PERFORMANCE EVALUATION FOR INDIRECT EVAPORATIVE COOLING ON CELLULAR SITES

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EXECUTIVE SUMMARY

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

The objective of this study is to conduct field evaluations of Indirect Evaporative Coolers (IECs) in California Climate Zone 8. IECs were installed at two different cellular sites in the cities of Cudahy, CA and Placentia, CA during the summer of 2014. The evaluation studied real-world equipment operation and developed characterizations of the overall system performance and energy efficiency across a range of operating conditions. The study is designed to investigate performance characteristics that cannot be captured by steady-state laboratory testing.

This evaluation carefully disaggregates performance in each mode of operation to consider efficiency of each system state, and to investigate the implications of the control strategies and field-selected settings that were applied. Characteristic performance metrics for the IECs were calculated from data collected over the month of September 2014. During September 2014, both field test sites experienced typical cooling season conditions with average outside air temperatures above 75°F. Tables 1 and 2 summarize the results for the IECs in five degree temperature bins ranging from 80 – 105°F.

The IECs tested use 100% Outside Air (OA) which makes it difficult to determine the true value of the room cooling metrics. For this study, the research team used an ideal room temperature of 80°F. The research team recognizes that 80°F is not always representative of the return air temperature seen by the baseline, but for comparison between the systems positive room cooling credit for the difference between the supply air temperature and 80°F. The results only focus on sensible cooling metrics. The company running the cell site has alarms for high humidity situations, but none of the Heating, Ventilating, and Air Conditioning (HVAC) equipment is set up to control the humidity in the space.

It is important to remember that with IECs, the room cooling capacity decreases with an increase in outside air temperature. Therefore, in certain applications, IECs are not suited for a one-to-one replacement with conventional direct expansion (DX) equipment. However, IECs will still provide substantial savings if used to partially replace or offset DX cooling.

The research team recommends IECs as an impactful measure to reduce energy consumption and peak demand for cooling in commercial buildings. However, we also recommend that utility efficiency programs, and other efforts to advance the technology, should remain cognizant of some of the challenges that can hinder performance and limit the persistence of savings. It is especially important that any installation of this measure be paired with a quality service agreement. In our observation, the current lack of industry familiarity with the technology can result in untimely failure or abandonment of the measure. For example, IECs use 100% outside air which typical results in filter changes on a shorter interval than conventional systems. If possible, the research team suggests that new systems be installed with a guaranteed system performance for set period of time. This

might necessitate a different type of capital and incentive structure but will address some of the challenges that currently plague performance for HVAC equipment in commercial buildings. In addition, IEC equipment needs to be drained to prevent freezing of water pipes in the winter. California Climate Zone 8 rarely frosts or freezes which negated the need for a seasonal service in this field test.

The research team also recommends the 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. The market penetration for IECs is still small and the benefits of such an incentive or rebate program will help end-users with the larger up-front costs.

Finally, the research team recommends that further research be conducted to translate the characteristic measurements from this study into a calibrated model for an indirect evaporative system. The results presented in this report stand as particular examples, but the characteristic observations allow for development and validation of a general map for the technology that can be applied to other scenarios through building energy modeling. This model can be used to simulate savings across an array of climates and applications. It can also help programs target strategic savings opportunities, identify approaches to optimize control of IECs, and inform the development of design guidelines to support broad and successful application of the measure.

TABLE 1. SUMMARY OF IEC RESULTS AT CUDAHY, CA

KEY METRIC	80°F - 85°F	85°F - 90°F	90°F - 95°F	95°F - 100°F	100°F - 105°F
Sensible System COP (–)	10.2	13.8	16.7	17.6	18.6
Sensible System Capacity (kbtu/hr)	20.0	26.9	32.2	33.9	35.6
Sensible Room COP (-)	8.6	8.4	7.3	4.0	2.5
Sensible Room Capacity (kbtu/hr)	16.9	16.5	14.1	7.7	4.8
Water Use Efficiency (Gal/Ton-hr)	4.4	3.8	3.5	3.7	3.4

TABLE 2. SUMMARY OF IEC RESULTS AT PLACENTIA, CA

KEY METRIC	80°F - 85°F	85°F - 90°F	90°F - 95°F	95°F - 100°F	100°F - 105°F
Sensible System COP (–)	7.0	8.7	10.6	11.6	12.2
Sensible System Capacity (kbtu/hr)	37.5	45.6	56.1	59.7	60.9
Sensible Room COP (-)	5.9	5.6	5.7	4.6	3.1
Sensible Room Capacity (kbtu/hr)	31.8	29.6	30.3	23.7	15.8
Water Use Efficiency (Gal/Ton-hr)	2.9	2.5	2.4	1.9	2.5

ABBREVIATIONS AND ACRONYMS

$\dot{H}$	Cooling Capacity, (Enthalpy Flow Rate) (e.g. *kBtu/h*)
$\dot{V}$	Volume Flow Rate (e.g. *scfm*)
$\dot{m}$	Mass Flow Rate (e.g. *lbm/h*)
CEC	California Energy Commission
c_p	Specific Heat Capacity (e.g. *Btu/lbm-°F*)
CPUC	California Public Utilities Commission
DX	Direct Expansion Vapor Compression
EXH	Exhaust Air
h	Specific Enthalpy (e.g. *Btu/lbm-dryair*)
HVAC	Heating, Ventilating, and Air Conditioning
IEC	Indirect Evaporative Air Conditioner (Indirect Evaporative Cooling)
OA	Outside Air
RA	Return Air
RH	Relative Humidity (%)
RTU	Rooftop Air Conditioning Unit
SA	Supply Air
Sq.ft.	Square Feet
T	Temperature (e.g. *°F*)
W	Watt
WBE	Wet Bulb Effectiveness
ΔP	Differential Static Pressure (e.g. *inWC*)

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INTRODUCTION

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

Buildings consume 70% of the electricity in the United States, 50% of which is used for commercial buildings. Air conditioning and ventilation is responsible for more than 25% of the annual electricity use from commercial buildings in California.

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

This report documents the results of a field study designed to characterize performance for two indirect evaporative air conditioning products, referred to here as the "IEC-C" and "IEC-P" equipment. These systems use indirect evaporative heat exchangers that use water evaporation to cool a building without adding moisture 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 30 years, in which time the technology has made major advancements.

This study emerges from a variety of efforts and innovation surrounding climate-appropriate cooling strategies. The California Energy Efficiency Strategic Plan sets aggressive targets to advance the presence and market adoption of such measures (California Public Utility Commission (CPUC 2011)). Climate appropriate measures leverage technology that may not be appropriate for all climates, but will have unique potential to decrease energy use and peak demand in specific scenarios.

This report describes the technology evaluated, experimental design, and technical methodology used in a field evaluation with this technology. Characteristic performance metrics results are presented and discussed. Technical advantages apparent to the technology, lessons learned through the process, and recommendations are included to foster further advancements and application for the technology.

PROJECT OVERVIEW

Indirect evaporative air conditioners employ specially designed heat exchangers that use water evaporation in one air stream to transfer sensible cooling to 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 also be referred to as "supply" air.

Indirect evaporative cooling can be very efficient. It is different from a direct evaporative cooling in three significant ways, including:

- It does not add moisture to the conditioned space;
- It can cool to a lower temperature; and
- It exhausts a portion of the air moved.

IECs consume more fan power per delivered air-flow (*W/cfm*) than a conventional direct evaporative cooler because a portion of the air is exhausted. 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. Other designs utilize a tube-in-flow approach similar to an evaporative fluid cooler, heat pipes, or 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 the inlet for the secondary air stream (Figure 1). These systems can generate primary air at a temperature lower than the wet-bulb temperature of the system inlet air. 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 the primary air toward the lower wet-bulb temperature. In theory, a system that repeatedly cascades flow in such a way can achieve primary air at the dew point temperature of the system inlet. Generally, as the amount of primary airflow diverted into the secondary side of the heat exchanger increases, primary temperature will decrease and the fan power required to deliver each unit of primary air flow into a space increases. This iterative process means that a heat exchanger design can optimize for cooling capacity, sensible efficiency, or for delivered temperature.

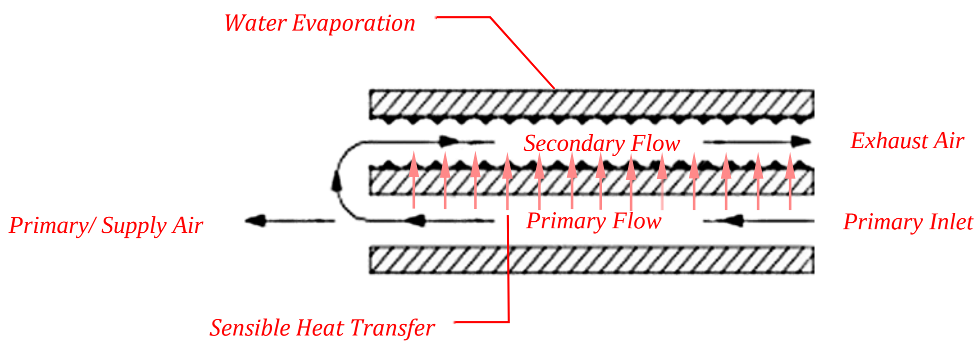


FIGURE 1. CONCEPTUAL SCHEMATIC FOR INDIRECT EVAPORATIVE COOLING

Indirect evaporative cooling systems can also be configured to 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 exhaust air exits these systems near saturation, meaning that the exhaust air-flow is cooler than outside air. The exhaust air can be applied for some useful purpose, such as cooling the condenser air for a vapor-compression system.

The two technologies studied here both utilize outside air only, and were not configured to process any return air, or to repurpose exhaust air flow. 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 primary 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.

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 IEC-P system allows all of the primary air-flow to pass through the heat exchanger before diverting a portion of the primary air 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 IEC-C 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 other important difference between the two technologies is the IEC-P utilizes a sump where the unevaporated water is mixed with make-up water. The sump water is reused until the water quality drops below a control threshold which causes the whole sump to be drained and refilled. The IEC-C does not have a sump; therefore all unevaporated water is drained after a single pass through the heat exchanger.

OVERVIEW OF FIELD TEST SITES

University of California Davis' Western Cooling Efficiency Center was retained to conduct the field assessment, analyze the collected data, and present the study results in this report.

Two field sites were selected for this field performance evaluation in the cities of Cudahy, CA and Placentia, CA. For the remainder of this report the sites will be referred to by the name of the city in which they exist. Both sites are cellular sites operated by a cellular telephone company. The companies operating standards hold the room to a temperature between 78 –

82°F during normal operation. Although the equipment data handling equipment can function properly at higher temperatures, the ~80°F room temperature is for battery backup stability and longevity. Each site is equipped with two HVAC units and each unit is sized to handle the cooling load generated by the cellular equipment. There are two units for mechanical redundancy. Per the building operator's procedure, the units are operated in a lead-lag scenario where the lead unit switches each week. The staging of the units is controlled by a lead/lag controller capable of calling for 4-stages of cooling. Although each unit is capable of providing enough cooling, the operator stages in the units in a lead-lag-lead-lag sequence in order to take advantage of running both units in an economizer mode when the outside air is below the set point of 70°F; this staging sequence also results in more consistent run times between units. The cellular sites often do not have potable water connections which posed an issue for testing indirect evaporative cooling. IECs need access to a supply of water and appropriate drainage to function properly. The second consideration is the static pressure of the equipment space. Before the addition of the IEC's, the HVAC equipment normally ran in recirculation only, meaning that there was no net static pressure increase inside the space. To account for the IEC's using 100% outside air, barometric relief dampers were installed.

PLACENTIA, CA

The Placentia site, (Figure 2), is located inside a large storage/commercial space. The total space is roughly 2,500 sq. ft. with a 20-foot ceiling. Tenant improvements have been made so that several enclosed rooms and an office take up ~ 80% of the space. All of the cellular equipment is located in a 250 sq. ft. conditioned room with a drop ceiling in the back half of the space (Figure 3). The current HVAC equipment is located on the roof (Figure 4). Each HVAC unit has a supply and return penetration through the roof with insulated flex ducting runs from the respective plenums to the registers in the drop ceiling. There was previously an office in the space so there was access to domestic water and sewage lines in which the IEC can connect.

