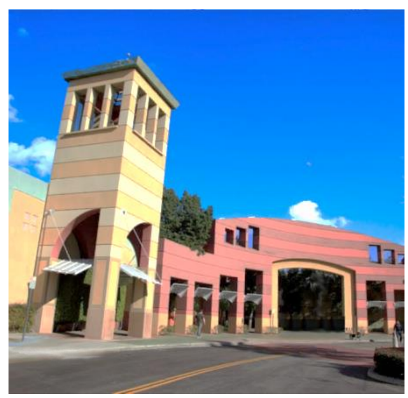
Emerging Technologies

# Hybrid Rooftop Air Conditioners with Dual Evaporative Pre-Cooling Performance Evaluation

*ET13SCE7090/ET13SCE7120*

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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. These RTUs are generally mass produced to meet federal efficiency standards, and then sold throughout the country. After having identified significant savings opportunities for equipment optimized for the California climate, the Western Cooling Efficiency Center created an RTU energy efficiency challenge. This efficiency target for this challenge, called the Western Cooling Challenge (WCC), was designed based upon adding evaporative pre-cooling to air entering the RTU condenser, and indirect evaporative pre-cooling to the outdoor air intake of the RTU. The results of these additions were sensible-cooling efficiency targets for annual energy consumption and peak electricity demand roughly 40% better than the performance of 2010 minimum-federal-standard equipment.

The purpose of this project was to gather field data to demonstrate and understand the performance of a hybrid rooftop air conditioner that uses dual-evaporative pre-cooling, essentially an RTU that employs the technology used to define the Western Cooling Challenge. The technology is expected to save energy and demand two ways:

a) by cooling the outdoor air being delivered to the RTU indoor coil, and thereby reducing how much cooling it needs to perform, and
b) by reducing the air temperature seen by the RTU condenser coil, thereby decreasing refrigerant pressure and the work that needs to be done by the compressor.

The key metrics used to characterize the performance of the retrofit include:

c) sensible Coefficient of Performance (COP)
d) sensible cooling capacity, and
e) electric power draw

Each performance metric is evaluated as a function of outdoor weather conditions. In addition, the project was designed to measure the on-site water consumption associated with achieving those performance improvements, as well as to calculate a key intermediate parameter, the evaporative effectiveness of the pre-cooling system. Evaporative effectiveness measures the ability of the evaporative media to cool the air entering the condenser toward the wet-bulb temperature (WBT) of the outside air, and is the key parameter used to characterize the performance of evaporative coolers for condenser air in the laboratory.

The approach chosen was a field test conducted on three new identical packaged rooftop units (RTUs) installed in Ontario, California. Two of these RTUs serve interior office spaces, and the third unit serves the kitchen of a restaurant and bakery. Monitoring involved minute-by-minute data collection on the RTUs with and without the evaporative media installed and operational. In addition to the key performance metrics, the internal workings were also investigated, including isolating the performance of the water-to-air heat exchanger for indirect evaporative cooling of ventilation air, monitoring the performance of the sump-pump control, as well as monitoring total water consumption and estimating the fraction of total water use associated with evaporation.

The energy use signature for each system with and without the dual-evaporative pre-cooler indicates energy savings between 13%-66% when outdoor-temperature conditions were between 100-105 °F, and average savings of 20%-64% for all operating hours above 70°F. For an installed cost of $350 per ton, the increase in efficiency demonstrated by this equipment would equate to a simple payback between 5-15 years. The payback period depends on application – it is most appealing in hot climates, in scenarios with longer runtime hours, for units with larger outside air fractions, and for customers with high peak demand charges. For the project evaluated here, we estimate a simple payback of 7.5 years (see Appendix A).

The following table summarizes savings , as well as other key performance metrics observed across a range of ambient conditions.

**TABLE ES-1: SUMMARY OF PERFORMANCE METRICS FOR EACH RTU (BINNED AVERAGE VALUES)**

| Equipment Tag | Location | 70-75 °F | 75-80 °F | 80-85 °F | 85-90 °F | 90-95 °F | 95-100 °F | 100-105 °F |
|---|---|---|---|---|---|---|---|---|
| Measured Percent Reduction in Hourly Energy Consumption | | | | | | | | |
| M 12-14 | Mall Security Offices | 87% | 63% | 47% | 52% | 56% | 69% | 66% |
| M 15-15 | Mall Admin Offices | 14% | 1% | -13% | -18% | 36% | 72% | 45% |
| AC 7 | Restaurant Kitchen | 40% | 27% | 17% | 18% | 14% | 8% | 13% |
| Sensible System Coefficient of Performance (–) | | | | | | | | |
| M 12-14 | Mall Security Offices | 2.3 | 2.2 | 2.3 | 2.3 | 2.4 | 2.3 | 2.3 |
| M 15-15 | Mall Admin Offices | 2.6 | 2.8 | 2.3 | 2.5 | 2.4 | 2.3 | 2.5 |
| AC 7 | Restaurant Kitchen | 2.0 | 2.8 | 3.0 | 3.3 | 3.6 | 3.7 | 3.8 |
| Sensible System Cooling Capacity (kBtu/hr) | | | | | | | | |
| M 12-14 | Mall Security Offices | 70 | 71 | 72 | 73 | 74 | 75 | 75 |
| M 15-15 | Mall Admin Offices | 120 | 125 | 130 | 140 | 81 | 75 | 82 |
| AC 7 | Restaurant Kitchen | 50 | 68 | 119 | 136 | 145 | 152 | 160 |
| Electric Power Draw (kW) | | | | | | | | |
| M 12-14 | Mall Security Offices | 9.0 | 9.1 | 9.1 | 9.1 | 9.2 | 9.2 | 9.4 |
| M 15-15 | Mall Admin Offices | 11.5 | 11.4 | 9.4 | 11.6 | 9.5 | 9.7 | 9.7 |
| AC 7 | Restaurant Kitchen | 9.2 | 11.8 | 12 | 12 | 12.1 | 12.2 | 12.5 |

These results indicate that the peak demand reduction and aggregate energy savings varies significantly between applications. The overall savings for AC7 is smallest, even though this unit operated at the highest COP. This apparent incongruity can be explained by the fact that for AC7, part of the efficiency increase resulted in increased cooling of the space instead of reduced electricity consumption. Despite the variability in measured performance, these three cases represent good examples of the savings that can be expected from this RTU. Water use for the three systems evaluated was found to be reasonable. In contrast to a parallel study with the same technology (Modera et al. 2014), the bleed rate was better controlled for this installation. Total water consumption only exceeded the estimated evaporation rate by ~25%, and the ratio of total water use to energy savings for M12-14 was roughly 5 gal/kWh. This should be compared to the state-wide average 1.34 to 2.76 gallons of water consumed per kWh of electricity generated (Pistochini, 2011).

This field evaluation suggests four key conclusions:

1. Hybrid rooftop air conditioners using dual-evaporative pre-cooling can deliver considerable energy savings, capacity improvement, and peak demand reduction.
2. The water evaporated to achieve these savings is comparable to the water that would be used to generate the electricity saved.
3. The actual savings realized depends on the application, patterns of use, and climate for which the measure is installed.
4. The savings potential, water use, and reliability for this technology would benefit from improved controls and system integration strategies.

Based upon the observed performance, the research team recommends hybrid dual-evaporative RTUs as a means to reduce energy consumption and peak demand for cooling in commercial buildings. However, we also recommend that utility efficiency programs address some of the challenges observed in this study, including:

1. Require a quality service agreement or manufacturer/installer performance guarantee in programs to avoid pre-mature failure, or abandonment of the pre-cooler functions within the RTU
2. Require a pre-application analysis designed to assure that any pre-existing performance issues be addressed along with the installation of the hybrid RTU (e.g. a poorly-maintained, inadequately-designed duct system)
3. Conduct further investigation of the three installations studied here: to optimize performance and produce practical guidelines for broader application of the technology

The results, observations, conclusions and recommendations from this study are generally consistent with those from a recent field study of the same dual-evaporative technology applied as a retrofit, and the COPs achieved in these field applications agree with laboratory observations for the same RTU. The authors suggest further work to translate the characteristic measurements from this study and others into a calibrated model for this hybrid system. The objective would be to provide a performance map and modelling platform that address all the various operating modes of this RTU, and that could be used to predict and analyze its performance in different climates and applications.

# ABBREVIATIONS AND ACRONYMS

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

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# INTRODUCTION

Energy used for air conditioning in buildings necessitates massive investments in electric generation and distribution capacity, draws on a substantial portion of our global fuel extraction, and subsequently contributes to a host of environmental, political, and economic challenges. There is a monumental need to improve efficiency for air conditioning. This report explores one technology opportunity that has previously demonstrated 40% on-peak demand savings for cooling in California climate conditions.

Buildings consume 70% of the electricity in the US, 50% of which is used for commercial buildings. Air conditioning and ventilation is responsible for more than 25% of the annual electricity use from commercial buildings in California. If gas consumption is considered, heating, cooling, and ventilation usually accounts for more than 50% of the annual energy use in these facilities.

Importantly, air conditioning can account for more than 50% of the on-peak electrical demand from commercial buildings. California's electric grid is especially stressed during summer periods when generation requirements can be twice as high as other periods in the year. 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.

Cooling and heating for commercial buildings is served predominately by packaged rooftop air conditioners (RTUs). This equipment is often the single largest connected load in a building, and uses 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 for rooftop air conditioners, including the measure that is evaluated through this study.

The intent of this project is to characterize the cooling performance and energy efficiency for three hybrid rooftop packaged air conditioners installed for commercial buildings in Ontario, California. The hybrid systems are standard 'high efficiency' model rooftop air conditioners that use 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 technology 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 emerges from a variety of efforts and innovation surrounding climate appropriate cooling strategies. The California Energy Efficiency Strategic Plan sets aggressive targets to advance the presence and market adoption of such measures (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 were 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 indicate that these measures can reduce annual electric energy consumption for heating, ventilation, and air conditioning (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 controls. 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, which means that utility efficiency programs will find it difficult to use 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 105ºF (73ºF wet-bulb), as compared to equipment that is minimally compliant with federal energy conservation standards. 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-osa/nominal ton*. 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, there has previously not been adequate field data 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.

Further, 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. Nevertheless, we offer a number of recommendations and identify some of the real world constraints for the product.

This report describes the technology evaluated, and the applications in which the measure was installed. Then, the experimental design and technical methodology is documented, and an array of performance results are presented and discussed. We identify the technical advantages apparent for the machine, and review lessons learned through the process before presenting general conclusions and recommendations to foster further advancements and application for the technology.

# PROJECT 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 on a rooftop unit. As condenser air is drawn across the media it cools by evaporation. Water supplied at the top of this media is also cooled by evaporation and drains down to a stainless steel sump, where it is pumped to a water-to-air heat exchanger located at the ventilation air inlet. Ventilation air is cooled as it passes through this heat exchanger, resulting in a cooler mixed air temperature at the evaporator coil inlet. The water warms as it exchanges heat with ventilation air. Then the water is circulated to the top of the direct evaporative media where the cycle begins again. Figure 1 provides a schematic illustration of the technology.

These dual processes work together to increase cooling capacity and to improve efficiency for the vapor compression system. The second effect is mainly caused by a lower heat sink temperature for the refrigeration cycle. The effect of a lower heat sink temperature is readily apparent for conventional air conditioners. Manufacturer performance tables, laboratory measurements, and most field observations show an efficiency increase of 1-2% for every degree Fahrenheit ($^{o}$F) decrease for air at the condenser coil inlet. The dual evaporative pre-cooler takes advantage of this natural fact by reducing condenser temperature with an evaporative cooling process that requires only 150 Watt (W) pumping power. The capacity increase is partly a result of reduced condenser air temperature, and partly a result of the sensible cooling delivered by indirect evaporative cooling of the ventilation air. 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 (RTUs), and only requires modest in-field integration efforts. The system uses a simple stand-alone control scheme that does not require integration with or revisions to existing RTU 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 a 20 *ton* RTU. In this study the measure was installed on new equipment, but it can also be installed for existing equipment. Actual installed costs appear to depend on the number of retrofits, equipment size, and the ease of installation. In addition, 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 than retrofits 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.

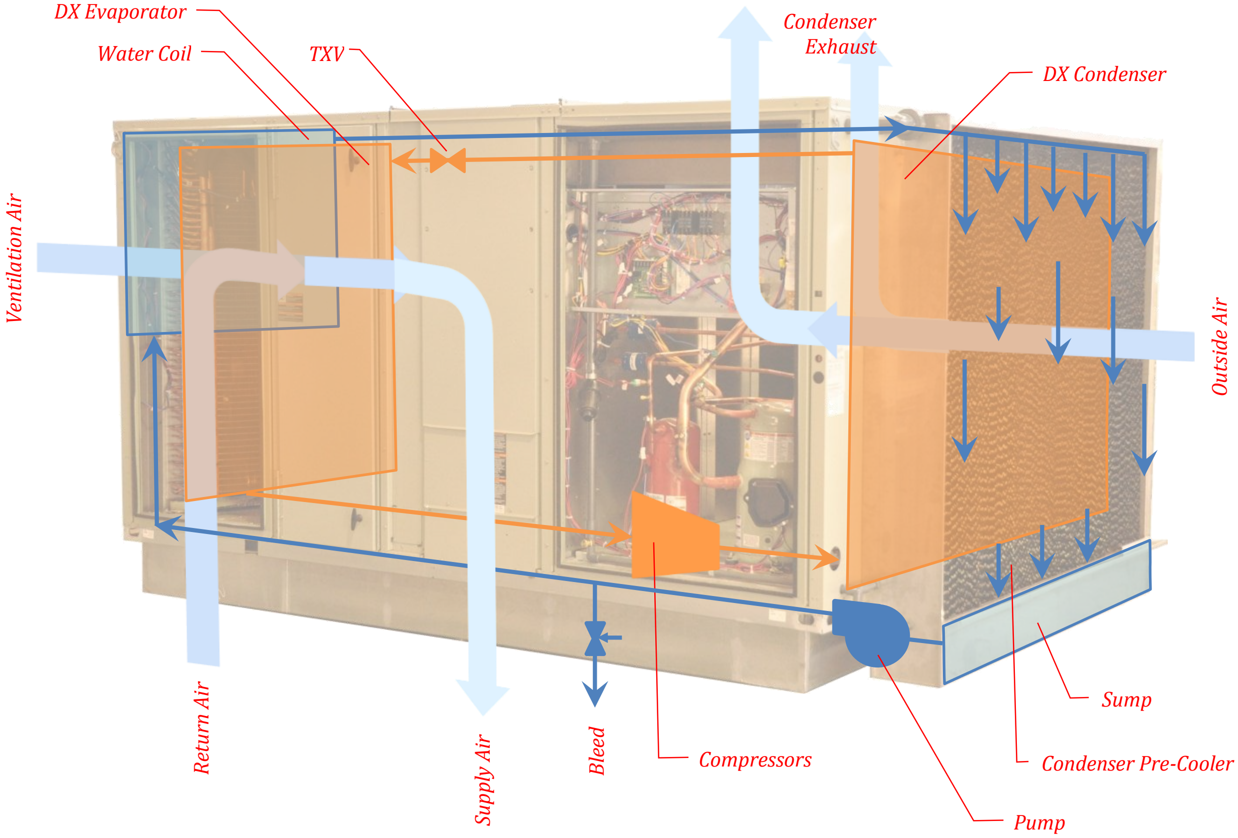


**Figure 1: Conceptual Schematic for Rooftop Unit with Dual Evaporative Pre-Cooling**

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 exerted 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 an air balance to maintain an appropriate ventilation air flow rate.

Second, the savings achieved by this measure is tied closely to the amount of outside air that is treated by the system. When it is an option, it should therefore be advantageous to group the ventilation needs for a building onto units that use this technology, and to shift other standard units to operate as recirculation only and in an 'AUTO' mode where the supply air blower only cycles with an active call for cooling. 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 mode 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. As for economizer operation, since the dual evaporative pre-cooler cools incoming ventilation air, this measure can also extend the range of outside air temperatures that are appropriate for operation with 100% outside air.

## *OPERATING MODES & SEQUENCE OF OPERATIONS*

A hybrid RTU that uses 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 a different 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. Table 1 summarizes each possible operating mode, and the corresponding function of each component. All of the active cooling modes described assume that the unit functions to provide continuous ventilation for indoor air quality.

It should be noted that while the simple outside air temperature control switch 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.

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 cooling effect for ventilation air is small, and there is less efficiency benefit from condenser-air cooling. Further, it has been observed that if the pump is allowed to circulate at temperatures well below 70°F, the media may not dry out adequately each day, which can allow biological growth on the media. The optimal changeover point would be that temperatures below which the compressor and fan energy savings benefits are smaller than the energy expended for pump operation.

In lieu of the temperature switch control strategy described above, the technology can also utilize a custom-programmed controller that measures key operating parameters on the rooftop unit (such as blower and compressor operation) in order make control decisions for the dual evaporative pre-cooler. There are a variety of ways that this type of integrated controller could be deployed. Two units evaluated in this project used a custom-programmed controller to enable the pump only if the blower is operating, and the outside air temperature is above the changeover set point. This approach can also employ a relay to switch the condenser fans on anytime the pump operates, even if there is not compressor operation. As noted in Table 1, this control strategy avoids pump operation when the unit is off but still uses the evaporative cooling components during ventilation, or economizer-only modes.

A schematic illustration of the control wiring for the custom-programmed controller approach is provided in Figure 2.

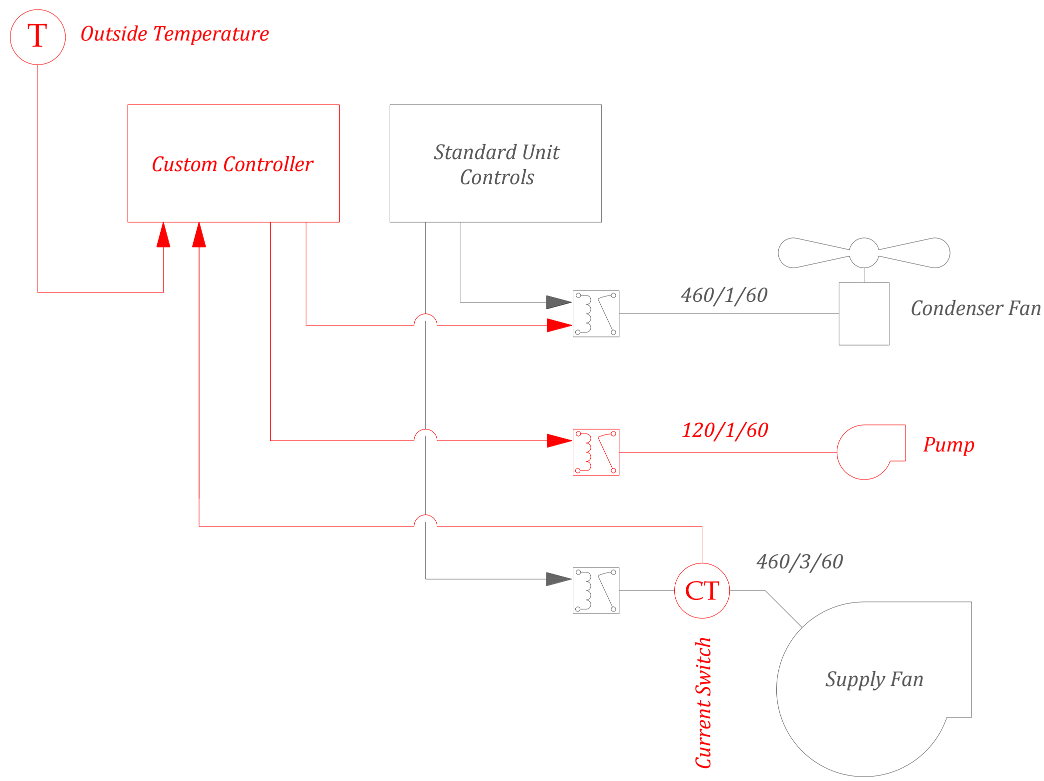


**FIGURE 2: CONTROLS SCHEMATIC FOR SYSTEMS USING CUSTOM PROGRAMED CONTROLLER (M12-14 & M15-15)**

**TABLE 1: DEFINITION OF EACH OPERATING MODE**

| Mode | Indoor Blower | OSA Damper | Cond. Fans[1,2] | | Compressors | Water Pump |
|---|---|---|---|---|---|---|
| | | | *Temp. Switch* | *Controller* | | |
| Off | OFF | CLOSED | OFF | | OFF | OFF |
| Pump While Off [2] | OFF | CLOSED | OFF | | OFF | ON |
| Ventilation Only | Indexed to field selected set point according to mode | Indexed to field selected set point according to mode | OFF | | OFF | OFF |
| Economizer | | | OFF | | OFF | OFF |
| DX1 | | | 0-2 | | 1 | OFF |
| DX2 | | | 0-2 | | 2 | OFF |
| Pump + Ventilation [2] | | | 0 | 1 | OFF | ON |
| Pump + Economizer [2] | | | 0 | 1 | OFF | ON |
| Pump + DX1 | | | 0-2 | 1-2 | 1 | ON |
| Pump + DX2 | | | 0-2 | 1-2 | 2 | ON |
| Integrated Economizer [3] | | | 0-2 | | 1-2 | OFF |
| Pump + Integrated Economizer | | | 0-2 | 1-2 | 1-2 | ON |
| Heating | | | 0 | | 0 | OFF |

[1] Condenser fan operation is controlled in part by head pressure measurements on the refrigerant circuit. Nominally, the number of condenser fans corresponds to the number of compressors operating. However, when head pressure is below a threshold, one or both condenser fans will cycle off, even while compressor(s) are running.

