PG&E's Emerging Technologies Program

HYBRID ROOFTOP AIR CONDITIONERS FOR A MALL, FAIRFIELD CA

Pacific Gas & Electric Company

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LEGAL NOTICE

ABBREVIATIONS AND ACRONYMS

AC	Alternating current
BTU	British thermal units
C	Concentration
CFM	Cubic feet per minute
COP	Coefficient of performance
c_p	Specific heat
CSV	Comma separated value (text file)
db	Dry-bulb
DC	Direct current
$\dot{E}$	Rate change of energy
FS	Full-scale
GPM	Gallons per minute
$\dot{H}$	Rate change of enthalpy
h	Specific enthalpy
Hr	Hour
$\dot{m}$	Mass flow rate
OA	Outside air
OSAF	Outside air fraction
P	Power
PA	Product air
RA	Return air
RH	Relative humidity
RTU	Roof top unit
SA	Supply air

T	Temperature
$\dot{V}$	Volume flow rate
VFD	Variable frequency drive
wb	Wet-bulb
WBE	Wet-bulb effectiveness
wc	Water coil

FIGURES

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EXECUTIVE SUMMARY

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 and 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.

An inordinate amount of energy is wasted each year as a result of the fact that packaged commercial air conditioners are designed as 'one-size fits-all' products that can function in any climate and are not designed to achieve maximum efficiency for a specific climate. Climate appropriate solutions recognize unique opportunities for efficiency that arise from the climatic patterns and characteristics in particular regions. In California, these solutions include technologies that use water evaporation strategically to achieve substantial gains in cooling efficiency

The research reported herein directly supports California Energy Efficiency Strategic Plan goals to accelerate marketplace penetration of climate appropriate air conditioning technologies. The report presents results from a field assessment of two Trane Voyager RTUs that have been retrofitted with an Integrated Comfort Dual-Cool. The Dual-Cool retrofit uses direct evaporative cooling to precool the condenser air and indirect evaporative cooling to cool the ventilation air. Two of these hybrid systems were installed on a mall in Fairfield, California. One of the units serves tenant spaces and the other serves the common areas in the mall. When compared to manufacturer data for a similar RTU the field data showed an improvement in peak capacity by as much as 48% and an improvement in average capacity by as much as 45%. The field data showed an improved overall efficiency in all modes of operation and a decrease in power consumption by as much as 19%. However, some modes of operation showed an increased power draw of as much as 10%.

PROJECT GOALS

The main objectives of this project were to:

1. Provide reliable energy and water performance data for the technology in real world operation
2. Facilitate, review, and document application and operation of the technology.
3. Develop practical design guidance and recommendations based on in-field experience with the technology.
4. Assess the whole building energy savings achieved by the project

TECHNOLOGY REVIEW

The Trane Voyager Dual-Cool is a hybrid rooftop air conditioner that uses dual-evaporative pre-cooling. The technology was expected to save energy two ways:

1. By cooling the outdoor air being delivered to the RTU evaporator coil by indirect evaporative cooling, thereby reducing how much cooling the vapor compression system must perform
2. By reducing the air temperature seen by the RTU condenser coil by evaporative pre-cooling, thereby decreasing refrigerant pressure and the work that needs to be done by the compressor.

APPROACH

The approach chosen for this evaluation was a field test conducted on two packaged rooftop units (RTUs) installed in Fairfield, California. One of these RTUs serves tenant spaces and the second unit serves the common areas of the mall. Monitoring of the installed RTUs began in March 2013. Monitoring involved minute-by-minute data collection on the RTUs. The key metrics used to characterize the performance of the retrofit include:

a) Coefficient of Performance (COP)
b) Cooling capacity
c) Electric power draw

In addition to the key performance metrics, the internal workings the Dual-Cool were also investigated, including isolating the performance of the water-to-air heat exchanger for indirect evaporative cooling of ventilation air, and monitoring the performance of the sump-pump control.

Recommendations

The research team strongly recommends that the technology be adopted by utility energy efficiency programs. Although the field monitoring revealed many commissioning and control issues that negatively impacted system performance, the technology demonstrated its ability to significantly improve capacity and efficiency, especially during peak hours of operation. While the benefits of the dual evaporative pre-cooler are clear, the results of this field demonstration show that to take full advantage of the improvements in capacity and efficiency, further research and development should be focused on developing a controller that optimizes the performance of both the RTU and the dual-evaporative pre-cooler.

INTRODUCTION

This report documents a field evaluation of the Trane Voyager with a Dual-Cool retrofit installed at a mall in Fairfield, CA. The technology explored in this evaluation has previously demonstrated 40% on-peak demand savings for cooling in California climate conditions (Wooley 2014). The study directly supports PG&E Emerging Technologies efforts to advance the California Energy Efficiency Strategic Plan goals related to broad marketplace application of climate-appropriate HVAC technologies.

Buildings consume 70% of the electricity in the US, 50% of which is used for commercial buildings. Cooling and ventilation account for more than 25% of the annual electricity consumption for commercial buildings in California. If natural gas use is also considered, heating, cooling and ventilation typically account for more than 35% of the annual primary energy footprint for a commercial building (EIA 2012). Furthermore, HVAC accounts for more than 30% of the greenhouse gas emissions associated with commercial buildings in California, amounting to statewide emission of more than 23 MMT CO_2e (CEC 2006, CARB 2014). Efficiency for these systems must improve in order to reach strategic energy and environmental goals, and state policy in response to climate change.

Moreover, air conditioning is the largest single contributor to peak electrical demand. Rooftop units are usually the largest single connected load in a commercial building, and can account for more than 50% of the on-peak demand from commercial facilities. California's electric grid is especially stressed during summer periods when generation requirements can be twice as high as other seasons. On the hottest summer days, air conditioning alone accounts for more than 30% of the peak demand on the statewide electric network (EIA 2014, CEC 2006). Grid management is anticipated to become more challenging as a larger number of intermittent renewable generators are brought on to the network. Since air conditioning loads are such a singularly large fraction of statewide demand, these systems will play a key role on the newly emerging paradigm of dynamic grid management.

Packaged rooftop air conditioners (RTUs) are predominately responsible for heating and cooling in commercial buildings. This equipment is often the single largest connected load in a building, and utilizes technology that has not evolved to keep pace with efficiency improvements for other key end use sectors. There are a variety of emerging technologies that can improve the efficiency of rooftop air conditioners. This study evaluates one option that is designed especially for cooling in California, and other hot-dry climates, where it promises to reduce peak electrical demand for air conditioning by 40%.

The intent of this project was to characterize the cooling performance and energy efficiency for two hybrid rooftop packaged air conditioners installed on commercial buildings in Fairfield, California. The hybrid systems are standard 'high efficiency' model rooftop air conditioners that have been retrofitted with a dual evaporative pre-cooling technology to cool air at the condenser inlet and at the ventilation air inlet. This pre-cooling process is executed in such a way that moisture is not added to the conditioned environment. Further, the retrofit can be applied directly to almost any conventional rooftop air conditioner with only minor revisions. We consider this a key advantage in comparison to other climate appropriate commercial cooling measures.

This study directly supports PG&E Emerging Technologies efforts to advance the California Energy Efficiency Strategic Plan goals related to broad marketplace adoption of climate-appropriate HVAC technologies (CPUC 2011). Climate appropriate measures leverage technology that may not be appropriate for all climates, but which have unique potential to decrease energy use and peak demand in specific scenarios. In the case of the technology studied here, water evaporation is used to decrease load and improve the efficiency of a conventional vapor compression air conditioner.

Traditionally, conventional vapor compression air conditioners have been designed to comply with federal minimum energy conservation standards, which require cooling performance at a single condition that is milder and more humid than design conditions experienced in California climates. However, power draw and cooling capacity for air conditioners are especially sensitive to outside air temperature. As a result, systems in California deliver less cooling and draw more power than what is predicted by standard ratings.

There are many new energy efficiency measures for rooftop air conditioners, including variable speed fans, advanced control schemes, and multi-stage or variable-capacity compressors. Recent studies have indicated that these measures can reduce annual electric energy consumption for HVAC by more than 50%. The greatest share of these savings is generally captured from reduced fan power at part capacity operation, and from appropriate economizer control. These measures are promising, however since the greatest portion of savings are captured at part load, these measures provide relatively little improvement at peak when all components must operate at full capacity and when air conditioning stands as the largest single end-use on a resource-stressed grid. Additionally, most of these capabilities are now required by California Building Energy Efficiency Standards, and as a result utility efficiency programs will find it difficult to utilize such technologies to advance savings beyond a standard baseline.

The climate appropriate technologies evaluated in this study and through other recent research improve the efficiency for cooling during all non-economizer hours, and offer substantial demand savings at peak. The dual-evaporative pre-cooling technology observed in the field for this study was also laboratory tested by UC Davis in 2012 (Woolley 2012). Measured performance in that evaluation indicated 43% demand reduction at an outside temperature of 105°F (73°F wet-bulb). This degree of peak demand reduction met the UC Davis Western Cooling Challenge performance targets, which consider a system's sensible cooling performance while providing 120 cfm of ventilation air per nominal ton of cooling equipment. In fact, the dual evaporative pre-cooling concept was used as the basis to develop the Challenge performance targets, and is recognized as a central opportunity for improving efficiency for rooftop air conditioners in California.

A variety of studies have suggested that evaporative cooling technologies can be an effective method to reduce electricity consumption and peak power demand from conventional vapor compression systems, however characteristic performance data from field testing is limited. For example, the measured performance of each component in the dual evaporative pre-cooler has not been well documented, and there is currently not adequate validation of the overall equipment performance characteristics across a comprehensive range of operating conditions. As a result, previous field data is not adequate to inform sophisticated modeling efforts that can estimate statewide energy savings and demand reduction potential. The results of this study expand our understanding of these characteristics and should provide a basis for further efforts to advance the measure. Despite the demonstrated energy efficiency advantages of the technology, we identify several technical opportunities to improve system control and equipment design in order to address some of the challenges that were observed in the field.

Furthermore, this study afforded an opportunity to observe real world equipment behavior and the impact of control sequences and interactive effects with other building energy systems. Similarly, we observed some challenges related to equipment application, commissioning, operation and maintenance. Many of the challenges observed are not unique to the dual-evaporative pre-cooling technology, and are common problems associated with conventional rooftop air conditioners, and building systems. Regardless, these challenges do constrain successful application of the measure; we offer a number of recommendations and identify some of the real world constraints for the product.

Project Objectives

The main objectives of this project were to:

1. Provide reliable energy and water performance data for the technology in real world operation
2. Facilitate, review, and document application and operation of the technology.
3. Develop practical design guidance and recommendations based on in field experience with the technology.
4. Utilize available data to capture information about whole building energy savings achieved.

To accomplish these objectives, the research team collaborated closely with the manufacturer of both the RTUs and the dual evaporative pre-cooler retrofit as well as the facilities manager of the mall.

