FIELD EVALUATION OF AN EVAPORATIVE CONDENSER AIR PRE-COOLER ADDED TO A PACKAGED ROOFTOP AIR CONDITIONER FOR A DATA CENTER

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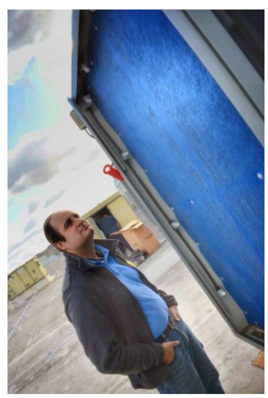

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DISCLAIMER

EXECUTIVE SUMMARY

BACKGROUND

As part of the California Energy Efficiency Strategic Plan, the California Public Utilities Commission introduced four Programmatic Initiatives as "Big Bold" strategies to advance energy efficiency in California. The first two strategies focus on advancing zero net energy for residential and commercial buildings. The third initiative envisions broad transformation for HVAC - mainly as a transition toward climate specific cooling solutions that have efficiency advantages unique to California's climate. The evaporative condenser air pre-cooling technology evaluated in this study is thought to be one climate appropriate solution that can be added to existing rooftop air conditioners as a retrofit. Laboratory studies have measured 27% decrease in energy intensity from these systems at typical peak conditions, and recent exploratory research has indicated that the combined impact of condenser pre-cooling and variable capacity controls could reduce energy intensity at peak by as much as 38% without sacrificing cooling capacity (Pistochini 2015).

OBJECTIVE

This study set out to carefully measure the in-field performance improvements for one evaporative condenser air pre-cooler, applied to improve the cooling efficiency for a data center. A secondary objective was to observe and document the real world experiences with this technology, and to develop some understanding about the longevity and maintainability for the technology.

APPROACH

The condenser pre-cooler was installed on an existing 17.5 ton rooftop air conditioner that cools a small data center in Rancho Santa Margarita – California Climate Zone 8. A thorough suite of instrumentation was deployed to monitor power consumption, water consumption, and temperature and humidity throughout the system. The monitoring plan was designed to capture characteristic performance data about the system in all modes of operation.

After several months of commissioning conducted in cooperation with the manufacturer, several weeks of data was collected for system operation across a range of ambient conditions. Toward the end of the cooling season, the condenser pre-cooler was removed, and the system was monitored for several weeks to capture a complete characterization of baseline performance across a similar range of operating conditions.

RESULTS

Analysis of system performance from several perspectives indicates that the evaporative condenser air pre-cooler did not achieve energy savings.

Generally, we encourage further efforts to advance programs that target broad application of condenser pre-coolers and other climate appropriate HVAC strategies. However, we highly recommend that utility programs and practitioners should distinguish carefully between those products that perform well and those that do not. One relevant effort is currently underway through ASHRAE SPC 212, to develop a standard test method for the evaluation of condenser air pre-coolers. Programmatic efforts and design specifications should reference these standards and should set minimum performance requirements for products in this space.

Observations from this study also suggest that certain scenarios may not benefit from evaporative condenser air pre-cooling, even if the product utilized performs exceptionally well. Power draw for the rooftop unit in this study did not increase with outside air temperature at the rate typically predicted for rooftop air conditioners. This phenomenon should be considered further, and efforts to apply similar technologies should be careful to target applications where they will have significant savings potential.

Abbreviations and Acronyms

ASHRAE	American Society of Heating Refrigeration and Air Conditioning Engineers
CEC	California Energy Commission
COP	Coefficient of Performance (*dimensionless*)
c_p	Specific Heat Capacity (e.g. *Btu/lbm-°F*)
C_X	Concentration (*of constituent X*) (eg. *ppm*)
DX	Direct Expansion Vapor Compression
$\dot{E}$	Electric Power, (Rate of Electric Energy Consumption) (e.g. *kW*)
EIA	Energy Information Administration
$\dot{H}$	Cooling Capacity, (Enthalpy Flow Rate) (e.g. *kBtu/h*)
h	Specific Enthalpy (e.g. *Btu/lbm-dryair*)
$\dot{m}$	Mass Flow Rate (e.g. *lbm/h*)
OSA	Outside Air
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

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INTRODUCTION

The California Energy Efficiency Strategic Plan advances goals for broad market deployment of climate appropriate cooling solutions. Technologies in this category offer greater energy savings in California's climates than other cooling efficiency measures. They are especially valuable for electricity savings during peak cooling hours when air conditioning alone can account for more than 30% of the peak demand on the statewide electric network (EIA 2014, CEC 2006). Many studies have demonstrated that add-on evaporative pre-coolers for conventional air conditioners can reduce energy use for cooling during peak periods by as much as 30-60% (Woolley 2014, Modera 2014, Pistochini 2014, Davis 2014). However, the savings achieved depends significantly on the technology and the application.

This study evaluates the measured performance of one evaporative condenser air pre-cooler installed on a packaged rooftop air conditioner at a data center in Rancho Santa Margarita, California – CA Climate Zone 08. The technology in this application achieved zero savings. This conclusion demonstrates that not all products in this technology class perform similarly, and underscores the need for a standard test methodology to rate performance of condenser pre-cooler systems from different manufacturers (Pistochini 2014, ASHRAE SPC 212 2014).

Cooling and heating for commercial buildings is served predominately by packaged rooftop air conditioners. These systems are often the single largest connected load in a building, and utilize technology that has failed to achieve the same degree of efficiency improvements advanced by other key end-use sectors, such as lighting. Cooling and ventilation is responsible for more than 25% of the annual electricity use in commercial buildings, and can account for more than 50% of the peak demand for typical commercial customers. These systems are responsible for a prodigious share of our energy consumption and annual greenhouse gas emissions. Intermittent operation of air conditioning systems is burdensome to grid demand management and has required immense public investment in a large and underutilized generation, transmission, and distribution infrastructure (Oldak 2012).

The realization of strategic energy and environmental goals, such as the California Greenhouse Gas Emission Standards, Renewable Portfolio Standard, and Energy Efficiency Strategic Plan, will necessitate a major market transformation for heating, cooling, and ventilation in California buildings. Evaporative cooling strategies could play a significant role in achieving these goals, and retrofit solutions may allow for quick application in buildings where existing equipment still has many years of useful life. However, programs and efforts designed to encourage broader application of climate appropriate technologies should be careful to distinguish between products that achieve good savings, and those that do not.

This report describes the technology evaluated, the application in which the measure was installed, and some of the design considerations that were made for setup and installation of the technology. Then, the experimental design and technical methodology for analysis is documented. Finally, the *Results & Discussion* section presents an array of performance metrics to characterize the overall system performance before and after the retrofit. The results presented also assess thermodynamic performance of the pre-cooler, separate from the rooftop unit, so that its performance characteristics might be applied to other scenarios.

PROJECT OVERVIEW

OVERVIEW OF THE EVAPORATIVE CONDENSER AIR PRE-COOLING TECHNOLOGY

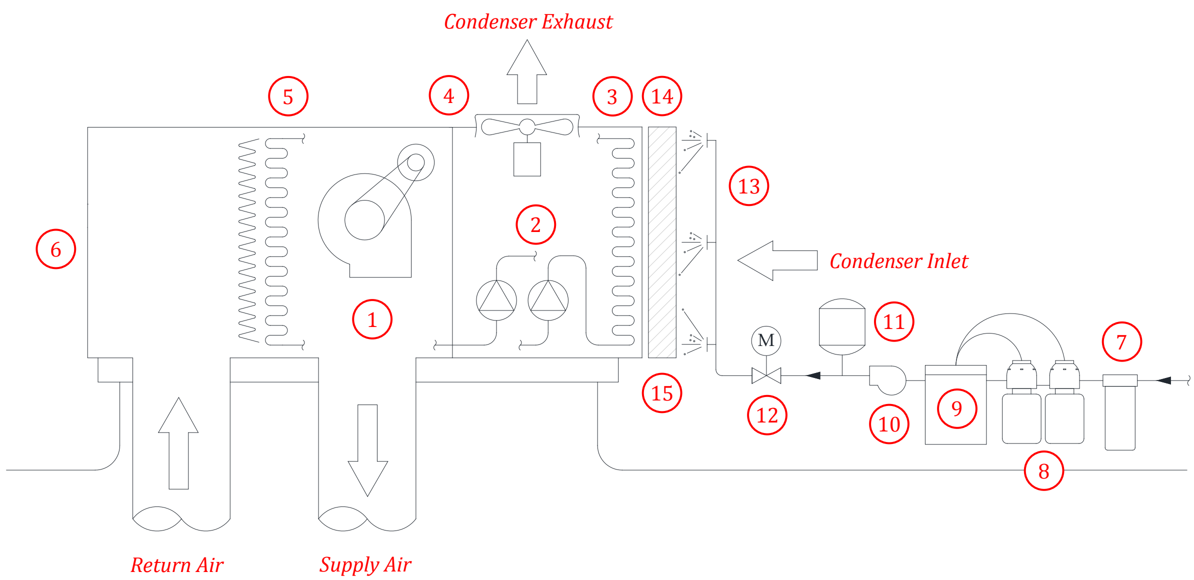
Evaporative condenser air pre-cooling reduces the temperature of the air stream used for heat rejection from a vapor compression air conditioner. Since efficiency of the ideal vapor compression cycle is driven by the temperature difference between refrigerant evaporation and refrigerant condensation, a reduction in condenser temperature can increase the cooling capacity and efficiency. This theoretical potential has been well studied, and many pre-cooling technologies have demonstrated substantial savings in practice (Wang 2014). There are many approaches to the physical design for evaporative condenser air pre-coolers which can impact on the energy savings achieved, the water use, the maintainability and longevity of the systems.

The technology studied here uses an array of nozzles to spray water onto a polymer 'filter media' that is installed across the face of the condenser air inlet for a vapor compression system. For this project, the pre-cooler was added to a rooftop air conditioner, but the same technology has been used for air cooled chillers in many applications. The ½" thick polymer media is installed in a plastic frame, which is mounted onto the air conditioner using powerful magnets. The technology is custom built for every application, so it can be added to almost any air conditioner, regardless of its size and arrangement. Water piping is routed inside the plastic frame, and specially selected spray nozzles oriented to spray water onto the media are arranged around the perimeter of each frame. Sometimes, multiple frames are used to cover a large condenser coil; breaking the area up into sections provides rigidity for the 'filter media', and allows spray nozzles to be located at the interior of a large condenser inlet area.

The technology also incorporates a simple screen at the inlet face to avoid entrainment of larger debris, and is designed to be easily removed so as to facilitate access to the condenser coils for cleaning or for service.

Condenser pre-coolers add some airflow resistance in front of the condenser, which can reduce condenser airflow considerably. Laboratory testing of some pre-coolers has indicated as much as 11% reduction in condenser airflow (Davis 2015) , which tends to reduce condenser heat transfer rate. Generally, the positive impact of reduced temperature is much more significant than any negative impact associated with reduced airflow, however, it is important that system design consider this factor. A parallel laboratory test of the technology evaluated here indicated 10-15% reduction in airflow (Pistochini 2014).

