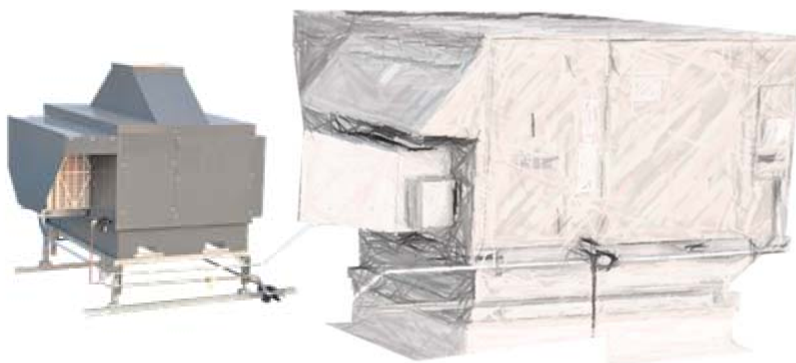
LABORATORY PERFORMANCE RESULTS:
INDIRECT EVAPORATIVE AIR CONDITIONING AND CONDENSER PRE-COOLING AS CLIMATE-APPROPRIATE RETROFITS FOR PACKAGED ROOFTOP UNITS

ET15SCE7050



Prepared by:

Emerging Products
Customer Service
Southern California Edison

November 2015

SOUTHERN CALIFORNIA EDISON
An EDISON INTERNATIONAL® Company

Acknowledgments

Southern California Edison's Emerging Products (EP) group is responsible for this project. It was developed as part of Southern California Edison's Emerging Technologies Program under internal project number ET15SCE7050, by University of California Davis's (UC Davis) Western Cooling Efficiency Center (WCEC). Jay Madden conducted this technology evaluation with overall guidance and management from Jerine Ahmed. Contact Jay.Madden@SCE.com for more information on this project.

EXECUTIVE SUMMARY

This report documents and discusses results of a detailed laboratory evaluation of an indirect evaporative cooler (IEC) coupled to a rooftop packaged air conditioner (RTU) that was retrofit with a condenser air pre-cooler. The scheme reduces air conditioner compressor energy use in two main ways:

- Reducing condenser inlet temperature and improving efficiency for the vapor compression cycle
- Meeting a substantial portion of the building cooling needs with indirect evaporative cooling

The primary goal of this project was to characterize the energy efficiency of the retrofit package in all modes of operation and possible configurations, as well as across a full range of operating conditions. To this end, the project team conducted extensive laboratory tests and carefully analyzed results to consider the technical opportunities and challenges related to the equipment and identify opportunities for improving the technology. These findings provide the basis for recommendations for utility efficiency programs, design engineers, and customers on applying this technology for managing indoor environmental quality in commercial buildings while simultaneously reducing energy consumption and peak demand.

The study showed that IEC coupled with condenser air pre-cooling has the potential to transform new or existing RTUs into climate-appropriate equipment for hot dry climates. The IEC delivers a very efficient means to cool outside air for ventilation or modest room cooling while the condenser air pre-cooler can significantly improve the performance of the RTU vapor compression cycle. The configuration tested demonstrated it could provide these benefits:

- Reduce annual energy use for cooling and ventilation by in 68% in California commercial buildings
- At peak cooling conditions (105°FDB/73°FWB), achieve equal capacity while reducing energy use for cooling by 26%
- At milder conditions (65°FDB/ 52.8°FWB), offer savings at large as 86%

Because the retrofit connects new loads, it increases electrical demand at every condition tested. On aggregate, the retrofit would decrease energy used during peak cooling conditions by reducing compressor operation. Nonetheless, the highest instantaneous electrical draw will be larger, potentially increasing customer demand charges. For regional and statewide grid management, the gains in system efficiency would reduce peak generation requirements. Moreover, the retrofit could reduce the total electrical demand for a building if the retrofit handled ventilation air for multiple units, thereby allowing the operation of existing units in recirculation mode or even disconnection of the existing units.

The equipment expands the number of potential operating modes by allowing multiple combinations of IEC and compressor stages—driving the need to develop smart controls that can determine the most appropriate and beneficial operating mode for a particular situation.

The project team recommends that the technology be adopted by utility energy efficiency programs. As is the case for most air conditioning technologies, some applications will be better suited than others, and programs that adopt this technology should be designed to avoid scenarios where overall energy performance is limited. A simulation study should be conducted to assess the savings potential for target customers and identify applications that ought to be avoided. A first-generation program that draws on customer smart meter data and limited field monitoring can help assess the actual savings achieved by installation of the equipment. Technology improvement efforts could focus on design guidance to facilitate proper physical application and optimized controls integration.

ABBREVIATIONS AND ACRONYMS

AHRI	Air-Conditioning, Heating, and Refrigeration Institute
ANSI	American National Standards Institute
ASHRAE	American Society of Heating Refrigeration and Air Conditioning Engineers
CEC	California Energy Commission
CFM	cubic feet per minute
COP	coefficient of performance (dimensionless)
cp	specific heat capacity (e.g. Btu/lbm-°F)
CPUC	California Public Utilities Commission
CX	concentration (of constituent X) (e.g., parts per million (ppm))
DX	direct expansion vapor compression
ε	sensible heat exchanger effectiveness (dimensionless)
$\dot{E}$	electric power, (rate of electric energy consumption) (e.g., kW)
EA	Exhaust Air
$\dot{H}$	cooling capacity, (enthalpy flow rate) (e.g. kBtu/h)
h	specific enthalpy (e.g., Btu/lbm-dryair)
HR	humidity ratio (e.g., lbmwater/lbmdryair)
HVAC	heating, ventilation, and air conditioning
IEC	indirect evaporative air conditioner (indirect evaporative cooling)
IEER	integrated energy efficiency ratio
kW	kilowatt
kWh	Kilo-watt-hour
$\dot{m}$	mass flow rate (e.g. *lbm/h*)
OSA	outside air
ΔP	differential static pressure (e.g. *inwc*)
RA	return air
RH	relative humidity (%)
RTD	resistance temperature detector
RTU	rooftop packaged air conditioning unit
SA	supply air
SCE	Southern California Edison