[2] The two rooftop units at the mall site use a custom-programed controller that only allows pump operation when the supply blower is active, and enables condenser fan operation whenever the pump circulates, whether or not compressors are active. Subsequently, these units do not operate in any scenario where the pump circulates water without condenser airflow or without supply airflow.

[3] The unit may run in full economizer mode (outside air only), or as an integrated economizer (outside air plus compressor cooling), with or without the dual-evaporative pre-cooling function. Since the retrofit provides some temperature reduction to the ventilation air stream, there is good reason to expect that equipment with the retrofit should benefit from operating with an elevated economizer changeover set point.

## OVERVIEW OF FIELD TEST SITES

Two field test sites in Ontario, CA were selected for this study. The first site is a restaurant and bakery, the second site is a mall facility. For the restaurant, a new rooftop unit with the dual evaporative pre-cooler was installed to serve cooling and ventilation for a portion of the kitchen. Two systems were installed for the mall, one to cool administrative offices, and another to cool the mall security office – both spaces are interior zones. The two units in the mall use a common return plenum, but use separate and independent thermostat controls. The mall security office is a 600 $ft^2$ space with large equipment loads from security electronics.

Table 2 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 2. Figure 6 in section "*Assessment Methodology: Water and Airflow Measurements*" documents the measured supply airflow and ventilation airflow rates for each unit.

**TABLE 2: EQUIPMENT SCHEDULE FOR ROOFTOP UNITS[1]**

| *Equipment Tag* | *Site & Location* | *Manufacturer* | *Model* | *AHRI Rated Total Net Cooling Capacity (kBtu/hr)* | *Design Supply Airflow (cfm)* | *Design Vent Airflow (cfm)* | *Compressors* | *V/ø/hz* |
|---|---|---|---|---|---|---|---|---|
| AC 7 | Restaurant Kitchen | Trane | YFD-151-F3-K | 145 | 5,000 | 1,500 | 2 Scroll | 460/3/60 |
| M 12-14 | Mall Security Offices | Trane | TFD-151-F4-L | 145 | 5,000 | 960 | 2 Scroll | 460/3/60 |
| M 15-15 | Mall Admin Offices | Trane | TFD-151-F4-L | 145 | 5,000 | 1,140 | 2 Scroll | 230/3/60 |

[1] The three units also include the following specifications:

- 12.5 ton 410A "high efficiency" models (before dual-evaporative pre-cooler)
- VFD on indoor fan
- Micro channel condenser coils
- TXV/face split evaporator coils
- Down-flow economizers (0-100% fan speed with barometric relief dampers)
- No variable speed condenser fans
- MC12-14 has an oversized indoor fan motor.

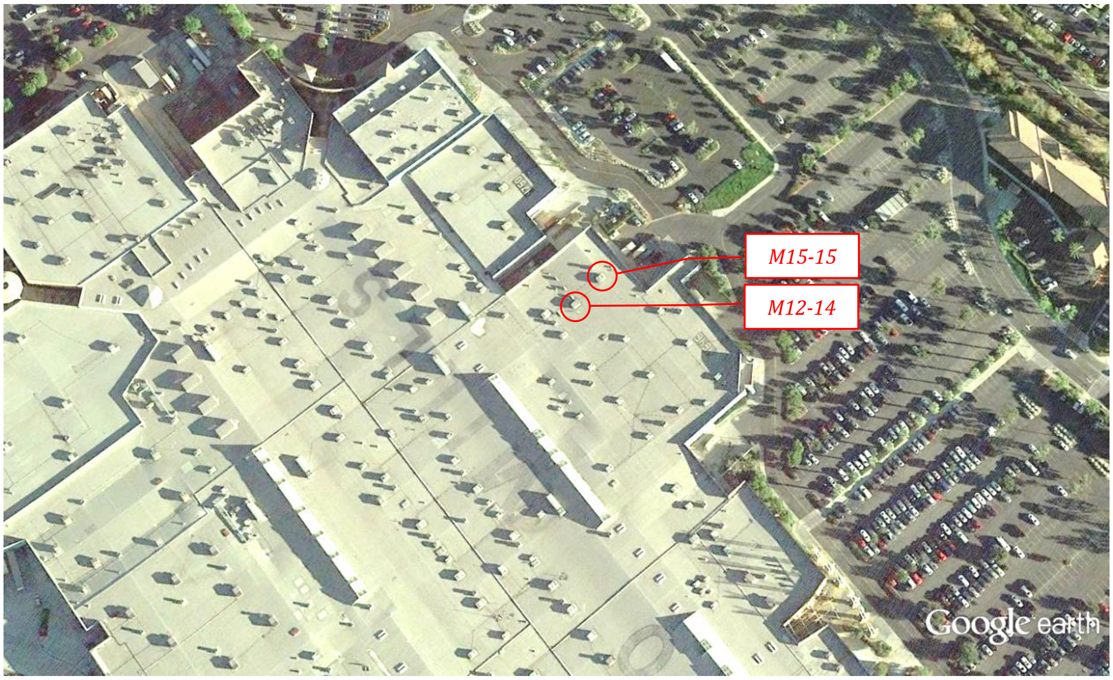


**FIGURE 3: LAYOUT OF ROOFTOP UNITS STUDIED AT MALL FACILITY IN ONTARIO**


**FIGURE 4: LAYOUT OF ROOFTOP UNIT STUDIED AT RESTAURANT FACILITY IN ONTARIO**

# ASSESSMENT OBJECTIVES

The primary objective of this investigation was to conduct multiple field evaluations of a new hybrid rooftop air conditioner that uses 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 three 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 to investigate 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 observation of the hybrid system in three different applications in order to 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.

In addition, the study presents a pre-post assessment of energy use to estimate savings from the three installations. This assessment is not an annual projection of savings, only a comparison of characteristic performance for each unit from periods of operation with and without the dual-evaporative pre-cooler. Mainly, results present a clear and reliable description of system performance for real world operation with the dual-evaporative pre-cooler.

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 on a broad scale. We identify the successes observed in these areas and recommend possible solutions where we recognize challenges.

# ASSESSMENT METHODOLOGY

## OVERVIEW OF THE TECHNICAL APPROACH

The three systems tested in this study were installed in November 2012. The equipment was commissioned by the installing contractor at that time, but the dual evaporative pre-cooling systems were not commissioned and enabled until August 2013 when the period of study began. In August 2013, UC Davis installed a thorough array of instrumentation on each of the three 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 for 14 months between August 2013 and October 2014. For the sake of clarity, the data presented in this report is drawn from September 2013, the month that experienced the widest range of ambient conditions.

Data analysis and visualization was conducted using a custom software developed in Python (Rossum 1995). Python is especially well-suited for manipulation and analysis of large time series datasets. A custom script and library of analysis functions was developed through this project that can be applied to analysis of similar monitoring and evaluation efforts in the future. In particular, the developments allow for straightforward definition of distinct operating modes and for filtering of data to extract performance results for periods of steady state operations. The research team also developed a library of psychrometric functions for Python, as well as an array of calculators for common analysis metrics such as cooling capacity, energy and water use efficiency.

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 that 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 technique. These measurements are described in more detail in the "*Water and Airflow Measurements*" section.

## 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 used for the study is illustrated schematically in Figure 5. Table 3 provides a simple description of each measurement marked in the instrumentation schematic, and documents the performance specifications for the sensors used for each corresponding measurement.

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 direct current-voltage signals used on-board the RTU to control the operation of various components.

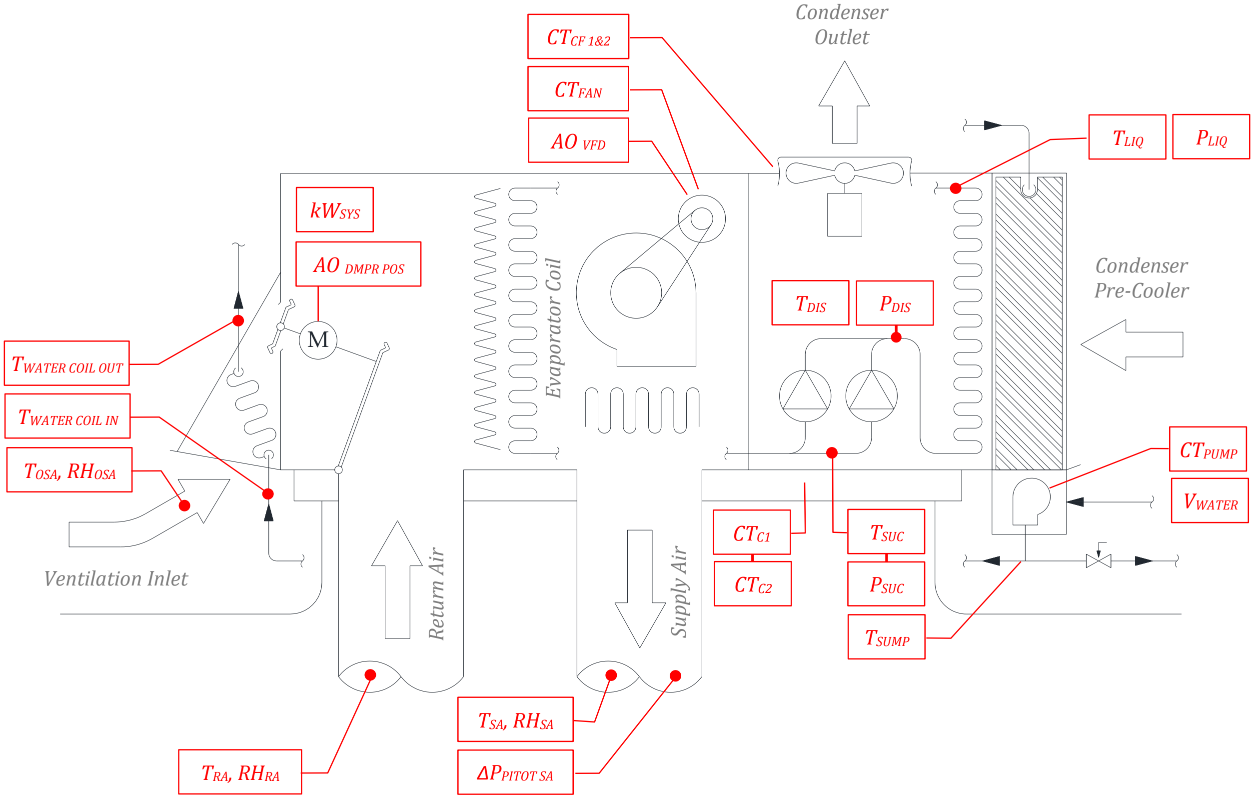


**FIGURE 5: INSTRUMENTATION SCHEMATIC**

**TABLE 3: INSTRUMENTATION TABLE**

| *Name* | *Measurement* | *Sensor* | *Uncertainty* |
|---|---|---|---|
| $T_{OSA}$ | Temperature – Outside Air | Vaisala HUMICAP HMP110 | +/- 0.2 °C |
| $RH_{OSA}$ | 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 Temperature | Omega 10k Ω TH-44031-40-T | +/- 0.1 °C |
| $V_{WATER}$ | System Water Consumption | OMEGA FTB 4105 A P 1 pulse per gallon | +/- 1.5% FS |
| $AO_{DMPR\ POS}$ | RA/OSA Damper Actuator Position | 2-10 V dc Analog Signal | NA |
| $kW_{SYSTEM}$ | System Power Draw, Voltage, Current & PF | Dent Powerscout 3 | +/- 0.5% |

## DETERMINATION OF OPERATING MODE

The performance of each system changes 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 combination of component states in each mode of operation is described in Table 1, in the "Operating Modes & Sequence of Operations" section. These definitions were used to separate the results categorically.

In some cases, when faults and errors occur, equipment can operate in unexpected ways that are not clearly defined by an intentional control sequence. In this project there were some instances of intermittent pump cycling are not defined by the conceptual description of each mode in Table 1. To address this behavior, these observations were tagged as if the equipment was operating continuously in a mode with the pump on, and the performance measured during these periods is presented alongside periods of more steady operation. In this way, the inconsistency is included as a characteristic of the intended mode of operation. These behaviors are not ideal. We explore the reason for these behaviors, and recommend solutions in the "*Results & Discussion*" section.

There are sometimes also instances where compressors operate without any condenser fans. In these cases we have marked the mode according to compressor function. These RTUs incorporate a head pressure control scheme that disables condenser fan operation when discharge pressure is too low in order to reduce heat dissipation, to keep the evaporator temperature from dropping too low, and to avoid return of liquid refrigerant to the compressor. Therefore, when one compressor is active the mode is tagged as "DX1" regardless of whether or not either condenser fan is on. In this way, any variation in performance is included as a characteristic of the intended mode of operation.

The explicit logic used to determine the state of each component and the system operating mode was slightly different for each unit. Although the three RTUs have the same model number and are nominally the same machine, they are each configured differently, and operate with different component power draw characteristics in each mode of operation. The research team found that each of the three units had noticeably different power draws on condenser and supply fans. For example, during normal operation the supply fan on M12-14 draws roughly 0.85 kilowatts (kW), M15-15 draws 3.85 kW, and AC-7 draws 1.4 kW. The team also found significant difference in the minimum outside air damper position in ventilation and economizer modes. Each unit has an outside air damper actuator that is controlled with a 0-10 Volt (V) dc signal. The research team found the minimum position for the units ranged from 3 – 7Vdc and the economizer position ranged from 8 – 10Vdc. Therefore, the logical details for determining operating modes were carefully chosen for each unit by examining patterns in the time series data for each control variable, and by correlating these patterns to records from on-site observations of each system in each mode.

## CUMULATIVE VERSUS STEADY-STATE PERFORMANCE ANALYSIS

Two types of performance results are presented in this report: 1) cumulative performance metrics, and 2) steady-state performance metrics calculated whenever the equipment had been running in a particular mode for more than ten minutes. The cumulative metrics account for every minute of the test period, whereas the steady-state results only show data acquired after 10 minutes of operation in a particular mode.

For example, Figure 9 presents the cumulative sensible cooling, and Figure 15 presents the cumulative water use for each day. These metrics account for the electricity use, water consumption, and thermal energy delivered during every minute of the study period. Figure 11 plots the sensible cooling capacity in each mode of operation as a function of outside air temperature, and Figure 10 presents the sensible COP. In order to reduce noise and ease visual interpretation of the results, these plots discard observations from the first 10 minutes of operation in any mode. The exception to this rule is for periods of intermittent pump cycling within a particular mode of operation, as described previously.

## DATA CONFIDENCE

The measurement accuracy for each instrument used for field monitoring is recorded as part of the monitoring plan in Table 3. Table 4 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 the "*Definition & Calculation of Performance Metrics*" section. The values in Table 4 do not account for any methodological uncertainty associated with features such as sensor placement, or transient interaction between equipment operation and sensor response.

**TABLE 4. UNCERTAINTY FOR KEY CALCULATED METRICS[1]**

| *Metric* | *Uncertainty* |
|---|---|
| Supply Airflow Rate | ±34 SCFM |
| Ventilation Airflow Rate | ± 117 SCFM |
| Absolute Humidity | ±0.00057 pound $(lb)_{m,\ water} / lb_{m,\ dry\ air}$ |
| Sensible System Capacity | ±6.2kilo British Thermal Unit per hour (kBtu) /hr |
| Sensible Room Capacity | ±4.5 kBtu/hr |

[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 = 1903.4 SCFM, Ventilation Flow Rate = 447.8 SCFM

| | |
|---|---|
| Sensible System COP | ±1.2 |
| Sensible Room COP | ±0.9 |
| Water Use | ±0.06 gallon per ton per hour (gal/ton/hr) |
| Wet Bulb Effectiveness | ± 0.08 |

# DEFINITION & CALCULATION OF PERFORMANCE METRICS

## *WATER AND AIRFLOW MEASUREMENTS*

Accurate calculation of cooling capacity, COP, and other metrics relies directly on accurate measurements 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 a constant speed, so the assumption is that the flow rate of water circulated through each dual evaporative pre-cooler remains consistent for all intervals. Table 5 summarizes the measurements made for each system. Note that the flow rate is fairly consistent between units.

**TABLE 5: 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)* |
|---|---|---|---|
| AC 7 | 25.1 | 38.7 | 11.1 |
| M 12-14 | 23.4 | 38.8 | 11.9 |
| M 15-15 | 23.8 | 39.3 | 11.9 |

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:

**EQUATION 1. VOLUME OF AIR INTO $CO_2$ INTO SUPPLY AIRSTREAM, CORRESPONDING RISE IN $CO_2$ CONCENTRATION DOWNSTREAM**

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

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 and fan speed scenario, $CO_2$ concentration was measured in the outside air and return air streams, then 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 Equation 2:

**EQUATION 2. MASS BALANCE CALCULATIONS**

$$OSAF = \frac{\dot{V}_{OSA}}{\dot{V}_{SA}} = \frac{C_{SA} - C_{RA}}{C_{OSA} - C_{RA}} \quad (3)$$

Figure 6 charts the airflow and outside air fraction maps developed for each of the three units. Plots in Column A show supply airflow rates, and plots in Column B show outside air fraction. It should be noted that while all three systems incorporate variable speed drives for the supply blower, both MC 12-14 and MC 15-15 were field configured in a way that precluded the fan from changing speed with each mode of operation. Since these two systems do not change fan speed, supply airflow rates were mapped as a function of outside air damper position only. The outside air pathway for the RTUs evaluated has a higher resistance to flow than does the return air pathway; as a result, supply airflow tends to decrease when the outside air damper opens. This trend is most obvious for M15-15.

As recorded in Figure 6, while the nominal supply airflow rate for these three units was specified at 5,000 *cfm*, they were each field-configured to operate very differently. M12-14, which serves the mall security office, runs with a very low fan speed and supplies only 2,000 *scfm*, while M15-15 moves between 6,000-8,000 *scfm*, depending on the outside air damper position. Neither machine changes fan speed with operating mode.

The restaurant unit was the only installation configured to adjust fan speed with certain modes of operation. Subsequently, supply airflow rate and outside air fraction for this system were mapped as a function of both fan speed and damper position. However, as discussed in section: "*Results & Discussion: System Power Draw*", the change in airflow rate does not correspond to compressor operation in the ways that one would expect. AC7 operates at approximately 5,600 *scfm* for both first and second stage cooling, and drops to 3,800 s*cfm* for economizer mode, and for periods above 75°F when the unit cycles between DX2 and economizer+DX1.

Further, none of the units were configured to operate in a ventilation-only mode during periods when there is no call for heating or cooling. Supply airflow rate and outside air fraction measurements were made for the mode, but no units operated to provide continuous ventilation throughout the course of observation.