Project Overview

Climate Appropriate Cooling

California's *Long Term Energy Efficiency Strategic Plan* outlines four major programmatic initiatives, as *"Big Bold Energy Efficiency Strategies*" to facilitate broad energy savings for our built environment:

- All new residential construction will be zero net energy by 2020
- All new commercial construction will be zero net energy by 2030
- HVAC will be transformed to ensure that its energy performance is optimal for California's climate
- All low-income customers will have the opportunity to participate in energy efficiency programs by 2020

The third initiative targets a 50 percent efficiency improvement for HVAC by 2020, and a 75 percent improvement by 2030. The plan recognizes that cooling and ventilation is the single largest contributor to peak electrical demand in California, which results in "*enormous and costly impacts on generation, transmission, and distribution resources as well as a concurrent lowering of utility load factors.*" Strategic goals to transform the HVAC industry focus on:

1. Code compliance
2. Quality installation and maintenance
3. Whole-building integrated design practices, and
4. Development and accelerated implementation of new climate-appropriate equipment and controls

The efficiency measure studied in this project specifically targets the fourth goal: it advances the evaluation and application of climate appropriate systems and controls. Air conditioning equipment has traditionally been designed and rated according to a single number efficiency metric that does not accurately represent the performance of air conditioners in California climates. Optimizing for this metric, manufacturers have mainly sold a single type of air conditioner that functions reliably in any climate, but is also inefficient in most climates. Luckily, there are many climate appropriate technologies and system design strategies that use far less energy than the "one-size fits all" approach. Climate appropriate air conditioning systems and controls are designed and tuned specifically for local climate conditions, and occupant comfort needs; they provide an equal (or better) quality of service with less energy input. Some examples of cooling strategies appropriate for California climates include:

- Sensible-only cooling measures that do not waste energy on unnecessary dehumidification
- Indirect evaporative cooling (and other evaporative measures), when water is used efficiently
- Advanced economizer controls, natural ventilation cooling, nighttime ventilation pre-cooling, and other passive or semi-active systems that capitalize on large diurnal outdoor temperature swings to reduce the amount of active cooling required at other periods.
- Adaptive comfort controls, and predictive control strategies that conserve energy by allowing indoor conditions to drift across a wider range, in concert with dynamic human comfort considerations.
- Any technology that uses substantially less energy for cooling (especially at peak) than the industry standard "*one size fits all*" minimum efficiency equipment.

Climate appropriate cooling technologies have reliably demonstrated peak demand reduction of more than 40% (Woolley 2013). Some solutions have shown annual cooling energy savings beyond 65% (Harrington 2015).

One should also note that current single number industry standard rating methods are generally not appropriate for describing performance of climate appropriate technologies. The problem is not that the limited range of standard test conditions are not exactly representative of every application in California; the issue is that the standard methods of test can actually portray climate optimized products as less efficient than traditional air conditioners. These standards unintentionally disadvantage climate appropriate strategies by misrepresenting their performance in comparison to the status quo. In many circumstances climate appropriate strategies cannot even be tested by industry standard methods because they operate in configurations that are fundamentally different than the scenario for which current standards were designed. This shortcoming is especially true for whole building integrated design practices.

The project reported here builds on a body of research, evaluation, and pilot demonstrations recently advanced by PG&E and other California entities to advance the understanding and market introduction of climate appropriate HVAC solutions. The findings from this project should guide the development and implementation of programs and policies designed to accelerate the broad and successful uptake of these solutions for new and existing buildings.

TECHNOLOGY OVERVIEW

OVERVIEW OF THE DUAL EVAPORATIVE PRE-COOLING TECHNOLOGY

The product tested in this project takes advantage of indirect evaporative cooling to cool the ventilation air stream on a conventional rooftop unit, and uses direct evaporative cooling to cool air at the condenser inlet. The system cools ventilation air sensibly; therefore it does not add moisture to the conditioned space. The combined system maintains latent cooling capacity for applications where dehumidification is required.

The dual evaporative pre-cooler adds a 12" deep direct-evaporative media at the inlet face of the condenser coil, a stainless steel sump with a pump and a water to air heat exchanger at the ventilation air inlet on a rooftop unit. The water for evaporation is supplied at the top of the media; as the water flows down the media some of it evaporates into the condenser air, cooling both the condenser air and the remaining water which drains into the sump. Before being recirculated to the top of the evaporative media, water in the sump is pumped through the water-to-air heat exchanger located at the ventilation air inlet. In the water-to-air heat exchanger heat is transferred from the ventilation air to the water pumped from the sump, resulting in a cooler mixed air temperature at the evaporator coil inlet. Figure 1 provides a schematic illustration of the technology.

These dual processes work together to increase cooling capacity and to improve efficiency of the vapor compression system. The capacity increase is partly a result of reduced condenser air temperature, and partly a result of the sensible cooling delivered by the water-to-air heat exchanger. The improved efficiency is mainly caused by the reduced condenser air temperature. Manufacturer performance tables, laboratory measurements, and most field observations show an efficiency increase of 1-2% for every 1°F by which the air at the condenser coil inlet is decreased. The dual evaporative pre-cooler takes advantage of this fact by reducing condenser temperature with an evaporative cooling process that requires only 150 W pumping power. Laboratory measurements for the dual evaporative pre-cooling technology installed on a similar rooftop air conditioner indicated 43% reduction in power draw at peak (Woolley 2012).

The product tested in this assessment is designed to be added to new or existing conventional rooftop units, and only requires modest integration efforts. The system uses a simple stand-alone control scheme that does not require integration with or revision to existing rooftop unit controls. It also uses relatively few materials and standardly available components, which helps to keep equipment costs low compared to other climate appropriate strategies.

Our review of various projects that have applied this technology indicate that installed cost of the dual evaporative pre-cooler can be between $350 – 450 *per nominal ton*; so it could cost $7,000-9,000 to add this technology to an existing 20 *ton* rooftop unit. Actual installed costs appear to depend on the number of retrofits, equipment size, and the ease of installation. Access to a water supply and a sewer drain are necessary. The addition of rooftop penetrations and plumbing interconnections can increase costs. Many of the component costs, construction, and installation costs are not sensitive to equipment size, so the retrofit of smaller units will be comparably more expensive per nominal ton than retrofit of larger equipment. It is also incrementally less expensive to retrofit several units in a single project. This strategy already has a reasonable first cost; however we believe the technology could become even more affordable were it manufactured at scale.

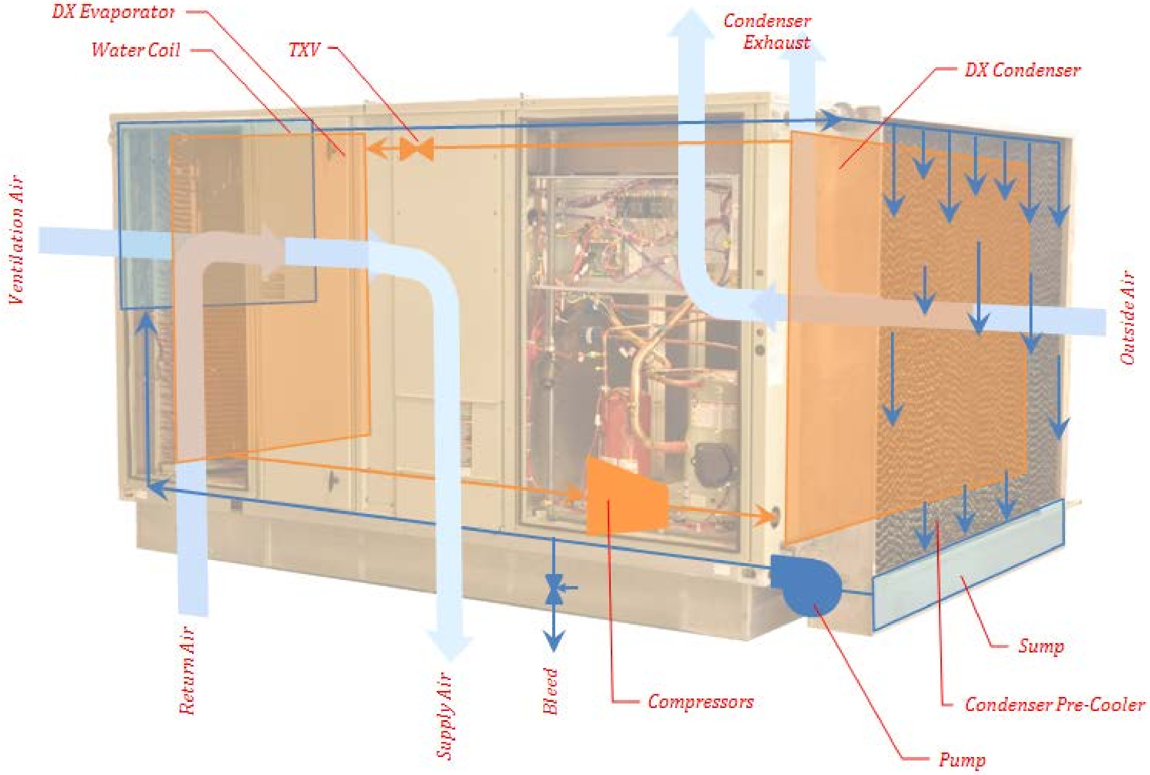

FIGURE 1 – CONCEPTUAL SCHEMATIC FOR ROOFTOP UNIT WITH DUAL EVAPORATIVE PRE-COOLING

There are some technical factors related to this equipment that designers and practitioners should consider.

First, the water-to-air heat exchanger adds some airflow resistance in the ventilation flow path. During normal operation, this resistance takes the place of resistance normally provided by the outside air damper. Therefore, the system does not increase fan power and does not decrease supply airflow during normal operation. Since the coil adds restriction in the ventilation flow path, installation of the technology does require the damper settings to be re-balanced to maintain an appropriate ventilation air flow rate.

Second, the savings achieved by this measure are tied closely to the amount of outside air that is conditioned by the system. When it is an option, it is advantageous to group the ventilation needs for a building onto units that utilize this technology, and to shift other standard units to operate as recirculation air only. This strategy is especially appropriate for big box retail stores and other buildings where a displacement ventilation strategy can maintain appropriate air change rates.

Finally, addition of the dual evaporative pre-cooler does add airflow resistance to the system in economizer mode. Subsequently, the supply airflow rate will be reduced in economizer modes unless the blower speed is adjusted to overcome the added resistance. Special consideration should be given to operation in this mode to ensure that the blower is capable of functioning reliably with the added resistance, and that the supply airflow rate is adequate for operation of each compressor stage in an integrated economizer mode. Since the dual evaporative pre-cooler cools the incoming ventilation air this measure can also extend the range of outside air temperatures that are appropriate for economizer operation with 100% outside air.

OPERATING MODES & SEQUENCE OF OPERATIONS

Controls for the dual evaporative pre-cooler typically consist of a single outside air temperature switch that enables pump operation anytime the measured temperature is greater than a field-selected set point. The appropriate set point may vary a little by application, but the manufacturer typically recommends 70°F as the changeover point. Below 70°F, the cooling effect for ventilation air is small, and there is less efficiency benefit from condenser-air cooling. Furthermore, it has been observed that if the pump is allowed to circulate at temperatures well below 70°F, the media may not dry out adequately each day, which can allow biological growth on the media. The optimal changeover point would be that temperature below which the compressor and fan energy savings benefits are smaller than the energy expended for pump operation.

The hybrid rooftop unit that utilizes the dual evaporative pre-cooler evaluated in this study can operate in many different modes. The pump for the dual evaporative pre-cooler is controlled to operate anytime outside air temperature is above a field-selected set-point. The rooftop unit is controlled to operate in an economizer mode anytime the outside air temperature is below an independent field-selected set-point. Simultaneously, the supply blower speed and compressor stages respond to programmed ventilation requirements and staged cooling signals from a room thermostat, or building Energy Management and Control System (EMCS). Controls that manage the dual evaporative pre-cooler are completely separate from controls that manage the rest of the rooftop unit functions; therefore the pump can operate in combination with any normal rooftop unit operating mode.

It should be noted that while the outside air temperature switch for the dual evaporative pre-cooler control provides for a simple retrofit, and helps to keep equipment costs relatively low, it also results in the possibility of some unanticipated operating modes. For example, the pump may operate while the rooftop unit is off, or during periods when the outside air damper is not open. Other studies have indicated that for some units, these operating modes may constitute a substantial number of operating hours (Modera 2014).

When building controls differentiate between occupied and unoccupied states, there may be periods when cooling is needed but when ventilation is not needed. These periods would likely occur when outside temperature is cool enough that the dual evaporative pre-cooler does not operate – one example is a pre-occupancy cool-down period. However, some of these hours could occur when it is warm enough for the dual evaporative pre-cooler to operate – one example is when cooling is needed in the late afternoon or weekend while a commercial space is vacant. It is not clear whether pump operation with the outside air damper closed would be of benefit or not. The system would provide no value in cooling ventilation air, but condenser air pre-cooling would still improve vapor compression efficiency.