Water flow for the technology can be driven by grid pressure, but water must be supplied between 60–100 PSI to ensure proper spray distribution. In applications where this water pressure is not available such as for this evaluation, the system incorporates a booster pump. The system evaluated in this study also incorporated a water softener, which is needed to keep mineral scale from depositing on the media. The water softener was not functional for several weeks during this study, in which time the media accumulated a substantial amount of scale. When the water softener was repaired the 'filter media' was also replaced.



1	*Constant Speed Supply Fan*	2	*2 Compressor Circuits*	3	*Condenser Coils*
4	*2 Condenser Fans*	5	*Evaporator Coils*	6	*No Outside Air Inlet*
7	*Coarse Filtration*	8	*Water Softener*	9	*Brine Reservoir*
10	*Booster Pump*	11	*Pressure Vessel*	12	*Solenoid Valve(s)*
13	*Spray Nozzles*	14	*Poly. 'Filter Media'*	15	*Drain Holes*

FIGURE 1: SCHEMATIC OF ROOFTOP UNIT WITH EVAPORATIVE CONDENSER AIR PRE-COOLER

Solenoid valves are used to manage water distribution to a group of nozzles on each frame. The pre-cooler system is programmed to operate only when outside air temperature is above 70°F, and whilst the unit is actively cooling. When the pre-cooler is enabled, water is not sprayed onto the media continuously; instead, these solenoid valves cycle in a periodic pattern intended to supply only as much water as will be evaporated. The frequency and duration of pulses to spray water on the media varies with outside air temperature and humidity, such that the total amount of water supplied to the media increases with the evaporation potential.

Controls for the pre-cooler are powered by low voltage from the rooftop unit, but otherwise stand alone and do not require complex integration with the unit controls or revision to the whole building energy management and control system. This is a major advantage in comparison to other efficiency measures that require more integral revisions to systems and controls. These evaporative pre-cooler controls are straightforward; they use a programmed microprocessor to adjust the patterns of water delivery in response to compressor operation and the measured outdoor temperature and humidity .

The installation studied also incorporated a monitoring system (separate from the instrumentation used as the basis for this study) that communicated real time performance data back to the manufacturer over the cellular network. This type of real time monitoring has the potential to serve valuably for fault detection and diagnostics, historical benchmarking, and for ongoing verification of performance. However, it should be noted that the availability of data about system operation does not guarantee good performance. Continuous monitoring is only valuable if the right information is measured, and if there is a reliable and verifiable process and method to review and apply the data. Although the continuous data stream was available to the manufacturer over the course of this project, the lack of savings achieved was not apparent until the research team developed a thorough review of all aspects of equipment performance.

Lastly, it should be noted that the evaporative pre-cooler evaluated here does not require a plumbed drain – any water that does not evaporate drips off the bottom of the filter media and drains out of the plastic frame onto the rooftop. Ideally, all of the water sprayed would be evaporated into the condenser air stream and no excess water would remain, however in practice some water does drain from the pre-cooler onto the rooftop.

Overview of Field Test Site

This project was conducted in Rancho Santa Margarita, California, in Climate Zone 8. Figure 2 A identifies the location on a map with other California Climate Zones. The region is characterized by warm summers and mild winters. Normal summer highs occasionally break 100°F, and daily average temperature in the summer is almost always between 70–80°F. Temperature in the winter rarely drops below 40°F, and the average temperature in the coolest months is usually around 60°F. Some days in the winter months are as high as 90°F. Humidity is relatively high compared to other California climates; summertime dew point is consistently around 60°F, while dew point in the wintertime is somewhat lower and much more variable. This limits the potential impact for evaporative condenser air pre-cooling, since the wet bulb depression for periods above 70°F averages about 15°F. Notwithstanding, the hottest days are driven by Santa Ana winds which also bring low humidity. The hottest afternoons – when condenser pre-cooling should be most valuable – are usually quite dry. This occurs mainly in the spring and autumn.

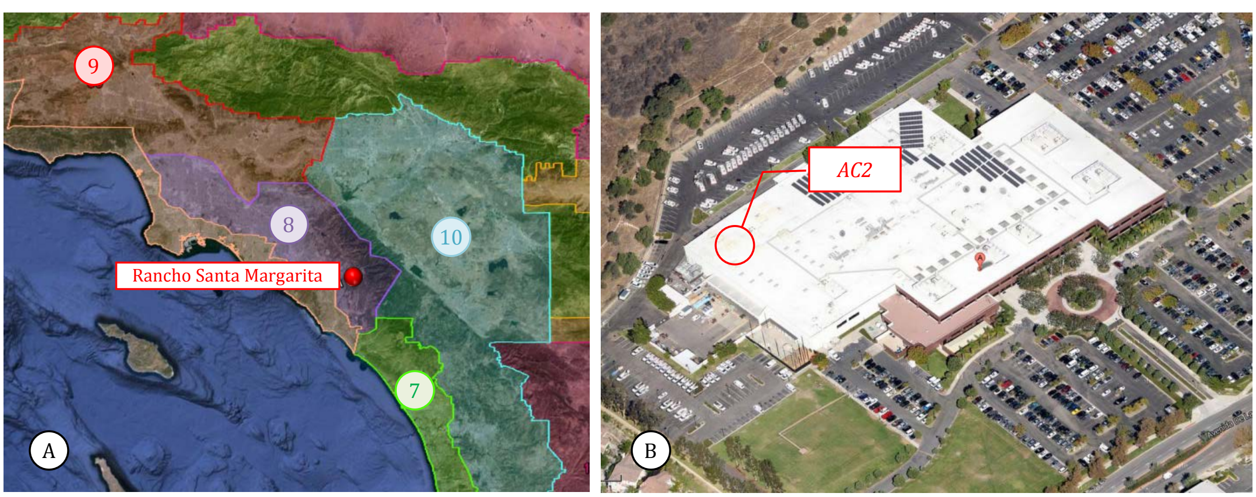


Figure 2: Field Test Location in California Climate Zone 08

The facility selected for the evaluation is a small data center with consistent internal cooling loads that are almost independent from ambient conditions. The building is pictured in Figure 2 B, and the location of the unit tested is indicated. The existing data center had two 17.5 ton cooling-only packaged rooftop air conditioners (Trane TDC210F400AA circa 2011) designed to provide (N+1) redundancy. Only one rooftop unit operates at a time, and the two are managed to provide cooling on alternate weeks.

Neither rooftop unit is intended to provide outside air, and although there are many hours in Rancho Santa Margarita when outside temperature is appropriate for economizer cooling, the units do not include outside air dampers or economizer controls. Incidentally, it should be noted that *2013 California Building Energy Efficiency Standards* now require economizers for computer rooms in all new buildings. Also, as described in section *Airflow Measurements*, although these rooftop units are not designed to provide ventilation air, measurements indicate that a substantial amount of outside air was introduced from an unidentified source.

The evaporative condenser air pre-cooler was added to one of the two rooftop units, and both units were monitored by the research team. The system installed is pictured in Figure 3 and Figure 4.

FIGURE 3: PHOTOS OF EVAPORATIVE PRE-COOLER INSTALLED ON ROOFTOP UNIT

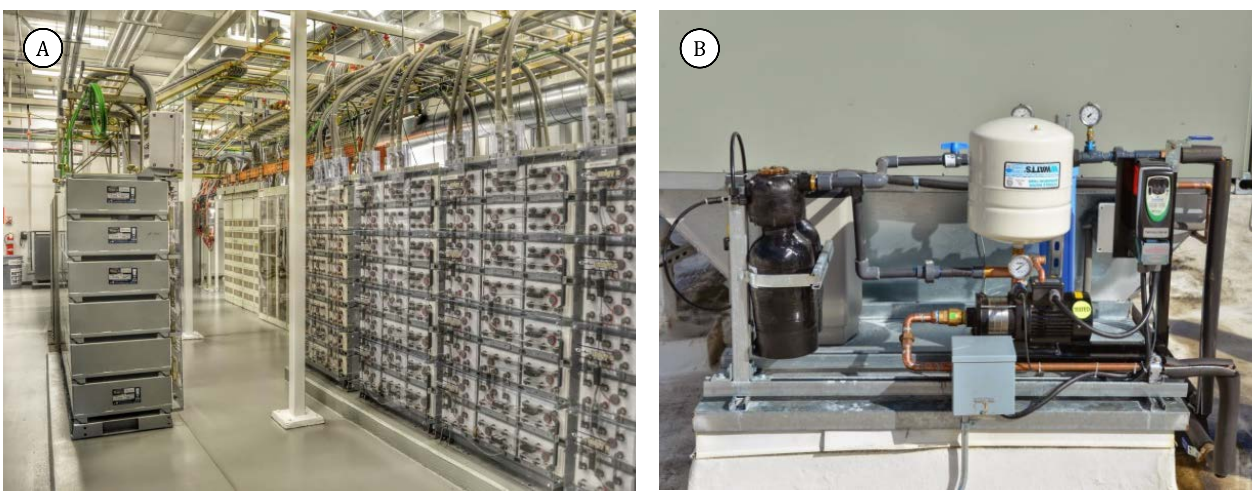


FIGURE 4: PHOTOS OF (A) DATA CENTER COOLED BY UNIT (B) WATER TREATMENT SYSTEM FOR EVAPORATIVE PRE-COOLER

ASSESSMENT OBJECTIVES

The primary objective of this project was to assess the efficiency improvement and changes in characteristic performance for a package rooftop air conditioner with the evaporative condenser air pre-cooler. The study measured all of the energetic inputs and outputs to the rooftop unit with and without the pre-cooler in operation in order to capture all impacts that could be attributed to the measure.

In addition to description of the overall performance impacts for the unit, a secondary objective of the study was to characterize performance of the evaporative pre-cooler component. This is presented in terms of wet-bulb effectiveness and water consumption.

Lastly, since the savings measured for this installation was negligible, the assessment also endeavored to explain whether the lack of savings was mainly due to poor performance for the condenser pre-cooler, or if the evaporative pre-cooler performed well, but other characteristics about the rooftop unit limited savings.

ASSESSMENT METHODOLOGY

OVERVIEW OF THE TECHNICAL APPROACH

Initially, the research team measured performance for two similar rooftop units that serve cooling for the same data center space. These units were designed as redundant systems and were controlled so that only one was enabled at a time. The condenser pre-cooler was added to one of the air conditioners, and the building control system was used to switch between each of the two units in one-to-two week intervals.

Part way through the study, data observations indicated that there was an unknown issue with the baseline unit that caused unexpectedly low return air and supply air temperatures. On the ground investigation observed a disconnect in the roof curb that allowed short circuit of air from the supply plenum to the return plenum. At this point, the assessment method changed course. Once adequate performance data had been captured for operation with the condenser pre-cooler, the measure was removed and performance data was collected from the same rooftop unit for several more weeks. Therefore, all data presented for this study is from measurement of the single unit with and without the condenser pre-cooler.

