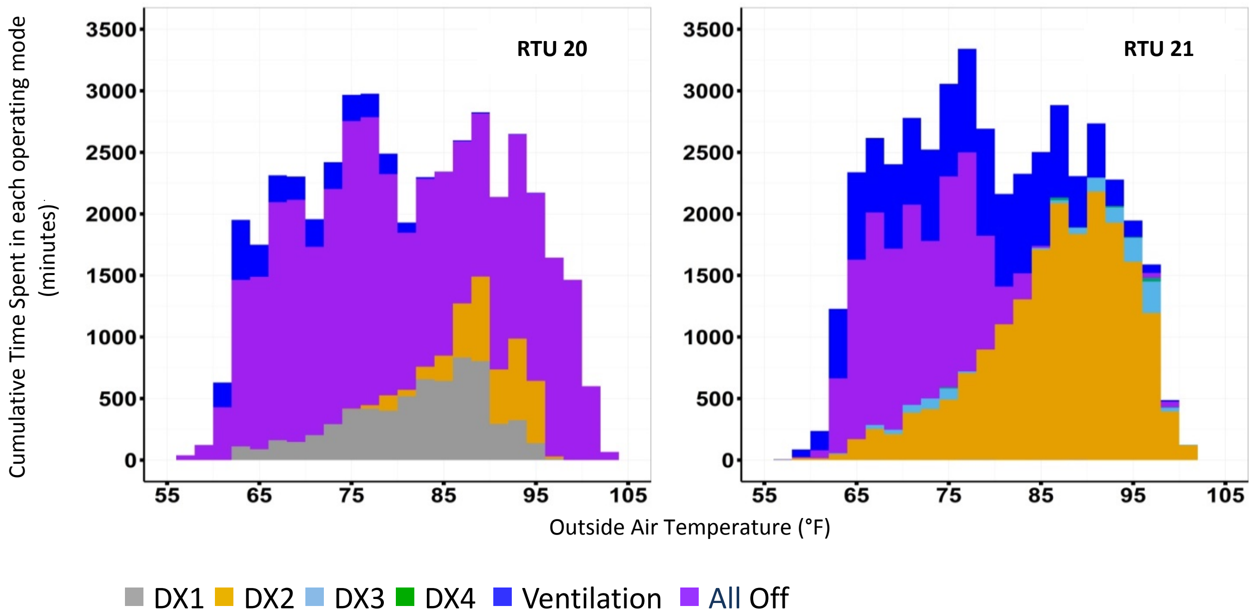


FIGURE 6: CUMULATIVE TIME SPENT IN EACH OPERATING MODE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (^{o}F) (RTU-20 AND RTU-21)

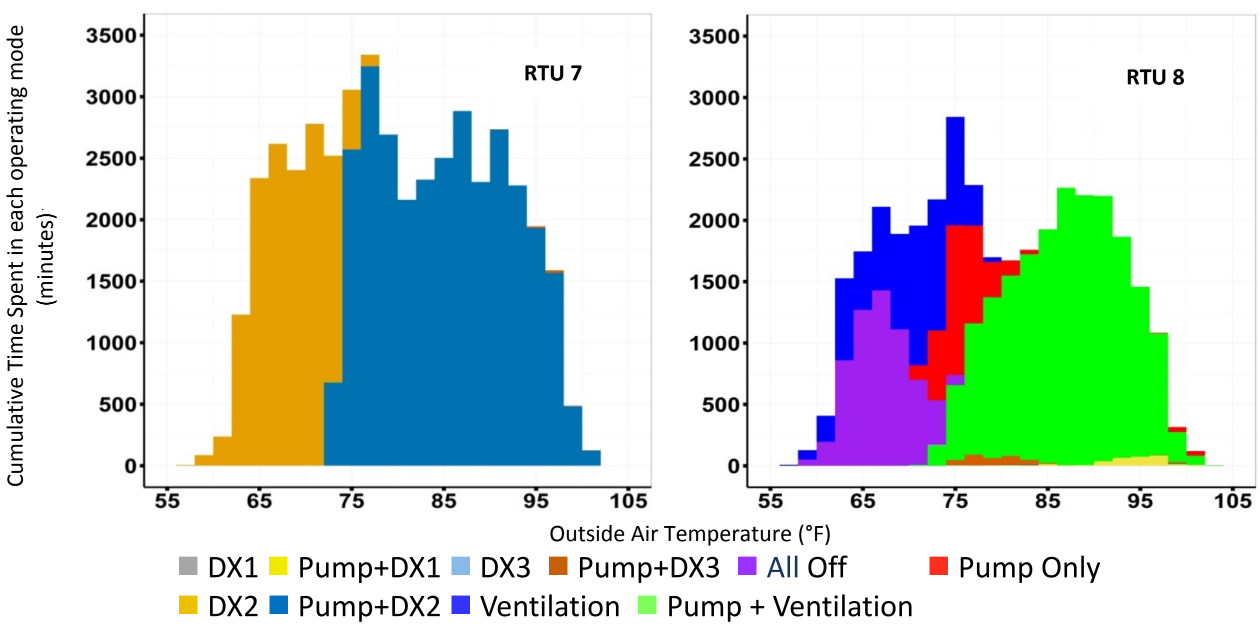


FIGURE 7: CUMULATIVE TIME SPENT IN EACH OPERATING MODE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (RTU -7 AND RTU -8)

Figure 7 shows the operating modes of two of the retrofitted RTUs, as does Figure 8 for the other two retrofitted RTUs. Similar to the baseline RTUs, it is clear that the retrofitted RTUs each have their own particular operating-mode profiles. However there are many more operating modes for the units with dual-evaporative pre-coolers. The simplest of the four is RTU-7 in Figure 7, which operates in two-compressor cooling only, and clearly shows the pump coming on consistently at approximately 75^{o}F outdoor air temperature. At the other end of the spectrum is RTU-8, which shows essentially no compressor use, operating almost exclusively in ventilation mode, and showing the same pump-on transition at approximately 75^{o}F outdoor air temperature. RTU-8 also shows a particularly inefficient operating mode around 75^{o}F outdoor air temperature, namely pump operation with the RTU off completely (red section of plot). This stems from the fact that the controls for the retrofit and RTU are neither integrated, nor interlocked.

The remaining two RTUs in Figure 8 show relatively complex behavior. RTU-10 behaves similarly to RTU-8, the difference being that it operates one and two compressors for a reasonable fraction of the time above an outdoor temperature of 75°F. RTU-11 shows the largestvariation with outside temperature, operating with multiple compressors above about 80°F, and transitioning off, or into ventilation-only mode below about 75°F.

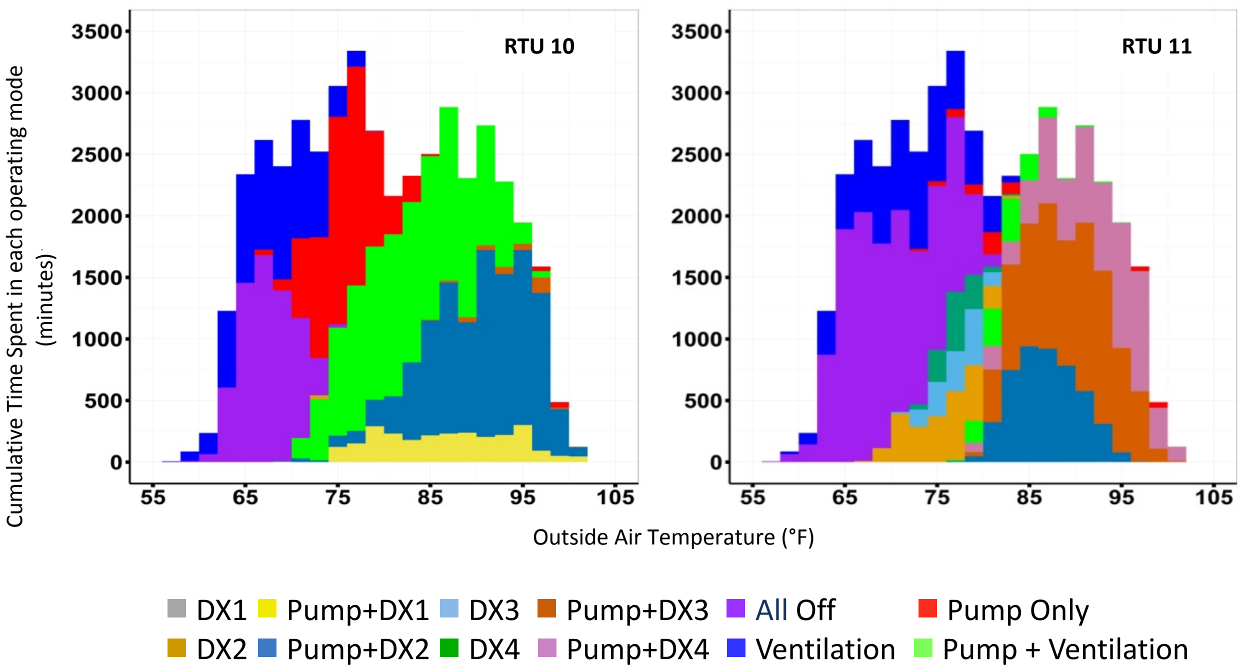


FIGURE 8: CUMULATIVE TIME SPENT IN EACH OPERATING MODE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (°F) (RTU -10 AND RTU -11)

PERFORMANCE METRICS FOR ALL SIX UNITS MONITORED

As noted above, the key metrics used to describe the performance of each of the RTUs are: a) sensible-cooling COP, b) sensible cooling capacity, and c) system power draw. These metrics are presented below as a function of outdoor drybulb temperature and operating mode.

COEFFICIENT OF PERFORMANCE

Figure 9 plots the COP of the baseline RTUs as a function of outdoor air temperature and operating mode. This figure indicates that the COP for both units is roughly 2, and that it does not depend on outdoor air temperature to any significant extent. This could be considered surprising, as laboratory tests of direct expansion (DX) RTUs typically show a strong negative correlation between outdoor air temperature and efficiency. Our explanation for this lack of a trend is that the outdoor air being introduced for ventilation purposes is also increasing the temperature seen by the indoor coil, which will tend to counteract the impact of increased air temperature at the condenser. Stated another way, the increase in outdoor temperature is not significantly increasing the temperature differential seen by the compressor. RTU-20 is running at 24% outdoor air, and RTU-21 is running at 21% outdoor air. One other potentially odd result is that the COP of RTU-20 seems to be somewhat higher with two compressors operating, as compared to one compressor.

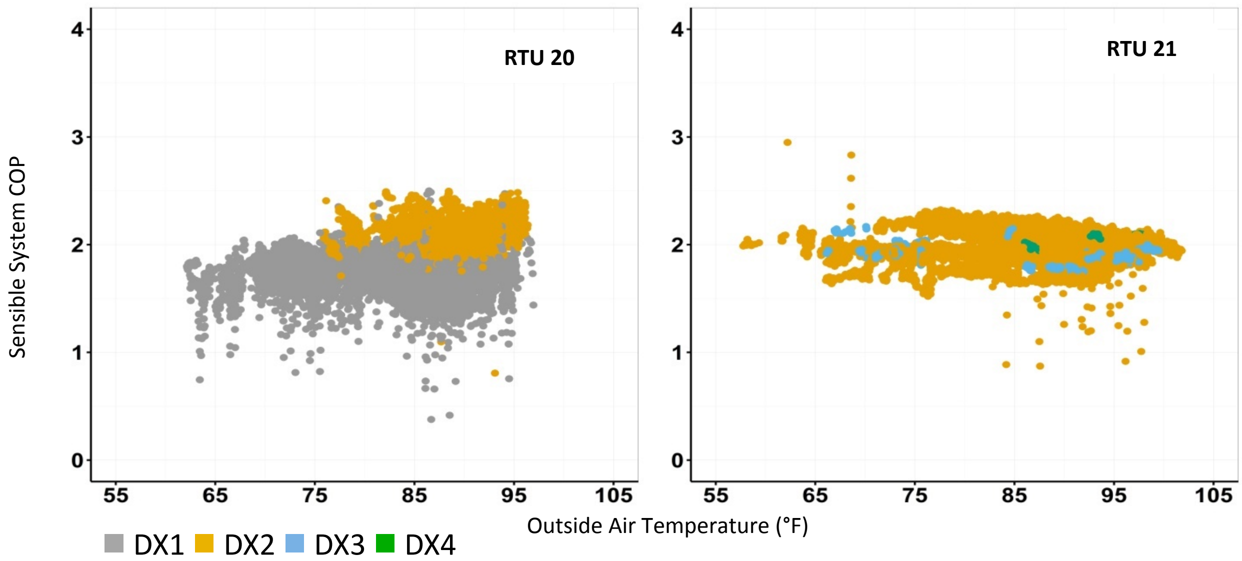


FIGURE 9: SENSIBLE SYSTEM COEFFICIENT OF PERFORMANCE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (RTU-20 AND RTU-21)

