SIDE-BY-SIDE EVALUATION OF TWO INDIRECT EVAPORATIVE AIR CONDITIONERS ADDED TO EXISTING PACKAGED ROOFTOP UNITS

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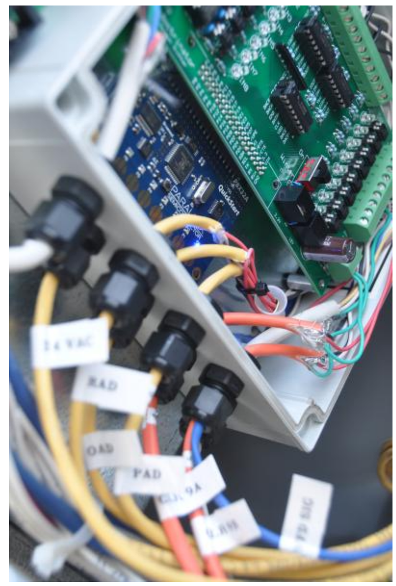

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Abbreviations and Acronyms

HVAC	Heating Ventilation & Air Conditioning
RTU	Rooftop Unit (Rooftop Packaged Air Conditioner)
IEC	Indirect Evaporative Cooling (Cooler)
PA	Product Air (Supply air from indirect evaporative cooler)
RA	Return Air (From the room)
SA	Supply Air (Supply air delivered to the space)
OSA	Outside Air
EA	Exhaust Air (Exhaust air from the IEC)
WCEC	UC Davis Western Cooling Efficiency Center
EDGE	Enhanced Data rates for GSM Evolution (Wireless network protocol)
DX	"Direct eXpansion" (Compressor based vapor compression cooling)
EMCS	Energy Management and Control System (Building wide system controls)
COP	Coefficient of Performance
C_p	Specific Heat Capacity (e.g.: *Btu/lbm-°F*)
$\dot{H}$	Enthalpy Flow Rate, (Cooling Capacity) (e.g.: *kBtu/h*)
h	Specific Enthalpy (e.g.: *Btu/lbm*)
HR	Humidity Ratio
$\dot{m}$	Mass Flow Rate (e.g.: *lbm/h*)

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INTRODUCTION

More than 50% of the peak electrical demand from commercial buildings in California can be attributed to air conditioning. Cooling and ventilation in these buildings is served predominately by packaged rooftop units (RTUs). Data from the California Energy Commission's *Commercial End Use Survey* indicates that more than 75% of commercial cooling systems in California are packaged rooftop units. On average, these systems account for 25% of the electricity use in commercial buildings. However, there is considerable variability in the overall energy consumption and electrical demand from this equipment in different buildings. For some customers with large cooling loads these systems can account for more than 75% of their electricity consumption during hot afternoons. Aggregated across the state, air conditioning results in summertime electricity generation requirements that are 40% higher than the winter. For inland regions the summertime electricity demand can be 250% larger than winter months. 60% of the electrical load during these times can be attributed to air conditioning (CA ISO 2013).

As California moves toward a larger share of renewable and intermittent generating resources, air conditioning is poised to become an even more significant challenge for the grid. Cooling loads lag behind daily trends for outside air temperature, so electricity demand for air conditioning can increase late in the day at the same time that electricity generation from solar begins to wane. The net result introduces new challenges for electric grid management, especially related to the rate at which conventional power plants are required to ramp up, and the potential for over-generation on the grid (CA ISO, 2013). The recent retirement of substantial base load generation from the San Onofre Nuclear Generating Station further complicates the dynamics for generation in California.

Addressing the economic and environmental challenges that surround these facts will require substantial changes in a variety of sectors. Since heating, cooling, and ventilation are at the root of more than half of the energy use and carbon emissions in buildings, this sector deserves significant attention and technical advancement for efficiency improvements. Utilities, industry, and end users are beginning to tackle the many challenges surrounding energy use in buildings, the need for change is arguably most acute for HVAC technology.

The California Public Utilities Commission has emphasized the need for a rapid industry-wide shift toward dramatically more efficient cooling technologies. Among those strategies, the California *Energy Efficiency Strategic Plan* calls for a market transition toward "climate-appropriate" cooling strategies. Climate appropriate technologies utilize various techniques designed to produce cooling in California's hot-dry climate with far less energy consumption than the conventional alternatives. In addition to the incremental efficiency improvements that are emerging for conventional rooftop packaged air conditioners, climate appropriate technologies avoid unneeded dehumidification, and use local environmental conditions for efficiency advantage whenever possible. These technologies promise to reduce annual energy consumption and peak electrical demand by more than 50%.

This report documents the results of a pilot field study designed to characterize performance for two indirect evaporative air conditioning products, referred to here as the "Type M" and "Type C" equipment. These systems each utilize indirect evaporative heat exchangers that enable water evaporation to cool a building without adding moisture to the air that is supplied to the conditioned space. These air conditioners do not have compressors, and the only major energy consuming component is a fan. Unlike conventional vapor-compression air conditioners, indirect evaporative systems actually become more efficient as outdoor temperature increases. Indirect evaporative cooling has been in development for more than thirty years, in which time the technology has made major advancements.

There are currently a variety independently developed products that accomplish indirect evaporative cooling in different ways. All of these strategies promise significant energy savings for cooling, though each presents unique advantages and challenges. The technology has reached a state of development that warrants broader market adoption, however product diversity is currently small, equipment costs are relatively high, professional familiarity with the technology is limited, and the requirements for custom engineering and systems integration is somewhat different from conventional rooftop units. These challenges are generally expected for emerging technologies.

This study provides independent field data on the energy performance of these products, offers insight into some of the technical challenges, and recommends strategies for manufacturers, customers, utilities, and regulators to accelerate successful and cost effective application of the technology. The level of efficiency achieved by the equipment studied here is striking, and the potential for energy savings and peak demand reduction is substantial. The research team encourages further efforts from all industry stakeholders to facilitate market adoption of the technology, and cautions that such efforts should be designed strategically so as to safely navigate a variety of technical and market challenges.

Project Objectives

The overarching goal of this pilot demonstration project was to explore and document the field application of indirect evaporative cooling applied as a retrofit to existing commercial HVAC equipment. Climate appropriate cooling is a key goal within the California Energy Efficiency Strategic plan. This work advances those goals and included several specific objectives:

1. Demonstrate an effective method to integrate a stand-alone indirect evaporative air conditioner as retrofit to existing rooftop packaged equipment.
2. Characterize the energy and water use efficiency of indirect evaporative cooling, separate from the performance of the rooftop packaged equipment. Describe performance according to metrics that can inform subsequent building modeling and simulation efforts for this type of equipment.
3. Provide explanation about any characteristic differences between the technologies that may impact design requirements, controls, and preventative maintenance practices.
4. Assess the technical opportunities and challenges related to installing an indirect evaporative air conditioners as a retrofit to a conventional rooftop packaged unit.
5. Develop recommendations for controls and design concepts to facilitate the application of indirect evaporative cooling. Consider the opportunities and tradeoffs related to alternative approaches.
6. Document challenges and qualitative lessons learned over the course of the pilot installation.

Project Overview

Technology Background

Indirect evaporative air conditioners employ specially designed heat exchangers that use water evaporation in one air stream to impart sensible cooling to a another air stream without any moisture addition to the conditioned space. The wetted air stream is generally referred to as the "secondary", "process", "scavenger", or "working" air-flow. At its outlet the secondary air stream from an indirect evaporative device is typically near 100% relative humidity and is exhausted to outdoors. The dry side of an indirect evaporative device is referred to as the "primary" air stream. When the outlet of the primary air stream is delivered to the space it may be referred to as "supply" air; for this report it is referred to more generally as the "product" air stream since it is delivered to the inlet of a subsequent cooling process.

Indirect evaporative cooling can be very efficient. It is different from a direct evaporative cooling in three significant ways: (1) does not add moisture to the conditioned space; (2) can cool to a lower temperature; and (3) exhausts a portion of the air moved. The later characteristics mean that indirect evaporative requires more fan power per delivered air-flow (*W/cfm*) than a conventional direct evaporative cooler. Since the fan(s) in an indirect evaporative cooler are the only significant energy consuming component(s), the details of heat exchanger design can result in significant differences for equipment performance and energy efficiency.

There are a variety of configurations for indirect evaporative systems. Some equipment is constructed using cross-flow plate heat exchangers similar to those utilized for exhaust heat recovery, others utilize a tube-in-flow approach similar to an evaporative fluid cooler, while other systems utilize heat pipes or runaround hydronic circuits to transfer heat between two physically separate airstreams. The two systems studied in this project use specially developed polymer heat exchangers that extract a portion of the primary air stream to be used as inlet for the secondary air stream. As a result of approach, these systems can generate product air at a temperature lower than the wet-bulb of the system inlet. This is possible because flow diverted from the primary air stream has already been cooled sensibly and therefore enters the secondary channels with a wet-bulb temperature that is lower than at the system inlet. As evaporation occurs in the secondary air stream the process drives product air toward the lower wet-bulb temperature. In theory, a system that repeatedly cascades flow in such a way could achieve product air at the dew point temperature of the system inlet. Generally, as the amount of primary air-flow diverted into the secondary side of the heat exchanger increases, product temperature will decrease. While the product temperature decreases, diversion of air flow out of the primary air stream increases the fan power required to deliver each unit of product air-flow. These facts mean that a heat exchanger design could optimize for cooling capacity, sensible efficiency, or for delivered temperature. These two systems divert roughly half of the primary air-flow to for secondary air-flow.

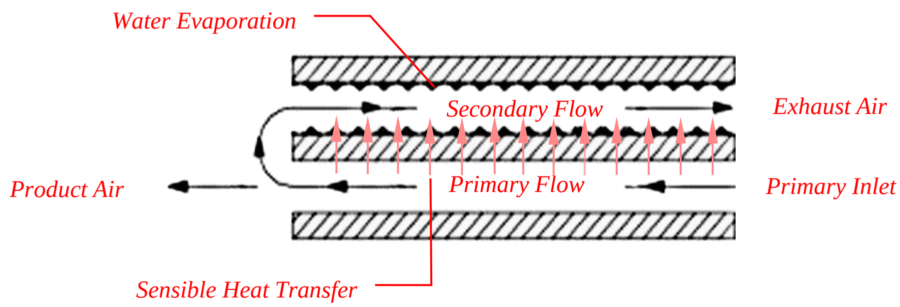

FIGURE 1. CONCEPTUAL SCHEMATIC FOR INDIRECT EVAPORATIVE COOLING

Some indirect evaporative cooling systems can utilize building exhaust air as the source for the secondary air stream. This approach is beneficial for system efficiency because it effectively combines heat recovery with indirect evaporative cooling to increase the system-cooling-capacity. Other systems have mixed return air with outside air as the source for primary air-flow, not unlike a conventional packaged rooftop unit.

The two technologies studied here both utilize outside air only, and were not configured to process any return air. Roughly half of the primary flow in these two systems is diverted as inlet for the secondary flow. The remaining primary flow is delivered as useful product air, and the secondary flow is exhausted. The systems therefore provide positive pressurization of the building, and require some air relief, or building exhaust to maintain air balance within the building. While exhaust air exits these systems near saturation, this air-flow is cooler than outside air, and so in some cases could be applied for some useful purpose. In some hybrid systems that also use vapor-compression cooling, exhaust air from the indirect evaporative system is used to cool the condenser.

The technologies studied here are similar in their overall conceptual operation, but they differ in a few important ways that result in unique performance characteristics, and engineering design constraints. The Type C system allows all of the primary air-flow to pass through the heat exchanger before diverting a portion of the product back into the secondary channels. Much of the heat exchanger operates in counter-flow, and part of the heat exchanger operates in cross-flow. The Type M system passes air from primary channels to secondary channels at a series of points throughout the heat exchanger, and operates entirely in cross-flow. The implication of these differences will be discussed through review of the engineering design for the pilot installation, as well as in the results, and conclusions.

As the results of this study show, indirect evaporative air conditioners can achieve much higher sensible cooling efficiencies than conventional vapor-compression systems. The technology has the greatest benefit when used for cooling code-required ventilation air. In fact, the full-speed system-cooling-capacity and efficiency of these systems increase as outside air temperature increases. However, while indirect evaporative air conditioners can sustain the room-cooling requirements in commercial applications for many hours, they are typically not able to cover the peak sensible room-cooling loads for commercial buildings. Accordingly, the current state of indirect evaporative cooling must usually be applied to operate in cooperation with vapor-compression systems. There are many applications where indirect evaporative cooling may be sufficient without supplementary mechanical cooling, however this pilot was mainly concerned with an application where vapor-compression cooling would be needed for adequate cooling.

The two indirect evaporative air conditioners evaluated here use variable speed fans. This allows the equipment to operate at part speed during part load conditions. Since fan power declines rapidly as air-flow decreases, part speed operation can achieve higher cooling efficiency. However, since the systems operate with 100% outside air it makes sense to use their product air to meet ventilation requirements for a space. Under such a scenario, the potential for part speed operation can be limited since the equipment must continue to provide ventilation regardless of load.

Since the cooling-capacity for these systems is directly coupled to their flow rate, there can be a mismatch between instantaneous room-cooling needs and ventilation requirements. In certain scenarios, and without proper control, this could result in overcooling a zone. When indirect evaporative air conditioners are used to supply ventilation air, and to operate together with vapor-compression, we found that there are several main technical constraints that must be addressed as part of the engineering design for the overall HVAC systems:

1. Systems must maintain ventilation requirements without overcooling the zone
2. Systems must maintain ventilation requirements even when heating is required
3. Controls should give priority to indirect evaporative cooling over vapor-compression cooling
4. Controls should give priority to economizer cooling over indirect evaporative cooling
5. System must maintain adequate evaporator coil air-flow for vapor-compression cooling (when needed)
6. System must maintain adequate condenser air-flow rates for efficient vapor-compression cooling

The Type M and Type C products evaluated here are designed as stand-alone indirect evaporative air conditioners, and application of the systems in a way that achieves these technical constraints currently requires custom engineering and controls. Both manufacturers can supply their heat exchangers for application in a custom air handler that could be designed to meet these constraints, but this demonstration focuses on application of their stand-alone equipment as retrofit to a building with existing rooftop packaged air conditioners.

Design Goals & Constraints for Pilot Installation

This pilot installation was developed through a collaboration between PG&E, the host (the PG&E customer that provided the site), UC Davis Western Cooling Efficiency Center (WCEC), the two manufacturers tested, and the installing design-build contractor EMCOR Mesa Energy Systems. The primary design goal was to apply indirect evaporative cooling as a retrofit to cover the ventilation requirements for a large retail store and grocery in Bakersfield CA (California Climate Zone 13). While large retail facilities usually provide ventilation air through the many rooftop packaged air conditioners spread across a store, this project aimed to centralize ventilation air-flow via a displacement ventilation scheme that would allow most rooftop units to operate as recirculation only. In this way, fans for the existing rooftop units only operate when there is a call for cooling or heating. Further, the cooling load for these existing units is reduced in two ways:

1. They no longer have to address ventilation cooling loads
2. Sensible room-cooling loads are reduced by the indirect evaporative systems

In many ways, the conceptual design approach was similar to application of Dedicated Outside Air Supply (DOAS) systems. However, to simplify the physical retrofits required, it was decided to utilize existing duct systems to deliver ventilation air, instead of adding new roof penetrations and ductwork for the new indirect evaporative cooling equipment. Instead of replacing existing rooftop units, the project retrofitted several existing units to operate in conjunction with the new indirect evaporative air conditioners. Doing this required substantial attention to design details and to a sequence of operation to control the paired equipment in an appropriate way. The combination needed to: (1) respond to signals from the existing building automation system; (2) maintain an appropriate amount of continuous ventilation; (3) give priority to room-cooling capacity from the indirect evaporative systems before enabling compressors; (4) ensure the ability to operate in an economizer mode; (5) maintain access to ventilation air during heating operation; (6) allow for a ventilation-only mode when no zone cooling was required; and (7) provide adequate supply air-flow for each stage of vapor-compression operation. Further, the two indirect evaporative systems are designed to operate against different external resistances so the retrofit required a design that was flexible enough to facilitate the needs for both systems.

WCEC provided project facilitation and design vision for the effort, and worked in close collaboration with each manufacturer, the customer's engineering team, and the installing contractor to develop all details for the pilot. WCEC developed conceptual mechanical designs for the installation, including layout plans, air-flow arrangements, physical retrofit specifications, and a complete sequence of operations for controls. In an effort to expedite the project, and to consolidate management of the experimental controls scheme, WCEC developed and constructed a custom controller that enabled integration of the systems without requiring any revision to the customer's existing Energy Management and Control System (EMCS), and without requiring retrofit of the on-board controls for the existing 10 ton Lennox Strategos rooftop units. A complete review of the engineering design concept is described in the following section. The prototype retrofits for this project required significant design and engineering, plus substantial contractor effort for custom installation work. Following installation and physical commissioning, WCEC invested dozens of hours for diagnostics and commissioning to setup and program the controls with appropriate fan speed and damper position indexes. The technology demonstrated in this project is technically mature, and shows exceptional efficiency marks, but the design details described here serve as a clear example of the current level of complexity associated with application of the technology. We believe that the need for a robust and simple systems-integration protocol is the most substantial technical challenge for successful market adoption of this climate appropriate cooling strategy. This report should serve as validation of the performance potential for the equipment, and also as a technical introduction to engineering design with these systems.

DESIGN APPROACH FOR PILOT INSTALLATION

INDIRECT EVAPORATIVE COOLING AS RETROFIT DOAS

A design was developed to utilize indirect evaporative air conditioning to supply all of the ventilation air cooling needs for the retail store, and so as to meet the aforementioned design constraints. The continuous ventilation target for the store is 12,000 *cfm*. Most of the existing rooftop units on the sales floor are 10 ton systems with design supply air-flow rate of 3,700 *cfm*. The new arrangement uses six of the 40 existing rooftop units to supply ventilation air flow.

Each of these six units was equipped with an indirect evaporative addition and made responsible for serving 2,000 *cfm* continuous ventilation. The Type M equipment selected for test is rated to supply 1,279 *cfm* at 0.1" ESP, but it is a modular system, so each of three existing rooftop units were retrofit with a pair of Type M systems (RTU 13, RTU 15 & RTU 17). The Type C system tested is rated to supply 2,500 *cfm* so it was added one-for-one to each of three existing rooftop units (RTU 16, RTU 18, RTU 19). This arrangement allowed for a clear side-by-side comparison of the alternative products since the pair of Type M systems supply a similar air-flow to the single Type C system.

The six units selected for the retrofit were located across the front of the sales floor. Since relief flow is readily available through other equipment across the store , through doors, and through the general building envelope ventilation flow from these six machines is distributed throughout the building by displacement. The use of six separate existing rooftop units for supply of ventilation air has some advantage over the conventional DOAS approach in that ventilation supply is spread across the entire width of the store, instead of from a single location.

In general the cooling loads in big box stores are highest at the front end, concentrated around checkout lines, and near the store entrances. We expect that by placing this ventilation air cooling equipment in the most significant cooling zone, the indirect evaporative air conditions would be called to operate at full speed more often. Other DOAS strategies prefer to deliver cooling across the back end of a sales floor, or over refrigerated cases in the grocery – which carries cooling from the refrigerated space to adjacent zones by displacement. These strategies have advantages, however the design selected was concerned that the indirect evaporative cooling may not run as often in the other locations. Placing ventilation supply over refrigerated cases allows excess cooling from these zones to be distributed throughout the store, but the research team has observed existing DOAS equipment supplying unconditioned ventilation air over top of refrigerated cases during peak cooling hours because so little active cooling is needed for these zones. Applying a very high efficiency system to cool a zone that is already cool for another reason does not result in savings. Therefore, the design advanced for this pilot places supply from the indirect evaporative equipment in the most significant cooling zone. The research was not concerned with characterizing and comparing the measured advantages of alternative displacement ventilation strategies, but these dynamics were considered in selection of the location and operating scheme for the indirect evaporative equipment.

FUNCTIONAL DESIGN CONSTRAINTS

Unfortunately, the retrofit of each rooftop unit was not as simple as ducting the indirect evaporative product into the outside air inlet and turning everything on. In addition to the overarching system integration constraints outlined previously, there are a number of very specific operating constraints that had to be addressed, and which led to a somewhat complicated retrofit.

Most importantly, operation in all modes needs to ensure proper positive external static pressure for the indirect evaporative systems. Both of the equipment studied rely on downstream resistance to maintain an appropriate balance between primary and secondary air-flow. At full speed, the Type M equipment is intended to operate with 0.2" ESP, while the Type C equipment targets 0.6" ESP. As the downstream resistance drifts from these design points, the product air-flow rate shifts, and the ratio between primary and secondary air-flow shifts, which changes the heat transfer characteristics within the heat exchanger and effects the net output of the indirect evaporative cooling process. Generally, if these systems are made to work against a larger downstream resistance, a larger fraction of the fan air-flow will pass through the secondary passages (since the relative resistance is reduced) and the system will supply a cooler temperature. However, the reduced supply air temperature is achieved only at the cost of reduced product air-flow. For the Type M equipment, manufacturer-stated data indicates that while the equipment typically operates with a wet-bulb effectiveness of 95%, the equipment will shift to 120% if made to operate against 1.00" ESP. Even while supply air temperature declines, product air-flow will decrease by 50% and cooling capacity will decrease by 40%. Thus, reaching for a lower product air temperature is not necessarily the optimal strategy.

On the other extreme, if these systems operate with a negative downstream pressure, air-flow may actually be drawn backward through the secondary channels, drawing moisture into the product airstream, and disrupting the entire conceptual air-flow scheme and thermodynamic function for the indirect evaporative heat exchanger. It is therefore very important to maintain an appropriate downstream resistance.

In a stand-alone application, the Type M equipment generally relies on a small amount of ductwork resistance and positive building pressurization to maintain a reasonable level of external static pressure. The Type C system has an advantage in that it can maintain airflow and high efficiency against a significantly higher external resistance, however in most stand-alone applications the downstream system resistance will not provide 0.6" ESP, and so the Type C device utilizes an internal manually-set balancing damper to add an appropriate amount of resistance at full speed. When adding these systems to an existing air handler, special care must be taken to maintain an appropriate amount of downstream resistance at all times. When the indirect evaporative product is supplied into an air handler upstream of the blower (through the outside air intake for example) it may be exposed to a negative static pressure, so some method must be used to exert positive back pressure on the indirect evaporative product stream.

Aside from ensuring proper flow dynamics for the indirect evaporative equipment, the combination also needed to function appropriately when indirect evaporative cooling was not needed. In particular, the building requires continuous ventilation even when cooling is not required. These systems could be controlled to supply ventilation without cooling, but doing this requires a significantly larger fan power because air-flow must pass through the indirect evaporative heat exchanger. This approach could eliminate the need for a separate ventilation airflow path, though care must be taken in system configuration and control strategy to avoid drawing air backward through the wetted channels. Moreover, in most climates the indirect evaporative equipment should be disabled through the winter months for freeze protection. Therefore, we recommend that indirect evaporative be applied to cool ventilation air, but that the building retain a separate path for ventilation when cooling is not needed.

For the pilot, it was decided that equipment should be installed such that each rooftop units would still have access to outside air even while the indirect evaporative equipment is shut down, or cooling is not required. In this way, the same six rooftop units could continue to serve as the sole means of ventilation air throughout the year. Alternative designs could switch back to a typical ventilation scheme during the winter months, or provide still another means for ventilation. Such an approach could simplify the physical retrofit (it would not require each rooftop unit to retain direct access to outside air) but it would also require a more complicated EMCS sequence of operations.

ALTERNATE DESIGN CONSIDERATIONS

Weighing these constraints, the research team considered several arrangements for the retrofit.

First, we considered supplying indirect evaporative product air into the rooftop unit supply plenum. However, such an arrangement would not allow the indirect evaporative cooler to operate in combination with the rooftop unit because supply plenum pressure could be too high for connection to the product stream, which would result in imbalance for the indirect evaporative cooler.

We considered supplying product air through the outside air inlet, however doing this closes off access to outside air for economizer-only cooling, and for ventilation supply when cooling is not required. This approach could be a viable option for future applications as long as ventilation and economizer cooling are supplied for the building in some other way. For buildings where there are no other rooftop units or other means for economizer cooling and ventilation, eliminating direct access to outside air is not an option. This research sought a design that would allow the exiting rooftop system to retain all modes of functionality.

It should also be noted that the implications of mismatch between air-flow from the indirect evaporative equipment and supply air-flow for the rooftop unit must be considered. If the rooftop unit were smaller, ductwork could be undersized for the indirect evaporative product flow. If the rooftop unit were larger than the indirect evaporative system, then the rooftop unit fan would need to operate when vapor-compression cooling is required. In this later case the indirect evaporative product is exposed to negative static pressure. For this pilot, the rooftop unit supply air-flow was larger than the indirect evaporative product air-flow. A physical design and control scheme was needed to allow variable speed operation of the indirect evaporative equipment plus also variable (indexed) speed operation of the existing rooftop unit. The indirect evaporative system in each combination supplies a maximum of approximately 2,500 *cfm* product air, while each existing rooftop unit has a design flow rate of 3,400 *cfm*. Operation of the rooftop unit with the first stage compressor requires approximately 2,750 *cfm* supply air, while second stage requires 3,400 *cfm*. Table 1 (next page) records the nameplate technical details for existing and new equipment.

Table 1. Equipment schedule

(E) Equipment							(N) Equipment Addition					
Unit Tag	Serves Area	Manufacturer	Model	Rated Airflow (cfm)	Electrical	Number of Compressors	IEC Manufacturer	Model	Quantity	Rated Airflow (cfm) each	Electrical	Min Outside Airflow (cfm)
RTU 15	Sales	Lennox	SCA120H4MS1G	3700	460/3/60	2	Type M	M50	2	1279	208	2046
RTU 16	Sales	Lennox	SCA120H4MS1G	3700	460/3/60	2	Type C	CW-H15	1	2500	460/3/60	2000
RTU 17	Sales	Lennox	SCA120H4ME1G	3700	460/3/60	2	Type M	M50	2	1279	208	2046
RTU 18	Sales	Lennox	SCA120H4MN1G	3700	460/3/60	2	Type C	CW-H15	1	2500	460/3/60	2000
RTU 13	Sales	Lennox	SCA120H4MS1G	3700	460/3/60	2	Type M	M50	2	1279	208	2046
RTU 19	Sales	Lennox	SCA120H4ME1G	3700	460/3/60	2	Type C	CW-H15	1	2500	460/3/60	2000

REVIEW OF SYSTEM DESIGN FOR PILOT

Ultimately the research team settled on a design that supplies product air from the indirect evaporative cooler through the relief opening of each existing rooftop unit, as illustrated in Figure 2. This arrangement leaves the outside air opening available for economizer operation and ventilation when indirect evaporative cooling is not needed. In order to avoid pushing air backward through the return ductwork, a relief damper was installed in the return plenum below the rooftop unit. This allows flow in the proper direction when the rooftop unit fan operates and the return plenum is at negative pressure. The damper closes passively when the rooftop unit is at idle and the return plenum is at positive pressure because the indirect evaporative cooler is serving flow. When the combination operates as indirect evaporative only, the existing return air damper is adjusted to provide an appropriate backpressure. However, since the return air damper and outside air damper were physically linked in the existing equipment, the retrofit required breaking the link and adding a separate motorized actuator to manage the outside air damper position separate from the return air damper.