The condenser pre-cooler system tested in this study was installed in November 2013. The measure was commissioned by the installing contractor at that time, but several operating concerns persisted until the end of July, when the manufacturer, installing contractor, and research team confirmed that the evaporative pre-cooling system was operating as intended. UC Davis installed a thorough array of instrumentation on the two rooftop units in March 2014, and monitoring persisted on both units from this time until December 2014.

TABLE 1: TIMELINE OF EVENTS

Date	Event
10/26/2013	Condenser pre-cooler installation and startup.
12/06/2013	Commissioning complete.
3/19/2014	Instrumentation installation and begin monitoring.
5/1/2014	Research team observes excessive mineralization and metal corrosion. Issues attributed to failure of water softener.
5/13/2014	Contractor repairs water softener, and measurement confirms that it is functioning correctly. Contractor observes that the pre-cooler media is impacted with mineral deposits because of prolonged operation without water treatment.
5/15/2014	Manufacturer and research team agree on action items before study will continue: (1) replace media (2) check booster pump pressure and bladder tank (3) check water softener to ensure proper performance.
6/6/2014	Data indicates that the evaporative pre-cooler system stops spraying water on this day.
6/24/2014	Contractor replaces brine tank for water softener, replaces media for the pre-cooler, tests water softener and pre-cooler to confirm proper operation, and returns all systems to operation.
7/18/2014	Observed short circuit from supply plenum to return plenum in unit without pre-cooler through leak in roof curb, decided to remove evaporative pre-cooler to establish baseline from the same unit that was used for retrofit.
7/25/2014	Contractor observes that the pre-cooler is not spraying water. Issue is attributed to failed controller output. Repairs are made. Observation of measure performance begins at this point.
9/9/2014	Research team shuts down pre-cooler and removes media. Observation of baseline performance begins at this point.

Analog and digital measurements from the rooftop unit were collected by a data acquisition module located on board the unit. Data was collected over the course of the study, with only minor gaps during maintenance periods for the equipment or data acquisition equipment. The minute interval data was stored on board the data acquisition module for 24 hours, then automatically uploaded over the 2G cellular network to an SFTP server hosted by UC Davis. Data for each unit was 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 multi-week time series data sets that correspond with the baseline and retrofit study periods. Data for the unit studied was divided into two periods: July 26 2014 – September 9 2014, and September 10 2014 – October 12 2014. These two data sets were used as the focus for analysis and visualization. Unless otherwise noted, the data presented in this report represents every minute of operation in each of these two periods.

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. The monitoring scheme utilized for the study is illustrated schematically in Figure 5. Table 2 provides a simple description of each measurement marked in the instrumentation schematic, and documents the performance specifications for the sensors utilized for each corresponding measurement.

Current transducers listed in the monitoring plan are used mainly for sensing component operations to determine system mode. The amperage, line voltage, and power factor for the complete system is also recorded to accurately determine the total power draw for each minute of operation. All temperature and humidity measurements are made at single-points, and not as space averages; however multiple temperature measurements from different locations at the condenser inlet and condenser exhaust were recorded separately, then averaged as a part of data analysis.   Refrigerant circuit temperature measurements were made with surface mounted thermistors, adhered tightly with foil tape and wrapped with 1” neoprene insulation. Refrigerant circuit pressures were measured with Schrader valve pressure transducers located on service ports for the suction, discharge, and liquid lines of each vapor compression circuit.

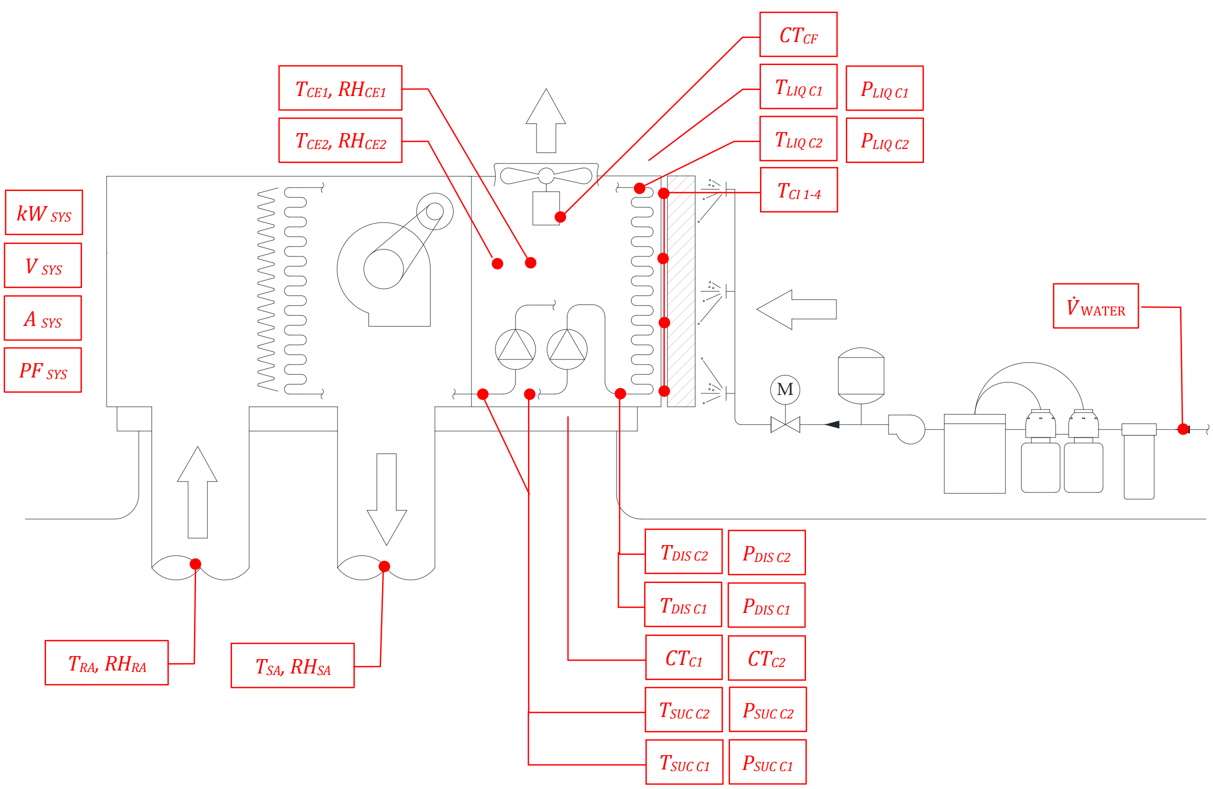


FIGURE 5: INSTRUMENTATION SCHEMATIC

Table 2: Instrumentation Schedule

Name	*Measurement*	*Sensor*	*Uncertainty*
T_{OSA}	Outside Air Temperature (°F)	Vaisala HUMICAP HMP110	± 0.36 °F
RH_{OSA}	Outside Air Relative Humidity (%)	Vaisala HUMICAP HMP110	± 1.7% RH
T_{RA}	Return Air Temperature (°F)	Vaisala HUMICAP HMP110	± 0.36 °F
RH_{RA}	Return Air Relative Humidity (%)	Vaisala HUMICAP HMP110	± 1.7% RH
T_{SA}	Supply Air Temperature (°F)	Vaisala HUMICAP HMP110	± 0.36 °F
RH_{SA}	Supply Air Relative Humidity (%)	Vaisala HUMICAP HMP110	± 1.7% RH
$T_{CA\,1}$	Condenser Outlet Temperature (loc.1) (°F)	Vaisala HUMICAP HMP110	± 0.36 °F
$RH_{CA\,1}$	Condenser Outlet Relative Humidity (loc.1) (°F)	Vaisala HUMICAP HMP110	± 1.7% RH
$T_{CA\,2}$	Condenser Outlet Temperature (loc.2) (°F)	Vaisala HUMICAP HMP110	± 0.36 °F
RH_{CA2}	Condenser Outlet Relative Humidity (loc.2) (°F)	Vaisala HUMICAP HMP110	± 1.7% RH
$T_{CI\,1}$	Condenser Inlet Temperature (loc.1) (°F)	Omega TH-44000	± 0.36 °F
$T_{CI\,2}$	Condenser Inlet Temperature (loc.2) (°F)	Omega TH-44000	± 0.36 °F
$T_{CI\,3}$	Condenser Inlet Temperature (loc.3) (°F)	Omega TH-44000	± 0.36 °F
$T_{CI\,4}$	Condenser Inlet Temperature (loc.4) (°F)	Omega TH-44000	± 0.36 °F
$\dot{V}_{WATER}$	Water Consumption Rate (gal/min)	OMEGA FTB 4605	± 1.5%
CT_{C1}	Compressor 1 Current (Amps)	NK AT1-005-000-SP	± 0.1 A
CT_{C2}	Compressor 2 Current (Amps)	NK AT1-005-000-SP	± 0.1 A
CT_{CF}	Condenser Fan(s) Current (Amps)	NK AT1-005-000-SP	± 0.1 A
kW_{SYSTEM}	System Power (kW, 3Ø)	Dent Powerscout 3	± 1%
A_{SYSTEM}	System Current (Amps, 3Ø)	Dent Powerscout 3	± 1%
V_{SYSTEM}	System Voltage (Volts, 3Ø)	Dent Powerscout 3	± 1%
PF_{SYSTEM}	System Power Factor (–, 3Ø)	Dent Powerscout 3	± 1%
$T_{SUCTION\,C1}$	Compressor 1 Suction Temperature (°F)	Omega TH-44000	± 0.36 °F
$T_{SUCTION\,C2}$	Compressor 2 Suction Temperature (°F)	Omega TH-44000	± 0.36 °F
$T_{DISCHARGE\,C1}$	Compressor 1 Discharge Temperature (°F)	Omega TH-44000	± 0.36 °F
$T_{DISCHARGE\,C2}$	Compressor 2 Discharge Temperature (°F)	Omega TH-44000	± 0.36 °F
$T_{LIQUID\,C1}$	Compressor 1 Liquid Temperature (°F)	Omega TH-44000	± 0.36 °F
$T_{LIQUID\,C2}$	Compressor 2 Liquid Temperature (°F)	Omega TH-44000	± 0.36 °F
$P_{SUCTION\,C1}$	Compressor 1 Suction Pressure (bar-g)	ClimaCheck, 35 bar	± 0.35 bar
$P_{SUCTION\,C2}$	Compressor 2 Suction Pressure (bar-g)	ClimaCheck, 35 bar	± 0.35 bar
$P_{DISCHARGE\,C1}$	Compressor 1 Discharge Pressure (bar-g)	ClimaCheck, 50 bar	± 0.50 bar
$P_{DISCHARGE\,C2}$	Compressor 2 Discharge Pressure (bar-g)	ClimaCheck, 50 bar	± 0.50 bar
$P_{LIQUID\,C1}$	Compressor 1 Liquid Pressure (bar-g)	ClimaCheck, 50 bar	± 0.50 bar
$P_{LIQUID\,C2}$	Compressor 2 Liquid Pressure (bar-g)	ClimaCheck, 50 bar	± 0.50 bar

AIRFLOW MEASUREMENTS

The supply airflow rate for this constant speed rooftop unit was 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:

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

This method has many advantages compared to conventional air balance techniques, the most significant of which is accuracy. The tracer gas airflow tool used can operate with a calculated uncertainty of less than ±2%.