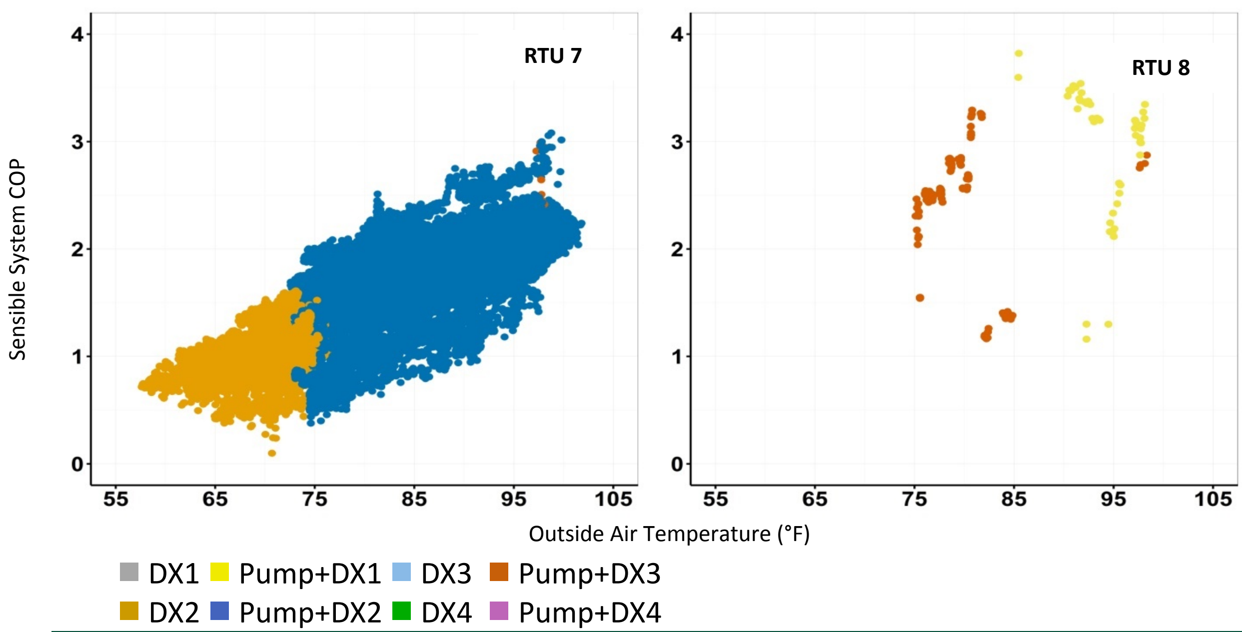


FIGURE 10: SENSIBLE SYSTEM COEFFICIENT OF PERFORMANCE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (^{O}F) (RTU -7AND RTU -8)

Figure 10 plots the COP of two of the retrofitted RTUs as a function of outdoor air temperature and operating mode. In the case of RTU-7, there appears to be a clear demarcation at approximately 75^{o}F between operation with and without the pump in operation, and there is also a clear trend towards higher efficiency at higher outdoor temperatures (NOTE: the demarcation is not as clear as it appears in the plot, as the overlap due to hysteresis between pump on-off cycle cannot be seen in this plot). This increase in efficiency with outdoor temperature is completely opposite to the outdoor temperature trend associated with a DX system operated at 0% outdoor air. For RTU-7, in addition to the impact of outdoor air (28% outdoor air) on the indoor coil temperature (as was seen for the baseline units), the retrofit is actually able to provide more evaporative cooling at higher outdoor air temperatures, due to larger wet-bulb depression at higher temperatures. As will be seen below, the average wet-bulb depression for RTU-7 was 20^{o}F between 75-85^{o}F outdoor dry-bulb temperature, versus 28^{o}F between 85-95^{o}F. One other observation on RTU-7 is that its efficiency is clearly lower than the baseline units at cooler temperatures. We investigated several possible explanations for this discrepancy (e.g. supply airflow, indoor wet-bulb temperature), but were not able to find a satisfactory explanation. However, we did not do a refrigerant charge test. As for RTU-8, there is very little COP data, as its compressors rarely operated.

Figure 11 provides COP plots for the two remaining retrofitted RTUs. In the case of RTU-10, essentially all of the compressor operation occurs at temperatures when the pump is in operation, and the efficiency appears to be pretty much independent of outdoor air temperature. However, this RTU had the lowest outdoor air fraction, which means that there is less potential for: a) capturing indirect evaporative cooling of the ventilation air, and b) the evaporator inlet temperature increasing with outdoor air temperature. This can explain why the efficiency did not increase with outdoor temperature, as was seen for RTU-7.

RTU-11 shows the largest derivative with respect to outside air temperature. Just like RTU-7, the efficiency increases with outdoor temperature, once again contrary to the performance of 0%-outside-air, DX-only RTUs. In addition, the impact of turning on additional compressors for RTU-11 is opposite to the performance trend observed for RTU-20. Namely, RTU-11 show a steady decrease in efficiency as more compressors are turned on. The decrease in efficiency of RTU-11 associated with turning on additional compressors is not surprising, as the sequence of operations is such that compressors 1 and 2 utilize the outside condensers served by the evaporative pre-cooler, whereas compressors 3 and 4 utilize the inside condensers that are not exposed to pre-cooled air. In addition, RTU-11 shows the clearest step change (increase) in efficiency for all compressor modes when the evaporative media is activated.

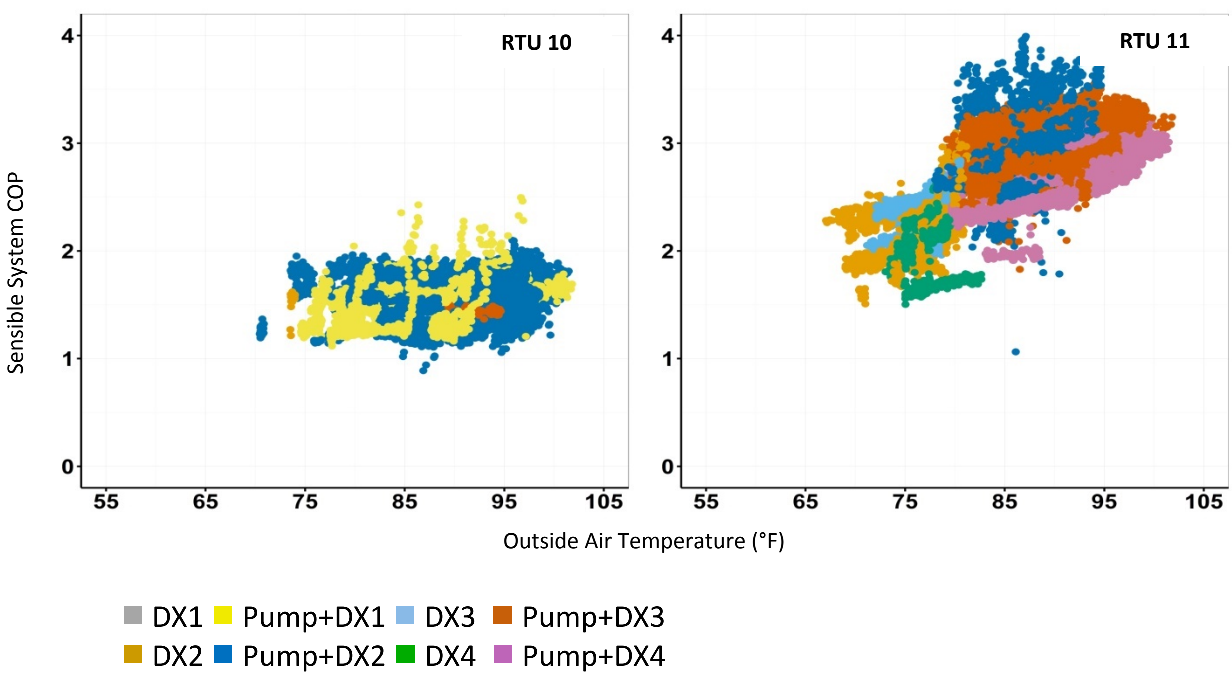


FIGURE 11: SENSIBLE SYSTEM COEFFICIENT OF PERFORMANCE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (^{O}F) (RTU -10 AND RTU -11)

SENSIBLE COOLING CAPACITY

Figure 12 – Figure 14 are plots of the sensible system cooling capacity of the baseline RTUs and retrofitted RTUs as a function of outdoor air temperature and operating mode. Figure 12 shows the sensible system capacity of the two baseline RTUs, and indicates the expected increases in capacity as more compressors are turned on, as well as roughly the same capacity for both RTUs with two compressors in operation. It is worth noting that the sensible system capacity, just like the COP, does not decrease with increasing outdoor air temperature, although this might not be the case if we were looking at total room capacity. shows that the sensible capacity of RTU-7 has a clear upward trend with increasing outdoor air temperature, consistent with the trend seen for the COP of this unit. The capacity trends in Figure 14 show that for RTU-11 the sensible system capacity increases steadily with increasing

outdoor air temperature, and that the upward trend is much less pronounced for RTU-10, consistent with the COP data. For both RTUs the sensible system capacity clearly increases as more compressors are turned on. RTU-10 operates almost exclusively with the water pump running, whereas RTU-11 indicates that the pump turns on at approximately 80°F outdoor temperature.

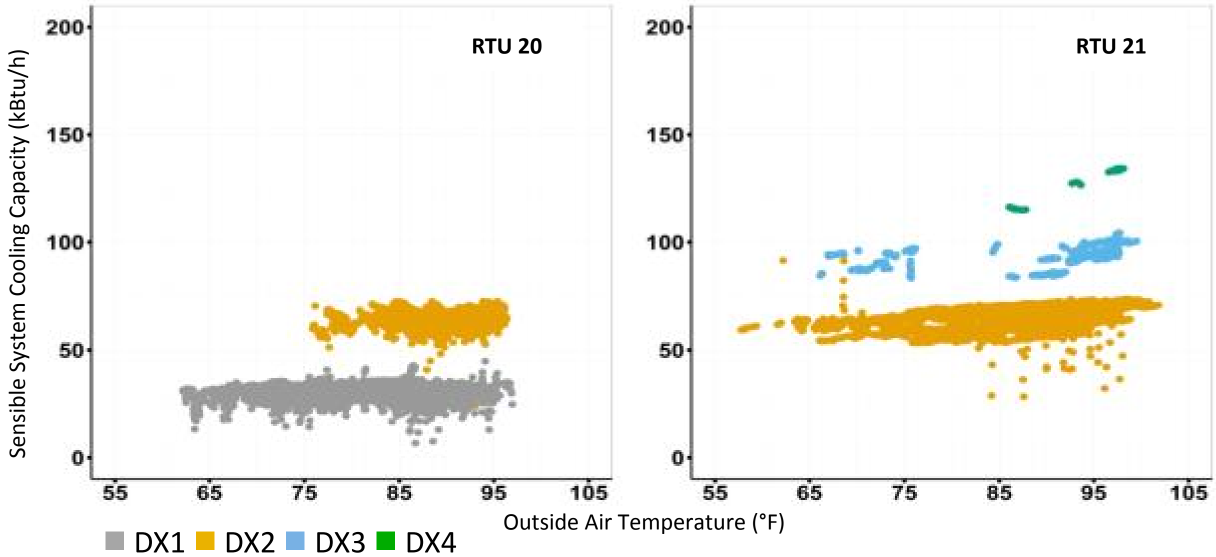


FIGURE 12: SENSIBLE SYSTEM COOLING CAPACITY AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (RTU-20 AND RTU-21)

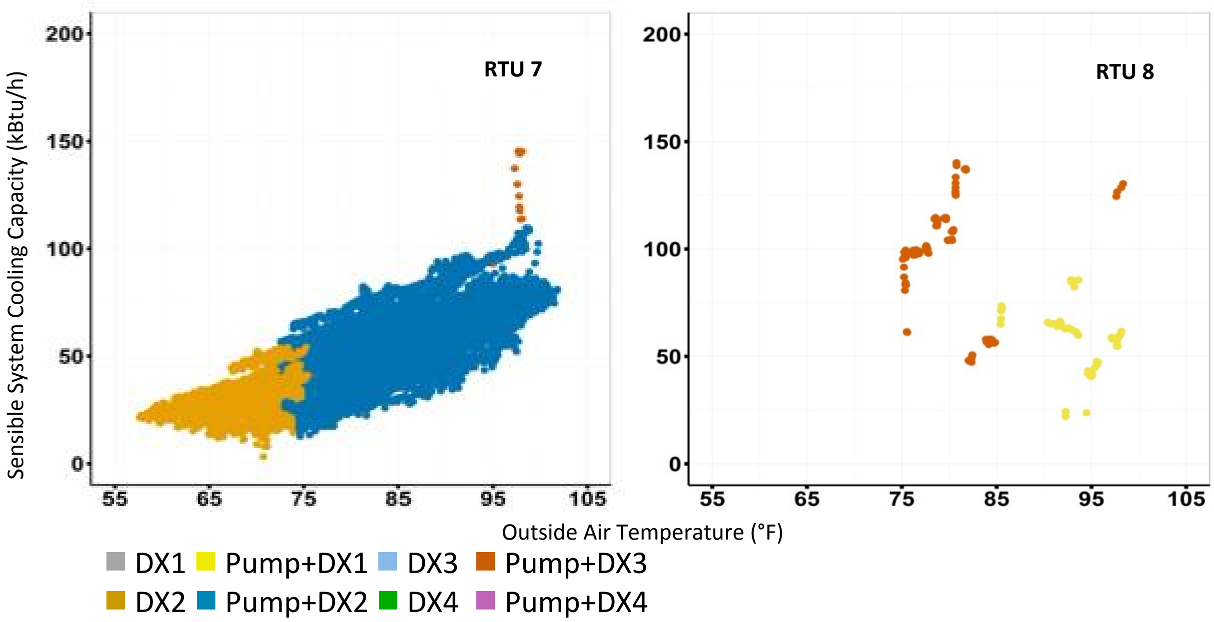


FIGURE 13: SENSIBLE SYSTEM COOLING CAPACITY AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (RTU -7 AND RTU -8)