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Overview of Field Test Site

The field test site selected for this study was a mall in Fairfield, CA. Two RTUs were installed on the mall, one that serves the common areas in the mall and one that serves a tenant space. The two units are controlled by separate and independent thermostats. The units replaced were nearing end of life, as such they were not monitored to establish a performance baseline for comparison. The performance of the equipment studied here is compared to the performance of a modern standard-efficiency air conditioner operating at similar conditions.

Table 1 summarizes the general design specifications for the equipment installed at each location. Measured supply airflow rates and ventilation airflow rates were significantly different than the design targets documented in Table 1.

Table 1 – As designed equipment schedule for rooftop units (before dual evaporative pre-cooler)

	Manufacturer	*Model*	*Total Cooling (kbtu/h)*	*Nominal Flow Rate (CFM)*	*Compressors*	*V/ø/hz/*
TAC2	Trane	TCD 420 B40 K6B3FC5A00D00GH0K000000	420	13,000	2 SCROLL	460/3/60
MAC9	Trane	YCD 330 B4L K6B2DC1A00D00GH0K000P00	330	10,000	2 SCROLL	460/3/60

The units installed have the following options:

TAC2

- Cooling only
- 35 tons of cooling with 410A (w/o dual pre-cooler)
- Microchannel condenser coil
- Downflow supply and return with 100% powered exhaust and VFD with bypass
- 15 HP supply fan motor
- 0-100% economizer with dry bulb control

MAC9

- Packaged cooling and natural gas heat (low heat)
- 27.5 tons of cooling with 410A (w/o dual pre-cooler)
- Microchannel condenser coil
- Downflow supply and return with 100% powered exhaust and VFD with bypass
- 10 HP supply fan motor
- 0-100% economizer with dry bulb control

TECHNICAL APPROACH & TEST METHODOLOGY

FIELD TESTING OF TECHNOLOGY

The primary objective of this investigation was to conduct a field evaluation of a new hybrid rooftop air conditioner that utilizes a dual evaporative pre-cooler. The evaluation studied real world equipment operation and developed accurate characterizations of the overall system performance and energy efficiency for two separate units in each mode of operation and across a range of operating conditions.

The study was designed to investigate performance characteristics that cannot be captured by steady state laboratory testing. For example, this evaluation carefully disaggregates performance in each mode of operation to consider the value and effectiveness of each system state, and investigates the implications of the control strategies and field-selected settings that were applied. There are a variety of field conditions that can impact equipment performance; this study documents our observation of the hybrid system in two different applications in order develop a broader understanding about the technology. Further, the study captures several specific metrics that can be used to model performance of the dual-evaporative pre-cooler.

This study also presents a comparison of the field data against manufacturer data for a similar RTU without any retrofits. Although it would have been ideal to use performance curves for the stock Trane Voyager, the manufacturer provided data was insufficient to fully describe its performance. The RTU selected for the comparison is the Lennox TCA240S which is a two stage unit with a rated cooling capacity of 25 tons and a rated COP of 3.45. The performance curves are easily scaled to match the capacity of the MAC9 and TAC2 units.

Beyond the technical assessment objectives, the study also documents observations related to water use, equipment reliability, quality installation, and maintenance requirements. These factors can have a significant impact on energy savings, and can play a substantial role in determining the successful application of a technology at a broad scale. We identify the successes observed in these areas and recommend possible solutions where we recognize challenges.

Monitoring Plan

Overview of the Technical Approach

The two systems instrumented in this study were installed in October 2012 by Trane contractors. The RTU and the dual evaporative pre-cooler were both installed and commissioned by at that time. In March 2013, UC Davis installed a thorough array of instrumentation on each of the systems and began monitoring operation and performance at one-minute intervals.

Analog and digital measurements from each rooftop unit were collected by a data acquisition module located on board each unit. Data was collected over the course of the study, with minor gaps during any period when the equipment was shut down for service, update, or maintenance. The minute interval data from each unit is stored on board the data acquisition module for 24 hours, then automatically uploaded over the EDGE cellular network to an SFTP server hosted by the University. Data for each unit is collected on this server as a separate CSV file each day.

Raw day-by-day datasets for each unit were concatenated into larger datasets that group minute interval data into month-long time series data sets. These month-long files were then used as manageable chunks for further analysis and visualization. Data was collected continuously between March 2013 and June 2015. For the sake of clarity, the data presented in this report is drawn from the year of 2014. Data analysis and visualization was conducted using Matlab.

In addition to the array of instrumentation for minute-by-minute performance monitoring, the research team also conducted a number of in-field diagnostic measurements in order to build calibrated maps for particular operating variables. These in-field measurements were used to supplement the minute-by-minute data with information that is not easily measured continuously, but which is required to calculate meaningful performance characteristics such as cooling capacity and coefficient of performance. Pumped water flow rates were determined by measuring the mass of water pumped through the system over a measured period of time. Airflow rates were measured using a tracer gas airflow measurement system.

MONITORING PLAN

The research team developed a monitoring plan that allowed for (1) assessment of overall performance for system inputs and outputs (2) evaluation of sub component performance characteristics, and (3) consideration of dynamic equipment operating behaviors. The monitoring scheme utilized for the study is illustrated schematically in Figure 2. Table 2 provides a simple description of each measurement marked in the instrumentation schematic, and documents the performance specifications for the sensors.

Current transducers listed in the monitoring plan are used mainly for sensing component operations to determine system mode. The system amperage, line voltage, and power factor are recorded to accurately determine the total power draw for each minute of operation. The analog output channels noted represent the non-invasive measurement of dc-voltage signals used on-board the rooftop unit to control the operation of various components.

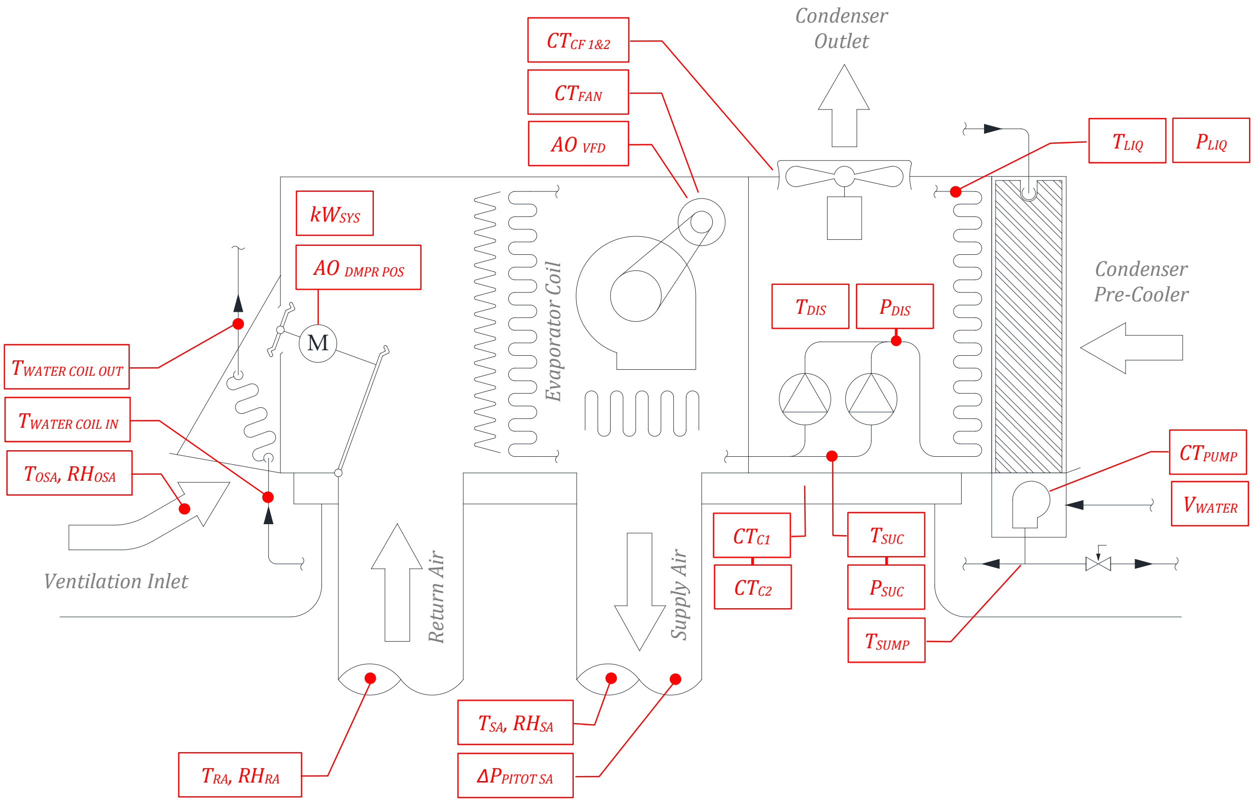

FIGURE 2 – INSTRUMENTATION SCHEMATIC

TABLE 2 – INSTRUMENTATION TABLE

Name	*Measurement*	*Sensor*	*Uncertainty*
T_{OA}	Temperature – Outside Air	Vaisala HUMICAP HMP110	+/- 0.2 °C
RH_{OA}	Relative Humidity – Outside Air	Vaisala HUMICAP HMP110	+/- 1.1% RH
T_{RA}	Temperature – Return Air	Vaisala HUMICAP HMP110	+/- 0.2 °C
RH_{RA}	Relative Humidity – Return Air	Vaisala HUMICAP HMP110	+/- 1.1% RH
T_{SA}	Temperature – Supply Air	Vaisala HUMICAP HMP110	+/- 0.2 °C
RH_{SA}	Relative Humidity – Supply Air	Vaisala HUMICAP HMP110	+/- 1.1% RH
$\Delta P_{PITOT\ SA}$	Pitot Tube in Supply Airstream	Dwyer 668-1	+/- 1% FS
CT_{C1}	AC Current – Compressor 1	NK AT1-005-000-SP	+/- 1% FS
CT_{C2}	AC Current – Compressor 2	NK AT1-005-000-SP	+/- 1% FS
CT_{PUMP}	AC Current – Pump	NK AT1-005-000-SP	+/- 1% FS
$CT_{CF\ 1\&2}$	AC Current – Condenser Fans	NK AT1-005-000-SP	+/- 1% FS
T_{SUC}	Suction Line Temperature	Omega 10k Ω TH-44031-40-T	+/- 0.1 °C
T_{DIS}	Discharge Line Temperature	Omega 10k Ω TH-44031-40-T	+/- 0.1 °C
T_{LIQ}	Liquid Line Temperature	Omega 10k Ω TH-44031-40-T	+/- 0.1 °C
P_{SUC}	Suction Line Pressure	ClimaCheck 200200 10*bar*	< 1% FS
P_{DIS}	Discharge Line Pressure	ClimaCheck 200100 35*bar*	< 1% FS
P_{LIQ}	Liquid Line Pressure	ClimaCheck 200100 35*bar*	< 1% FS
T_{SUMP}	Sump Water Temperature	Omega 10k Ω HSTH-44031	+/- 0.1 °C
$T_{WC\ IN}$	Water Coil Inlet Water Temperature	Omega 10k Ω TH-44031-40-T	+/- 0.1 °C
$T_{WC\ OUT}$	Water Coil Outlet Water	Omega 10k Ω TH-44031-40-T	+/- 0.1 °C
V_{WATER}	System Water Consumption	OMEGA FTB 4105 A P 1 pulse per	+/- 1.5% FS
$AO_{DMPR\ POS}$	RA/OA Damper Actuator Position	2-10 Vdc Analog Signal	NA
kW_{SYSTEM}	System Power Draw, Voltage, Current & PF	Dent Powerscout 3	+/- 0.5%

WATER AND AIRFLOW MEASUREMENTS

Accurate calculation of cooling capacity, coefficient of performance, and other metrics relies directly on accurate measurement of mass flow rates for water and air throughout a system.

Pumped water flow rates were determined by measuring the mass of water pumped out of the sump over a measured period of time. The pump operates at constant speed, so it was assumed that the flow rate of water circulated through each dual evaporative pre-cooler remains consistent throughout the monitoring period. Table 3 summarizes the measurements made for each system.