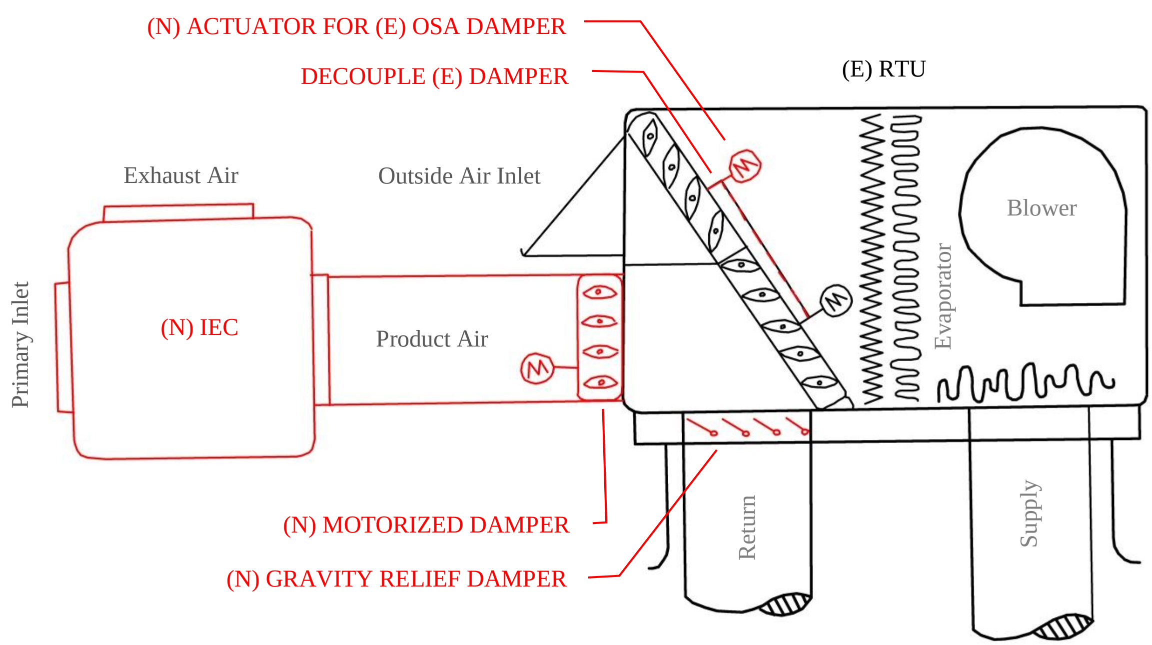

FIGURE 2. SCHEMATIC OF PHYSICAL RETROFITS CONDUCTED FOR INDIRECT EVAPORATIVE COOLER ADDITION

The system was controlled to specific damper position set points and fan speed set points for each operating mode. In an indirect evaporative only mode the combination of fan speed and damper position was selected to maintain an appropriate back pressure condition for each indirect evaporative equipment. When commissioning these set points, the research team was able to set the rooftop unit variable speed supply fan to a minimum idling speed, and was required to apply a substantial amount of damper closure to reach target pressures in the product plenum. The rooftop unit supply fan could have easily been turned off for this mode of operation.

The nominal full speed product air-flow rate from the six separate indirect evaporative systems exceeds the continuous ventilation requirement of 12,000 *cfm*. Therefore, each indirect evaporative unit is controlled to operate at part speed during a first stage call for cooling, then ramp to full speed only with a second stage call for cooling. This control feature was debated, since the additional cooling delivered at full speed would be spread to adjacent zones and likely offset more compressor cooling. However, to avoid any risk of overcooling, and to demonstrate the higher efficiency of part speed operation, the equipment was controlled to a part speed mode, and a full speed mode. When additional cooling is needed from compressor stages, the indirect evaporative systems remain at full speed.

In order to reach required supply air-flow rates for operation in each compressor stage, the rooftop unit supply fan speed and damper positions must be adjusted. Since the rooftop unit fan requires a custom speed for each of the modes designed, the rooftop units supply fan was controlled directly, instead of relying on the indexed set points for each mode within the on-board controller. In order to make up the proper air-flow for each compressor mode, some return flow is required, and the return plenum must operate at negative pressure. In order to maintain positive pressure for the indirect evaporative product plenum, a new damper and motorized actuator were added at the point of connection with the rooftop unit. When in either compressor mode, the return air damper is opened fully, and the product air damper is adjusted. Since the product plenum must remain at positive pressure and the return plenum must operate at negative pressure this damper has to serve as substantial air-flow restriction. For second stage compressor the damper should need to operate with more than 1.0 *"WC* pressure drop.

When there is no call for cooling, but the building still requires ventilation, the product damper closes fully, and the outside air damper, return air damper, and rooftop unit supply fan modulate to provide the target ventilation air-flow.

The building's existing Energy Management and Control System (EMCS) controlled each rooftop unit using 24*VAC* signals, not unlike a conventional thermostat. However the EMCS sequence also includes a signal for economizer control. In this way every economizer on the store can be enabled at once, using a single temperature measurement and a centrally controlled changeover set point. When signaled for economizer operation, the retrofit designed utilizes return air damper to provide back pressure for the indirect evaporative system, then opens the outside air damper and controls rooftop unit fan speed to reach an air-flow appropriate for each compressor stage. In this configuration the return plenum is positively pressurized, the relief damper remains closed, the indirect evaporative cooler is allowed to operate and additional supply air-flow is drawn from outside. There is a point for ambient temperature at which it becomes more efficient to operate purely as economizer and not use indirect evaporative cooling. The pilot did not provision for this, and always allowed indirect evaporative cooling to operate as the first priority for cooling from these six units.

Airflow schematics for each mode of operation are included in Appendix A. Appendix B documents the detailed sequence of operations, denoting placeholders for each index that was selected during diagnostic field measurements for setup and commissioning. Appendix C records the complete code deployed to control the combination. Figure 4 (next page) illustrates the sequence of operations as a decision tree, and Table 2 describes each mode of operation for the combination by defining the state of each controlled component in each mode.

FIGURE 3. PHOTOS OF COMPLETE INSTALLATION FOR TYPE C (LEFT) AND TYPE M (RIGHT) INDIRECT EVAPORATIVE AIR CONDITIONERS. SIX ROOFTOP UNITS WERE RETROFIT, THREE WITH EACH EQUIPMENT TYPE

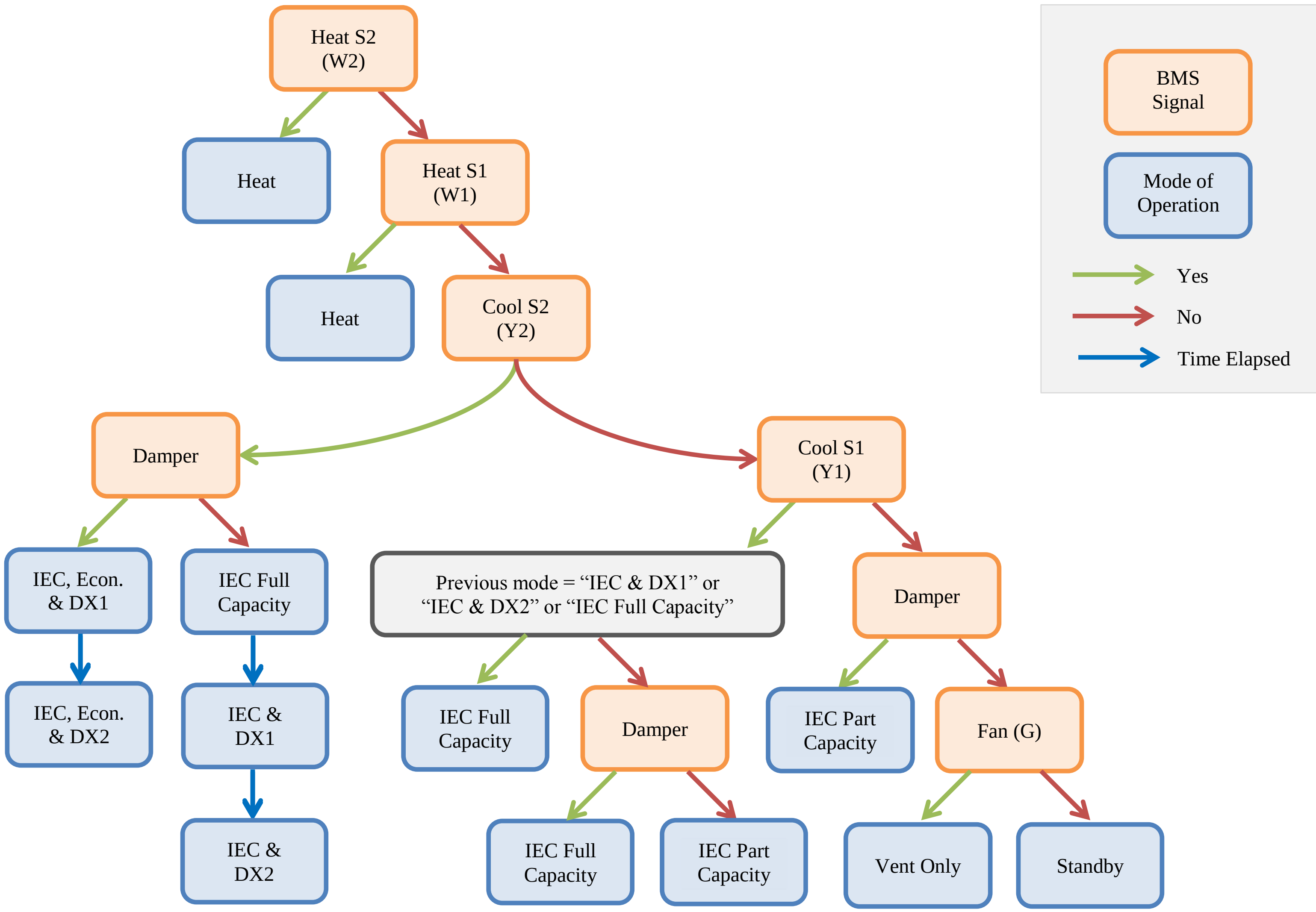

FIGURE 4. DECISION TREE FOR SEQUENCE OF OPERATIONS

TABLE 2. TRUTH TABLE TO DEFINE COMPONENT OPERATIONS IN EACH MODE OF OPERATION

Cooling Mode	*Independent Conditions*			*Component Operations*							
	ΔT=(Troom-Tsp)	Outside Temp (°F)	Time Elapsed	IEC Cooling	IEC Fan Speed	IEC Product Damper	RTU OSA Damper	RTU RA Damper	RTU Fan Speed	CMPR 1	CMPR 2
Off	0	NA	NA	OFF	OFF	CLOSED	CLOSED	OPEN	OFF	OFF	OFF
Ventilation Only	0	NA	NA	OFF	OFF	CLOSED	OPEN	CLOSED	55%	OFF	OFF
IEC Part Capacity	>0.5	NA	NA	ON	80%	OPEN	CLOSED	28.5%	37.3%	OFF	OFF
IEC Full Capacity	>1.0	NA	NA	ON	100%	OPEN	CLOSED	28.5%	37.3%	OFF	OFF
IEC & DX1	NA	>75	10m	ON	80%	66%	CLOSED	OPEN	64%	ON	OFF
IEC & DX2	NA	>75	10m	ON	80%	71.5%	CLOSED	OPEN	83%	ON	ON
IEC, ECON, & DX1	>1.5	<75	NA	ON	80%	OPEN	OPEN	84%	64%	ON	OFF
IEC, ECON, & DX2	NA	<75	10m	ON	80%	OPEN	OPEN	NA	83%	ON	ON

CUSTOM CONTROLS DEVELOPMENT

In order to operate both equipment according to the sequence designed, the retrofit required a custom-programmed controller. Hardware and software for the building's existing EMCS would not have easily adapted to serve the sequence of operations, nor would the on board controller for the rooftop air conditioner. For the purposes of this pilot, WCEC developed a custom controller that intercepts all 24*VAC* signals from the EMCS and makes logical decisions about how each system component should operate. The controller passes on appropriate 24*VAC* signals to enable each system and modulates 0-10 *Vdc* analog signals to control fan speeds and damper positions. All internal safeties and timing sequences for the packaged rooftop unit are maintained because the new controller does not cut in to manage each component separately. From the perspective of the rooftop unit, the controller acts almost like a thermostat; it signals 24*VAC* to the standard input terminals (G, Y1, Y2, W1) to activate the on board sequence for each stage of operation. The retrofit does take direct control of the supply fan VFD and all damper positions. All logic is housed on a bare bones open source programmable microcontroller, the Parallax Propeller QuickStart P8X32A. This device interfaces with and manages the solid state relays and analog output voltage controls on a custom built circuit board with terminal blocks for all low voltage inputs and outputs. The board is powered by 24*VAC* from the RTU.

FIGURE 5. PHOTO OF WCEC CONTROLLER

TEST METHODOLOGY

FIELD TESTING OF TECHNOLOGY

The overarching intent of this pilot was to explore and advance the technical opportunities for successful application of indirect evaporative cooling as a retrofit for existing high efficiency rooftop units on commercial buildings. This technology class has been laboratory tested by PG&E and others previously; the field study described here focused on characterization of field performance for two commercial products, exploration of their interactions with other air conditioning equipment, and consideration of the challenges to broader adoption for the technology.

The study does not provide a pre-post comparison to determine annual energy savings for the measure. Instead, it focuses carefully on developing a clear map of equipment performance in all modes of operation for the meteorological conditions experienced. The quantitative results presented here should provide a clear and justifiable basis for building energy use simulations or other analysis to project savings in various applications and climates.

The host provided all funds for equipment and installation for the study, and worked hand-in-hand with PG&E and WCEC to consider several possible host sites within PG&E service territory. The store selected for the demonstration in Bakersfield includes a retail sales floor and also a complete grocery. The building size required installation of multiple systems in order to adequately serve the entire ventilation needs of the sales floor. This allowed for a side-by-side comparison of two similar indirect evaporative products: referred to here as the "Type C" and the "Type M" systems. The host provided input on the overarching design constraints for the pilot, and WCEC worked closely with each manufacturer to develop a specific design that would address the unique operational requirements of each machine. The final design installed both systems in equal physical arrangements, and controlled the Type M and Type C equipment in precisely the same way, is that a clear comparison could be drawn without confusion due to differences in the operating scheme and method of installation.

WCEC worked closely with the installing contractor to see the design implemented according to plan, then worked with the contractor and each manufacturer to install controls and commission the equipment operations in each mode. WCEC designed and installed an extensive data acquisition system to monitor each of the six indirect evaporative air conditioners, in addition to the rooftop units to which they were integrated. Figure 7 and Figure 8 provide a schematic of the instrumentation plan deployed. The monitoring system is programmed to collect minute-by-minute data for each measurement, which is uploaded daily to a secure server hosted by UC Davis. Beginning in June, the equipment was monitored in operation over the course of the 2013 cooling season. Daily data was shared directly with each manufacturer in order to solicit ongoing input and feedback about the operating behavior and observed performance. This collaboration proved invaluable for the pilot, as the team was able to identify and troubleshoot challenges within a reasonable time period.

The project required a significant amount of attention by the research team, including several site visits throughout the course of study to diagnose and repair issues with equipment operations, controls, and data acquisition systems. The timeline of significant events associated with the project is summarized in the following section. Data was collected over the course of the entire summer, but the analysis presented here is limited to a period of several weeks at the end of the summer when all six systems were operating reliably.

Further, the quantitative results presented focus mainly on performance of each indirect evaporative air conditioner, separate from the rooftop unit to which each is attached. Some of the analysis does describe interaction with the rooftop units, but no attempt is made to develop an analysis of capacity and coefficient of performance for the combination. We believe it is most fair to compare the two indirect evaporative products according to this narrow characterization in order to avoid confounding results that should be attributed to performance differences between each rooftop unit. Similarly, the study does not draw conclusions about the extent to which indirect evaporative cooling reduced compressor operation for each rooftop unit. It is apparent that these systems do reduce compressor operation, but each unit operates differently according to the loads in each zone, so a direct comparison of the aggregate compressor runtime for each unit does not offer a decent measure of the impact of each system.

While quantitative analysis focuses mainly on the performance of each indirect evaporative system, observations of the equipment in combination provide a wealth of qualitative insight into the advantages and challenges of the design approach that was applied. For example, the measurements presented indicate substantial thermal losses across the rooftop unit, an impact that should be avoided in future applications.

PROJECT TIMELINE

Observations in June led everyone reviewing data to conclude that the initial control scheme implemented should be revised. The controller was initially programmed to return the indirect evaporative systems to part capacity operation whenever the call for cooling dropped from second stage (Y2) to first stage (Y1). This resulted in repetitive cycling behavior where the indirect evaporative system would run at part speed for several minutes, then full speed for several minutes, then first stage compressor, then second stage compressor, then return to part speed and begin the cycle over again. It was clear from this behavior that the indirect evaporative system should remain at full speed until the zone is completely satisfied. By dropping back to part speed the controller was effectively under-predicting the steady state load, and thus causing the compressors to operate later on in order to catch back up. The control scheme was revised the first week in July, following which all units began to exhibit a much more steady behavior, with far less compressor operation. In fact, for the rest of the year it was very rare to see any of the rooftop units run with second stage compressor. It is clear that many second stage compressors could be disconnected permanently without any downside. Each second stage compressor represents roughly 5 kW demand at peak, this measure could allow for a substantial reduction of connected load.

Observations from the same time identified a handful of recurring faults with some systems. Through collaboration with each manufacturer these faults were addressed in early July. One fault observed was the result of installation, one was caused by a faulty solenoid valve, and one resulted from a failed drain water flow meter installed by the research team. These issues were resolved in early July when the controls were revised, then data collection proceeded for several more weeks.

In late August, manufacturer's review of the data indicated that measured performance for one unit was not meeting manufacturer performance expectations. After two weeks of pondering over the data it was predicted that the filters had not been replaced since installation, which could significantly disrupt air-flow. The research team visited the site in cooperation with each manufacturer and found the air filters were so heavily soiled that air-flow was substantially impeded. In fact, the filters on a number of units had collapsed. Upon review of the data, it appears that the equipment should require filter change once each month. After roughly six weeks of operation, data analysis reveals a slow but observable decline in fan power and system pressure, corresponding with reduced air-flow. Since data in July and August represented operation with low air-flow and soiled filters, the period of performance assessment was restricted to the several weeks following filter replacement for each unit (September-October 2013). *Operating Pressures and Power Draw* in the *Results* section records system operating pressures and fan power measurements with and without soiled filters for one Type C system and one Type M system.

FIGURE 6. FILTERS REQUIRE REPLACEMENT EVERY 30 DAYS OR ELSE AIR-FLOW WILL BE RESTRICTED AND MAY LEAD TO FILTER FAILURE AND DEGRADED SYSTEM PERFORMANCE

Table 3 records the occurrence and duration of significant events throughout the course of the pilot. The performance results presented herein are restricted to September – October 2013, when all six units were operating to specification reliably, and according to the desired sequence of operations. Although all equipment were operating and actively cooling in July and August, our analysis excluded these periods because air-flow could not be well quantified for the periods after filters had become so soiled.

TABLE 3. TIMELINE OF PROJECT ACTIVITIES AND SIGNIFICANT EVENTS

Project Activity & Significant Events	*Dates*
Initial site visits and project scoping	October 2012
Design development and project preparation	November – December, 2013
Controller design, construction, and testing	January 2013
Equipment installation	January 2013
Startup, commissioning, controls programming	February 4th – 6th, 2013
Monitoring setup, troubleshooting, and final commissioning	May 13th – June 6th, 2013
Official beginning for data collection	June 8th, 2013
Revision to sequence of operations	July 5th, 2013
Review of data shows first signs of filter degradation	July 7th, 2013
Revise controls sequence	July 1st, 2013
Obvious signs of trouble for some units prompts on-site investigation	September 1st, 2013
Troubleshooting, filter replacement, and flow diagnostics	September 12th – 13th, 2013
Data period used for analysis	September 14th – October 24th, 2013
Mnfr. replaces heat exchanger for RTU 19	October 15, 2013
Data analysis and reporting:	October 2013 – April 2014
Water distribution on roof shut off following significant freeze damage	January 2014
Final on site diagnostic measurements to support analysis:	April 2014

MONITORING PLAN

The research team developed a monitoring plan to measure electricity consumption, thermal energy service, water use and component status in order to determine energy and water use efficiency in each mode of operation. Ambient temperature and humidity conditions were measured in order to map performance against environmental variables, and a number of intermediate points, such as static pressures, water temperatures, and refrigerant temperatures and pressures were captured in order to develop clearer insight about the specific mechanisms that effect performance. For example, static pressure measurements throughout the system clarify the combined roles of air-flow resistances and fan speeds on heat exchanger flow rates and performance. These intermediate measurements are not needed to capture the overall cooling capacity and energy efficiency for the system, but they allowed the team to better review the operation on control sequences in order to ensure that static pressures were maintained in an appropriate range.

Electricity consumption was measured separately for each rooftop unit and each indirect evaporative cooler. The monitoring package utilized three phase power meters that measure voltage, current, and power factor on each leg of the three phase systems. This allows for analysis of the both the real and reactive power. Since power factor for motor systems is often between 0.6 – 0.9, this characteristic is important to understand. The operation of sub components such as fans and compressors were monitored with individual current transducers on a single leg. These measurements are used mostly for determining the mode of operation for any given instance, though they can also be used to estimate sub-component power use where desired. The temperature and humidity of each air-flow stream is measured at a single point, and static pressure in each location is measured as a physical average.

Table 4 summarizes all of the measurements made on each system and records the specific instrument utilized. Figure 7 and Figure 8 mark out conceptual monitoring schematics to show the general layout of instrumentation for the Type C retrofits and Type M retrofits. RTU 16 and RTU 17 received the complete monitoring plan described in order to investigate the more nuanced details about internal system functions. A more limited monitoring package was installed for the remaining four units. The limited package still allowed for complete assessment of cooling capacity, water and energy consumption, but excluded refrigerant circuit measurements and differential static pressure measurements throughout the system.

Table 4. Instrumentation Schedule for the Six Units

Measurement	*Device*	*Details*	*Uncertainty*	*Units*
T_{OSA}	Vaisala HUMICAP HMP110	-40°F ≤ T_{OSA} ≤ 176°F	± 0.36 °F	All 6
RH_{OSA}	Vaisala HUMICAP HMP110	0% ≤ RH_{OSA} ≤ 100%	± 1.7% RH	All 6
T_{RA}	Vaisala HUMICAP HMP110	-40°F ≤ T_{OSA} ≤ 176°F	± 0.36 °F	All 6
RH_{RA}	Vaisala HUMICAP HMP110	0% ≤ RH_{OSA} ≤ 100%	± 1.7% RH	All 6
T_{SA}	Vaisala HUMICAP HMP110	-40°F ≤ T_{OSA} ≤ 176°F	± 0.36 °F	All 6
RH_{SA}	Vaisala HUMICAP HMP110	0% ≤ RH_{OSA} ≤ 100%	± 1.7% RH	All 6
T_{PA}	Vaisala HUMICAP HMP110	-40°F ≤ T_{OSA} ≤ 176°F	± 0.36 °F	All 6
RH_{PA}	Vaisala HUMICAP HMP110	0% ≤ RH_{OSA} ≤ 100%	± 1.7% RH	All 6
T_{EXH}	Vaisala HUMICAP HMP110	-40°F ≤ T_{OSA} ≤ 176°F	± 0.36 °F	All 6
RH_{EXH}	Vaisala HUMICAP HMP110	0% ≤ RH_{OSA} ≤ 100%	± 1.7% RH	All 6
$\Delta P_{PITOT,\ PA}$	Dwyer	0-0.25 "WC = 4-20 *mA*	± 0.0025 "WC	All 6
$\Delta P_{PITOT,\ SA}$	Dwyer	0-0.25 "WC = 4-20 *mA*	± 0.0025 "WC	All 6
ΔP_{FAN}	Dwyer	0-2.5 "WC = 4-20 *mA*	± 0.025 "WC	16, 17
ΔP_{1st}	Dwyer	0-2.5 "WC = 4-20 *mA*	± 0.025 "WC	16, 17
$\Delta P_{2nd\ (CW)}$	Dwyer	0-1 "WC = 4-20 *mA*	± 0.01 "WC	16
$\Delta P_{2nd\ (CO)}$	Dwyer	0-0.25 "WC = 4-20 *mA*	± 0.0025 "WC	17
ΔP_{RD}	Dwyer	0-0.25 "WC = 4-20 *mA*	± 0.0025 "WC	16, 17
$\dot{V}_{SUPPLY}$	OMEGA FTB 4105 A P	1 pulse per gallon	± 1.5%	All 6
$\dot{V}_{DRAIN}$	OMEGA FTB 4107 A P	1 pulse per gallon	± 1.5%	16, 17
$AO_{OSA\ Damper}$	OA Damper control signal	0–10 V_{DC}	± 0.1V	All 6
$AO_{PA\ Damper}$	PA Damper control signal	0–10 V_{DC}	± 0.1V	All 6
$AO_{RA\ Damper}$	RA Damper control signal	0–10 V_{DC}	± 0.1V	All 6
$AO_{IEC\ FAN}$	IEC fan speed signal	0–10 V_{DC}	± 0.1V	All 6
$AO_{RTU\ FAN}$	RTU VFD fan speed signal	0–10 V_{DC}	± 0.1V	All 6
CT_{C1}	NK AT1-005-000-SP	0-10 A_{AC} to 0-5 V_{DC}	± 0.1 A	All 6
CT_{C2}	NK AT1-005-000-SP	0-10 A_{AC} to 0-5 V_{DC}	± 0.1 A	All 6
CT_{CF}	NK AT1-005-000-SP	0-10 A_{AC} to 0-5 V_{DC}	± 0.1 A	All 6
$T_{LOW\ C1}$	Thermocouple Type T, 3 AVG	Tape and Insulate	± 0.52 °F	16, 17
$T_{LOW\ C2}$	Thermocouple Type T, 3 AVG	Tape and Insulate	± 0.52 °F	16, 17
$T_{HI\ C1}$	Thermocouple Type T, 3 AVG	Tape and Insulate	± 0.52 °F	16, 17
$T_{HI\ C2}$	Thermocouple Type T, 3 AVG	Tape and Insulate	± 0.52 °F	16, 17
$T_{CD\ OUT\ 1}$	Thermocouple Type T, 3 AVG	Tape and Insulate	± 0.52 °F	16, 17
$T_{CD\ OUT\ 2}$	Thermocouple Type T, 3 AVG	Tape and Insulate	± 0.52 °F	16, 17
$P_{LOW,\ C1}$	ClimaCheck, 35 *bar*	0-35 *bar* to 1-5 V_{DC}	± 0.35 *bar*	16, 17
$P_{LOW,\ C2}$	ClimaCheck, 35 *bar*	0-35 *bar* to 1-5 V_{DC}	± 0.35 *bar*	16, 17
$P_{HI,\ C1}$	ClimaCheck, 50 *bar*	0-50 *bar* to 1-5 V_{DC}	± 0.50 *bar*	16, 17
$P_{HI,\ C2}$	ClimaCheck, 50 *bar*	0-50 *bar* to 1-5 V_{DC}	± 0.50 *bar*	16, 17
$P_{CD\ OUT,\ C1}$	ClimaCheck, 50 *bar*	0-50 *bar* to 1-5 V_{DC}	± 0.50 *bar*	16, 17
$P_{CD\ OUT,\ C2}$	ClimaCheck, 50 *bar*	0-50 *bar* to 1-5 V_{DC}	± 0.50 *bar*	16, 17
CT_{PUMP}	NK AT1-005-000-SP	0-10 A_{AC} to 0-5 V_{DC}	± 0.1 A	All 6
T_{SUMP}	Thermistor	10kΩ	± 0.32 °F	16,18,19
KW_{IEC} & KW_{RTU}	Dent Powerscout 3	RS485 connection to dataTaker	± 1%	All 6