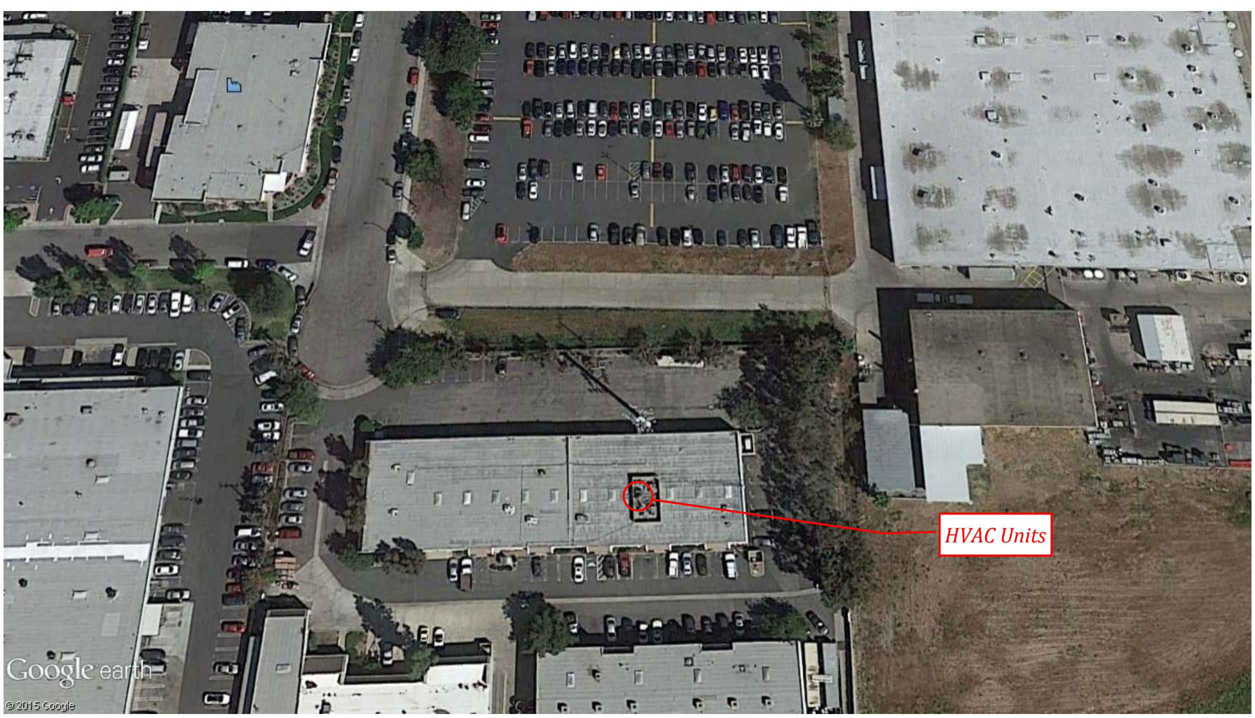


FIGURE 2. LAYOUT OF ROOFTOP UNIT AT PLACENTIA (GOOGLE 2015)

FIGURE 3. VIEW FROM ABOVE THE DROP CEILING OF CELL SITE ROOM AT PLACENTIA

FIGURE 4. RTUS ON THE PLACENTA ROOFTOP

CUDAHY, CA

The Cudahy site, (Figure 5), is located on a small plot of land shared by the local water utility and a small office building. This site consists of a Fibrebond shelter with two wall pack units mounted on the side (Figure 6). The walls of the shelter consist of approximately 2” on concrete material and 6” of insulation. Prior to the IEC installation the only penetrations in the shelter were for the cellular antennae, the door, HVAC electrical runs, and the supply and return register. The internal area is less than 150 sq. ft. This type of installation is much more typical than the installation at Placentia. Water was available on site, but had to be plumbed to the IEC unit. Also, the mechanical contractor installed a French drain to allow for appropriate drainage of water from the IEC unit.

FIGURE 5. LAYOUT OF CUDAHY SITE (GOOGLE 2015)

FIGURE 6. WALL PACK UNITS ON FIBERBOND STRUCTURE AT CUDAHY

TABLE 3. BASELINE EQUIPMENT AND RETROFIT UNITS AT CUDAHY AND PLACENTIA

EQUIPMENT TAG	TYPE	AHRI TOTAL NET COOLING CAPACITY (KBTU/HR)	DESIGN SUPPLY AIRFLOW (CFM)	STAGES OF COOLING	V/Ø/HZ
AC1-C	Wall-Mounted Packaged AC	56.5	850 - 1700	2	230/1/60
AC2-C	Wall-Mounted Packaged AC	56.5	850 - 1700	2	230/1/60
IEC-C	-	N/A	1700	Variable	480/3/60
AC1-P	Roof-Mounted Packaged AC	34.6	1500	1	230/3/60
AC2-P	Roof-Mounted Packaged AC	34.6	1500	1	230/3/60
IEC-P	-	N/A	2300	Variable	480/3/60

OPERATING MODES & SEQUENCE OF OPERATIONS

The baseline equipment at both the Cudahy and Placentia sites can operate in one of following five modes:

- **Fan-only:** From the standard operating procedure, the lead unit will run the supply fan continuously with the outside air damper closed to increase mixing and reduce stratification of the air within the space.
- **Economizer:** The unit will run with the fan on and the outside air damper fully open if the outside air temperature is below the economizer changeover set point of 70°F. The standard operating procedure is if one economizer is not able to satisfy the call for cooling, the second stage call for cooling will activate the economizer for the lag unit.
- **Economizer + Direct Expansion (DX):** The unit will run with the outside air damper open and the compressor on. If economizers running simultaneously on both units are unable to satisfy the second stage cooling call, the third stage will active the DX system on the lead unit.
- **DX:**The unit will run with the outside air damper closed and the compressor on.
- **Heating:** The unit will run with the outside air damper closed and the heater on. No instances of heating were recorded during this field test.

Both IECs have variable speed fans and were controlled by the manufactures thermostat. Both manufacturers have their own control algorithms to step the fan through various speeds based on the set point temperature and the current room temperature. For clarity, IEC operation will be simplified into the following two modes:

- **Full-speed:** Anytime the fan is operating within 10% of full-speed.
- **Part-speed:** Anytime the fan is on and is not operating within 10% of full-speed.

Typically, when retrofitting or adding an IEC to an existing system a controller capable of controlling all of the equipment is used. However, in this field test, the cellular sites had to remain active and it was decided to not integrate the control due to the possibility of affecting the cellular site performance. Instead, the research team developed a sequence of operations based on the room temperature (Figure 7). Before the field test, the cell sites operated with a set point of 78°F with an additional set point two degrees higher for each successive stage of cooling. The goal after the addition of the IEC was to set the thermostat to be 3°F below the baseline units to act as the first stage of cooling. However, it was discovered during the commissioning phase that the IEC control algorithm will not ramp the fan up quickly enough so the baseline units turn on very quickly after the IEC. After discussion with the company responsible for the cellular site, the research team shifted the set point for the baseline units up to 80°F with the IEC set point remaining at 75°F.

While designing the sequence of operations the fact that the IEC has a larger fan and therefore uses more energy than the baseline units running in economizer mode was considered. Because the cellular sites cooling load do not trend with outside air temperature, it was recognized that there can be a significant number of hours in Climate Zone 8 where it would be more efficient to run the economizers than the IEC. To achieve this, the research team used a second thermostat to lock out the IEC below an outside air temperature of 65°F.

The resulting sequence of operations is illustrated in Figure 7. Below 65°F there is no IEC operation. All cooling is provided by the baseline units once the room temperature goes above 80°F. Above 65°F, the IEC will activate and try to keep the room at 75°F. If the room temperature reaches 80°F, the IEC will continue to run, but the lead baseline unit will turn on. The sequence of operations is also explained in tabulated form in Table 4.

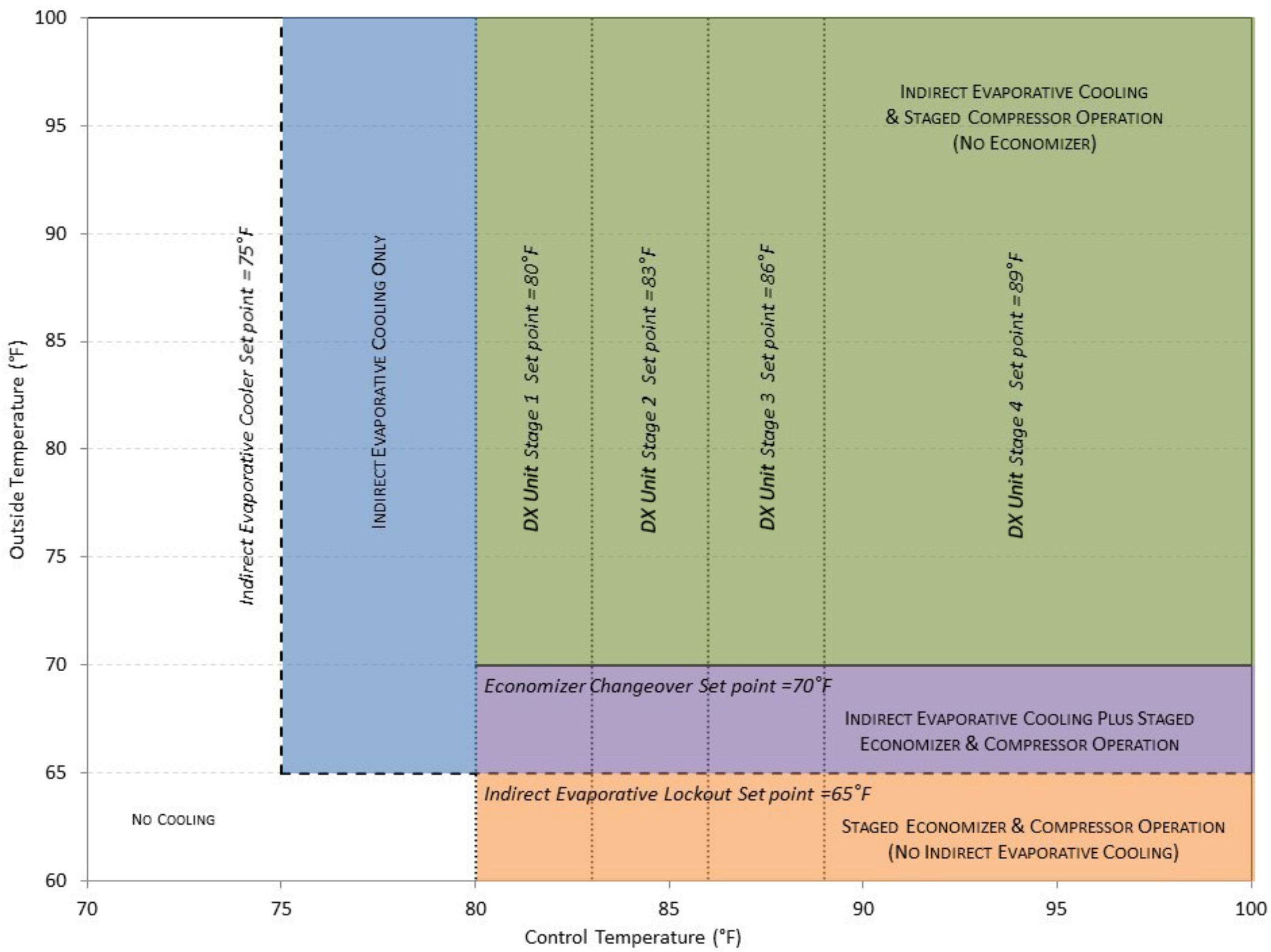


Figure 7. Sequence of Operation Schematic for Cudahy and Placentia

Table 4. Definition of each operating mode for Cudahy and Placentia

IEC Setpoint = 75°F ACs Setpoint = 80°F	A Control Temperature (°F)	Outside Temperature (°F)	IEC Cooling	IEC Fan Speed	RTU OSA Damper	RTU Fan Speed	Compressor 1	Compressor 2
OFF	0	NA	OFF	OFF	CLOSED	OFF	OFF	OFF
Economizer	>80	<70	OFF	OFF	OPEN	ON	OFF	OFF
Economizer & DX1	>80	<70	OFF	OFF	OPEN	ON	ON	OFF
Economizer & DX2	>80	<70	OFF	OFF	OPEN	ON	ON	ON
IEC (0-100% Speed)	75-80	>70	ON	0-100%	CLOSED	OFF	OFF	OFF
IEC & DX1	>80	>70	ON	100%	CLOSED	ON	ON	OFF
IEC & DX2	>80	>70	ON	100%	CLOSED	ON	ON	ON

ASSESSMENT OBJECTIVES

The primary objective of this study is to conduct field evaluations of the IECs in California Climate Zone 8. The evaluation studied real-world equipment operation at two different field sites and developed characterizations of the overall system performance and energy efficiency across a range of operating conditions.

The study is designed to investigate performance characteristics that cannot be captured by steady-state laboratory testing. This evaluation carefully disaggregates performance in each mode of operation to consider the performance and efficiency of each system state, and to investigate the implications of the control strategies and field-selected settings that were applied.

This study also applied a day-on, day-off control scheme where the IEC was locked out every other day to capture the performance of the baseline equipment versus the retrofit performance. This assessment is not an annual projection of savings but is only a comparison of characteristic performance for the baseline equipment versus the IEC addition. The results present a clear and reliable description of system performance for real-world operation of the IEC equipment.

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

ASSESSMENT METHODOLOGY

OVERVIEW OF THE TECHNICAL APPROACH

IEC-C and IEC-P were installed in June 2014 and July 2014, respectively. During the installations the project team was on site installing a thorough array of instrumentation on each of the three systems. The installing contractor commissioned IEC-C and IEC-P in June 2014, and August 2014, respectively.

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

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

In addition to the array of instrumentation for minute-by-minute performance monitoring, the research team also conducted a number of in-field diagnostic measurements in order to build calibrated maps for particular operating variables. These in-field measurements were used to supplement the minute-by-minute data with information that is not easily measured continuously, but is required to calculate meaningful performance characteristics such as cooling capacity and coefficient of performance. Airflow rates were measured using a tracer gas airflow technique described in more detail in the section Airflow Measurements

MONITORING PLAN

The research team developed a monitoring plan that allowed for:

- Assessment of overall performance for system inputs and outputs;
- Evaluation of sub component performance characteristics: and
- Consideration of dynamic equipment operating behaviors.