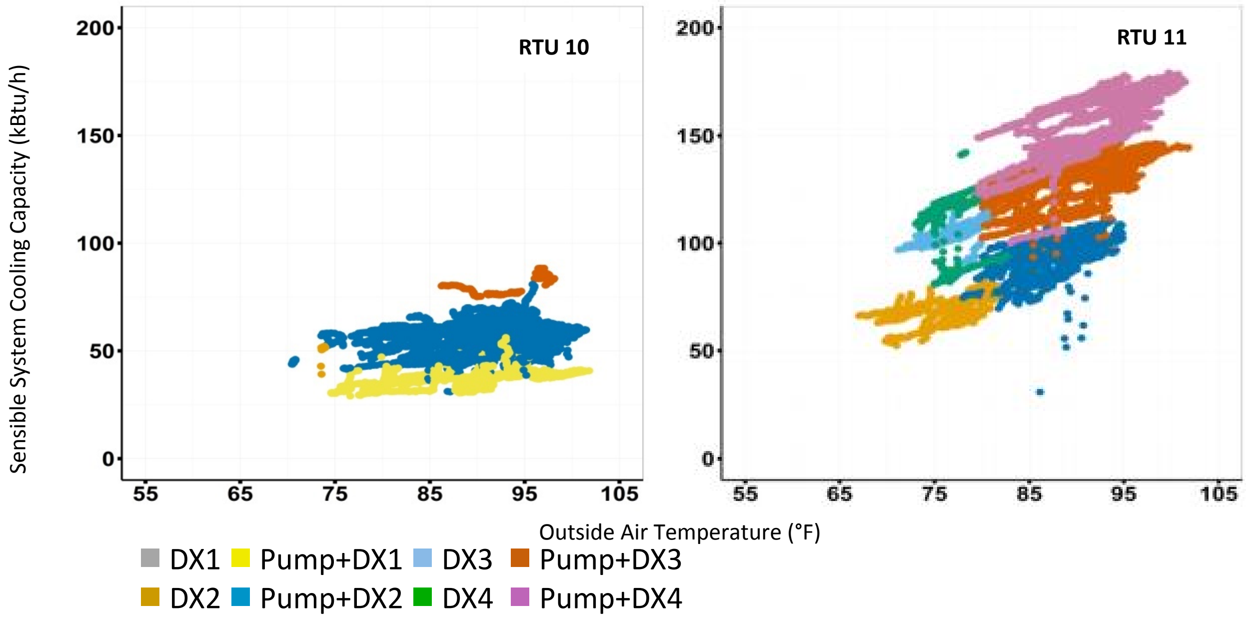


FIGURE 14: SENSIBLE SYSTEM COOLING CAPACITY AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (RTU -10 AND RTU -11)

Figure 15 – Figure 17 are plots of the total sensible cooling delivered (kBtu) in each mode of operation for the month of August 2013 as a function of time of day. Figure 15 shows the sensible system capacity of the two baseline RTUs, while and Figure 17 show the same for the retrofitted RTUs. Although the relative amount of cooling provided in the different modes of operation could be inferred from the number of points in each mode of operation in Figure 12 – Figure 14, Figure 15 – Figure 17 provide the cumulative amount of sensible cooling provided by each mode (capacity times time). Looking at Figure 15 – Figure 17 all together, it is clear that some RTUs are providing much more cooling than others. It is also clear that only one of the units (RTU-11) is doing much cooling with more than two compressors in operation.

The negative values for RTU-21 and RTU-8 (as well as the smaller negative values for the other RTUs) stem from periods when the ventilation fans are running without any compressors in operation, in which case the heat generated by the fans shows up as negative sensible cooling. It should be noted that the air being brought in from outside during these periods is generally cooler than room air, however in calculating sensible system cooling, we are counting the thermal energy difference across the RTU, not the cooling effect for the room (see Equation 6).

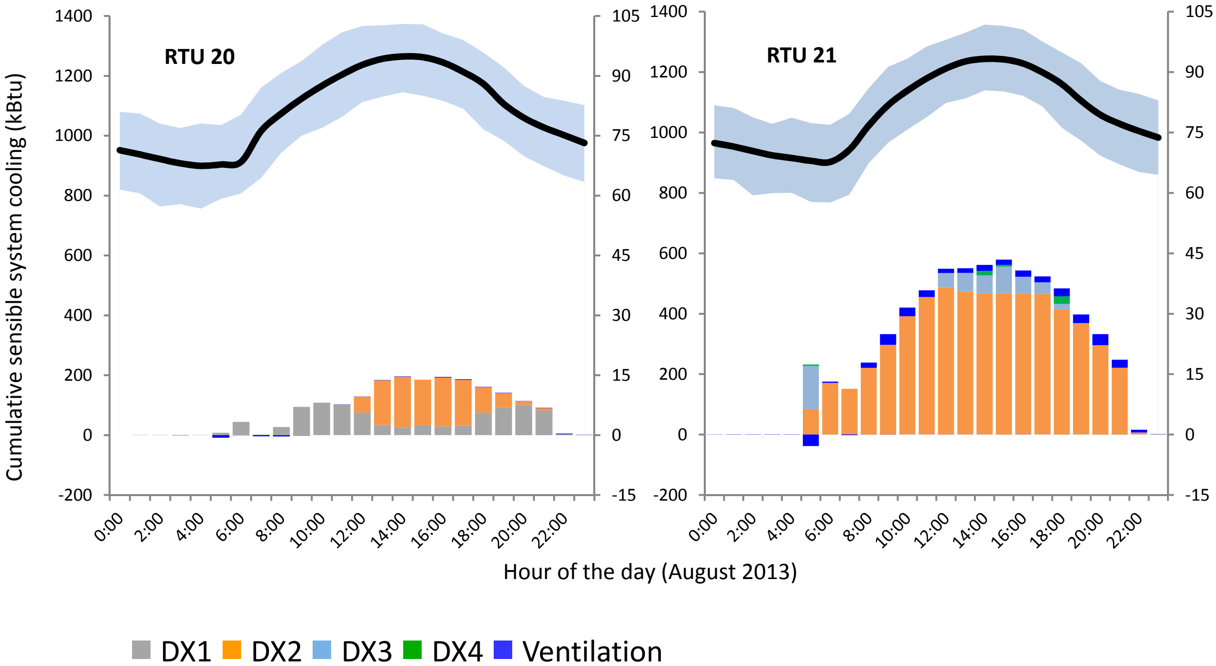


FIGURE 15: CUMULATIVE SENSIBLE SYSTEM COOLING IN EACH MODE OF OPERATION (LEFT AXIS) AND MIN MAX & AVERAGE OUTSIDE TEMPERATURE (RIGHT AXIS) AS A FUNCTION OF HOUR OF THE DAY (RTU 20 & RTU 21)

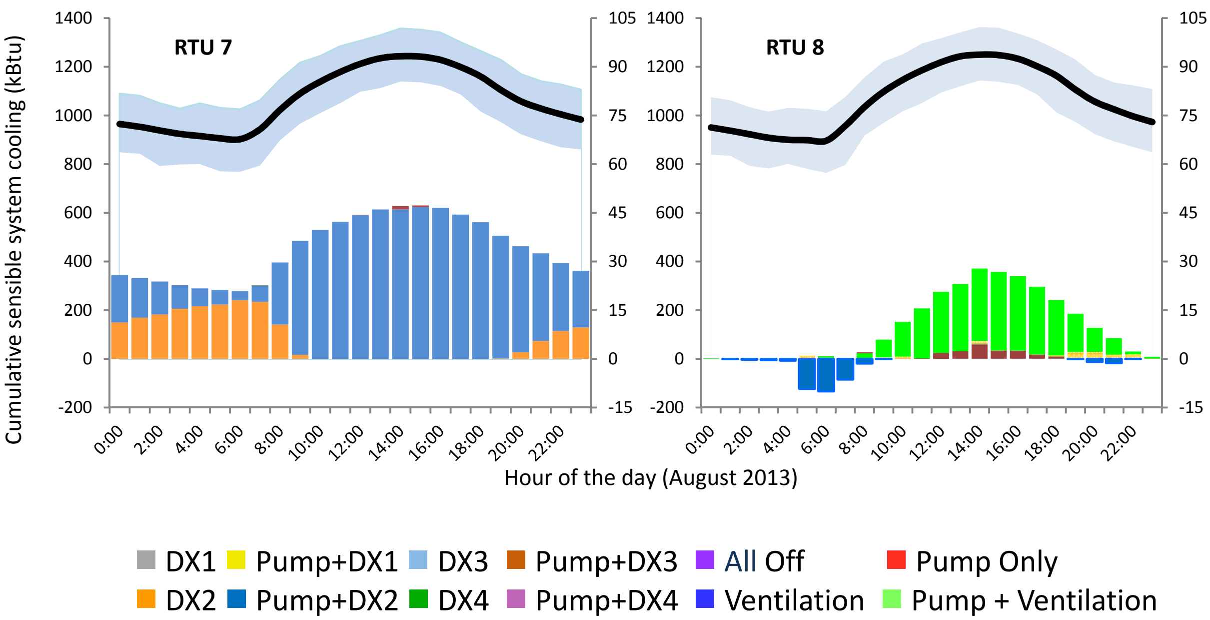


FIGURE 16: CUMULATIVE SENSIBLE SYSTEM COOLING IN EACH MODE OF OPERATION (LEFT AXIS) AND MIN MAX & AVERAGE OUTSIDE TEMPERATURE (RIGHT AXIS) AS A FUNCTION OF HOUR OF THE DAY (RTU 7 & RTU 8)

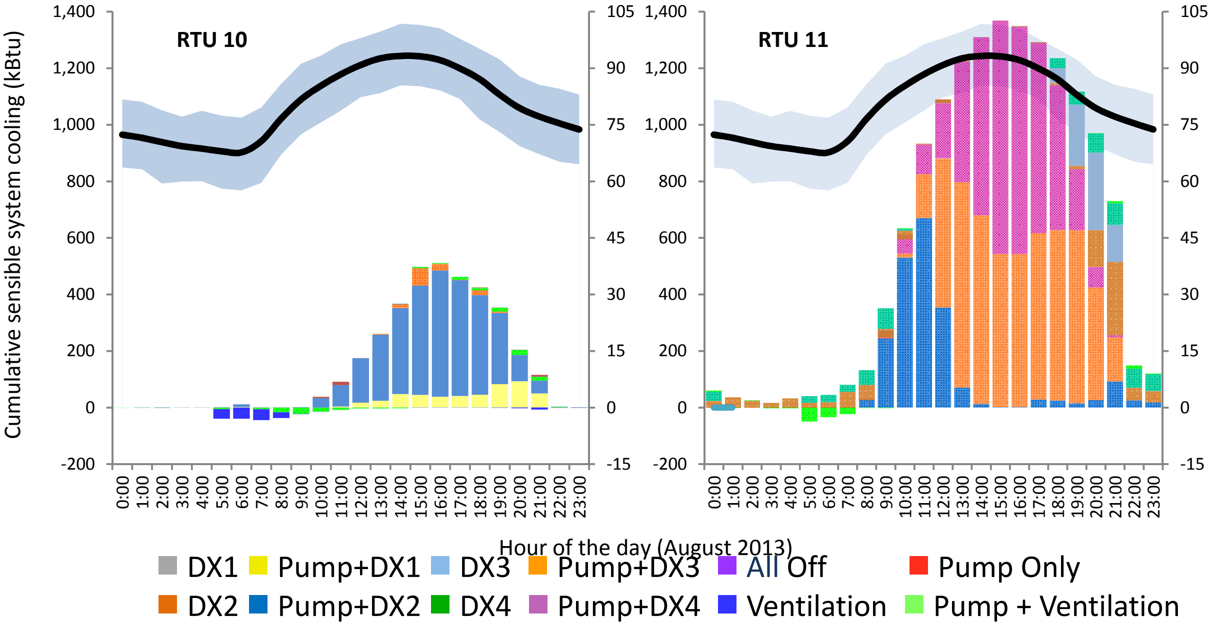


FIGURE 17: CUMULATIVE SENSIBLE SYSTEM COOLING IN EACH MODE OF OPERATION (LEFT AXIS) AND MIN MAX & AVERAGE OUTSIDE TEMPERATURE (RIGHT AXIS) AS A FUNCTION OF HOUR OF THE DAY (RTU 10 & RTU 11)

ELECTRICAL USAGE

Figure 18 – Figure 20 are scatter plots of the electrical power draw of each of the RTUs in the different modes of operation, once again as a function of outdoor air temperature. Looking at all of the plots together, two things become apparent: a) the increased power draw associated with turning on additional compressors is clearly indicated, and b) some of the RTUs shows the expected increase in power draw with increasing outdoor air temperature, and others do not. Expanding upon the second observation, in Figure 18, RTU-20 shows no impact of outdoor temperature on power draw, whereas RTU-21 shows a clear trend with outdoor temperature. In Figure 19, RTU-7 shows a mild but apparent increase in power draw with outdoor temperature, with and without the retrofit pump in operation, whereas RTU-8 shows a trend with three compressors in operation, but not with two compressors (note that the data for RTU-8 is very limited).

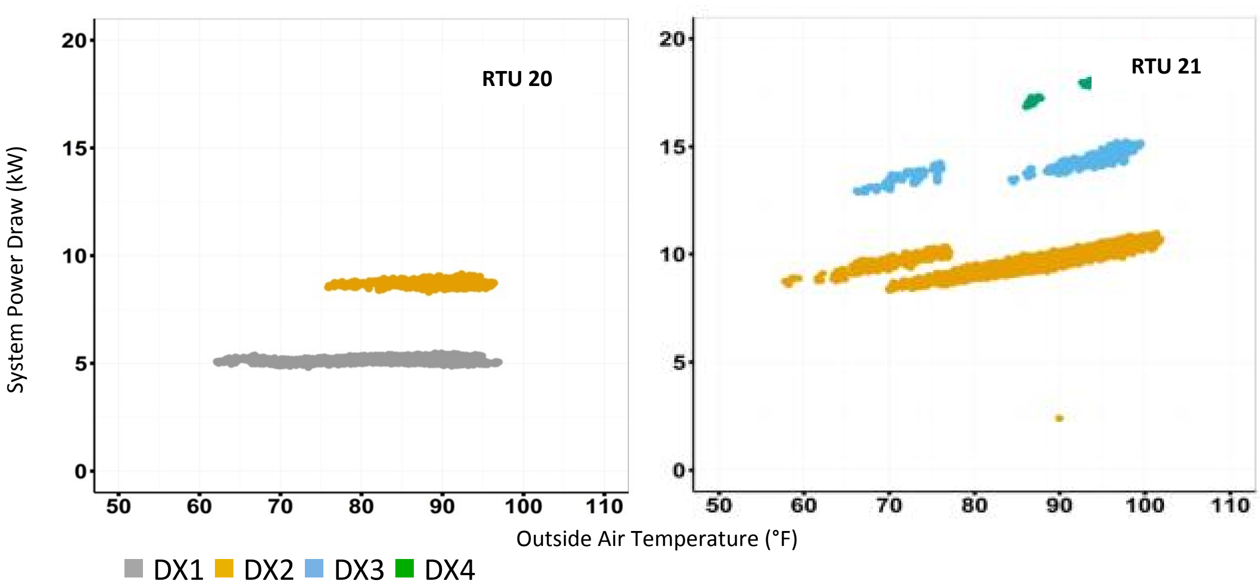


FIGURE 18: SYSTEM POWER DRAW IN EACH MODE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (RTU-20 AND RTU-21)