Importantly, these airflow analysis also measure damper and cabinet leakage. When M12-14 and M15-15 switch to economizer mode, only 65% of the supply airflow is drawn from outside, the remainder is drawn from return air. AC7 operates with 90% outside air fraction in economizer mode. This does not achieve California Building Efficiency Standard requirements that "air economizers [shall be] capable of modulating … dampers to supply 100% of the design supply air quantity as outside-air" (CEC 2012). This fact is mostly a result of the baseline RTU construction, and installation practices, however the trend is worsened somewhat by the added resistance from the ventilation cooling coil. When outside air dampers are fully closed, approximately 5% of the supply airflow for these systems comes from outside.

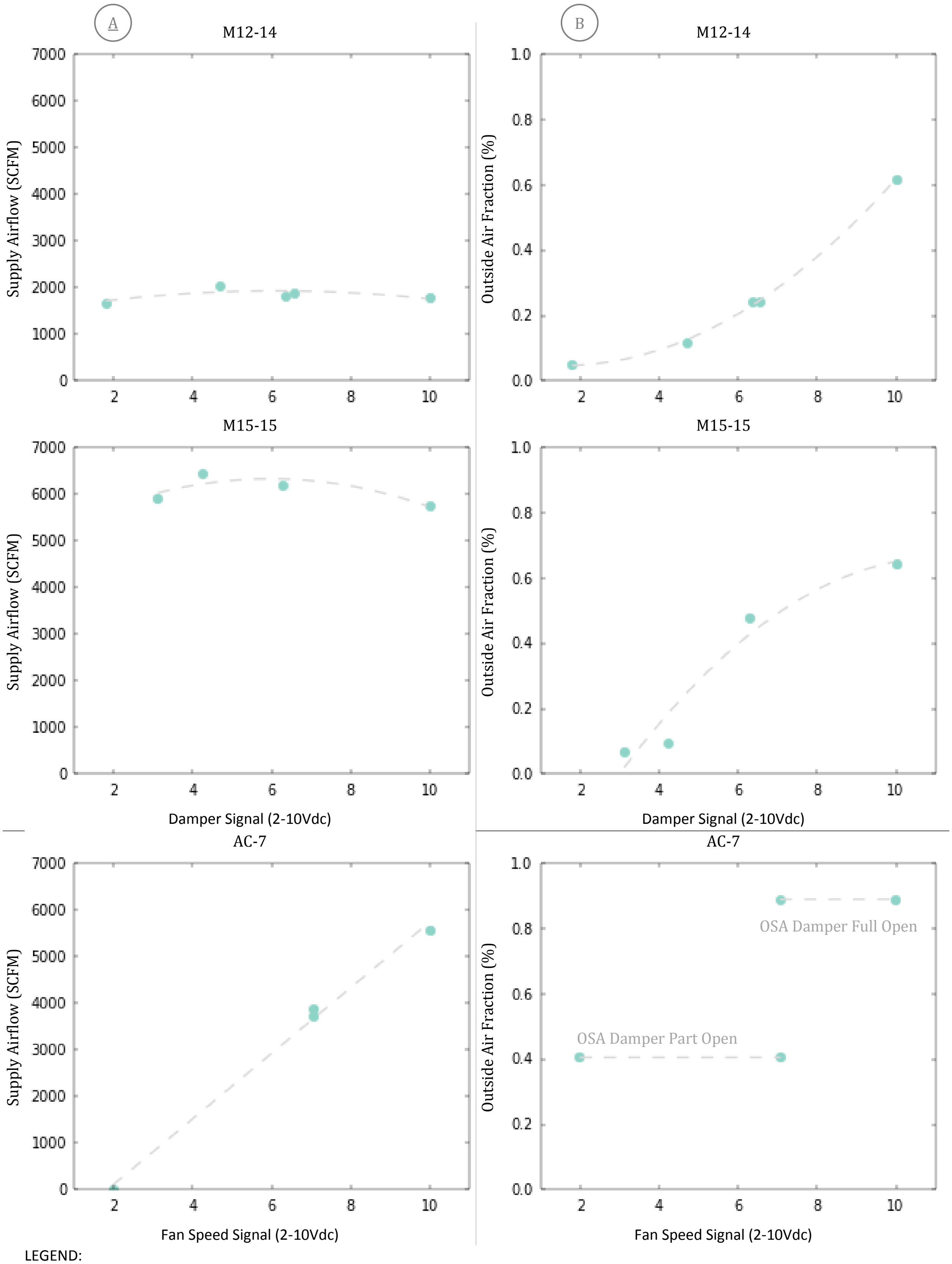


**FIGURE 6: (A) SUPPLY AIRFLOW MAP AND (B) OUTSIDE AIR FRACTION AS A FUNCTION OF OUTSIDE AIR DAMPER POSITION (FOR M12-14 AND M15-15) AND AS A FUNCTION OF FAN SPEED AND DAMPER POSITION (FOR M12-14 AND M15-15)**

### *Calculating Cooling Capacity*

The system-cooling-capacity for the hybrid equipment 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 4. 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.

**Equation 4. Supply airflow rate and Enthalpy difference between air and supply streams**

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

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 RTU 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 5 calculates the specific enthalpy for the 'virtual' mixed air condition.

**Equation 5. Calculation of specific enthalpy for virtual mixed air condition**

$$h^*_{MA} = \text{OSAF} \cdot h_{OSA} + (1 - \text{OSAF}) \cdot h_{RA}$$

The room-cooling capacity, given by Equation 6, is the cooling that is actually of service to the zone. This metric discounts the portion of the system-cooling-capacity that goes toward cooling ventilation air to the room air condition. In the case when outside air is cooler than return air, room-cooling may be greater than the system cooling (this should occur in any economizer mode).

**Equation 6 Room-cooling capacity**

$$\dot{H}_{room} = \dot{m}_{SA} \cdot (h_{RA} - h_{SA})$$

Since ambient humidity in most western climates is low enough that dehumidification is not necessary to maintain occupant comfort in most commercial buildings, the assessment presented here focuses on the system's ability to produce sensible cooling (ASHRAE 2010). Furthermore, since thermostat controls for this application only respond to temperature and do not control humidity, it is not appropriate to credit any latent cooling when considering the performance advantages of the dual-evaporative pre-cooling system studied here. The net sensible system cooling capacity is determined according to Equation 7:

**Equation 7. Calculating the Net Sensible System Cooling Capacity**

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

Concomitantly, the latent system cooling is determined as:

**Equation 8. Determining the Latent System Cooling**

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

### *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 COP:

**Equation 9. Determining the Coefficient of Performance**

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

Analysis in this report focuses on the sensible cooling generated by the equipment. This metric discounts the enthalpy associated with reduced humidity. The Sensible Coefficient of Performance can be expressed as:

**EQUATION 10. SENSIBLE COEFFICIENT OF PERFORMANCE**

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

Further, performance results are described both in terms of the Sensible System COP, and the Sensible Room COP. The first metric considers the ratio of electricity consumed to the sum of sensible cooling generated by the machine. The second metric compares the electricity consumed to the sensible cooling effect on the room.

**EQUATION 11. RATIO OF ELECTRICITY CONSUMED TO THE SUM OF SENSIBLE COOLING GENERATED BY THE MACHINE**

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

**EQUATION 12. COMPARISON OF ELECTRICITY CONSUMED TO THE SENSIBLE COOLING EFFECT ON THE ROOM**

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

## *ANALYSIS OF VENTILATION AIR COOLING PERFORMANCE*

Since the ventilation air mixes with return air immediately after passing through the water coil, 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 using Equation 13.

**EQUATION 13. CALCULATING PRODUCT AIR TEMPERATURE**

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

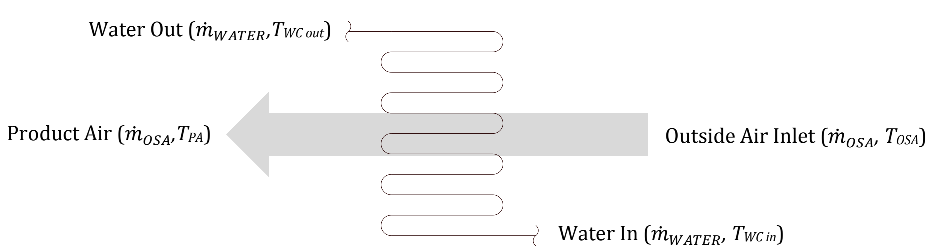


**FIGURE 7: SCHEMATIC FOR ENERGY BALANCE USED TO CALCULATE PRODUCT AIR TEMPERATURE**

## *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 14. 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.

**EQUATION 14. CALCULATING VENTILATION AIR**

$$\mathrm{WBE}_{IEC} = \frac{T_{OSA} - T_{PA}}{T_{OSA} - T_{wb\,OSA}}$$

### *CALCULATING SENSIBLE HEAT EXCHANGER EFFECTIVENESS*

The sensible heat exchanger effectiveness, which is the most common metric to describe heat exchanger performance, is calculated by using Equation 15.

**EQUATION 15. CALCULATING SENSIBLE HEAT EXCHANGER EFFECTIVENESS**

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

Where:

$\dot{m}_{OSA}$ = mass flow rate of the ventilation air stream

$c_{p\,air}$ = specific heat capacity for air

$T_{OSA}$ = dry bulb temperature of outside air

$T_{PA}$ = dry bulb temperature of product air (outlet from the water coil)

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

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

### *WATER USE INTENSITY FOR ENERGY SAVINGS*

The dual evaporative pre-cooler studied here makes substantial gains for energy efficiency, but consumes some water on site. Previous research suggests that site energy savings can offset upstream water consumption from the generation of electricity, even to the extent that total net water consumed may be less for evaporative cooling systems than for conventional air conditioners (Pistochini 2011, Torcellini 2003). Determination of this balance is most sensitive to the water use intensity for electricity generation, to the energy efficiency improvement attributed to the measure, and to the rate of water use on site. The water use intensity for energy savings is defined as:

**EQUATION 16. DETERMINING WATER USE INTENSITY FOR ENERGY SAVINGS**

$$\eta_{water\,use} = \frac{\dot{V}_{water}}{kWh_{savings}}$$

The volume water consumption term in this metric counts all of the water that is consumed by the equipment. This includes the amount of water evaporated for cooling, plus the amount of water drained for bleed in order to manage water quality.

# RESULTS & DISCUSSION

Monitoring for the three dual-evaporative pre-cooler systems studied here was conducted over a 14 month period from August 2013 – October 2014. However, the results presented in this report focus mainly on data from a single representative month of operation – September 2013. The month was selected because it included observations over a wide range of ambient conditions, which allowed for clear analysis of the characteristic performance for each system across a manageable slice of time. The extended data set from all months of monitoring is available as a resource to develop robust empirical performance maps for equipment performance in each mode and to validate modeling and simulation efforts for the equipment.

Although most of the results presented here focus explicitly on operation during September 2013, section "*Comparison of Energy Use with and without the Dual-Evaporative Pre-Cooler*" compares data from September 2013 with data from September – October 2014, a period when the dual-evaporative pre-cooler had been disabled to establish a clear performance baseline for each unit. Table 6 summarizes the key results from this comparison, including the hourly kWh savings across different temperature bins, the sensible COP, sensible system Capacity ventilation coil cooling effect, and hourly water consumption.

**TABLE 6: SUMMARY OF KEY RESULTS**

| *Key Metric* | *70-75 °F* | *75-80 °F* | *80-85 °F* | *85-90 °F* | *90-95 °F* | *95-100 °F* | *100-105 °F* |
|---|---|---|---|---|---|---|---|
| M 12-14 Mall Security Offices | | | | | | | |
| Baseline Avg. Hourly Energy (kWh) | 6.10 | 7.35 | 8.22 | 9.50 | 10.69 | 12.26 | 13.80 |
| Measure Avg. Hourly Energy (kWh) | 2.40 | 3.82 | 5.11 | 5.59 | 6.04 | 5.98 | 6.93 |
| Hourly kWh Savings (%) | 87% | 63% | 47% | 52% | 56% | 69% | 66% |
| Sensible System COP (–) | 2.1 | 2.3 | 2.3 | 2.3 | 2.4 | 2.3 | 2.2 |
| Sensible System Capacity (kBtu/hr) | 59.7 | 63.8 | 67.2 | 69.1 | 69.6 | 68.8 | 64.2 |
| Ventilation Coil ΔT (°F) | NA | NA | NA | NA | NA | NA | NA |
| Hourly Water Consumption (gal) | 0 | 2.1 | 3 | 11.5 | 18.4 | 27.2 | 37.8 |
| M15-15 Mall Admin Offices | | | | | | | |
| Baseline Hourly Energy (kWh) | 0.72 | 1.17 | 2.24 | 4.11 | 7.54 | 10.39 | 8.89 |
| Measure Hourly Energy (kWh) | 0.63 | 1.16 | 2.54 | 4.90 | 5.23 | 4.88 | 5.61 |
| Hourly kWh Savings (%) | 14% | 1% | -13% | -18% | 36% | 72% | 45% |
| Sensible System COP (–) | 2.2 | 2.1 | 2.2 | 2.4 | 2.5 | 2.4 | 2.4 |
| Sensible System Capacity (kBtu/hr) | 71.3 | 67.9 | 71.9 | 75.9 | 81.8 | 79.9 | 80.8 |
| Ventilation Coil ΔT (°F) | (-) 0.8 | (-) 0.4 | (-) 0.4 | 0.5 | 3.25 | 10.3 | 18.2 |
| Hourly Water Consumption (gal) | 0.1 | 1.2 | 4.1 | 6.3 | 12.2 | 17.9 | 37.8 |

| | AC 7 Restaurant Kitchen | | | | | | |
|---|---|---|---|---|---|---|---|
| Baseline Hourly Energy (kWh) | 9.49 | 11.40 | 12.26 | 12.73 | 13.22 | 13.27 | 14.15 |
| Measure Hourly Energy (kWh) | 6.32 | 8.66 | 10.36 | 10.63 | 11.44 | 12.2 | 12.41 |
| Hourly kWh Savings (%) | 40% | 27% | 17% | 18% | 14% | 8% | 13% |
| Sensible System COP (–) | 2.0 | 2.5 | 3.3 | 3.6 | 3.7 | 3.6 | 3.8 |
| Sensible System Capacity (kBtu/hr) | 64.1 | 92.8 | 114 | 129.4 | 140.6 | 148.1 | 159.3 |
| Ventilation Coil ΔT (°F) | 5.2 | 6.5 | 8.8 | 10.7 | 12.5 | 13.5 | 15.3 |
| Hourly Water Consumption (gal) | 0.2 | 5 | 17.5 | 23.2 | 26.3 | 27.3 | 30.8 |

## COMPARISON OF ENERGY USE WITH AND WITHOUT THE DUAL-EVAPORATIVE PRE-COOLER

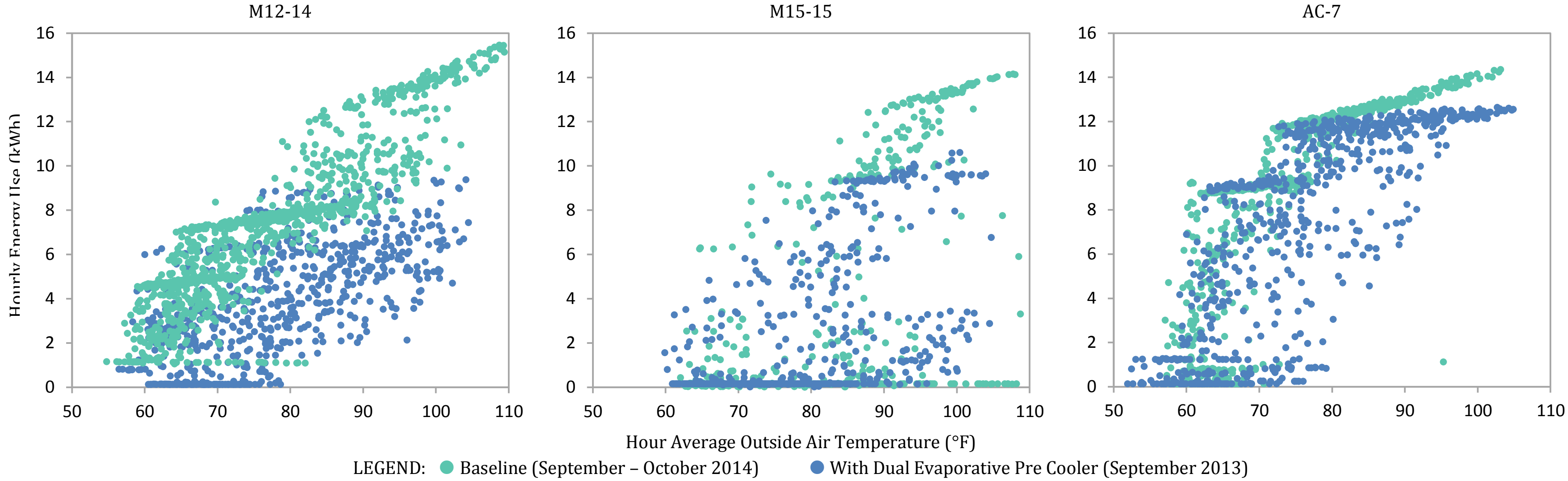


**FIGURE 8: PRE-POST COMPARISON – ENERGY USE SIGNATURE FOR EACH UNIT – THE HOURLY SUM OF ENERGY CONSUMPTION AS A FUNCTION OF HOUR AVERAGE OUTSIDE TEMPERATURE**

Figure 8 compares the energy use signatures for each system during September 2013 with a later period when the dual-evaporative pre-coolers had been disabled. The energy use signature plots the hourly sum of energy consumption as a function of the corresponding hourly average outside temperature. This analysis illustrates how the total system energy consumption changes with outside air temperature, and serves as a fingerprint for the whole system performance in a particular application. The energy use signature cannot be generalized for projection of efficiency gains in alternate applications because it is influenced by more factors than just the characteristic performance for the machine. However, it does allow for straightforward assessment of the impacts to energy use for a particular measure applied to a particular system.

The comparisons presented here captures all things that may have changed for each system between two periods – the primary difference is operation of the dual-evaporative pre-cooler. Notwithstanding any other minor differences between the periods, this pre-post comparison of energy use signature indicates substantial savings can be attributed to the dual-evaporative pre-cooler.

Below about 75°F the differences are minor, but as outside temperatures increase, energy savings clearly increase.  For periods above 100°F, M12-14 and M15-15 show hourly energy use reduction of 66% and 45%, respectively. Peak savings for AC7 was only about 13%; however this appears to result from the fact that the system is never able to meet set point for the kitchen during peak periods. Since the unit runs flat out regardless, the added capacity afforded by the dual-evaporative pre-cooler does not promulgate energy savings. Instead, part of the efficiency for this enhanced system is invested in an increased level of service.

## Cumulative Sensible Cooling

Equation 9 presents the cumulative amount of sensible cooling energy delivered by each hybrid RTU in each mode of operation and across a range of outside air temperature bins during the month of September 2013. Charts in the first column present the sensible system cooling effect, while those in column B present the sensible room cooling effect.

Notably, while the three units are nominally equal in capacity size – they are in fact all the same model number – they have different mode operating patterns and carry distinctly differ loads. The restaurant unit, AC7, provides 400% more sensible room cooling than the administrative office unit, M15-15. The security office unit, M12-14, carries 200% more sensible load than M15-15, and it tends to carry a substantial load 24 hours a day. The base load for sensible room cooling by M12-14 is approximately 16.67 *kBtu/hr*, which equates to a continuous internal heat gain from electronic equipment of 4.88 *kW*.

The dual-evaporative pre-cooler runs predominately during the day, though there are some overnight hours when it continues to operate. There are also some daytime periods when the pump does not operate because the outside air temperature is not warm enough. For M15-15 and AC7 the large majority of cooling occurs during hours when it is warm enough for the dual-evaporative pre-cooler to operate. For M12-14, which services continuous internal gains from electronics equipment, the cumulative amount of sensible cooling is split evenly between periods with and without the dual-evaporative pre-cooler.

For AC7, economizer mode is able to cover 20% of the room cooling needs during milder hours. Integrated economizer cooling is responsible for another 30-50% of the cumulative room cooling during these times. The remainder is covered by compressor modes, with and without the dual-evaporative pre-cooler. According to the intended changeover set points for each component, there should never be instances when compressors operate outside of an economizer and without the pump. However, there is a significant mismatch in changeover control for the economizer and pump, so there are many hours when this occurs.