INSTRUMENTATION SCHEMATIC FOR TYPE C RETROFITS

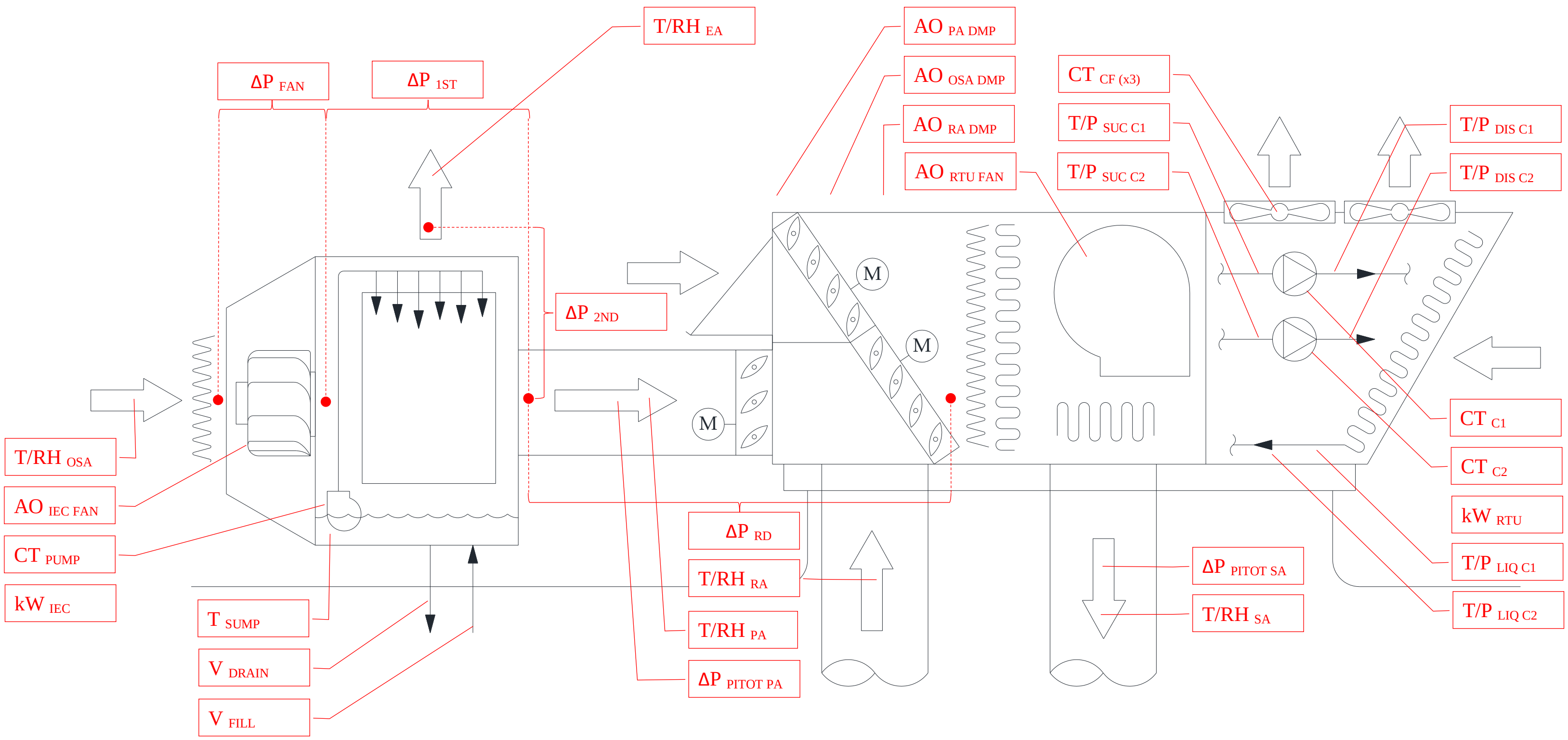

FIGURE 7. INSTRUMENTATION SCHEMATIC FOR TYPE C SYSTEMS (RTU 16, RTU 18 & RTU 19)

INSTRUMENTATION SCHEMATIC FOR TYPE M RETROFITS

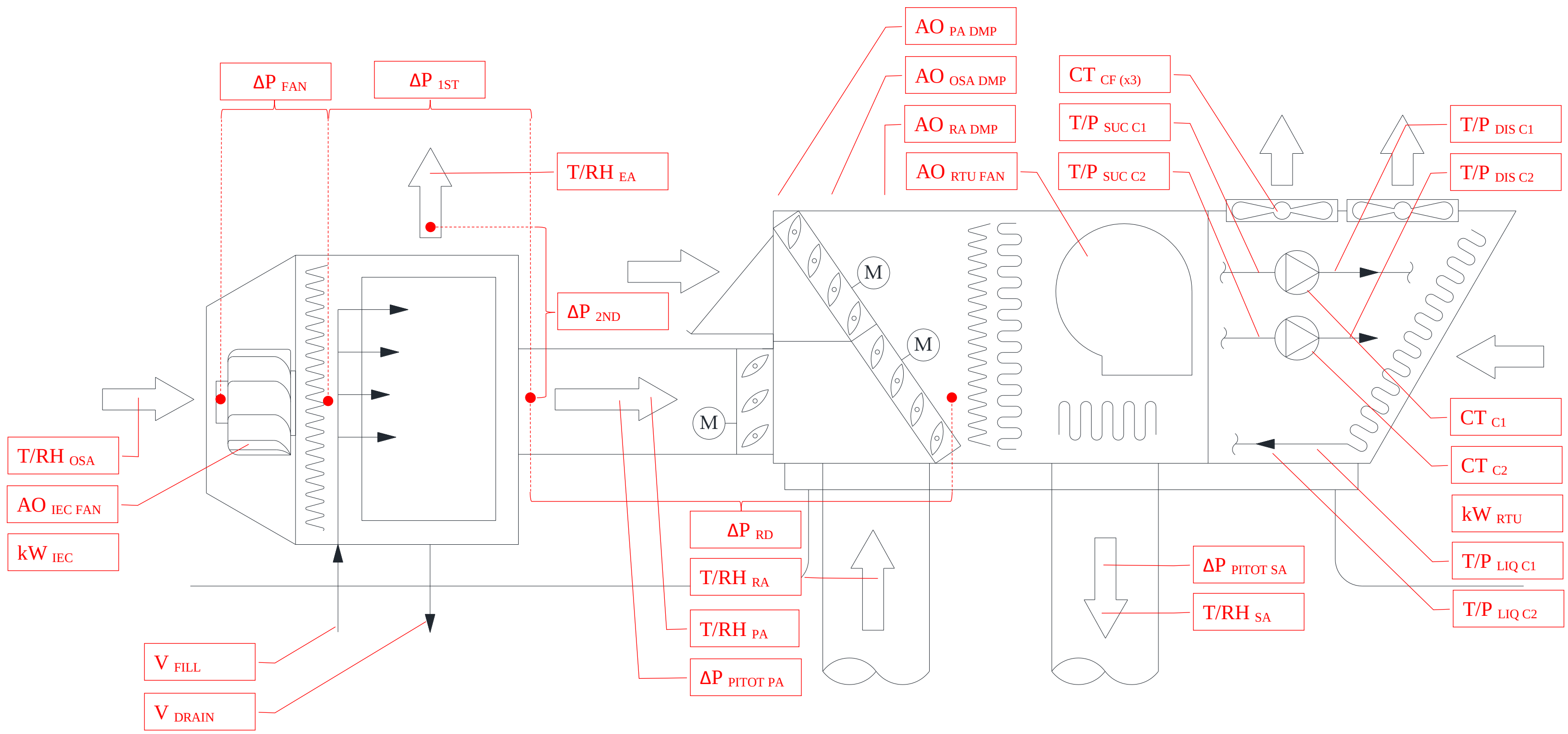

FIGURE 8. INSTRUMENTATION SCHEMATIC FOR TYPE C SYSTEMS (RTU 16 RTU 18 & RTU 19)

Data Confidence

The manufacturer calibrated accuracy of each instrument used in this monitoring effort is listed in Table 5. These measurements are used to calculate a number of performance metrics for each system, the uncertainty for which is determined by propagation of error. Since accuracy for each device is dependent on the magnitude of the measurement, the uncertainty of each metric also changes as operating conditions change. Table 5 summarizes the calculated uncertainty for several key metrics for a single condition and operating scenario[1]. The values recorded here indicate the calculated uncertainty resulting from manufacturer stated performance for the instrumentation used, and from the calculation methods documented in section "*Definition and Calculation of Performance Metrics*". The degree of accuracy listed here does not account for any methodological uncertainty associated with features such as sensor placement, or transient interactions between equipment operation and sensor response.

Table 5. Uncertainty for Key Measurements and Calculated Metrics

Metric	*Uncertainty*
Measured Values	
Electrical Power	±0.0155 kW
Analog Output	±0.0101 V_{DC}
Air Temperature	±0.36 °F
Air Relative Humidity	±1.5 %RH
Differential Pressure	±0.0025 inWC
Supply Water	±0.01 gal
Drain Water	±0.01 gal
Product Airflow	±2.00 %
Calculated Values	
Absolute Humidity	±0.000406 $lb_{m,\ water}$ / $lb_{m,\ dry\ air}$
Sensible IEC System Capacity	±6.24 kBTU/hr
Sensible IEC Room Capacity	±4.51 kBTU/hr
Sensible IEC System COP	±1.19
Sensible IEC Room COP	±0.861
Water Use Efficiency	±0.0562 gal/ton hr

[1] Uncertainties recorded in Table 5 were calculated at for the following conditions: T_{OSA}= 85.17°*F*, RH_{OSA}=14.29 %, T_{PA}=60.82 °*F*, RH_{PA}=32.81 %, Power=1.548 *kW*, supply water flow = 1 *gallon*, drain water flow =1 *gallon*, differential pressure = 0.1184 *inWC*, HR_{OSA}=0.0037 *lbm, water / lbm, dry air*, Sensible IEC System Capacity=73.21 *kBtu/hr*, Sensible IEC Room Capacity=51.63 *kBtu/hr*, and Water Use Efficiency=2.049 *gal/ton hr*.

DATA ANALYSIS

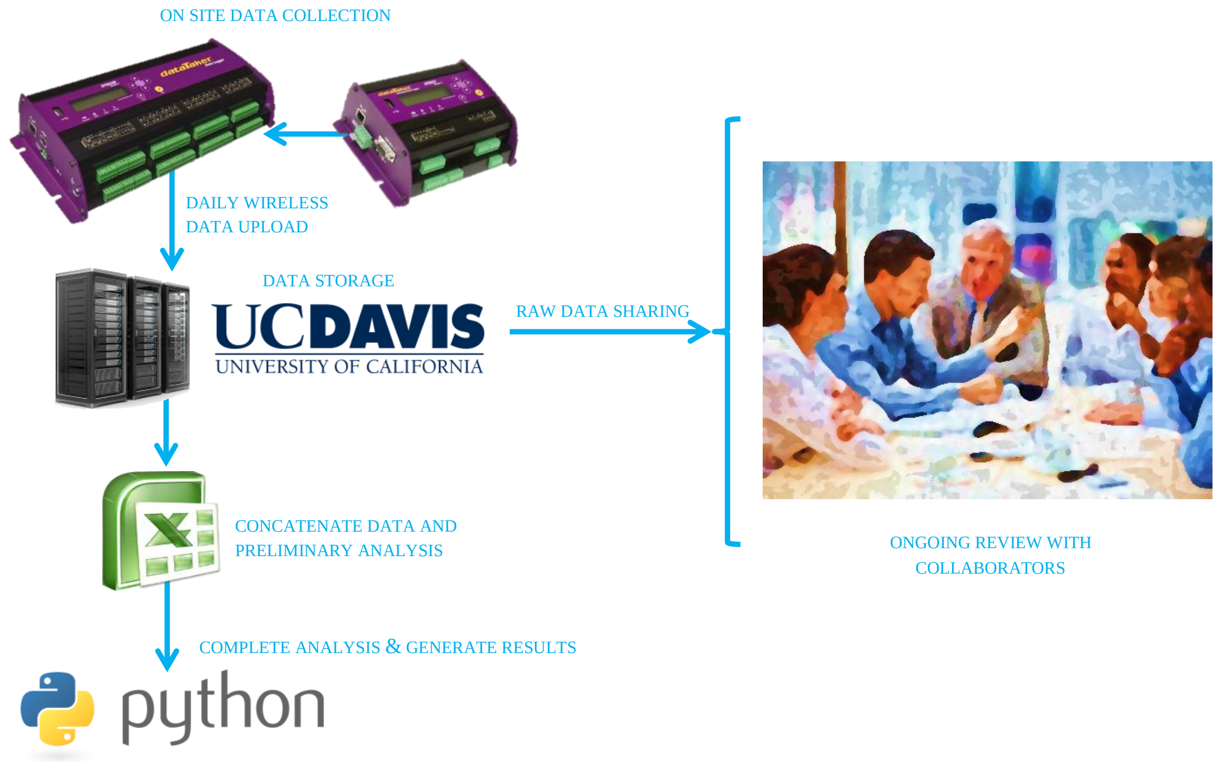

FIGURE 9. FLOW CHART FOR DATA COLLECTION AND ANALYSIS

Analog and digital measurements from each rooftop unit combination were collected by a data acquisition module located on board each unit. Data acquisition modules from the three Type M systems were grouped onto one hard wired data acquisition network, and the modules from the three Type C systems were grouped onto a separate hard wired data acquisition network. These two networks are connected wirelessly to the EDGE cellular network and communicate records to an SFTP sever hosted by UC Davis. One minute interval data was collected from each unit 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. Data for each unit is collected on this server as a separate CSV file each day. Each manufacturer and project partner was given read access to relevant data, which was valuable for continuous review of the equipment operations and allowed the team to identify and address faults within a reasonable time.

Raw day-by-day datasets for each unit were concatenated into larger datasets that group minute interval data into month long time series sets. These month-long files were then used as manageable chunks for further data analysis and visualization. As discussed previously, the most useful period for analysis and comparison of the two products was from September and October 2013. Therefore, day-by-day datasets for each unit were concatenated into time series groups for this custom period.

Data analysis and visualization was conducted using custom developed algorithm for data processing in Python (Rossum). The tools and functions available through Pandas are especially well-suited for manipulation and analysis of large time series datasets. A custom script and library of analysis functions was developed for this project that can be applied to analysis of similar monitoring and evaluation efforts in the future. In particular, the custom 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.

DEFINITION AND CALCULATION OF PERFORMANCE METRICS

CALCULATING WET-BULB EFFECTIVENESS

Wet-bulb effectiveness (WBE) measures the extent to which an evaporative cooler is able to cool toward the wet-bulb temperature of the inlet air. For conventional evaporative coolers, this metric remains relatively steady for a given system configuration even while meteorological conditions and system-cooling-capacity varies. For indirect evaporative systems WBE tends to change significantly with operating conditions. However, WBE is the most common metric to describe performance of an evaporative system and is currently used as the key performance input to model indirect evaporative equipment in most building energy simulation tools.

$$WBE = \frac{T_{DB\,OSA} - T_{DB\,PA}}{WBD_{OSA}} = \frac{T_{DB\,OSA} - T_{DB\,PA}}{T_{DB\,OSA} - T_{WB\,OSA}} \qquad 1$$

The metric has traditionally been used to describe performance of direct evaporative coolers, but it can also be applied to indirect evaporative equipment. Since indirect evaporative heat exchangers use a secondary air stream that can have an inlet wet-bulb temperature that is lower than that of the primary inlet, it is possible to achieve better than 100% effectiveness. The metric provides a good conceptual platform for comparing alternative products, but it does not tell the entire story for a system. Some indirect evaporative heat exchangers may sacrifice air-flow for a better wet-bulb effectiveness, which may have either positive or negative effects on the overall system efficiency.

CALCULATING COOLING CAPACITY

The system-cooling-capacity for the indirect evaporative equipment is determined at any operating condition according to the product air-flow rate and the specific enthalpy difference between the outside air inlet and product air, as described by equation 2. This is the net cooling produced by the device, including what is lost due to fan heat.

$$\dot{H}_{IEC\ System} = \dot{m}_{PA} \cdot (h_{OSA} - h_{PA}) \qquad 2$$

For a conventional rooftop unit, the system cooling is measured by the difference between the mixed air enthalpy and the supply air enthalpy, given some mixture of return air with outside air. For an indirect evaporative cooler the system cooling represents the difference between outside air enthalpy and product air enthalpy. In either case, the metric represents the total amount of cooling that is actively produced by a piece of equipment. The metric does not describe the amount of cooling delivered to a conditioned zone, since some cooling may arrive for free when outside air is lower energy than return air, and since a significant amount of capacity must be used for cooling ventilation air when outside air is warmer than return air.

The room-cooling capacity, given by equation 5, is the cooling that is actually of service to the zone. This metric accounts for 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. Note that the analysis presented here centers on the indirect evaporative equipment, separate from the rooftop air conditioners to which they were attached, so the room-cooling capacity is calculated as the enthalpy difference between room air and product air.

$$\dot{H}_{Room} = \dot{m}_{PA} \cdot (h_{RA} - h_{PA}) \qquad 3$$

Since there was not return air-flow in every mode of operation, and since data indicated some leakage of product air into the return air path during "*IEC Only*" modes, return air measurements did not provide a reliable description of the zone temperature. Further, since the indirect evaporative systems deliver ventilation and cooling by displacement it is difficult to measure the temperature of air displaced. Therefore, room air temperature for this study is assumed to be 78°F in order to provide a steady foundation for comparison of each system's room cooling capability.

As will be discussed in the results section, our observations show that there are significant thermal losses as product air from the indirect evaporative equipment flows through each rooftop unit. For the comparison of each indirect evaporative product presented here, it is therefore important to separate the performance of each indirect evaporative cooler, from the external losses incurred. To calculate the actual cooling service provided by the combination, one would supplant the product air terms in equations 2–5 with the supply air characteristics at the outlet of the rooftop unit.

The assessment presented here is generally concerned with a system's ability to produce sensible cooling; since ambient humidity in hot-dry climates doesn't typically require dehumidification for comfort. Also, thermostat controls for this application only respond to sensible effects, so it is therefore not appropriate to compare the performance presented here to the sensible-plus-latent capacity from conventional air conditioners. Thus the sensible room-cooling is determined according to:

$$\dot{H}_{Room}^{sensible} = \dot{m}_{PA} \cdot C_p \cdot (T_{RA} - T_{PA}) \quad (4)$$

And the latent room-cooling is determined as:

$$\dot{H}_{Room}^{latent} = \dot{H}_{Room} - \dot{H}_{Room}^{Sensible} \quad (5)$$

The ventilation cooling capacity is the difference between the system cooling and room-cooling, and it can also be calculated according to equation 6

$$\dot{H}_{ventilation} = \dot{m}_{SA} \cdot (h_{OSA} - h_{RA}) \quad (6)$$

In the event that outside air is cooler than return air, the ventilation cooling capacity calculates as negative. This thermal energy can be applied usefully as "free cooling" for the room when there is a room-cooling load. Since the system-cooling-capacity is split between room-cooling and ventilation cooling, when ventilation cooling is negative, the room-cooling can be greater than the system-cooling-capacity.

Calculating Coefficient of Performance

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

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

The COP numbers presented here should not be confused with or compared directly to the AHRI nominal EER, SEER, or IEER values for conventional air conditioners, since those metrics are developed to describe performance at very specific conditions which are not appropriate for the indirect evaporative equipment assessed here. For this report, COP is presented as a generic metric that varies with conditions, and with frame of reference.

For comparison to conventional rooftop air conditioners in western climates, performance ought to only credit a portion of the total cooling delivered by a system. First, cooling of any excess ventilation air is should not be counted as useful thermal energy service since the conventional alternative would not have to carry the increased load associated with this excess outside air. Second, the latent cooling capacity of a conventional air conditioner should not be credited since it represents unneeded dehumidification that does not impact thermostat control of the building. The question of whether or not a zone requires dehumidification should be considered separately.

Since the equipment studied here is designed to fill ventilation needs for the store, the system does not provide excess outside air, and the therefore the entire cooling effect is credited. Further, the analysis presented here only credits sensible cooling capacity, since dehumidification is usually not required for the application in question, and is generally not required for commercial buildings in California. These assumptions align with methods presented for other evaluations for the Western Cooling Challenge. The Sensible Coefficient of Performance can be expressed as:

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

Further, performance results are described both in terms of the Sensible System EER, and the Sensible Room EER. The first metric considers how much energy is consumed to generate a specific amount of cooling across the machine; the second considers the ratio of that energy consumption to the cooling effect on the room:

$$COP_{system}^{sensible} = \frac{\dot{H}_{system}^{sensible}}{\dot{E}_{system}} \tag{9}$$

$$COP_{room}^{sensible} = \frac{\dot{H}_{room}^{sensible}}{\dot{E}_{system}} \tag{10}$$

WATER USE EFFICIENCY

The evaporative equipment studied here makes substantial gains for energy efficiency, but supplants that electricity saved with water consumption on site. Previous research indicates 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. However, this metric is most sensitive to the embodied water content in energy generation, and to the water use efficiency on site. The water use efficiency metric measures the amount of water consumed relative to degree of cooling delivered:

$$WUE = \frac{\dot{V}_{water}}{\dot{H}_{sensible}} \tag{11}$$

The water use 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 also that amount of water drained for bleed in order to manage water quality.

RESULTS & DISCUSSION

The following pages document the results of a thorough analysis of equipment performance. The metrics assessed focus mainly on the indirect evaporative cooling equipment apart from the rooftop air conditioners to which they are attached. This analysis is done by considering measured conditions and air-flow rates at the inlet and product outlet of the indirect evaporative equipment. The analysis endeavors to present a clear characterization of system performance over a range of operating conditions and in different modes. This level of performance characterization is sorely needed to inform the sound estimate of energy savings potential for the technology in various climates and applications.

Interactions between each indirect evaporative air conditioner and the rooftop unit to which it is combined are also assessed here. This analysis is based mostly on qualitative assessment, field observation, and professional experience. For the sake of brevity, the measured cooling capacity and energy efficiency of the combination is not presented here. Since this equipment can be applied in various ways, the results are presented in a way that performance characteristics for these products can be applied to alternate scenarios, without concern for the effects of this specific application. The results do not indicate the extent to which compressor runtime is reduced for the rooftop units in focus. Accordingly, they also do not quantify the impact on other rooftop units in the store, and do not present an estimate of overall energy savings for the building.

The results describe water use efficiency for the systems tested. Since the internal design and operating sequence for the two indirect evaporative systems tested are different in several important ways, these results present measurements and general observations to explain some of the features and dynamics that make each system unique.

The results section is organized to present a side-by-side comparison of all six indirect evaporative air conditioners for each metric of analysis. All six charts on each full page in the results section present a single performance metric. For each page, all of these charts share similar vertical and horizontal axis. They are scaled in common. Each group of charts is given a common figure title, but results from each unit are marked separately for clarity. Some results, such as presentation of the psychrometric performance and analysis of product temperature dynamics, consolidate data from multiple units but present results for the Type M and Type C systems separately.

Largely, we find that the two indirect evaporative coolers operate with similar energy efficiency, but that there are apparent differences for cooling capacity, product temperature, evaporative effectiveness, and water use efficiency. The level of performance achieved by both systems is exceptional. Full load sensible system cooling efficiency at peak conditions appears to reach COP=15 (EER>50), while part load efficiency is as high as COP=25 (EER>85). Although the indirect evaporative air conditioners do not cover all of the cooling requirements for the building, they easily cover ventilation cooling needs under all conditions, and while also providing a substantial amount of room-cooling. Notably, for the period of analysis presented, second stage compressor was never required in these zones, and operation of the first stage compressor was a rare occurrence.

COEFFICIENT OF PERFORMANCE FOR SENSIBLE SYSTEM-COOLING

The results in Figure 10 chart coefficient of performance: the ratio of energy consumption by each indirect evaporative unit to the sensible cooling generated by the system. The metric is presented as a function of outside air temperature. Most importantly, the level of efficiency shown by these measurements is remarkable. The results indicate that full load COP = 15 a peak conditions and that part speed operation can reach COP = 25. This should be compared to vapor-compression cooling which will operate with COP=4 for a similar scenario.

It is first of all clear that performance results for the Type C system appear to be much more varied than the results for the Type M equipment. Upon thorough review of the data we conclude that this is as a result of variation in the air-flow measurement for the Type C equipment. The variation in air-flow measurement appears to be partly due to an actual variation in air-flow, and partly due to measurement error. According to on-site field diagnostics, we observe that air-flow in the location where the measurement is made is very turbulent. Since the air-flow measurement for each minute is taken as an instantaneous point, this turbulence results in measurement noise, even while the bulk air-flow rate may not be shifting in such a substantial way.

Coefficient of performance increases significantly when each system operates as part speed. This is due mostly to a significant reduction in fan power draw at part speed, and to an increase in evaporative effectiveness.

Note that system coefficient of performance declines as ambient temperature decreases. Since power draw for the indirect evaporative systems is basically constant for any given fan speed and cooling capacity is tied closely with wet-bulb depression, as ambient temperature decreases, wet-bulb depression decreases, until the point that wet-bulb depression is eventually very small. As ambient temperature decreases so does cooling load, such that a smaller cooling capacity may suffice. This trend is apparent in the data set presented here, at some point between outdoor temperature 60-65°F, part speed fan operation is adequate for cooling and the zone never calls for full speed operation. Similarly, above about 85°F it appears that part speed operation is never adequate for maintaining set point, but that that added capacity from full speed operation is usually able to provide sufficient cooling.

It is interesting to note the diverging trends between the part speed performance and full speed performance. This trend is most apparent for the Type M systems, but appears to hold true for both technologies. We expect this behavior is a result of the fact that as ambient temperature increases, wet-bulb depression increases faster than wet-bulb temperature. Since wet-bulb effectiveness remains relatively steady across a range of temperatures, but increases significantly at part speed, energy efficiency will respond proportional to the wet-bulb effectiveness.

Since power draw for the indirect evaporative air conditioner is steady for any particular operating speed, efficiency changes only as a result of change in cooling capacity. Cooling capacity changes in direct proportion to wet-bulb effectiveness and wet-bulb depression. Therefore, when wet-bulb effectiveness is higher at part speed, a change in outside air temperature, which is effectively a change in wet-bulb depression results in a larger efficiency gain.

Notably, since part speed performance is so much more efficient than full speed performance, 'oversized' equipment will result in substantial efficiency gains. This is in contrast to conventional rooftop units where part load operation rarely provides substantial efficiency improvements. The part load efficiency characteristics certainly benefited overall performance in this application. For the period of analysis, more than half of the cumulative cooling was delivered at part speed. Ventilation requirements would have been met with only five systems fixed to full speed operation, but the sixth unit allowed all systems to spend a fraction of time operating at part speed, where efficiency roughly doubled.

During the period of analysis, first stage compressor cooling (DX1) operated for short periods of time on RTU 18, 19, 13, 15 and 17, but not RTU 16. Second stage vapor compression cooling (DX2) was practically never required. These results chart the sensible system coefficient of performance for the each indirect evaporative unit, separate from the rooftop air conditioner. These results do not describe overall performance of the combination, though they do capture any effect that rooftop unit operation has on performance of the indirect evaporative air conditioner. For most units, it appears that operation in DX1 does not impact the indirect evaporative equipment. However, for RTU 19, efficiency drops precipitously for operation with DX1. The team became aware of this fact during the course of study, and at first believed it was related to the filter failures discussed earlier. Since the effect persisted even after filter replacement, the manufacturer eventually elected to replace the heat exchanger. However, since there was no obvious issue with the heat exchanger following replacement, our best judgment leads us to believe that damper set points within the custom control program were not selected properly, which resulted in low back pressure in the product plenum and degraded cooling performance. This emphasizes the importance of commissioning and controls

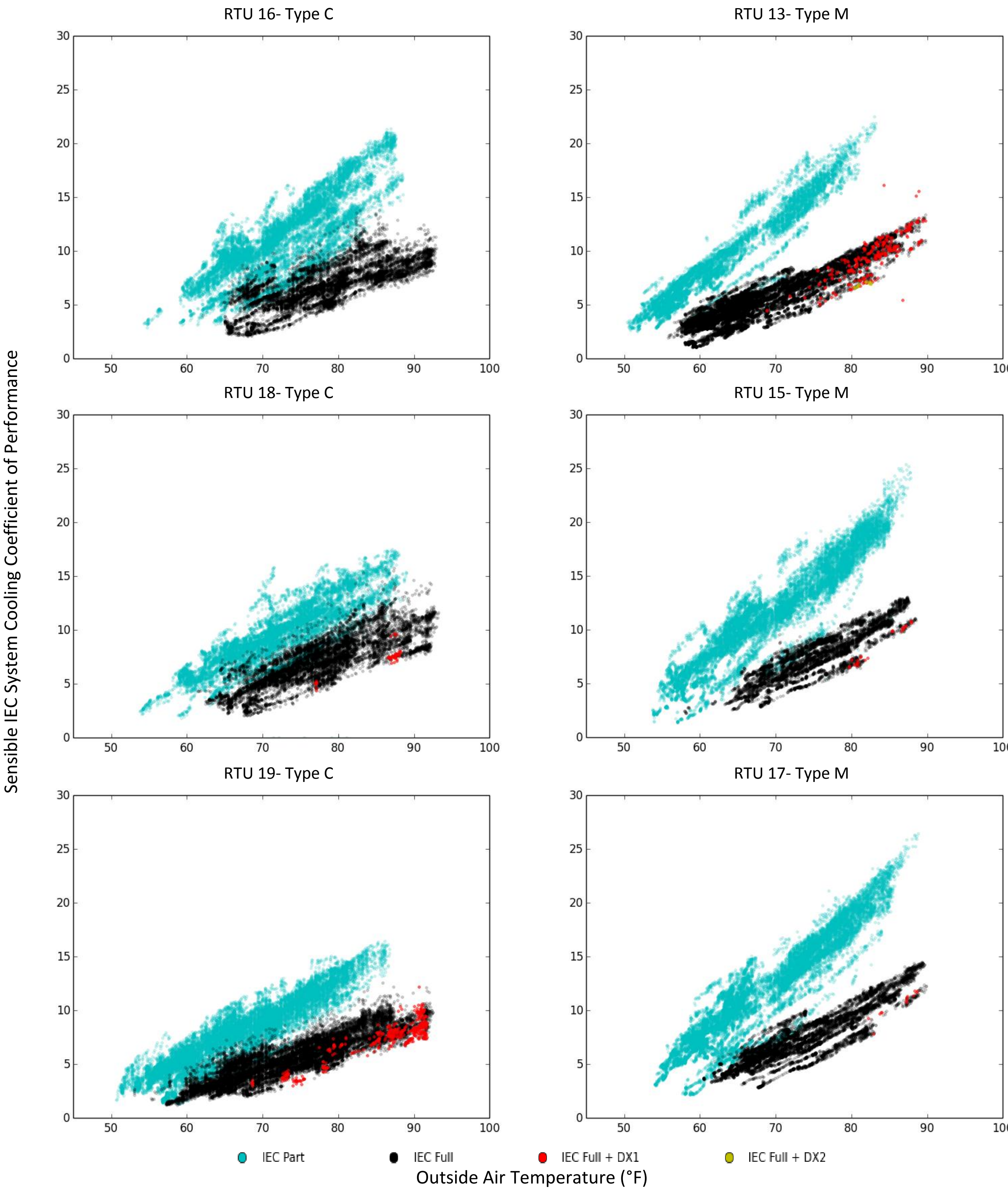

Figure 10. Sensible Indirect Evaporative Cooler System Cooling Coefficient of Performance by Operating Mode versus Outside Air Temperature

for RTU 19, especially during DX operation. This observation speaks to the fact that performance for these systems is highly sensitive to downstream pressure, and that systems must be controlled carefully.