The monitoring scheme utilized for the study is illustrated schematically in Figure 8. Table 5 provides a simple description of each measurement marked in the instrumentation schematic, and documents the performance specifications for the sensors utilized for each corresponding measurement.

The current transducers listed in the monitoring plan are used mainly for sensing component operations to determine system mode. The system amperage, line voltage, and power factor are recorded to accurately determine the total power draw for each minute of operation. The water usage is measured to determine the total amount of water used for cooling and water quality management. The analog output channels noted represent the dc-voltage used to control the outdoor air damper position for the ACs and the speed of the IEC fan. The pitot tubes are used to calculate airflow by correlating the pressure measurement to a field measured airflow map. Finally, the temperature arrays are used to monitor the air temperature around the equipment within the cell site.

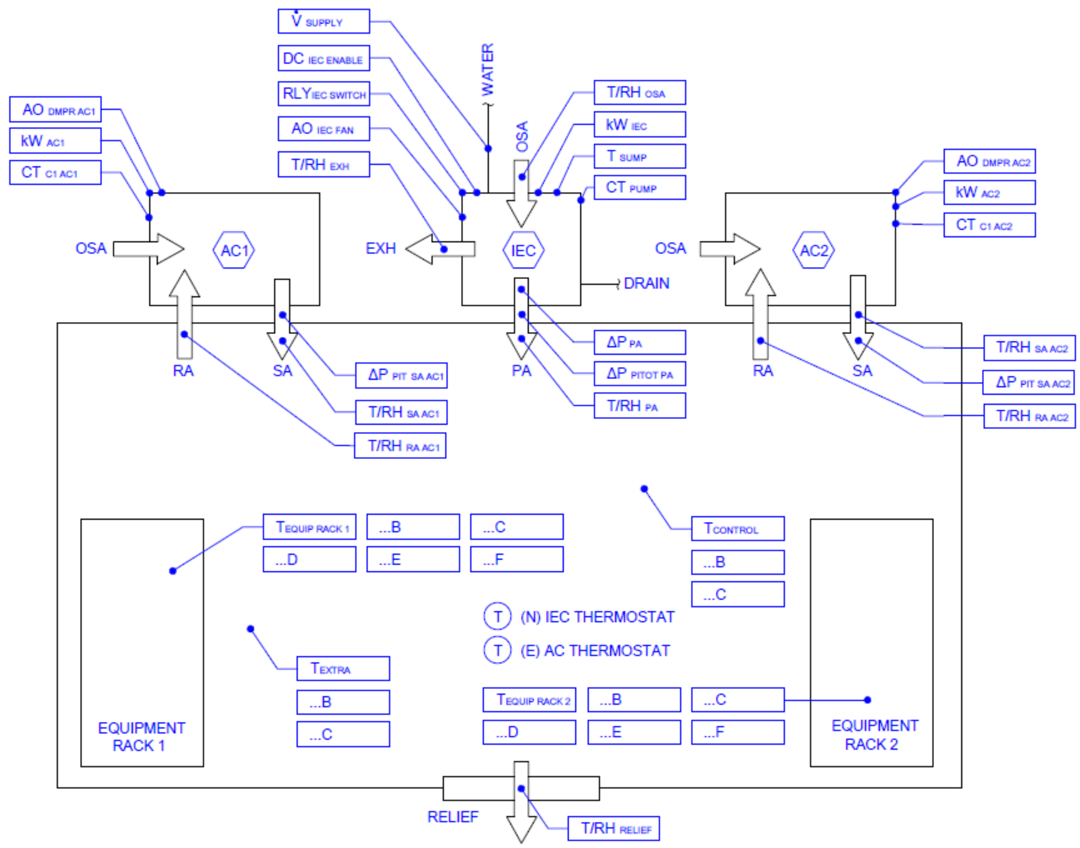


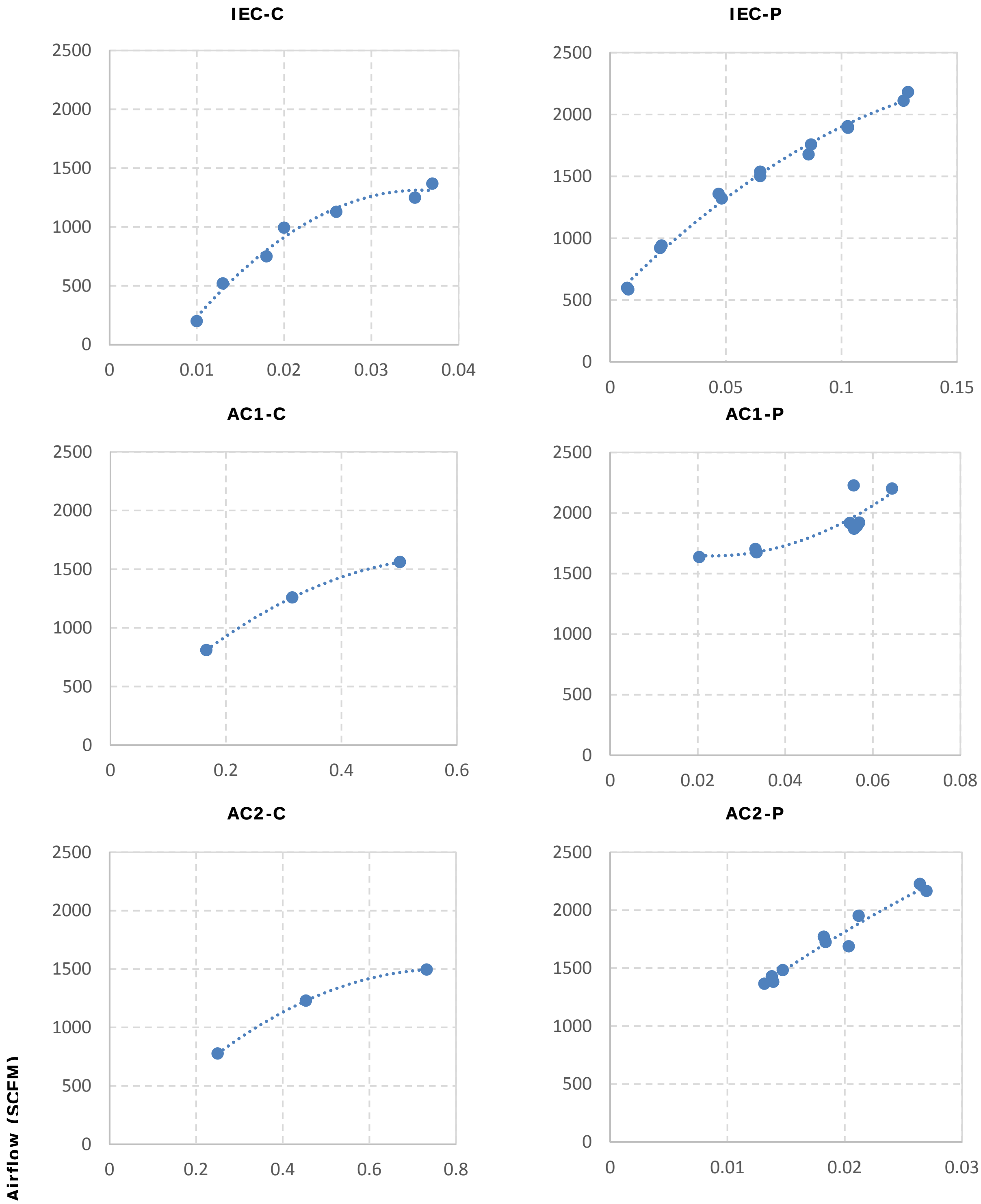
FIGURE 8. INSTRUMENTATION SCHEMATIC

TABLE 5. INSTRUMENTATION TABLE

Name	Measurement	Sensor	Uncertainty	Unit install on
T_{OSA}	Temperature – Outside Air	Vaisala HUMICAP HMP110	± 0.36°F	AC 1
RH_{OSA}	Relative Humidity – Outside Air	Vaisala HUMICAP HMP110	± 1.7% RH	AC 1
T_{RA}	Temperature – Return Air	Vaisala HUMICAP HMP110	± 0.36°F	Both ACs
RH_{RA}	Relative Humidity – Return Air	Vaisala HUMICAP HMP110	± 1.7% RH	Both ACs
T_{SA}	Temperature – Supply Air	Vaisala HUMICAP HMP110	± 0.36°F	Both ACs
RH_{SA}	Relative Humidity – Supply Air	Vaisala HUMICAP HMP110	± 1.7% RH	Both ACs
T_{PA}	Temperature- IEC Product Air	Vaisala HUMICAP HMP110	± 0.36°F	IEC only
RH_{PA}	Relative Humidity- IEC Product Air	Vaisala HUMICAP HMP110	± 1.7% RH	IEC only
T_{EXH}	Temperature- IEC Exhaust Air	Vaisala HUMICAP HMP110	± 0.36°F	IEC only
RH_{EXH}	Relative Humidity- IEC Exhaust Air	Vaisala HUMICAP HMP110	± 1.7% RH	IEC only
$\Delta P_{PITOT,\ SA}$	Pitot Tube- Supply Air	Dwyer	± 0.0025 "WC	Both ACs
$\Delta P_{PITOT,\ PA}$	Pitot Tube- IEC Product Air	Dwyer	± 0.0025 "WC	IEC only
ΔP_{PA}	Pressure- IEC Product Air	Dwyer	± 0.0025 "WC	IEC only
$\dot{V}_{SUPPLY}$	IEC Water Consumption	OMEGA FTB 4105 A P	± 1.5%	IEC only
AO_{DMPR}	Analog Output- Damper Position	OA Damper control signal	± 0.1V	Both ACs
$AO_{IEC\ FAN}$	Analog Output- IEC Fan Speed	Analog output	± 0.1V	IEC only
CT_{C1}	AC Current – Compressor	NK AT1-005-000-SP	± 0.1 A	Both ACs
CT_{PUMP}	Current Transducer- IEC Pump Status	NK AT1-005-000-SP	± 0.1 A	IEC-P only
T_{SUMP}	Temperature- IEC Water Temperature	Omega Thermistor	± 0.36°F	IEC only
kW	System Power Draw, Voltage, Current & PF	Dent Powerscout 3	± 1%	All 3
$T_{EQUIP\ RACK\ 1}$ (6)	Temperature array- Cell Equipment	Omega Thermistor	± 0.36°F	N/A
$T_{EQUIP\ RACK\ 2}$ (6)	Temperature array- Cell Equipment	Omega Thermistor	± 0.36°F	N/A
T_{RELIEF} (3)	Temperature array- Relief Damper	Omega Thermistor	± 0.36°F	N/A
$T_{CONTROL}$ x3	Temperature array- Thermostat	Omega Thermistor	± 0.36 °F	N/A

AIRFLOW MEASUREMENTS

All the field measured airflows were mapped against the differential pressure measurement from the pitot tube in the supply air duct (



Differential Pressure (in WC) **Differential Pressure (in WC)**

Figure 9). For IEC-C and IEC-P, the map was developed by stepping the variable speed fan through the various fan speeds. It should be noted that although it is a variable speed fan, the control sequence for the unit makes discrete steps based on information received from the manufacturers thermostat. AC1-C and AC2-C have two fan speeds; one for fan-only and the second for all other modes of operation. The airflow measurements were made with the damper closed at both fan speeds and with the damper open at the higher fan speed. AC1-P and AC2-P have constant speed fans. To account for changes in airflow resistance over the course of the field test, the research team measured airflow in the normal operating modes with the damper fully open or closed and with adding or removing resistance from the system. Less resistance was achieved by removing the outside air damper and resistance was added by taping off return air registers. The physical location of the installed units and the lack of ducting caused the research team to use a different method for measuring airflow at each field site. The method used at each site is describe in the following sections.

PLACENTIA

Supply airflow rates at Placentia were determined using a tracer gas airflow measurement, conducted according to ASTM E2029 *Standard Test Method for Volumetric and Mass Flow Rate Measurement in a Duct Using Tracer Gas Dilution* (ASTM 2011). This method mixes a measured mass flow rate of CO_2 into the supply air stream and then measures the corresponding rise in CO_2 concentration downstream. The volume flow of air into which the tracer gas is mixed can be calculated by the formula in Equation 1.

EQUATION 1. VOLUME AIR FLOW

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

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

Comparing the results with the nominal airflow listed in Table 3 the AC units are moving roughly 100 – 200 standard cubic feet per minute (SCFM) more than their nominal fan rating and the IEC at full-speed is roughly 100 SCFM under the manufacturer's rating.

A similar method was used to measure the outside air fraction. The CO_2 concentration in the room was increased above normal outdoor air concentrations. Then, while operating in each mode and fan speed scenario, CO_2 concentration was measured in the outside air and return air streams, and the resultant concentration of their mixture was measured in the supply air stream. The ratio of outside air to the total supply airflow can be determined according to a conservation of mass. The mass balance calculations can be reduced to the following equation as shown in Equation 2.

EQUATION 2. OUTSIDE AIRFLOW FRACTION

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

It is important to note that the outdoor airflow fraction analysis includes damper and cabinet leakage. By design it is expected that the OSAF fraction for the AC units be 0% when the damper is closed and 100% when the damper is open. However, the results show an outdoor air fraction of 6% and 11% when the damper is closed and 89% and 92% when the damper is open (Table 6).