The economizer-only mode does function for M12-14 and M15-15, but this mode carries very little of the cumulative room cooling load for either system. There is hardly any load for M15-15 during hours when economizer-only cooling would provide much benefit, and the internal gains for M12-14 are large enough that economizer-only cooling from this machine is usually not sufficient. Economizer cooling would be adequate for M12-14 for more of the overnight hours, but the supply airflow rate for this system was set so low that the room cooling capacity delivered by economizer-only mode was very small. Further, the integrated economizer modes for M12-14 and M15-15 were apparently not enabled – there were many hours for both machines when it would have been appropriate to operate with compressors and 100% outside air. M12-14 would have benefitted the most from this mode of operation, especially during overnight hours.

As one might expect, the cooling loads appear to peak sometime after the peak outdoor temperature. It is interesting to see that in all cases the cumulative system cooling load peaks 1-2 hours after the average peak ambient temperature, and the cumulative room cooling load peaks 1-2 hours later.

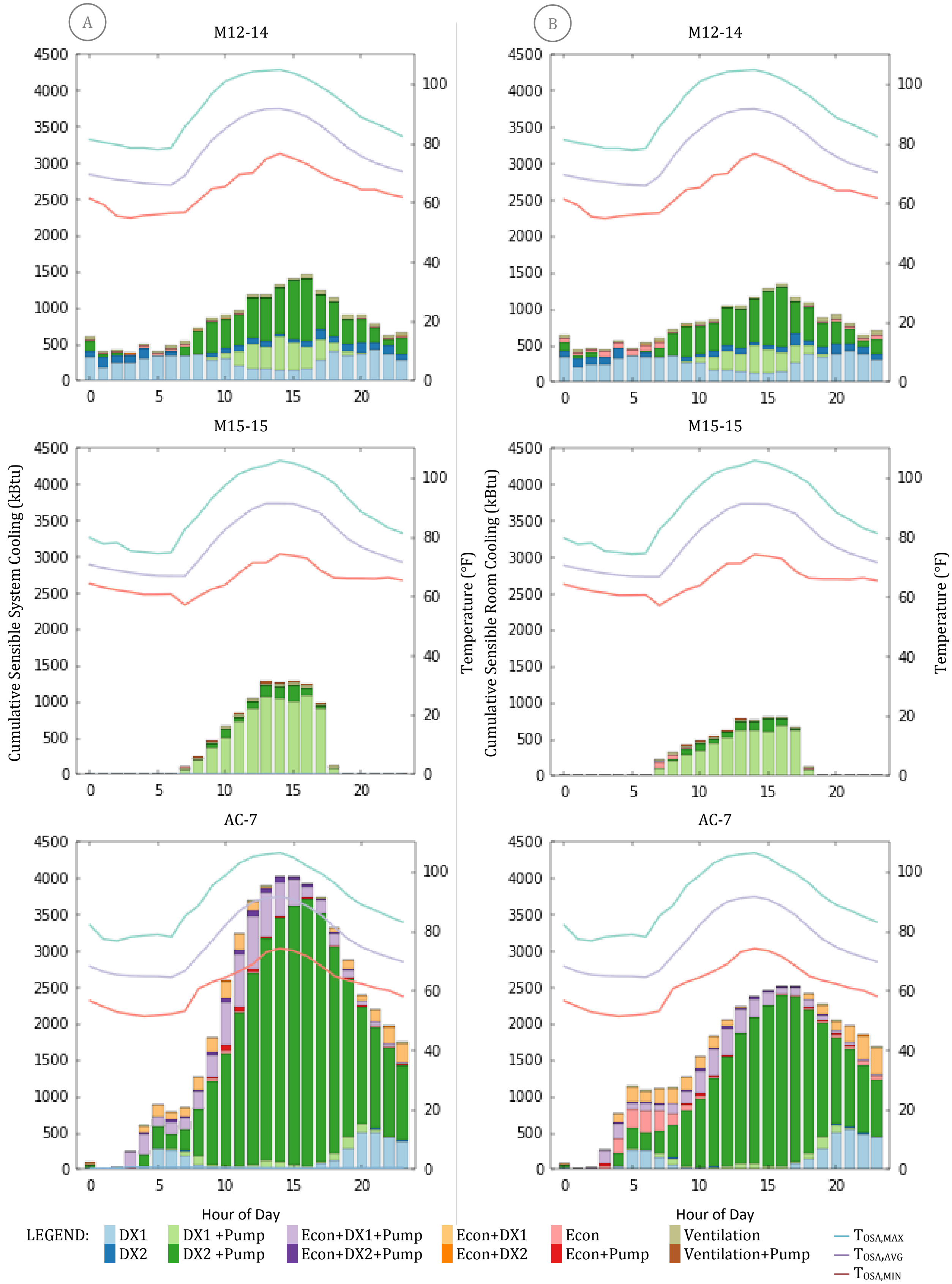


**FIGURE 9: DISTRIBUTION OF CUMULATIVE SENSIBLE COOLING IN EACH MODE AND MIN-MAX-AVERAGE OUTSIDE AIR TEMP. BY HOUR OF THE DAY FOR EACH UNIT. (A) SENSIBLE SYSTEM COOLING (B) SENSIBLE ROOM COOLING**

## COEFFICIENT OF PERFORMANCE

When compressors operate, measured sensible system COP ranges from 0.5–6.0 for the three systems tested. The COP for sensible room cooling ranges from 1.0–3.0 when compressors are operating, and reaches as high as COP=8 at 55°F in economizer-only mode. There are significant differences between each system tested. Figure 10 plots the efficiency for each system in each mode of operation as a function of outside air temperature. Plots in the first column (A) show the COP for sensible system cooling, while those in the second column (B) show the COP for sensible room cooling.

For AC7 there is a strong relationship between sensible system COP and outside temperature. More importantly, sensible system COP increases as the outside temperature increases. AC7 operated with a sensible system COP=1.0 at 65*°F* in mode "*Economizer+DX1*"; this increases to COP=4.0 at 100–105*°F* in mode "*DX2+DC*". This agrees nicely with laboratory tests that measured COP=4.0 at 105°F in the same mode.

Surprisingly, the sensible system COP for mode "*Economizer+DX+DC*" reaches 6.0 at 95*°F*. It is not clear why AC7 operated in an integrated economizer mode for some instances when outside temperature was well above a typical economizer changeover set point, and the efficiency gain is equally puzzling. Despite the high marks, operation in this mode doesn't necessarily result in savings because excess ventilation air introduces additional load. These instances are discussed further in the "*Irregularities in System Operations*" section.

In striking contrast to performance trends for AC7, the sensible system COP for M12-14 is mostly lower and not at all sensitive to outside air temperatures. The sensible system COP for this unit remained between 2.0–2.5 for outside air temperatures from 55–105°F. There is a minor downward trend with outside air temperature for the efficiency of compressor modes without the dual-evaporative pre-cooler, but for all periods when the pump operates sensible system COP in each mode remains steady. The difference between M12-14 and AC7 can be attributed to the supply airflow rate. M12-14 operated with approximately one-third the supply airflow rate, even though equipment was equally sized. The low supply airflow reduces both total cooling capacity and sensible heat ratio – this is especially clear by comparison of the two units in Figure 11.

Sensible room cooling efficiency is more consistent between units. In contrast to the trend for sensible system COP, the sensible room COP for AC7 is not especially sensitive to outside air temperatures. The sensible room COP in mode "*DX2+DC*" shows significant variation – measurements range between 1.5 – 2.75 – but the difference is not mainly caused by outside air temperatures. The variation can be attributed to outside air wet bulb temperature – a pattern that is discussed with greater detail in the "System Air Temperatures" section. This performance trend is a major shift from traditional air conditioner efficiency that tends to be poorest when outside temperature is highest.

Measurements of room cooling efficiency for AC7 in other modes presents some surprising results.

First, some of the instances for mode "*Economizer+DX1+DC*" at high outside air temperatures show very good performance – better than mode "*DX2+DC*" despite the fact that outside air is much warmer than return air. Measurements and calculations for these instances are correct, but the characteristic trend that can be seen in Figure 10 may be misleading. One should note that not all of the data from this mode of operation shows such compelling performance, and that most of the instances with very good efficiency are from periods shortly after operation in DX2. For these instances, it appears that there is some sensible capacity carryover from the previous mode of operation.

Second, at mild outside air conditions between 60–75*°F* "*DX1*" appears to achieve a better room cooling COP than mode "*Economizer+DX1*". It ought to be beneficial to operate with an integrated economizer at these conditions, but the results measured here indicate otherwise.

Finally, for M12-14, the only unit that spends an appreciable amount of time in first stage modes and second stage modes, it appears that DX1 is generally somewhat less efficient than DX2, across the entire operating range, and whether or not the dual-evaporative pre-cooler is running. This could be partly attributed to the fact the variable speed fan was not configured to change speed with each compressor stage, so fan power would be a larger fraction of the total consumption during first stage cooling.

The results presented in Figure 10 should be contrasted with performance for a conventional RTU of similar size that should have sensible room cooling COP 1.5-1.6 at peak conditions with a similar fraction of outside air. The conventional system would have a corresponding sensible system COP of 2.5-3.0.

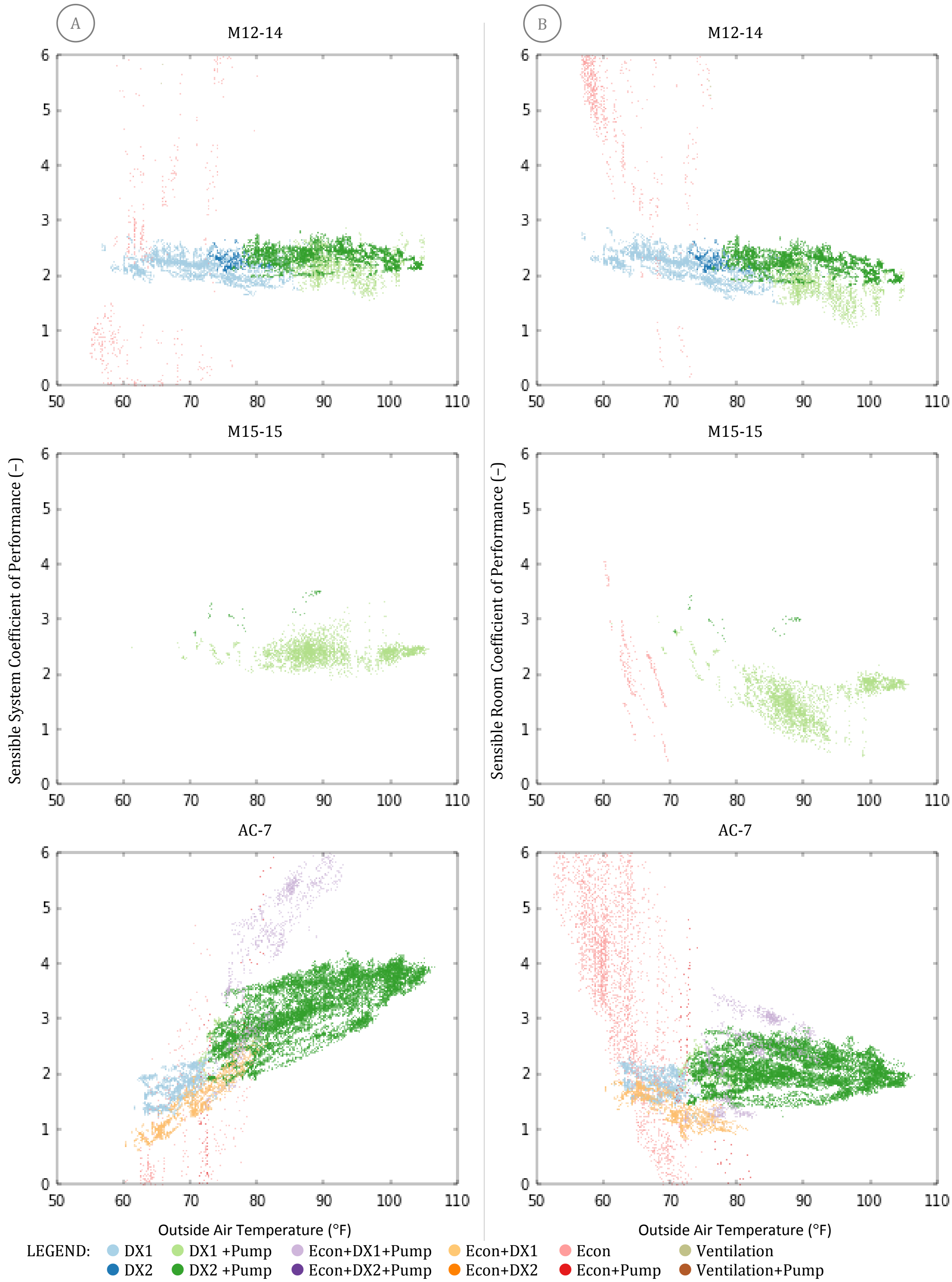


**FIGURE 10: COEFFICIENT OF PERFORMANCE FOR SENSIBLE COOLING IN EACH OPERATING MODE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE FOR EACH UNIT. (A) SENSIBLE SYSTEM COP (B) SENSIBLE ROOM COP.**

## Cooling Capacity

Figure 11 presents the sensible cooling capacity for each unit. The left column (A) includes plots for sensible system cooling capacity from each unit, while the right column (B) plots the sensible room cooling capacity.

Most importantly, operation with the dual-evaporative pre-cooler influences equipment performance such that room cooling capacity remains more or less steady across a range of outside temperatures. This is most apparent for AC7. The behavior is significantly different from what would be expected for a conventional rooftop air conditioner. For the amount of ventilation air supplied in these scenarios, sensible room cooling capacity for a conventional system would decrease by more than 30% on a shift from $70°F$ to $100°F$.

M12-14 and M15-15 exhibit a slight decrease in capacity as outside air temperature increases; the trend is more apparent for operation with one compressor. This pattern is most likely attributed to the fact that the water pumps for these two systems did not operate appropriately when only one compressor was active. This behavior is discussed with much greater detail in the "*Irregularities in System Operations*" section. The pump does operate intermittently for these periods, so these two units benefit somewhat from the dual-evaporative pre-cooler with only one compressor active, but capacity is effected compared to the trend for AC7.

For AC7, the sensible room cooling capacity for all modes with economizer decreases as outside temperatures increase. This is a result of operation with additional outside air, which has less and less capacity benefit as outside temperatures increase.

The variation in sensible room capacity for AC7 when in mode "*DX2+DC*" is much larger than what is observed for other modes and other units. This corresponds to similar observations for sensible room COP. The variation is attributed to differences in outside air wet bulb temperatures, an effect discussed in greater detail in the "*System Air Temperatures*" section.

At peak conditions the sensible system cooling capacity for M12-14 in mode "*DX2+DC*" is less than half of what was measured for AC7 in the same scenario. This difference is attributed to the very low supply airflow rate for M12-14, which reduces both heat exchanger effectiveness and sensible heat ratio. This adjustment was made by field technicians to overcome challenges with a problematic and non-functional multi-zone duct distribution system, however it resulted in a substantial decrease in system performance.

In economizer mode, supply air temperature is always warmer than outside air temperature as a result of fan heat, and due to leakage through the return air damper. For AC7, supply air temperature tends to be roughly $5°F$ warmer. As a result, the trend for sensible room cooling capacity drops below zero for outside temperatures as low as about 69°F. This also occurs, in part, on account of the fact that AC7 tends to operate with a relatively low return air temperature. In comparison, economizer mode for M12-14 always achieves a positive sensible room cooling capacity – supply air temperature is always cooler than return air temperature. This is partly because M12-14 operates with a smaller temperature rise across the fan – because airflow is so low on that unit – and partly because M12-14 operates with higher return air temperatures in the afternoons. The overall impact on room cooling from a mismatch between the economizer changeover set point and the actual room temperature is relatively small, but these instances do reinforce that a differential dry bulb economizer control could provide added savings (Taylor, 2010).

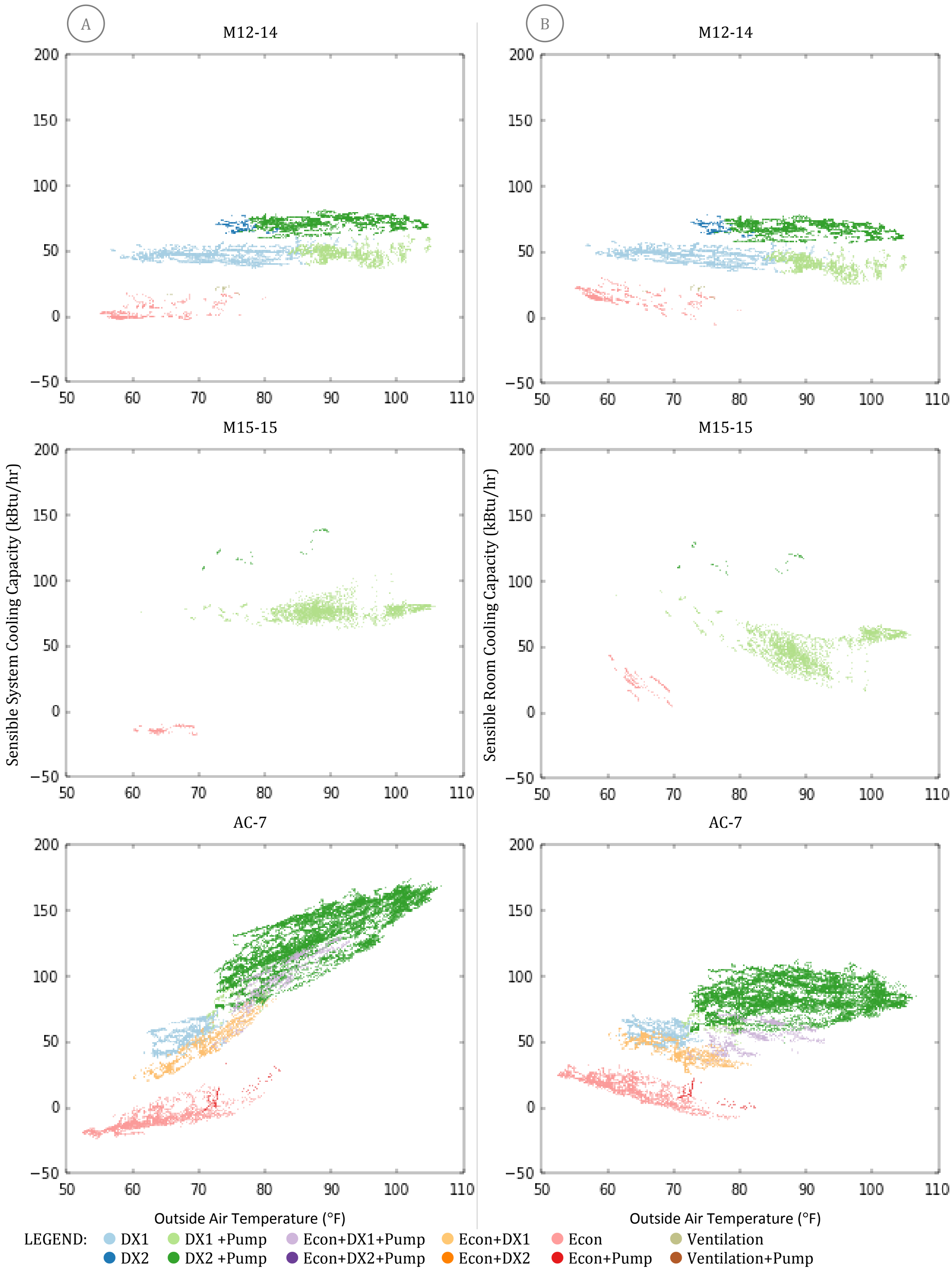


**FIGURE 11: SENSIBLE COOLING CAPACITY IN EACH OPERATING MODE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE FOR EACH UNIT. (A) SENSIBLE SYSTEM CAPACITY (B) SENSIBLE ROOM CAPACITY**

## CONTROL OF CHANGEOVER TO DUAL-EVAPORATIVE PRE-COOLING MODES

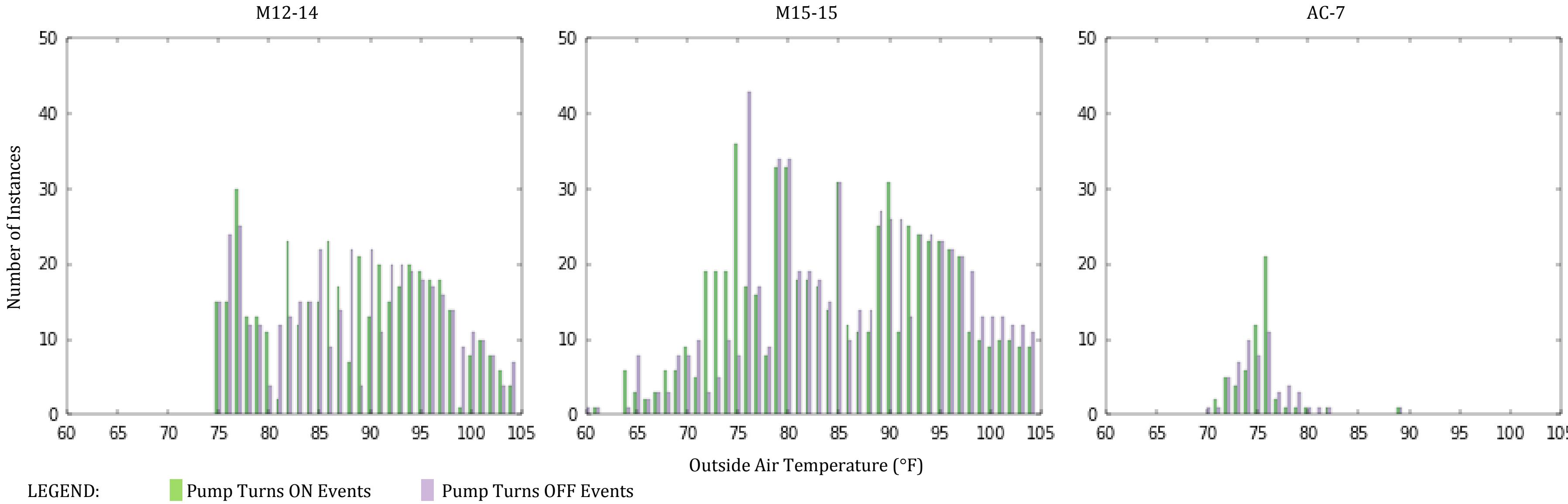


**FIGURE 12: DISTRIBUTION BY TEMPERATURE FOR THE NUMBER OF INSTANCES WHEN THE PUMP CYCLES ON AND OFF**

It is intended that the dual-evaporative pre-cooling pump only operates above a particular outside air temperature. The pump only switch on if the outside air temperature is above this point, and it should remain off for all periods when outside air temperature is below this point. Additionally, the two mall systems M12-14 and M15-15 incorporate controls that only allow the pump to operate whenever the supply fan is also operating. This feature keeps the pump from circulating unnecessarily while the rooftop unit cycles off. The manufacturer recommends 75°F as the changeover point.