TABLE 3 – MEASUREMENT OF WATER FLOW RATE CIRCULATED THROUGH DUAL EVAPORATIVE PRE-COOLER

Tag	*Elapsed Time (seconds) (Avg. of 5 tests)*	*Water Weight Collected (lbs) (Avg. 5 tests)*	*Flow Rate (GPM)*
TAC2	21.3	43.2	14.6
MAC9	23.6	38.8	11.8

Supply airflow rates were determined using a tracer gas airflow measurement, conducted according to ASTM E2029 *Standard Test Method for Volumetric and Mass Flow Rate Measurement in a Duct Using Tracer Gas Dilution* (ASTM 2011). This method mixes a measured mass flow rate of CO_2 into the supply air stream then measures the corresponding rise in CO_2 concentration downstream. The volume flow of air into which the tracer is mixed can be calculated by the following relation:

$$\dot{V}_{Airflow} = \frac{\dot{V}_{CO_2}}{C_{CO_2\,downstream} - C_{CO_2\,background}} \qquad 1$$

This method has many advantages compared to conventional air balance techniques, the most significant of which is accuracy. The tracer gas airflow tools used can measure with a calculated uncertainty of less than ±2%. The tracer gas measurement was conducted across a range of fan speeds and operating conditions in order to build an airflow map for the system in all possible scenarios. While it is often overlooked, the outside air damper position can have a significant impact on supply airflow rates by changing the overall airflow resistance for the fan. The tracer gas measurements conducted here account for this characteristic by measuring supply airflow across a range of damper positions and fan speeds.

A similar method was used to measure the outside air fraction. While operating in each mode the CO_2 concentration was measured in the outside air and return air streams and the resultant concentration of their mixture was measured in the supply air stream. The ratio of outside air to the total supply airflow can be determined according to a conservation of mass. The mass balance calculations can be reduced to the following equation:

$$OSAF = \frac{\dot{V}_{OSA}}{\dot{V}_{SA}} = \frac{C_{SA} - C_{RA}}{C_{OSA} - C_{RA}} \qquad 2$$

Figure 3 and Figure 4 show the air flow rates measured for each unit. It should be noted that while both systems incorporate variable speed drives for the supply blower, they were field configured in a way that precluded the fan from changing speed with each mode of operation. Therefore, the supply air flow rates are a function of outside air damper position only. The outside air pathway for the rooftop units evaluated have a higher resistance to flow than the return air pathway; as a result, supply air flow tends to decrease when the outside air damper opens and the return air damper closes.

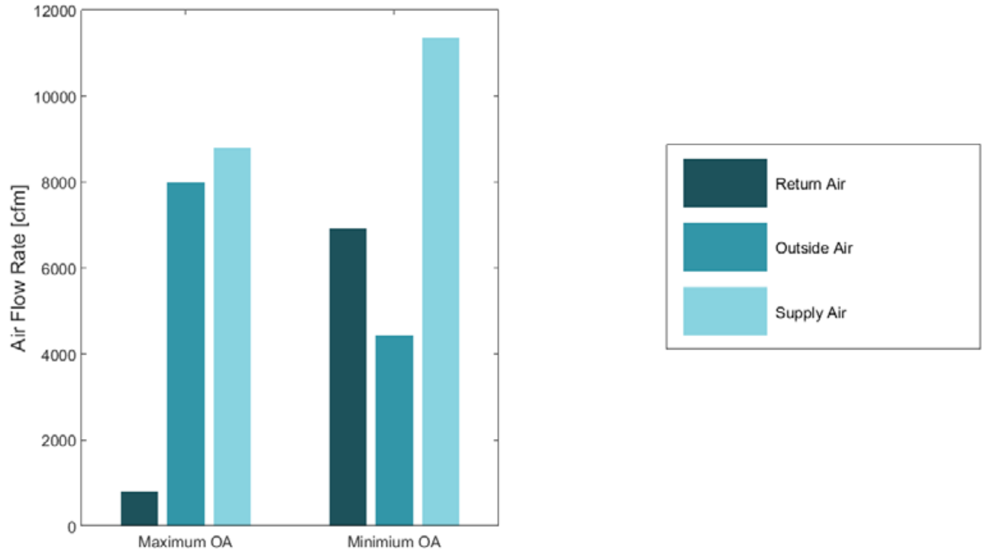

FIGURE 3 – MAC9 – AIR FLOW MEASUREMENTS

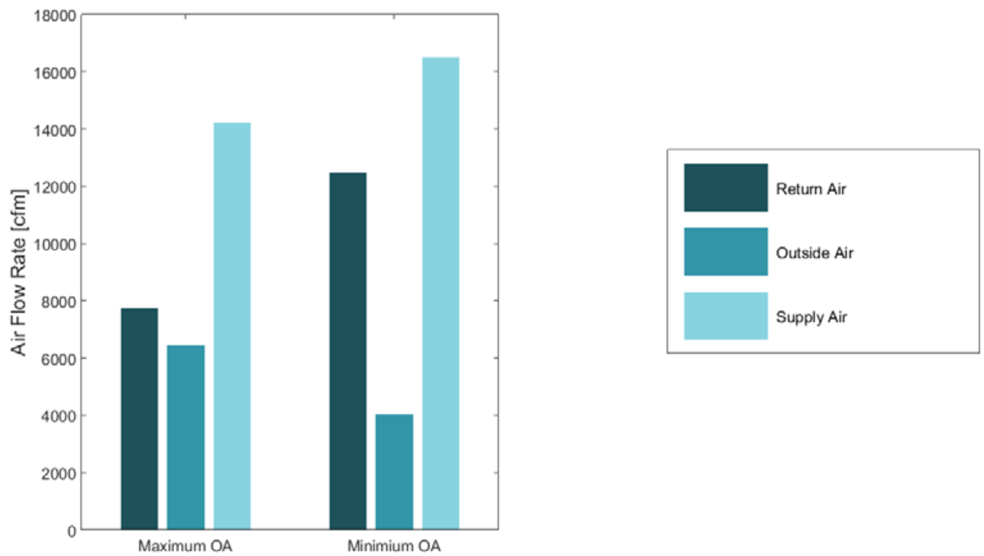

FIGURE 4 – TAC2 – AIR FLOW MEASUREMENTS

Table 4 compares the measured flow rates in both damper positions to the manufacturer specified nominal supply air flow rate.

Unit	Nominal Supply Air Flow Rate (CFM)	Damper Position	Measured Supply Air Flow Rate (CFM)	Deviation from Nominal (%)
MAC9	10,000	Minimum Outside Air	11,350	+13.5%
		Maximum Outside Air	8,800	-12%
TAC2	13,000	Minimum Outside Air	16,500	+27%
		Maximum Outside Air	14,200	+9%

TABLE 4 – MANUFACTURER SPECIFIED AND MEASURED SUPPLY AIR FLOW RATES

The RTU controllers were not configured to control their VFD to maintain a constant supply air flowrate for all modes of operation. Since the outside air flow path has a higher resistance than the return air path, a larger outside air fraction results in a smaller supply air flow rate. The configuration of the MAC9 unit resulted in a lower than nominal supply air flow rate when operating at the maximum outside air fraction and a larger than nominal supply air flowrate when operating at the minimum outside air fraction. The configuration of the TAC2 unit resulted a larger than nominal supply air flowrate for both outside air fractions.

Importantly, this airflow analysis also measures damper and cabinet leakage. When operating in economizer mode, the MAC9 unit achieves an outside air fraction of 91% while the TAC2 unit achieves an outside air fraction of only 45%. This fact is mostly a result of the baseline rooftop unit construction, and installation practices, however the trend is worsened somewhat by the added resistance from the ventilation cooling coil. When operating at minimum ventilation, the MAC9 and TAC2 units achieve outside air fractions of 39% and 24% respectively.

Determination of Operating Mode

The performance of each system changes discretely with mode of operation, therefore it is important to discretize results by the separate modes so that the observations may be analyzed and explained clearly. The operating mode for each one-minute interval record was determined by examination of several variables used as indicators for the function of each system component.

The explicit logic used to determine the state of each component and the system operating mode was slightly different for each unit. The two rooftop units are each configured differently, and operate with different component power draw characteristics in each mode of operation. The mode of operation was determined based on the power draw of the RTU, the damper positions and the current draw of the pump.

Data Confidence

The measurement accuracy for each instrument used for field monitoring is recorded as part of the monitoring plan in Table 2. Table 5 summarizes the degree of confidence for the key calculated metrics presented in this report. These values are calculated by propagation of uncertainty at a single operating condition. The values recorded here indicate the uncertainty resulting from manufacturer stated performance for the sensors used, and from the equations documented in section "Definition & Calculation of Performance Metrics". The values in Table 5 do not account for any methodological uncertainty associated with features such as sensor placement, or transient interaction between equipment operation and sensor response.

Table 5 – Uncertainty for Key Calculated Metrics[1]

Metric	*Uncertainty*
Supply Airflow Rate	±34 SCFM
Ventilation Airflow Rate	± 117 SCFM
Absolute Humidity	±0.00057 $lb_{m,\ water}$ / $lb_{m,\ dry\ air}$
Sensible System Capacity	±6.2 kBTU/hr
Sensible Room Capacity	±4.5 kBTU/hr
Sensible System COP	±1.2
Sensible Room COP	±0.9
Water Use	±0.06 gal/ton hr
Wet Bulb Effectiveness	± 0.08

[1] Uncertainty for each metric is calculated for the following conditions: $T_{DB\ OSA}$=105°F, $T_{WB\ OSA}$=73°F, $T_{DB\ RA}$=78°F, $T_{WB\ RA}$=64°F, $T_{DB\ SA}$=55°F, $T_{WB\ SA}$=52°F, Supply Airflow Rate = 1903 SCFM, Ventilation Flow Rate = 448 SCFM

DEFINITION & CALCULATION OF PERFORMANCE METRICS

CALCULATING COOLING CAPACITY

The system-cooling-capacity for the hybrid equipment is determined at any operating condition according to the supply air-flow rate and the specific enthalpy difference between the combined air streams entering the system and the supply air stream, as described by Equation 3. This is the net cooling produced by the equipment, including the capacity from cooling by the water coil, and including what is lost due to fan heat.

$$\dot{H}_{system} = \dot{m}_{SA} \cdot (h^*_{MA} - h_{SA}) \qquad 3$$

Where:

$\dot{m}_{SA}$	=	Mass flow rate of the supply air
h_{SA}	=	Specific enthalpy of the supply air
h^*_{MA}	=	Specific enthalpy of the mixed air

Where h^*_{MA} is the specific enthalpy of the 'virtual' mixed air, a parameter that does not physically exist. Generally, the system cooling capacity for a conventional rooftop unit is measured by the difference between the mixed air enthalpy and the supply air enthalpy. However, for the hybrid machine tested here, the ventilation air stream is cooled before it mixes with return air. The 'virtual' mixed air condition represents the combined enthalpy from all inlets to the equipment, and allows for accounting of the cooling delivered by the ventilation cooling coil. Equation 4 calculates the specific enthalpy for the 'virtual' mixed air condition.

$$h^*_{MA} = \mathrm{OSAF} \cdot h_{OA} + (1 - \mathrm{OSAF}) \cdot h_{RA} \qquad 4$$

Where:

$OSAF$	=	Outside Air Fraction
h_{OA}	=	Specific enthalpy of the outside air
h_{RA}	=	Specific enthalpy of the return air

Since ambient humidity in most western climates is low enough that dehumidification is not necessary to maintain occupant comfort in most commercial buildings, the system's ability to produce sensible cooling (ASHRAE 2010) is presented as well. The net sensible system cooling capacity is determined according to Equation 5:

$$\dot{H}^{sensible}_{system} = \dot{m}_{SA} \cdot C_{p\ air} \cdot (T^*_{MA} - T_{SA}) \qquad 5$$