CUDAHY

At Cudahy, the research team was able to complete the OSAF measurement using the same method described in Placentia, however, the tracer gas method could not be used to measure the airflow. The lack of ducting for the wall pack units and the IEC prevented CO_2 injection and mixing. Therefore, the team used a flow hood with an uncertainty of ± 3% to measure airflow. For the IEC, the primary airflow was calculated by the difference of the measured airflow between the inlet of the fan and the wet channel exhaust (Equation 3).

EQUATION 3. PRIMARY AIRFLOW

$$\dot{V}_{PA} = \dot{V}_{Fan\ Inlet} - \dot{V}_{EXH}$$

Though the IEC physically has two exhaust ports, the installation position only allowed for one to be measured with the flow hood. The flow through the exhaust ports should be equal and will only change slightly due to differencing resistance of the IEC heat exchanger media. The research team encountered similar issues with both of the baseline units as well. For AC1-C the supply air register was blocked by equipment inside the space. The supply airflow was calculated by measuring the air flowing through the return register and outside air inlet during all modes of operation (Equation 4).

EQUATION 4. SUPPLY AIRFLOW

$$\dot{V}_{SA} = \dot{V}_{OSA} + \dot{V}_{RA}$$

For AC2-C both the supply and return registers were blocked by equipment. The airflow was calculated by measuring the outside air flow in each mode and using the outside air fraction measurement to determine the supply airflow for each mode (Equation 5).

EQUATION 5. SUPPLY AIRFLOW FRACTION

$$\dot{V}_{SA} = \frac{\dot{V}_{OSA}}{OSAF}$$

Although different methods were used for measuring the AC1-C and AC2-C airflow, the measured airflows differed by roughly 5%, which can be expected from manufacturing and installation differences (

A ·	IEC-C	IEC-P

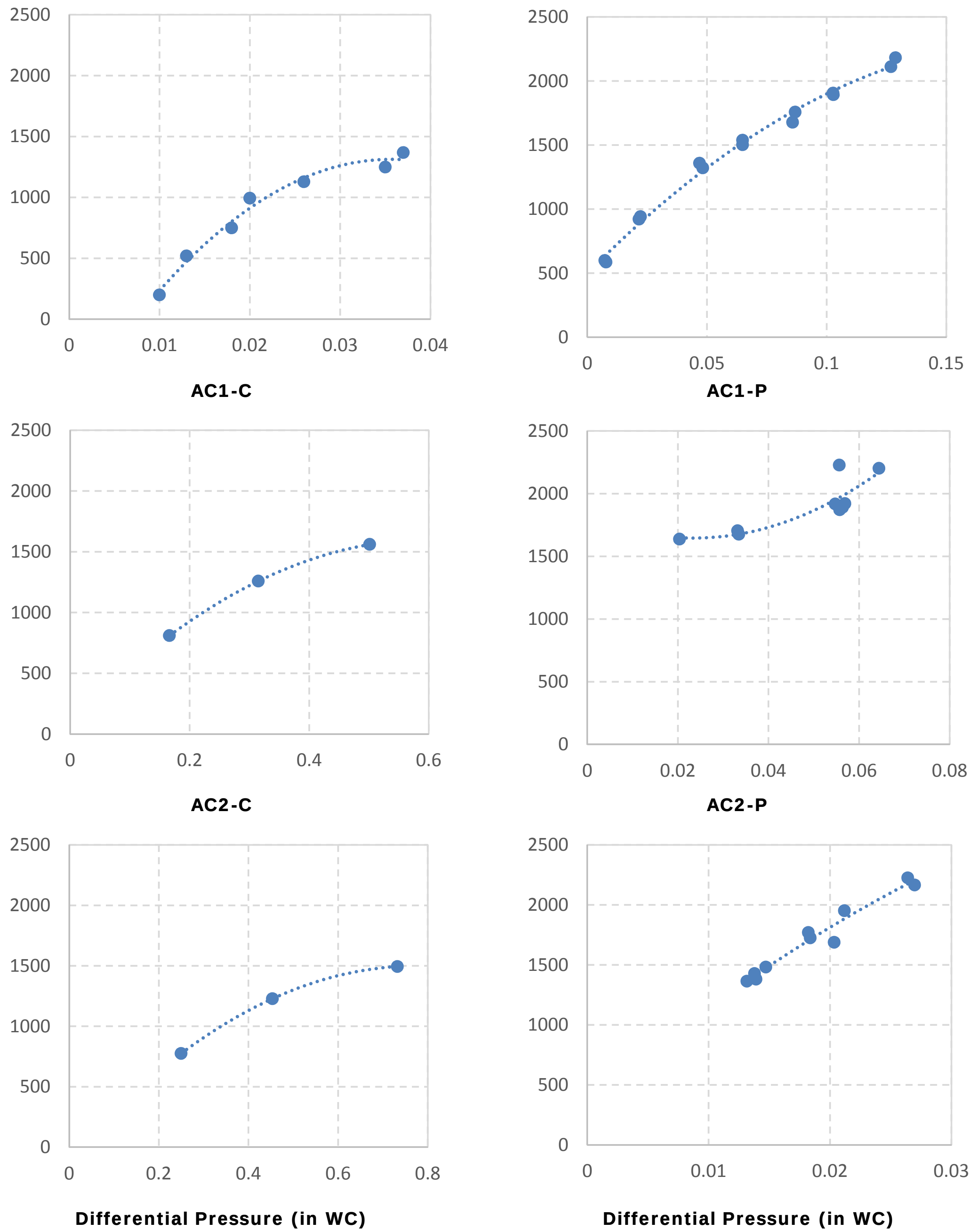


Figure 9). Comparing measured results to the manufacturer specifications in Table 3 show that the IEC-C is moving approximately 300 SCFM less than expected, while AC1-C and AC2-C match the manufacturer rating on the lower fan speed while delivering 100 SCFM less than expected at the highest fan speed.

It is important to note that the outdoor airflow fraction analysis includes damper and cabinet leakage. By design it is expected that the OSAF fraction for the AC units will be 0% when the damper is closed and 100% when the damper is open. However, the results show an outdoor air fraction of 13-14% when the damper is closed and 91% when the damper is open (Table 6).

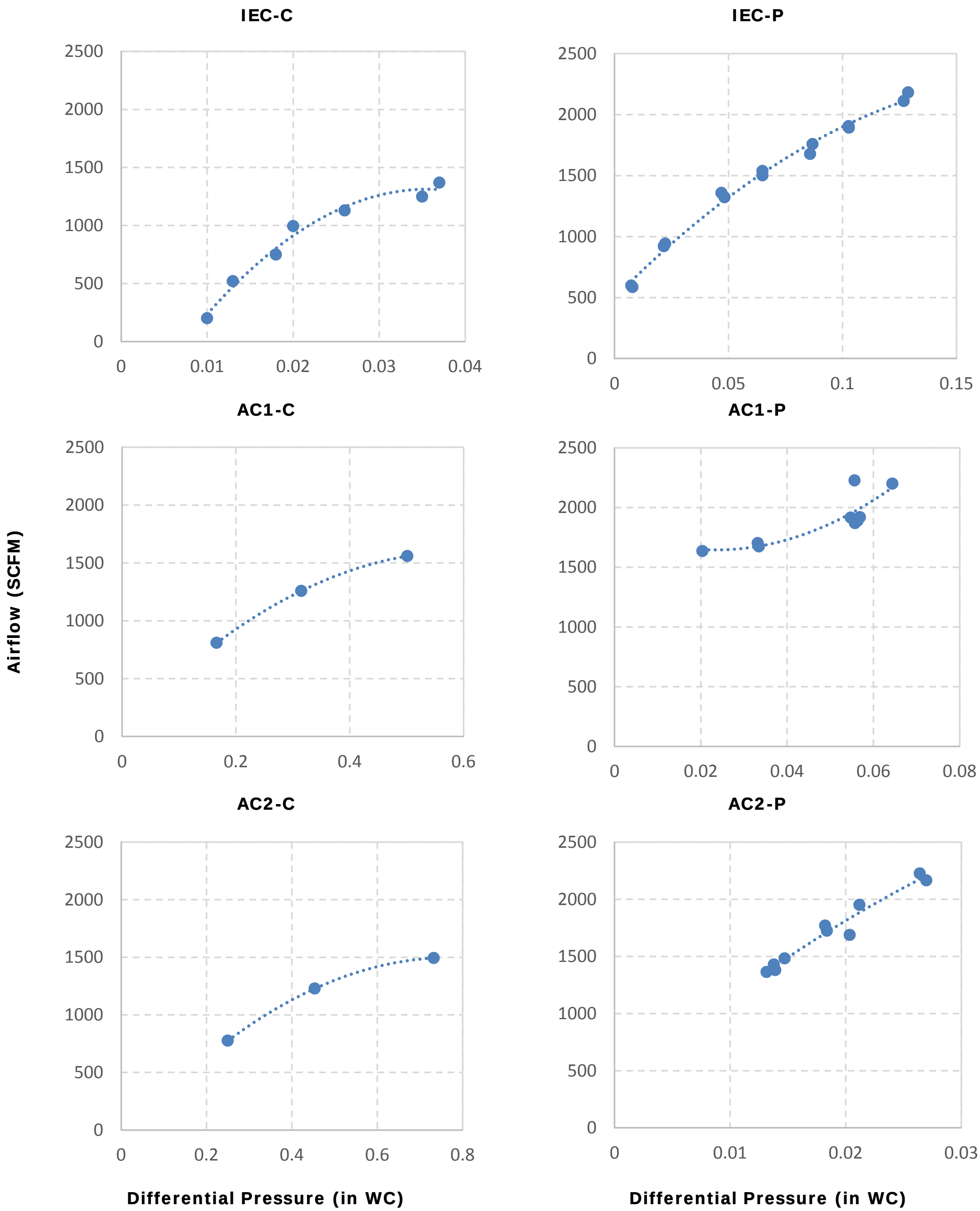


FIGURE 9. SUPPLY AIRFLOW AS FUNCTION OF DIFFERENTIAL PRESSURE AT CUDAHY AND PLACENTIA

TABLE 6. OUTSIDE AIR FRACTIONS FOR CUDAHY AND PLACENTIA

OUTSIDE AIR DAMPER POSITION	AC1-C	AC2-C	IEC-C	IEC-P	AC1-P	AC2-P
Closed	13%	14%	IEC units use 100% Outside Air	6%	11%	Closed
Open	91%	91%		89%	92%	Open

DETERMINATION OF OPERATING MODE

The performance of each system changes with the mode of operation. Therefore, it is important to discretize the results by the separate modes so that the observations may be analyzed and explained clearly. The operating mode for each one-minute interval recorded was determined by the state of the variables used as indicators for the component operation presented in Figure 4.

CUMULATIVE VERSUS STEADY-STATE PERFORMANCE ANALYSIS

Two types of performance results are presented in this report: cumulative performance metrics, and steady-state performance metrics. Cumulative performance metrics include the total capacity delivered and the total water used for the piece of cooling equipment summed over the entire day. Steady-state performance, such as capacity or efficiency, is calculated over a short period of time where the outdoor air and return air conditions are assumed to be constant. In order to reduce noise and present a clearer visual interpretation of the results, these plots discard observations from the first several minutes of operation in any mode. Normally, the research team will discard the first ten minutes of operation within a mode. However, in this field study the cell sites are small and some of the baseline equipment does not operate for more than ten minutes at a time. For example, the equipment at the Cudahy site consistently runs in a cooling mode for only three minutes (which is the minimum runtime for the compressor). To capture steady-state behavior, the research team used two steady-state thresholds. For Cudahy, the baseline threshold was three minutes and the IEC threshold was five. For Placentia, all thresholds were five minutes.

DATA CONFIDENCE

The measurement accuracy for each instrument used for field monitoring is recorded as part of the monitoring plan in Table 5. Table 7 and Table 8 summarize the degree of confidence for the key calculated metrics for Cudahy and Placentia. These values are calculated by propagation of uncertainty at a single operating condition. The values recorded indicate the uncertainty resulting from manufacturer-stated performance for the sensors used and from the equations documented in section “Definition and Calculation of Performance Metrics”. The values in Table 7 and Table 8 do not account for any methodological uncertainty associated with features such as sensor placement or sensor response time.

TABLE 7. UNCERTAINTY FOR KEY CALCULATED METRICS AT CUDAHY[1]

METRIC	UNCERTAINTY
IEC-C Supply Airflow Rate	±80 SCFM
AC1-C Supply Airflow Rate	±33.6 SCFM
AC2-C Supply Airflow Rate	±50 SCFM
ACs OSAF	±5%
Absolute Humidity	±0.00062 lbm, water / lbm, dry air
Sensible System Capacity	±0.89 kBTU/hr
Sensible Room Capacity	±0.56 kBTU/hr
Sensible System COP	±0.48
Sensible Room COP	±0.28
Water Use	±0.3 gal/ton hr
Wet Bulb Effectiveness	±0.03

TABLE 8. UNCERTAINTY FOR KEY CALCULATED METRICS AT PLACENTIA[2]

METRIC	UNCERTAINTY
IEC-P Supply Airflow Rate	±34 SCFM
ACs Supply Airflow Rate	±37 SCFM
ACs OSAF	±6%
Absolute Humidity	±0.00063 lbm, water / lbm, dry air
Sensible System Capacity	±1.0 kBTU/hr
Sensible Room Capacity	±0.71 kBTU/hr
Sensible System COP	±0.21
Sensible Room COP	±0.13
Water Use	±0.3 gal/ton hr
Wet Bulb Effectiveness	±0.04
IEC-P Supply Airflow Rate	±34 SCFM

[1] Uncertainty for each metric is calculated for the following conditions: $T_{DB\ OSA}$=95°F, $T_{WB\ OSA}$=70°F, $T_{DB\ RA}$=80°F, RH_{RA}=44%, $T_{DB\ SA,\ AC}$=72°F, $RH_{SA,\ AC}$=58%, $T_{DB,\ PA}$ = 72°F, IEC-C Airflow Rate = 1370 SCFM, AC1-C Airflow Rate = 1260 SCFM, AC2-C Airflow Rate = 777 SCFM.