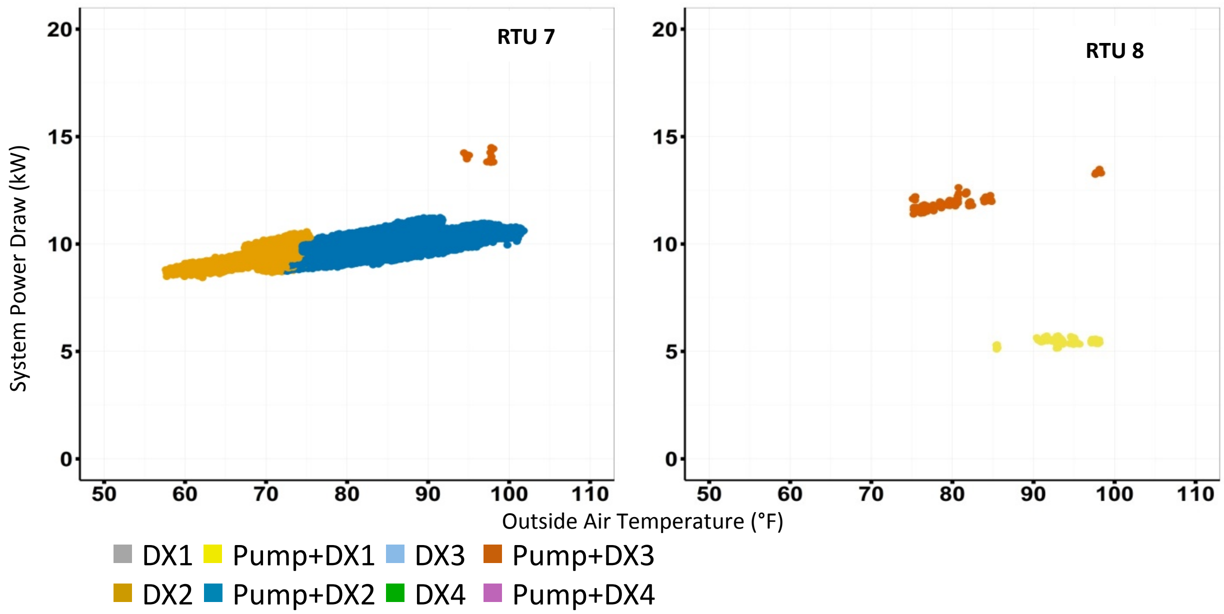


FIGURE 19: SYSTEM POWER DRAW IN EACH MODE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (RTU -7 AND RTU -8)

Turning to Figure 20, the data for RTU-10 is ambiguous, showing no trend with outdoor temperature with one compressor in operation, yet a potential trend with three compressors running. RTU-11 has the broadest data set and shows the clearest trends with respect to number of compressors, outside air temperatures, and operation with and without the pump in operation. Looking closely at this data it appears that: a) there are two groupings of data for each operating mode (similar to RTU-21 and RTU-7), b) there is a downward shift in all groupings associated with the pump turning on that is particularly clear with less compressors in operation, and c) the slope with respect to outdoor temperature seems to increase as the number of compressors in operation is increased.

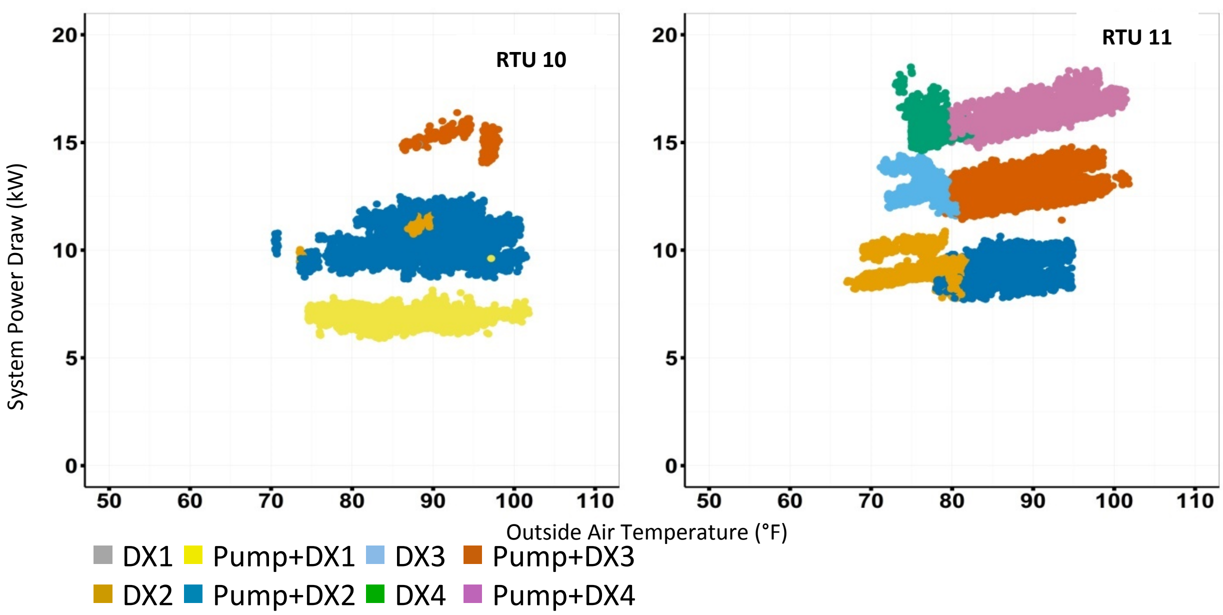
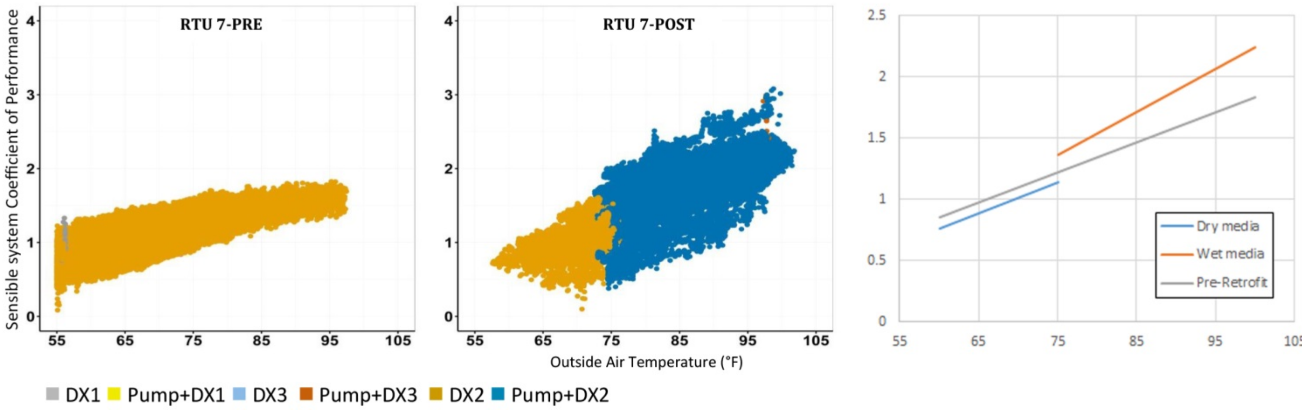


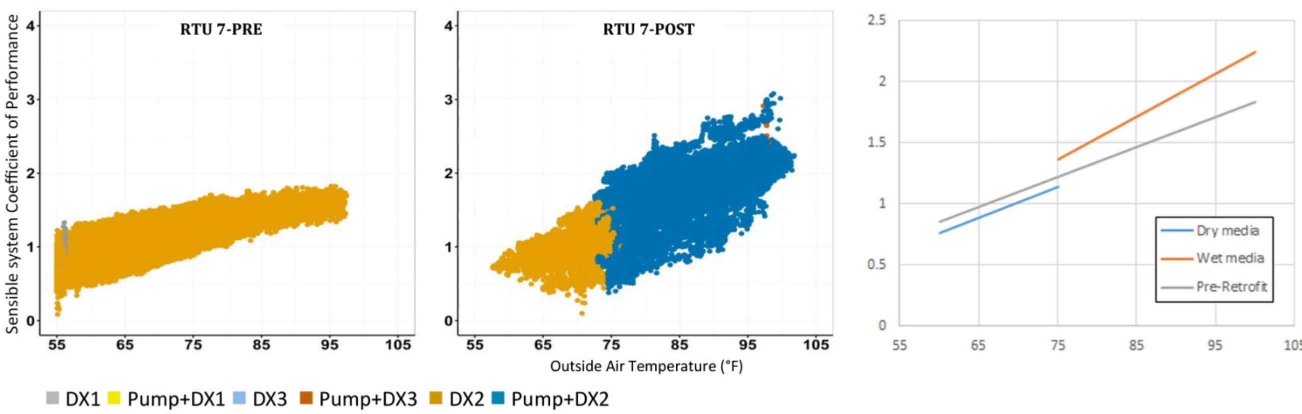
FIGURE 20: SYSTEM POWER DRAW IN EACH MODE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (RTU -10 AND RTU -11)

COMPARISON OF PRE- AND POST- RETROFIT PERFORMANCE FOR FOUR MONITORED RTUS (OCTOBER 2012 VS AUGUST 2013)

In addition to comparing the performance of different RTUs with and without the dual-evaporative pre-cooling system during the same time period, data was taken for all four retrofitted RTUs before and after retrofit. Figure 21 and Figure **22** compare the power draw of each RTU in October 2012 (pre-retrofit) with its power draw in August 2013, both as a function of outdoor air dry bulb temperature. In Figure 21, the pre-post comparison for RTU-8 does not provide any useful information because the system rarely operates with any compressor cooling. For the performance summary analysis, the data from July 2013 was used to augment the August data for RTU-8. On the other hand, RTU-7 appears to have directly comparable data pre- and post-retrofit. There appears to be approximately 10% less power draw at 95^{o}F after the retrofit. Figure **22** does not provide much in the way of comparable data pre and post retrofit.



provides an analogous comparison of pre and post sensible capacity data for RTU-7, which is the RTU for which the pre-post comparison provides the most robust data. Figure 24 provides a comparison of pre- and post-retrofit COP for RTU-7. The comparisons in



and Figure 24 show that sensible capacity increases by approximately 15-20% at 95^{o}F OSA due to the retrofit, and that efficiency (or COP), which depends on both power draw and capacity, increases by 25-30% at 95^{o}F OSA due to the retrofit. These figures also include linear regressions of the data pre- and post-retrofit. These regressions clearly illustrate that the retrofit reduces capacity and efficiency somewhat when the media is dry (due to condenser air flow resistance). They also clearly show immediate increases in capacity and efficiency when the water is turned on, as well as ever increasing improvements in capacity and efficiency as the outdoor air temperature increases. Figure 25 shows the COP data for RTU-11 pre and post retrofit, however there is not much overlap of operating modes at the same outdoor temperature for the two periods (we do not have an explanation for why RTU-11 appeared not to operate at high outdoor air temperatures during the pre-retrofit period).

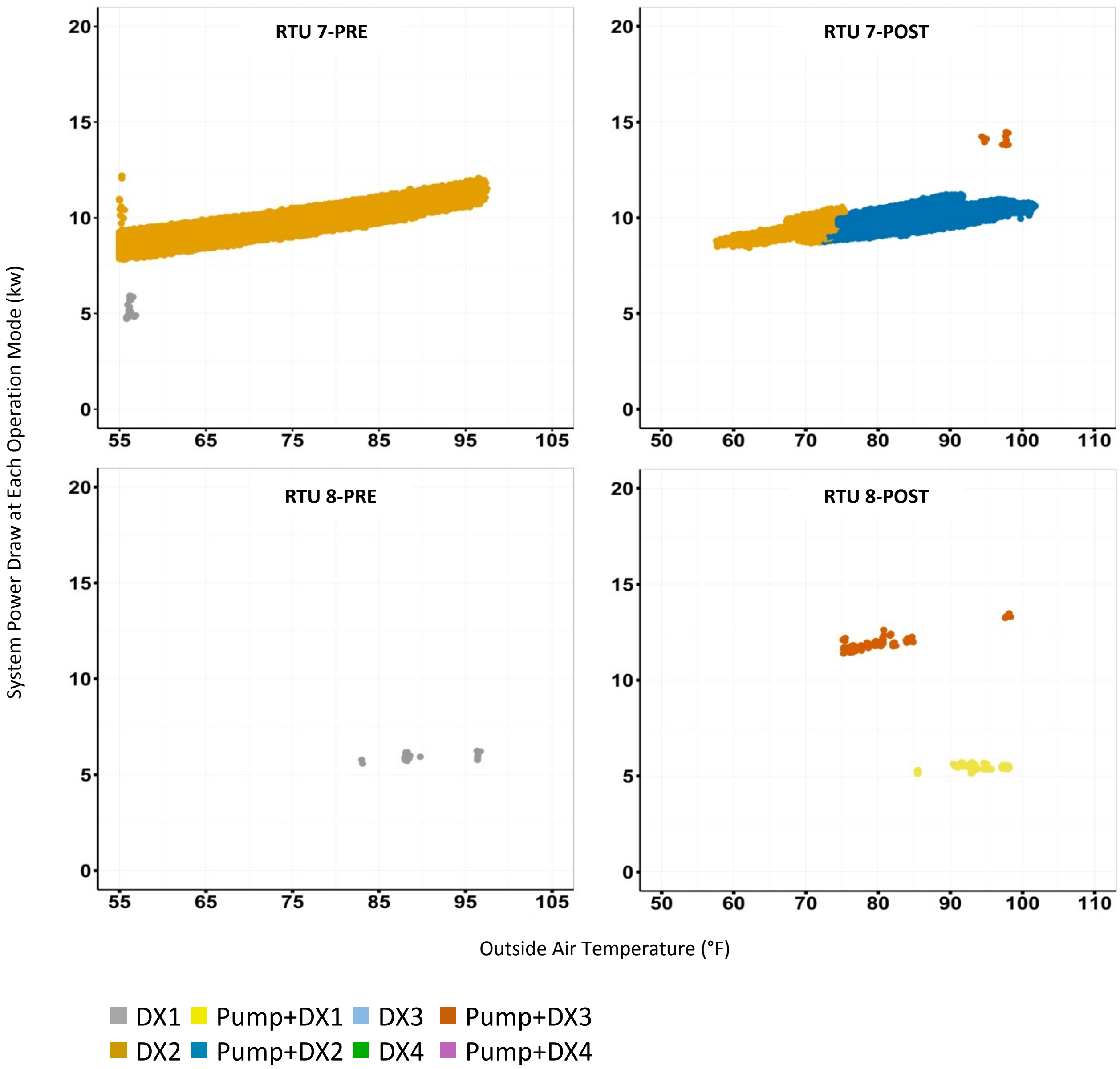


FIGURE 21: PRE-POST COMPARISON OF SYSTEM POWER DRAW IN EACH OPERATING MODE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (RTU-7 AND RTU-8)