A similar method was used to measure outside air fraction for the system. Although the unit measured does not have outside air dampers and is not designed to provide ventilation air, the cabinet does leak. Outside air fraction measurements are used as part of the calculations for system cooling capacity, since the difference between return air temperature and supply air temperature alone would slightly misrepresent the cooling capacity generated by the system. CO_2 concentration was measured in the outside air and return air streams, then 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 6 charts the airflow and outside air fraction measurements for the unit. These measurements were made once with the condenser pre-cooler in place. It is assumed that the airflow and outside air fraction are not impacted by the presence of the condenser pre-cooler, therefore the same airflow rates were used for the baseline period. For some instances during the baseline period it appears that supply airflow was being restricted – very possibly by an accumulation of ice on the evaporator coil during cool morning hours. These instances are omitted from all results that use airflow as an input.

The results for outside air fraction are surprising. The unit was not equipped with outside air dampers that could leak, and there were not visible holes, or obvious noise that would indicate the presence of such a substantial leakage flow into the cabinet. The measurement provides no explanation for where this leakage might be from. It is possible that the measurement is erroneous, but the surprising result is not adequate reason to disregard the measurement. Importantly, the conclusions reached about the technology tested in this study would not be different if the real outside air fraction were different from what was measured.

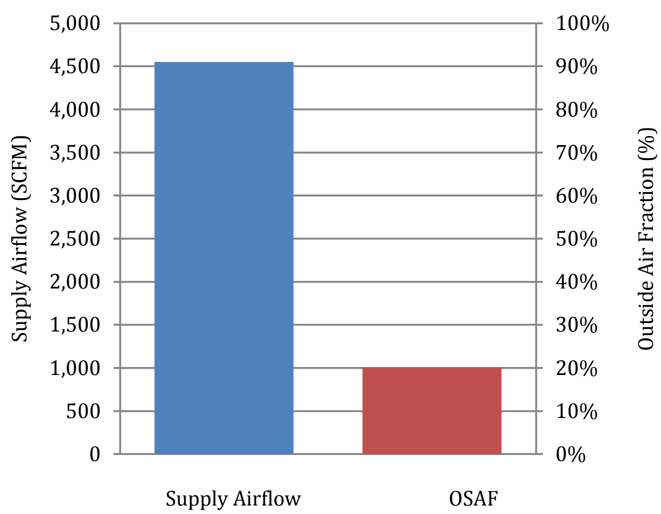


FIGURE 6: SUPPLY AIRFLOW RATE AND OUTSIDE AIR FRACTION MEASURED FOR UNIT

DEFINITION & CALCULATION OF PERFORMANCE METRICS

CALCULATING COOLING CAPACITY

The system cooling capacity for the rooftop unit is determined at any operating condition according to the supply airflow 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 what is lost due to fan heat.

$$\dot{H}_{system} = \dot{m}_{SA} \cdot \left(h_{MA} - h_{SA}\right) \tag{3}$$

In this equation, h_{MA} is the specific enthalpy of the mixed air. This unit does not have outside air dampers, but the cabinet does leak, so the mixed air condition is calculated as the apparent mixture of return air conditions and outside air conditions. Equation 4 calculates the specific enthalpy for the mixed air condition.

$$h_{MA} = \mathrm{OSAF} \cdot h_{OSA} + (1 - \mathrm{OSAF}) \cdot h_{RA} \tag{4}$$

The assessment presented here focuses on the system's ability to produce sensible cooling, and discounts the value of any dehumidification. There is no significant source of humidity in the data center cooled by this equipment. Since the thermostat controls only respond to temperature and do not control for humidity , it is not appropriate to credit any latent cooling when considering the performance of the machine with and without the evaporative condenser air pre-cooler. The net sensible system cooling capacity is determined according to Equation 5:

$$\dot{H}_{system}^{sensible} = \dot{m}_{SA} \cdot C_p \cdot \left(T_{MA} - T_{SA}\right) \tag{5}$$

Concomitantly, the latent system cooling is determined as:

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

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:

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

Analysis in this report focuses on the sensible cooling generated by the equipment, and ignores any cooling capacity associated with dehumidification. Efficiency is expressed as the sensible system coefficient of performance, which is calculated by the following equation:

$$COP_{system}^{sensible} = \frac{\dot{H}_{system}^{sensible}}{\dot{E}_{system}} \tag{8}$$

Analysis of Wet-Bulb Effectiveness

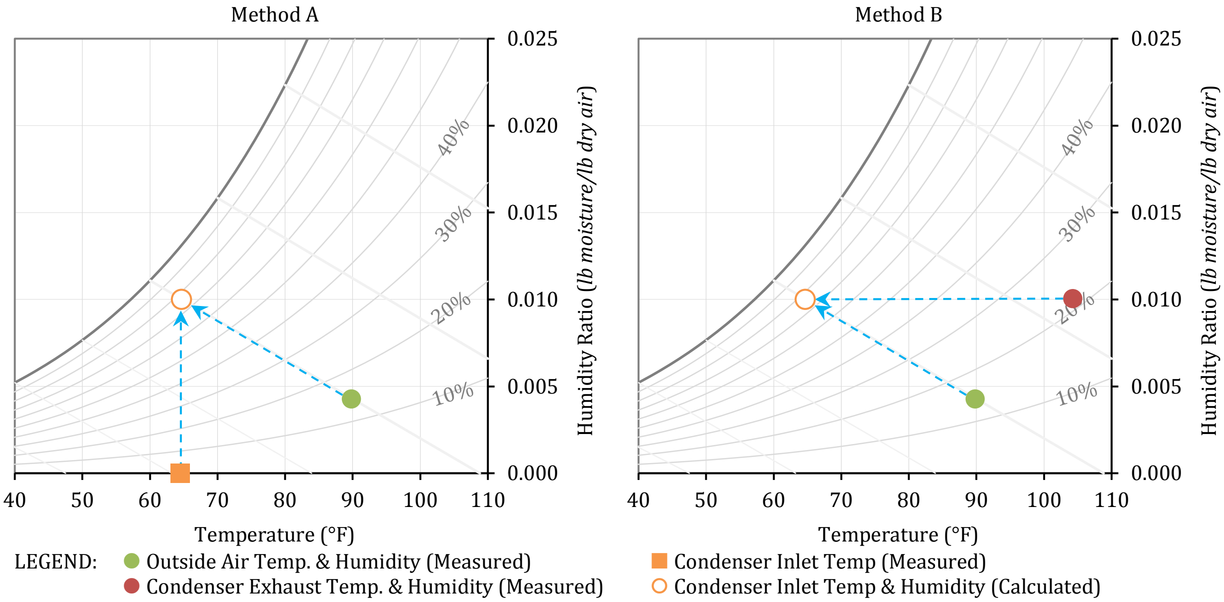


Figure 7: Psychrometric Charts Explaining Alternate Methods for Wet Bulb Effectiveness Calculations

The performance of evaporative cooling systems is most commonly described by the wet bulb effectiveness. This metric represents the degree to which air flowing through a system is cooled toward the wet bulb temperature. It is calculated according to Equation 9, as the ratio of the change in dry-bulb temperature to the wet-bulb depression. For a condenser pre-cooler, the rated wet bulb effectiveness can be used to predict the air temperature at the inlet of the condenser coil, which subsequently allows for estimation of the energy savings that will be achieved.

$$\text{WBE} = \frac{T_{OSA} - T_{CI}}{T_{OSA} - T_{wb\ OSA}} \qquad 9$$

While wet bulb effectiveness can be measured for an evaporative component under controlled conditions in the laboratory, it can be difficult to ascertain from direct measurement for condenser pre-coolers in the field. There is very little space between a condenser pre-cooler and the condenser coil. Physical temperature measurement in this space are subjected to radiation effects from the condenser, as well as potential carry over effects from the evaporative pre-cooler. Moreover, airflow and evaporation are not equally distributed across the face of an evaporative pre-cooler, so temperature sensors placed in this area are subject to measurement location bias.

This study uses two separate methods to estimate wet bulb effectiveness. The first method (illustrated as "*Method A*" in Figure 7) uses direct measurement of the condenser air inlet temperature from four separate thermistors (indicated in Figure 5) to calculate wet bulb effectiveness according to Equation 10.

$$\text{WBE}_A = \frac{T_{OSA} - \frac{\sum_{i=1}^{N}\left(T_{CI_i}\right)}{N}}{T_{OSA} - T_{wb\ OSA}} \qquad 10$$

The second method (illustrated as "*Method B*" in Figure 7) uses measurement of the temperature and humidity in the condenser exhaust to infer what the condenser inlet temperature must have been. This method relies on two assumptions: (1) change in psychrometric conditions across the evaporative pre-cooler is an adiabatic sensible-latent exchange, and (2) temperature rise across the condenser is sensible only. Basically, this method assumes that the condenser inlet has the same specific enthalpy as outdoor air, and the same absolute humidity as condenser exhaust air. As indicated in Figure 5, temperature and humidity in the condenser exhaust were measured at two locations and averaged. Similar to the methodological uncertainties associated with "*Method A*", this approach is also subject to some measurement location bias.

RESULTS & DISCUSSION

For the scope of this project, a combination of MATLAB and Excel were used to conduct calculations and analysis. First, minute interval data from the unit evaluated was concatenated into two separate groups for a 'baseline' data set (September 10 2014 – October 12 2014) and a 'retrofit' data set (July 26 2014 – September 9 2014). Then, every minute in each of these data sets was assigned a value of "enabled" or "disabled" to identify whether or not the unit evaluated had been selected as the lead unit by the building Energy Management and Control System. Afterward, a Boolean algorithm was used to identify the mode of operation in each minute so that performance data could be separated and considered accordingly. Finally, capacity, coefficient of performance, and wet bulb effectiveness were calculated for each minute in both data sets.

Analysis explored the characteristic performance of the unit with and without the condenser pre-cooler. Figures in this section present the results of this comparison, interpretive explanation is given for the behaviors observed, and the extended technical implication of these observations is discussed.

Unless otherwise noted, the figures presented in this section include all one minute data points from each set. However, in the 'retrofit' case there were periods in which the unit appeared to be operating with a frozen evaporator coil. It seems that the supply airflow rate was reduced during these periods in a way that could not be accounted for analytically, which subsequently skewed the calculated results for capacity and coefficient of performance. In order to avoid misrepresentation of system performance, these instances were identified then filtered from the calculated results. Thus, plots for capacity and coefficient of performance do not include the periods with reduced airflow. Charts that show direct measurements such as supply air temperature or system power draw still include these periods.