[2] Uncertainty for each metric is calculated for the following conditions: $T_{DB\ OSA}$=96°F, $T_{WB\ OSA}$=73°F, $T_{DB\ RA,\ AC}$=76°F, $RH_{RA,\ AC}$=57%, $T_{DB\ SA,\ AC}$=66°F, $RH_{SA,\ AC}$=76%, $T_{DB,\ PA}$ = 70°F, IEC-C Airflow Rate = 1823 SCFM, ACs Airflow Rate = 1880 SCFM.

DEFINITION AND CALCULATION OF PERFORMANCE METRICS

CALCULATING COOLING CAPACITY

The system-cooling-capacity for the IEC equipment is determined at any operating condition according to the supply air-flow rate and the specific enthalpy difference between the outside air and the supply air stream, as described by Equation 6. This is the net cooling produced by the equipment and includes what is lost due to fan heat.

EQUATION 6. NET COOLING CAPACITY LESS FAN HEAT

$$\dot{H}_{system} = \dot{m}_{SA} \cdot (h_{OSA} - h_{SA})$$

The system-cooling-capacity for the baseline equipment is determined at any operating condition according to the supply air-flow rate and the specific enthalpy difference between the combined air streams entering the system and the supply air stream, as described by Equation **Error! Reference source not found.**. This is the net cooling produced by the equipment.

EQUATION 7. NET COOLING CAPACITY OF EQUIPMENT

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

where h^*_{MA} is the specific enthalpy of the 'virtual' mixed-air, a parameter that does not physically exist. Generally, the system cooling capacity for a conventional rooftop unit is measured by the difference between the mixed air enthalpy and the supply air enthalpy. The 'virtual' mixed-air condition represents the combined enthalpy from all inlets to the equipment, and allows for accounting for leakage across the baseline units. Equation 8 calculates the specific enthalpy for the 'virtual' mixed air condition.

EQUATION 8. SPECIFIC ENTHALPY FOR VIRTUAL MIXED AIR

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

The room-cooling capacity, given by Equation 9, is the cooling that is actually of service to the zone. In the case when outside air is cooler than return air, room-cooling may be greater than the system cooling. (This should occur in any economizer mode.)

EQUATION 9. ROOM COOLING CAPACITY

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

Since ambient humidity in most western climates is low enough that dehumidification is not necessary to maintain occupant comfort in most commercial buildings, the assessment presented here focuses on the system's ability to produce sensible cooling (ASHRAE 2010). Furthermore, since thermostat controls for this application only respond to temperature and do not control for humidity, it is not appropriate to credit any latent cooling when considering of the performance comparison for the IEC systems studied here. The net sensible system cooling capacity is determined according to Equation 10.

EQUATION 10. NET SENSIBLE SYSTEM COOLING CAPACITY

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

Equation 11 shows how the latent system cooling is determined.

EQUATION 11. LATENT SYSTEM COOLING

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

CALCULATING COEFFICIENT OF PERFORMANCE

Energy efficiency at any given operating condition is expressed as the dimensionless ratio of useful thermal capacity delivered to electrical power consumed by the system – the Coefficient of Performance (Equation 12).

EQUATION 12. COEFFICIENT OF PERFORMANCE

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

Analysis in this report focuses on the sensible cooling generated by the equipment. This metric discounts the enthalpy associated with reduced humidity. The Sensible Coefficient of Performance can be expressed as shown in Equation 13.

EQUATION 13. COEFFICIENT OF PERFORMANCE - SENSIBLE

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

Further, performance results are described both in terms of the Sensible System COP, and the Sensible Room COP. The first metric (Equation 14) considers the ratio of electricity consumed to the sum of sensible cooling generated by the machine. The second metric (Equation 15) compares the electricity consumed to the sensible cooling effect on the room.

EQUATION 14. COEFFICIENT OF PERFORMANCE – SENSIBLE SYSTEM

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

EQUATION 15. COEFFICIENT OF PERFORMANCE – SENSIBLE ROOM

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

CALCULATING WET-BULB EFFECTIVENESS FOR INDIRECT EVAPORATIVE COOLING

The wet-bulb effectiveness for the indirect evaporative cooling of the ventilation air is calculated according to Equation 16. This metric represents the degree to which ventilation air is cooled toward the outside air wet bulb temperature. It is calculated as the ratio of the change in ventilation-air dry-bulb temperature across the indirect evaporative cooler, to the wet-bulb depression of the outside air.

EQUATION 16. WET-BULB EFFECTIVENESS FOR INDIRECT EVAPORATIVE COOLING

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

CALCULATING 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 (Equation 17).

EQUATION 17. WATER USE EFFICIENCY

$$WUE = \frac{\dot{V}_{water}}{\dot{H}_{Sensible}}$$

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

RESULTS AND DISCUSSION

The monitoring for IEC-C and IEC-P was conducted over a nine-month period from September 2014 to May 2015. During those nine months, September 2014 was the only month where both field sites experienced typical cooling season conditions with average outside air temperatures above 75°F. Thus, the data presented in this report is from September 2014 only and provides the clearest characteristic performance for IEC-C and IEC-P.

The results focus on sensible cooling only. The company running the cell site has alarms for high humidity situations, but none of the HVAC equipment is set up to control the humidity in the space. The characteristic metrics for all systems include, Sensible System COP, Sensible Room COP, Sensible System Capacity, Sensible Room Capacity, and System Power Draw. For the IEC units, Wet-Bulb Effectiveness, Water Use Efficiency, and Total Water-Use are discussed. Throughout the report, it is important to remember the IEC units use 100% OSA which makes it difficult to determine the true value of the room cooling metrics. Therefore, for this report the research team used the ideal room temperature of 80°F. The research team recognizes that 80°F is not always representative of the return air temperature seen by the AC units, but for comparison between the systems at each site the room metrics will only give positive room cooling credit for the difference between the supply air temperature and 80°F. Furthermore, the two AC units at each test site are used as the baseline comparison for their respective IEC units.

SENSIBLE SYSTEM COEFFICIENT OF PERFORMANCE

It is quickly noticed that IEC-C and IEC-P had a higher sensible system COP than their respective baseline units and that the IEC's COP trends upward with outside air temperature (Figure 10). At 95°F, IEC-C and IEC-P are operating with an increase in system COP of 158% and 113% compared to their respective baseline. This result is characteristic of IECs in general because of the difference between the increase in outside air temperature and outside air wet-bulb depression. At Cudahy, the COP for IEC-C at full-speed ranges from 15 – 18 while baseline units AC1-C and AC2-C have a COP between 2 – 3. At Placentia, the COP for IEC-P ranges from 8 – 17 while baseline units AC1-P and AC2-P have a COP between 2-4.

While comparing both IECs to their respective baseline units, it is noticeable that IEC-C has data in temperatures not seen on the baseline graphs. IEC-P is the opposite. This occurrence is just pure happenstance from the switching the IECs on and off on a day-to-day basis. At Placentia, the hottest days of the test were baseline days. Conversely, at Cudahy, the hottest days were retrofit days.

SENSIBLE ROOM COEFFICIENT OF PERFORMANCE

Evaluating the sensible room COP results illustrates that the IECs trend downward with outside air temperature. This is expected due to the design room temperature of 80°F. As the outside air temperature increases, the IEC is able to generate a larger temperature drop across the heat exchanger, but the primary air temperature is also increasing. Therefore, the sensible room COP trends towards zero as the primary air approaches 80°F. Although there is a downward trend, at 95°F, the sensible room COP for IEC-C and IEC-P is still higher than the baseline by 112% and 53%, respectively.

Sensible System Cooling Capacity

The system and room capacity results follow the same trends and the sensible system and room COP. However, looking at the capacities the difference between IECs and conventional DX systems is more pronounced. For Cudahy, as the outside air temperature increases, the cooling capacity provided by the baseline units is relatively constant. For Placentia, there is a noticeable downward trend for both system and room capacities. Although there is a downward trend, the baseline units are able to maintain more of their room-cooling capacity. For example, over the 80 – 110°F plotted, the room-cooling capacity for IEC-C and IEC-P decreased by 109% and 62%, respectively. For comparison, the average room capacity of the baseline equipment at Cudahy and Placentia only decreased by 42% and 16%, respectively. Also, at 95°F, at Cudahy, IEC-C delivered 14.7 kBTU/hr while the baseline equipment delivered an average of 29.5 kBTU/hr. At Placentia, IEC-P delivered 30.3 kBTU/hr while the baseline equipment delivered an average of 33.4 kBTU/hr.

System Power Draw

The steady-state power draw trends for each unit can be seen in Figure 14. Noticeable trends include the IECs having no noticeable power draw increase with outside air temperature. This is as expected with the only major component being the supply fan. IEC-C, which has a lower airflow than IEC-P, draws 600W at full-speed. The 600W for full-speed is a 140% reduction in power draw compared to the baseline unit. IEC-P has a higher power draw and maxes out at 1.8 kW. The dual bands seen in the IEC-P chart are from the water pumps turning on to re-wet the heat exchanger. In full-speed mode, the fan speed will decrease for a minute while the pumps are running. If the fan is not at full-speed the speed will not change during the wetting cycle. Furthermore, the baseline units at Cudahy have a steeper trend with outside air temperature compared to the baseline units at Placentia.

Both types of IECs require a substantially lower power draw compared the baseline units currently installed, and suggest opportunities for both peak demand and total energy savings.

Figure 14 also provides a good indication to how IECs are operated with respect to outside air temperature. At Cudahy, IEC-C only had steady-state operation in the full-speed mode above 83°F. At Placentia, IEC-P only had steady-state operation at full-speed above 80°F.

Psychometric Conditions

Figure 15 illustrates the psychometric conditions experienced during instances of operation during the field test. The IEC and baseline charts are from retrofit and baseline days, respectively. Although both sites are in California Climate Zone 8, Placentia had some instances that were hotter and more humid than Cudahy.

The Placentia graphs reveal the units cycling through the various modes as the outside air temperature increases. Conversely at Cudahy, only a large cloud of fan appears. Figure 15, shows that the equipment at Cudahy cycles back and forth between fan-only and cooling at all operating conditions. Whereas at Placentia, the equipment rarely runs in fan-only mode above 80°F and the cooling load is not satisfied by the equipment until the outside air temperature drops below 80°F.