Despite the attention and research efforts put toward the design, commissioning, operation and troubleshooting of the equipment, this mechanism was still not optimized because it required too much time in setup and commissioning. We expect that more reliable performance might be had from a system that actively controls fan speed and damper position to maintain appropriate static pressure splits and air-flow ratios. The control scheme that was deployed for this pilot utilized fixed damper set points, which could not adjust to unanticipated changes in downstream resistance, differences in operating pressures between each unit, or physical adjustments made to the equipment over time. An automated controller would eliminate the potential for these errors, and would reduce the time required for setup.

Sensible System-Cooling-Capacity

Figure 11 charts the sensible system-cooling-capacity for each indirect evaporative air conditioner as a function of outside air temperature. Cooling capacity increases as outside air temperature increases because the wet-bulb potential expands as outside temperature rises. Warmer air at the inlet to the indirect evaporative air conditioner has a greater capacity for water evaporation, and therefore larger potential for evaporative cooling. Therefore, as cooling load increases, capacity also increases without the need for additional electrical input.

Although limited, data from RTU 13 and RTU 16 indicate that when the rooftop unit engages compressor cooling and the blower speeds up, the control strategy installed is able to balance air-flows appropriately so that there is no significant impact on cooling capacity from the indirect evaporative system supplying to the inlet of the rooftop.

The analysis in Figure 11 does not show cooling capacity for the rooftop unit. As results later will indicate, there is a significant loss in cooling capacity as the indirect evaporative product air passes through the rooftop unit. Also, while compressor operation certainly adds cooling capacity, this analysis does not assess the impact that the indirect evaporative cooler has on vapor-compression performance. For high ambient conditions we expect that unloading the vapor-compression system has a beneficial effect on performance. However, for lower ambient conditions, cooler air at the evaporator coil inlet could result in increased latent cooling. If coil air-flows are not managed appropriately, there is risk that the refrigerant system could drift toward a low suction pressure, or could fail to vaporize all liquid refrigerant. There is no indication that these concerns occurred for the systems studied here, but the mechanisms should be carefully avoided by design in future projects.

To be clear, part speed operation slows the fans down to a point that reduces air-flow to 77%. However, it is apparent from this analysis that the decrease in air-flow does not always result in an equal decrease for cooling capacity. While there is a significant drop in cooling capacity for the Type M equipment at part speed, sensible cooling capacity for the Type C system at part speed is virtually indistinguishable from full speed. This effect is likely due to a significant increase in the wet-bulb effectiveness at part speed operation, such that even while air-flow declines, product temperature declines and system-cooling-capacity remains relatively steady.

The results in Figure 11 also show that system-cooling-capacity diminishes significantly at low ambient temperatures. It is clear that at some point, the small cooling capacity derived by the indirect evaporative cooler may not be worth the fan power that is invested to generate the effect. Figure 12 and Figure 13 explore the room-cooling capacity for the system. When ambient temperature is already cooler than room conditions economizer cooling can offer a significant room-cooling effect. Near the economizer changeover condition, indirect evaporative cooling may still offer added value, but at much lower ambient temperature it is likely better to bypass the indirect evaporative heat exchanger and shift to full economizer cooling. This type of control driven performance optimization represents an area for significant enhancements.

Again, RTU 19 records different behavior. In this case, part speed cooling capacity is actually greater than cooling capacity at full speed. This is a strange effect that isn't fully explained by an increase in wet-bulb effectiveness. It may also be the case that full speed air-flow is lower than anticipated, or that the air-flow measurement does not respond appropriately to a decrease in air-flow. Similarly, cooling capacity decreases yet again when the equipment engages DX cooling. As will be discussed later, wet-bulb effectiveness is also lower for this unit, but doesn't change perceptibly between operation at full speed and operation with DX cooling, so air-flow must be the only explanation. It is not fully clear whether the variation in air-flow is as a result of poor measurement or actual system behavior. Our best judgment leads us to believe that damper set points were not commissioned appropriately for

RTU 19. This would result in improper air-flow balance through the system, especially when the RTU fan is active for DX cooling and the product plenum is exposed to negative static pressure.

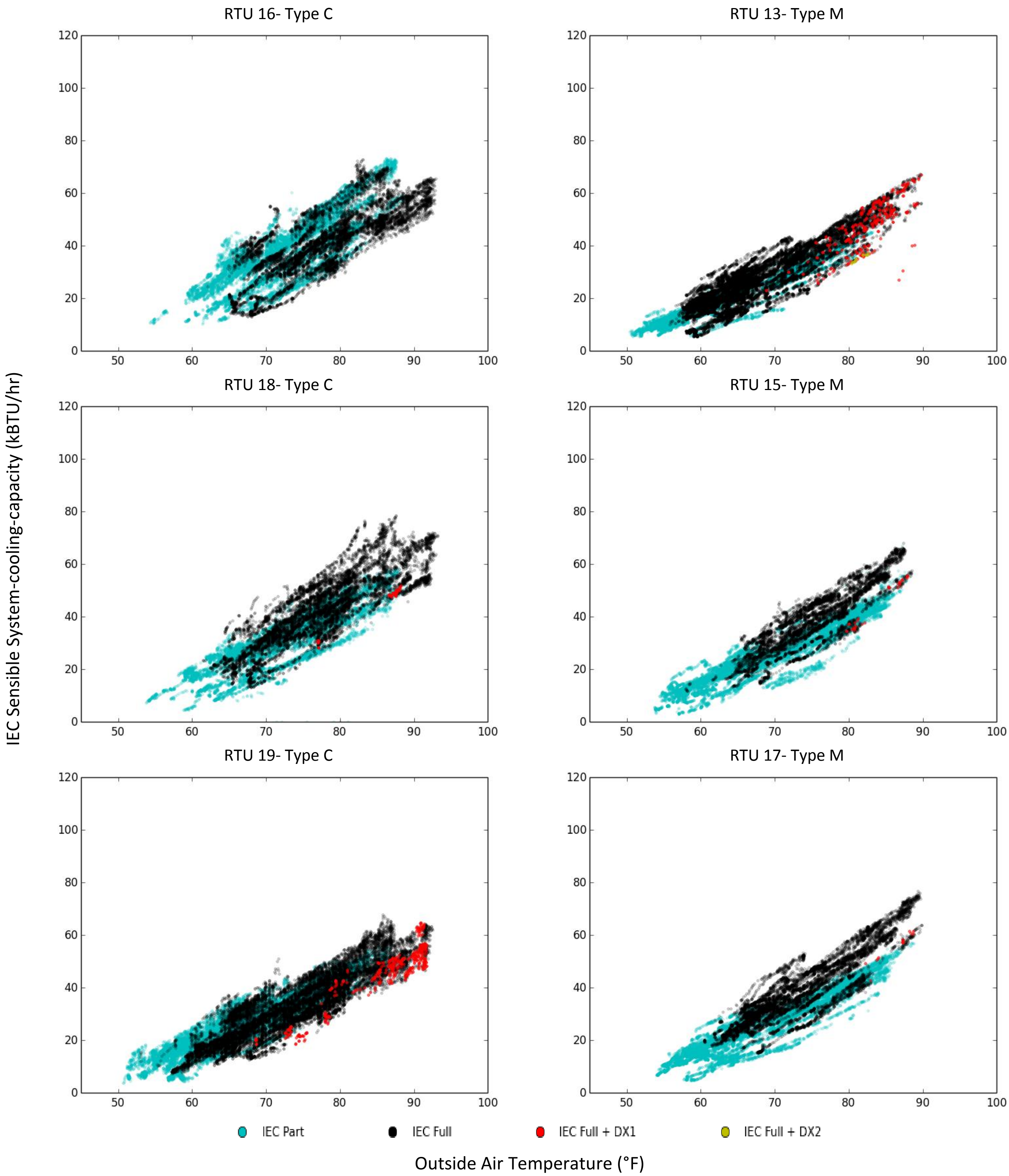

FIGURE 11. SENSIBLE IEC SYSTEM-COOLING-CAPACITY VERSUS OUTSIDE AIR TEMPERATURE

Room-Cooling-Capacity and Coefficient of Performance

One should note that even while system-cooling-capacity increases with increasing outdoor air temperature, the potential for cooling to serve sensible room loads decreases as outdoor air temperature increases. To look at this fact from a different perspective, the indirect evaporative product air temperature is most closely coupled with the outdoor wet-bulb temperature, and so increases as ambient dry bulb temperature and ambient wet-bulb temperature increase. This increase in product air temperature means that the system has less potential for sensible room-cooling at high outdoor temperatures. This, of course, coincides with the times that room-cooling loads are most significant.

For the meteorological conditions studied in this pilot the indirect evaporative air conditioners are always capable of supplying product air that is cooler than the room set point. Therefore the systems are able to cover the entire ventilation cooling load, and also always provide some benefit for room-cooling. For this application, the room-cooling potential is almost always enough to cover room-cooling loads in the zones served by the equipment, but compressor cooling is required for a small amount of time. This compressor cooling is restricted to periods of higher ambient temperature when room loads are high and room-cooling capacity is diminished. Although the effect was not measured in this pilot, because the equipment was installed to provide ventilation to the store by displacement, the room-cooling capacity delivered by these indirect evaporative systems should also reduce compressor runtime for other equipment in adjacent zones and throughout the store.

Indirect evaporative cooling can provide the largest energy savings impact when applied for cooling ventilation air. However, for periods when these systems can provide useful room-cooling, they will do so with far less energy consumption than a conventional air conditioner. Even in periods when additional compressor cooling is required, the capacity served by the indirect evaporative system comes with far less energy expense. The sensible room-cooling coefficient of performance, presented in Figure 12 describes the efficiency of each system using the measured room-cooling capacity as the frame of reference. This metric describes the performance of the system in an application that replaces a recirculation only air conditioner where no ventilation is required. The metric discounts the amount of cooling that is applied toward bringing outside air down to room air conditions, and counts only the capacity that has a sensible impact on the room temperature. From this perspective, even at 95°F, these systems achieve COP>5 for sensible room-cooling. A conventional rooftop unit would achieve COP<2 for a similar operating scenario.

Similar to the sensible-system-cooling results, the Type M and Type C equipment mark very similar capacity and efficiency trends for sensible-room-cooling at full speed. At part speed (77% product air-flow) , the Type M system gains an efficiency advantage. Interestingly, since the wet-bulb-effectiveness for the Type C device climbs at part air-flow, the cooling capacity does not decrease in proportion to the reduced air-flow. In fact, since the product air temperature declines rapidly with part speed operation, the Type C system appears to maintain a relatively steady room cooling capacity. Figure 12 and Figure 13 indicate that the coefficient of performance for sensible-room-cooling for these systems can be more than doubled at part speed with only a modest decrease in room-cooling capacity.

These results reinforce the fact that while the Type C technology does achieve a higher evaporative effectiveness, and therefore lower product air temperature, this appears to come at the expense of higher fan power, and results in full speed sensible-room-cooling efficiency that is very similar to the Type M equipment. The tracer gas air-flow measurements presented in Figure 24 indicate that full speed product air-flow for the Type C device is somewhat lower than for the Type M combination, even while fan power is approximately 19% higher.

One key takeaway from Figure 12 and Figure 13 is that while covering all ventilation requirements for the store, each of the six indirect evaporative systems also contributes significantly to cooling the space. At 95°F outside temperature each system contributes approximately two tons of sensible cooling to the space; at 80°F the room-cooling contribution from each increases to roughly 3.5 tons. The primary role of the systems may be to cover ventilation cooling requirements, but it is clear that they make a substantial contribution to addressing room-cooling loads as well, and do so for much lower power draw than would be required for conventional recirculation only vapor-compression cooling.

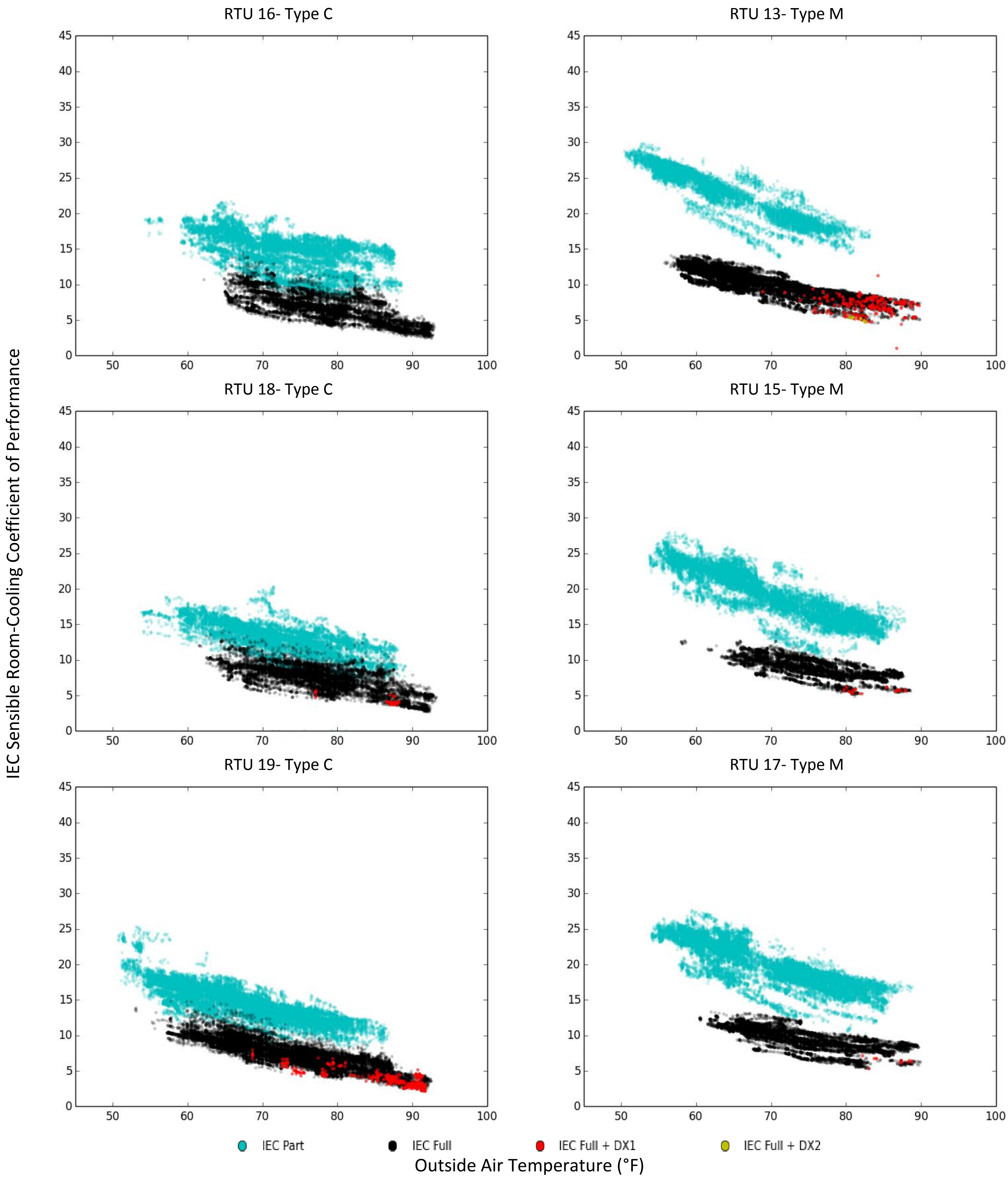

FIGURE 12. IEC SENSIBLE ROOM-COOLING COEFFICIENT OF PERFORMANCE VERSUS OUTSIDE AIR TEMPERATURE

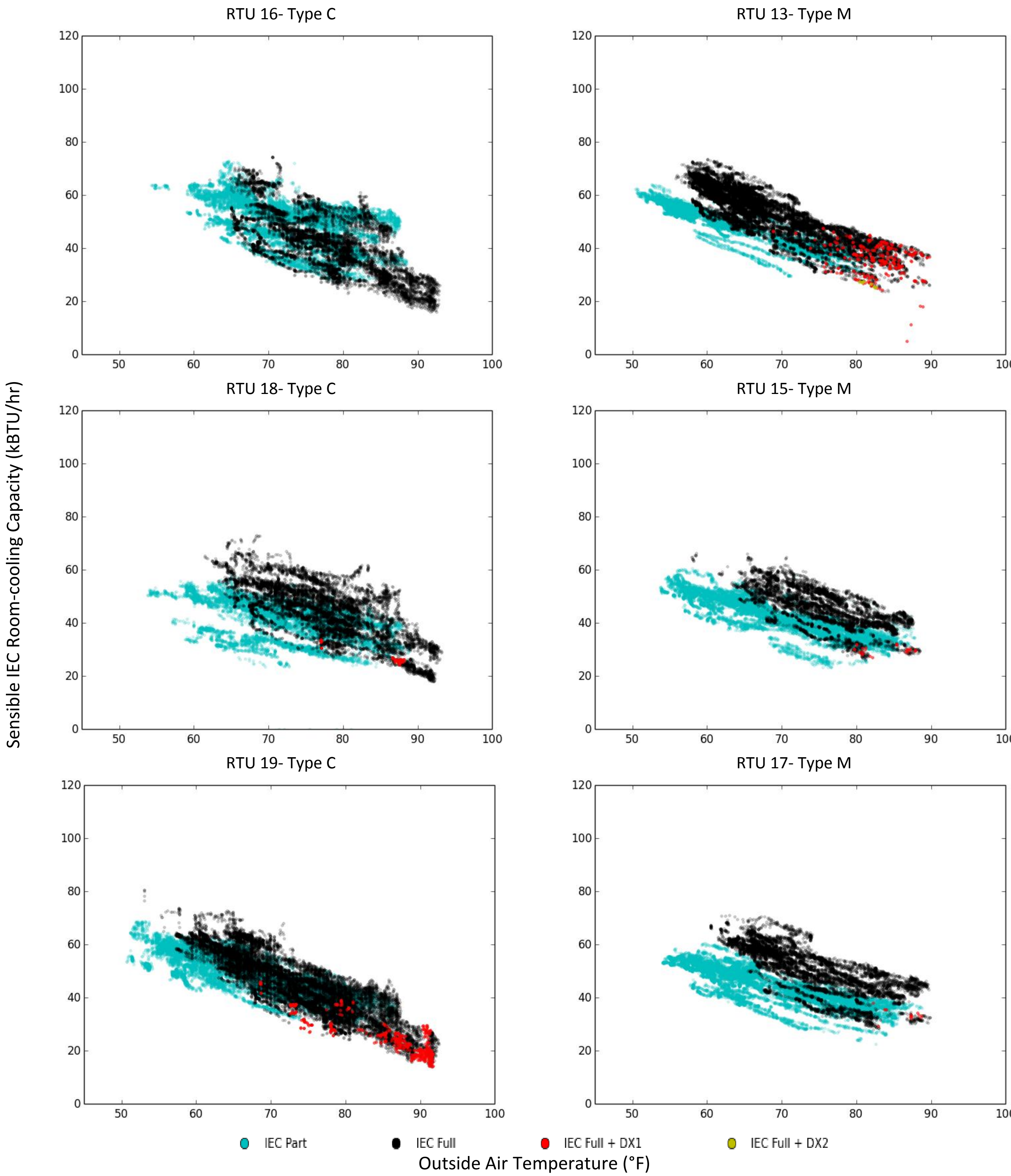

FIGURE 13. SENSIBLE IEC SYSTEM ROOM-COOLING CAPACITY VERSUS OUTSIDE AIR TEMPERATURE

Sensible and Latent Interactions with the Rooftop Unit

Figure 14 charts humidity ratio in the product air and supply air steam for each combined system as a function of the coincident outdoor air humidity ratio. Indirect evaporative cooling should not change the absolute humidity ratio as air passes across the primary side of the heat exchanger, this analysis reinforces that fact. In theory, all of the points representing product air humidity in Figure 14 should land on a 1:1 line where outside air humidity ratio and product air humidity ratio are equal. While the data shows some small deviation from this relationship, most of the observations are well within tolerances for the instruments and measurements. There does appear to be a very slight moisture addition from all units (the product conditions consistently measure slightly higher than outside conditions on all units), and it is reasonable to expect a very small amount of crossover (as is typical for heat recovery ventilators), but any effect is too small to measure with statistical significance.

Figure 14 does show several 'whiskers' of data where supply air humidity and outside air humidity differ significantly. Sometimes supply air humidity is higher than outside air humidity, while at other times it is lower. These instances only occur for measurement in the supply air, and can be attributed to the vapor-compression evaporator coil. When compressors operate the unit provides dehumidification, and when compressors shut off condensate retained on the evaporator coil evaporates, increasing the supply air humidity ratio for a period of time.

Figure **15** charts the relationship between product air temperature at the outlet of the indirect evaporative equipment, and supply air temperature at the outlet of the rooftop unit. Ideally, when compressors are off these two measurements would be equal. However, this is not the case in reality. Field measurements indicate that temperature effects across the rooftop unit can be significant.

When compressors are off, flow through the rooftop unit experiences a consistent temperature rise of 1-5 °F. We expect that this rise is due to thermal losses and leakage across the envelope of the rooftop unit, in addition to heat addition from the supply blower motor. Rooftop air conditioners are typically not especially well sealed, particularly at access panels and damper sets. Since the supply blower operates whilst in indirect evaporative mode, most sections of the unit operate at negative pressure, and we expect that a significant amount of outside air leaks into the rooftop unit. In fact, diagnostic air-flow measurements indicate that the supply air-flow rate in indirect evaporative only mode is 100-200 *cfm* higher than the product air-flow. The 1-5 °F rise does not seem to have a significant dependence on outside air conditions. This appears to be partly because the temperature effect of infiltration is damped by the fact that the product air temperature scales with outside air temperature, and partly because blower heat is responsible for a good portion of the rise. For the indirect evaporative only modes, this blower operates at an idling speed and consumes approximately 0.3 *kW*.

The same 'whiskers' that appear in Figure 14 are apparent in Figure **15**. For periods of time well after the compressor shuts off, the supply air temperature is remains several degrees below the indirect evaporative product air temperature. This effect can be attributed to evaporation of condensate retained on the evaporator coil. This re-evaporation of condensate is a well-documented phenomenon that must be carefully accounted for in simulation of air conditioners since it can have significant effect on the predicted capacity for a system when cycling.

The temperature rise presented in Figure **15** is undesirable. Such a significant thermal gain substantially reduces cooling capacity for the indirect evaporative system. It is important to note that this thermal gain would exist whether or not the indirect evaporative equipment were in place, but it is still unfortunate that the indirect evaporative system must contend with such a loss. We observe that the rooftop unit fan could probably be shut off when the vapor-compression cooling is not required. For the application in this trial, the indirect evaporative equipment would have easily overcome the air-flow resistance presented by the rooftop units. The research team commissioned fan speed and damper positions to maintain appropriate static pressure for each indirect evaporative system. The blower speed set point signal was brought all the way to zero, but the rooftop fan VFD was not configured to output zero *Hz*. Therefore, the blower was left to operate at a minimum speed. For future projects, we recommend disengaging the rooftop unit fan for operation without compressors. This would eliminate unneeded thermal gain from blower heat and unneeded power draw.

Further, for future applications we recommend avoiding the losses and thermal gains associated with passing product air through an existing rooftop unit. Wherever possible, it should be more effective to supply product air directly to the sales floor through a dedicated distribution system. This could be achieved by installing indirect evaporative equipment as a unitary DOAS system designed specifically to serve all of the ventilation needs for a building throughout every season. This approach would also simplify physical systems integration and controls.