Concomitantly, the latent system cooling is determined as:

$$\dot{H}^{latent}_{system} = \dot{H}_{system} - \dot{H}^{sensible}_{system} \qquad 6$$

Where:

$\dot{H}$	=	Rate of change of enthalpy
$C_{p\ air}$	=	Specific heat of air
T^*_{MA}	=	Temperature of the mixed air
T_{SA}	=	Temperature of the supply air

CALCULATING COEFFICIENT OF PERFORMANCE

Energy efficiency at any given operating condition is expressed as the dimensionless ratio of useful thermal capacity delivered to electrical power consumed by the system – the Coefficient of Performance:

$$COP = \frac{Thermal\ Energy\ Delivered}{Electrical\ Energy\ Consumed} = \frac{\dot{H}}{\dot{E}_{system}} \quad (7)$$

The sensible cooling generated by the equipment discounts the enthalpy associated with reduced humidity. The Sensible Coefficient of Performance can be expressed as:

$$COP_{sensible} = \frac{\dot{H}_{system}^{sensible}}{\dot{E}_{system}} \quad (8)$$

Where:

$\dot{E}_{system}$ = Power draw of the system

ANALYSIS OF VENTILATION AIR COOLING PERFORMANCE

The ventilation air mixes with return air immediately after passing through the water coil, as shown in Figure 5. Thus, it is not practical to measure the average temperature of this product air stream with confidence. Instead, analysis of the ventilation air cooling performance is determined indirectly according to conservation of energy, measurement of the water-side temperatures and the measured mass flow rates for water and ventilation air. Since the water temperature for this system must always remain above the outside air wet bulb temperature, it can be safely assumed that all heat transferred to the water stream corresponds to a sensible cooling effect in the ventilation air stream. The resulting product air temperature can be calculated by Equation 9.

$$T_{PA} = T_{OA} - \frac{\dot{m}_{water} \cdot C_{p\ water} \cdot (T_{WC\ out} - T_{WC\ in})}{\dot{m}_{OA} \cdot C_{p\ air}} \quad (9)$$

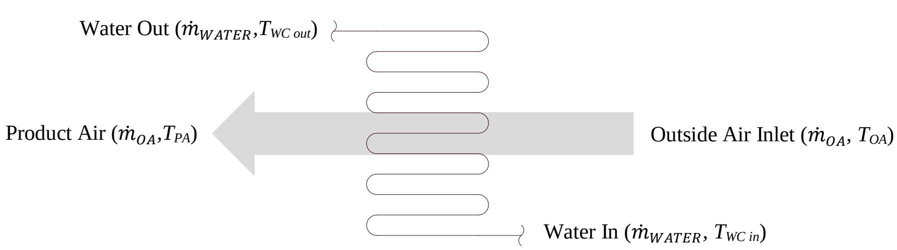

FIGURE 5 – SCHEMATIC FOR ENERGY BALANCE USED TO CALCULATE PRODUCT AIR TEMPERATURE

Where:

T_{PA} = Temperature of the product air

T_{OA} = Temperature of the outside air

$\dot{m}_{water}$ = Mass flow rate of water in the water coil

$C_{p\ water}$ = Specific heat of water

$\dot{m}_{OA}$ = Mass flow rate of the outside air

$T_{WC\ out}$ = Water temperature at the outlet of the water coil

$T_{WC\ in}$ = Water temperature at the inlet of the water coil

Calculating Wet-Bulb Effectiveness for Ventilation Air Cooling

The wet bulb effectiveness for the indirect evaporative cooling of the ventilation air is calculated according to Equation 10. This metric represents the degree to which ventilation air is cooled toward the outside air wet bulb temperature. It is calculated as the ratio of the change in ventilation-air dry-bulb temperature across the water coil, to the wet-bulb depression of the outside air.

$$\mathrm{WBE}_{IEC} = \frac{T_{OA} - T_{PA}}{T_{OA} - T_{wb\,OA}} \qquad 10$$

Calculating Sensible Heat Exchanger Effectiveness

The sensible heat exchanger effectiveness, which is the most common metric to describe heat exchanger performance, is determined by Equation 11.

$$\varepsilon_{sensible} = \frac{\dot{m}_{OA} \cdot c_{p\,air} \cdot (T_{OA} - T_{PA})}{(\dot{m} \cdot c_p)_{min} \cdot (T_{OA} - T_{WC\,in})} \qquad 11$$

Where:

$(\dot{m} \cdot c_p)_{min}$ = smaller of $\dot{m}_{OSA} \cdot c_{p\,air}$ and $\dot{m}_{water} \cdot c_{p\,water}$

COMPARISON TO A TYPICAL RTU

The performance of a typical RTU with capacity and characteristics similar to an unmodified Trane Voyager was simulated to compare against the field data. The RTU selected for the comparison is the Lennox TCA240S which is a two stage unit with a rated cooling capacity of 25 tons and a rated COP of 3.45. The two bi-quadratic curves used to calculate the performance of the Lennox TCA240S describe the total cooling capacity and power consumption as functions of outdoor dry-bulb temperature and return air wet-bulb temperature. The performance curves are easily scaled to match the capacity of the MAC9 and TAC2 units.

The performance of the Lennox RTU was calculated using the air conditions at each steady-state minute in the field data. Since the modeled Lennox RTU does not provide any ventilation air the mixed air wet-bulb temperature was used instead of the return air wet-bulb temperature. The bi-quadratic curves are shown in equation 12 and 13. The coefficients used for each curve are shown in Table 6 and Table 7.

$$\dot{H}_{system} = \dot{H}_{rated} \cdot \left(A + B \cdot T_{MA,wb} + C \cdot T_{MA,wb}^2 + D \cdot T_{OA,db} + E \cdot T_{OA,db}^2 + F \cdot T_{MA,wb} \cdot T_{OA,db}\right) \quad 12$$

TABLE 6 – TOTAL COOLING CAPACITY CURVE COEFFICIENTS FOR A LENNOX TCA240S RTU

	$\dot{H}_{rated}$		A	B	C	D	E	F
MAC9	228	Stage 1	0.9915	-0.0109	0.0011	-0.0007	0.0000	-0.0002
TAC2	290							
MAC9	330	Stage 2	1.0117	-0.0157	0.0013	0.0010	0.0000	-0.0002
TAC2	420							

$$P = \frac{\dot{H}_{rated}}{COP_{rated}} \left(A + B \cdot T_{MA,wb} + C \cdot T_{MA,wb}^2 + D \cdot T_{OA,db} + E \cdot T_{OA,db}^2 + F \cdot T_{MA,wb} \cdot T_{OA,db}\right) \quad 13$$

TABLE 7 – POWER DRAW CURVE COEFFICIENTS FOR A LENNOX TCA240S RTU

	COP_{rated}		A	B	C	D	E	F
MAC9 TAC2	3.47	Stage 1	0.5325	0.0034	-0.0002	0.0108	0.0003	-0.0004
MAC9 TAC2	3.45	Stage 2	0.6088	0.0104	-0.0003	0.0010	0.0005	-0.0005

Where:

$T_{MA,wb}$ = Wet-bulb temperature of the mixed air

$T_{OA,db}$ = Dry-bulb temperature of the outside air

RESULTS

The processed field data for the month of June 2014 is shown for both retrofitted RTUs (MAC9 and TAC2) in Figure 6 and Figure 7, respectively. The minute interval data was filtered to remove any points within 5 minutes of a change in operating mode. The following modes are plotted as separate data series:

1. Maximum outdoor air (OA) damper position with no compressor cooling
2. Maximum OA damper position with stage 1 (S1) cooling
3. Maximum OA damper position with stage 2 (S2) cooling
4. Minimum OA damper position with no compressor cooling
5. Minimum OA damper position with S1 cooling
6. Minimum OA damper position with S2 cooling

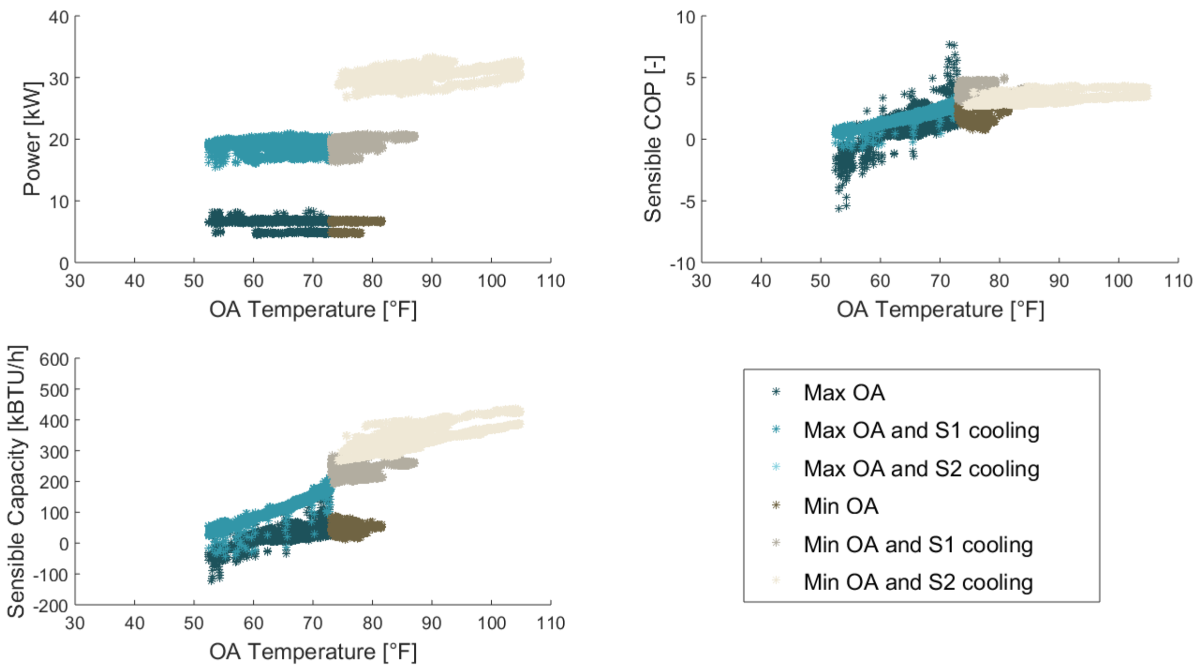

FIGURE 6 – MAC9 – PERFORMANCE VERSUS OUTSIDE AIR DRY-BULB TEMPERATURE (JUNE 2014)

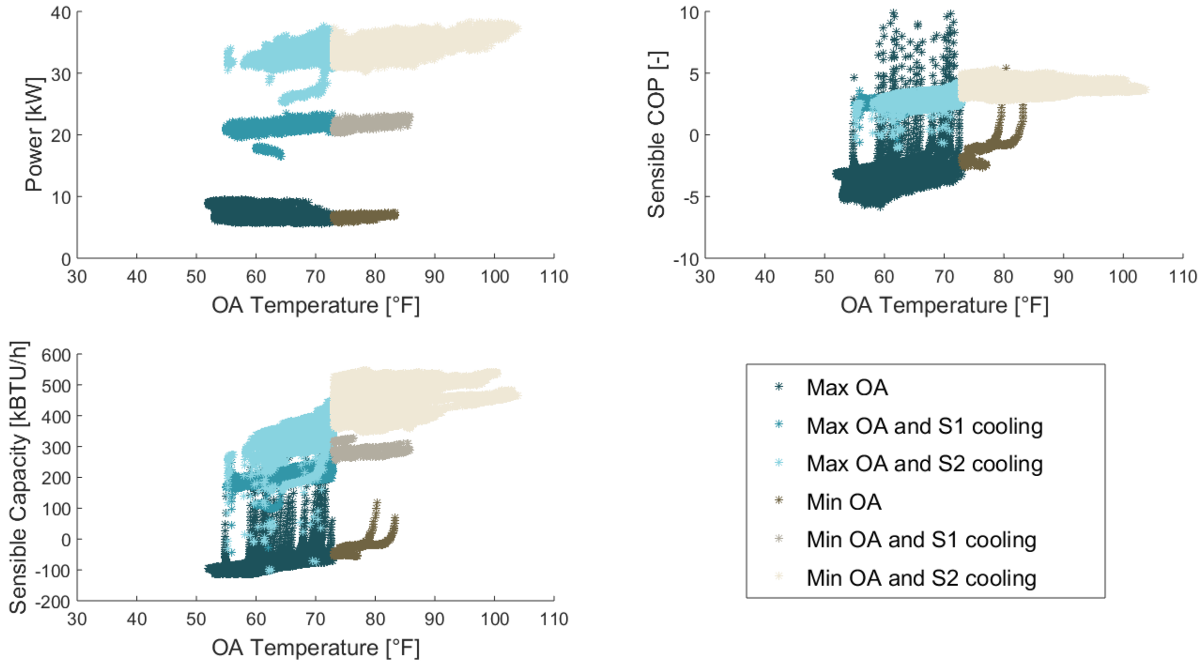

FIGURE 7 – TAC2 – PERFORMANCE VERSUS OUTSIDE AIR DRY-BULB TEMPERATURE (JUNE 2014)

By examining Figure 6 and Figure 7 it can be seen that the sensible capacity of both units is often negative when the outdoor air damper is fully open. This occurs when the supply air temperature is greater than the mixed air temperature. These results indicate that the heaters in both the MAC9 and TAC2 units operate some of the time. Additionally, since the capacity is occasionally negative even when the stage 1 and stage 2 compressors are on, it appears that there is some cycling between heating and cooling modes such that there is still heat retained in the furnace when the compressor turns on. Since the heater was not instrumented it is not possible to determine exactly how the heater is functioning or the effect that it is having on the cooling capacity.