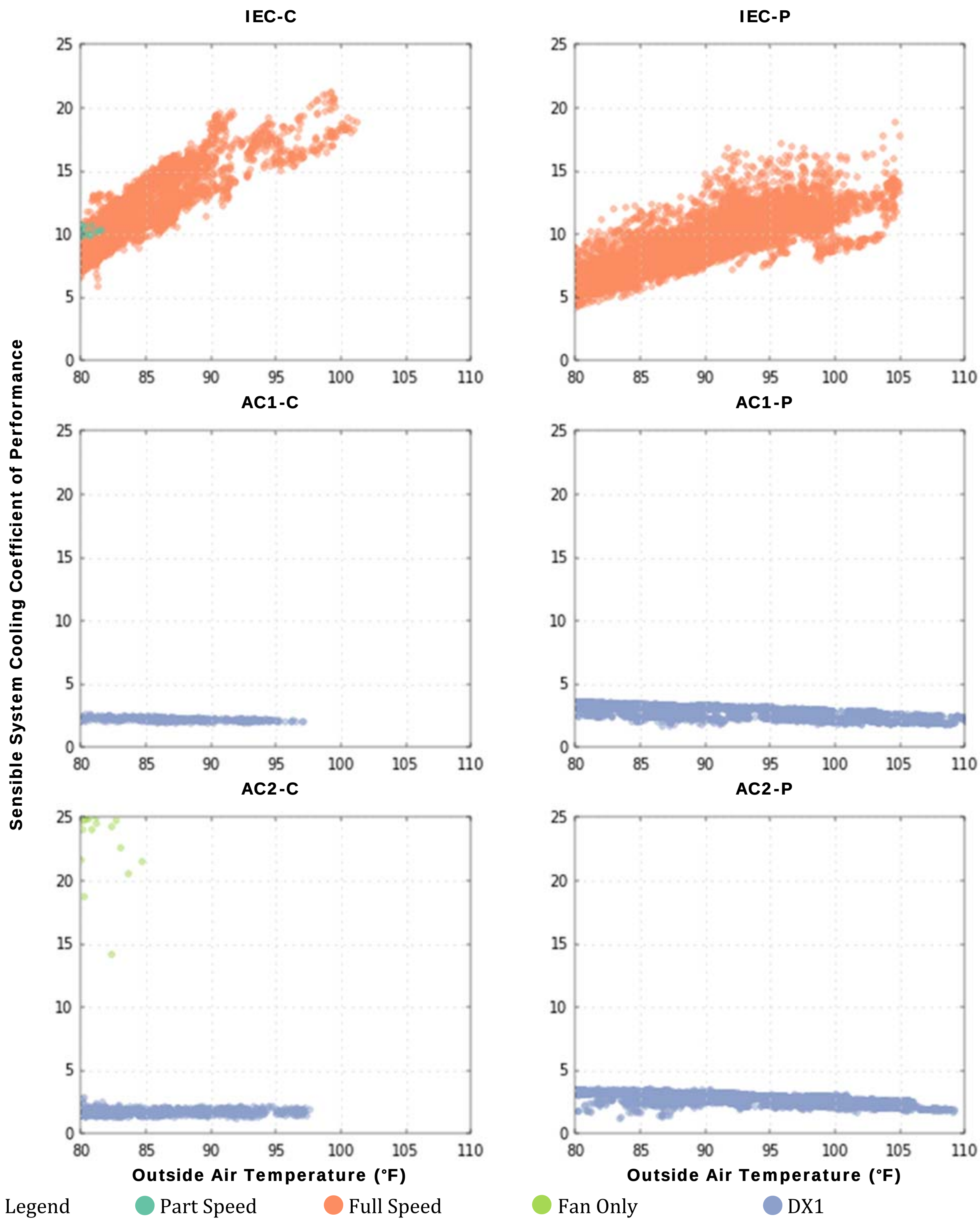


FIGURE 10. SENSIBLE SYSTEM COEFFICIENT OF PERFORMANCE

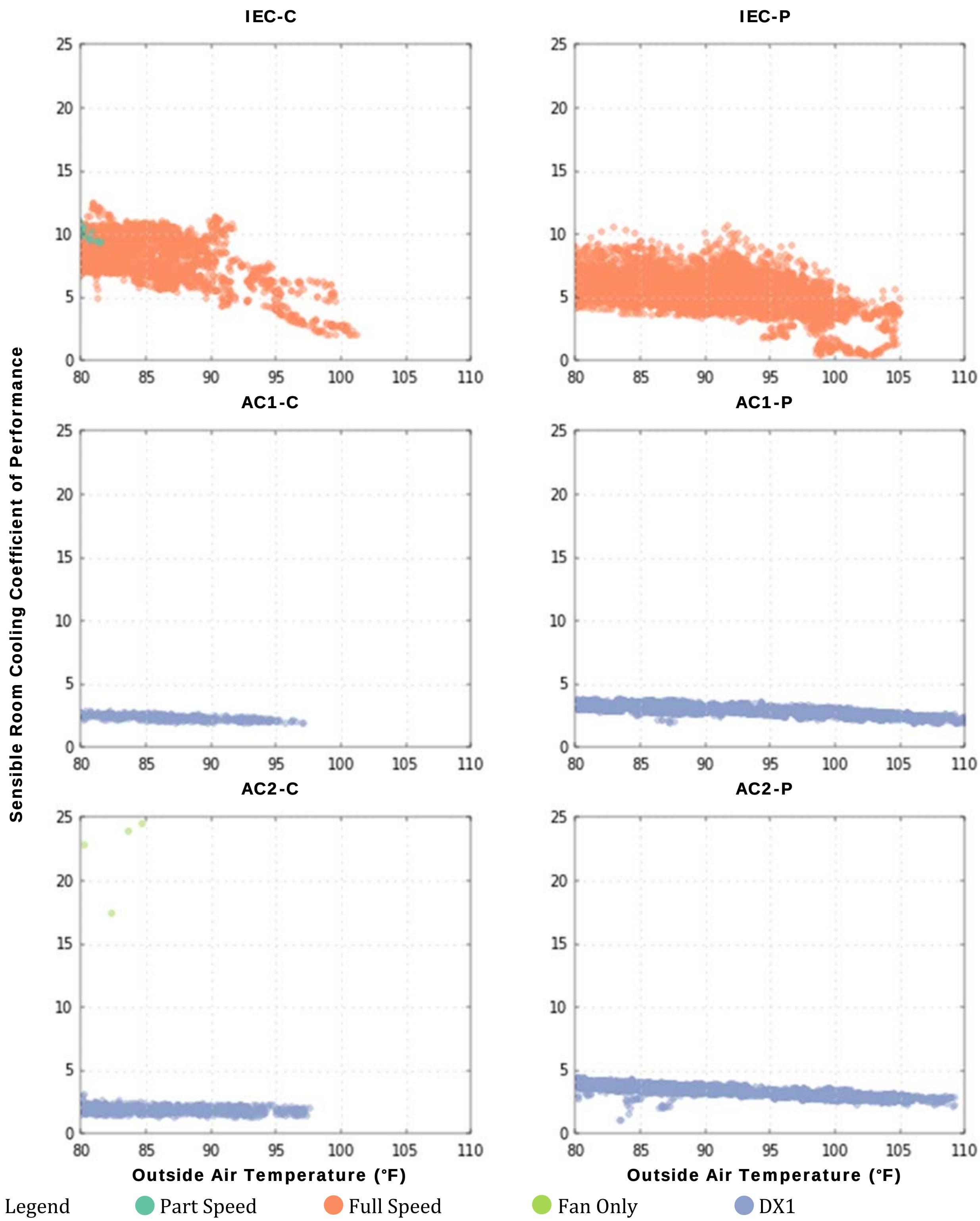


FIGURE 11. SENSIBLE ROOM COEFFICIENT OF PERFORMANCE

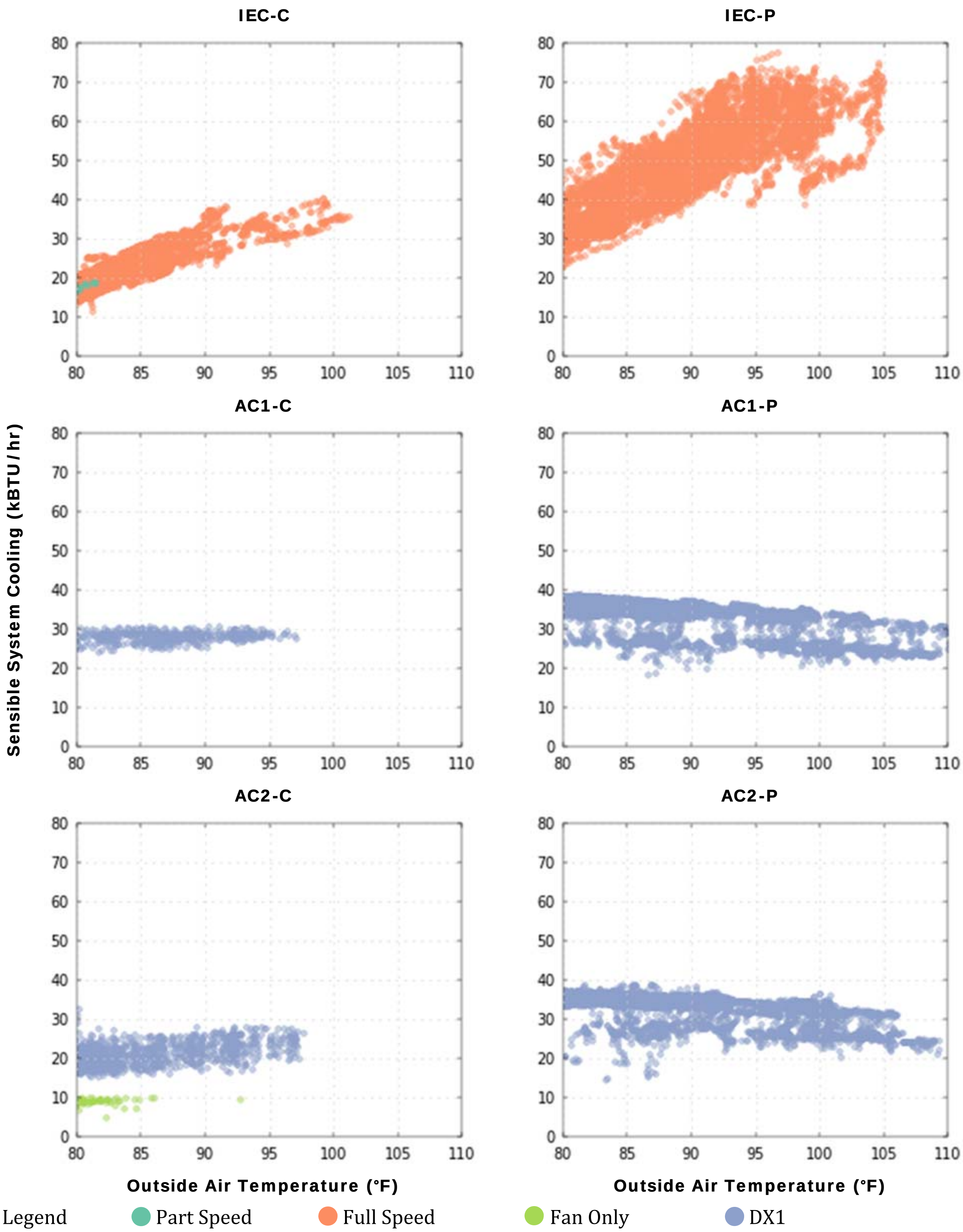


FIGURE 12. SENSIBLE SYSTEM COOLING CAPACITY

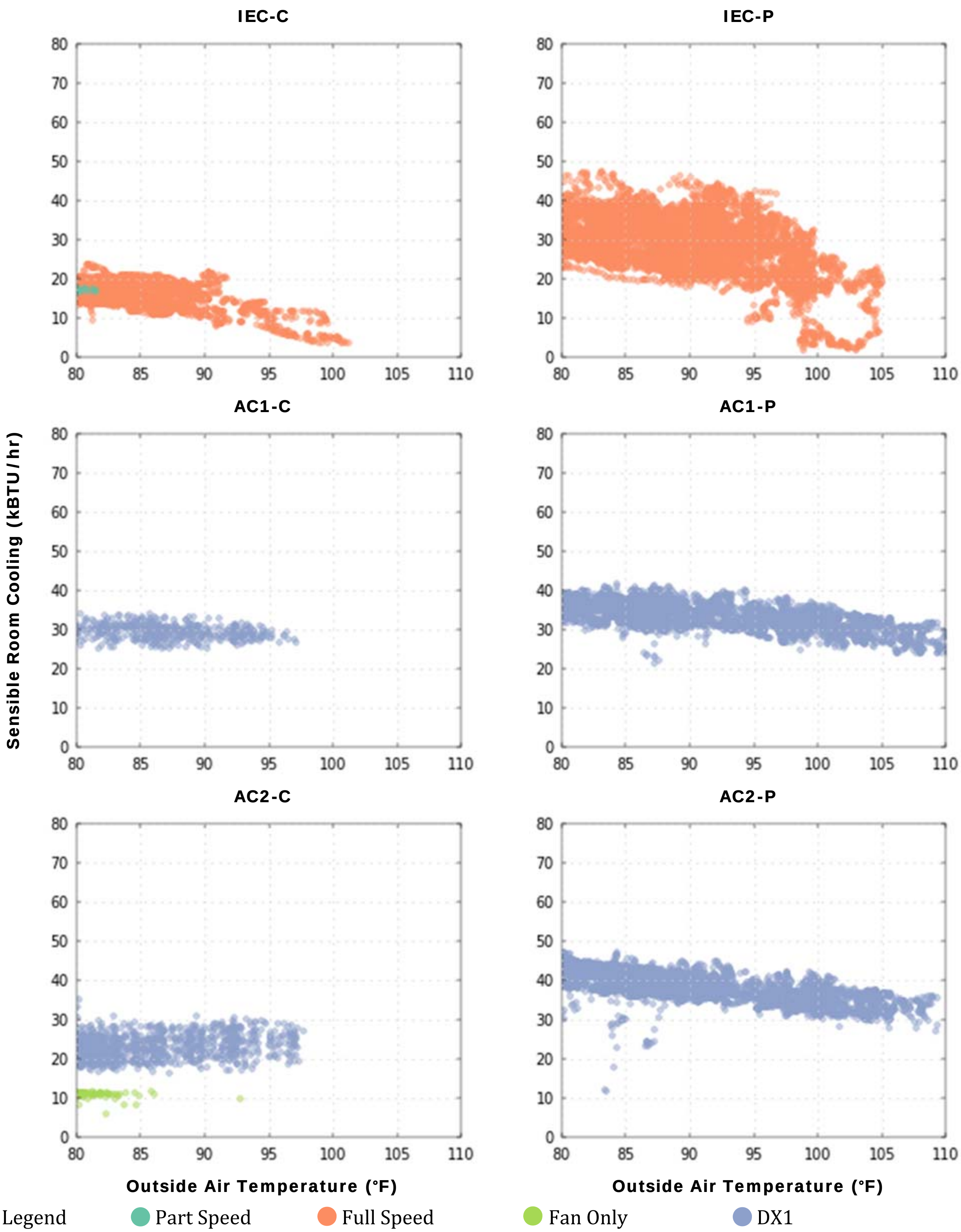


FIGURE 13. SENSIBLE ROOM COOLING CAPACITY

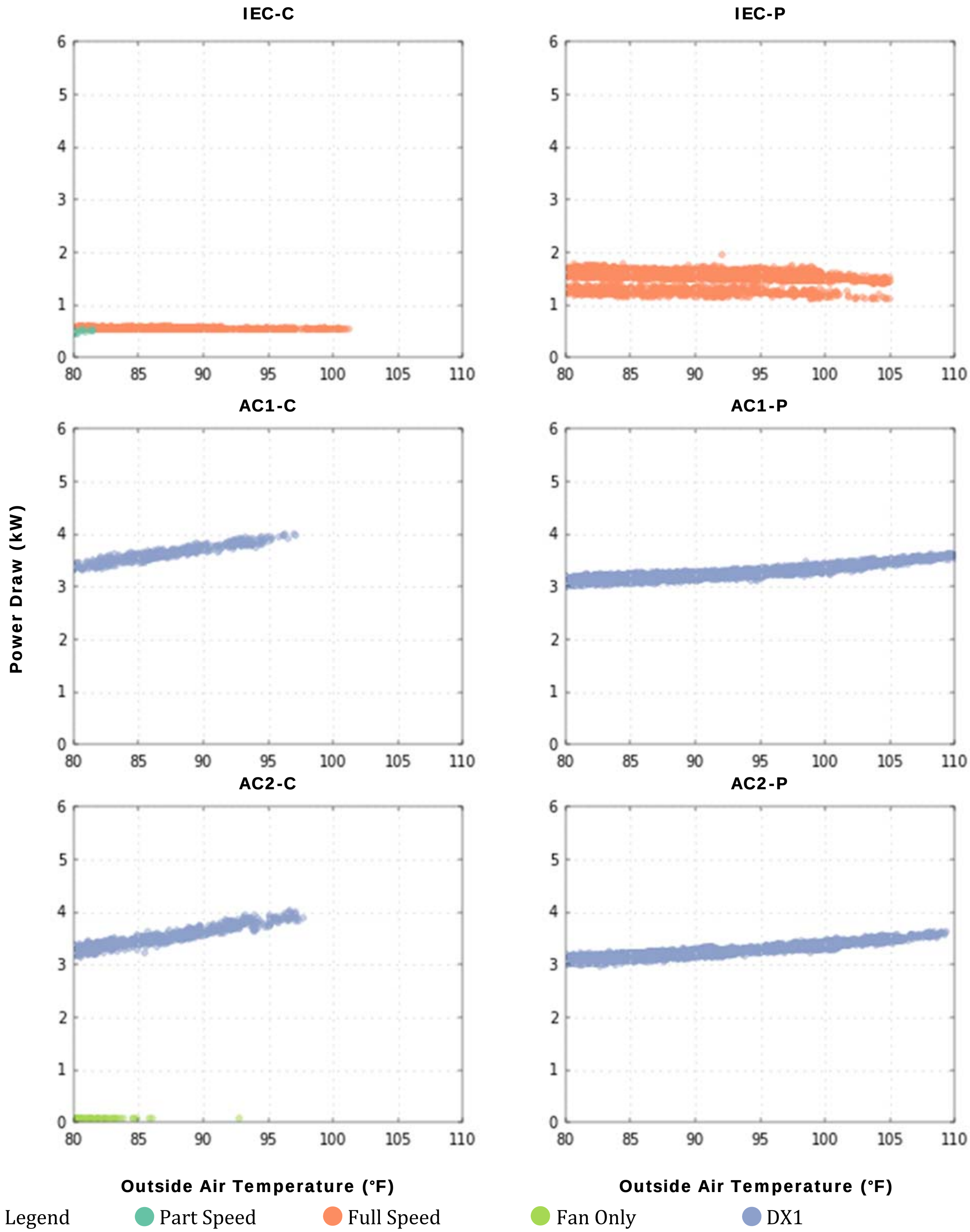


FIGURE 14. SYSTEM POWER DRAW

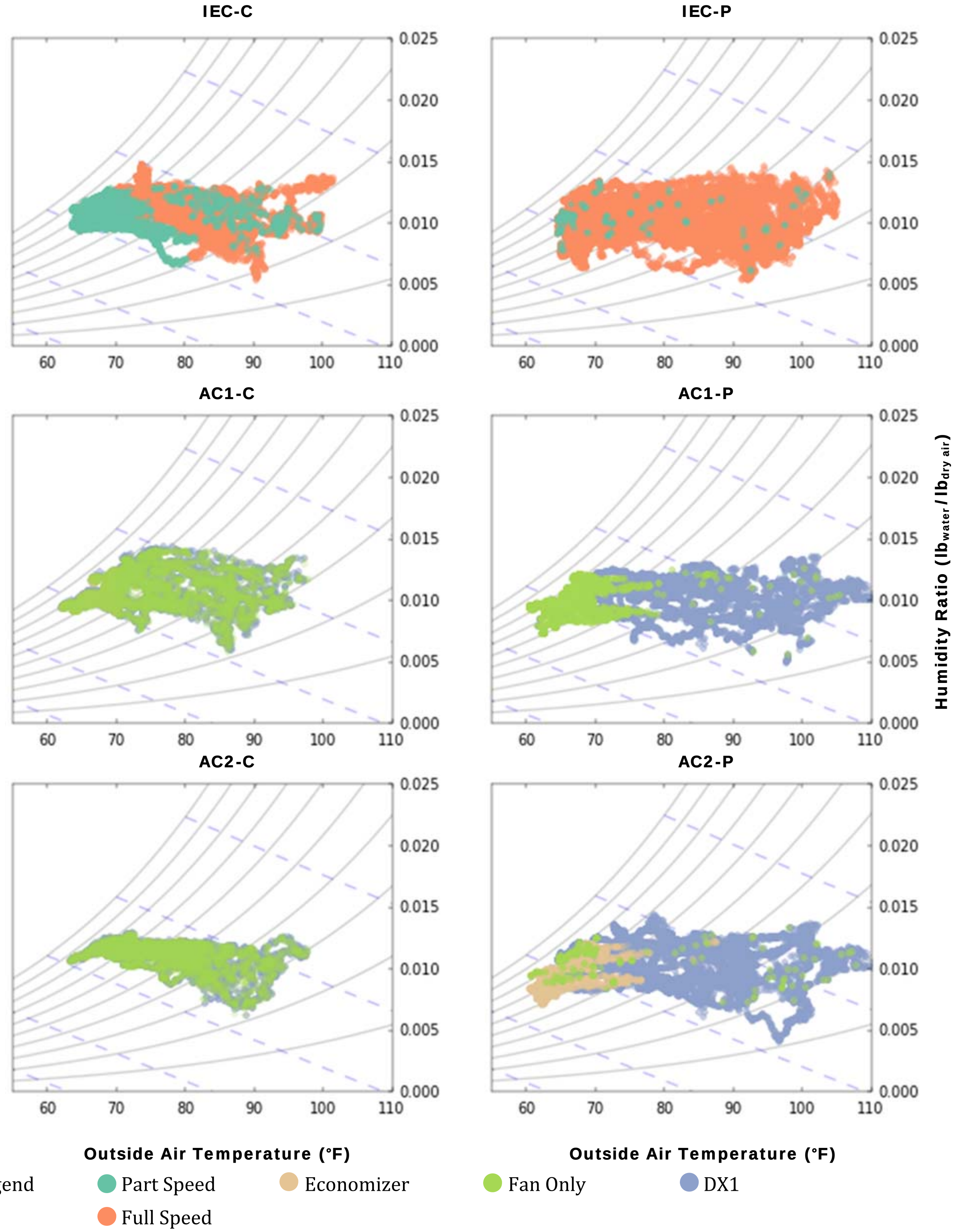


FIGURE 15. PSYCHROMETRIC CONDITIONS WHILE EQUIPMENT WAS OPERATING

WET-BULB EFFECTIVENESS

Figure 16 shows the wet-bulb effectiveness observed for both the IEC-C and IEC-P. In this study, IEC-P performed between 0.6 – 1.2 and IEC-C preformed between 0.4 – 1. Typically, the evaporative effectiveness is higher at part-speed, because the air is moving slower and thus has more contact with the water in the wet channels. However, in the field test, IEC-C was the only unit with part-speed steady-state performance and those instances corresponded to outside air condition with a lower wet-bulb depression.

The results from the field study show the wet-bulb effectiveness for IEC-C and IEC-P type units are 16% and 18% higher than a previous field study conducted in California Climate Zone 13 (Woolley 2014). Conversely, making direct comparisons and conclusions between this test and previous tests are difficult because this study had far less steady-state operation than previous field studies.

WATER-USE EFFICIENCY

Figure 17 displays the water-use efficiency of the IEC units based on the gallons of water used per ton-hour of sensible system cooling. The water-use efficiency is not constant, but it follows the trend of the daily average outside air temperature. Each IEC's water-use efficiency is dependent on the design of the system. As previously discussed in the ***Error! Reference source not found.