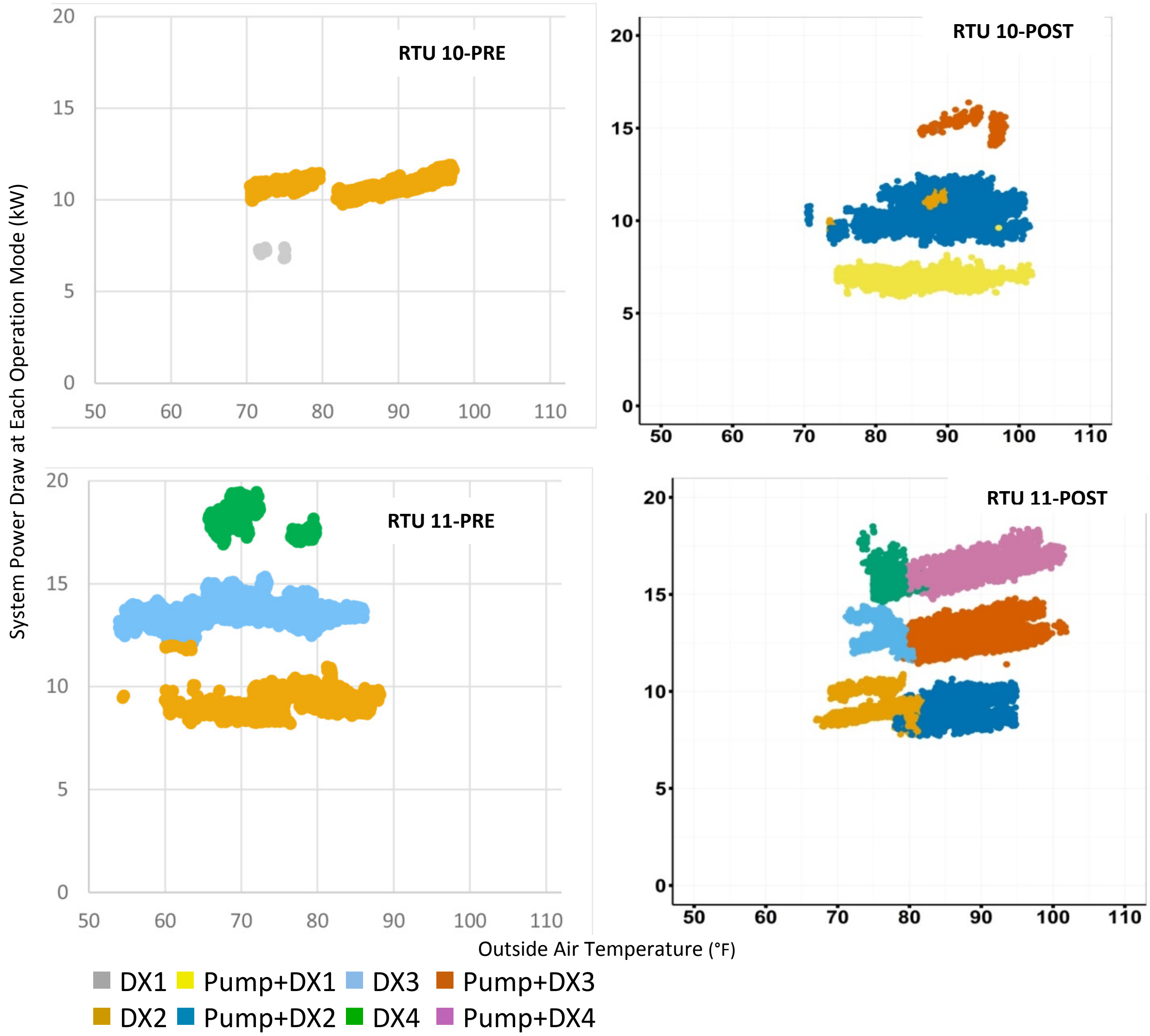


FIGURE 22: PRE-POST COMPARISON OF SYSTEM POWER DRAW IN EACH OPERATING MODE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (RTU-10 AND RTU-11)

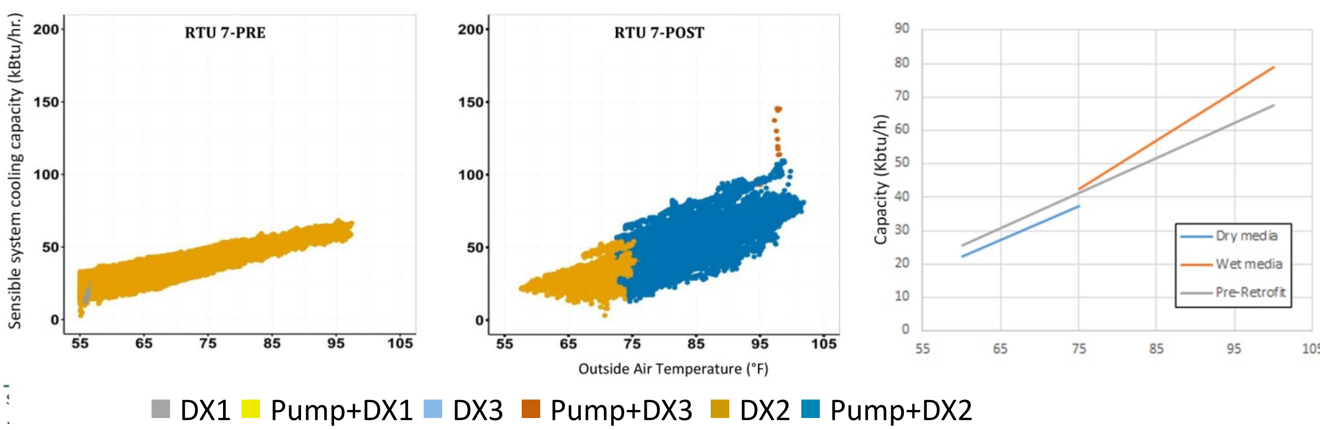


FIGURE 23: PRE-POST COMPARISON OF SENSIBLE SYSTEM COOLING CAPACITY IN EACH OPERATING MODE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE FOR RTU-7

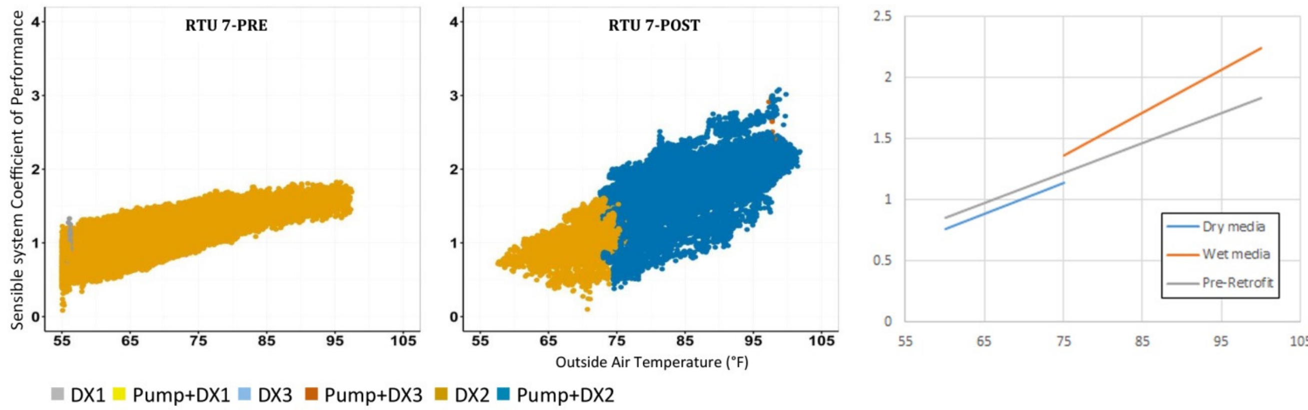


FIGURE 24: PRE-POST COMPARISON OF SENSIBLE SYSTEM COEFFICIENT OF PERFORMANCE IN EACH OPERATING MODE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE FOR RTU-7

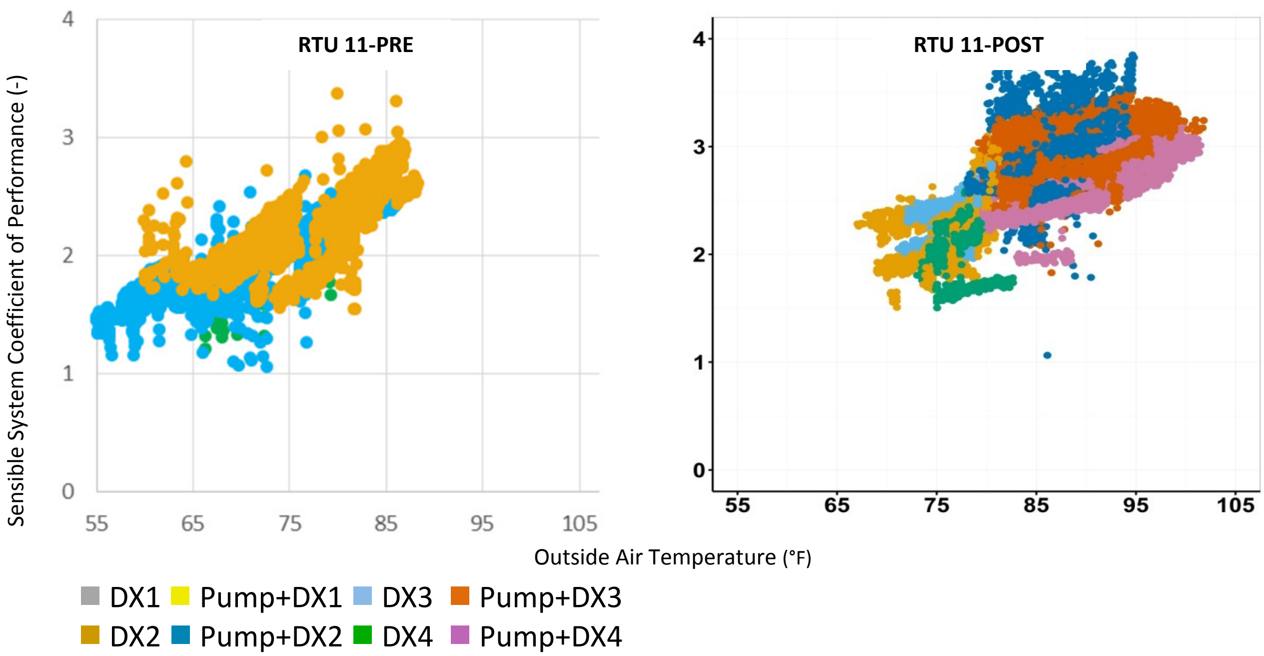


FIGURE 25: PRE-POST COMPARISON OF SENSIBLE SYSTEM COEFFICIENT OF PERFORMANCE IN EACH OPERATING MODE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE FOR RTU-11

PERFORMANCE CHARACTERISTICS FOR THE DUAL-EVAPORATIVE PRE-COOLING SYSTEM

In addition to analyzing the overall impact of the retrofit on the RTUs, data was taken to analyze the internal performance of the dual-evaporative pre-cooling technology. Specifically, the operation of the pump was monitored directly, and the performance of the water coil heat exchanger for the ventilation air was analyzed. Starting with the operation of the pump, Figure 26 presents the distribution of temperatures at which the pump turned on and turned off, for all four retrofitted RTUs. Taken as a whole, there are three key observations to be made based upon Figure 26: a) there is significant hysteresis between the pump turning on and turning off, and b) the transition temperature is not consistent between RTUs, and c) the transition temperature is not consistent for an individual RTU. More specifically, RTU 11 differs from the others in that it shows the least amount of hysteresis, and transitions the pump on and off at a noticeably higher outdoor air temperature. RTU-11 seems to be transitioning around 80°F, whereas the other three RTUs all transition around the target temperature of 75°F. Some implications of these results are discussed in detail in "Discussion'.