Figure 12 plots the distribution by temperature for the number of instances when the pump turns on and off for each unit during the month of August. For AC7, these pump changeover events center mostly around 75°F, though there are a number of instances as low as 70°F, and some as high as 82°F. There are far more on and off events for the pumps on M12-14 and M15-15. This occurs because the pumps on these units cycle on and off in sync with supply fan operation. As long as the outside temperature is above the prescribed temperature the pump turns on when the fan turns on then turns off when the fan turns off. Since the thermostat controls for these units were set on "auto" mode, the fan and pump turn on and off every time the compressor cycles. As discussed in the "*Irregularities in System Operations*" section, the pumps also cycled irregularly during periods of steady compressor operation. The unintentional instances are not included in Figure 12. This figure only records the events that are associated with changing from a system mode that uses the pump, to a mode that does not use the pump. The transient temperature dynamics associated with such frequent pump cycling do impact thermal performance of the system.

M12-14 shows a clear and hard cutoff at 75°F, while M15-15 experiences some pump operation as low as 60°F. These observations indicate high variation for pump changeover for each unit, as well as between units.

## System Power Draw

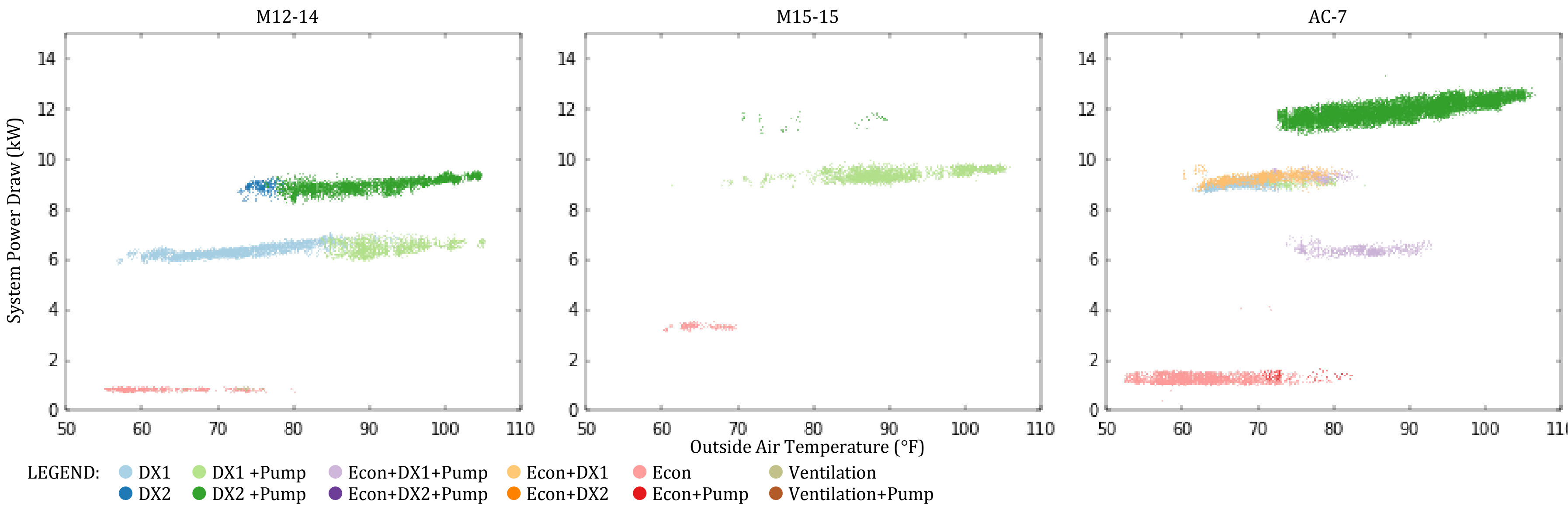


**Figure 13: System power draw in each mode of operation as a function of outside air temperature for each unit**

Figure 13 plots system power draw for each unit in each mode of operation. These charts illustrate the step change in power draw for each mode, and illustrates the sensitivity to outside air temperature. In economizer mode, power consumption is almost entirely from the supply fan, and the differences in power draw for this mode on each unit corresponds to the differences in supply airflow rate for each unit. In economizer mode, M12-14 moves approximately 1800 s*cfm* with 850 *W*, M15-15 moves 6,250 s*cfm* with 3,600 *W*, and AC7 moves 3,900 s*cfm* with 1,400 *W*. The differences in power draw for each unit in other modes is also due mainly to differences in supply airflow rate and the corresponding fan power. The effect of fan power associated with changing airflow rates can be seen especially well for AC7 in "*Economizer+Pump+DX1*" mode where the unit operates for some periods at 5,700 s*cfm* and for other periods at 3,650 s*cfm*. For these instances, there is a corresponding change in power draw of 3 *kW*.

For all three units, the first stage compressor draws between 5.5 – 6.0 kW on top of fan power, depending on outside air conditions. The second (smaller) compressor reliably adds an additional 2 *kW*. Power draw for operation with the dual-evaporative pre-cooler appears to be somewhat less sensitive to outside air temperatures than operation at lower temperatures with the pump off. Comparison of the trends with and without pump operation indicate that the system reduces power draw for compressors by 15-20%.

While the power draw for any particular outside air condition is less than it would be without the dual-evaporative pre-cooler, there is still some sensitivity to temperature because the wet bulb temperature tends to increase at the same time as the dry bulb temperature.

## WATER CONSUMPTION

Water consumption for each of the three systems was measured each minute during the period of observation. This measurement captured all water that was delivered from the city water supply to each of the systems. The measurement includes all water evaporated by each dual-evaporative pre-cooler, plus the volume of water drained from each system.

Since consumption of water for evaporation and for bleed only occurs when the pump is running, water consumption is restricted to periods with outside temperatures above the change point for each dual-evaporative pre-cooler. For M12-14 and AC7 this change point is very close to 70°F, while the pump for M15-15 was observed to run as low as 60°F. This low temperature operation costs pump energy but provides very little cooling and uses very little water.

Figure 15A (next page) charts the total daily water consumption by each unit for each day in September 2013. The chart also shows the minimum, maximum, and average outside air temperature for each day. Overall, total water consumption for a particular unit depends most significantly on the outside temperature, the bleed rate set for each system, and the amount of time spent with the pump operating. The outside humidity, the ventilation rate, the sensible load transferred into the water circuit, and the water temperature should also have important but lesser effects. For the three systems studied here, the total amount of water consumed in each hour trends most closely with outside dry bulb temperature. Figure 15B (next page) plots the hourly water consumption for each unit as a function of the hourly average outside dry bulb temperature. The aforementioned variables, and transient dynamics lead to some scatter in the data, but the major trend is an increase in hourly consumption as outside temperatures increase.

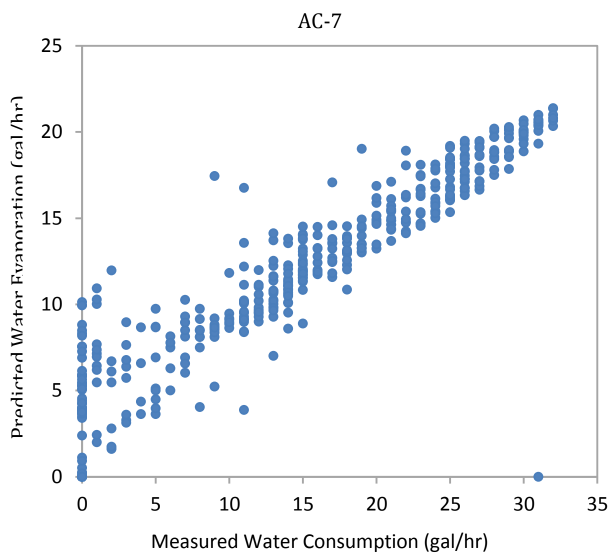


**FIGURE 14: PREDICTED EVAPORATION AND MEASURED WATER CONSUMPTION**

The bleed rate plays an important role for the total amount of water consumed by the dual-evaporative pre-cooler. A continuous water bleed is used to manage the concentration of dissolved solids in order to avoid deposits of mineral scale throughout the system. The bleed rate for the equipment is adjusted by a manual valve that allows some water to drain from the water circuit any time the pump circulates water. Since this is a manually set valve position it is prone to maladjustments by field technicians. When the research team arrived to install instrumentation for the three units in Ontario, the ball valve controlling bleed rate was found fully open. The bleed rate was adjusted at that time and the valve handle was removed to discourage future manipulations. Other field evaluations of this technology have observed that an excessive bleed rate can result in overall water use that is more than twice as large as the predicted water use for evaporation. Another study also observed significant changes in bleed over time.

The amount of bleed water required will be different for each site, but it is reasonable to expect that a system should consume 10-30% more water than what is needed for evaporation. Water use for the systems studied in Ontario appears to be within this range. Figure 14 compares the measured hourly water consumption for AC7 against the predicted hourly water evaporation for the unit. The prediction for water evaporation is based on the water use for a direct evaporative cooler with wet bulb effectiveness of 50% and the manufacturer stated condenser airflow rate of 11,000 *cfm*. Fifty percent wet bulb effectiveness is representative of the laboratory measured performance for this system. Over the period of observation the total water consumption was 25% larger than the predicted evaporation rate. The fraction of water used for maintenance appears to be reasonable for M12-14 and M15-15 as well since these two units follow a similar trend for water use as a function of outside air temperature.

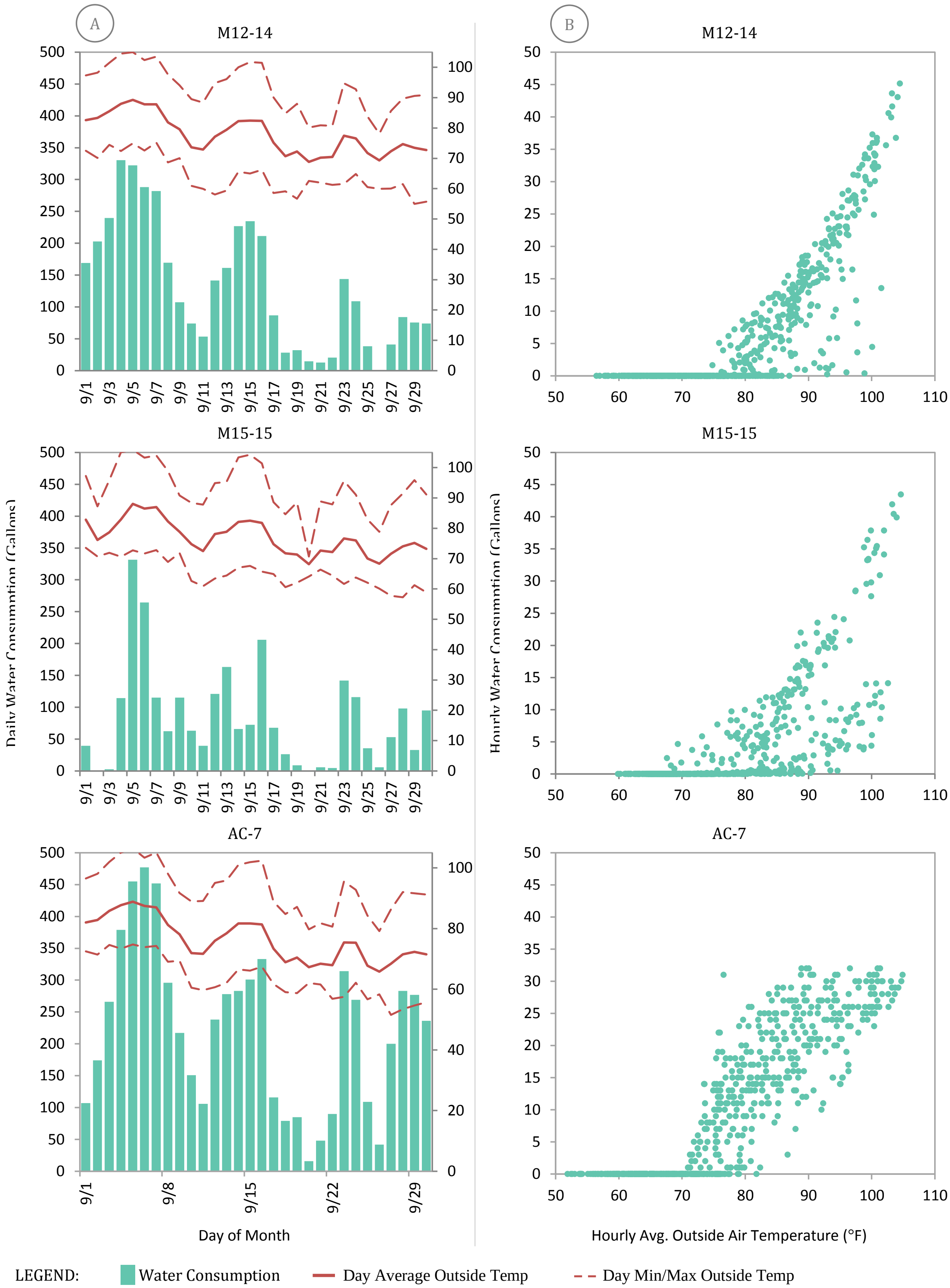


**FIGURE 15: WATER CONSUMPTION FOR EACH SYSTEM IN AUGUST 2013 (A) DAILY WATER CONSUMPTION (B) HOURLY WATER CONSUMPTION AS A FUNCTION OF OUTSIDE AIR TEMPERATURE**

## WATER TEMPERATURES

Figure 16 (next page) describes the water side temperature performance for each dual-evaporative pre-cooler system. Figure 19A plots the sump water temperature for each minute interval observation as a function of outside air wet bulb temperature. Figure 16B plots the water temperature rise across the ventilation cooling coil in each mode with pump operation. Both figures plot steady state measurements after consistent operation in a particular mode for at least ten minutes.

Again, since the pump on AC7 operated most consistently, this unit provides the clearest representation of performance for this type of hybrid rooftop air conditioner. Despite the fact that the evaporative cooling process must reject the sensible load acquired from cooling outside air (OSA), as long as the condenser fans were operating water returned to the sump very near the ambient wet bulb condition. For steady state operation, the sump water temperature remained within 1-2 °F of the outside air wet bulb temperature. These observations are consistent with laboratory measurements. When the dual evaporative pre-cooler pump functioned, but there was no condenser airflow, sump water temperatures increased substantially.

Since the pumps for M12-14 and M15-15 operated irregularly when only one compressor was active, the sump water temperature for the two units in “DC+DX1” mode shows quite a bit of scatter. There are still some periods for both units when sump water temperature approaches the wet bulb limit, indicating that for instances when the pump did run long enough, these units exhibited appropriate thermal performance capabilities. M12-14 spent a significant amount of time in “DC+DX2” mode, and since the pump operated continuously whenever the second stage compressor was on, the trend for sump water temperature in this mode remains within a couple of degrees of the concurrent wet bulb temperature.

Figure 16B plots the water temperature rise across the ventilation cooling coil as a function of outside air wet bulb depression. Observations for AC7 present a very clear trend for each mode of operation and reach a water temperature rise of 9°F with wet bulb depression between 30–35°F. The temperature rise in “Econ + DC” mode is significantly lower as a result of the fact that the condenser fans do not operate in this mode, and so the evaporative cooling capabilities for the system are greatly reduced. These results also show that water temperature rise is 1–2°F higher for “Econ+DC+DX1” mode than for “DC+DX2” mode. This occurs because “Econ+DC+DX1” mode operates with the outside air damper fully open, which increases the airflow rate across the coil and allows for a larger temperature rise for the water flow. It is important to note that the additional sensible load resulting from increased ventilation airflow does not affect the sump water temperature trend.

M15-15 shows two distinct trends for “DC+DX1” mode in Figure 16B. One trend surrounds a 0°F water temperature rise across the coil, while the other, albeit with quite a lot of scatter, indicates temperature rise in the same range as was observed for AC7. The 0°F trend occurs as a result of periods when the pump fails to move water through the coil, as discussed in the “Irregularities in System Operations” section. The second trend emerges from the instances when the pump does finally manage to circulate water.

M12-14 shows a clear trend for water temperature rise across the coil for “DC+DX2” mode since the pump for this system operated consistently in that mode. However, the trend for temperature rise on this system is much smaller than what was observed for the other systems, and much smaller than what has been observed in laboratory tests and other field evaluations. The small temperature rise can be attributed to the very low supply airflow rate for this system, which results in a ventilation airflow rate that is less than 25% of what was observed for M15-15 and AC7.

These observations highlight the need for proper setup and commissioning of the baseline vapor compression equipment in combination with any installation of the dual-evaporative pre-cooler system. The very low airflow rate on M12-14, and the inconsistent pump controls on both M12-14 and M15-15 limited the value of the dual-evaporative pre-cooler. The results for AC7 agree very nicely with laboratory measurements and other field evaluations for this technology.