Figure 6 and Figure 7 show that the sensible cooling capacity tends to increase with outdoor air dry-bulb temperature. The condenser air pre-cooler maintains reduced air temperatures at the condenser coil while the increase in the mixed air temperature results in a larger temperature difference across the evaporator coil, resulting in a larger capacity.

The MAC9 RTU never operates the second stage compressor when the outdoor air damper is fully open. The TAC2 RTU appears to be serving a larger load and often operates the second stage compressor when outdoor air damper is fully open. The TAC2 heater also operates far more frequently than the MAC9 heater.

At least two discrete power bands can be observed in each mode of operation. The discrete changes in power draw are likely due to the powered exhaust fans which maintain the building pressure. These were not instrumented separately but their power draw is included in the total measurement.

Despite the data filtering, transient effects can still be observed in the fingers extending out from the clouds of data. This shows that even five minutes after a change in operating mode the coils have not reached a steady state temperature.

Figure 8 and Figure 9 show the cumulative capacity and power for each hour of the MAC9 and TAC2 RTUs during the month of June 2014. The mean outdoor air temperature and one standard deviation above and below the mean at each hour are shown as well.

Figure 8 shows that the MAC9 RTU only operates between 6:00 am and 5:00 pm with a peak in delivered capacity at 3:00 pm which corresponds with the peak in outdoor air temperature. Figure 9 shows that the TAC2 RTU fan operates continuously and the compressors operate between 9:00 am and 10:00 pm. The operation of the heaters identified in Figure 6 and Figure 7 have an impact on the cumulative capacity for several modes of operation which could not be filtered out because the heater was not instrumented. However, since the effects of the heater can only be observed when the outdoor air damper is open, it can be assumed that when the unit is operating with the outdoor air damper in its most closed position the cumulative capacity is unaffected by the heater.

The cumulative capacity delivered to the space for both RTU's follows the curve of the mean outdoor air temperature. The TAC2 RTU uses the second stage compressor the majority of the time, regardless of damper position. When the outdoor air damper is in its most closed position, the MAC9 RTU operates the second stage compressor the majority of the time. When the outdoor air damper is open only the stage one compressor is used.

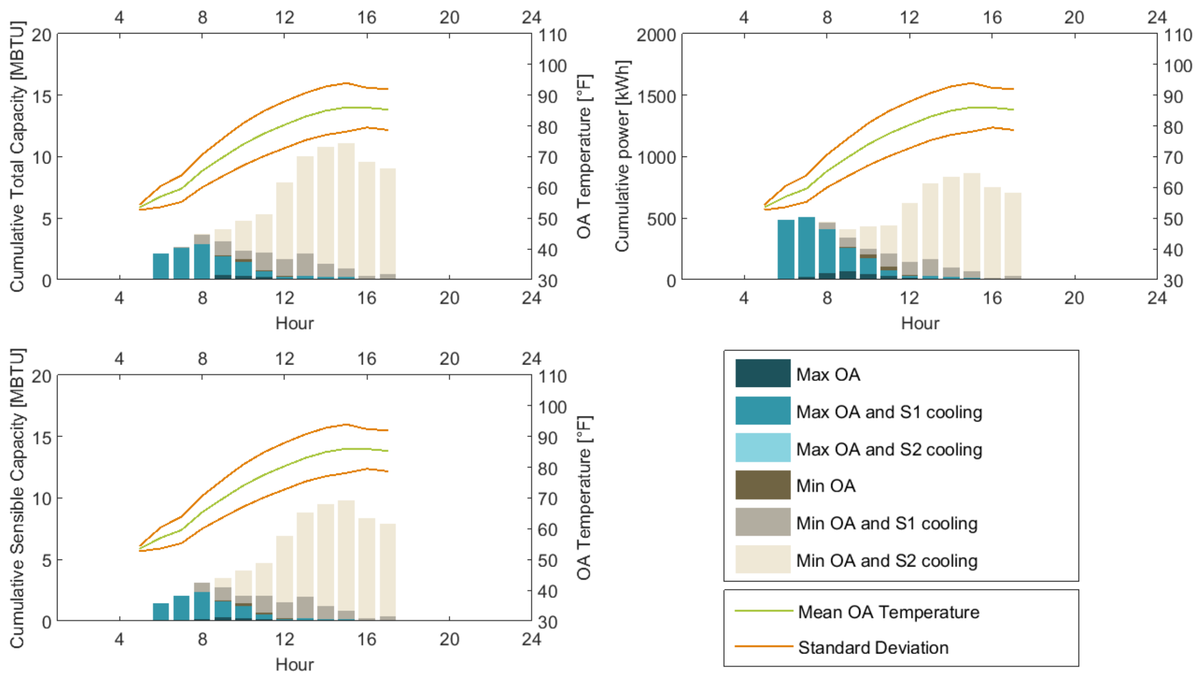

FIGURE 8 – MAC9 – CUMULATIVE PERFORMANCE FOR JUNE 2014

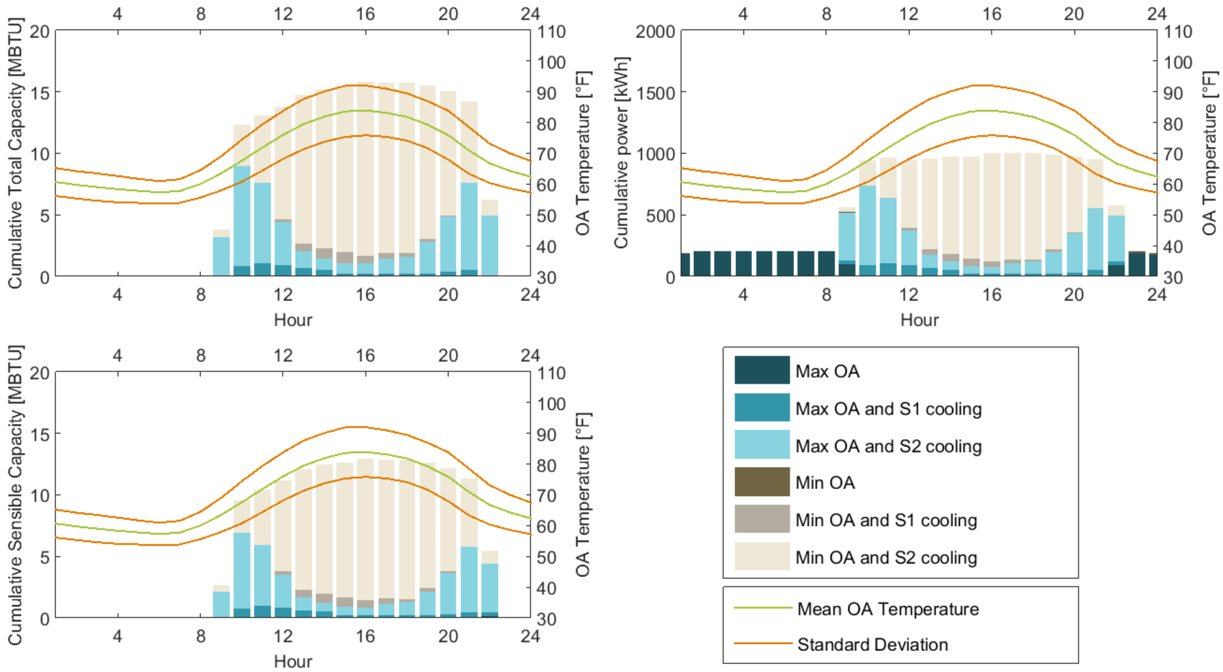

FIGURE 9 – TAC2 – CUMULATIVE PERFORMANCE FOR JUNE 2014

A comparison of the average by hour of the total cooling capacity and power between the field data for the MAC9 and TAC2 RTU's against the performance curves for the Lennox TCA240S is shown in Figure 10 and Figure 11. This comparison focuses on the RTU performance when the outdoor air damper is in its most closed position. As shown in Figure 6 and Figure 7, the heater does not operate in these operating modes. Additionally, the outdoor air damper is only open when it is cool outside and the cooling performance of the RTU is less critical.