SUMMARY OF KEY RESULTS

The following list summarizes key results and observations presented in this section. Table 3 records specific quantitative values measured with and without the evaporative condenser air pre-cooler.

- Similar environmental conditions were encountered during the pre and post retrofit periods.
- Hourly energy consumption is not noticeably improved by the pre-cooler.
- Hourly energy consumption increases somewhat with outside air temperature.
- Coefficient of performance is not noticeably improved by the pre-cooler.
- Cooling capacity is not noticeably improved by the pre-cooler.
- The pre-cooler does not reduce system power draw.
- The characteristic power draw for this system does not increase with outside air temperature as much as is typically expected. Between 75-100°F power draw only increases by 3%.
- Return air temperatures are 2-3°F warmer during the retrofit periods.
- Wet bulb effectiveness is low.
- Refrigerant temperatures indicate that the pre-cooler may have a small effect; discharge temperature is about 3°F lower for the retrofit period.
- The amount of water consumed at peak corresponds to a 3°F cooling effect for the condenser airflow.

TABLE 3: SUMMARY OF KEY RESULTS

Measurement @ T_{DB}=95°F, T_{WB}=65°F	*With Pre-Cooler*	*Without Pre-Cooler*
Energy Use (kWh)	20.03	20.04
Sensible System COP (—)	2.79	2.8
Sensible System Cooling Capacity (kbtu/hr)	192.4	192.77
System Power Draw (kW)	20.13	20.18
Supply Air Temperature (°F)	48.03	47.54
Liquid Temperature (°F) (*Circuit 2*)	92.12	96.91
Wet Bulb Effectiveness (%) (*Avg of Method A & B*)	12 %	NA
Water Consumption (gal/hr)	7.53	NA

DISTRIBUTION OF CONDITIONS ENCOUNTERED

The number of operating hours spent in each mode is charted in Figure 8 as a histogram to show the distribution of runtime across the range of outside air conditions encountered. Overall, the conditions encountered are similar but not identical. There are more operating hours in the baseline data set, and more instances of first stage compressor mode in the retrofit data set. These observations should not lead to the conclusion that the condenser pre-cooler reduced compressor runtime because there are other confounding variables that should be considered, namely:

- The evaporator coil appears to have frozen during many low temperature periods in the baseline data set, which reduced capacity for second stage cooling, and would have reduced the fraction of time spent in first stage cooling.
- There are more total hours in the baseline data set, and so more time spent cooling.

The most valuable observations to be drawn from these charts are that:

- The system does not deliver adequate cooling capacity to satisfy the load. Cooling operation is continuous regardless of outside temperature, and for all instances above 70°F the system runs flat out in second stage. System temperatures corroborate the fact that continuous full capacity operation is almost never able to satisfy the set point. This may be because the rooftop unit is undersized for the application. It was also observed that duct sizes and configurations in the facility were not adequate for the design flow rates.
- The baseline and retrofit sets include adequate data across a similar range of outside air conditions to allow for comparison of the characteristic performance for each data set.

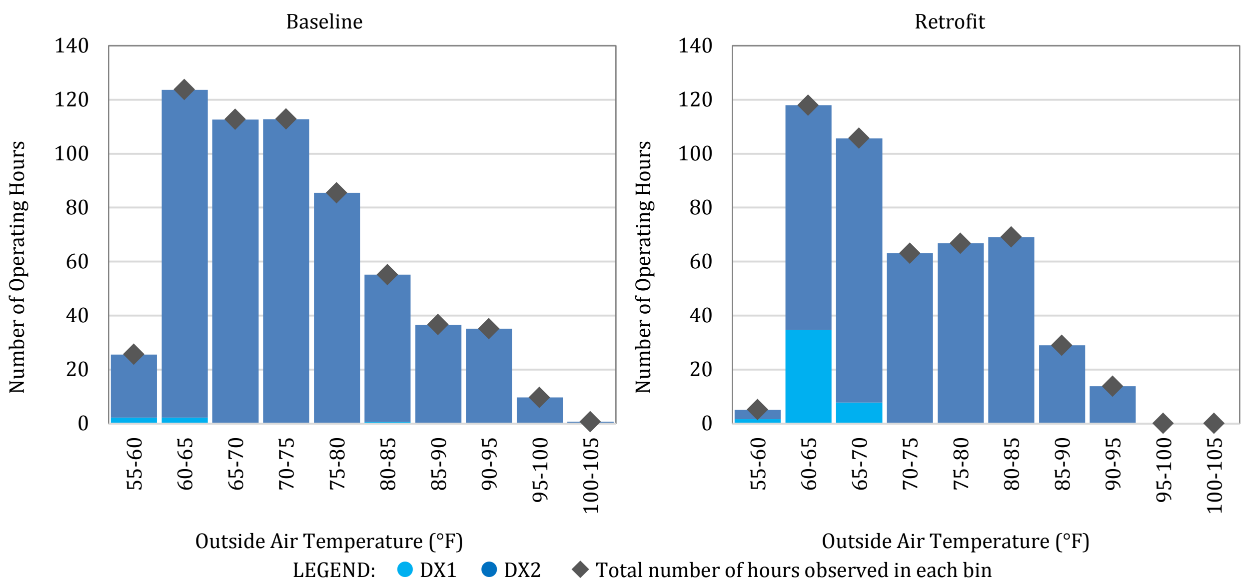


FIGURE 8: DISTRIBUTION OF TIME SPENT IN EACH MODE AS A FUNCTION OF OUTSIDE AIR CONDITIONS DURING EACH TEST PERIOD

COMPARISON OF ENERGY USE WITH AND WITHOUT THE PRE-COOLER

Figure 9 presents the so called 'energy signature' (NBI 2009) for the unit studied with and without the evaporative pre-cooler. These plots record the total energy consumption in each hour as a function of the corresponding hourly average outside air temperature. The energy use signatures for the baseline and retrofit cases are very similar. In fact, for those periods with second stage compressor operation, there is no statistically significant difference in the results. The pre-cooler did not reduce hourly energy consumption.

There are a number of instances between 60–70°F where hourly energy consumption for the retrofit case is lower than the baseline case. However, this effect should not necessarily be attributed to the pre-cooler because there are other confounding factors. Most importantly, in the baseline case, it appears likely that airflow and cooling capacity for second stage was often reduced by moisture freezing on the evaporator coil. These circumstances appear to have eliminated all opportunities for first stage operation.

Aside from the fact that there is no significant difference between the baseline and retrofit case, it should be noticed that the hourly energy consumption for the system only increases a little as outside temperature increases. For most applications the hourly energy use for cooling changes significantly with outside air temperature since the cooling load, sensible cooling capacity, and room cooling efficiency all tend to increase energy use as outside temperature increases. Most applications will have almost zero cooling energy use below about 65°F.   For data centers, room cooling requirements generally persist regardless of the outside temperature, and so runtime remains much more consistent. However efficiency and capacity still tend to decrease as outside temperature increases, so electric energy used for cooling a data center should still increase as outside temperature increases.

One should also note that although the energy use signature plainly shows that the condenser pre-cooler studied did not reduce hourly energy consumption in this circumstance, it does not provide a conclusive assessment of the potential impacts in other applications. For example, condenser pre-cooling should reduce instantaneous power draw and simultaneously increase cooling capacity; both factors tend to increase efficiency. Since the rooftop unit in this data center appears to hav been undersized for the load, any increase in cooling capacity would be absorbed by the room and the equipment would still run continuously. In this case, the energy use signature would mask the value of increased cooling capacity.

Future projects to apply condenser pre-cooling should be cognizant of the technology's parallel impact on capacity, and instantaneous power draw. In many instances, a portion of the efficiency advantage may result in increased cooling, which will reduce the degree to which theoretical energy savings potential is achieved. This data center would certainly be one such example.

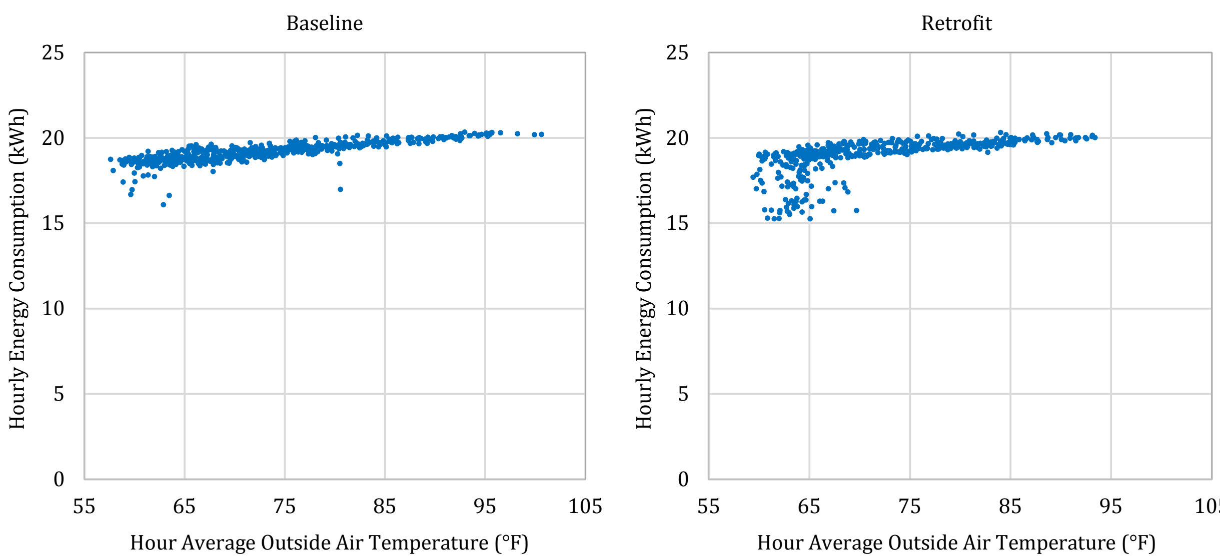


FIGURE 9: ENERGY USE SIGNATURE FOR THE UNIT WITH AND WITHOUT CONDENSER PRE-COOLER

COOLING CAPACITY

Figure 10 presents the sensible system cooling capacity for the unit with and without the evaporative condenser pre-cooler. Similar to later plots results for the baseline period omit those instances when it appears that the coil is frozen, during which time it is presumed that supply airflow is reduced.

There is no noticeable difference in cooling capacity between the baseline and retrofit period. The retrofit period has more instances of first stage operation between 60–70°F, but again, this cannot be attributed to the condenser pre-cooler.

Despite traditional assumptions about performance trends for air conditioners, sensible system cooling capacity increases somewhat as outside air temperature increases. This occurs in both the baseline and retrofit case. The behavior can be attributed to two main factors:

- Return air temperature increases in this building as outside temperature increases.
- The unit was observed to operates with a significant amount of outside air.