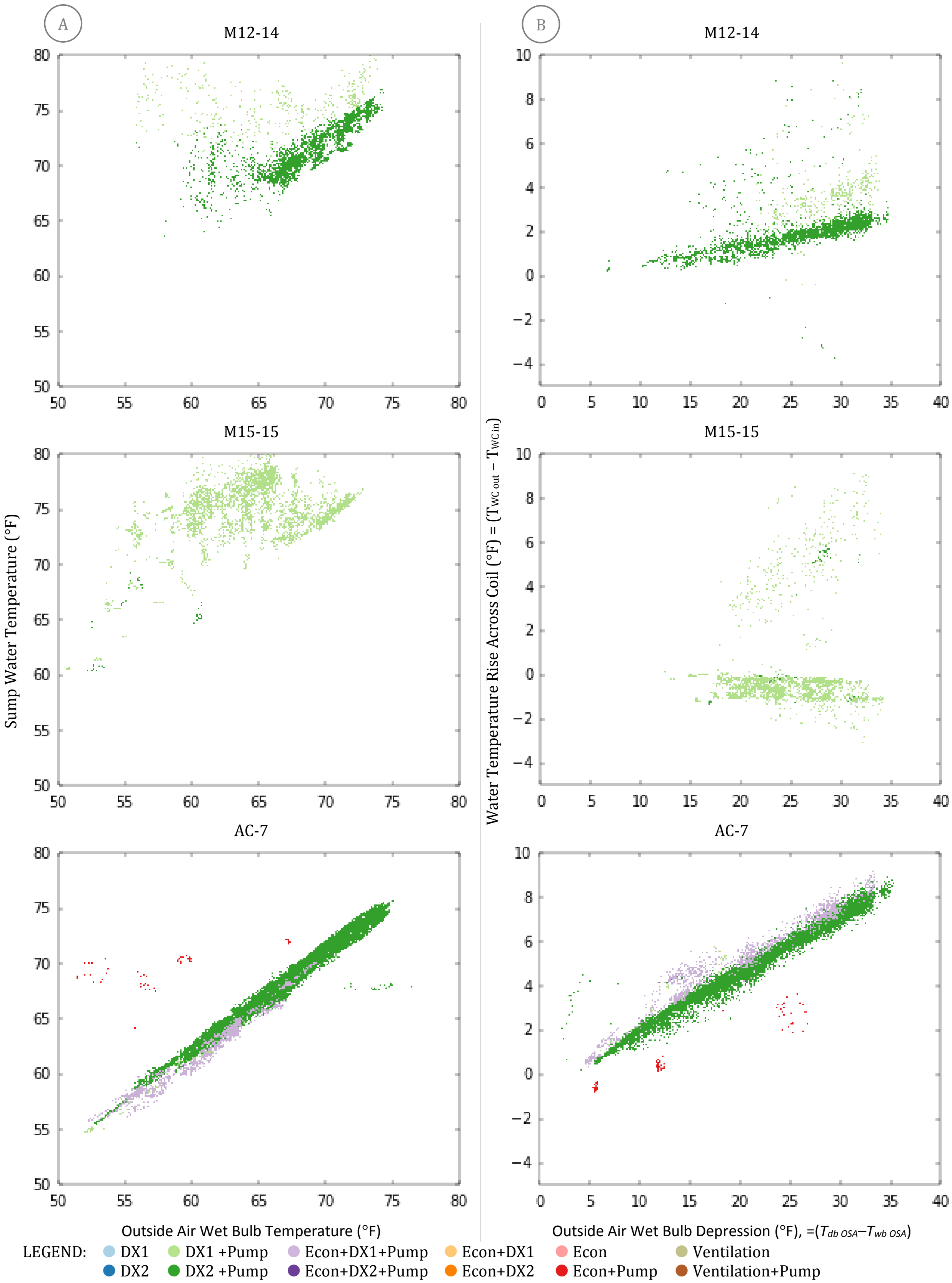


**FIGURE 16: WATER TEMPERATURES FOR THE DUAL-EVAPORATIVE PRE-COOLING CIRCUIT ON EACH SYSTEM.(A) SUMP WATER TEMPERATURE (B) WATER TEMP. RISE ACROSS VENTILATION COOLING COIL**

## SYSTEM AIR TEMPERATURES

### *SUPPLY AIR TEMPERATURE*

Figure 17A (next page) presents supply air temperature in each mode of operation for the three systems.

M12-14 runs with a very low supply airflow, which results in a very low supply air temperature. During the period of observation presented, the unit generates supply air as low as 36°F in mode "DX2+DC". Most of the time this mode generates supply air between 40-45°F. This is detrimental to system efficiency. The other units in this study operate with a sensible system COP that is 50% higher than M12-14 (refer to Figure 10). Although the total system power draw with second stage compressor is approximately 25% lower than other units, the sensible system cooling capacity is reduced by 50%. This occurs because the low airflow and low supply air temperature decrease vapor compression cooling capacity and result in a significant amount of unnecessary dehumidification. Since the unit must still cover all room cooling loads, and because distribution efficiency is reduced as a result of increased thermal losses, this unit must run longer hours to satisfy the thermostat, which results in a higher aggregate demand on peak. These observations underscore the fact the dual-evaporative pre-cooler studied here is only one aspect of an effective efficiency program, and that deployment of such a technology ought to be coordinated with efforts for quality maintenance and performance optimization for the existing equipment and building systems.

M15-15 shows two distinct bands for supply air temperature in the "DX1+DC" mode. This occurs because the first five days in the period of observation presented here operated with the outside air damper fully closed. Review of the monitoring data indicates that the ventilation set point was adjusted on Sept. 5. For all observations before this date (July – August) the system had been operating without ventilation air. While the outside air damper was closed, the unit operated with a somewhat lower supply airflow rate because the overall airflow resistance was higher than during periods when the outside air damper was partly open. The lower supply airflow results in a correspondingly lower supply air temperature for this period. The separate bands seen for supply air temperatures in Figure 17A can also be seen in the plot of sensible cooling capacity (refer to Figure 11), and in Figure 17B, which shows that the air temperature difference across the ventilation cooling coil was zero for the period when the damper was closed.

Supply air temperature changes significantly with ambient conditions, room conditions, and operating mode. As illustrated in Figure 17A, supply air temperature for AC7 ranges between 48-67°F when operating in "DX2+DC" mode. For any particular outside air temperature, supply air temperature for this unit spanned a range of 10-15°F. This is a surprisingly large distribution for supply air temperature compared to what would be expected from a conventional air conditioner and does not present a particularly strong trend to support accurate regression analysis of supply air temperature as a function of outside air temperature.

### *VENTILATION PRE-COOLING TEMPERATURE*

Figure 17B (next page) presents the temperature difference across the ventilation cooling coil in each mode that uses the pump. Temperature of the ventilation air is determined according to Equation 12, by performing an energy balance for the air and water streams across the ventilation cooling coil. Since the mass flow rate of ventilation air for M12-14 was so low (464 scfm), the uncertainty for calculation of product air temperature was ±31.2°F, which is unacceptably large. However, both M15-15 and AC7 operate with a significantly larger ventilation air mass flow rate. The calculated uncertainty for the temperature difference across the ventilation coil for M15-15 and AC7 is ±4.3°F and ±0.9°F, respectively.

AC7 operated most consistently and showed the type of temperature characteristics that should be generally expected for a hybrid RTU with dual-evaporative pre-cooling. The air temperature difference across the ventilation cooling coil correlates solidly with the outside air wet bulb depression for any particular mode of operation. Similar to wet bulb effectiveness, the temperature difference across this cooling coil is also impacted by the ventilation airflow rate. When the unit operates with one hundred percent outside air, the temperature split declines as a result of increased airflow. In these cases, the ventilation cooling capacity actually increases somewhat, however there is no benefit to using additional outside air preferentially since the overall effect of increased ventilation air will be an increase in energy use.

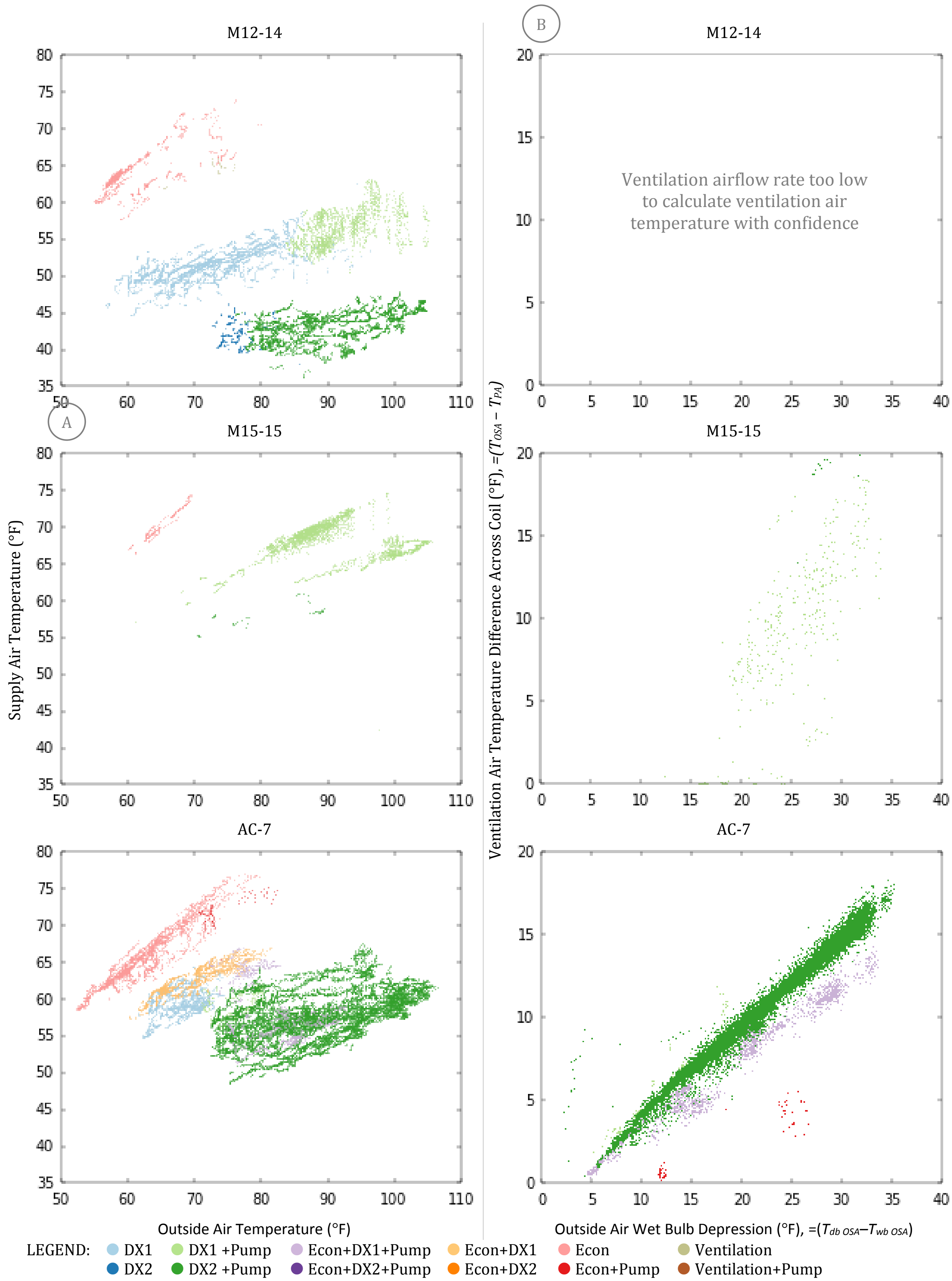


**FIGURE 17: AIR TEMPERATURE PERFORMANCE FOR EACH SYSTEM (A) SUPPLY AIR TEMPERATURE, (B) TEMP. DIFFERENCE ACROSS VENTILATION COOLING COIL**

The fact that ventilation air temperature split changes with outside airflow rate, effects the selection of an optimal economizer changeover point. Since the product air from the dual evaporate pre-cooler is lower than the outside air temperature, there is an opportunity to raise the economizer changeover set point. However, since the product air temperature rises with a switch to 100% outside air, the economizer should not be controlled to change over as soon as the product air is cooler than return air. The optimal control scheme would modulate outside air damper position to keep product air temperature below the return air conditions. In lieu of this ideal control scheme, we recommend a simpler approach – to shift the changeover temperature on a typical dry bulb economizer controller to 80°F.

Further, when the pump operates in sequence with economizer mode and without compressor operation, the temperature split for AC7 is even smaller. This can be attributed to the fact that the condenser fans do not run in this mode. It may be beneficial to operate condenser fans without the compressors to gain the advantage of the added evaporative effectiveness. This will only be true if the added cooling capacity comes at a lower energy expense than would cooling capacity from compressor operation. The sensible room cooling COP in this extended economizer mode is already significantly higher than the efficiency for a simple economizer, but if the condenser fans were controlled to operate in sequence, the cooling capacity in this mode could be more than doubled with the addition of only 350 $W$ condenser fan energy. While MC12-14 and MC15-15 have the capability to operate the pump and condenser fans independent from compressor operation – the condenser fans can be enabled by a relay on the controller – neither system ever operated in such a mode.

### *Relationship Between Supply Temperature and Environmental Conditions*

For conventional systems, supply air temperature for a particular unit trends most closely with outside air temperature. However, for the hybrid systems studied here, supply air temperature trends much more closely with outside air wet bulb temperature. This results partly from the fact that output from the direct evaporative condenser air pre-cooler is driven by the wet bulb temperature. The sump water temperature is effected by the wet bulb temperature, and so the temperature of ventilation air is driven by the wet bulb temperature. A lower wet bulb temperature results in lower ventilation air temperature, reduced mixed air temperature, and lower condenser air temperature. All of these factors tend to drive a lower supply air temperature. Figure 18A plots the supply air temperature for AC7 as a function of the corresponding outside air wet bulb temperature for each minute interval in "DX2+DC" mode. There is much less variation in this trend than there is for supply air temperature in Figure 17A. Figure 18B plots the time series trend for outside air dry bulb temperature, outside air wet bulb temperature, return air temperature, supply air temperature and system power draw from three days in September 2013. The patterns illustrated in this chart indicate that supply air temperature responds most directly to the outside air wet bulb temperature.

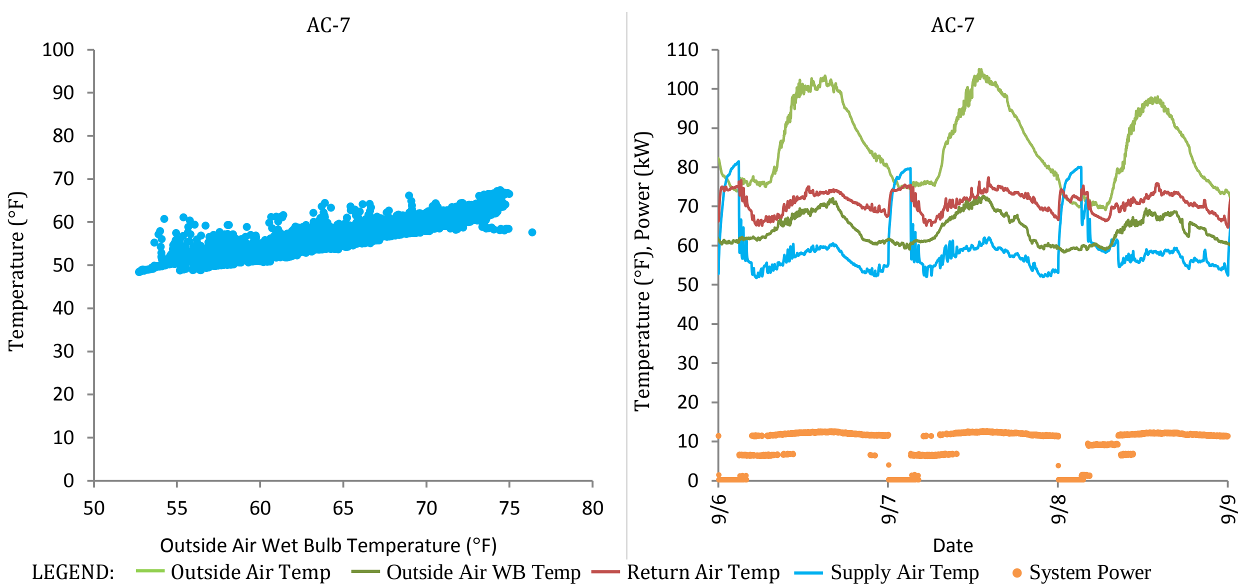


**Figure 18: (A) Supply air temperature in "DX2+DC" mode as a function of outside air wet bulb temperature (B) Time series trend for system temperatures and power draw for three days in September 2013**

## VENTILATION COIL PERFORMANCE

Wet bulb effectiveness for ventilation air cooling is a measure of how near the product air temperature at the outlet of the ventilation cooling coil approaches wet bulb for the outside air. Wet bulb effectiveness is defined by Equation 13. Sensible heat exchanger effectiveness is a measure of how near the product air temperature at the outlet of the ventilation cooling coil approaches the theoretical maximum heat transfer potential for the air flow and water flow at the inlets to the heat exchanger. Sensible heat exchanger effectiveness is defined by Equation 15. Figure 19 A (next page) plots the wet bulb effectiveness metric for each unit, and Figure 19 B (next page) plots the sensible heat exchanger effectiveness. Since the ventilation airflow rate on M12-14 was so low, the uncertainty for these calculated metrics is too high to warrant consideration.

For AC7, wet bulb effectiveness is grouped tightly between 45–55% when operating in “DC+DX2” mode with wet bulb depression above 10°F. Wet bulb effectiveness declines sharply when wet bulb depression is below this point. This observation agrees well with previous laboratory results (Woolley, 2012). The “Economizer+DC+DX1” mode operated with a lower wet bulb effectiveness between 30–40%. This is a result of the much higher outside airflow rate in this mode. As airflow across the coil increases, product air temperature increases and wet bulb effectiveness decreases. This phenomenon is also apparent in the previous assessment of temperatures split across the ventilation cooling coil as presented in Figure 17B.

As a result of these observations, we recommend that any simulation effort that intends to model performance of these dual-evaporative pre-cooling systems using a wet-bulb effectiveness metric should not use a fixed value for wet bulb effectiveness. Instead, the performance metric should be allowed to change as a function of airflow, water flow rate, and outside air wet bulb depression. The results presented in Figure 19 (next page) offer valuable insight into the ways that wet bulb effectiveness for ventilation cooling behaves across a range of ambient conditions, and system operating parameters. This data should serve as a strong basis for validation of models designed to simulate performance of this dual-evaporative pre-cooler technology.

Heretofore, modeling efforts have used very simple assumptions about the wet bulb effectiveness for indirect evaporative cooling from these systems. These results indicate that those previous efforts have overestimated performance in some instances and underestimated performance in others. These new results provide the opportunity to develop a more sophisticated empirical model of behavior for these systems that can be used by EnergyPlus modeling efforts such as those being advanced by NREL’s Technology Performance Exchange (TPEx) (NREL 2014) or by Lawrence Berkeley National Library (LBNL) and UC Davis with the Hybrid Black Box Model (HBBM) for EnergyPlus (Dutton 2014).