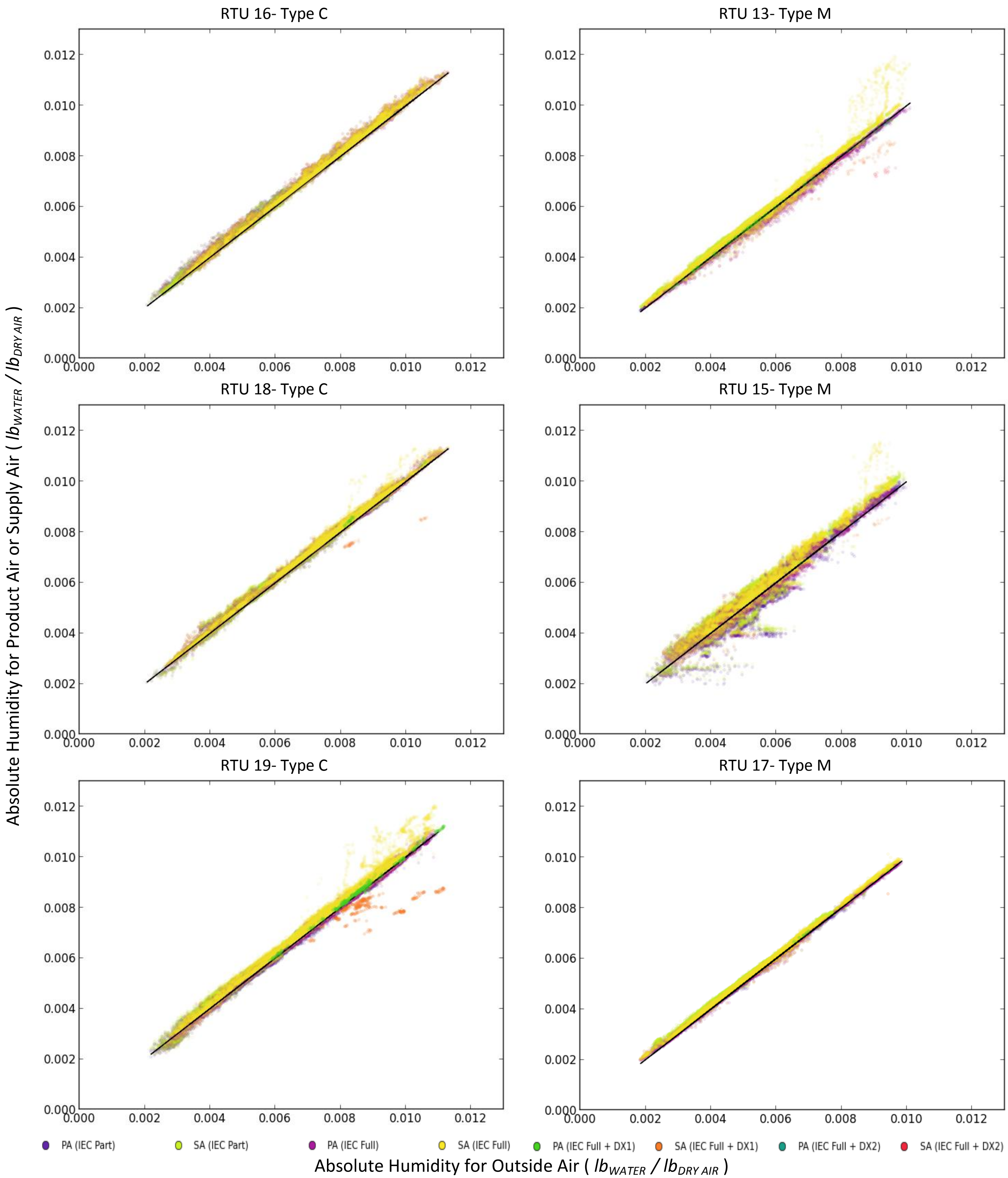

FIGURE 14. RELATIONSHIP BETWEEN PRODUCT AIR ABSOLUTE HUMIDITY AND OUTSIDE AIR ABSOLUTE HUMIDITY

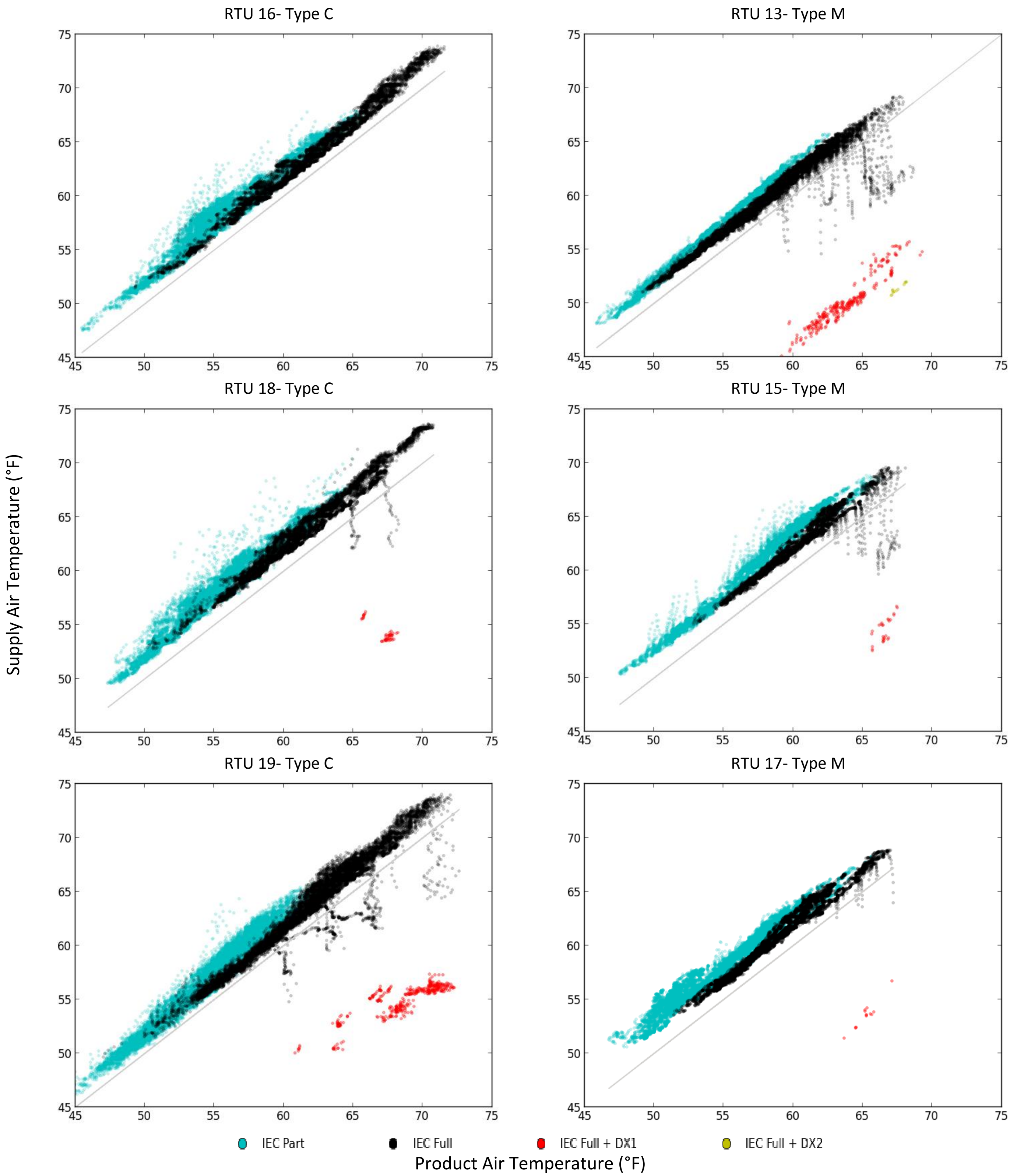

FIGURE 15. RELATIONSHIP BETWEEN PRODUCT AIR TEMPERATURE AND SUPPLY AIR TEMPERATURE BY OPERATING MODE

WET-BULB EFFECTIVENESS

Wet-bulb effectiveness is probably the most common metric for describing performance of evaporative systems. The metric describes the extent to which a machine is able to cool air toward the wet-bulb condition at the inlet of the system. Traditionally, product temperature from evaporative systems has been fundamentally limited by wet-bulb, however indirect evaporative air conditioners utilize more sophisticated heat transfer processes. Under certain circumstances some indirect evaporative air conditioners, including the products tested here, can generate product air temperature that is cooler than the wet-bulb at the inlet of the machine.

Figure 16 charts wet-bulb effectiveness for each machine as a function of the outside air wet-bulb depression. At full speed, the Type M system generally achieves wet-bulb effectiveness of 0.7 – 0.9, with a strong correlation to wet-bulb depression. Under similar conditions, the Type C system tends to mark wet-bulb effectiveness of 0.8 – 1.0. The variation in wet-bulb effectiveness at any particular condition appears to be larger for the Type C, a phenomenon we trace to the cyclic air-flow adjustments and product temperature fluctuations associated with the media wetting and drying cycle for this system. This fact was discussed previously and is explored further in the section *Transient System Dynamics.*

At part speed operation, which corresponds to 77% air-flow rate, the Type C marks a significant increase in wet-bulb effectiveness. For some periods, effectiveness exceeds 1.15. Wet-bulb effectiveness for the Type M equipment is also sensitive to air-flow rate, though the change is much smaller. Generally, measurements indicate that effectiveness for the Type M only increases by an average of 1.5% when the unit is shifted to 77% air-flow rate. Figure 16 also charts the wet-bulb effectiveness for each indirect evaporative system for periods when the rooftop unit operates in compressor cooling. These records shows that performance does not suffer when the rooftop unit blower is enabled, indicating that controls are able to adjust damper positions to maintain static pressure in the product plenum.

It should be noted that the Type C equipment on RTU 19 consistently showed poorer wet-bulb effectiveness than the two other Type C systems. Diagnostic investigation led to the conclusion that control systems developed for this evaluation were not maintaining proper static pressure in the product air plenum. This appears to have resulted from the fact that the external controls for all three Type C systems were programmed with equal damper set points and fan speed indexes; this allowed small physical differences between the three systems to result in different operating pressures within the product air plenum for each unit. RTU 19 was observed to operate with 0.45*"WC* static-to-ambient, instead of the manufacturer recommendation of 0.6*"WC*. Ostensibly, this decrease in resistance resulted in an increase for product air-flow, a shift in the primary-to-secondary air-flow balance, and an increase in product temperature.

These measured results demonstrate the fact that performance for these systems is sensitive to operating static pressure. In instances where the downstream resistance does not change, if needed, a fixed damper can be commissioned at installation to provide for the appropriate operating pressure. In applications where the downstream resistance changes, such as in this demonstration, it is important to actively manage pressure, and especially to avoid exposure to negative pressure. The open-loop indexed control scheme we applied resulted in less-than-optimal system performance. We recommend that future applications could use active pressure sensing in a simple closed-loop control sequence to adjust dampers or fans to always maintain operating pressures within design conditions.

Although the installation reported here appears to have managed product plenum pressure adequately for all most of the systems, it is still important to note that controls should carefully maintain operation within the intended pressure envelope. An increase in downstream resistance will generally increase wet-bulb effectiveness, but simultaneously reduce product air-flow, cooling capacity, and efficiency. Exposure to negative pressure, by supplying product air-flow directly to the negative pressure inlet of an air handler, can result in significant imbalance between the primary and secondary air-flows.

Most building energy simulation tools use a fixed effectiveness as the key term to describe performance for all types of evaporative equipment. However, accurate modeling for indirect evaporative air conditioners requires a more sophisticated methodology that can describe performance as a function of air-flow rate, and inlet conditions. Figure 16 indicates that effectiveness for the Type M system increases as wet-bulb depression increases. For both systems effectiveness declines sharply when wet-bulb depression is small. For the Type C system, it appears that effectiveness for full speed operation may peak between 15-25°F wet-bulb depression, then decrease somewhat when wet-bulb depression is high. Effectiveness changes with fan speed for both systems, though it appears that this effect is more pronounced for the Type C systems in this application.

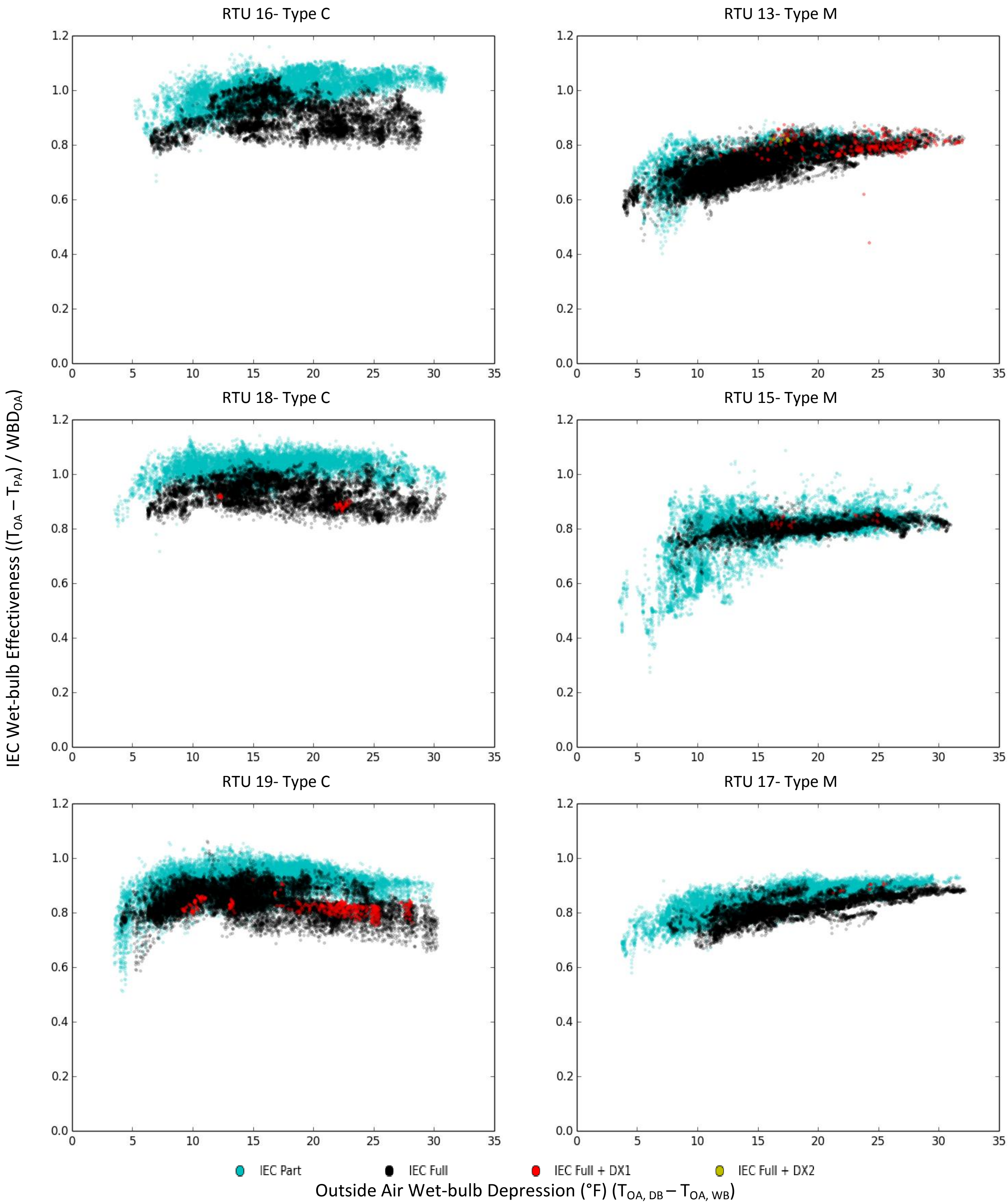

FIGURE 16. WET-BULB EFFECTIVENESS AS A FUNCTION OF OUTSIDE AIR WET-BULB DEPRESSION.

System Temperatures

Figure 17 charts product air temperature for each system as a function of outside air temperature. Full speed operation is charted separate from results for part speed operation. In order to eliminate variation associated with transients, the data charted is only for steady state periods where the equipment has been running consistently in a single mode for at least ten minutes. These results show that product air temperature for a particular air-flow rate is most sensitive to outside air dry bulb temperature, though variation in the trend indicates that it is also effected by humidity. For any particular outside air temperature observations indicate that product air temperature varies by as much as 10°F. This variation is attributed mostly to the differences in ambient humidity ratio, which ranges from 0.0025 – 0.011 during the periods of observation.

At part speed, when ambient temperature is between 70–80°F, these systems generate product air temperature between 50–60°F. At 85°F part speed operation achieves temperature difference as large as 25°F, similar to the results for wet-bulb effectiveness. At 90°F, full speed operation generally supplies product air in the 60°F – 70°F range, depending on humidity. The product air temperature rises as outside air temperature rises, though the temperature difference across the heat exchanger also increases. The trends indicates that at 95°F, full speed operation would achieve 25°F cooling effect. These results clarify the reasoning that indirect evaporative cooling is usually applied in combination with conventional cooling strategies for commercial buildings. For the air-flow rates in typical ducted cooling systems, supply air temperature near 70°F will not provide adequate cooling at peak.

For most climate zones in California, it appears that the equipment evaluated here will always provide some room-cooling capacity (relative to 78°F return condition). Although the room-cooling-capacity is limited at peak, it should usually make sense from an energy efficiency perspective to take as much cooling as possible from the indirect evaporative system, operating at full speed and providing supplementary room-cooling with a vapor-compression system. For buildings with cooler return air temperatures, in more humid climates, and for extremely hot conditions, there will be a point beyond which product air temperature is not cooler than room conditions. In this case, indirect evaporative cooling would still provide substantial ventilation-cooling-capacity, but the overall system efficiency would benefit from returning to part speed operation in order to supply only the minimum ventilation rate.

Figure 18 charts the relationship between the inlet air condition, product air condition, and exhaust air condition for all six units studied. The entire field of measurements for each point is plotted on a psychrometric chart for each unit and three specific instances are pulled from those clouds of data and highlighted. The conditions chosen for this comparison are 70°F | 30% RH, 85°F | 15% RH, and 80°F | 40% RH.

The results in Figure 18 clearly illustrate that the product air is cooled sensibly, with no measureable addition of moisture. The sensible cooling of primary air scales mostly with wet-bulb depression, where hotter and drier conditions achieve a larger cooling capacity. At the same time, the absolute product air temperature increases as ambient temperature increases.

The secondary air stream is exhausted with a wet-bulb condition that is higher than at the inlet. The air in an evaporative cooling process is usually thought to progress along a wet-bulb line, but the secondary air stream in an indirect evaporative cooling process does not behave this way because the process is not adiabatic. In addition to translating thermal energy from sensible to latent through evaporation, sensible heat is also transferred from the primary air stream to the secondary air stream. In order for the primary air stream to cool sensibly without moisture addition, the secondary air stream must exit with a higher specific enthalpy than the inlet. In fact, conservation of energy requires that the mass weighted enthalpy difference between outside air and product air must be exactly equal to the mass-weighted enthalpy difference between outside air and exhaust air.

A final important observation from Figure 18 is that the climate conditions experienced during the period of study for the Type C equipment is somewhat different from the conditions observed during the period of study for Type M system. The periods of study overlap, but the data for Type C includes observations from nine extra days in early September, while the Type M data includes observations from ten extra days in late October. For any given outside air temperature, the Type M dataset is slightly less humid on average. Accordingly, the product air temperature results, cooling capacity and efficiency results, and water use results documented throughout this report present Type M performance for conditions that are somewhat drier on average than those reported for the Type C system. The differences should not have a major impact on the general observations presented here, but efforts to extract performance characteristics from the data should carefully account for performance sensitivity to all environmental conditions.

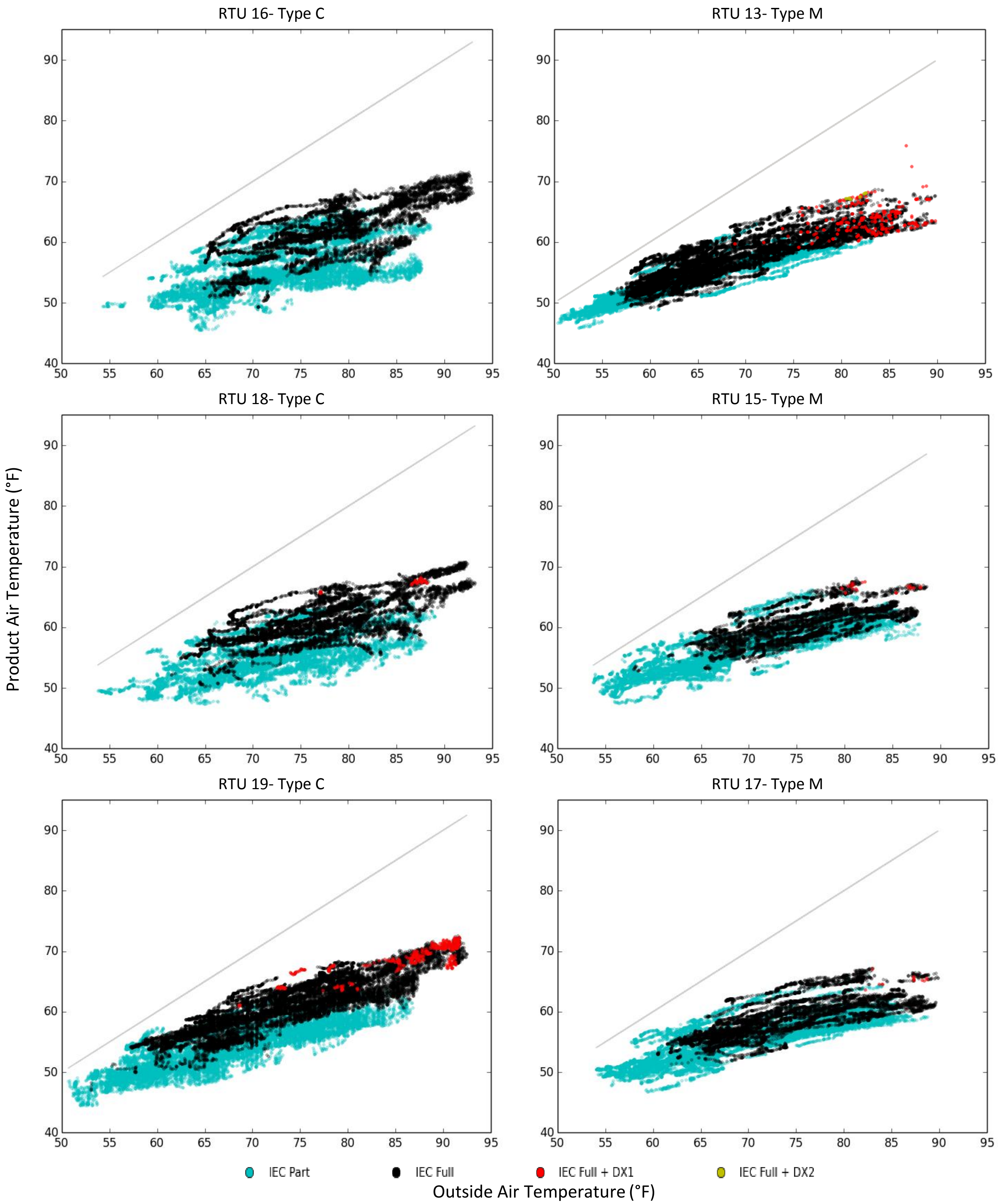

FIGURE 17: PRODUCT AIR TEMPERATURE AS A FUNCTION OF OUTSIDE AIR TEMPERATURE

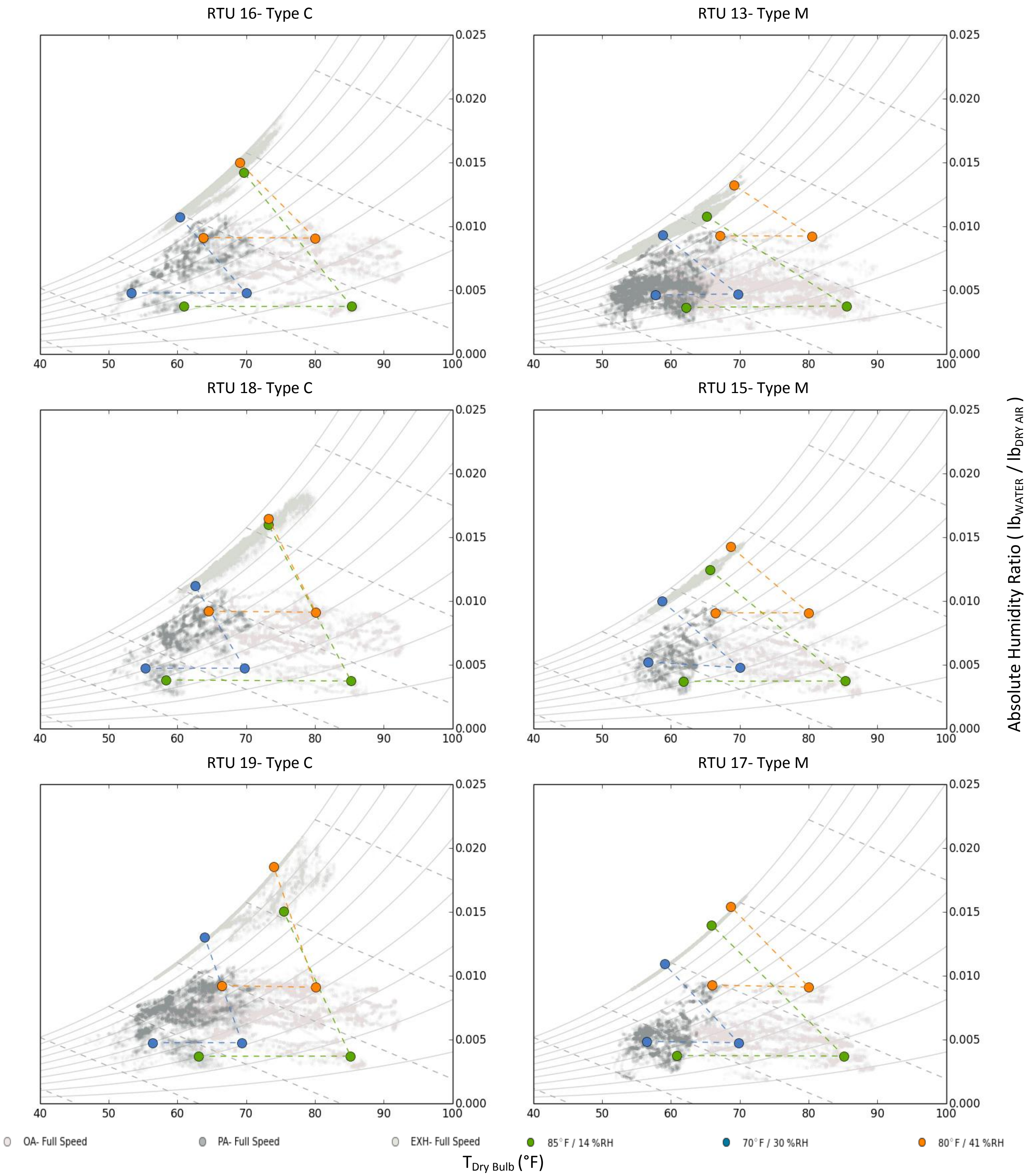

FIGURE 18: PSYCHROMETRIC PERFORMANCE FOR EACH SYSTEM AT THREE OUTSIDE AIR CONDITIONS

Cumulative Sensible Cooling

Figure 19 charts the cumulative sum of sensible-system-cooling delivered by each indirect evaporative system in each mode of operation for every day during the period of test. These results represent the total amount of cooling generated by each system. For operation in "*IEC Full + DX*" modes, these charts only mark the sum of sensible-cooling generated by the indirect evaporative component, and do not include the supplemental cooling delivered by the vapor-compression systems. The period of observation for the Type C system began earlier than for the Type M equipment. Figure 19 charts results from the full dataset for each unit, and highlights the time period that overlaps.

The daily sum of cooling energy for each unit trends roughly with the daily peak outside air temperature. For the period represented here, vapor-compression is only needed for a small portion of time. Some units call for more vapor-compression cooling than others, though this cannot be attributed to the performance characteristic of either technology since the cooling loads for each zone in a big box store can vary significantly. For example, on cooler days when part speed cooling from the Type M system on RTU 15 is adequate to serve the daily room-cooling requirements for the zone, the Type M on RTU 13 runs mostly at full speed, generates more than twice as much cooling over the course of the day and even requires a fair amount of vapor-compression cooling.

These results indicate that for most zones indirect evaporative cooling is capable of maintaining comfort with very little compressor operation even on days as warm as 90–95°F. These systems are also providing for the ventilation requirements of the entire sales floor. Not only are the observed zones carried mainly by indirect evaporative cooling, compressor operation for other rooftop units on the store should also be reduced since those systems do not have to cover ventilation loads, and since some of the sensible-room-cooling delivered by the indirect evaporative equipment is transferred across the sales floor to other zones according to a displacement ventilation design.

Figure 20 charts the distribution of cumulative sensible-system-cooling for the indirect evaporative equipment, binned by outside air temperature. Figure 21 charts the same distribution for cumulative sensible-room-cooling. These results show that cooling below 85°F is dominated by indirect evaporative only, compressor cooling for these zones is held off until ambient temperatures are above 90°F.

Note that for RTU 13, the largest portion of cooling delivered by the indirect evaporative system is in the 65°F bin. As discussed previously, this zone appears to have much higher internal loads, or is held to a lower set point temperature. While the five other units require only 500 – 2,000 *kBtu* sensible-room-cooling in the 65°F bin over the period of study, RTU 13 delivers more than 8,500 *kBtu* room-cooling. It is important to note that while the control system designed does allow for economizer cooling in parallel with the indirect evaporative systems, the building EMCS never requested economizer operation for the store. Insofar as we can tell, this is a failure of the EMCS for the store. At 65°F it is very likely that an appropriately controlled economizer cooling mode would have played a substantial role for RTU 13.

Below 75°F the cumulative sensible-room-cooling for each unit exceeds the cumulative sensible-system-cooling; this occurs because outside air is already cooler than return air. The trend is reversed above 75°F since only a portion of the system-cooling effect has an impact as room-cooling-capacity. The decrease in cumulative cooling at higher outside air temperatures is related to the number of hours spent at these conditions; the relationship between instantaneous cooling capacity and outside air temperature are discussed previously.

Each unit exhibits a unique distribution of operating modes, related to the dynamics within the corresponding zone. Notwithstanding, these figures indicate that the majority of sensible-cooling energy from all units is delivered at part speed operation, where the indirect evaporative systems generate cooling with a COP as high as 25 (see Figure 10 and Figure 12). As should be expected, part speed operation serves a larger portion of the cooling needs at lower outside air temperatures, but still carries a substantial portion of the cooling requirements in the 85°F bin. Part speed operation plays a minor role for some units in the 90°F bin. Full speed indirect evaporative cooling plays a minor role at low temperatures, and dominates the cumulative cooling effects above 85°F. Vapor-compression cooling is needed for some systems above 85°F, and appears to play at least a small role for the cooling capacity from all systems above 95°F.