*** section, IEC-P utilizes a sump and IEC-C does not. The results in Figure 17 show that this design difference has a bigger impact on water-use efficiency on days where the outside air temperature is lower. On days where the daily average outside air temperature was 75°F or lower, the IEC-P water-use efficiency was almost double IEC-C – 3.8 gal/ton-hr compared to 2.1 gal/ton-hr. This result stems from the design of IEC-C having to keep the heat exchanger wet at all times and thus consuming water when there is no call for cooling. During periods of non-use, IEC-P stores water in the sump. Therefore, extra water is not required until there is a demand for cooling. The impact of the design difference is reduced during periods when the daily average outside air temperature was above 75°F and the IECs operated in a more continuous fashion. In these instances, both IEC-C and IEC-P used 4-5 gal/ton-hr of sensible system cooling.

TOTAL WATER CONSUMPTION

Figure 18 displays the total water-use by day. It is quickly noticed that IEC-C uses water every day even when, for this field test, the unit only operates every other day. This is another example of the design difference between the two units. The research team recognizes that the experimental design influenced this behavior and that it will not be consistent with a typical installation. However, the experimental design allows the research team to show the minimum amount of water-use per day for IEC-C. Independent of cooling demand, IEC-C requires 75 gallons per day to keep the heat exchanger wet. On days with cooling, the water-use increases up to an additional 75 gallons of water. Due to having a sump, IEC-P has a noticeably different water-use pattern. IEC-P also uses more water on retrofit days than IEC-C, however, the water-use efficiencies are similar as shown in Figure 17.