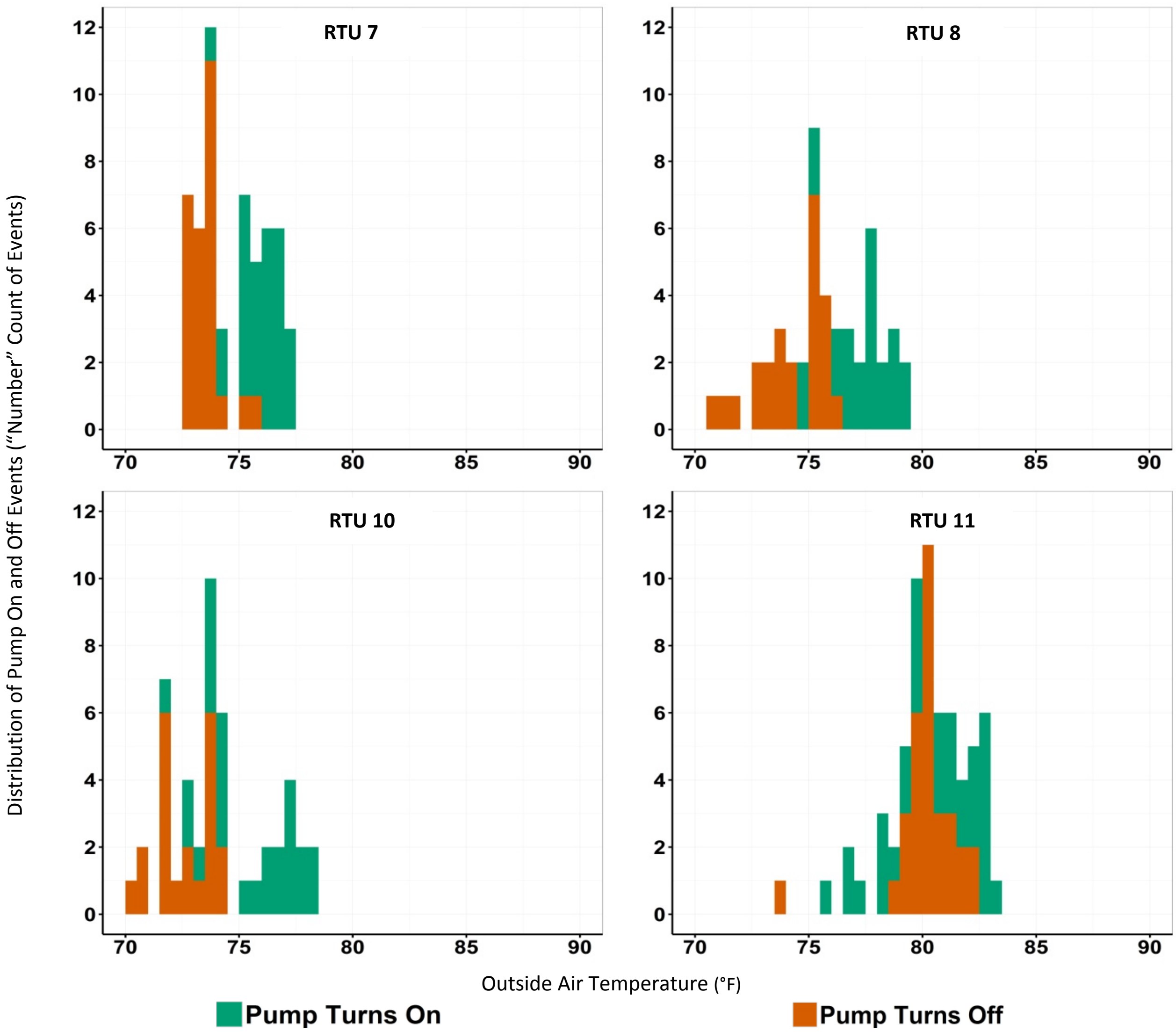


FIGURE 26: DISTRIBUTION OF PUMP ON AND OFF EVENTS AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (^{O}F) FOR ALL RETROFITTED RTUS (AUGUST 2013)

ANALYSIS OF WATER SIDE PERFORMANCE (SEPTEMBER 2013)

Although the previously reported performance principally corresponded to observations with the dual-evaporative pre-cooler in August 2013, data for the water temperature measurements from this period was not reliable because some of the sensors were exposed to solar radiation for part of the day. Instrumentation was upgraded in September 2013, so results for water-side performance are based upon data from September 2013. The results of these measurements are presented in **Error! Reference source not found.** through Figure 29.

Error! Reference source not found. presents the measured sump temperature for all four RTUs fitted with the dual evaporative system. These results provide confirmation that there is a major difference between operating in ventilation-only mode versus compressor mode. Namely, in ventilation-only mode the evaporative media does not bring the sump water as close to the outdoor-air wet-bulb temperature. For all RTUs, the sump temperature is much closer to the outdoor wet-bulb temperature whenever the compressor and condenser-fan are running. One way to think of this is that the evaporation rate when the condenser fans are off is too low to remove as much heat from the ventilation air as compared to when those fans are running.

One final point relative to **Error! Reference source not found.** is that these sump water temperatures measured in the field (even with the condenser fan running) are somewhat higher than what was measured by laboratory tests. Lab tests indicated that sump water was reliably 1.5 F warmer than wet-bulb (Woolley 2012) across a range of conditions, while the measurements from this study indicate sump water temperature can be between 1-5°F warmer than wet bulb. Although we did not find concrete evidence for it, the higher temperatures observed in the field may have been caused by solar gains.

Error! Reference source not found. presents the maximum possible wet-bulb effectiveness for indirect evaporative cooling of ventilation air, estimated based upon the sump-water temperature. This metric basically represents how close the ventilation air could possibly approach the wet-bulb temperature of the outdoor air, which would correspond to the maximum potential for the dual evaporative pre-cooler to absorb heat from the ventilation air stream. The results in **Error! Reference source not found.** mirror those in **Error! Reference source not found.**, which fall into two distinct categories, points measured when there is condenser airflow (i.e. with at least one compressor running), and points measured when the pump is circulating water and there is ventilation airflow, but no condenser fans are running. In general, the wet-bulb effectiveness is significantly higher when the condenser fan is moving air.

RTU-7 operates only with the condenser fan running, while RTU-8 operates almost exclusively in ventilation-only mode, and the other two RTUs operate in both modes. These results suggest that when the condenser fan is not running there isn’t enough air flow across the evaporative media to cool the water down as close to the wet-bulb temperature of the outdoor air. Note that the pump and indoor evaporator fan always run at the same speed, so the only things that change at the water coil are the incoming outdoor air temperature and the incoming water temperature. The results for RTU-11 show the same difference between operation with and without the condenser fans in operation.

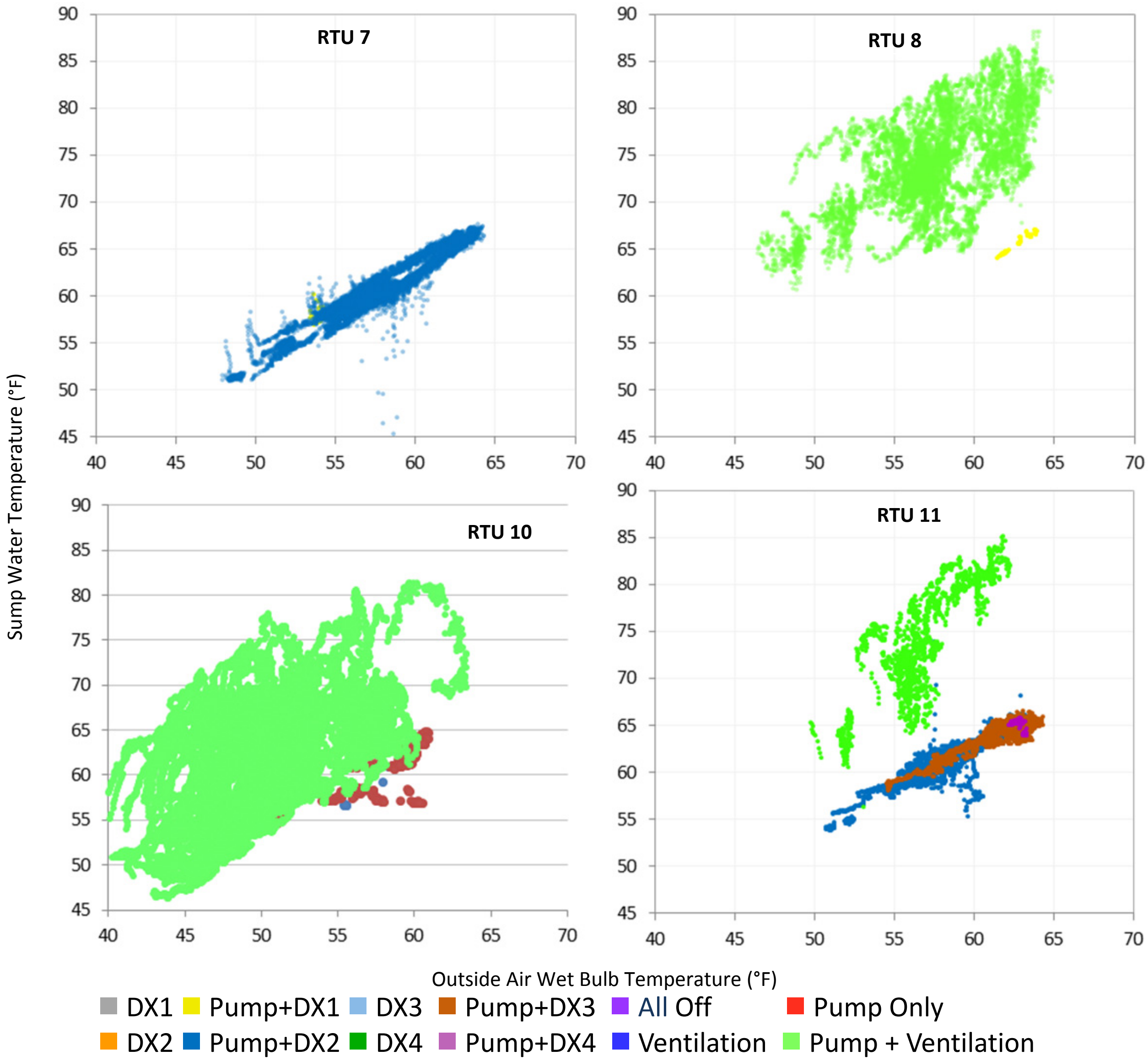


FIGURE 27: SUMP WATER TEMPERATURE AS A FUNCTION OF OUTSIDE AIR WET-BULB TEMPERATURE FOR DIFFERENT OPERATING MODES (SEPTEMBER 13-30 2013)

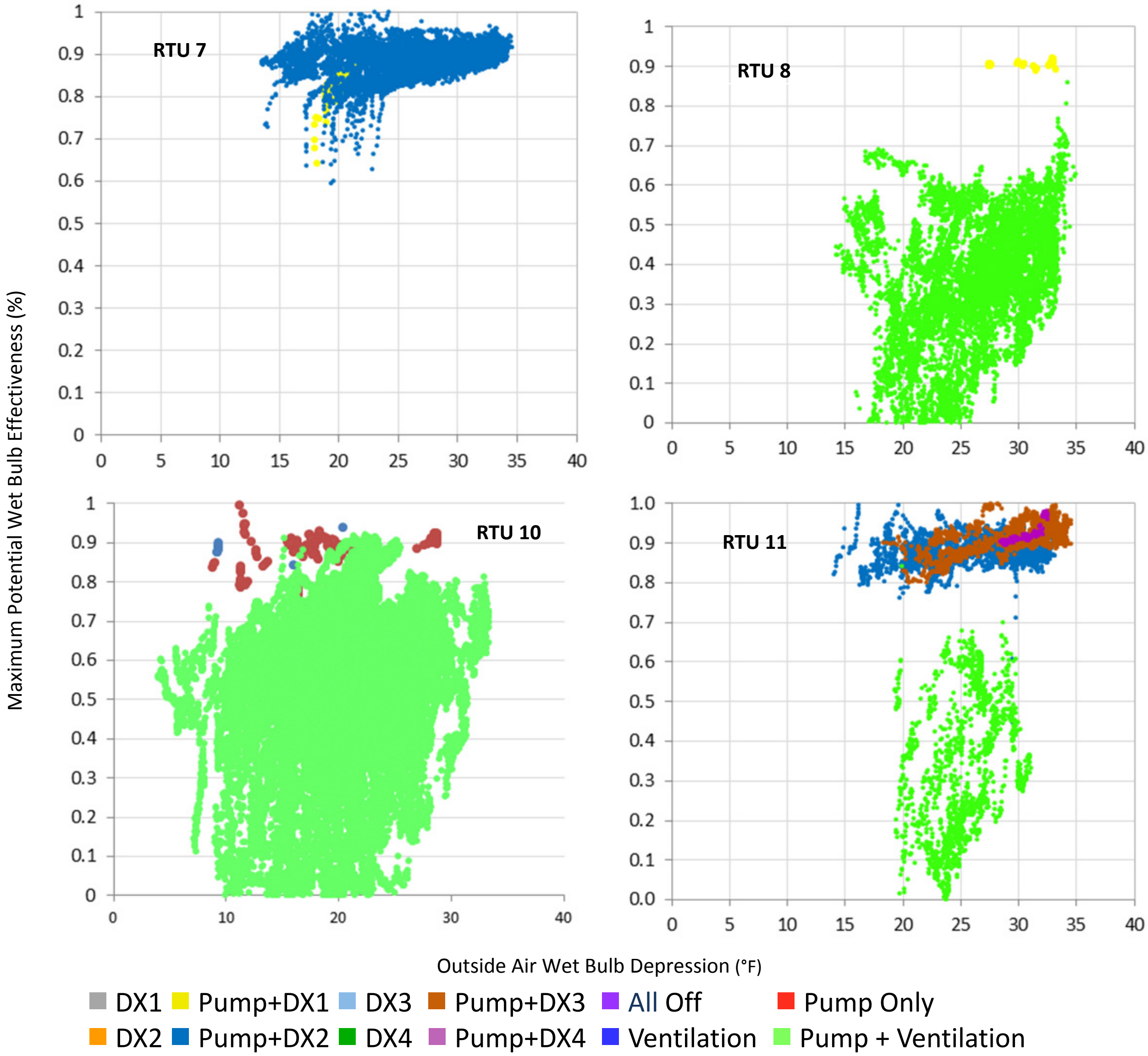


FIGURE 28: MAXIMUM POTENTIAL WET-BULB EFFECTIVENESS FOR EITHER CONDENSER AIR PRE COOLING OR VENTILATION AIR COOLING, AS PREDICTED FROM SUMP WATER TEMPERATURE (SEPTEMBER 13-30 2013)

Figure 29 presents the sensible ventilation cooling provided by each of the RTUs, calculated based upon the water flow rate through the ventilation coil and the temperature rise of that water across the coil whenever the pump is operating. It is noteworthy that the absolute ventilation cooling is the highest for RTU-7 and RTU-11 when operating in a compressor mode with condenser fans, and lowest for RTU-8 and RTU 10 which operated mostly in ventilation-only mode, without any condenser fans in operation. Overall, average ventilation cooling rates range between 4 and 7 kBtu/h with the condenser fans off and 22 to 25 kBtu/h when the condenser fans are running (depending upon the RTU). RTU-11 has the largest self-consistent data set that illustrates the impact on ventilation cooling of having the condenser fans in operation. When the data is based upon the average wet bulb depression, turning on the fans appears to increase the ventilation cooling rate by a factor of 4.6 for RTU-11. A similar analysis for RTU-8 indicates a ratio of 2.5, although there are very few data points with the condenser fans operating for RTU-8. There is not enough data to verify this, however it is worth noting that the ventilation air flow rate should affect the impact of condenser fans on ventilation cooling rates. When the pump runs but the condenser fans do not run, water is still cooled by evaporation, and this cooling is transferred to the ventilation air, but the cooling capacity is significantly reduced because the evaporation rate is reduced. Evaporation from the media without the condenser fans is driven by air flow created by wind and free convection. Variations in wind speed and direction may be cause of the large scatter in ventilation cooling rates when the condenser fans are not operating.