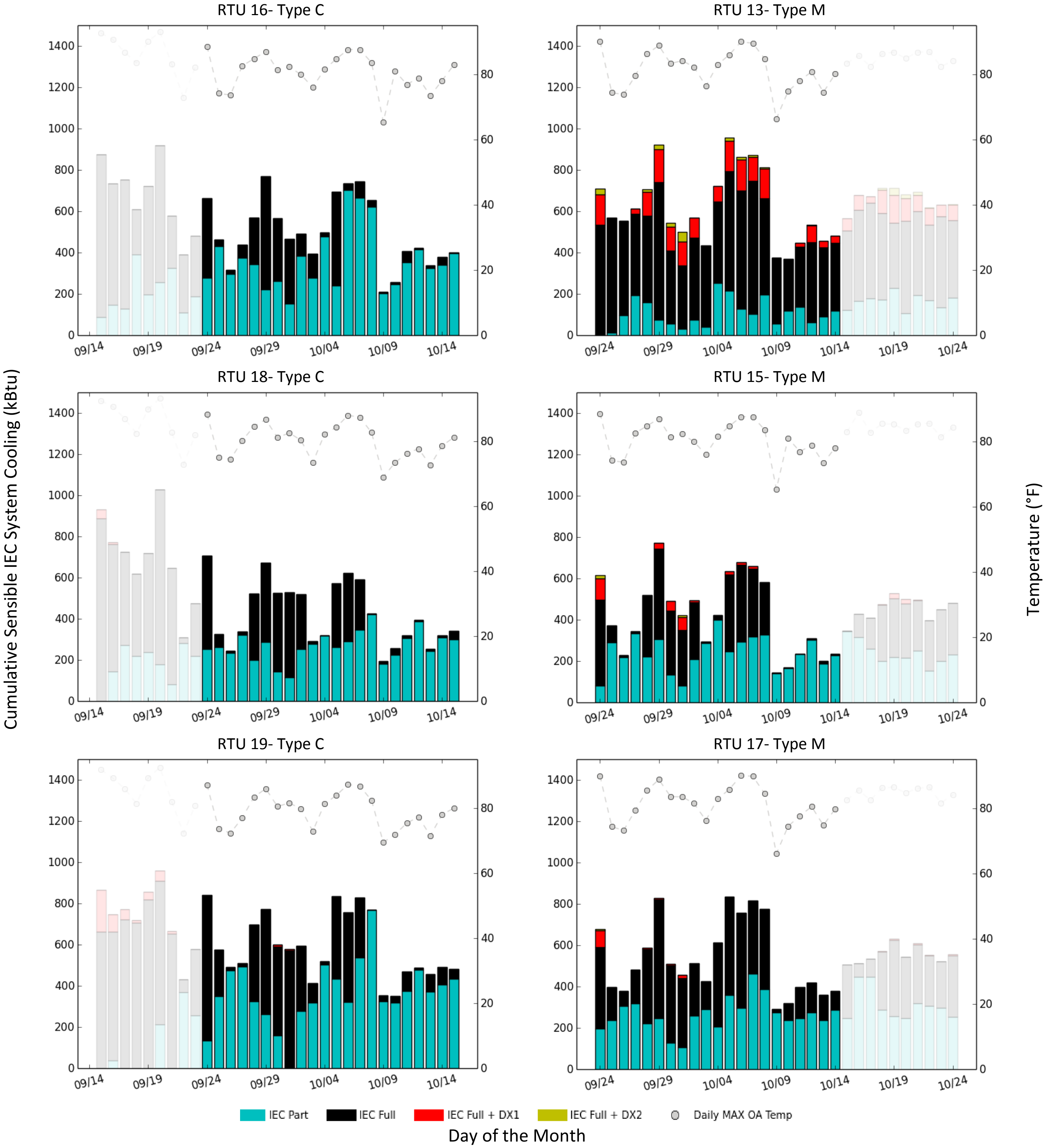

FIGURE 19. THE DAILY SUM OF SENSIBLE-SYSTEM-COOLING FROM INDIRECT EVAPORATIVE SYSTEMS IN EACH OPERATING MODE

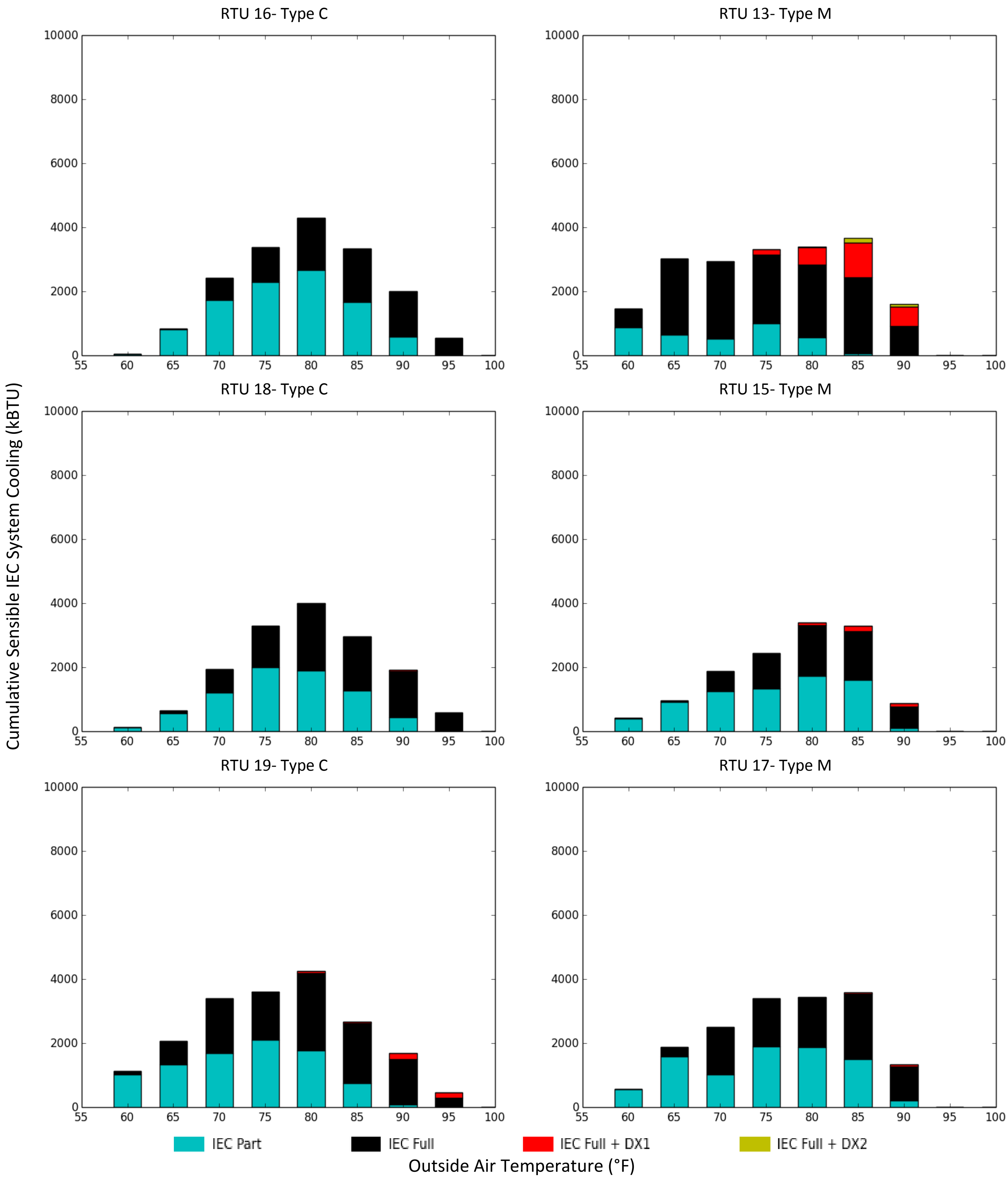

Figure 20. Cumulative sensible-system-cooling from indirect evaporative in each mode by outside temperature

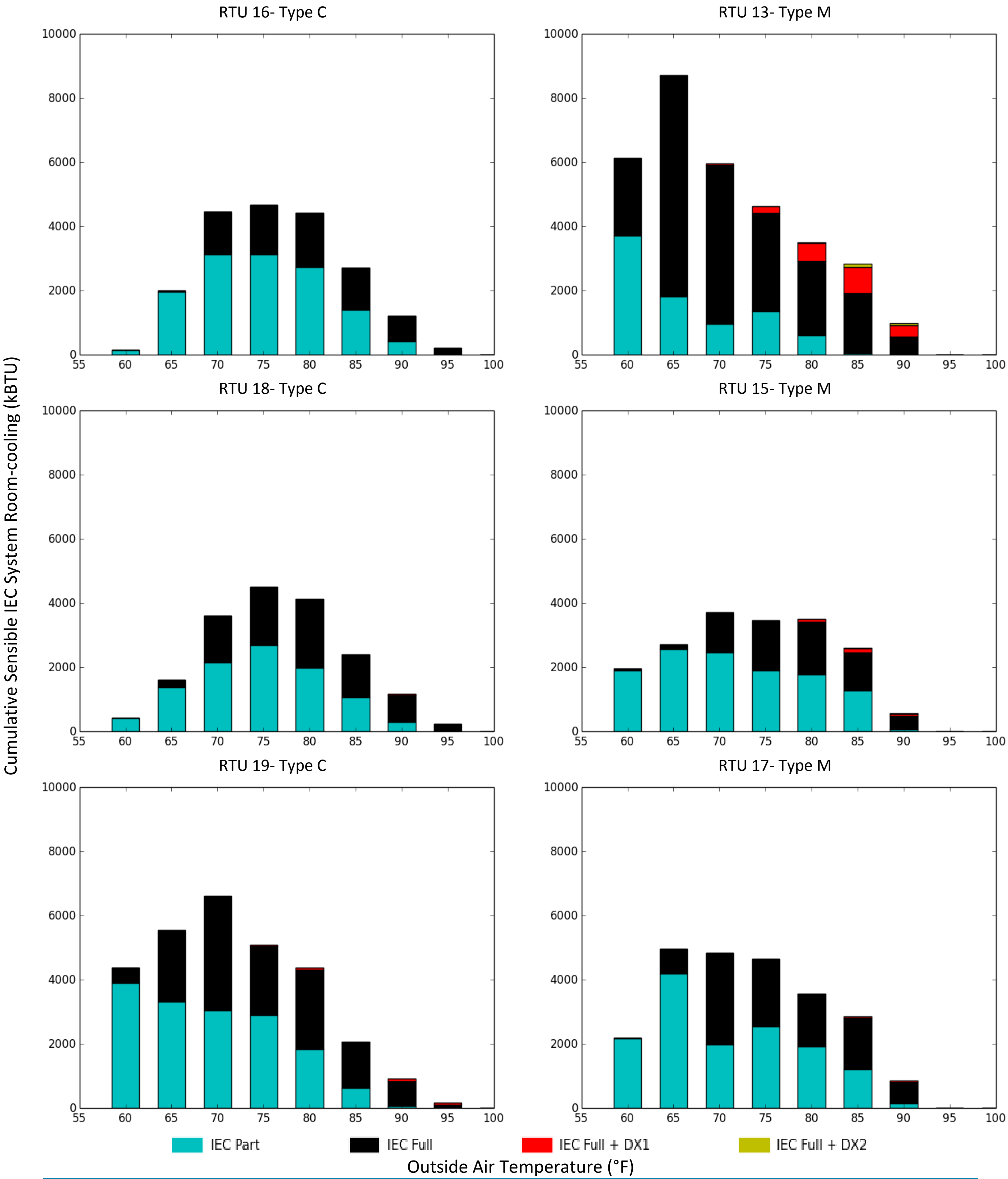

FIGURE 21. CUMULATIVE SENSIBLE-ROOM-COOLING FROM INDIRECT EVAPORATIVE IN EACH MODE BY OUTSIDE TEMPERATURE

WATER EFFICIENCY

The consumption of water supplied to each unit was measured for every minute. The volume of water drained in each minute was measured for the Type C equipment on RTU 16. The same measurement was intended for the Type M on RTU 15, but the meter was removed after it became plugged with debris which prevented the system from draining properly. Figure 22 charts the daily total water consumption for the Type C and Type M systems during the periods of monitoring in September-October 2013. The water drained from the Type C system is shown, but the water drained for Type M system was not recorded. The corresponding daily maximum outside air temperature is also plotted for comparison. RTU 16 and RTU 17 were selected for comparison, since they have very similar distribution of daily cooling loads and operating modes.

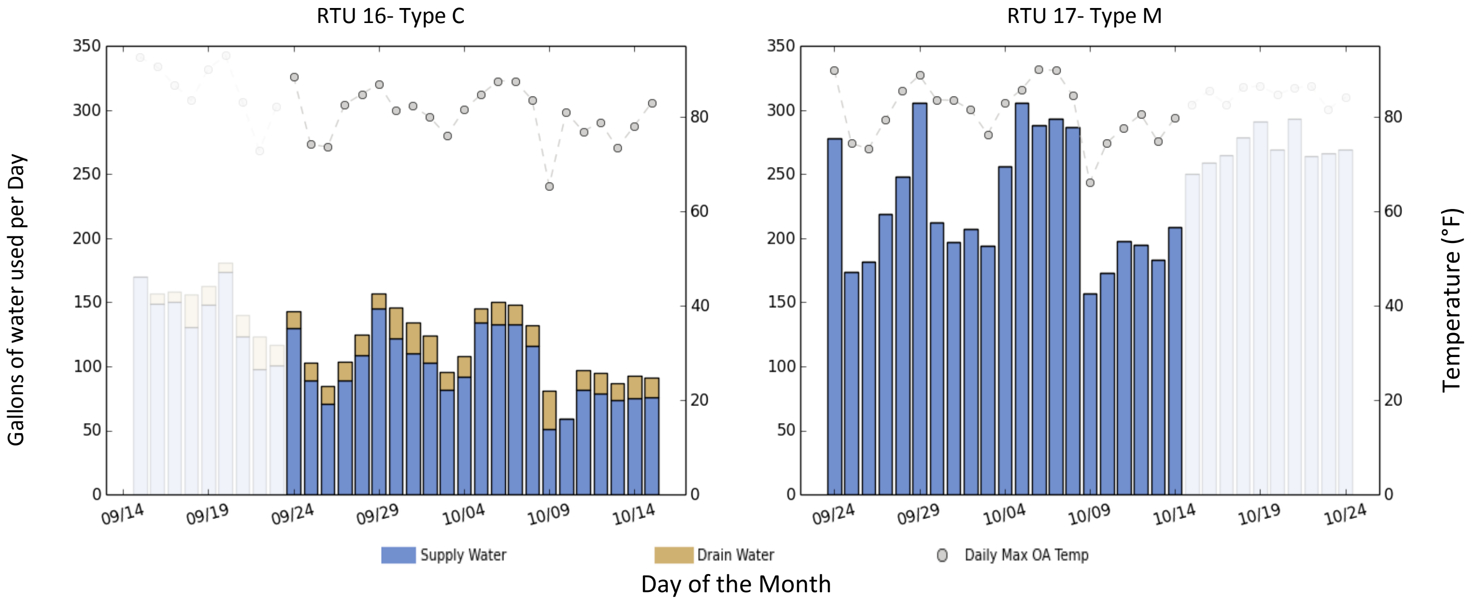

FIGURE 22. DAILY WATER CONSUMPTION WITH CORRESPONDING DAILY MAXIMUM OUTSIDE AIR TEMPERATURE

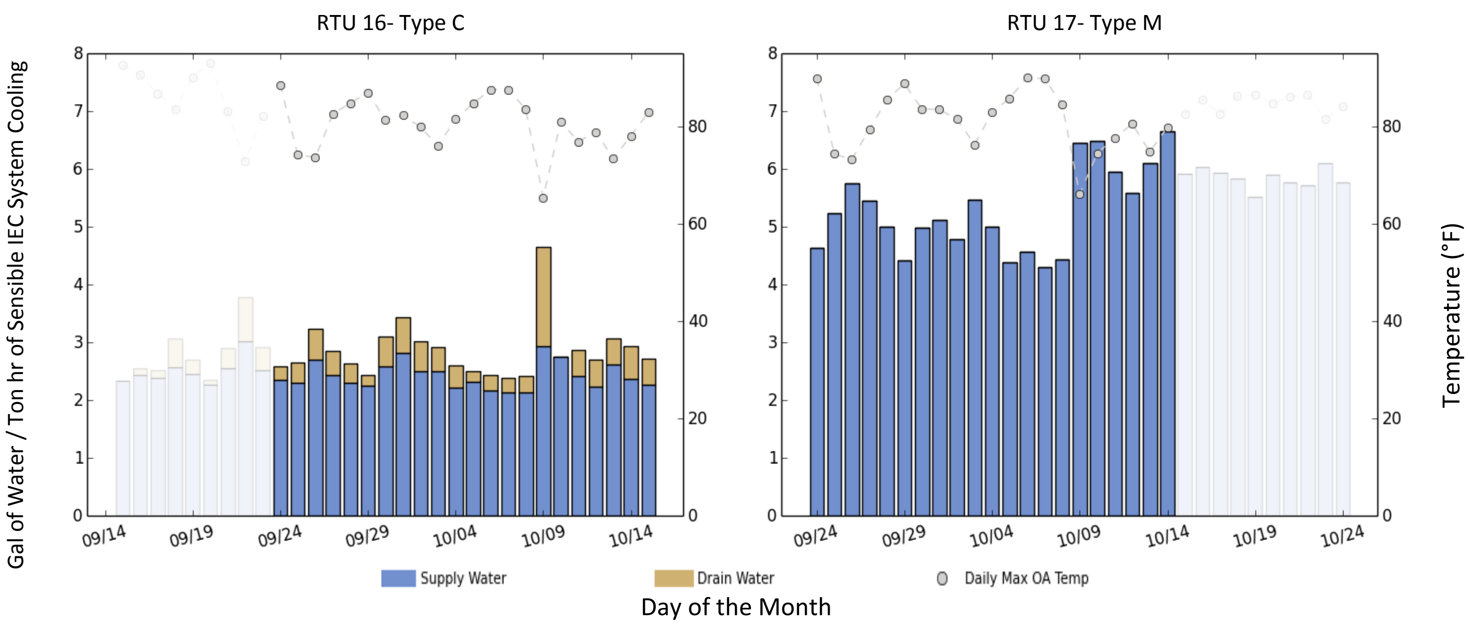

FIGURE 23. NORMALIZED DAILY WATER CONSUMPTION WITH CORRESPONDING DAILY MAXIMUM OUTSIDE AIR TEMPERATURE

As one should expect, the daily water consumption for both units corresponds coarsely to the daily maximum outside air temperature. Water consumption should also be a function of cooling load, air-flow rate and ambient humidity conditions, but outside temperature is the major driver. The Type C system consumes 75–175 gallons per day over the test period. Less than 15% of that water is drained, the balance is evaporated. The Type M system uses between 175–300 gallons per day during the same period. The field study did not yield measurements for what fraction of this water is evaporated and what fraction is drained, however some estimates can be made according to the enthalpy of phase change required to generate the measured sensible cooling capacity during the corresponding period observed. It is estimated that the Type M system consumed 30-40% more water than what was evaporated.

The Type M and Type C systems use distinctly different water management strategies in order to maintain heat exchanger wetting, and to avoid deposition and accumulation of solids. The Type C equipment uses multiple small pumps to spray water over the top of the heat exchanger which drains by gravity back to a sump. Supply water to the sump is maintained with a level switch and solenoid, and water is drained from the sump when the system controller senses high conductivity in the water. The threshold for conductivity, and the intermittent drain cycle for the unit are field selectable. Type M equipment uses a predictive strategy for water management. There is no sump, and no pump. Water is supplied to the unit through a solenoid valve at a rate that will provide water for evaporation, plus ensure enough additional flow to avoid deposition of solids. The anticipated rate of evaporation is predicted according to measurement of ambient psychrometric conditions, and the amount of additional water is selected at startup in order to suit the water quality observed on site.

Since the cooling capacity and energy efficiency for the Type M and Type C systems are very similar, we believe that the difference in on-site daily water consumption can be mainly attributed to the water management strategies for each system.

The amount of daily water consumption for this equipment may seem large (750 – 1,425 gallons per day for the entire site) however, it is important to consider the broader implications of this use in order to assign some relevance to the result. There are three important aspects to consider:

1. The economic cost of water is small compared to the value of energy saved
2. The net impact to statewide water consumption can be positive when one considers the upstream water use associated with electrical generation.
3. The relative magnitude of water consumption associated with cooling is very small compared to other water end uses such as agriculture, landscape irrigation, manufacturing, and food production.

For the US average water tariff of .00272 *$/gal* (EPA, Community Water System Survey) we estimate the cost of cumulative site water consumption at $2.04 - $3.88 */day*. Given the energy savings from cooling operation at an average sensible system COP=10, our calculations indicate that water costs are very small in comparison the value of energy savings. The ratio of water costs to energy cost savings will depend on the baseline for comparison, and on local water rates, and water quality, but the take-back appears to be on the order of 1 – 10%. These estimates do not consider the substantial value of demand savings or other benefits associated with improved air quality, reduced greenhouse gas emissions, improved public health, and reduced environmental and social costs of energy extraction.

In order to consider the net impact on statewide water consumption, one must account for the water consumption intensity for electricity generation (*gal/kWh*), as well as the energy savings outcome of water use on site (*gal/kWh*). It also is helpful for this accounting to normalize water consumption by the amount of sensible cooling that was delivered as a result of the water use. Figure 23 charts the normalized daily water consumption for each unit as *gallons per sensible-ton-hour* of cooling delivered by the indirect evaporative equipment. The measurements range from 2.25 – 6.5 *gal/sens-ton-hr*. This is roughly consistent with the water use efficiency targets established by the Western Cooling Challenge, and align with estimates for what performance could result in regionally neutral water consumption from a statewide perspective. In fact, our estimates for net water consumption for this project indicate that the measure results in somewhere between 469 *gal/day* increase in water consumption to a 322 *gal/day* savings.

Review of literature indicates the average water consumption intensity for electricity generation in California is somewhere between 1.33 – 2.76 *gal/kWh*, including evaporative losses from hydroelectric generation. The average water use intensity for generation from thermal plants alone in California is estimated at 0.44 *gal/kWh*. (Torcellini 2003, Pistochini 2011). Depending on the input assumptions and system efficiency afforded by a water consuming cooling technology, our estimates for the equipment studied here indicate that the balance for regionally-neutral-water consumption should occur between 1 – 10 *gal/sens-ton-hour*.

IEC Product Airflow Results

Airflow plays an important role in measuring system cooling capacity and efficiency. The results presented here rely on tracer gas air-flow measurement of the product stream from each unit at part speed operation and at full speed operation. The tracer gas airflow measurement tool meters a precise mass flow rate of CO_2 into an air-flow stream, then measures the downstream CO_2 concentration and directly calculates the volume flow rate of the bulk air-flow into which the tracer gas was mixed. This method can achieve a calculated uncertainty of less than 2%.

Product air-flow rate may vary from minute to minute, and it is therefore important that interval data for temperature power draw are referenced against an appropriate air-flow for the corresponding minute. Changes in product air-flow may result from slow changes in system resistance over time (e.g.: associated with filter soiling) or from sudden and intermittent shifts in system operation (e.g.: associated with downstream pressure, especially for each mode of operation). The research team was especially cognizant of the fact that when operating at full speed, the Type C equipment will automatically decrease air-flow rate once every 9-min associated with the wetting cycle for the heat exchanger. The effect of this function is discussed further *Transient System Dynamics*.

In order to capture dynamic variations in air-flow rate, single instance flow measurements were used to map the relationship between fan power draw and product air-flow over a range of scenarios. In this way, for each minute the fan power measurement is used as a proxy for air-flow rate. It is best to map air-flow as a function of the differential static pressure measured across some fixed resistance (e.g.: fan inlet ring), or as a function of an airflow velocity measurement, but the measurements of these variables was not reliable for this project. As a result, the airflow maps used are only appropriate for the mode of operation, and the system resistance characteristics for which they were developed. Most importantly, the maps are only appropriate for periods when the equipment operates with clean filters, and cannot be applied accurately for the periods with soiled filters. All of the data presented in this report is from periods immediately after filters had been replaced.

Figure 24 charts the resulting box-and-whiskers distribution of airflow for each unit at part speed and at full speed during the September-October periods of evaluation presented in this report. These distributions are the product of single point tracer-gas airflow measurements, mapped against fan power, and propagated into the minute-by-minute data set according to the fan power measured at each interval. The limits of each box represent the first and third quartiles in the data, the whiskers represent 99.9% of the intervals – outliers are ignored. We find that the average product air-flow for each unit is very close to manufacturer specifications; for some units the flow appears to be is 100 *cfm* lower, and for others 50 *cfm* higher. The flow rate for each unit in full speed varies from the average by only 3–12%, and the average air-flow from unit to unit varies by less than 5%. Fan speed for the Type C does decrease for the heat exchanger wetting cycle, the air-flow typically decreases by approximately 350 *cfm* for less than one minute.

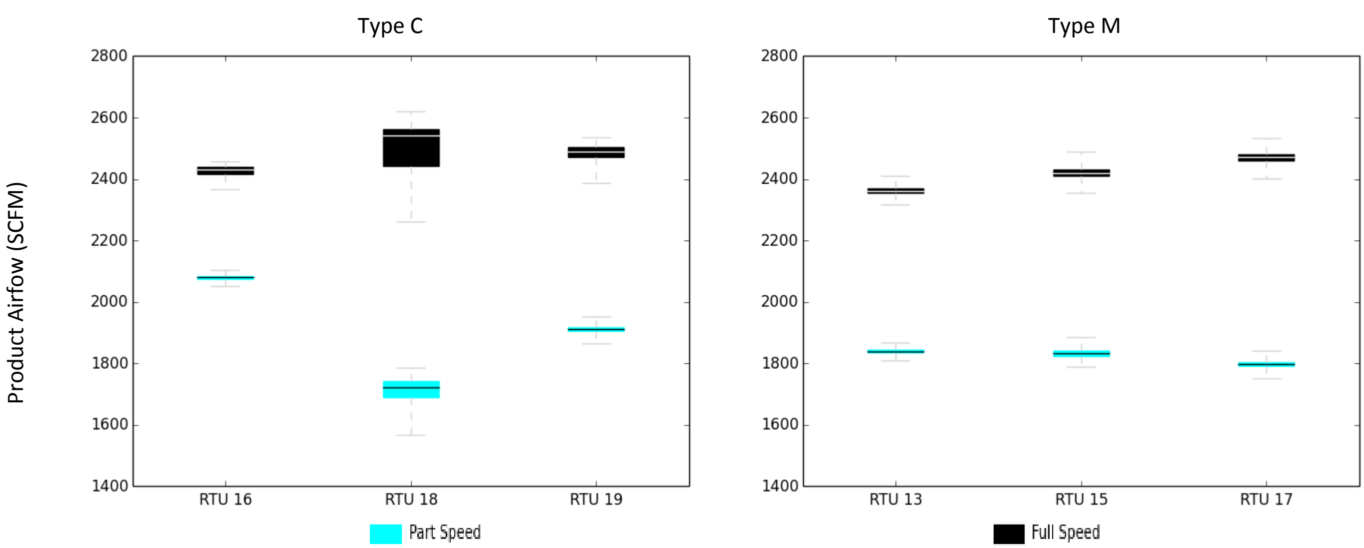

Figure 24: Box-and-whiskers distribution of product air-flow for each unit at full speed and part speed

OPERATING PRESSURES AND POWER DRAW

In August 2013, ongoing review of performance data from the site hinted that some units were not performing as well as expected. In particular, it was observed that when one unit shifted from "*IEC Full*" in "*IEC+DX1*" mode, the product air temperature would increase suddenly by nearly 5°F. If everything is functioning and controlled correctly, a switch to compressor operating mode should not have any impact on product temperature because dampers would adjust as the RTU blower speeds up to maintain appropriate operating pressure for the indirect evaporative cooler.

A review of several weeks of the minute-interval performance data showed that the pattern had developed slowly over the course a few weeks in early July, then ebbed over a few days in late August, and finally emerged again suddenly in early September. The only phenomenon that could cause such major a disturbance in product temperature at any particular inlet condition would be a pressure-air-flow imbalance for the heat exchanger. Initially we expected a hiccup in the controls algorithm, or a physical failure of one of the control dampers which could expose the indirect evaporative heat exchanger to negative pressure from the RTU blower. After several days of scrutiny we dispatched both manufacturers and the research team for a site visit to investigate.

On site we quickly identified that the air-flow imbalance was caused by soiled filters. The maintenance service provider for the store had changed filters for all other units, but not for the indirect evaporative equipment. Filters for two of the Type C systems had soiled so completely that suction pressure caused them to collapse. After collapsing, the air-flow path was free open, so monitored performance of these two systems showed no signs of failure. Filters for the unit that caught our attention had collapsed in such a way that the fan inlet was largely blocked, causing substantial air-flow resistance. The IEC fan could not maintain adequate air-flow, and when the RTU blower was enabled for "*IEC+DX1*" the product plenum was exposed to negative pressure, causing an imbalance in the primary-to-secondary air-flow ratio which resulted in the sharp rise for product temperature that was observed. Filters for the three Type M systems had also become soiled, and product air-flow was significantly reduced, but the effect did not cause an obvious primary-to-secondary air-flow imbalance when in "*IEC+DX1*" mode.