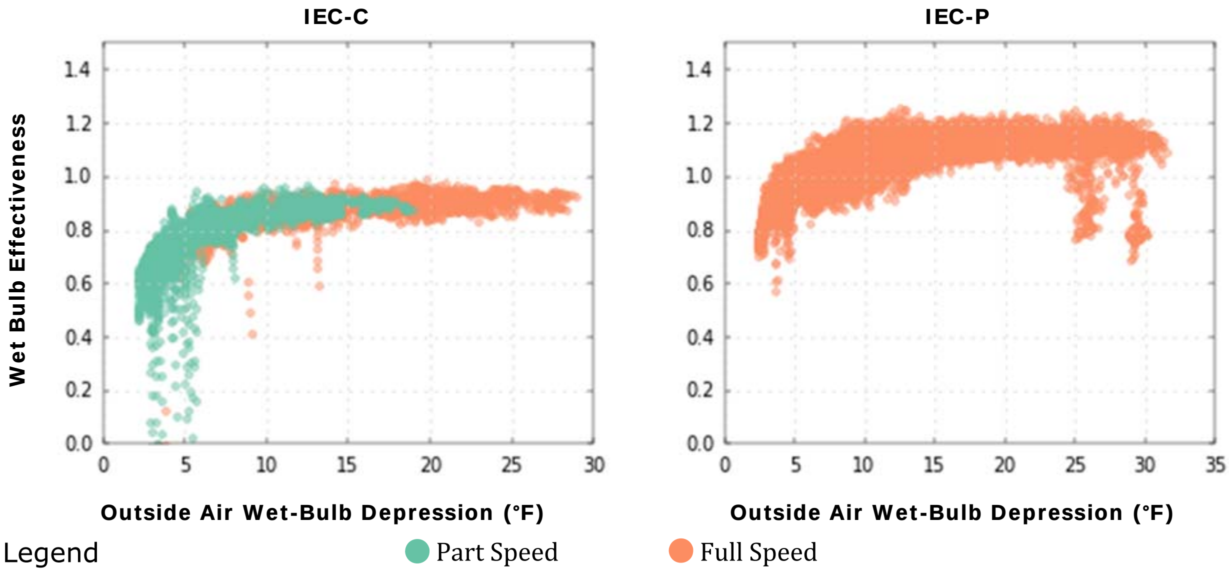


FIGURE 16. WET-BULB EFFECTIVENESS

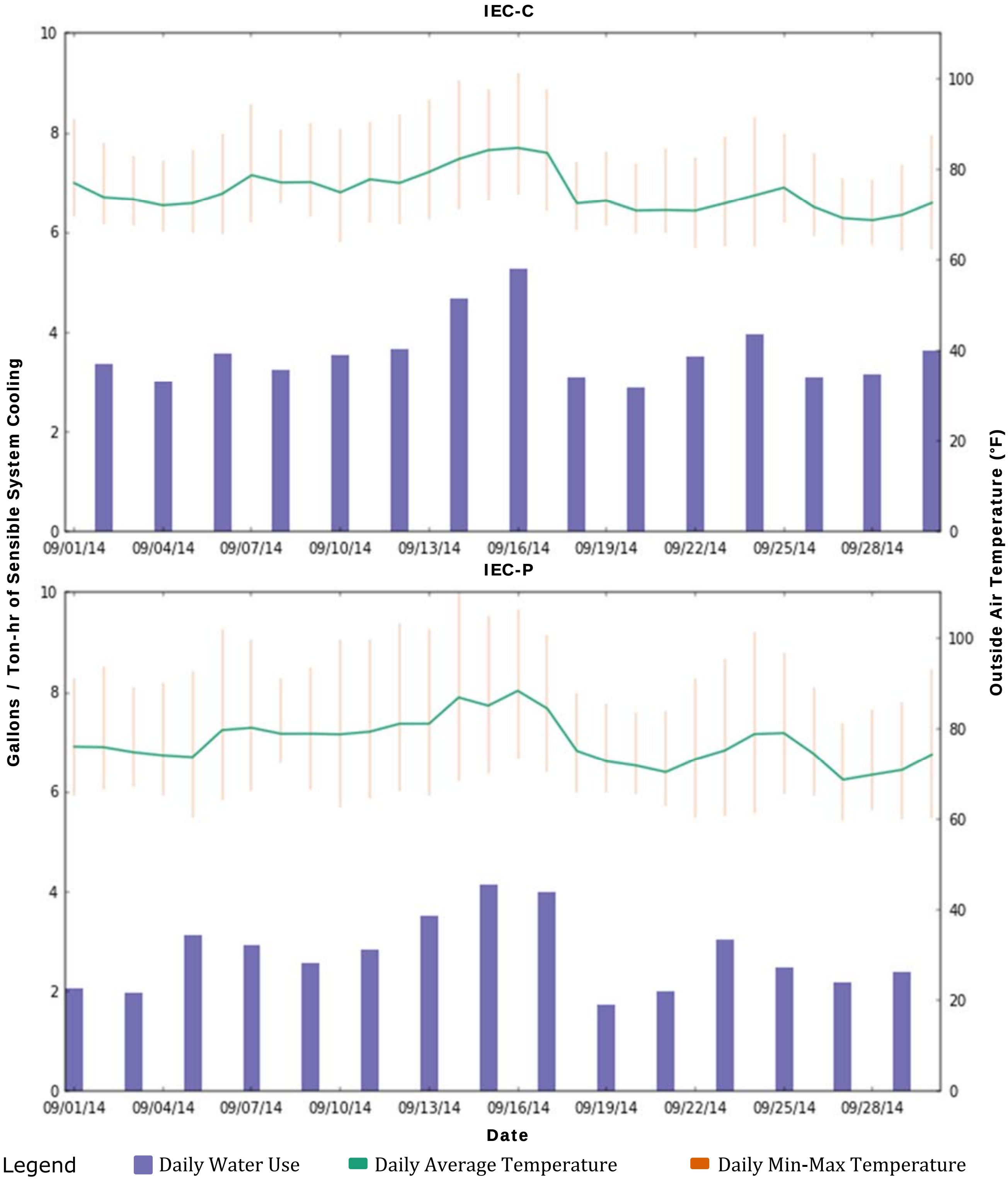


FIGURE 17. WATER-USE EFFICIENCY

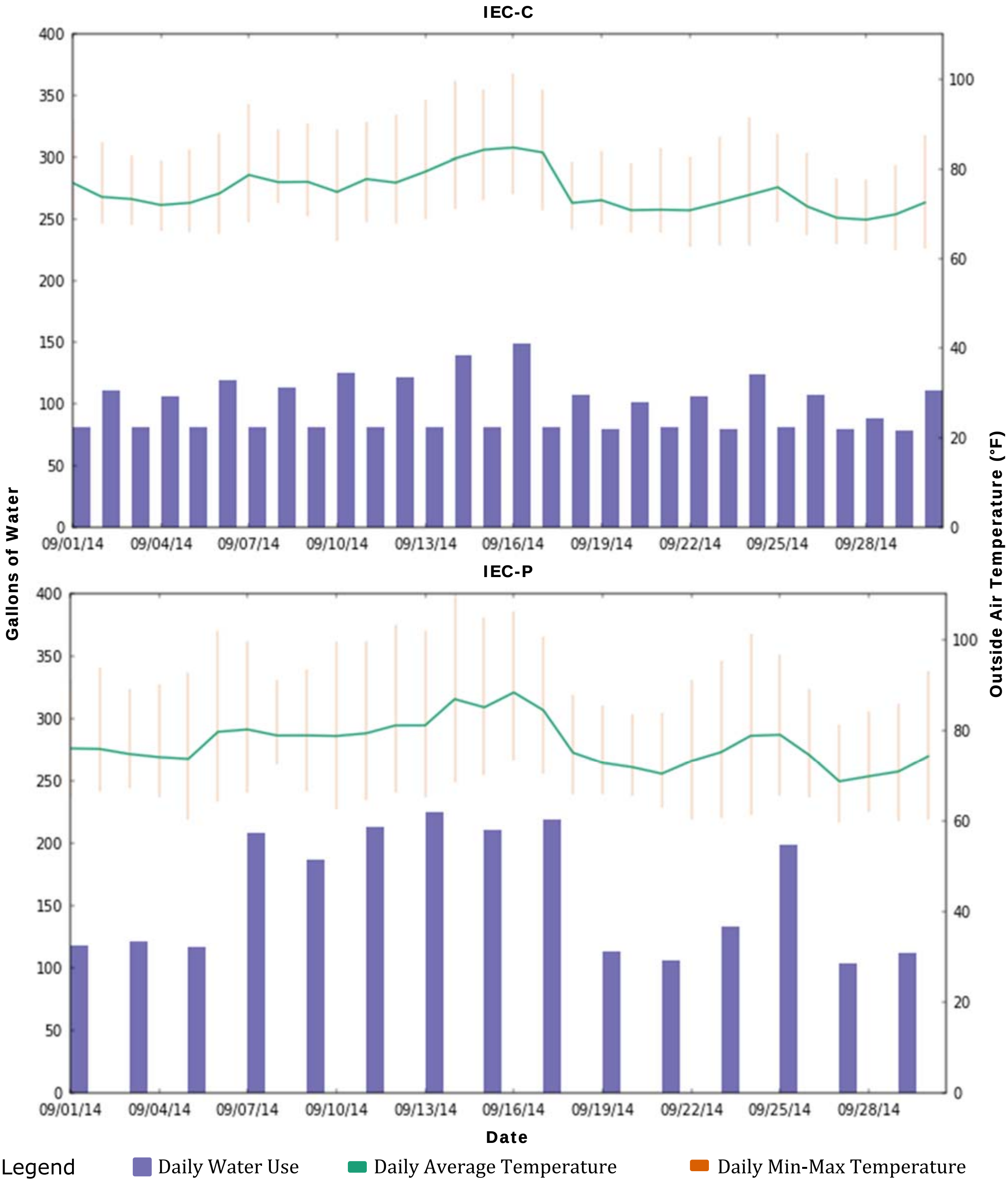


FIGURE 18. TOTAL WATER CONSUMPTION

CONCLUSIONS

As the results of this study show, indirect evaporative air conditioners can achieve much higher sensible cooling efficiencies than conventional vapor-compression systems.

Table 9 and Table 10 provide a summary of the comparison between the equipment at each field site. The values are averaged over all steady-state values within the given temperature range. For ease of comparison, IEC values are for full-speed operation and the baseline units are operating with the compressor on.

It is important to remember that the room cooling capacity will decrease with an increase in outside air temperature. Therefore, in certain applications IECs are not suited for a one-to-one replacement with conventional DX equipment. However, IECs will still provide substantial savings if used to partially replace or offset DX cooling. Another suitable application for IECs is cooling ventilation air for commercial spaces. This field test had no ventilation requirement. However, removing the extra outside air load from conventional DX systems and using IECs instead for ventilation air cooling will result in substantial savings.

TABLE 9. SUMMARY OF KEY RESULTS FOR CUDAHY

KEY METRIC	80-85°F	85-90°F	95-100°F	100-105°F	105 - 110°F
IEC-C					
Sensible System COP (–)	10.2	13.8	16.7	17.6	18.6
Sensible System Capacity (kbtu/hr)	20.0	26.9	32.2	33.9	35.6
Sensible Room COP (-)	8.6	8.4	7.3	4.0	2.5
Sensible Room Capacity (kbtu/hr)	16.9	16.5	14.1	7.7	4.8
Water Use Efficiency (Gal/Ton-hr)	4.4	3.8	3.5	3.7	3.4
Total Water Usage (Gal)	285	255	203	91	51
Power Draw (kW)	0.6	0.6	0.6	0.6	0.6
AC1-C					
Sensible System COP (–)	2.4	2.3	2.2	2.1	2.0
Sensible System Capacity (kbtu/hr)	28.2	28.2	28.8	28.8	28.5
Sensible Room COP (-)	2.6	2.4	2.2	2.1	1.8
Sensible Room Capacity (kbtu/hr)	30.4	29.7	29.3	28.5	25.8
Power Draw on Baseline Days (kW)	3.5	3.6	3.8	4.0	N/A
Power Draw on Retrofit Days (kW)	3.5	3.6	3.8	4.0	4.2
Sensible System COP (–)	2.4	2.3	2.2	2.1	2.0
AC2-C					
Sensible System COP (–)	1.8	1.8	1.8	1.8	N/A
Sensible System Capacity (kbtu/hr)	20.5	21.8	23.3	24.1	N/A
Sensible Room COP (-)	2.0	2.0	1.9	1.8	N/A
Sensible Room Capacity (kbtu/hr)	22.9	23.7	24.7	24.6	N/A
Power Draw on Baseline Days (kW)	3.3	3.5	3.7	3.9	N/A
Power Draw on Retrofit Days (kW)	3.3	3.5	N/A	N/A	N/A

TABLE 10. SUMMARY OF KEY RESULTS FOR PLACENTIA

KEY METRIC	80-85°F	85-90°F	95-100°F	100-105°F	105 - 110°F
IEC-P					
Sensible System COP (–)	7.0	8.7	10.6	11.6	12.2
Sensible System Capacity (kbtu/hr)	37.5	45.6	56.1	59.7	60.9
Sensible Room COP (-)	5.9	5.6	5.7	4.6	3.1
Sensible Room Capacity (kbtu/hr)	31.8	29.6	30.3	23.7	15.8
Water Use Efficiency (Gal/Ton-hr)	2.9	2.5	2.4	1.9	2.5
Total Water Usage (Gal)	278	320	290	281	265
Power Draw (kW)	1.6	1.6	1.6	1.5	1.5
AC1-P					
Sensible System COP (–)	3.2	3.0	2.9	2.7	2.4
Sensible System Capacity (kbtu/hr)	35.0	33.1	32.7	30.4	27.6
Sensible Room COP (-)	3.3	3.2	3.1	2.8	2.7
Sensible Room Capacity (kbtu/hr)	35.8	34.7	34.1	32.2	31.1
Power Draw on Baseline Days (kW)	3.2	3.2	3.3	3.3	3.4
Power Draw on Retrofit Days (kW)	3.2	3.2	3.3	3.4	3.4
AC2-P					
Sensible System COP (–)	3.2	3.1	3.0	2.7	2.5
Sensible System Capacity (kbtu/hr)	34.7	33.9	33.4	30.1	28.8
Sensible Room COP (-)	3.9	3.6	3.5	3.2	3.0
Sensible Room Capacity (kbtu/hr)	41.5	39.3	38.6	36.2	35.0
Power Draw on Baseline Days (kW)	3.1	3.2	3.3	3.4	3.4
Power Draw on Retrofit Days (kW)	3.1	3.2	3.3	3.4	3.4
Sensible System COP (–)	3.2	3.1	3.0	2.7	2.5

RECOMMENDATIONS

The research team recommends IECs as an impactful measure to reduce energy consumption and peak demand for cooling in commercial buildings. However, the team also recommends that utility efficiency programs, and other efforts to advance the technology, should remain cognizant of some of the challenges that can hinder performance and limit the persistence of savings. It is especially important that any installation of this measure be paired with a quality service agreement. In our observation, the current lack of industry familiarity with the technology can result in untimely failure or abandonment of the measure. For this specific field test, the cellular telephone company had a standing service contract. Adding the IEC equipment that uses 100% outside air requires filter service on a shorter interval. When possible, it is suggested that new systems be installed with a guaranteed system performance for a minimum time horizon. This might necessitate a different type of capital and incentive structure but will address some of the challenges that currently plague performance for HVAC equipment in commercial buildings. In addition, IEC equipment needs to be drained to prevent freezing of water pipes in the winter. California Climate Zone 8 rarely frosts or freezes which negated the need for a seasonal service to prevent the IEC water lines from freezing.

The research team also recommends the 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. The market penetration for IECs is still small and the benefits of such an incentive or rebate program will help end users with the larger up-front cost.

Lastly, the research team suggests that further research be conducted to translate the characteristic measurements from this study into a calibrated model for an indirect evaporative system. The results presented in this report stand as particular examples, but the characteristic observations allow for development and validation of a general map for the technology that can be applied to other scenarios through building energy modeling. This model can be used to simulate savings across an array of climates and applications. It can also help programs target strategic savings opportunities, identify approaches to optimize control of IECs and inform the development of design guidelines to support broad and successful application of the measure.

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