To put the data in Figure 29 into perspective, when the condenser fans are operating the cooling capacity from the ventilation coil increases consistently as the outside air wet bulb depression increases, such that the cooling capacity from the ventilation coil is greatest under peak outdoor air conditions. For RTU-7, the peak ventilation cooling capacity is almost 40 kBtu/h, which represents roughly 35% of its cooling capacity (DX-2). For RTU-11, ventilation cooling represents 20-25% of its peak cooling capacity, depending upon how many compressors are operating (20% for DX-4 vs. 25% for DX-3).

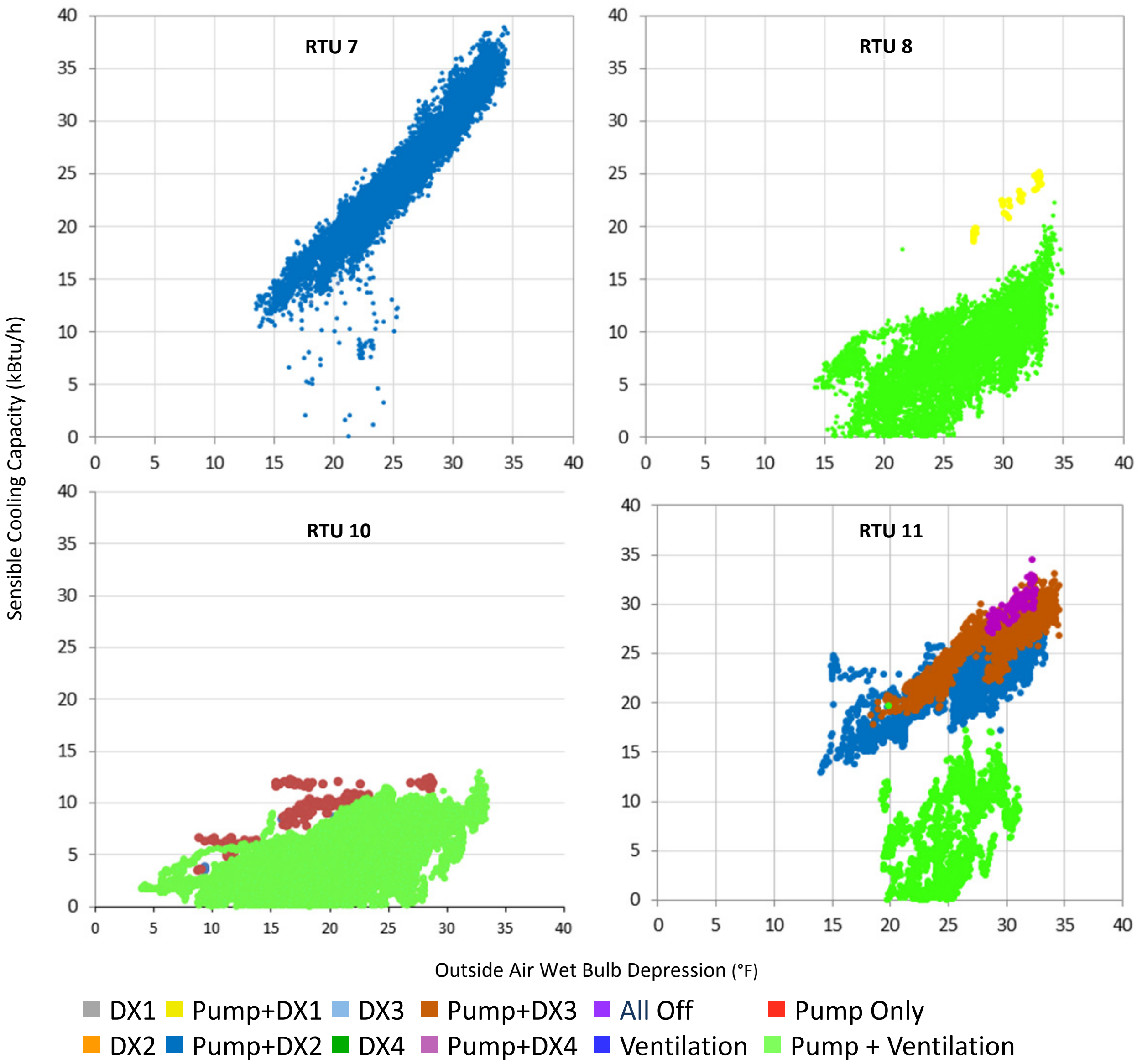


FIGURE 29: SENSIBLE COOLING CAPACITY FOR VENTILATION AIR COOLING COIL AS A FUNCTION OF OUTSIDE AIR WET-BULB DEPRESSION (°F) (SEPTEMBER 13-30 2013)

ANALYSIS OF WATER CONSUMPTION (JULY-AUGUST 2013)

The water consumed by the retrofitted RTUs was monitored by measuring the make-up water delivered to each of the RTUs using paddle-wheel flow meters. The make-up water includes evaporated water, as well as all bleed water used to maintain water quality. We discovered that the water use of the different RTUs differed dramatically, and that, due to the fact that this system maintains water level in the sump by trickling water in at a very low flow, the water meters on RTU-10 and RTU-11 were unable to record water flow rates. Based upon the minimum resolution of the flow meters (0.22 gal/min), the total water use by RTU-10 could not be more than 10 gal/h on average, and the total water use by RTU-11 could not be more than 7 gal/h on average, both of which are above our estimates of evaporative water use based upon Equation 11. The total water use limits were arrived at by multiplying the 0.22 gal/min minimum resolution of the paddle wheels by the number of minutes that the sump pump ran for RTU-10 and RTU-11.

To distinguish evaporated water from maintenance water, the monthly evaporated water was calculated based upon the product of: a) the on-time of the sump pumps, b) manufacturer-reported condenser-fan air flow rates (pre-cooled section only), c) the average wet-bulb effectiveness of the media based upon laboratory tests for this media, and measured sump water temperatures (described below) , and d) the humidity ratio of saturated outdoor air at wet-bulb temperature minus the humidity ratio of outdoor air. This was accomplished by summing the following product over the course of the month of August 2013 whenever the fans and pump were on:

$$\dot{m}_{evap} = \dot{m}_{air} * WBE * \left(w_{OA,WB-Sat} - w_{OA}\right) \qquad 11$$

Evaporated water can be calculated most accurately in this manner for RTU-7, as its compressors, and therefore condenser fans, run essentially continuously. The mass flow of air used for this calculation was 25% of the manufacturer-reported condenser fan flow, as 50% of the condenser air passes through the evaporative media (due to the V-coil nature of the condensers and a physical divider), and fan-power measurements which indicate that only two of the four condenser fans were operating during this test period. The wet-bulb effectiveness utilized to calculate evaporative water consumption for RTU-7 was 67%, based upon (currently unpublished) laboratory measurements on the evaporative media section of the same retrofit product. **Error! Reference source not found.** indicates an evaporative effectiveness of closer to 90% for RTU-7, however, as noted above, this is based on the sump water temperature, and represents an upper limit on the wet-bulb effectiveness for the air exiting the media.

The evaporation estimates for RTU-8 represent upper limits, as RTU-8 operates almost exclusively with the condenser fans off, which means that the air flow rate across the media is driven only by the wind and free convection, and will be considerably lower than during fan operation. The evaporative water consumption estimate used for RTU-8 was based upon an average wet-bulb effectiveness of 20%, which is once again lower than that observed in **Error! Reference source not found.**, based upon discounting for the fact that the sump-water temperature represents an upper limit on the wet bulb effectiveness. It should be noted that all of the estimated water evaporation rates are higher than the lower limit established by the evaporative cooling provided to the ventilation air (see Figure 29), with the estimated evaporation rates typically being a factor of two higher than the lower limit established by the cooling provided.

The water use determined using these measurements and methods is summarized in Figure 30 for RTUs 7 and 8.

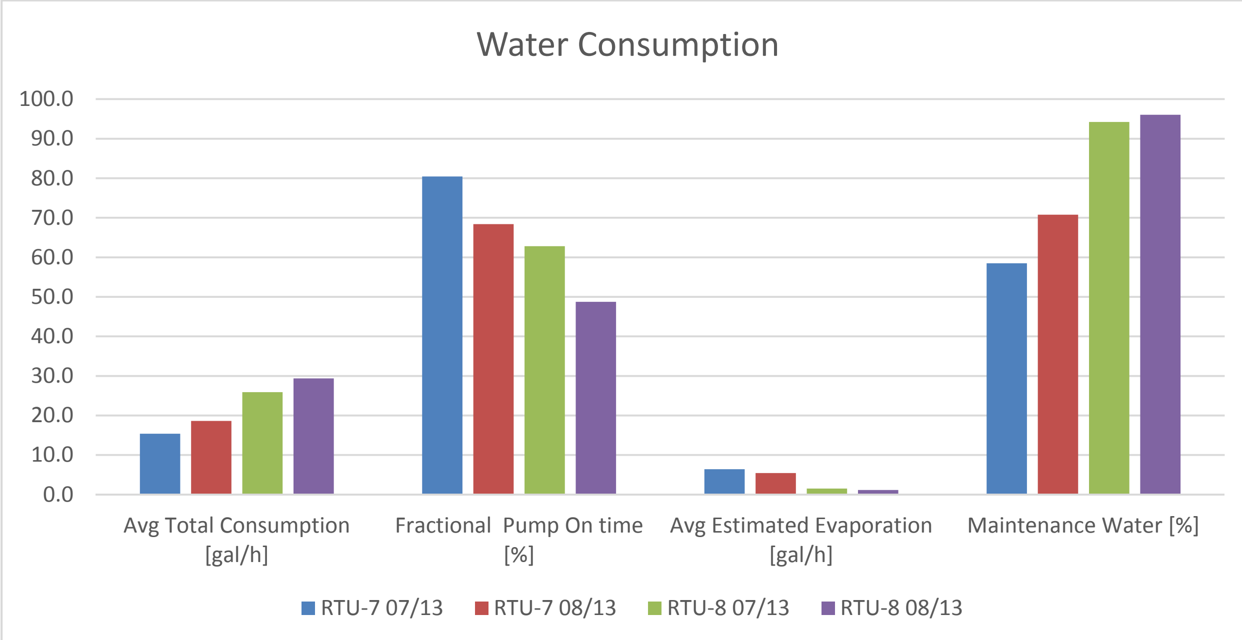


FIGURE 30: WATER USE ESTIMATE FOR THE DUAL-EVAPORATIVE PRE-COOLER ON EACH RTU- EVAPORATIVE WATER USE ESTIMATED BASED UPON 25% OF DESIGN CONDENSER FLOW, 67% WET-BULB EFFECTIVENESS FOR RTU-7 AND 20% WET-BULB EFFECTIVENESS FOR RTU-8 (JULY/AUGUST 2013)

The main takeaway from Figure 30 is that most of the water use for both RTU-7 and RTU-8 was for maintenance purposes rather than evaporation. In the case of RTU-8, virtually all of the water use is for maintenance, yet there does not seem to be any obvious rationale for an intentional variation of this magnitude in the amount of water being used for maintenance (see discussion section below).

SUMMARY OF PERFORMANCE FOR THE DUAL-EVAPORATIVE PRE-COOLER

A summary of the performance of all the retrofits is presented in Table 8. The data presented is based upon comparing measured performance from October 2012 with the retrofit evaporative media removed, with measured performance with the retrofit in place in July and August 2013. The results in Table 8 are based upon minute-by-minute data, aggregated into two outside-air temperature bins, only when the sensible cooling is positive (negative values can occur in ventilation-only mode when fan heat is not overcome by evaporative cooling) , and when particular numbers of compressors are in operation. The results include the number of minute-by-minute data points utilized.

Considering the large diversity in the load seen by each RTU monitored and the resulting differences in operating mode, it is difficult to summarize overall retrofit performance in one table. However, the two key metrics are summarized in Table 8. Although it does not capture the entire picture, the total cooling provided by each RTU is important for putting the reported results in perspective. Consistent with the plots shown above, this first row shows that RTU-7 and RTU-11 do dramatically more cooling than RTU-8 and RTU-10. The retrofit performance is also most clear for these two RTUs, with RTU-7 providing the cleanest results. On the other hand, despite the large variations in operating patterns, the increase in COP, and the increase in sensible cooling capacity are relatively consistent between RTUs 7, 10 and 11. The increase in sensible cooling capacity between 75^{o}F and 85^{o}F ranges between 7% and 11%, while the sensible cooling capacity increase between 85^{o}F and 95^{o}F ranges between10% and 18%. Similarly, Table 8 shows the increase in COP between 75^{o}F and 85^{o}F to be 12% to 20%, and to be 10% to 27% between 85^{o}F and 95^{o}F. The systematic difference between percentage capacity and percentage COP increases is due to the coincident decrease in power draw associated with the retrofit. The 85-95^{o}F savings results were consistently greater than the savings between 75-85^{o}F, which is not surprising considering that the average outdoor-air wet-bulb depression was roughly 8^{o}F larger at the higher outdoor air temperatures.