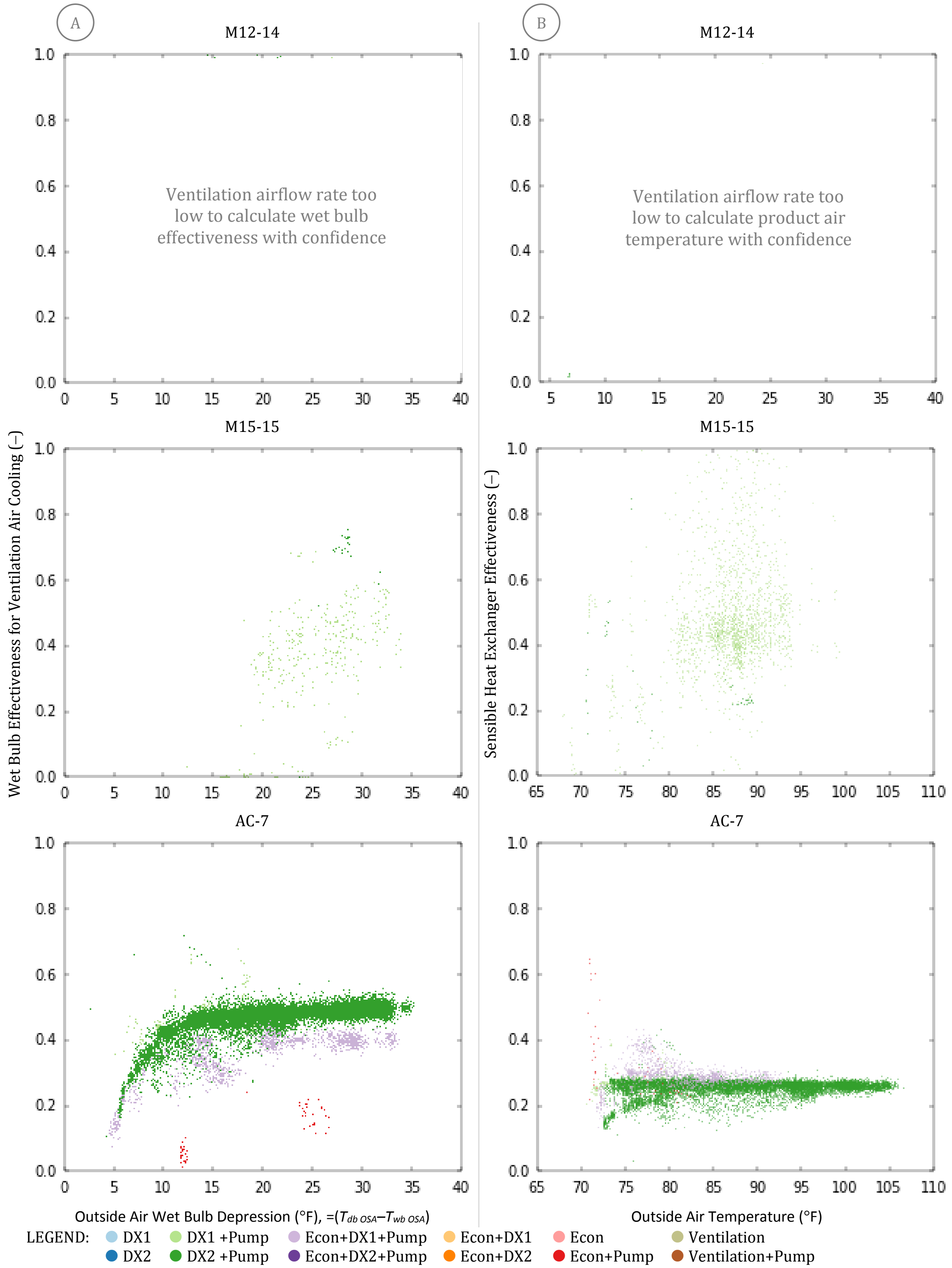


**FIGURE 19: VENTILATION COOLING COIL PERFORMANCE FOR EACH SYSTEM**
**(A) WET BULB EFFECTIVENESS FOR VENTILATION AIR COOLING, (B) SENSIBLE HEAT EXCHANGER EFFECTIVENESS**

## IRREGULARITIES IN SYSTEM OPERATIONS

The research team observed a few irregularities in system operation for each unit that deserve discussion and explanation. These irregularities are separate from the differences in supply airflow rate for each system, the differences in ventilation rate, and intentional differences in control scheme for the dual-evaporative pre-cooler. These inconsistencies caused confusion for the research team because they had significant impacts to system performance, but could not be explained immediately by differences in system design or setup. The following sections highlight irregularities that stood out for each system, and discusses the impact of each.

### *INTERMITTENT PUMP OPERATION FOR M12-14*

M12-14 employed a control strategy designed to operate the pump only when both the supply fan was running and the outside air temperature was above 75°F. In theory, this would cause the pump to turn on once in the morning after the fan had begun operating and after the outside air temperature rose above the changeover point, then turn off in the evening once the outside air temperature lowered below the changeover point or as soon as the supply fan turned off. However, in addition to the fact that this unit was controlled in "auto" mode as discussed in the "*Control of Changeover to Dual-Evaporative Pre-Cooling Modes*" section, data indicates that the pump on M12-14 also cycled spasmodically during warm periods with fan and compressor operation.

After much sleuthing, the research team traced this behavior to a poor reading from the current switch on the supply fan (see Figure 2 for controls schematic). When the current switch successfully output a signal, the unit controller would enable pump operation. However, the output signal from the current switch appears to have wavered between ON and OFF repeatedly even while the fan ran continuously. Except for the fact that the behavior occurred more often during first stage cooling than it did during second stage cooling the events do not seem to correspond to any other operating variables or environmental conditions. Figure 20A plots minute-by-minute measurements for ampere draw for the pump, system power draw, and outside air temperature for September 15. Figure 20B focuses on power draw and pump amperes from 09:30 – 14:30.

Figure 20 shows that the pump on M12-14 cycled intermittently without much explanation whilst all other components operated as expected. This discontinuous pump operation resulted in large transient variations in water circuit temperatures when the pump did start, then by the time water temperatures reached steady state the pump usually cut out. When water is not flowing, the ventilation cooling load increases, and the system does not benefit from evaporative condenser air cooling. Accordingly, Figure 20B shows a cyclic rise in system power draw associated with the pump cycling off, as a result of these decreases in performance.

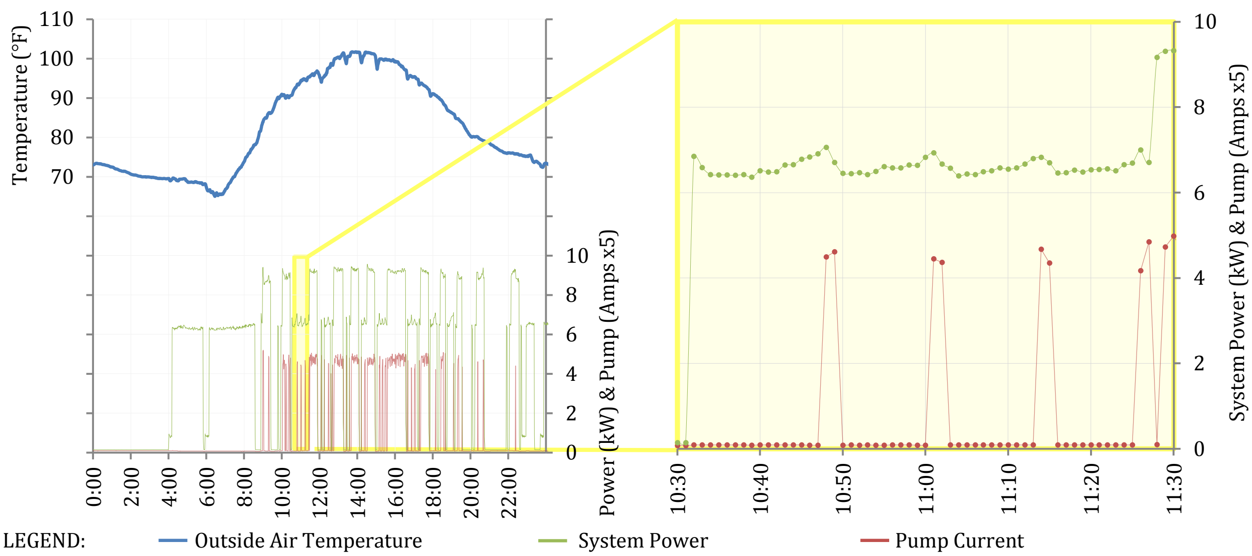


**FIGURE 20: INTERMITTENT PUMP OPERATION FOR M12-14**

### *INTERMITTENT PUMP OPERATION FOR M15-15*

The dual-evaporative pre-cooler pump also operated intermittently for M15-15, however the behavior appears to have occurred for a different reason than the patterns observed for M12-14. For M15-15, while the pump appears to run continuously for all periods with compressor operation, there are some periods when the pump draws approximately 3 *amps*, and other periods when the pump only draws 2 *amps*. The instances of high current draw correspond to periods when the water coil inlet and outlet temperatures remain roughly equal. These observations indicate that although the pump is drawing current during these periods, there is no water flow through the coil.

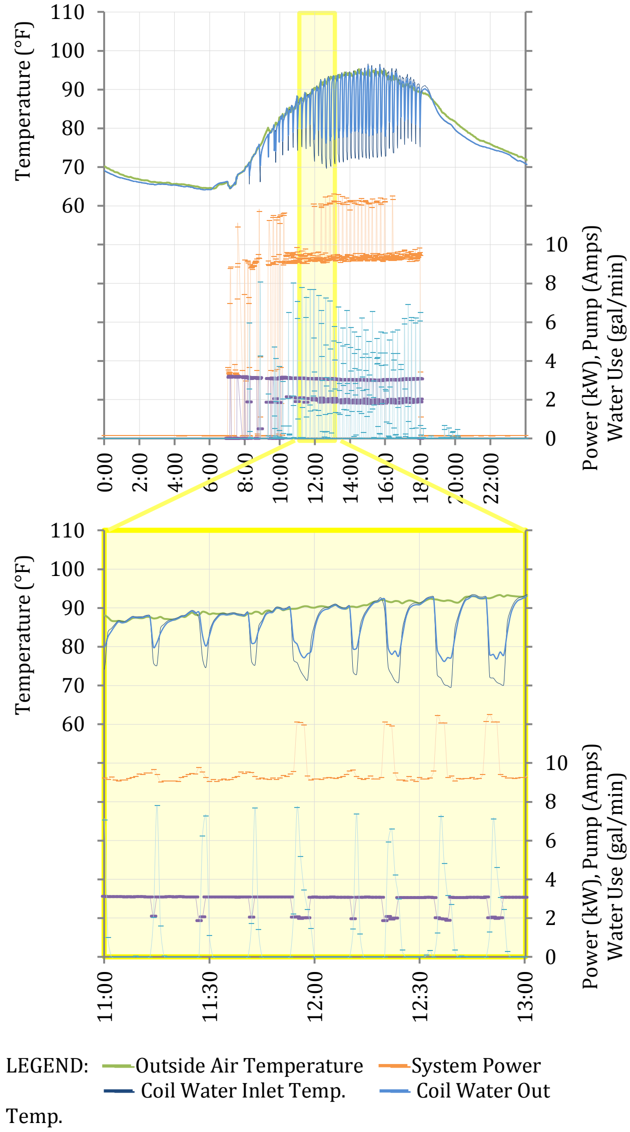


FIGURE 21: INTERMITTENT PUMP OPERATION FOR M15-15

This high-amp-draw and zero-flow scenario is consistent with a pump that has lost its prime, or is experiencing electrical issues. However, it is not clear exactly why this behavior occurred. The pump did operate correctly for intermittent periods each day, but it was rare for the pump to operate steadily for longer than one hour. On most days in the 2013 cooling season, the pump only moved water for several minutes at a time. For the day plotted in Figure 21, Sept. 13, 2013, while the pump was active and drawing amperage from 07:00 – 18:00, water only flowed intermittently throughout the day. It should be noted that the water always flowed when the second compressor was activated, but cycled inconsistently whenever the first compressor operated alone. The second compressor never ran for more than several minutes at a time and flow through the pump always ceased once the unit returned to first stage.

Make up water consumption corresponds to those periods when the pump is running and bleeding water. As soon as the pump manages to move water, the sump level declines rapidly and new water is let in through the float valve. This behavior is illustrated in Figure 21 where every occurrence of pump flow is met with a pulse of water use.

Similar to M12-14, M15-15 also exhibits a rise in power consumption that corresponds to periods that water does not flow. The recurring rise in power consumption can be seen in Figure 21.

The magnitude of the rise in power consumption corresponds to the length of time that the pump does not flow, which indicates that the evaporative media continues to have some effect for several minutes after the pump turns off. Of course, this effect erodes over time as the media dries out. Overall, the intermittent pump operation does not allow the system to achieve its full steady state performance potential. It should be noted that M12-14 and M15-15 are the only two units from multiple field evaluations that exhibited these pump failures; they are also the only two units to use the custom programmable unit controller described earlier, instead of the manufacturer's simpler standard controls approach use for the restaurant unit. These patterns are present for both units for all observations from 2013. Observations from 2014 indicate that the pump operated with more consistency, in synch with compressor operation, as intended. There was no known repair or adjustment made to either system during the time in between.

### *HIGH AMBIENT ECONOMIZER OPERATION FOR AC7*

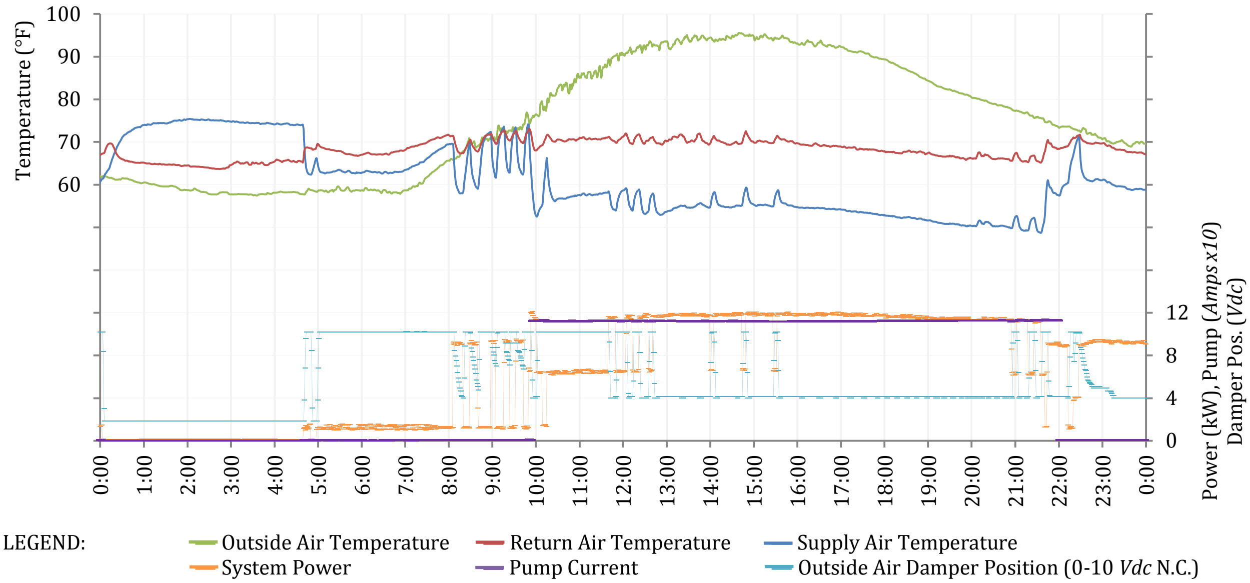


**FIGURE 22: UNUSUAL ECONOMIZER OPERATION FOR AC7**

The dual-evaporative pre-cooler at the restaurant operated appropriately during the period of observation presented here. However, that rooftop unit cycled between operating modes in a surprising way. Most importantly, it was found that the system operated in an integrated economizer mode at outside air temperatures well above an appropriate economizer changeover set point. This behavior can be seen throughout the characteristic performance plots presented earlier, especially in Figure 13, where the system power draw in this mode clearly stands apart from the other expected modes of operation.

Figure 22 plots a time series trend for several system variables for Sept. 23, 2013 to illustrate the behavior. On this day, the unit started up at about 04:40 while the outside temperature was below 60°F. The system ran continuously in an economizer-only mode from 05:00–08:00 while the outside temperature rose to approximately 65°F. At this point, economizer-only mode did not provide adequate cooling capacity to satisfy the indoor loads, but outside air was still cooler than return air, so the unit switched to an integrated economizer mode with first stage compressor. In this mode, the unit modulates outside air damper position even though the outside temperature is still below the economizer changeover temperature – this may be the result from a control sequence designed to keep supply air temperature from dropping too low. Once the room set point was satisfied, the unit switched back to economizer-only mode. This cyclic behavior persisted until about 10:00, about the same time that the outside temperature increased above both the economizer changeover point and the control point for the dual-evaporative pre-cooler pump. The pump turned on at this time, and after switching briefly to second stage cooling with a minimum ventilation rate, the system chose to operate in an integrated economizer mode again with the first stage compressor. This time, the fan speed was reduced. This mode persisted beyond 11:30 by which time the outside temperature had increased to approximately 90°F. At about 11:40 the unit closed the outside air damper to a minimum ventilation position, and engaged the second stage compressor. Operation for the second stage compressor persisted for the rest of the afternoon, except for several brief periods of approximately 10 minutes when the unit chose to switch back to an integrated economizer mode with the first stage compressor and a reduced fan speed.

While sensible cooling capacity decreases during these high ambient economizer periods, it is surprising to note that the sensible room cooling efficiency actually tends to increase slightly compared to operation with second stage compressor and minimum ventilation. The increase in efficiency most likely results from reduced fan power draw, in combination with a thermal energy carry over from the previous mode, and likely also some latent to sensible energy recovery by evaporation off the coil. It is not clear whether or not this switch to an economizer at high ambient conditions is an intentional mode of operation, but the example shown was typical behavior throughout the period of observation.

## MAINTENANCE AND RELIABILITY FOR EVAPORATIVE COMPONENTS

Evaporative cooling systems introduce some needs for service and maintenance that are not required for conventional rooftop air conditioners. While these technologies can provide substantial energy impacts, proper service and maintenance over the long term is essential to ensure persistent savings. The service needs for the system studied here are not burdensome, but if they are not addressed sincerely the technology may be abandoned prematurely.

Seasonal maintenance for the dual evaporative pre-cooler includes the following elements:

- Shutdown system, drain sump, and drain rooftop water lines
    - For freeze protection in freeze prone areas
    - To avoid standing water for extended periods without operation
- Wash sump, media, emitters (water distribution header), ventilation coil, and condenser coil
- Cleanout filter for water bleed system
- Startup system again in spring
- Adjust water bleed, set point temperature, and re-commission operations
- Mid-season checkup to confirm operation and cleanup if needed

Mineral deposition is a significant concern for evaporative systems. However, for the installations observed here, there was not substantial accumulation of scale on the media or anywhere on the system. There was some accumulation of solids at the bottom of the sump, but this was easily washed out with seasonal maintenance. There are a variety of methods to manage solids deposition in evaporative equipment. The system studied here bleeds some water to drain whenever the pump operates, this allows fresh makeup water let into the sump to dilute the concentration of dissolved solids in the sump water. The bleed rate is continuous and is selected by the contractor according to the local water quality characteristics. The bleed rates selected for the installations observed here were adequate to avoid the formation of scale.

Importantly, the dual evaporative pre-cooler systems did not damage the condenser coils. No corrosion and no mineral deposition was observed for the three hybrid rooftop units studied. However, the evaporative media doesn't necessarily keep the condenser cleaner than usual – the research team observed significant soiling on condenser coils during spring startup. Cleaning of condenser coils should still follow regular maintenance practices.

On the other hand, since the ventilation cooling coil is located upstream of any filtration it tends to accumulate a significant amount of dirt and debris. Over time, this accumulation could reduce ventilation flow and could reduce performance of the dual-evaporative pre-cooler. Accumulation of dust and debris was observed for all three of the systems evaluated in this study. The photograph in Figure 24B documents the dirty coil. A consistent service program is needed to keep coils clean.

Another general challenge for evaporative systems is that direct evaporative media can be damaged by long term sun exposure, and may require replacement more often than is preferred. However, the dual evaporative pre-cooler in this assessment uses media with an ultra violet (UV) resistant coating; the research team observed no physical degradation of the media in this study as a result of exposure to the sun. After 24 months of operation, the media showed no signs of deterioration. However, the manufacturer does recommend a schedule for media replacement ever 5 years.

Lastly, water based systems are naturally at risk of biological contamination. Precaution must be taken in system design and operation to avoid biological growth. There are different types of micro-organisms that may take root in evaporative systems, and a number of methods to avoid this biological growth.

Previous study of this dual-evaporative pre-cooler has indicated that if the pump runs continuously and the media is never allowed to dry out, it can host algae growth. One example of the research team's observation of this from a different site is recorded in Figure 23A. When the system is controlled properly, and when an algae resistant media is used this does not occur. The equipment in this study showed no indication of algae growth on the media, as illustrated in Figure 23B.

Although the media in this study remained clean and clear of algae growth, the research team consistently observed significant accumulation of biofilm in the sump water for these three units. The accumulation observed was substantial, although it did not cause any physical damage to the equipment and was easily cleaned out by seasonal maintenance. This type of biofilm growth has not been observed by the research team for other installations with the same equipment, but it was persistent for the systems studied in Ontario. This observation does not form a major concern about system longevity or health risk; however, it would be preferable to manage the system in a way that inhibits biological growth.

There are a variety of chemical and non-chemical methods to avoid biological growth in evaporative equipment. Some systems use electronic chlorination, while others use chemical biocides. We recommend a simple strategy that eliminates the potential for significant biological growth in the water by draining from the sump anytime the pump is not in use. Since the dual-evaporative pre-cooler typically shuts down for several hours each night, draining the sump should usually provide an effective means to avoid accumulation of biofilm. In circumstances where nighttime temperatures are not reliably cool enough to allow the pump to shut off, and where the fan runs continuously, we recommend an alternative means of biological control.