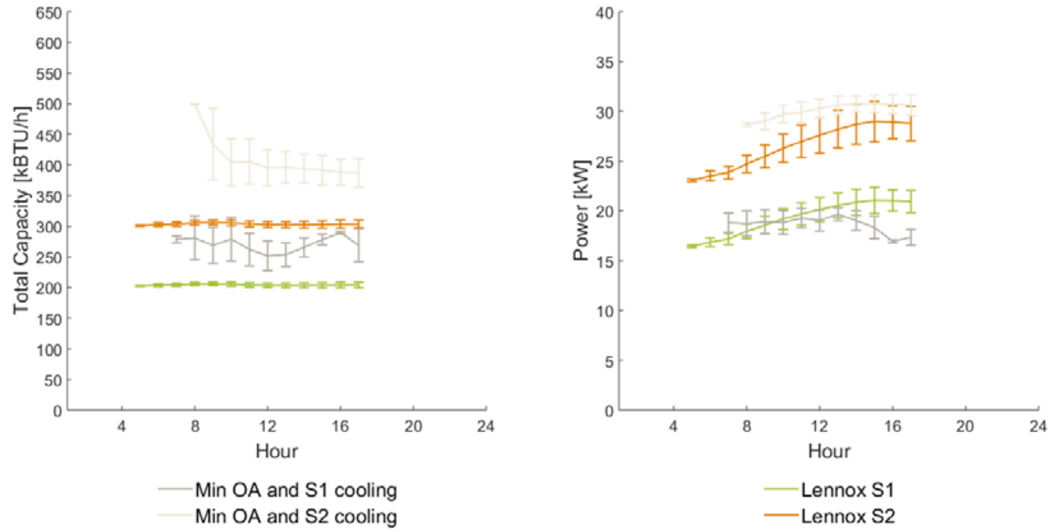

FIGURE 10 – MAC9 – AVERAGE DAILY PERFORMANCE – ERROR BARS INDICATE ONE STANDARD DEVIATION

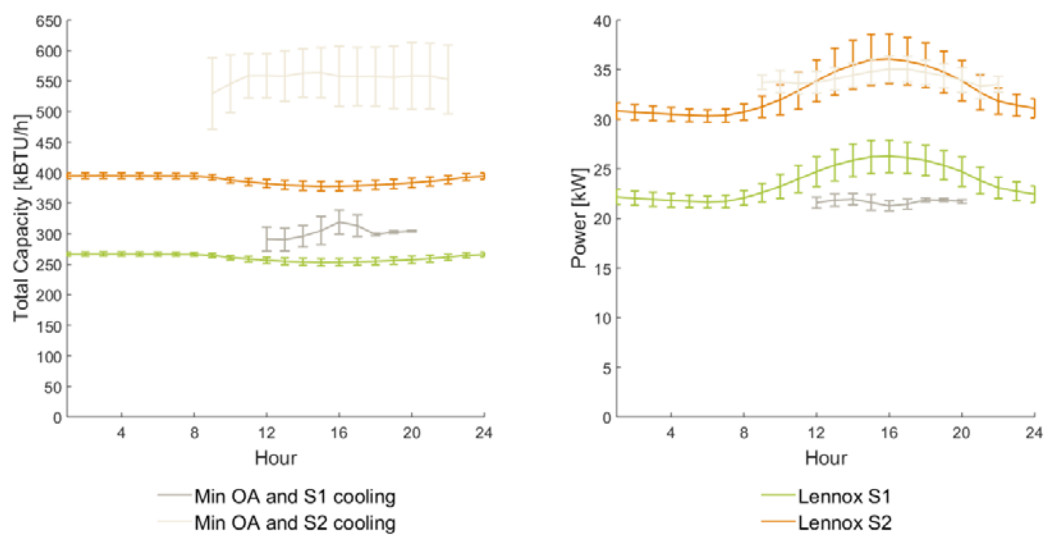

FIGURE 11 – TAC2 – AVERAGE DAILY PERFORMANCE – ERROR BARS INDICATE ONE STANDARD DEVIATION

As shown in Figure 10 and Figure 11 the cooling capacity of both RTU's is greater than the equivalently rated Lennox performance curves predict. When the first stage compressor is operating the MAC9 RTU consistently draws more power than the Lennox performance curves predict for the same mode of operation. The MAC9 RTU with the second stage compressor operating and the TAC2 RTU with the first stage compressor operating show similar power draws to those predicted by the Lennox performance curves. The TAC2 RTU with the second stage compressor operating shows consistently reduced power draw when compared to the Lennox performance curves for the same mode of operation.

TABLE 8 – CAPACITY AND POWER COMPARISON BETWEEN FIELD DATA AND LENNOX PERFORMANCE CURVES

	Mode	Average Capacity Difference	Average Power Difference	Peak Capacity Difference	Peak Power Difference
MAC 9	Stage 1	33%	-5%	42%	-5%
	Stage 2	35%	9%	28%	10%
TAC 2	Stage 1	19%	-15%	26%	-19%
	Stage 2	45%	0%	48%	-3%

The COP of the MAC9 and TAC2 RTUs are compared to the performance curves for the Lennox TCA240S in Figure 12 and Figure 13.

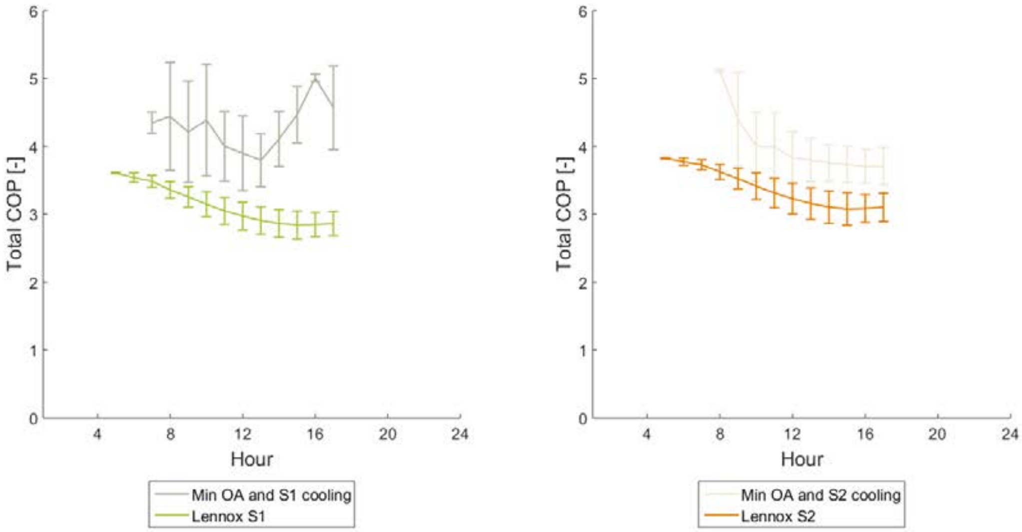

FIGURE 12 – MAC9 – AVERAGE DAILY COP – ERROR BARS INDICATE ONE STANDARD DEVIATION

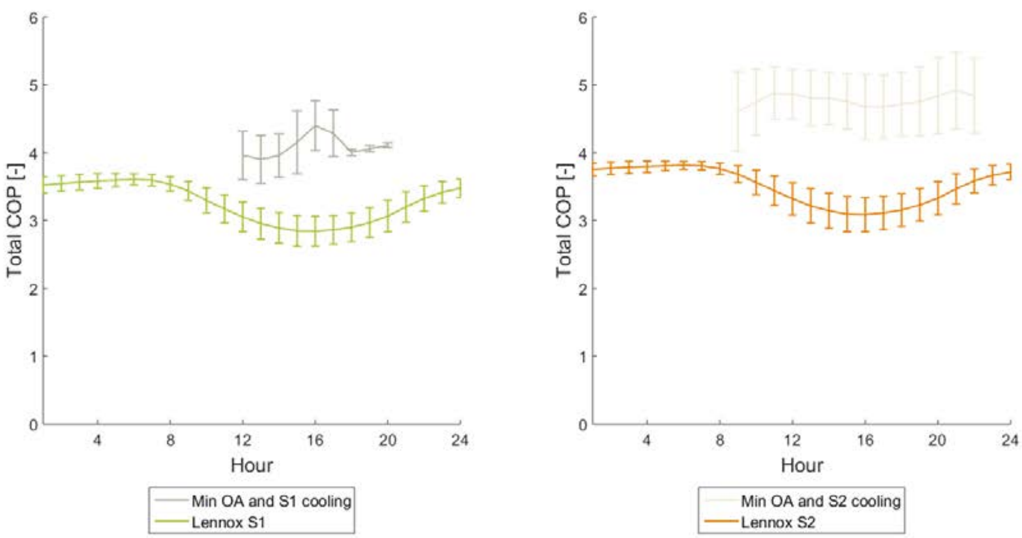

FIGURE 13 – TAC2 – AVERAGE DAILY COP – ERROR BARS INDICATE ONE STANDARD DEVIATION

The COPs of the both RTUs are consistently better than the predicted performance of the Lennox RTU. The COP is highest during peak hours in all cases except for the MAC 9 RTU with the second stage compressor operating.

The performance of the MAC9 outdoor air pre-cooler water coil is shown in Figure 14 and Figure 15. Similar results for the TAC2 RTU could not be produced due to a faulty current transducer installed on the TAC2 water pump.

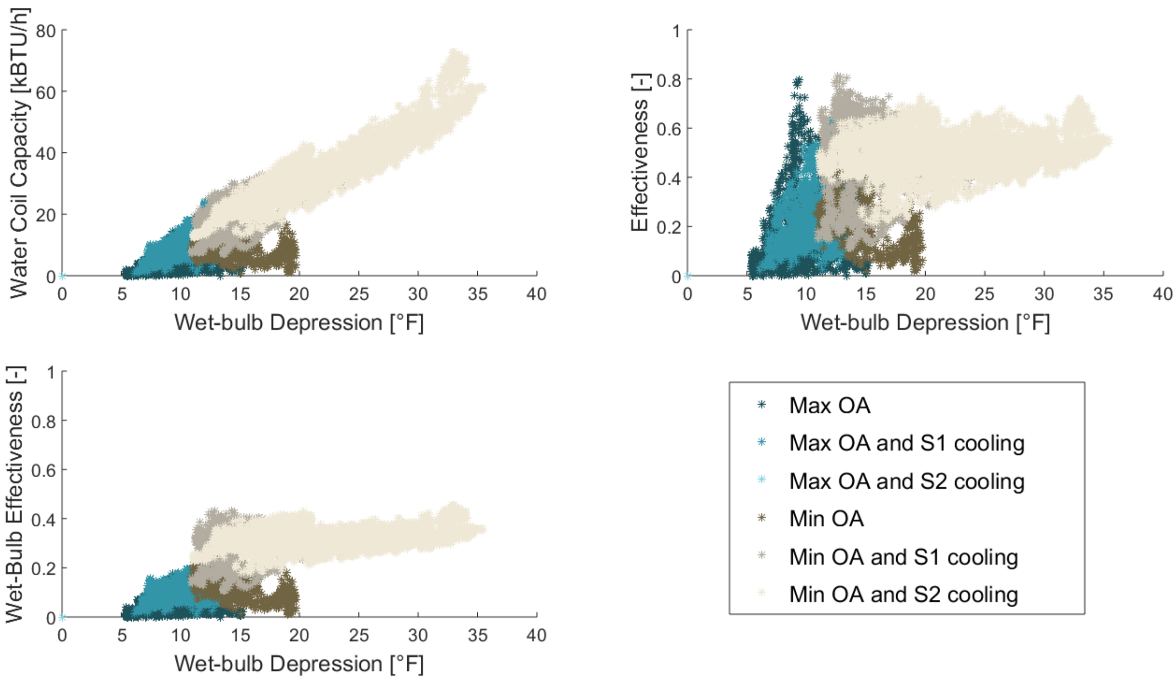

FIGURE 14 – MAC9 – DUAL PRE-COOLER WATER COIL PERFORMANCE VERSUS WET-BULB DEPRESSION

The water coil capacity increases with the wet-bulb depression. The heat exchanger effectiveness and wet-bulb effectiveness of the water coil do not have a strong trend with the wet-blub depression. However, there is a noticeable improvement when the outdoor damper is in its most closed position. This result indicates that there is ample airflow for the heat exchanger installed on this unit and that the performance could be improved by a larger heat exchanger.

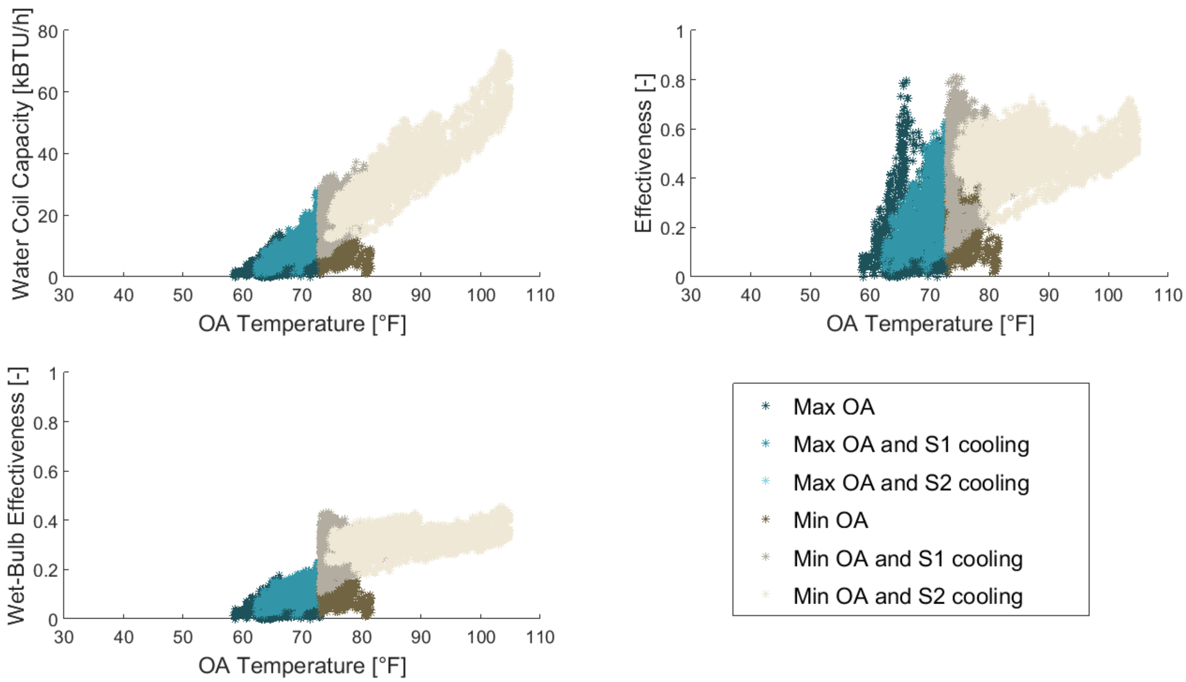

FIGURE 15 – MAC9 – DUAL PRE-COOLER WATER COIL PERFORMANCE VERSUS OUTDOOR AIR TEMPERATURE

Since wet-bulb depression tends to increase with outdoor air temperature the results in Figure 14 and Figure 15 are very similar. What is noteworthy in Figure 15 is that the results show that the water pump is operating at outdoor air temperatures as low as 60 °F despite the controller being configured to operate the pump only when the outdoor air temperature is above 70 °F. Since outdoor air temperature measurements were made using a sensor housed in a radiation shield it reasonable to assume that the cabinet housing the pump controller does not adequately insulate the temperature sensor resulting in the unintended operation shown in Figure 15.