Figure 7 and Figure 8 chart static pressure measurements throughout the Type C and Type M systems, with clean filters and with soiled filters. The effect of added air-flow resistance is quite evident. Product air-flow measurements at these conditions indicated that air-flow was reduced by more than 20%. Finer review of the data from June-August shows a slow decrease in full speed fan power draw for the indirect evaporative equipment, indicative of dwindling fan air-flow rate. Although it is not charted in Figure 7, it should be noted that for the soiled filters scenario, the exhaust plenum for RTU 19 was measured at a slightly negative pressure when the unit was shifted to "*IEC+DX1*" mode. It appears that depressurization of the product plenum was significant enough that air-flow was drawn backward through the secondary passages of the heat exchanger.

The key takeaway from these observations is that regular filter service for this equipment is very important. As filters soil the delivered air-flow rate decreases. Product temperature may not be impacted, and could in fact get colder, so this measurement cannot be used as a certain indicator for an air-flow problem. According to our observations, when applied to serve the continuous ventilation requirements for a retail store, we recommend replacing filters once each month. Failure to do so will have substantial impacts on cooling capacity and efficiency for the equipment, not to mention the impacts on indoor air quality. This is not a unique challenge for indirect evaporative air conditioning. Vapor-compression cooling also requires regular filter service, especially for Dedicated Outside Air Supply (DOAS) systems.

Operating Pressures for Type C System (RTU 19)

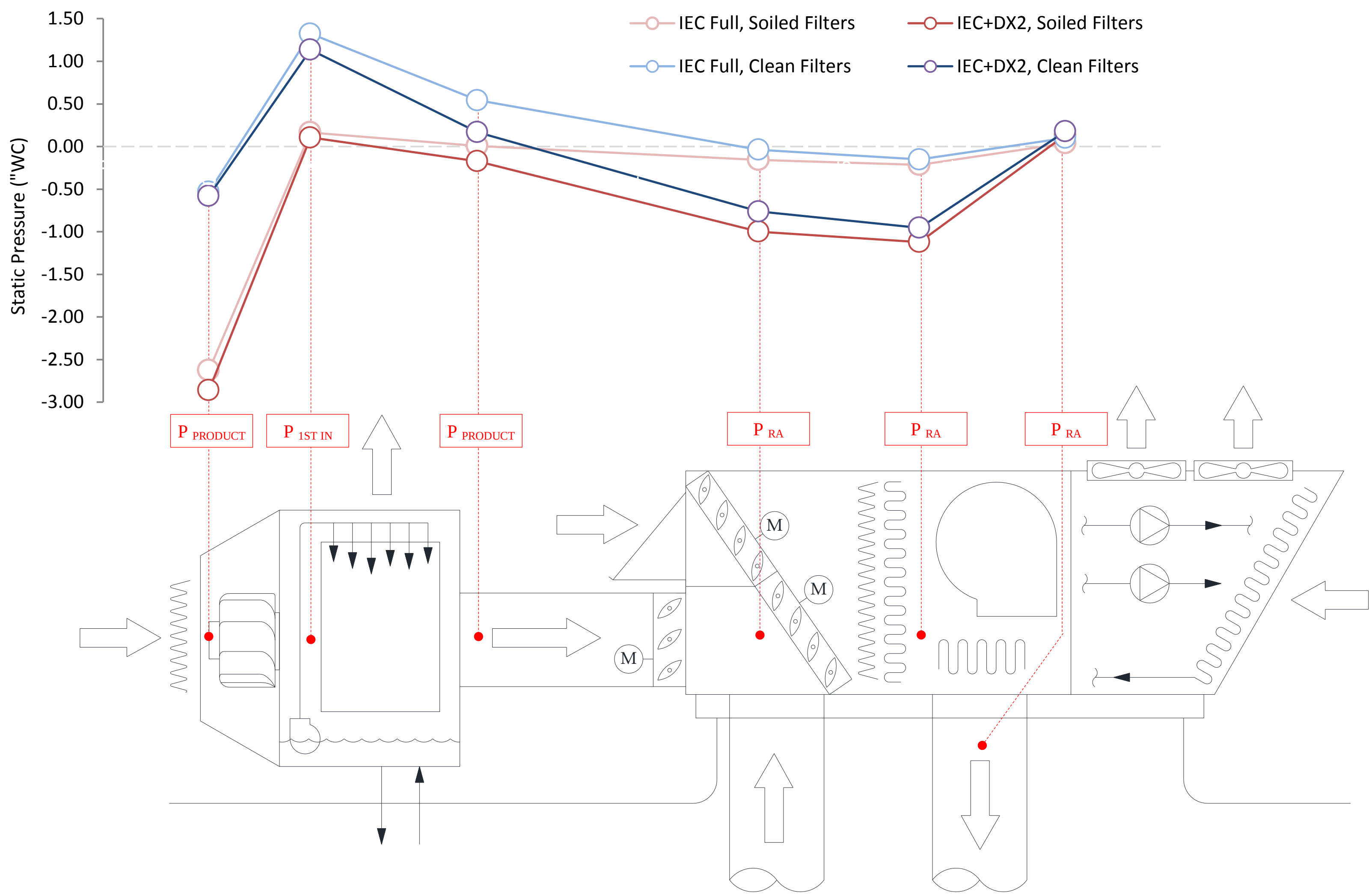

Figure 25. Static pressure measurements for Type C (RTU 16) with dirty filters and with clean filters

OPERATING PRESSURES FOR TYPE M SYSTEM (RTU 17)

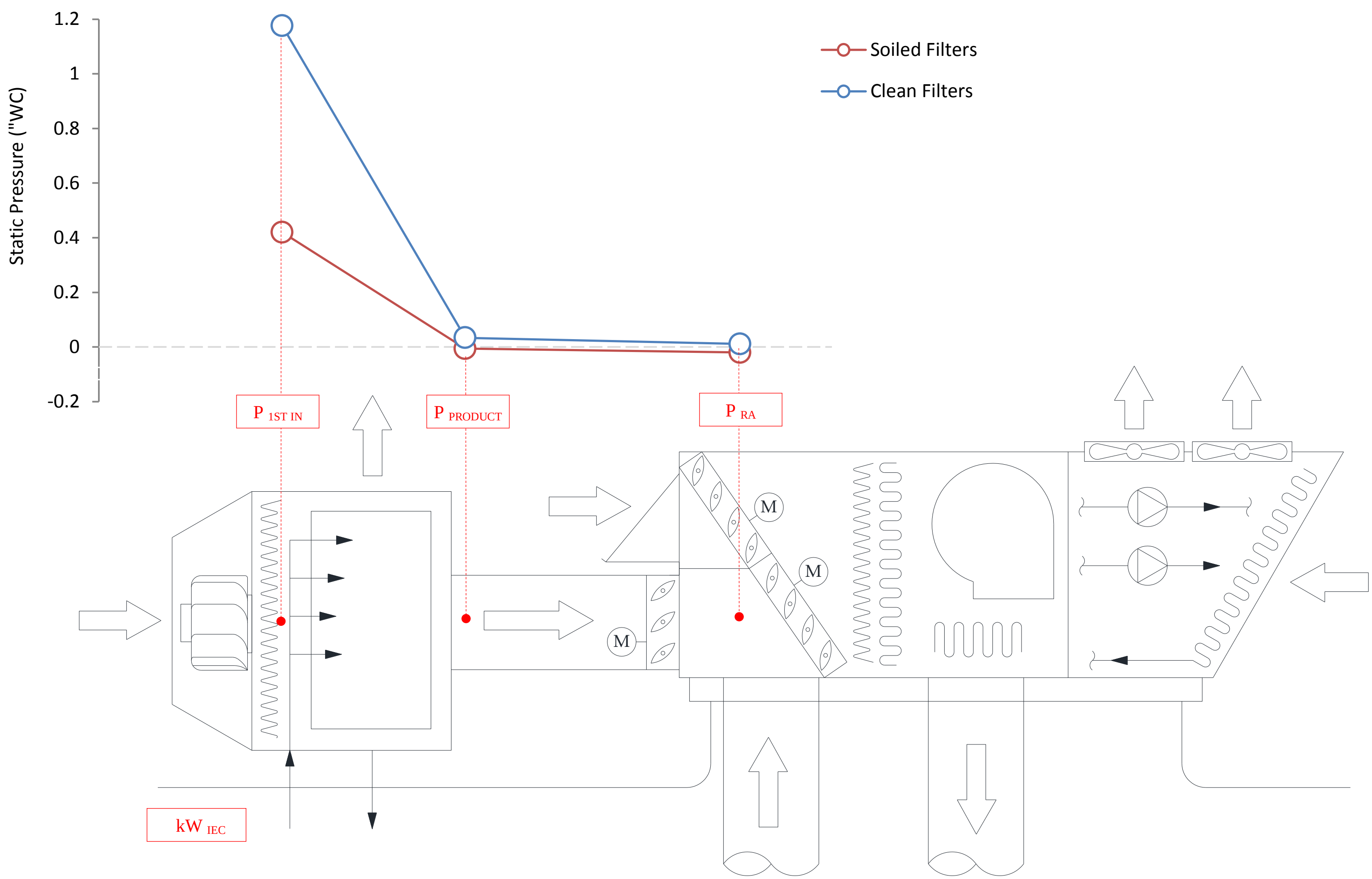

FIGURE 26. STATIC PRESSURE MEASUREMENTS FOR TYPE M (RTU 17) WITH DIRTY FILTERS AND WITH CLEAN FILTERS

INSTANTANEOUS POWER DRAW AND PEAK DEMAND CONSIDERATIONS

Figure 27 charts the electric power draw for each system for every minute of steady state operation during the periods of observation. Each mode of operation for the system exhibits a clear and distinct level of overall power draw, and there are no major differences in magnitude for each of the six systems evaluated.

Notably, full speed power for the IEC equipment does not change with outside air temperature. This is because the fan is the only major energy consuming component for the equipment, so full speed power draw for the system is independent from environmental conditions. The Type C equipment draws between 1.7 – 1.9 *kW* at full speed (0.72 W/cfm_{PA}). The Type M draws between 1.5–1.6 *kW* at full speed (0.6 W/cfm_{PA}). At 80% speed, the Type C drops to 0.95–*1.05 kW* (0.48 *W/cfm*), and Type M drops to 0.6–0.7 *kW* (0.36 *W/cfm*).

For the Type C equipment, "*IEC Full Speed*" marks two distinct power bands. This occurs when the unit reduces fan speed during the heat exchanger wetting cycle. The same pair of power bands shows up in the "*IEC+DX*" modes for the same reason. In these instances, fan power drops to *0.25kW*, corresponding to estimated 350 *cfm* drop in product airflow. The Type C systems on RTU 18 and RTU 19 also show two slightly different levels of power draw in "*IEC Part Speed*" mode, but the Type C on RTU 16 does not. Each of these distinct levels of power draw corresponds to a change in product airflow rate, as described in section *IEC Product Airflow Results.*

Power increases to the 6–8 kW range when the first stage compressor is activated, then increases to 10–12 *kW* when the second stage compressor is activated. Demand in "*IEC+DX*" modes increases significantly with outside air temperature. This rise is consistent with the decrease in efficiency and increase in power draw for conventional air conditioners at high outside air temperature. We do not expect that addition of the indirect evaporative system has a substantial effect on the compressor power draw characteristics for the rooftop unit. Compressor power draw is driven mainly by the condenser temperature, which is not changed by addition of the indirect evaporative system. The evaporator coil inlet condition is certainly different than what it would be in a baseline scenario; this will reduce sensible cooling capacity from vapor compression cooling, but only has a minor impact on compressor power draw.

It is worth considering the fact that when an indirect evaporative air conditioner is added to an existing system, the total connected load increases, and there is therefore a distinct possibility that peak electricity demand for some sites could increase. This is true whether the equipment is installed as a direct addition to an individual rooftop unit, or when the equipment is designed to operate in parallel with multiple units. In theory, the peak demand from a site should decrease substantially because cooling capacity from the indirect evaporative systems will keep compressors from turning on. We believe that this expectation should hold true for large facilities and for systems with part load capabilities, such as this pilot installation. However, there are some scenarios where we expect that peak demand from a site might actually increase. For example, such an increase may occur if an indirect evaporative air conditioner were added to a single zone retail space to operate in combination with an existing single stage rooftop air conditioner. During periods when the indirect evaporative cooler is not able to cover all room loads on its own, existing compressors would operate concurrent with the indirect evaporative cooler and the instantaneous electric load would be larger than if the rooftop unit operated on its own. Even in instances when the added capacity from indirect evaporative cooling should offset the need for the highest capacity compressor stages on peak, unless controls specifically restrict their operation it is possible that high capacity stages could operate at certain times.

For the circumstances where we expect a potential for increase in the peak instantaneous electrical demand, the compressor runtime during peak hours would certainly decrease as a result of the added capacity from the indirect evaporative system. The only exception would be for buildings where the existing conditioning systems did not provide adequate cooling capacity, and the addition of an indirect evaporative air conditioner serves to increase the level of service – improving comfort at peak. The aggregate effect from similar installations at a number of sites should be a reduction in total instantaneous load on the grid, even if in some instances demand at certain sites is higher than baseline.

Notwithstanding, we recommend that utility programs intending to advance indirect evaporative air conditioners as an efficiency measure should also consider ways to reduce connected load. Projects that install indirect evaporative air conditioners should also incorporate the intentional elimination of other cooling equipment capacity. This could be accomplished by disconnecting unneeded compressor stages, or by removing the fraction of rooftop air conditioners on large facilities whose capacity should be displaced by the indirect evaporative systems. One straight forward approach would be to remove aging rooftop units and to replace them directly with indirect evaporative equipment. This would also allow for re-use of existing roof curbs, penetrations, and ductwork systems.

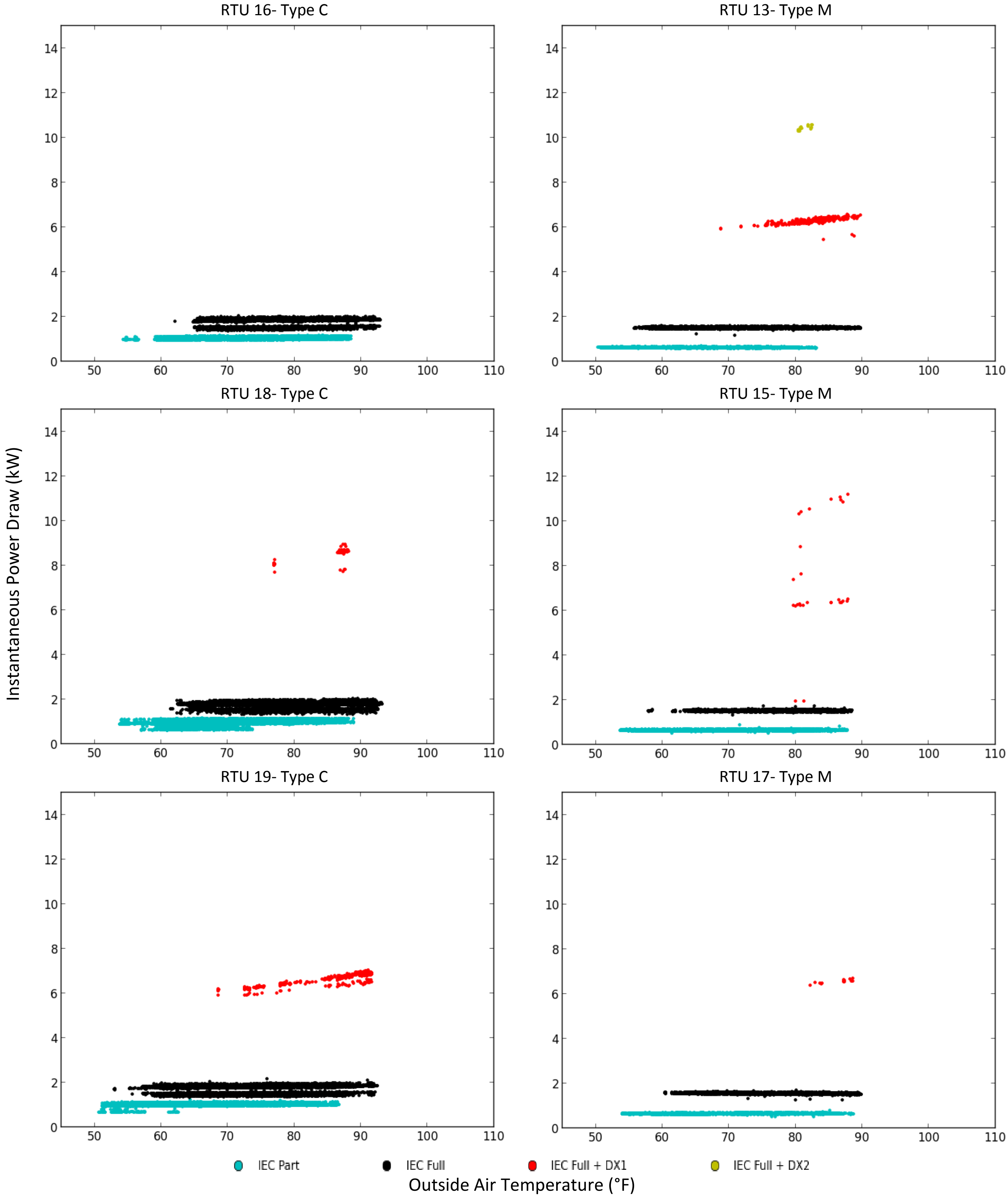

FIGURE 27. INSTANTANEOUS POWER DRAW BY OPERATING MODE VERSUS OUTSIDE AIR TEMPERATURE

TRANSIENT SYSTEM DYNAMICS

When operating at full speed, the Type C technology adjusts fan speed periodically to accommodate a wetting cycle, during which time the pumps cycle water from the sump through the heat exchanger. It is important to account for these dynamics in assessment of the overall cooling capacity and energy efficiency for the system. Our monitoring and in-field diagnostic measurements indicate that the wetting cycle occurs once every nine minutes and persists for approximately one minute. During this time fan power drops from 1.8 *kW* to 1.5 *kW* and pumps are enabled (drawing an additional 100 *W*) to circulate water . The product airflow drops by approximately 15% and the product temperature changes in response to the change in airflow and media wetting. Ultimately, this corresponds to a small and regular product air temperature fluctuation of no more than 2°F. The fan does not slow down for wetting when operating at part speed.

Figure 28 charts the time-series of product air temperature, outside air temperature, and exhaust air temperature measurements for each of the six systems over a one hour period on the evening of September 29. This is an hour that all six units were operating in the exact same mode without interruption. The small product temperature fluctuation associated with the periodic wetting cycles for the Type C is evident. These periodic changes in airflow rate and temperature are all accounted for in calculated results for cooling capacity and energy efficiency in each minute of observation, which explains the somewhat broader variation in those metrics for the Type C equipment.

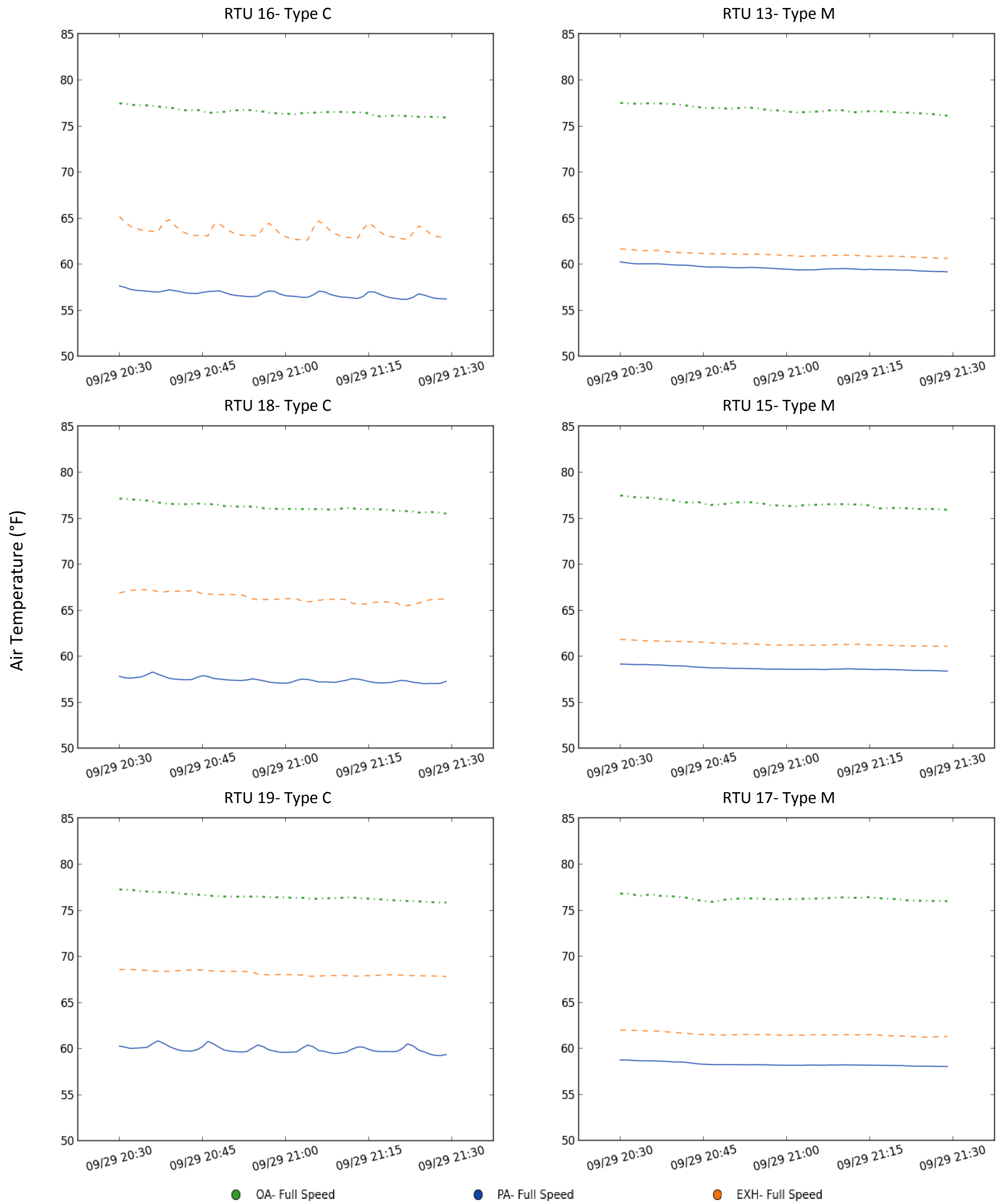

FIGURE 28. EXAMPLE TIME SERIES OF TEMPERATURE MEASUREMENTS SHOWING TRANSIENT BEHAVIOR

DISCUSSION AND CONCLUSIONS

ENERGY EFFICIENCY, EQUIPMENT PERFORMANCE, AND RELIABILITY ARE OUTSTANDING

This report presents a detailed assessment of cooling performance for two types of indirect evaporative air conditioners added onto existing rooftop packaged units. The energy efficiency demonstrated by both manufacturers tested is outstanding. For outside air conditions of 100°F, at full capacity these systems will achieve COP > 15. This degree of efficiency surpasses the practical limits for efficiency from both air and water-cooled vapor compression cooling. A modern high efficiency rooftop air conditioner would deliver cooling in a similar scenario with roughly COP=4. This equates to a demand reduction of more than 75% at full capacity. For part capacity operation at 90°F the Type M systems were observed to operate with COP as high as 25, while delivering about 75% capacity. Part capacity performance for the Type C system is somewhat lower.

The potential for annual energy savings and peak demand reduction achieved by these systems will depend on climate, application, and building type. It is important to note that these systems capture the greatest savings for cooling ventilation air. They do also achieve high efficiency for room cooling, though in applications with little or no ventilation requirements the savings potential is smaller. The results presented in this report provide a clear and thorough characterization of performance for these systems over a range of conditions. This information should form the solid basis for building energy simulations to assess the overall value of the technology in different applications and climates.

The measured water use and cooling efficiency of both indirect evaporative air conditioners align with estimates for what could achieve regionally neutral water consumption. That is, the equipment appears to save enough energy to reduce up stream water use associated with energy production by at least as much as is consumed on site. The balance for this metric changes substantially from region-to-region, but it is clear that the measured on-site water use for these systems in relation to energy savings achieved is consistent with the with the range of accepted estimates for up-stream water use associated with energy production and extraction. There may be region specific concerns, or differing valuations for different types of on-site and upstream water use. The results presented in this report provide the clear basis from which more through assessment of these questions could be made.

Aside from the compelling energy use and demand reductions demonstrated by these air conditioners, we also note that there are a number of considerations that should be held in mind for any project that applies the technology, as well as several technical and non-technical barriers to their broader adoption. These challenges persist, even while the equipment has proven to function reliably. Some of the main obstacles to broader adoption are discussed in the following section.

NEEDS FOR BROAD AND SUCCESSFUL TECHNOLOGY ADOPTION

There are not yet standard practices for how to install, integrate, and control indirect evaporative cooling as part of a larger building system. Each manufacturer provides general design guidance, but the best strategies for physical integration and control are still evolving. For example, while the technology generates cooling with very high efficiency, most market available indirect evaporative cooling equipment does not currently incorporate a way to heat ventilation air. Therefore, a building must provide separate physical apparatus to maintain ventilation when cooling is not required, and must incorporate custom controls that can manage the changeover from one system to another.

Indirect evaporative air conditioners require regular and seasonal maintenance. In our experience this is a small additional requirement and could be executed in coordination with other service efforts for conventional rooftop units. In many ways, service for these systems is more straightforward than proper service for refrigerant based vapor-compression equipment. However, lack of industry familiarity with the function for indirect evaporative air conditioners and their service needs appears to be a considerable challenge. In our observation there are two types of service required:

1. *Monthly Service:* The systems require regular filter replacement. Without filter replacement, system performance will suffer. For the pilot presented here, we estimate that filters required replacement once every 750 hours of operation – or approximately once each month. The required frequency for filter replacement will be different in each application. The application presented here probably defines the worst case scenario: 24/7 operation in a very dusty.

2. *Seasonal Service:* Equipment must be shut down for the winter, then setup again in the spring. The Type C equipment will drain automatically when there is no call for cooling, so could theoretically be left to sit throughout the winter without any trouble. The Type M continues to use some water to keep the heat exchanger wetted even when there is no call for cooling, so it is even more important to shut this system down for the winter. In either case, rooftop water distribution generally requires winterization to avoid freezing.

The seasonal shutdown and startup also serves as a good opportunity to provide some minor cleanup for the equipment. The heat exchangers on these machines appeared to remain clean throughout the period of test, but some solids deposits and debris accumulated around the heat exchanger inlets and in the sump or drain pan for each system. At the conclusion of the 2013 cooling season, the solenoid valves for some units were clogged with silt and required cleaning. These are small issues, but do require some regular attention to maintain equipment in good working order. It is important that service providers be cognizant of the specific needs for each system, and also that the service be conducted with consistency. For example, the Type M requires that service providers periodically refill a bottle of surfactant used to prime the heat exchanger. If these needs are neglected, cooling capabilities will suffer, efficiency advantages will be lost, and equipment could fail. Service was not provided as needed for the equipment studied in this project. Subsequently, all six units experienced air filter failures during the summer from over loading. Later on, the rooftop water supply was damaged from freeze during a series of cold winter nights.

These service requirements are small but important. The efforts required are well within the capabilities of normal service technicians, though it appears that the current lack a familiarity, and standardized practice do pose a major challenge to successful application on a broader scale.

Further, the overall system design deployed for this pilot proved to be rather complex. The physical retrofit was challenging as it required the addition of multiple dampers and actuators, as well as physical modification to the existing rooftop unit to manage airflow in all potential modes of operation. The approach also required a custom built controller to manage function of the rooftop unit together with the indirect evaporative cooler. The series of indexed set points for each component in each mode of operation required a time consuming commissioning effort. In the end, even with all the attention given to carefully selecting fan speeds and damper positions for every mode, some units were not held within their ideal operating conditions during all periods of time. We believe there are simpler approaches to incorporating indirect evaporative air conditioners in commercial building.

RECOMMENDATIONS

Foremost, we must emphasize the dramatic cooling energy efficiency achieved by these indirect evaporative air conditioners. Their impact is most substantial during peak cooling periods where these results indicate that systems will reduce energy used for cooling ventilation air by more than 75%. Peak demand reduction represents a strategic need for management of California's electric grid. The equipment also provides substantial annual energy savings, which will depend on climate and application.