These factors both tend to increase the evaporator coil entering air temperature when outside temperature increases, which can result in an increase for sensible cooling capacity. Also, the sensible cooling metric does not credit latent cooling, as described by Equation 5. Since the sensible heat ratio decreases as coil entering air temperature decreases, a larger portion of the cooling capacity results in latent cooling at lower temperatures.

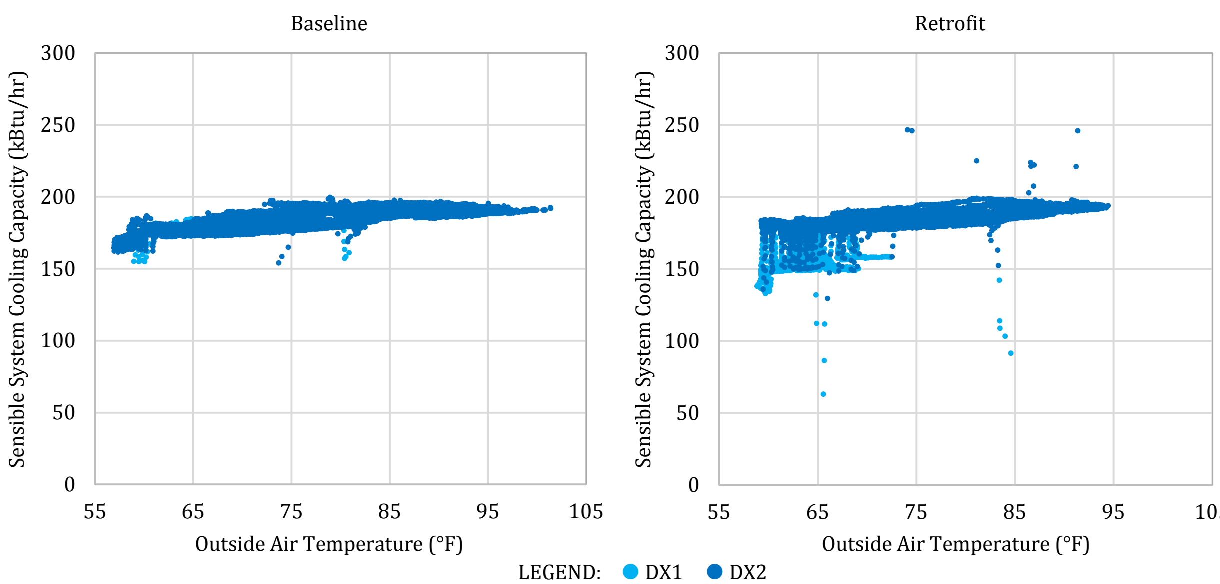


FIGURE 10: SENSIBLE SYSTEM COOLING CAPACITY FOR THE UNIT WITH AND WITHOUT CONDENSER PRE-COOLER

SYSTEM POWER DRAW

Figure 11 presents electric power draw for the system in each mode of operation, as a function of outside air temperature for the baseline and retrofit periods. This characteristic performance map is different than the 'energy use signature' presented earlier. This figure shows the instantaneous power draw that can be expected for the system for any given mode and condition, whereas the energy use signature showed the total amount of electricity consumed in each hour of operation.

Addition of the evaporative pre-cooler has no effect on electric power draw characteristics for either mode.

Aside from the fact that the pre-cooler appears to have no impact, it should be noticed that the characteristic power draw does not increase very much as outside temperature increases. For a temperature increase from 55–100°F power only increases by 15%, and for an increase from 75–100°F power only increases by 3%.

This observation is somewhat troubling, it suggests that no pre-cooler would result in substantial energy savings for this machine, regardless of how well it might perform.. At peak conditions a pre-cooler with 100% wet bulb effectiveness would only reduce power draw by 3%. This observation is contrary to general expectations about the potential impact for condenser pre-coolers. By comparison, recent laboratory tests for a standard efficiency rooftop unit have recorded 40% increase in power for an increase in outside temperature from 55–100°F, and a 24% increase for an increase from 75–100°F (Pistochini 2014).

At least in part, this surprising characteristic can be attributed to the fact that this rooftop unit appears to be undersized for the load. Since the unit cannot provide enough cooling to satisfy the set point, the return temperature rises as outside air temperature rises. Figure 13 analyzes the supply and return air temperatures observed. Since compressor power draw is driven mainly by the difference between evaporating temperature and condensing temperature, the fact that return air temperature rises steadily with outside air temperature deteriorates the tendency for power to increase with outside air temperature. For high ambient temperatures, the difference between outside air and evaporator coil entering air temperature is between 30–50% smaller than it would be for laboratory tests that use a fixed coil inlet temperature.

This same tendency may also have implications for equipment that is properly sized since ventilation tends to increase the evaporator coil inlet temperature similarly. Pre-cooler assessments have typically been made for equipment without ventilation air and a steady return air temperature.

There may also be other reasons for the comparably flat power draw characteristic observed for this machine. The phenomenon deserves further consideration in order to ensure that efforts to apply evaporative pre-cooling are able to target the most appropriate applications.

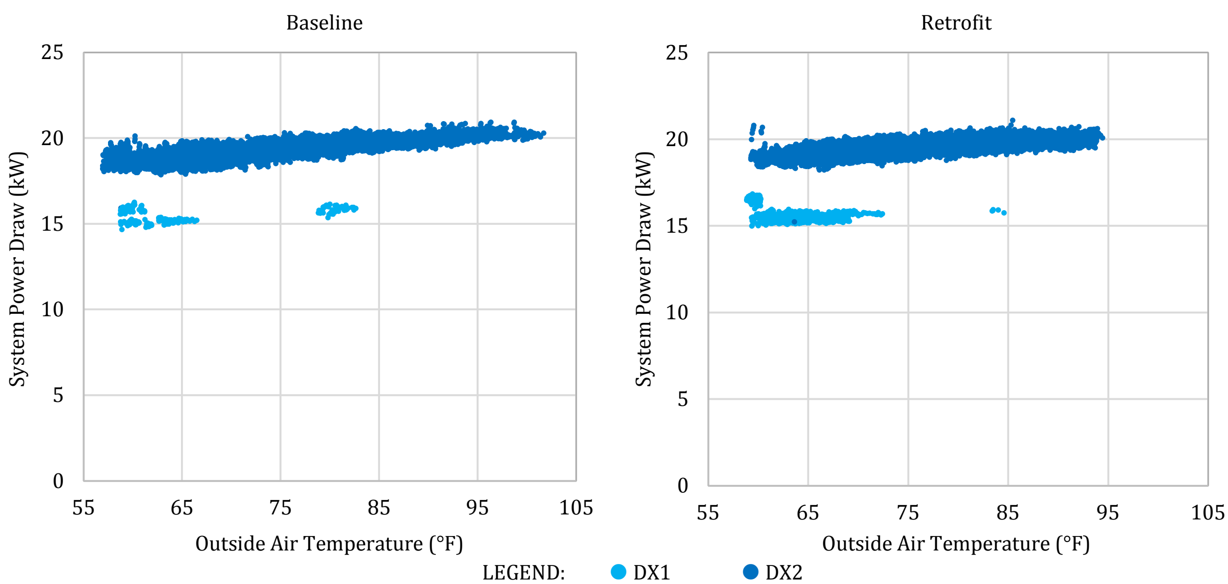


FIGURE 11: SYSTEM POWER DRAW FOR THE UNIT WITH AND WITHOUT CONDENSER PRE-COOLER

COEFFICIENT OF PERFORMANCE

Sensible system coefficient of performance describes the efficiency of the machine. Comparison of the efficiency in the baseline and retrofit periods describes the potential energy savings that could be achieved by the measure, regardless of external confounding factors that might impact the energy use signature presented previously. Figure 12 plots the sensible system coefficient of performance from each period as a function of outside air temeprature. The comparision shows that there is no noticeable change in efficiency that can be attributed to the condenser pre-cooler.

In both cases, there are some instances between 60–70°F associated with first stage cooling where COP is as high as 3.5-4.0. These are real occurrences, however, they arise from the transient thermodynamics associated with switching from second to first stage. For a period of time after this mode change the cooling capacity measured with only one compressor benefits from cooling that was left over from the previous state. A similar effect reduces coefficient of performance for a period when switching from first to second stage. At steady state, the sensible system coefficient of performance is similar for both stages.

In the baseline period, it appears that there were a number of instances when supply airflow rate was reduced as a result of frost accumulation on the evaporator coil. Since the airflow rate was not known accurately during these events, all corresponding intervals were filtered from the results presented in Figure 12. To filter the data, measurements from the retrofit data set were used to train a linear model of supply air temperature as a function of outside air temperature. It was observed that in the retrofit period, almost all supply air temperature measurements were within ± 2.5°F of the linear model. Accordingly, instances in the baseline period were filtered out if measured supply air differed from the linear model by more than ± 2.5°F.

It is important to note that the sensible system coefficient of performance for the the unit does not change substantially across the range of outside air temperature observed. For a recirculation only air conditoner, one would normally expect the sensible system coefficient of performance to decrease as outside air temperature increases.

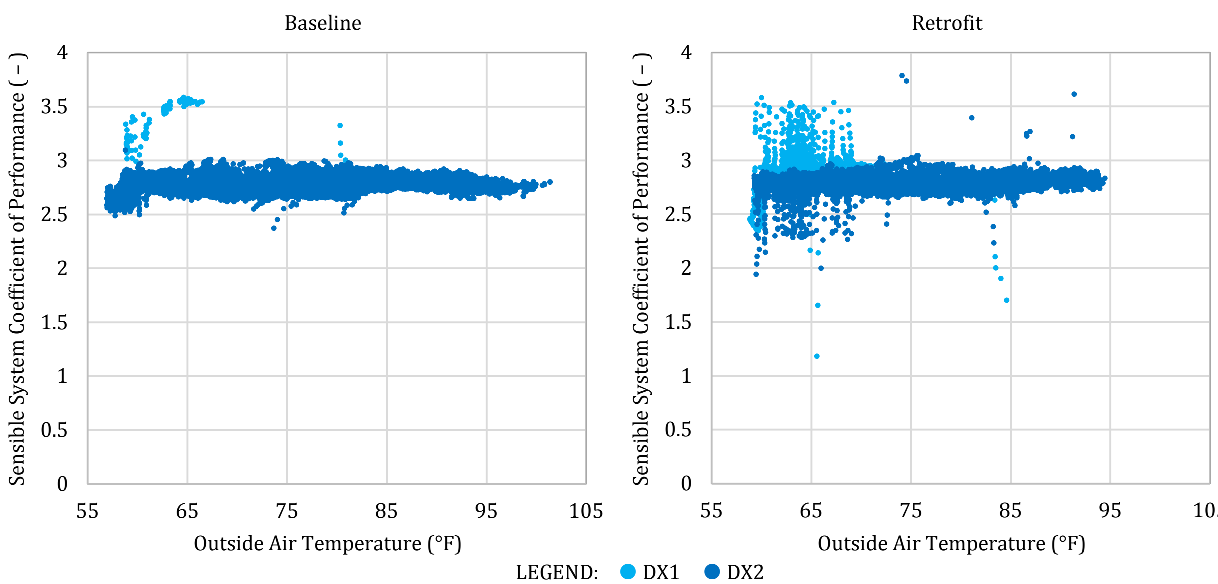


FIGURE 12: COEFFICIENT OF PERFORMANCE FOR SENSIBLE SYSTEM COOLING WITH AND WITHOUT CONDENSE PRE-COOLER

System Air Temperatures

Figure 13 presents supply air temperature and return air temperature from every measured one minute interval with compressor operation during the baseline and retrofit observation periods. Generally, the two periods are very similar and there are no differences that can be attributed to the condenser pre-cooler. However, there are some small differences that should be noted.