Figure 16 shows the average contribution of the water coil to the total capacity of the MAC9 RTU. The error bars show the portion of the capacity that is a result of the outdoor air pre-cooler.

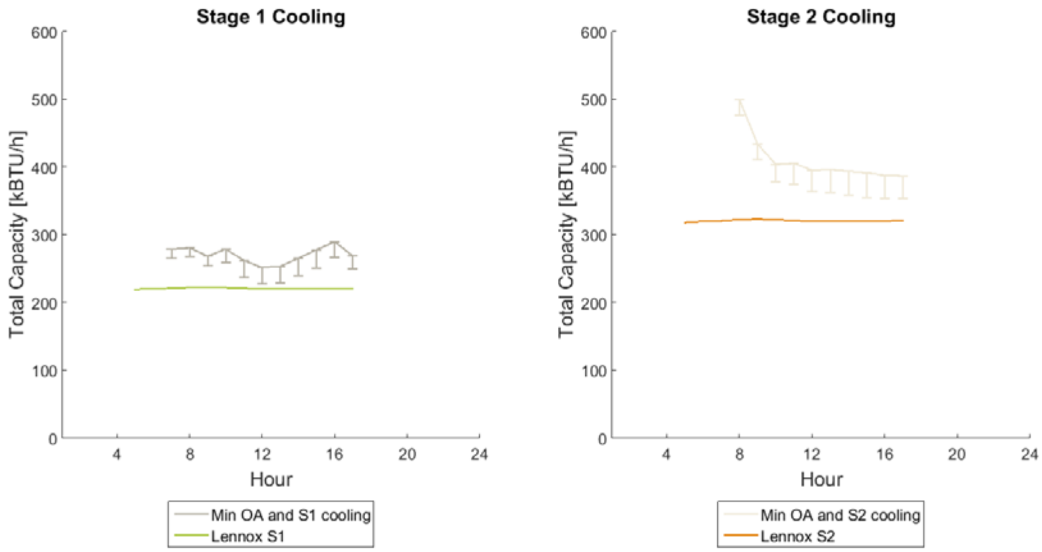

FIGURE 16 – MAC9 – PERFORMANCE DURING DUAL PRE-COOLER PUMP OPERATION

The capacity of a conventional vapor compression cycle decreases with the outdoor air dry-bulb temperature. It was shown that the capacity of the outdoor air pre-cooler increases with the wet-bulb depression. Since the wet-bulb depression increases with the dry-bulb temperature it makes sense that the capacity of the outdoor air pre-cooler increases as the capacity of the base RTU decreases. Therefore, the outdoor air pre-cooler is most effective at peak conditions and helps to offset the drop in performance of the base RTU.

CONCLUSION

The results of the field monitoring show that the dual evaporative pre-cooler can provide an increase to the total cooling capacity and efficiency of the RTUs, especially during peak hours. However, the results also show that there are significant improvements that can be made to the controller for both the pre-cooler and the RTU.

When compared to manufacturer data for a similar RTU the field data showed an improvement in peak capacity by as much as 64% and an improvement in average capacity by as much as 45%. The field data showed an improved overall efficiency in all modes of operation and a decrease in power consumption by as much as 15%. However, some modes of operation showed an increased power draw of as much as 16%. Additionally, the performance of the two RTU's differed from each other significantly.

One major difference in the operation of the two RTUs was the speed at which their fans were set to operate. Flow measurements revealed that the additional resistance imposed by the pre-cooler water coil resulted in a decrease in overall air flow rate of approximately 20-25% when the outdoor air damper was fully open. The MAC9 RTU was commissioned such that the flow rate was approximately 12% below nominal when the outdoor air damper was fully open and 12% above nominal when the outdoor air damper was fully closed. Conversely, the TAC2 RTU was commissioned such that the flow rate was approximately 9% above nominal when the outdoor air damper was fully open and 27% above nominal when the outdoor air damper was fully closed. The increased airflow may explain why the TAC2 RTU showed much better performance than the MAC9 RTU.

The field monitoring data reveals that the controls implemented to govern the operation of the dual evaporative pre-cooler and the RTUs are not optimized to take full advantage of the benefits that the technology could offer.

Although both units have VFDs on the indoor fans, neither are actively controlled in any way other than to turn the fan on and off. To maximize capacity and performance, the VFDs could be controlled to maintain a constant supply air temperature when the compressors are operating. When properly commissioned, controlling the fan in this way would allow the system to compensate for the added flow resistance imposed by the pre-cooler water coil and maximize the efficiency of the vapor compression cycle.

The heaters of both RTUs operate while the outdoor air damper is fully open. The introduction of additional cold outdoor air beyond what is needed for ventilation results in a larger and unnecessary heating load. To avoid this unnecessary load the outdoor air damper should only open fully when there is a call for cooling.

Since the heaters were not instrumented it is hard to tell exactly when the heater operates; however, even if the heaters and the compressors never operate simultaneously the data shows that the RTUs cycle between heater and compressor operation in such a short amount of time that the air is still being heated by the residual heat in the heater coil while the compressors are operating. This is likely the result of a small or non-existent dead band between the heating and cooling setpoints for the thermostat or a sudden change in setpoint temperatures in the thermostat schedule. Either way, the inefficiencies could be significantly reduced by implementing a minimum amount of time between when the RTU switches from heating to cooling or vice versa.

The capacity of the evaporative ventilation air pre-cooler was shown to increase with outdoor air up to 70 kBTU/hr at an outdoor air temperature of 100 °F. However, at outdoor air temperatures less than 70 °F the capacity of the pre-cooler proved to be less than 20 kBTU/hr and at 60 °F the pre-cooler capacity was close to 0 kBTU/hr. This steep decline in performance with outdoor air temperature is the reason the pre-cooler was configured with a controller that would turn the water pump off at outdoor air temperatures below 70 °F. Despite this control strategy the water pump often operates when the outdoor air temperature is as much as 10 °F below the threshold temperature. The reason the pre-cooler pumps do not operate as intended is likely because the temperature sensor in the controllers are not properly shielded from solar and radiant effects.

While the benefits of the dual evaporative pre-cooler are promising, the results of this field demonstration show that improvements in capacity and efficiency are possible. Further research and development should be focused on developing a controller that optimizes the performance of both the RTU and the dual-evaporative pre-cooler.

REFERENCES

A2B. Accurate Air Balancing LLC. Test and Balance Report. Whole Foods Market. 100 Sunset Dr. April 24, 2014.

ARB. California Greenhouse Gas Emission Inventory. Air Resources Board. California Environmental Protection Agency. 2014. Online. http://www.arb.ca.gov/cc/inventory/inventory_current.htm

Architectural Energy Corporation, and Fisher Nickel Inc. Commercial Kitchen Ventilation Design Guides. Improving Commercial Kitchen Ventilation System Performance. Updated March 2004. Appendix to report: Makeup Air Effects on the Performance of Kitchen Exhaust Systems. CEC #500-98-031. December 2002. Online: http://www.fishnick.com/ventilation/designguides/

ASHRAE. 2009. ASHRAE Handbook – Fundamentals. IP Edition. Amercian Society of Heating and Air-Conditioning Engineers, Inc. Atlanta GA. 2009.

ASHRAE. Advanced Energy Design Guide for Grocery Stores – Achieving 50% Energy Savings - Toward a Zero Net Energy Building. American Society of Heating Refrigeration and Air Conditioning Engineers. 2015. Online. https://www.ashrae.org/standards-research--technology/advanced-energy-design-guides

California Energy Commission. 2013 Building Energy Efficiency Standards for Residential and Nonresidential Buildings. Title 24, Part 6 and Associated Administrative Regulations in Part 1. CEC-400-2012-004-CMF-REV2. Edmund G. Brown Jr., Governor. California Energy Commission. May 2012. Online. http://www.energy.ca.gov/2012publications/CEC-400-2012-004/CEC-400-2012-004-CMF-REV2.pdf

California Energy Commission. California Commercial End-Use Survey. CEC 400-2006-005. March 2006. Online. http://www.energy.ca.gov/ceus/

Energy Information Administration. Commercial Buildings Energy Consumption Survey. 2012. Online. http://www.eia.gov/consumption/commercial/

Fricke, B. Becker, B. Energy Use of Doored and Open Vertical Refrigerated Display Cases. Purdue University. International Refrigeration and Air Conditioning Conference. 2010. Online. http://docs.lib.purdue.edu/iracc

Google. Imagery ©2015 Goolge, Map data ©Google. Online. https://www.google.com/maps

Hale, E.T. Macumber, D.L. Long, N.L. Griffith, B.T. Benne, K.S. Pless, S.D. Torcellini, P.A. Technical Support Document: Development of the Advanced Energy Design Guide for Grocery Stores – 50% Energy Savings. National Renewable Energy Laboratory. Technical Report NREL/TP-550-42849. September 2008.

Harrington, C. Woolley, J. Davis, R. Laboratory Performance Results: Indirect Evaporative Air Conditioning & Condenser Pre Cooling as Climate Appropriate Retrofits for Packaged Rooftop Units. Southern California Edison. Customer Programs and Services. 2015. Online. http://www.etcc-ca.com/ (submission pending publication)

Hirsch, A. Hale, E. Leach, M. Strategies to Save 50% Site Energy in Grocery and General Merchandise Stores. ACEEE Summer Study, 2012.

NOAA. Quality Controlled Local Climatological Data. National Climatic Data Center. National Oceanic and Atmospheric Administration. 2015. Online. http://www.ncdc.noaa.gov/data-access/land-based-station-data/land-based-datasets/quality-controlled-local-climatological-data-qclcd.

Wang, W. Katipamula, S. Huang, Y. Brambley MR. Energy Savings and Economics of Advanced Control Strategies for Packaged Air Conditioning Units with Gas Heat. U.S. Department of Energy. DE-AC05-76RL01830. Pacific Northwest National Laboratory. PNNL-20995. December 2011. http://www.pnnl.gov/main/publications/external/technical_reports/PNNL-20955.pdf

Woolley, J. Davis, R. Western Cooling Challenge Laboratory Performance Results: Munters EPX 5000 Hybrid DOAS. ET12PGE3101. Pacific Gas & Electric Company. 2013. Online. http://www.etcc-ca.com/

Woolley, J. Modera, M. Western Cooling Challenge Laboratory Results: Trane Voyager DC Hybrid Rooftop Unit. Southern California Edison, Design and Engineering Services. HT.12.SCE.017. 2012. Online. http://www.etcc-ca.com/