Generally, the dual-evaporative pre-cooler studied is a robust and reliable system. The research team did not observe any mechanical failures, or anything that caused obvious risk for equipment damage. However, in order to capture persistent savings, there is a need to ensure for consistent and reliable service for the long term. There is currently not adequate industry familiarity with the service requirements for these components; the research team is aware of some installations that have been abandon after a few years if service contracts are not renewed.  Other installations have been subjected to freeze damage when systems were not winterized early enough.

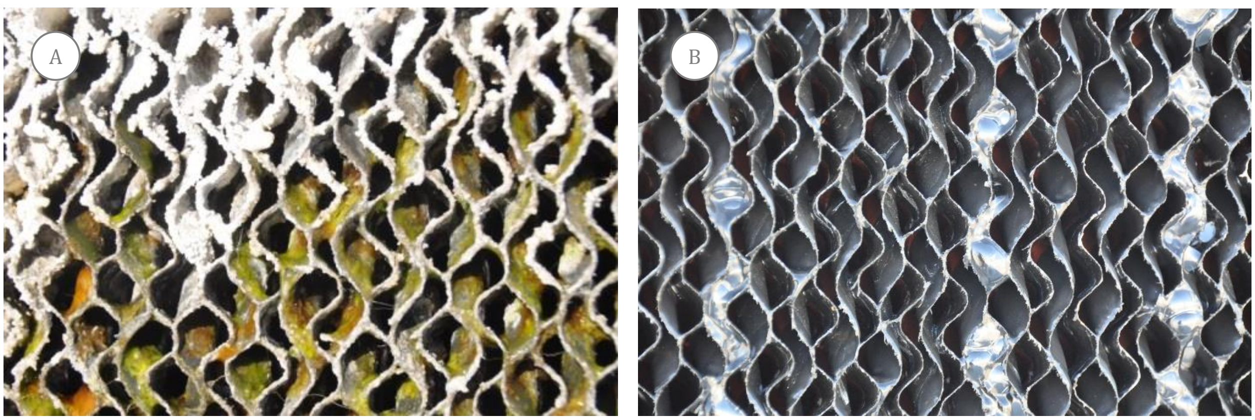


**FIGURE 23: (A) MEDIA WITH ALGAE GROWTH AND INCREASED SCALE ACCUMULATION AS A RESULT OF IMPROPER SET-POINT (B) MEDIA WITH PROPER SET-POINT THAT ALLOWS MEDIA TO DRY OUT DAILY**


**FIGURE 24: (A) BIOFILM ACCUMULATION IN SUMP (B) BUILDUP OF DIRT AND DEBRIS ON VENTILATION COIL**

# DISCUSSION AND CONCLUSIONS

This study provides three examples of real world performance for one type of hybrid rooftop packaged air conditioner that uses dual-evaporative pre-cooling to extend the performance of a conventional vapor compression air conditioner. The study demonstrated a clear reduction in measured energy consumption for each unit, and also allowed for an in-depth assessment of the characteristic performance of the hybrid unit, and the sub-components for the dual evaporative pre-cooler. The results of this study stand as a strong example to encourage future efforts surrounding advancement and market-transfer of hybrid rooftop air conditioners. The study results also provide detailed performance characteristics that are essential for development and validation of emerging modeling and simulation tools focused on this type of technology.

## *COEFFICIENT OF PERFORMANCE*

The sensible system COP measured in this evaluation generally agrees with efficiency that has been reported for this type of hybrid air conditioner in laboratory tests (Woolley 2012). It is important to note that the system sensible COP actually increases as outside air temperature increases. This behavior stands apart from general expectations about air conditioner performance, which predicts that system efficiency should decrease as outside temperature increases. The room cooling COP remains steady with outside air temperature.

## *ENERGY AND DEMAND SAVINGS*

A pre-and-post comparison of the energy use signature for the three systems indicates on-peak reduction in hourly energy consumption of 66% for M12-14, 45% for M15-15, and 13% for AC7. Of the three systems, AC7 achieved the highest sensible system COP, yet this unit showed the smallest reduction in overall energy consumption. This occurred because the equipment was undersized for the load, so part of the efficiency gained with addition of the dual-evaporative pre-cooler resulted in increased cooling, but not measured energy savings. This observation highlights the fact that real savings achieved by any efficiency measure hinges significantly on the specifics of its application and use.

The savings recorded for AC7 stand as an example that the full savings potential for the measure may not be reached when applied in a scenario where a part of the efficiency benefit is invested in extending the level of service, instead of reducing overall energy consumption. Future applications should be careful to target scenarios where the added capacity will decrease runtime, or decrease operation in higher capacity modes. One simple approach would be to ensure that a system is not undersized for the load that it is serving.

## *WET BULB EFFECTIVENESS*

The results herein indicate that wet bulb effectiveness for cooling ventilation air is not a fixed value. For AC7, wet bulb effectiveness ranged from 10-55%. Most importantly, effectiveness changes as the airflow rate across the coil changes; and for a variable speed multi-stage system with the capability to modulate damper position, this ventilation airflow rate can change with each mode of operation. For a particular ventilation airflow rate, wet bulb effectiveness also changes with the outside air wet bulb depression – it remains relatively steady above a certain wet bulb depression and declines sharply below this point. This last characteristic is a new understanding that emerges directly from the observations in this field evaluation, and which was not apparent from previous studies.

## *SUPPLY AIR TEMPERATURE*

Measurements indicated that supply air temperature for these hybrid air conditioners trends most closely with the outside air wet bulb temperature. Since capacity and performance for a vapor compression system is driven most significantly by the temperature difference between the evaporator and condenser, the outside air dry bulb temperature is usually the single best predictor for supply air temperature from a conventional air conditioner. In this case, the dual-evaporative pre-cooler shifts system performance in such a way that the outside air wet bulb temperature becomes a much better predictor.

### *VENTILATION COOLING*

When the pump cycles on and off in sequence with compressor cycling for RTUs that operate in "auto" mode, it can result in startup delays that can limit effectiveness of the dual-evaporative pre-cooling system. For systems that provide continuous ventilation, the wet bulb effectiveness for ventilation air cooling suffers when the pump continues to run while the compressors and condenser fans are off because there is no airflow across the evaporative media. We recommend that the dual-evaporative pre-cooler system be applied on units that provide continuous ventilation, and suggest that future generations of the technology include the capability to run condenser fan(s) to enhance ventilation cooling even while compressors remain off. Consideration should be given to whether such a capability should be deployed for all operating modes, or if it will only have advantage above a particular ambient condition.

### *SYSTEM CONTROLS*

M12-15 and M15-15 deployed a control scheme that interlocked pump operation with blower operation in order to ensure that the pump did not operate uselessly during periods when the rooftop unit was otherwise off. This is an important feature and an improvement over a previous generation of the technology that allowed some units to cycle pumps continuously for hundreds of hours each month while the rooftop unit was otherwise off. However, the way that the new feature was deployed resulted in spasmodic pump behavior that limited performance for the dual-evaporative pre-cooler. We recommend that future efforts incorporate a more reliable method for an interlock between the fan and pump.

### *WATER CONSUMPTION*

The electrical energy savings achieved by the dual-evaporative pre-cooler comes at the expense of increased on-site water consumption. Hourly water consumption for the system increases predictably with outside temperature, and bleed from the systems measured here is only 25% larger than the predicted water evaporation rate. For savings of 7 kWh/h, as measured for M12-14 at on-peak conditions, the coincident rate of water consumption equates to 5 gallons/kWh savings. For context, estimates for the water use intensity for electricity generation range from by more than an order of magnitude, depending on the source mix for electricity generation, but the most well founded research estimates 1.41 gal/kWh for California's grid mix, on average (Pistochini 2011, Torcellini 2003, Larson 2007). This means, from a statewide water use perspective, the local water consumption of this technology is partially offset by the water savings associated with not generating the electrical energy conserved.

The bleed rate for the three systems appears to be reasonable and consistent, but other studies have observed excessive bleed rates that are more than double the estimated water evaporation rate (Modera, 2014). The most effective and water efficient methods for managing sump water quality use measured conductivity to control drain and fill cycles for the sump. Given the current drought in California, and the continued pressure on water resources generally, in order to ensure efficient water consumption for this technology deployed in mass, we highly recommend the addition of a more sophisticated method for management of maintenance water.

### *STRATEGIC APPLICATIONS*

Parallel studies indicate that when multiple air conditioners serve the same general space, such as for big-box retail facilities, some units may end up operating far more often than others. In such a scenario, the dual-evaporative pre-cooler will have little value if installed on a system that rarely operates. Further, when there is an opportunity, the responsibility for ventilation could be grouped onto some units, instead of distributed across all rooftop units on a facility. The dual-evaporative pre-cooler could then be applied to the units providing continuous ventilation, and the systems without ventilation could operate in an 'auto' mode to reduce total fan runtime hours. Those units without ventilation could then use a less costly condenser air pre-cooler without ventilation pre-cooling that would still benefit the vapor compression efficiency.

This solution offers the most substantial savings potential at high ambient temperatures. From the observations in this study it appears that the measure has very little value below 70°F. Also, previous studies have observed algae growth and mineral scale on the media when the pump is allowed to operate at low ambient temperatures and the media is never allowed to dry. This technology can be applied especially well as a peak demand reduction measure, with the added benefit of significant energy savings over the whole course of operation, but operation should be limited to periods when ambient temperature is above 70°F. An even higher changeover set point might be appropriate when applied specifically for peak demand reduction, or where it is deemed that water use should be limited to periods when it has the greatest value.

The actual savings afforded by the dual-evaporative pre-cooler system depends on application. We recommend that efforts to advance the technology should target opportunities with the most significant potential for energy and peak demand impacts

### *INSTALLATION AND COMMISSIONING*

All three of the installations evaluated suffered from improper setup and commissioning. Some of the issues observed were related to setup for the dual-evaporative pre-cooling system, but most of the challenges were related to the baseline rooftop air conditioner. The most significant issues were related to program settings for fan speed and damper positions in each mode of operation. None of the systems were setup according to design intentions, and as a result they operated with low supply airflow rates, inadequate ventilation rates, or inappropriate economizer controls. We recommend that any effort to advance dual-evaporative pre-cooling should also include a quality optimization effort that follows a well vetted commissioning protocol to address concerns for the whole HVAC system, including distribution and controls.

### *MAINTENANCE AND LONGEVITY*

After almost two full years of operation, the dual-evaporative pre-cooler systems show very little physical degradation. The evaporative media is not degraded or damaged and shows no significant accumulation of mineral scale. In this test, and all other installations that the research team is familiar with, the technology does not result in any damage to the condenser coil. The technology is robust and reliable; it stands as a good opportunity to provide persistent energy savings and substantial peak demand reduction.

The one significant physical concern observed for these installations is related to the potential for biological growth in the sump. On every visit made to the site, the research team consistently found a substantial accumulation of biofilm. This accumulation did not damage any system components, and was easily cleaned by occasional maintenance, but it would be preferable to manage the system in a way that inhibits biological growth.

Given the need for attentive service for system cleanup and seasonal shutdown, we highly recommend that programs designed to advance broader application for the dual-evaporative pre-cooler and other evaporative cooling technologies should also incorporate an ongoing service element to ensure proper system operation and persistence of savings. However, there are also some opportunities for added features that could reduce the maintenance burden. For example, in climates where freeze shutdown is necessary, we recommend that any evaporative system should incorporate an automatic temperature-controlled drain back of all supply lines, drain for the sump, and shutdown of all related systems.

Furthermore, it is unlikely that any technology that allows for accumulation of biofilm will find broad market acceptance. We adamantly recommend that all evaporative systems incorporate features or management strategies that inhibit biological growth. There are chemical and non-chemical methods to achieve this end. One simple approach is to avoid any opportunity for standing water. If the sump for the dual-evaporative pre-cooler were automatically drained every day, or anytime the pump is shut off, the sump and media would dry out and water borne biology would not have an opportunity to colonize.

### *NON-TECHNICAL CHALLENGES*

Lastly, observation of the network of market actors involved in the process of design, installation, commissioning and management for this and other HVAC efficiency measures indicates that there are various failures in the standard deployment process that allow significant inefficiencies to go un-noticed and un-addressed. We noticed that each individual and group in this process is responsible for a narrow aspect of the work required to deploy a technology, and that the cumulative sum of decisions made by the whole network may not adequately address the details that are necessary to realize the full potential for savings. For example, a manufacturer builds a machine and warrantees that parts will not fail as a result of manufacturing defect, but they usually have zero input into how a machine is actually setup and used in the field. An installer connects everything and ensures that the system does turn on and off, and service contractors perform the bare minimum to ensure that the system provides comfort. Unfortunately, there is no place in this equation where someone is given the responsibility to ensure that the system operates in an efficient way. This study demonstrates that small changes in the particulars for an application can have significant impacts on the efficiency for a system and the savings achieved. Unless efforts to advance application of this type of technology can also advance a model for deployment that ensures effective application and management, it is clear that the full potential value for this type of measure will not be realized. One approach to addressing this challenge would be to structure incentives, deployment models, and technology features so that at least one stakeholder is given explicit responsibility for ensuring a minimum performance threshold.

# RECOMMENDATIONS

The research team strongly recommends broader adoption of the dual-evaporative pre-cooler technology evaluated here – especially as a measure to reduce peak electrical demand. The technology achieves substantial efficiency improvements. It shows clear energy and demand savings, operates with a reasonable amount of water consumption, and has demonstrated good reliability in the field.

At the same time, there are some challenges surrounding the measure, including the need for improved water use controls, and needs for more reliable commissioning. Efforts to advance broader application of the strategy should consider the following summary recommendations.

1. Deploy the technology strategically to reduce peak electricity demand for cooling
2. Utilize dual evaporative pre cooling for air conditioners that provide continuous ventilation
3. In buildings with many RTUs, aggregate the ventilation supply onto fewer units in order to maximize the value generated by each dual evaporative pre-cooling device. Utilize condenser pre-coolers for equipment without ventilation.
4. Structure incentives, deployment models, and technology features in a way that ensures equipment is commissioned and maintained.
5. Target applications where the increase in capacity afforded by dual-evaporative pre-cooling will decrease runtime, or decrease operation in higher capacity mode.
6. Develop new modeling tools and industry standard methods of evaluation that are capable of rating and projecting broad scale savings for hybrid air conditioners.
7. Develop training resources, design guidelines, and codes and standards that will build industry familiarity with this and other advanced HVAC concepts.
8. Incorporate automated fault detection for the measure, and require that all installations include a reliable ongoing service element to ensure proper system operation and persistence of savings.
9. Any installation of the measure should incorporate a quality optimization effort that follows a well vetted commissioning protocol to address concerns for the whole HVAC system.
10. In any freeze risk area, evaporative equipment should incorporate an automatic temperature-controlled drain back of all supply lines, drain for the sump, and shutdown of all related systems.
11. Any evaporative system should incorporate features that inhibit biological growth
12. The measure studied here should seek a more sophisticated method to control maintenance water.
13. Further study of the technology should focus on capturing:
    a. Measured whole building annual kWh savings, and site scale peak demand reduction
    b. Simulation of energy and demand savings in various applications and climates

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# APPENDIX A – SIMPLE PAYBACK ESTIMATE

| **General Assumptions** | | |
|---|---|---|
| Area = | 5700 | sq.ft. |
| OSA = | 960 | CFM |
| Density = | 100 | sq.ft./person |
| Room Temp = | 75 | degrees F |
| Jnit Size, Nominal | 12.5 | tons |

| **Sensible loads** | | | **BTU/hr** | |
|---|---|---|---|---|
| Lights | 1 | W/sq.ft | 19437 | |
| People | 250 | BTU/hr-per | 14250 | |
| Misc Elect. | 1 | W/sq.ft | 19437 | |
| Supply Fan | | BHP | 0 | (Net sensible capacity provided in catalog) |
| Outside Air | | (TBD) | | |
| | | | 53124 | BTU/hr +OSA Load |

**Baseline Scenario**

| | | OSA Temp | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | 65 - 70 | 70 - 75 | 75 - 80 | 80 - 85 | 85 - 90 | 90 - 95 | 95 - 100 | 100 - 105 | |
| Sensible Load | OSA | -5184 | 0 | 5184 | 10368 | 15552 | 20736 | 25920 | 31104 | |
| | Others | 53124 | 53124 | 53124 | 53124 | 53124 | 53124 | 53124 | 53124 | |
| | Qsens | 47940 | 53124 | 58308 | 63492 | 68676 | 73860 | 79044 | 84228 | |
| Net Qsens Cap. | | ( doesn't operate in this range) | | | | 103500 | 102000 | 101000 | 100000 | |
| % Run Time | | | | | | 66% | 72% | 78% | 84% | |
| | kW | | | | | | | | | |
| | compr | | | | | 8.7 | 9.1 | 9.7 | 10.1 | |
| | indoor fan | | | | | 1.0 | 1.0 | 1.0 | 1.0 | |
| | cond fan | | | | | 1.1 | 1.1 | 1.1 | 1.1 | |
| | Total | | | | | 10.8 | 11.2 | 11.8 | 12.2 | |
| hours/yr(Office) | | | | | | 249 | 274 | 168 | 51 | |
| kWh/yr(abv.85 degrees) | | | | | | 1784 | 2222 | 1551 | 524 | 6082 |

**Dual Evaporative Pre Cooling Scenario**

| | | OSA Temp | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | 65 - 70 | 70 - 75 | 75 - 80 | 80 - 85 | 85 - 90 | 90 - 95 | 95 - 100 | 100 - 105 | |
| | Coincid. WB Temp | | | | | 66 | 67 | 68 | 71 | |
| | Wet bulb Depression | | | | | 21 | 25 | 29 | 31 | |
| | Wet Bulb Efficiency | | | | | 50% | 50% | 55% | 55% | |
| | Condenser Air Temp | | | | | 77 | 80 | 81 | 85 | |
| | OSA | -5184 | 0 | 5184 | 10368 | 15552 | 15552 | 20736 | 25920 | |
| | Others | 53124 | 53124 | 53124 | 53124 | 53124 | 53124 | 53124 | 53124 | |
| | Qsens | 47940 | 53124 | 58308 | 63492 | 68676 | 68676 | 73860 | 79044 | |
| Net Qsens Cap. | | | | | | 103500 | 103500 | 103500 | 103500 | |
| % Run Time | | | | | | 66% | 66% | 71% | 76% | |
| kW | | | | | | | | | | |
| compr | | | | | | 6.3 | 5.5 | 5.9 | 6.3 | |
| indoor fan | | | | | | 1.0 | 1.0 | 1.0 | 1.0 | |
| cond fan | | | | | | 1.1 | 1.1 | 1.1 | 1.1 | |
| Total | | | | | | 8.4 | 7.6 | 8.0 | 8.4 | |
| | | | | | | 22% | 32% | 32% | 31% | |
| hours/yr(Office) | | | | | | 249 | 274 | 168 | 51 | |
| kWh/yr(abv.85 degrees) | | | | | | 1388 | 1382 | 959 | 327 | 4056 |

**Savings and Payback Calculations**

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | peak$/kWh | kWh Savings |
| Savings, kWh | | | 397 | 840 | 592 | 197 | 2026 | $ 0.13 | $ 253.27 |
| | | Month | Peak Temp | kWbase | kWmeas | kWsaving | | demand $/kW | $ demand saving |
| **Normalized Savings** | | May | 95 | 11.8 | 8.0 | 3.8 | 32.2% | $ 21.00 | $ 79.80 |
| 0.3 | kW/ton | June | 100 | 12.2 | 8.4 | 3.8 | 31.1% | $ 21.00 | $ 79.80 |
| 162 | kWh/ton | July | 100 | 12.2 | 8.4 | 3.8 | 31.1% | $ 21.00 | $ 79.80 |
| | | Aug | 100 | 12.2 | 8.4 | 3.8 | 31.1% | $ 21.00 | $ 79.80 |
| | | Sept | 100 | 12.2 | 8.4 | 3.8 | 31.1% | $ 21.00 | $ 79.80 |
| | | | | | | | | | $ 399.00 |
| | | | | | | | | **Electrical S** | **$ 652.27** |
| | | | | | | | | Annual wat | $ (66.15) |
| | | | | | | | **Net Annual Savings** | | **$ 586.12** |