First, return air temperature rises as outside temperature rises. This trend indicates that the rooftop unit and duct system does not deliver adequate cooling capacity for the load. As discussed earlier, this rise in return air temperature appears to reduce the tendency for power draw to increase as outside temperature increases, which in turn, reduces the potential value of condenser pre-cooling. Also, return temperature during the retrofit period is consistently higher than in the baseline period. When outside air temperature is 60°F, return temperature was approximately 76°F during the baseline period, compared to 79°F during the retrofit period. The increasing trend was similar for the two periods such that at 95°F outside temperature, return temperature was 80°F in the baseline period and 83°F for the retrofit period. The reason for this difference is not obvious, however, we expect that thermal gains from electronic equipment in the data center may have been somewhat higher during the retrofit period. It doesn't appear that these differences can be attributed to the evaporative condenser pre-cooler, nor does it seem that the difference has a significant impact on performance.

One should also note that in the retrofit period, there are a number of instances for outside air temperature between 60–70°F where supply temperature is several degrees warmer. These correspond to periods of operation with only one compressor. It is not clear why the observation period with higher return air temperature would have more instances of operation with first stage compressor.

Also, in the baseline period there are many hours where supply air temperature was much lower than what was observed in the retrofit period. For some of these occurrences supply temperature was as low as 32°F. This can only occur if the evaporating temperature is so low that ice is forming on the coil. When this happens, it appears that supply airflow declines. These conditions seem topersist for several hours at a time, even after outside air and return air conditions rise to a point that would not typically cause coil freezing. The root cause of this problem was not identified, but for the sake of consistent comparison, instances from these periods were omitted from the results for cooling capacity and coefficient of performance in the baseline period.

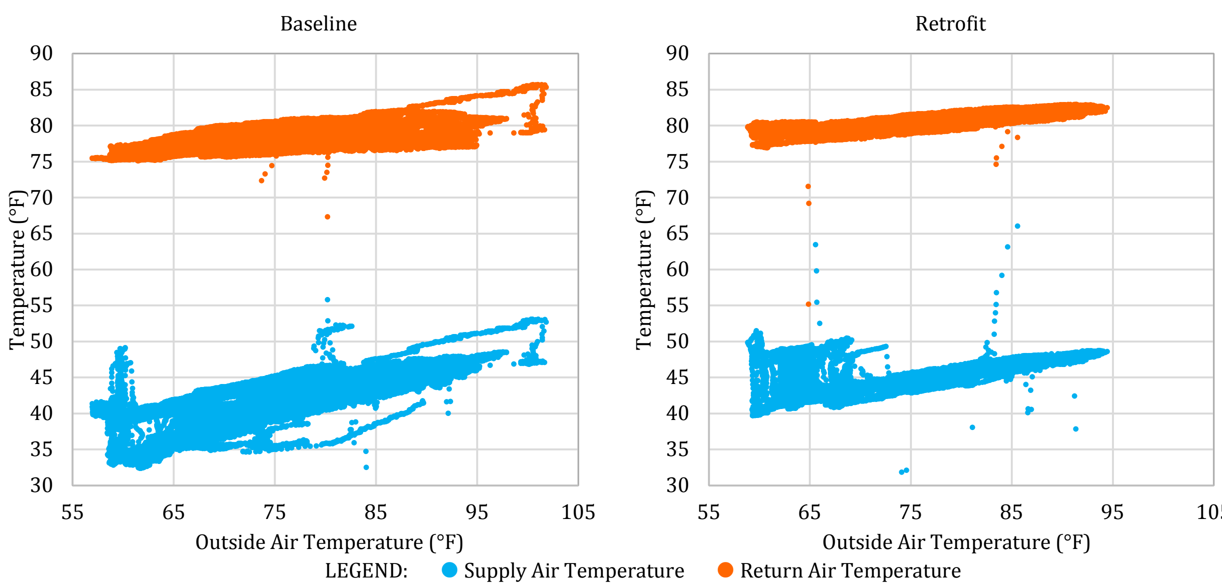


Figure 13: Supply air temperature and return air temperature for AC2 with and without the evaporative condenser pre-cooler

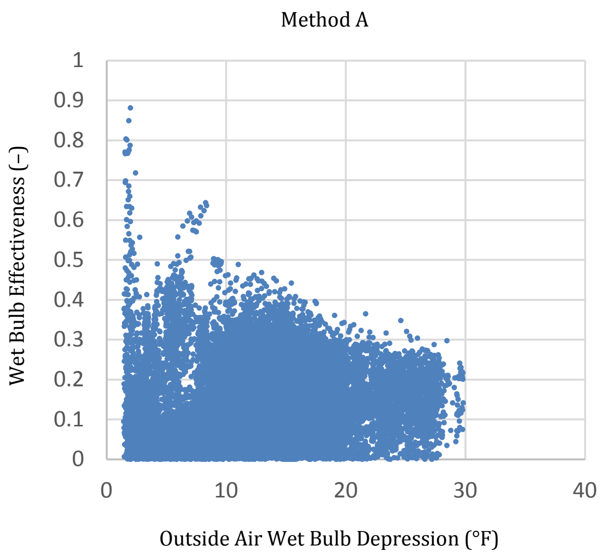
WET BULB EFFECTIVENESS FOR PRE-COOLER

Wet bulb effectiveness for cooling pre cooler is a measure of how near the condenser air inlet temperature, at the outlet of the pre cooler approaches wet bulb for the outside air. As described in the methodology section, the wet bulb effectiveness was calculated by two methods, each of which have their own limitations.

Method A relies on a direct measurement of the air temperature between the evaporative pre-cooler and the condenser. In this case, the space available for measurement is only about two inches deep. This does not allow for adequate mixing, and since the rate of evaporation is certainly not even across the face of the pre-cooler, any single measurement will be an uncertain representation of the average. In this study, the measurement for Method A represents the average of four individual points at the interior of the inlet cross section, but there is still no guarantee that these four points represent a true average.

Method B relies on psychrometric assumptions about how the evaporation and heat transfer processes occur as air passes through the pre-cooler and across the condenser. This method measures temperature and humidity in the condenser exhaust in order to calculate what the condenser inlet temperature must have been. The method assumes that water sprayed on the media is at the wet bulb temperature, which is not exactly true for this technology. However, measurement at this point is more indicative of the average.

Both measurement methods show that the condenser pre-cooler has very little effect on temperature at the condenser inlet. In fact, the difference between outside temperature and condenser inlet temperature is so small that it is often difficult to determine whether the difference should be attributed to the pre-cooler or to exogenous variance in the measurement.



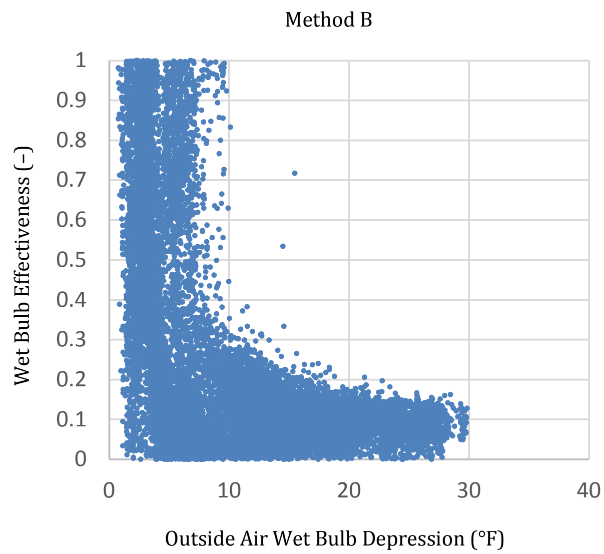


FIGURE 14: WET BULB EFFECTIVENESS FOR THE EVAPORATIVE CONDENSER PRE-COOLER CALCULATED BY TWO METHODS

Figure 14 plots wet bulb effectiveness as a function of wet bulb depression for both methods. For small wet bulb depressions, the variance is so large by comparison that the calculation of wet bulb effectiveness gives a meaningless result. Despite these challenges, the two methods agree well in regions where both can be considered reliable. For larger wet bulb depressions, although there is still large variation, the pre-cooler appears to have a discernable effect on average. At wet bulb depression of 30°F Method A results in a wet bulb effectiveness of 15% on average, while Method B calculates 10%.

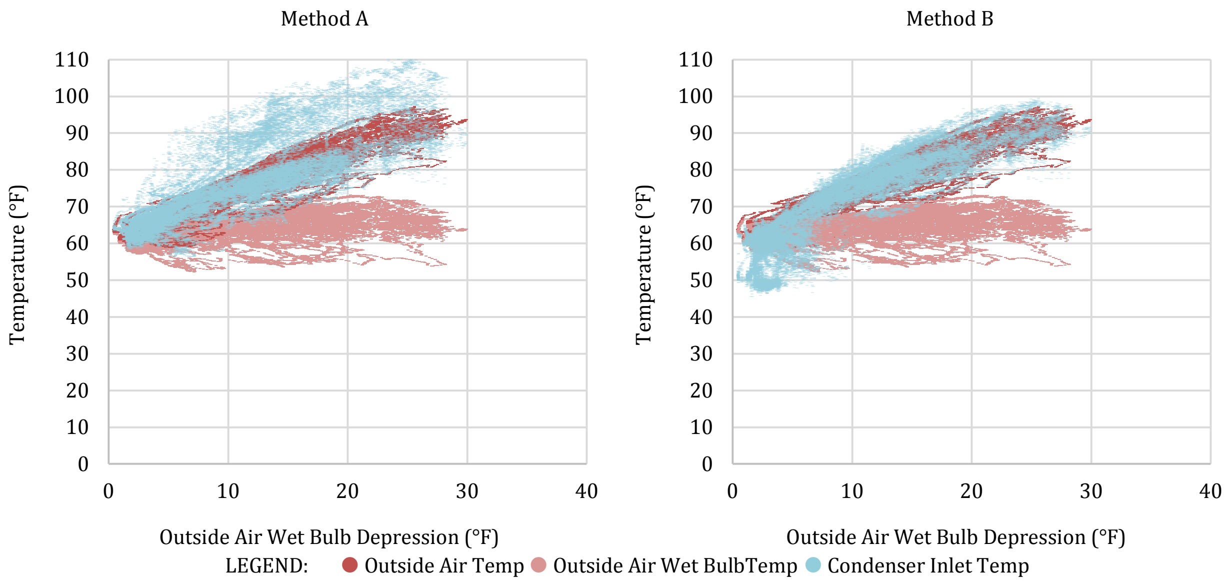


FIGURE 15: CONDENSER INLET TEMPERATURE CALCULATED BY TWO METHODS

Figure 15 presents the dry bulb and wet bulb temperatures measured for the outside air in each instance, along with the corresponding condenser inlet temperature determined by each method. This plot offers a little more clarity than the previous figure since the ratio of small differences can result in exaggerated results. Again, the temperature impact of the condenser pre-cooler is very small and there are many instances with both methods where condenser inlet is actually higher than the outside air temperature. These observations corroborate recent laboratory measurements for the same technology.