As illustrated in Table 8, the most robust data set was for RTU-7, and therefore the RTU-7 results are likely the best estimate of retrofit performance (at least for the V condensers on this building). However, even though we have the most confidence in the RTU-7 results, several observations can be made based upon the other RTUs. For example, the 2-compressor savings results for RTU-10 are consistent with the RTU-7 results, as are the 75-85 °F results for RTU-11 (85-95°F results for RTU-11 are based on very few pre-retrofit data points). Also, the largest percentage savings are observed for RTU-8, which is consistent with RTU-8 being the only unit to operate in single-compressor mode. The percentage savings should be highest for single-compressor operation due to the layout of the condenser coils in a V. One side of the V is not submitted to pre-cooling, and single-compressor operation uses only the pre-cooled side of the V. Another possible contributing factor is that ventilation cooling is a larger fraction of sensible system capacity with only one compressor in operation. That said, it must be noted that the results for RTU-8 are based upon limited data points, and therefore have a higher level of uncertainty

TABLE 8: SUMMARY OF SENSIBLE CAPACITY AND COP PERFORMANCE OF ALL RETROFITTED RTUS (JULY/AUG 2013 VERSUS OCT 2012) NOTE HIGH UNCERTAINTY ASSOCIATED WITH RESULTS BASED UPON VERY FEW DATA POINTS

RTU#	RTU 7 2-compressor	RTU 8 1-compressor	RTU 10 2-compressor	RTU 11 2-compressor	RTU 11 3-compressor
Total Cooling in all modes (Aug 2013) [KBtu]	38,700	10,700	12,000	49,000	
Change in Sensible Capacity at Outside Air Temperature of 75-85°F [pre/post data points]	8% [7116/28850]	53% [22/66]	10% [182/2993]	7% [756/8047]	11% [624/8162]
Change in Sensible Capacity at Outside Air Temperature of 85-95°F [pre/post data points]	18% [2393/24360]	56% [87/1093]	17% [554/11575]	10% [117/6417]	14% [14/10865]
Change in COP at Outside Air Temperature of 75-85°F [pre/post data points]	14% [7116/28850]	72% [22/66]	18% [182/2993]	12% [756/8047]	20% [624/8162]
Change in COP at Outside Air Temperature of 85-95°F [pre/post data points]	27% [2393/24360]	69% [87/1093]	23% [554/11575]	10% [117/6417]	20% [14/10865]

DISCUSSION

Based upon the data collected in this project, there are several key observations concerning the tested retrofit. First, the retrofit can produce significant energy savings and sensible cooling capacity improvements. The savings appears to be higher at higher outdoor air temperatures, which also correspond to larger wet-bulb depressions for the outdoor air, which also correspond to peak electricity demand conditions. The second key observation is that the variability of the percentage improvement in RTU COP (10-27%) was not the only key determinant of absolute saving. The other key determinants of absolute savings were the pre-retrofit energy consumption of the RTU, and the mode of operation of that RTU before and after retrofit. Starting with the most obvious, retrofitting an RTU whose compressors never run clearly impacts absolute savings, although the improvement in COP is actually highest for a unit that is only providing ventilation, as the only additional energy use required to provide the ventilation cooling is the pump power (see below). Finally, it is worth noting that this building had RTUs with a relatively uncommon configuration, namely V coils, which meant that the percentage savings would be reduced whenever cooling was being provided by a compressor whose condenser was not seeing pre-cooled air.

On a more basic level, it is important to note that the change in coefficient of performance for each particular operating mode does not necessarily present a complete picture of the energy savings provided by this measure. One reason is that such an analysis does not account for the fact that the increased cooling capacity provided by the retrofit should impact the amount of time that the RTU operates in each operating mode, in principle decreasing the amount of time that it operates with multiple compressors running. Stated another way, when the capacity increase changes the amount of time spent in each mode of operation, the change in COP in a particular mode does not capture the extent to which a retrofitted system will operate in a lower capacity and ostensibly more efficient mode of operation. Last, but not least, since the dual-evaporative pre-cooler adds cooling capacity during periods when compressors are OFF (i.e., ventilation-only cooling), the COP improvement while the compressors are ON does not capture the savings provided by operation in ventilation-only cooling mode while compressors are off.

ABSOLUTE ENERGY SAVINGS

Using only COP improvements for each compressor stage, the absolute savings achieved by RTU-7, RTU-10 and RTU-11 ranged from 700 – 1,600 kWh over the month of August 2013. These values were calculated using a time weighted (time in each compressor stage) COP improvement based upon the numbers in Table 8. As discussed above, this approach underestimates the savings achieved because it does not account for the fact that the retrofit would be expected to shift the distribution of system operating modes to higher-efficiency, lower-capacity compressor stages. In addition, this absolute savings does not include the savings produced by pre-cooling of ventilation air when the compressors are off. This is especially pertinent for RTU-8, which almost never operates in any compressor mode, but which generates a substantial amount of indirect-evaporative cooling for ventilation air.

Making a comparison of the pump energy used to provide ventilation-air cooling, to the compressor energy that would have been needed to generate the same amount of cooling using the average COP of these RTUs, indicates a 65-78% electricity savings associated the cooling provided in this ventilation-only operating mode. For RTU-8, applying the 78% savings value computed in September to the estimated 8,100 kBtu of ventilation-only cooling provided by RTU-8 in August, the absolute energy savings for RTU-8 would be 530 kWh, which is roughly one third the energy savings due to COP improvement for RTU-7 in August.

WATER CONSUMPTION

Maintenance water use for the dual-evaporative pre-cooler was controlled by a valve that is set by the installing technician. The particular valve and process employed seemed to result in two problems: a) a large variability in maintenance water use between RTUs, and b) inconsistent maintenance water use for the same RTU. The first problem is easily identified by the monitored water consumption and evaporation analysis described above. Specifically RTU-8 appears to be employing roughly 95% of its water use for maintenance purposes, whereas for RTU-7, maintenance water appears to represent 60-70% of its water use. On the other hand, a water evaporation analysis for RTU-11 suggests that it is not using any water for maintenance purposes (estimated water evaporation is almost exactly equal to the minimum detectable total consumption). For RTU-10, there is a gap between the estimated evaporation and the minimum detectable flow, which means that it may or may not have consumed some water for maintenance. Based upon the minimum detectable water flow, the maintenance water use for RTU-10 could not be more than 40% of its total water consumption. Our observations about inconsistent maintenance water use for the same RTU were confirmed by another pilot evaluation for the same technology in Northern CA. For one unit at that site, elevated water consumption was identified from monitored data, however, by the time the research team visited the site to investigate the reason for excess water use, bleed water for the system had declined to zero because the filter upstream of the controlling valve had clogged. Our recommendation based upon these observations is that the manufacturer should utilize a better system for regulating maintenance water. That said, it is also worth noting that we did not find any evidence of water coming into contact with the condenser coil on any of the retrofitted RTUs, and therefore did not observe any coil damage. Neither did we observe any scale formation or physical degradation of the evaporative media, nor any corrosion or failure in either the retrofit system or the existing RTU. We did not find any algae or apparent biological growth on the media, with the only biological growth being observed in a sump that remained full of water during the winter.

Based upon general industry practices, and visual observations of all four retrofitted RTUs, the level of water use for maintenance for RTU-7 and RTU-8 was dramatically more than what was necessary. Both RTU-11 and RTU-10 did not show evidence of consequential scaling over an entire year of operation, despite having essentially zero maintenance water use for RTU-11, and maintenance water use of less than 67% of evaporated water for RTU-10. By comparison, the maintenance water use for RTU-7 was 190% of evaporated water, and was 2400% of evaporated water for RTU-8.

In terms of understanding the overall impact of the retrofit, it seems appropriate to compare the cost of water consumption with the electricity savings provided. Once again we will use RTU-7 for this analysis. To do this, we calculate water cost three different ways: a) using total water consumed, b) using evaporated water only, and c) using a reasonable maintenance water strategy (15% of evaporated water). Using the month of August 2013, the total water consumption for RTU-7 was 13,800 gallons, the estimated evaporated water was 4,800 gallons, and with 15% maintenance water use, the total would be 5500 gallons. At a marginal cost of $2.50/hundred cubic feet, the water cost would be: $46, $16 or $18, respectively. Using the average COP of RTU-7, the estimated total electricity savings of 1620 kWh in August 2013, and an electricity cost of $0.20/kWh, the electricity cost savings would be $325. Thus, using the best estimate for water use (5500 gallons), roughly 6% of the energy savings goes towards paying for water. Looked at another way, the evaporated water use was 2.9 gal/kWh saved, which with 15% bleed would become 3.4 gal/kWh, which should be compared with the average1.34 gal/kWh water consumed for generating electricity in California.

SEQUENCE OF OPERATIONS AND CONTROLS

One of the key observations associated with this field test was that there was a large difference in the savings obtained from each RTU. This variability seemed to stem mainly from how a particular RTU was being utilized prior to retrofit, and how the retrofit control interacts with the operation of that unit. As noted above, for at least one retrofitted RTU (RTU-8), the compressors hardly ever operated, which means that the absolute savings associated with that RTU was limited by not making use of the condenser-air precooling feature of the retrofit. This could have been avoided by either choosing a different RTU to retrofit (based upon pre-retrofit data), or by employing some sort of integrated control of the different RTUs and retrofit.

There are several potential advantages of integrated control, one being reduced cost from replicating sensors and systems, although the current controls for this retrofit are very inexpensive. Another is that integrated control can eliminate conflicts. Yet another is a commissioning advantage, as the integration procedure might force the installer to determine how the existing installation is working prior to enacting the retrofit.

In terms of sequence of operations, it should also be noted that the wet-bulb effectiveness for indirect evaporative cooling of ventilation air is strongly impacted by operation of the condenser fans. If the condenser fans are not operating, but the water pump is operating, there is some evaporation from the wetted media, which provides some indirect evaporative cooling of the ventilation air (see and Figure 29). However **Error! Reference source not found.** clearly shows that the effectiveness of cooling water by evaporation without active flow across the media is significantly reduced. The maximum potential wet-bulb effectiveness for RTU-8 is highly variable and averages about 35%, as compared to a more consistent maximum potential wet-bulb effectiveness of about 90% for RTU-7.

Although there may be some value associated with condenser fan operation during periods when the compressors are not active (i.e. ventilation mode) so as to increase evaporative cooling capacity, that option was not tested in this project. Prior to implementing such a controls change, an analysis of pumping power, fan power, and ventilation cooling would need to be performed. This strategy and analysis should also include periods when the system operates in economizer mode. This is especially important for units that that spend a significant amount of time in ventilation only, where zones don't call for much compressor cooling. RTU-8 spent virtually all 363 hours of its pump operation in this low-wet-bulb effectiveness mode during August 2013, and RTU-10 spent 206 hours out of 540 pump hours in this mode in August 2013.

Further, as the water circulation pump is controlled separately from all other systems in the rooftop unit, there are even some periods when the pump operates while the rest of the rooftop unit is OFF. There is no benefit to this mode of operation. For RTU-8 and RTU-10 the pump operates while the rest of the unit is off for nearly 100 hours in August 2013, which for a 120W pump amounts to 12 kWh of energy consumed during the month. More importantly, since the water bleed rate (maintenance water use) is connected directly to pump operation, the equipment continues to consume water during these periods for no reason. There are different options to avoid inappropriate pumping scenarios. The pump could be installed on an interlock with the indoor fan, the outdoor fan, or the compressors. However, interlocked with the compressors, the retrofit wouldn't provide useful capacity in an indirect-evaporative-only mode (pump+ventilation) because it would only operate when the compressors operate.

CONCLUSIONS

There are four key conclusions that can be drawn from this field study about the dual-evaporative pre-cooling system tested:

a) considerable energy savings, capacity improvement, and peak demand reduction can be achieved with the technology,

b) the water evaporated to achieve these savings is of the same magnitude as the water used to generate the saved kWh,

c) the actual savings realized depends on the usage and operating mode of the RTU to which the system is applied, and

d) the savings potential for this technology might benefit from improved controls and integration strategies.

Moreover the results of this field test suggest that the building owner could disconnect one or more RTUS, due to the increase in cooling capacity of other units, thereby reducing their connected electrical load.

There are some additional conclusions to be drawn from this investigation. One is the fact that the water consumption for evaporation was relatively modest for the retrofit, corresponding to a cost penalty of roughly 5% of the electricity savings for the unit with the best pre/post retrofit data (RTU-7) (at $2.50/hundred-cubic-feet and $0.20/kWh). That said, maintenance water use was not adequately controlled for the monitored RTUs, such that it dramatically exceeded water evaporation in some cases. Using RTU-7 as our example once again, at the observed excessive maintenance water use rate, the estimated water cost would increase to 14% of the electricity savings, whereas using appropriate maintenance water would result in a total water cost of 6% of electricity savings. The conclusion here is that maintenance water use is an important factor that needs to be set up properly at the outset, and maintained on a regular basis (e.g. at the same time as the media), or perhaps be controlled by a reliable automated system.

RECOMMENDATIONS

Based upon the results of this field test, the authors have developed several recommendations. First, whenever there is more than one RTU serving a space, pre-retrofit observations should consider the typical runtime behavior of each RTU and ensure that the retrofit is only applied where it will have a beneficial effect. Ideally this decision should be part of an overall savings potential analysis (e.g. investigating the status of RTU operation including economizer and compressor use). This analysis could also result in some RTUs being turned off completely (or even disconnected from the grid). If some RTUs are hardly operating, then they will become even more superfluous when the extra capacity associated with the retrofit is added into the network of RTUs.

We would also recommend testing, or possibly simply analyzing, different control strategies and sequences of operations. One such strategy would be the following:

1. run the sump pump only if the indoor blower is running (could be interlocked with indoor blower)
2. whenever the pump is running, turn on the condenser fan if it is not already on

The first part of this strategy would not require any interaction with the RTU controls, but the second part would require getting into the power circuit for the condenser fan.

Finally, based upon a very large observed variability in the use of water for maintenance purposes, the authors also recommend that the manufacturer develop more robust hardware and complementary installation procedures to manage maintenance water use.

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