We recommend development of utility programs and other efforts that can support the broader adoption of these technologies. Such programs should give significant weight to the value of peak demand reduction, and the fact that demand reduction for cooling offsets the need for increased electric generation capacity. Each MW of demand reduction for cooling can be thought of as one less MW of peak generating capacity requirement. Some utilities, such as Xcel Energy, and Con Edison are currently providing incentive rebates for peak demand reduction at $400–$1,200 *per kW*. Indirect evaporative air conditioners currently have a high incremental cost compared to conventional air conditioners, but programs that give generation-value incentives for savings at peak will make the measure cost competitive with code-minimum baseline practices.

We conclude that the technology studied is well designed and robust. We have no recommendations for needed improvements to the equipment as manufactured. However, there are some needs related to implementation, system integration, and operation and maintenance.

The installation and controls integration of this equipment needs to be simpler. The improvements for controls and sequence of operation could be accomplished by including operation of the indirect evaporative air conditioners as part of the integrated energy management system for a building. Alternatively, manufacturers could develop on-board controls that allow them to interface directly with an existing EMCS without requiring revision to the building's existing sequence of operations. Ideally, these indirect evaporative air conditioners could directly replace existing rooftop units and operate correctly off of the same control inputs. The later approach could be difficult; as we have discussed throughout this report there are needs for coordination with other systems that cannot be achieved solely by control of the indirect evaporative system. Most importantly, the building must maintain ventilation requirements when there is no a need for cooling, and there must be a way to temper ventilation air when outdoor conditions are too cold.

To simplify the physical installation challenges associated with this demonstration, we recommend that indirect evaporative air conditioners could be installed to replace individual rooftop units, instead of as an addition to existing equipment. Several systems could also be installed as a single bank with a new roof penetration and new ductwork for ventilation distribution. Either option would avoid the complicated series of dampers used for this pilot. However, both options still require careful controls integration as discussed above.

The two manufacturers tested here provide stand-alone thermostats which could be utilized to schedule and control equipment separately from a building's EMCS, or in parallel with the room thermostats that may control other rooftop units in a building. We caution against using these controls as the sole means of managing the indirect evaporate air conditioners when other conventional rooftop units also serve the same zone. In order to achieve energy savings, it is important to prioritize cooling from the indirect evaporative systems over cooling from conventional equipment. The separate thermostat controls do not guarantee that this will happen. We have observed instances where cooling set-points for the more efficient system are set higher than the conventional equipment, in which case the more efficient option is never given the opportunity to cool.

Beyond the various options for integration of these indirect evaporative air conditioners in their current form factor, we recognize that there are important opportunities for further product development in the future.

First, we recommend that the equipment evaluated here could be adapted into the construction of hybrid packaged rooftop units that integrate indirect evaporative cooling and vapor-compression within the same box. The major advantage for such a system would be that it could directly replace existing rooftop units one-for-one without the need for other complications and systems integration. This type of packaged hybrid equipment has been tested and demonstrated previously for the Western Cooling Challenge, and could resolve many of technical challenges experienced in this project. A packaged hybrid unit could also allow for application in smaller commercial settings where indirect evaporative equipment might not be cost effectively deployed in addition to a mostly redundant vapor-compression air conditioner.

Further, we believe it would be advantageous to incorporate these indirect evaporative heat exchangers into a Dedicated Outside Air Supply (DOAS) air handler. This type of hybrid air handler should include a heating section, and might also benefit from incorporating vapor-compression cooling. Ideally, such a DOAS system would handle all ventilation needs for a building throughout the year, and would be designed to integrate seamlessly with existing building controls. Both Type M and Type C manufacturers have recently developed just such equipment. Many large retail facilities already utilize vapor-compression DOAS air handlers. A hybrid DOAS machine could replace these systems one-for-one, with substantial energy and demand savings. One unique advantage of the DOAS option is that it can allow for certain indirect evaporative heat exchangers could utilize return air as the source for the secondary air stream, which could improve cooling capacity and efficiency. A DOAS that uses return air as the source could extend the geographical range in which indirect evaporative cooling could be applied, and could double as an exhaust heat recovery system in the wintertime (Woolley 2014).

For applications that would install the stand-alone indirect evaporative air conditioners in series with another rooftop unit (as was done for this study), we recommend that manufacturers provide a packaged retrofit solution that would include all dampers and controls to allow for straightforward integration. Given that performance is sensitive to downstream pressure, we suggest that such a retrofit package would utilize active pressure sensing control scheme adjust damper positions and fan speeds. The indexed set point approach designed for this study did not provide optimal control, and was onerous to commission.

Regardless of the approach that is applied for future applications of this technology, we recognize that there is a significant need for standard specifications and design guidelines. We recommend that manufacturers, utilities, and industry organizations cooperate to develop guidelines and standards that describe appropriate physical system designs and sequence of operations. For example, these standards would ensure that the most efficient systems and modes of operation are given priority, and that ventilation needs are handled appropriately throughout the year. Standards could also specify the functionality for Fault Detection Diagnostics to ensure the persistence of high efficiency operation. Without such guidelines, each project will require custom design and development on the part of project engineers and contractors.

Lastly, as utility programs and other efforts take actions to encourage the broader application of these technologies, we recommend that such programs make every effort to address market barriers to ensure successful and persistent energy and demand savings. These efforts should clearly identify prerequisite requirements to ensure appropriate application and ongoing management of the systems. Among other details, these requirements should include:

1. Project design, installation, controls and commissioning will be conducted by manufacturer-trained engineers and contractors.
2. All installations will include a multiple-year service agreement with manufacturer-trained service provider. We recommend that this service program be paid for in part by rebates, and managed and coordinated by each associated manufacturer.
3. Customer will agree to maintain regular service for the equipment in line with manufacturer guidelines for the life of the equipment. We suggest that programs consider tying future customer incentives to ensuring that previously supported installations have been maintained appropriately.
4. A portion of all supported installations will be audited by an independent third party to ensure proper application. We recommend that ongoing programs for upstream rebates be adjusted in proportion to the distribution of audits that show appropriate operation.
5. Require certain prescriptive design elements and operating capabilities, such as: automatic freeze protection for exposed water distribution, priority operation for the most efficient systems and modes of operation, or mandatory reduction of total connected load.
6. Fault Detection & Diagnostic capabilities with the capacity to communicate alarms off of the rooftop.

The energy and demand savings potential demonstrated by the equipment studied here is compelling. We recommend further attention and support surrounding the technology as it appears to hold significant promise to support the strategic energy goals established by the California Energy Efficiency Strategic plan AB 32, and many other policy initiatives. Successful implementation and broader uptake for the technology will require navigating a variety of technical and non-technical complications. We hope that the findings from this pilot can help to guide the ways forward.

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APPENDIX A

AIRFLOW SCHEMATICS FOR EACH MODE OF OPERATION

MODES: IEC PART CAPACITY | IEC FULL CAPACITY

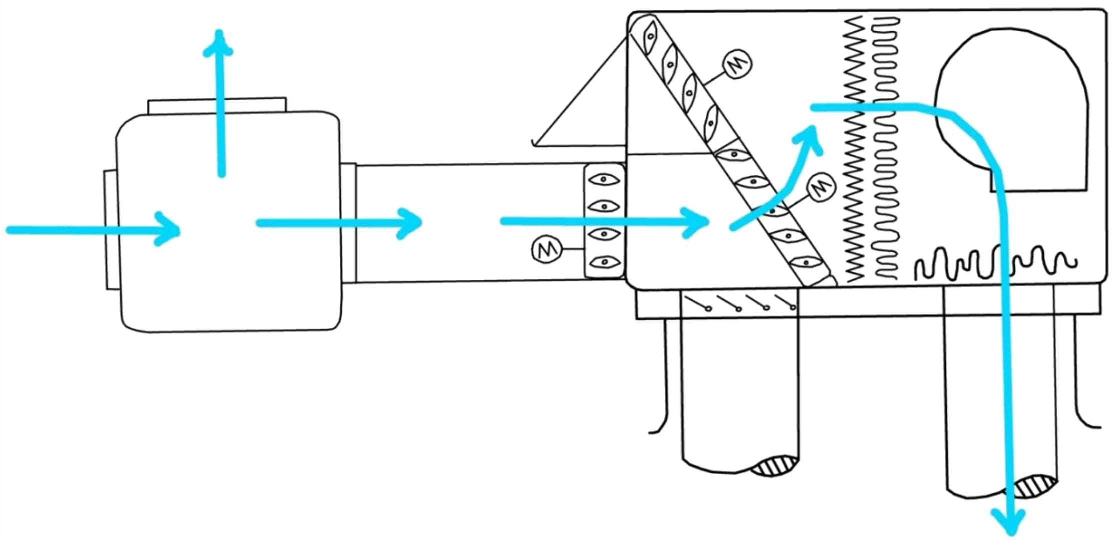

FIGURE 29. AIRFLOW SCHEMATIC FOR INDIRECT EVAPORATIVE COOLING ONLY (PART CAPACITY AND FULL CAPACITY)

MODES: IEC & DX1 | IEC & DX2

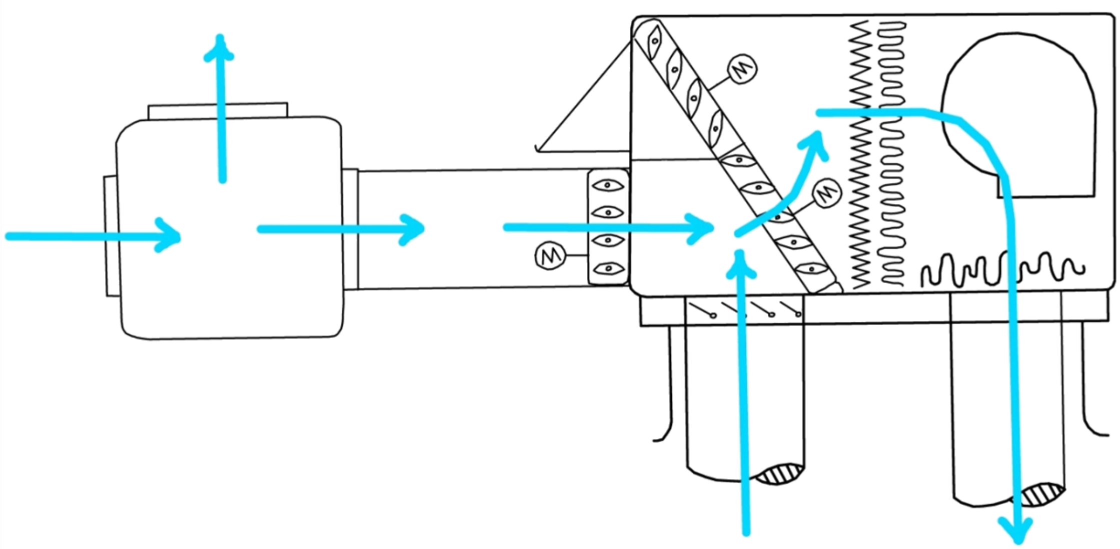

FIGURE 30. AIRFLOW SCHEMATIC FOR INDIRECT EVAPORATIVE PLUS VAPOR-COMPRESSION COOLING

MODES: IEC, ECONOMIZER & DX1 | IEC, ECONOMIZER & DX2

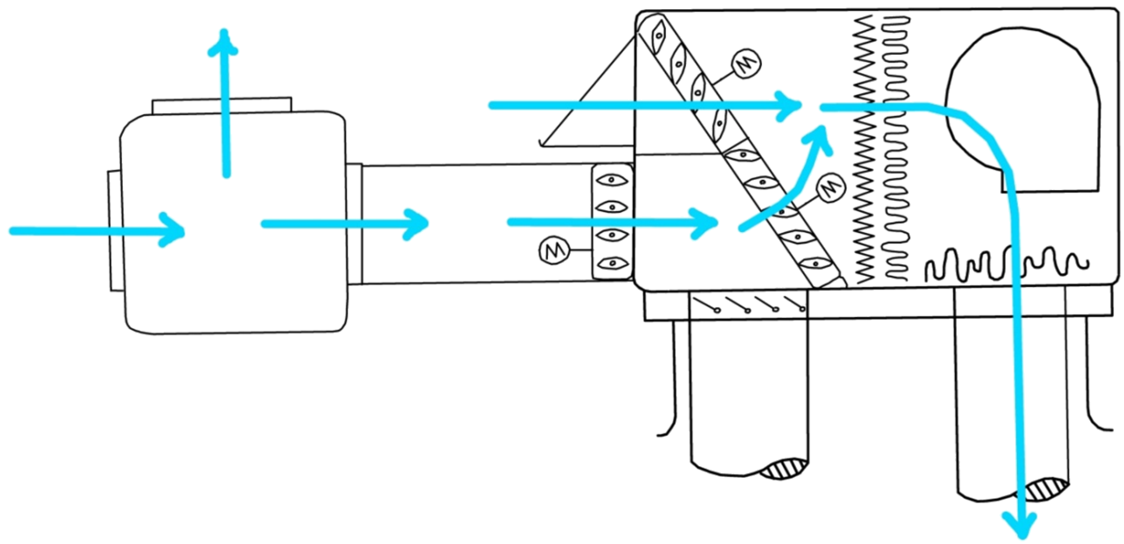

FIGURE 31. AIRFLOW SCHEMATIC FOR INDIRECT EVAPORATIVE, ECONOMIZER, PLUS VAPOR-COMPRESSION COOLING

MODE: VENTILATION ONLY

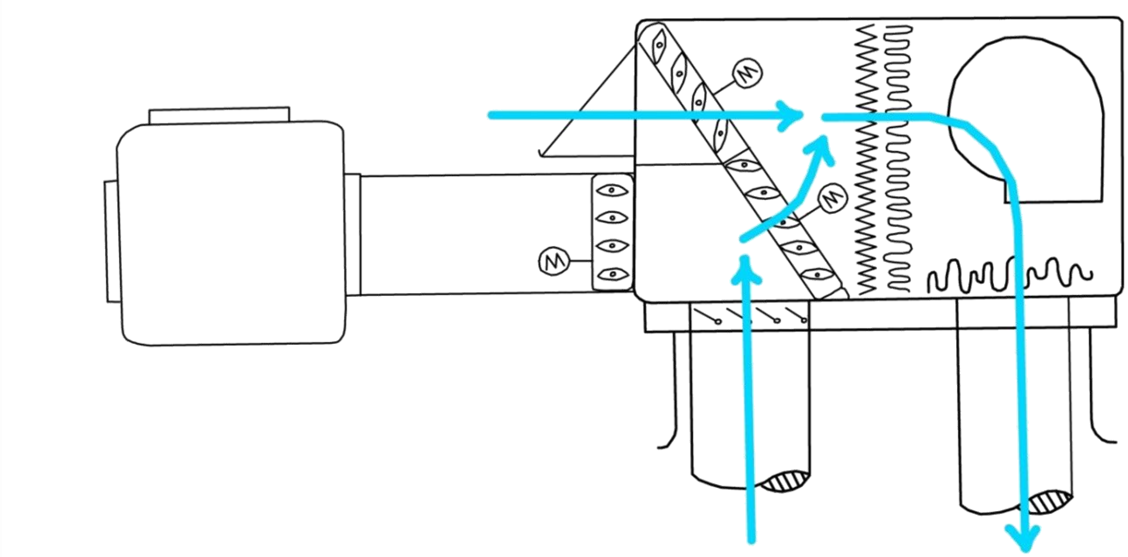

FIGURE 32. AIRFLOW SCHEMATIC FOR VENTILATION ONLY MODE

MODE: HEATING

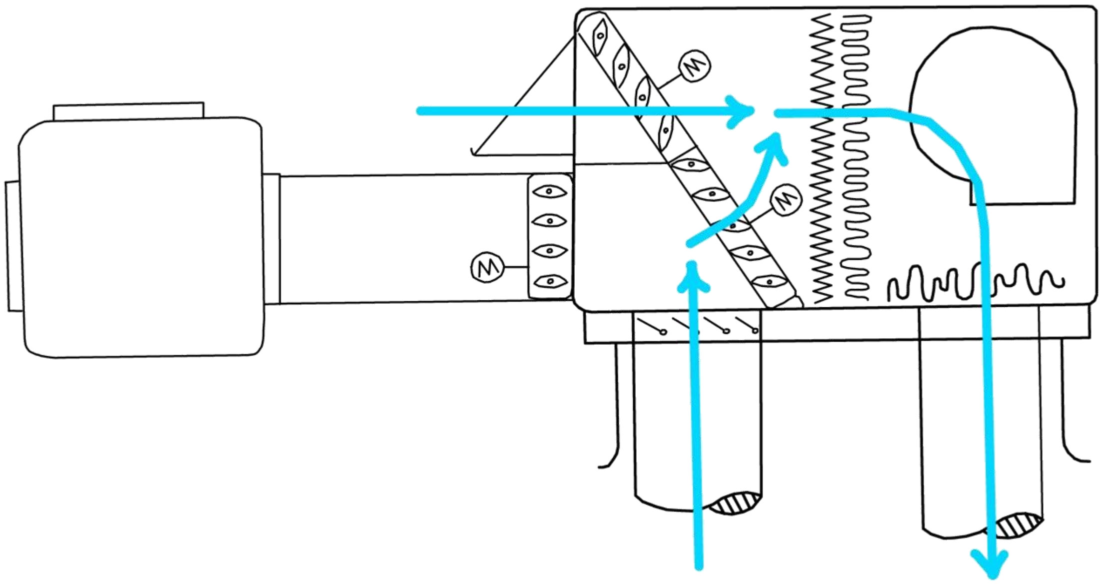

FIGURE 33. AIRFLOW SCHEMATIC FOR HEATING

APPENDIX B

COMPLETE SEQUENCE OF OPERATIONS

NORMAL OPERATION, NON-ECONOMIZER MODES

In normal operation (*as long as the evaporative equipment has not been winterized*), when the space is scheduled as occupied, and when the outside air temperature is above the economizer changeover point (*so that Novar ETM 2024 does not signal 24V VAC on relay "Damper"*):

MODE: VENTILATION ONLY:

As long as the cooling and heating set point temperatures are satisfied:

Host EMS shall signal 24*VAC* on Novar ETM 2024 relay "*Fan*"

The systems will operate to provide fresh air ventilation ("*Ventilation Only*" mode):

The outside air damper shall actuate fully open

The return air damper shall actuate to a (field selected "F%") position to maintain appropriate mixing of outside air with return air.

The (E) RTU supply fan shall modulate to a (field selected "N%") speed to maintain the scheduled minimum outside air-flow

The IEC fan shall remain off

The IEC product damper shall actuate fully closed

MODE: IEC PART CAPACITY:

When the space cooling setpoint is exceeded (SP_{COOL}+0.5°F):

Host EMS shall signal 24 *VAC* on Novar ETM 2024 relays "*Fan*" and "*Cool S1*"

Indirect evaporative cooling will be initiated with an air-flow that meets the scheduled minimum outside air-flow requirement ("*IEC Part Capacity*" mode):

The outside air damper shall actuate fully closed

The product damper shall actuate fully open

The IEC fan shall modulate to a (field selected "A%") speed that provides the scheduled minimum ventilation requirement.

The (E) RTU supply fan shall modulate to a (field selected "O%") speed that overcomes excessive resistance to flow through the (E) RTU and ductwork.

The return air damper shall actuate to a (field selected "G%") position that maintains appropriate static pressure in the product air plenum(M50 = (+)0.1*"WC;* CW-H15=(+)0.8*"WC*)

The IEC's internally controlled cooling sequence will be initiated.

If operation in "*IEC Part Capacity*" mode cools the space (to SP_{COOL} –0.5°F):

Systems will return to operation in "*Ventilation Only*" mode.

MODE: IEC FULL CAPACITY:

If the space temperature rises further (SP_{COOL}+1.0°F):

Host EMS shall signal 24 *VAC* on Novar ETM 2024 relays "*Fan*", "*Cool S1*" and "*Cool S2*"

IEC will modulate to full air-flow ("*IEC Full Capacity*" mode):

The outside air damper shall remain fully closed

The product damper shall remain fully open

The IEC fan speed shall modulate to 100%

The (E) RTU supply fan shall modulate to a (field selected "P%") speed that overcomes excessive resistance to flow through the (E) RTU and ductwork.

The return air damper shall actuate to a (field selected "H%") position that maintains appropriate static pressure in the product air plenum(M50 = (+)0.1 *"WC;* CW-H15=(+)0.8 *"WC*)

The IEC internally controlled cooling sequence shall remain enabled.

MODE: IEC & DX1

If operation in "*IEC Full Capacity*" mode cools the space (to SP_{COOL}+0.0°F):

Systems will return to operation in "*Ventilation Only*" mode.

If operation in "*IEC Full Capacity*" mode persists for "AE" minutes without cooling the space (to SP_{COOL}+0.0°F):

Host EMS shall continue to signal 24 VAC on Novar ETM 2024 relays "*Fan*", "*CoolS1*" and "*CoolS2*"

IEC shall shift to minimum ventilation air-flow and DX1 shall initiate ("*IEC & DX1*" mode):

The outside air damper shall remain fully closed

The return air damper shall actuate fully open

The IEC fan speed shall adjust to a (field selected A%) speed that provides the scheduled minimum ventilation requirement.

The IEC internally controlled cooling sequence shall remain enabled

The (E) RTU supply fan shall modulate to a (field selected "Q%") speed that provides the greater of:

A. The minimum allowable evaporator coil air-flow for DX1
B. The (field selected "P%") speed used for "*IEC Full Capacity*" mode

Compressor 1 and corresponding condenser fans shall operate

The product damper shall actuate to a (field selected "B%") position that maintains appropriate static pressure in the product air plenum(M50 = (+)0.1*"WC;* CW-H15=(+)0.8*"WC*)

If operation in "*IEC & DX1*" mode cools the space (to SP_{COOL}+0.0°F):

Systems will return to operation in "*Ventilation Only*" mode.

MODE: IEC & DX2

If operation in "*IEC & DX1*" mode persists for "AF" minutes without cooling the space (to SP_{COOL}+0.0°F):

Host EMS shall continue to signal 24 VAC on Novar ETM 2024 relays "*Fan*", "*CoolS1*" and "*CoolS2*"

IEC shall remain at minimum ventilation air-flow and DX2 shall initiate ("*IEC & DX2*" mode):

The outside air damper shall remain fully closed

The return air damper shall remain fully open

The IEC fan speed shall remain at a (field selected "A%") speed that provides the scheduled minimum ventilation requirement

The IEC internally controlled cooling sequence shall remain enabled

The (E) RTU supply fan shall modulate to a (field selected "R%") speed that provides the greater of:

A. The minimum allowable evaporator coil air-flow for DX2
B. The (field selected "P%") speed used for "*IEC Full Capacity*" mode

Compressor 1, compressor 2, and corresponding condenser fans shall operate

The product damper shall actuate to a (field selected "C%") position that maintains appropriate static pressure in the product air plenum (M50 = (+)0.1*"WC;* CW-H15=(+)0.8*"WC*)

If operation in "*IEC & DX1*" mode cools the space (to SP_{COOL}+0.0°F):

Systems will return to operation in "*Ventilation Only*" mode.

NORMAL OPERATION, ECONOMIZER MODES

In normal operation (*as long as the evaporative equipment has not been winterized*), when the space is scheduled as occupied (*always for Host 1624*), and when the outside air temperature is below the economizer changeover point:

MODE: IEC PART CAPACITY

When the space cooling setpoint is exceeded (SP_{COOL}+0.5°F):

Host EMS shall signal 24 *VAC* on Novar ETM 2024 relays "*Fan*" and "*Damper*"

Indirect evaporative cooling will be initiated with an air-flow that meets the scheduled minimum outside air-flow requirement ("*IEC Part Capacity*" mode):

The outside air damper shall actuate fully closed

The product damper shall actuate fully open

The IEC fan shall modulate to a (field selected "A%") speed that provides the scheduled minimum ventilation requirement.

The (E) RTU supply fan shall modulate to a (field selected "O%") speed that overcomes excessive resistance to flow through the (E) RTU and ductwork.

The return air damper shall actuate to a (field selected "G%") position that maintains appropriate static pressure in the product air plenum(M50 = (+)0.1*"WC;* CW-H15=(+)0.8*"WC*)

The IEC's internally controlled cooling sequence will be initiated.

If operation in "*IEC Part Capacity*" mode cools the space (to SP_{COOL} –0.5°F):

Systems will return to operation in "*Ventilation Only*" mode.

MODE: IEC FULL CAPACITY

As the space temperature rises further (SP_{COOL}+1.0°F):

Host EMS shall signal 24 VAC on Novar ETM 2024 relays "*Fan*", "*Damper*" and "*Cool S1*"

IEC will modulate to full air-flow ("*IEC Full Capacity*" mode):

The outside air damper shall remain fully closed

The product damper shall remain fully open

The IEC fan speed shall modulate to 100%

The (E) RTU supply fan shall modulate to a (field selected "P%") speed that overcomes excessive resistance to flow through the (E) RTU and ductwork.

The return air damper shall actuate to a (field selected "H%") position that maintains appropriate static pressure in the product air plenum(M50 = (+)0.1 *"WC;* CW-H15=(+)0.8 *"WC*)

The IEC internally controlled cooling sequence shall remain enabled.

If operation in "*IEC Full Capacity*" mode cools the space (to SP_{COOL} + 0.0°F):

Systems will return to operation in "*Ventilation Only*" mode.

Modes: IEC, Economizer & DX1

If the space temperature rises further (SP_{COOL}+1.5°F):

Host EMS shall signal 24VAC on Novar ETM 2024 relays "*Fan*", "*Damper*", and "*Cool S2*"

IEC shall shift to provide minimum ventilation air-flow, and DX1 shall initiate ("*IEC, Economizer & DX1*" mode):

The outside air damper shall actuate fully open

The product damper shall actuate fully open

The return air damper shall actuate to a (field selected "K%") position that maintains appropriate static pressure in the product air plenum(M50 = (+)0.1 *"WC;* CW-H15=(+)0.8 *"WC*)

The IEC fan shall modulate to a (field selected "A%") speed that provides the scheduled minimum ventilation requirement.

The IEC internally controlled cooling sequence shall remain enabled

The (E) RTU supply fan shall modulate to a (field selected "U%") speed that provides the greater of:

A. The minimum allowable evaporator coil air-flow for DX1
B. The (field selected) speed used for "*IEC Full Capacity*" mode

Compressor 1 and corresponding condenser fans shall operate

If operation in "IEC, Econ. & DX1" mode cools the space (to SP_{COOL} + 0.5°F):

Systems will return to operation "*IEC Part Capacity*" mode.

Mode: IEC, Economizer & DX2

If operation in "IEC, Econ. & DX1" persists for more than "AG" minutes without cooling the space (to SP_{COOL} + 0.5°F):

Host EMS will continue to signal relays "*Fan*", "*Damper*", and "*Cool S2*"

IEC shall remain at minimum ventilation air-flow and DX2 shall initiate ("*IEC, Economizer & DX2*" mode):

The outside air damper shall remain fully open

The product damper shall remain fully open

The return air damper shall actuate to a (field selected "L%") position that maintains appropriate static pressure in the product air plenum(M50 = (+)0.1 *"WC;* CW-H15=(+)0.8 *"WC*)

The IEC fan shall remain at a (field selected "A%") speed that provides the scheduled minimum ventilation requirement.

The IEC internally controlled cooling sequence shall remain enabled

The (E) RTU supply fan shall modulate to a (field selected "V%") speed that provides the greater of:

A. The minimum allowable evaporator coil air-flow for DX2
B. The (field selected) speed used for "*IEC Full Capacity*" mode

Compressor 1, compressor 2, and corresponding condenser fans shall operate

NORMAL OPERATION, HEATING MODE

In normal operation (*as long as the evaporative equipment has not been winterized*), when the space is scheduled as occupied (*always for Host 1624*), and when the space demands heating:

MODE: HEATING

If space temperature drops below the setpoint (to $SP_{HEAT} - 0.5°F$):

Host EMS will signal 24*VAC* on Novar ETM 2024 relays "*Fan*" and "*Heat S1*" and/or "*Heat S2*"

Systems will activate to provide heat ("*Heat 1*" and "*Heat 2*" modes):

The IEC fan and cooling sequence shall both remain off

The IEC product damper shall actuate fully closed

The outside air damper shall modulate to a (field selected "D%") position to provide minimum ventilation while heating is active.

The return air damper shall be fully open

The RTU fan speed shall modulate to a speed appropriate for Stage 2 heating