REFRIGERANT CIRCUIT TEMPERATURES

One of the main reasons that condenser pre-coolers can improve efficiency is that they tend to reduce refrigerant condensing temperature, which subsequently reduces compressor head and allows for less expansion valve resistance. Figure 16 plots the suction temperature, discharge temperature, and liquid temperature for each vapor compression circuit during the baseline and retrofit periods.

While there is no obvious difference in the trends from each period, careful inspection of the data does indicate that for high ambient conditions suction, discharge and liquid temperatures tend to be 1-3 °F degrees lower in the retrofit period. This does appear to be a result of the condenser pre-cooler, however the difference is small, and doesn't seem to translate into a measureable increase in efficiency for all of the reasons discussed previously.

The variation for refrigerant temperatures in the second circuit at lower ambient conditions occurs as a result of cycling into first stage for some periods. The same type of variation does not occur for circuit one because that compressor almost never shuts off during the period of observation.

For low outside air temperatures during the baseline observation period, these refrigerant measurements corroborate the theory that the evaporator coil was freezing on many occasions – in many instances suction temperature for the first circuit measured at the compressor inlet is as low as 35°F. It is also clear from these observations that only the evaporator coil on the first circuit could have been freezing since the suction temperature for the second circuit never drops below 45°F. These conclusions are corroborated by the fact that suction pressure for the first circuit often operated between 110–115 psia, which corresponds to an evaporating temperature of 28.7–31.4°F. In the post retrofit period, suction pressure never droped below 115 psia.

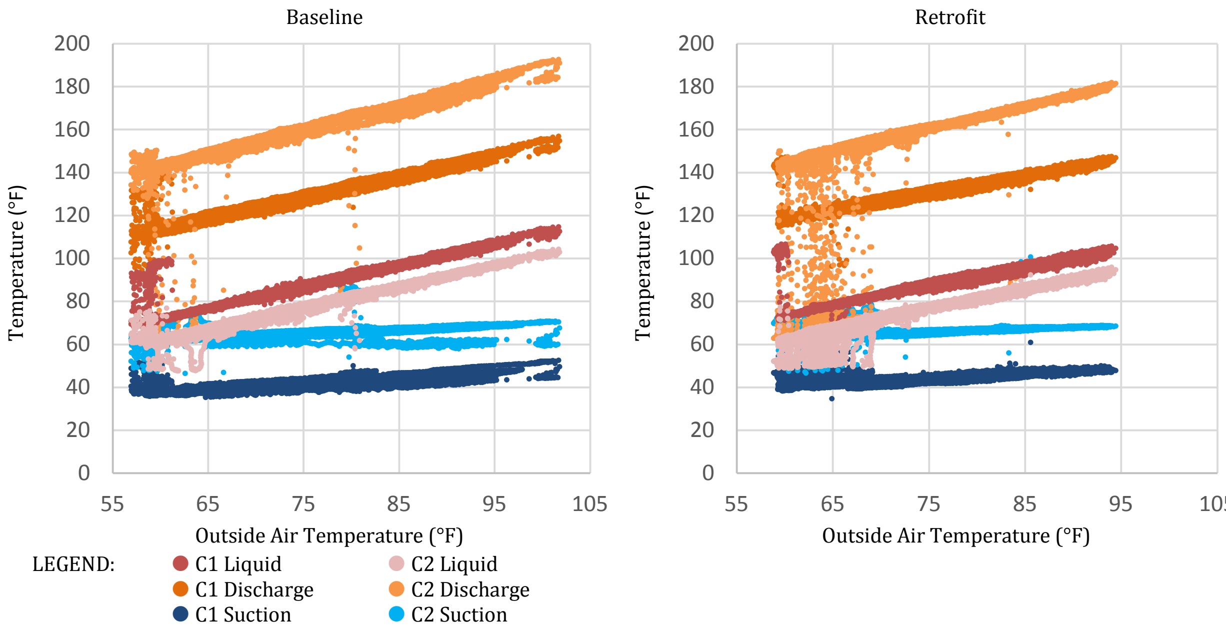


FIGURE 16: REFRIGERANT CIRCUIT TEMPERATURES WITH AND WITHOUT THE CONDENSER PRE-COOLER

WATER CONSUMPTION

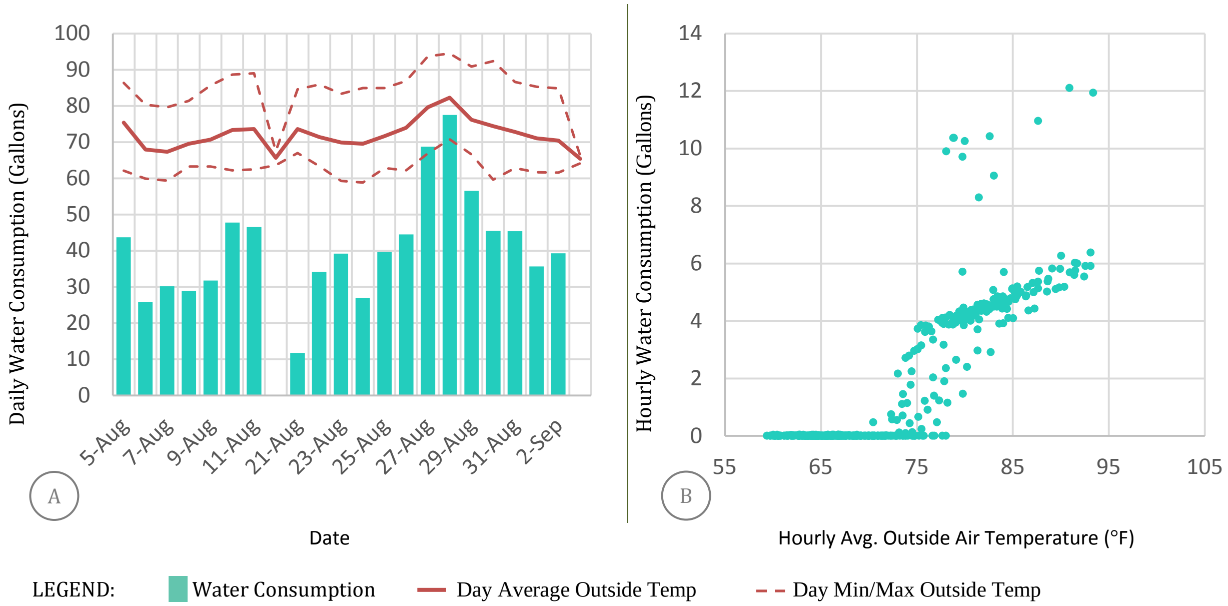


FIGURE 17: WATER CONSUMPTION FOR THE CONDENSER PRE COOLER (A) DAILY WATER CONSUMPTION (B) HOURLY WATER CONSUMPTION AS A FUNCTION OF OUTSIDE AIR TEMPERATURE

Although no energy savings were attainted, the amount of water used by the system is still relevant. However, for context, it is worthwhile to consider the amount of water used by the system. Figure 17 A charts the water consumed by the system for each day in the period of observation, and Figure 17 B plots the amount of water consumed each hour as a function of the corresponding outside air temperature. These charts illustrate the clear relationship between water use and outside air temperature which results from the control sequence used by the evaporative pre-cooler system.

Of course, the amount of water evaporated into the condenser air stream has a direct impact on the cooling effect that is achieved. It is interesting then, to note that for the manufacturer rated condenser airflow rate of 13,400 cfm, the cooling effect yielded by 6 gallons evaporation per hour would only be 3°F. For a wet bulb depression of 30°F, this corresponds very closely to the wet bulb effectiveness that was measured for this system. This observation would suggest that the poor performance observed for this condenser pre-cooler results mainly from the amount of water that is evaporated into the condenser air stream.

CONCLUSIONS & RECOMMENDATIONS

While other studies have measured large energy savings from condenser pre-coolers on rooftop units, this study shows one pre-cooler product that did not improve performance for a rooftop unit on a data center. Observed through several lenses, it is clear that the pre-cooler studied has almost no impact on overall performance of the air conditioner. These observations corroborate results from recent laboratory tests for the same technology.

The root of this poor energy performance can be attributed mainly to the fact that the amount of water used by the system has a small cooling potential for the condenser airflow rate on this rooftop unit.

However, it was also observed that the power draw and cooling efficiency for this system were not especially sensitive to outside air temperature. At peak conditions, an evaporative pre-cooler with 100% effectiveness would only reduce power draw by approximately 3%.

These results underscore the need for a standard method of test by which alternate condenser pre-cooler technologies can be equally compared to one another. Such an effort is underway with ASHRAE SPC 212. It also highlights the fact that condenser pre-cooling may have differing impacts in each application.

Despite the poor performance marks for this technology, we recommend that utility programs and other efforts work to advance broader application of condenser pre-coolers. Other studies have shown efficiency increase as large as 36% at typical peak conditions (Pistochini 2014, Davis 2014) – corresponding to a reduction in energy use intensity of 27%. Moreover, recent research has indicated that if condenser pre-cooling is coupled properly with variable fan speed and variable capacity compressor controls, efficiency could be increased by more than 60% without a sacrifice in sensible cooling capacity. This corresponds to a reduction in energy intensity of 38% (Pistochini 2015).

It is very important that any effort to advance these technologies make an effort to distinguish between those technologies and applications that will achieve savings, and those that will not.

It is concerning that the electric demand and efficiency for the rooftop unit observed here do not change substantially with outside air temperature. This study is the first indication that savings may depend significantly on application. It appears that the characteristics observed in this study are related to the fact that the equipment and ductwork systems are undersized for the load, however, this behavior should be considered in more depth. We suggest that standard methods of test should consider some of the physical scenarios that might erode the savings potential for condenser pre-cooling. We also recommend that efforts to apply the technology should be careful to ensure that the scenario in which the technology is applied has the potential for savings.

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