SEER	seasonal energy efficiency ratio
T	temperature (e.g. °*F*)
$\dot{V}$	volume flow rate (e.g. *scfm*)
VRF	variable refrigerant flow
WBE	wet bulb effectiveness

CONTENTS

FIGURES

TABLES

EQUATIONS

INTRODUCTION

OVERVIEW

This study documents the laboratory-measured performance for a rooftop package air conditioner (RTU) outfitted with an indirect evaporative air conditioner (IEC)—ducted to the outdoor air supply damper of the RTU—and an evaporative condenser air pre-cooler. This work directly supports Southern California Edison's (SCE's) Emerging Technologies efforts related to the California Energy Efficiency Strategic Plan (CA-EESP) goals developed by the California Public Utilities Commission (CPUC) to accelerate marketplace penetration of new climate-appropriate heating, ventilation, and air conditioning (HVAC) technologies [CPUC 2011]. The aggressive goals regarding climate-appropriate technologies in the CA-EESP reflect the potential benefits of this approach: reduced energy consumption, improved management of electrical demand, and eased integration of renewable and environmentally responsible energy resources.

CONTEXT

Cooling and ventilation create significant energy challenges for California. These activities consume more than 25% of the electricity used the state's commercial buildings each year [CEC 2006], and air conditioners can account for more than 50% of the peak electric load of commercial buildings during the hottest summer afternoons [CEC 2006]. California's electric grid typically experiences the highest 20% of demand for fewer than 5% of the hours each year [CA ISO 2015], and this high demand can be attributed almost exclusively to cooling throughout the state.

Further, HVAC can account for more than 50% of the annual greenhouse gas footprint for a commercial building [EIA 2012]. With the rapid efficiency increases for lighting and electronics, HVAC is projected to become the largest annual energy end use in most commercial buildings.

As the single largest individually connected load in many buildings, rooftop air conditioners represent a substantial opportunity for energy efficiency improvements. However, because these units have a very low turnover rate, the efficiency of a system installed will very likely persist for more than 20 years—highlighting the need for a retrofit solution. Further, vapor compression air conditioners have traditionally not varied much in design according to climate zone. Designing a system to work in concert with regional meteorological conditions can further reduce the energy consumption for cooling and ventilation.

SOLUTION

This research looks at a retrofit package that can have an immediate impact on the energy performance of existing RTUs, potentially transforming them from climate-agnostic technologies to products that are better tuned and more appropriate for a specific set of climate zones.

Because all California climate zones are relatively dry, efficient use of modern and sophisticated evaporative cooling technologies is a possible climate-appropriate method to reduce cooling energy consumption. This study focuses one of the many possible approaches: an innovative application of an IEC by ducting the device into the outdoor air supply damper of an RTU.

IEC can be part of a building HVAC system in many ways. For this laboratory test, the IEC was set up to deliver cool air through the outside air inlet of an existing RTU. In addition, a condenser air pre-cooler was used to improve the performance of the refrigeration cycle of the RTU. The system tested is capable of many modes of operation that use various combinations of the RTU components, the IEC, and the pre-cooler. The various modes are outlined in Table 1.

Equipment that handles outside air is especially well-suited to employ indirect evaporative cooling. Whereas high outdoor air temperatures generally result in reduced cooling capacity and efficiency of vapor compression systems, hot outdoor air typically increases the cooling capacity and efficiency of IEC. Consequently, the largest incremental savings can be achieved when IEC is applied to cooling of hot ventilation air. When integrated with an RTU, IEC can also cover sensible room cooling loads, but will achieve less energy savings compared to those of a conventional baseline in this configuration.

REPORT CONTENTS

This report presents analysis of laboratory testing of indirect evaporative air conditioner and condenser air pre-cooler retrofitted onto an existing RTU—one example of a retrofit package that incorporates IEC, vapor compression, and condenser air precooling. Tests measured the cooling capacity and energy consumption for the system in each mode of operation and across a range of operating conditions. Further, the tests characterized the performance of each major subcomponent as a function of the relevant independent conditions. For example, fan performance was mapped separately from tests of thermal performance. Laboratory tests were designed to gather a level of detail appropriate to inform and validate the simulation of system performance in various climate conditions and applications.

BACKGROUND

CALIFORNIA ENERGY EFFICIENCY STRATEGIC PLAN

CA-EESP outlines four major programmatic initiatives as Big Bold Energy Efficiency Strategies to facilitate broad energy savings for our built environment:

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

The third initiative targets a 50% efficiency improvement for HVAC by 2020, and a 75% improvement by 2030. The plan recognizes that cooling and ventilation is the single largest contributor to peak electrical demand in California, resulting in "…enormous and costly impacts on generation, transmission, and distribution resources as well as a concurrent lowering of utility load factors."

Strategic goals to transform the HVAC industry focus on the following:

- Code compliance
- Quality installation and maintenance
- Whole-building integrated design practices
- Development and accelerated implementation of new climate-appropriate equipment and controls

Climate-appropriate technologies offer greater energy savings in California's climates than do other cooling efficiency measures; they are especially valuable for electricity savings during peak cooling hours when air conditioning alone can account for more than 30% of the peak demand on the statewide electric network [EIA 2012, CEC 2006]. Many studies have demonstrated that add-on evaporative pre-coolers for conventional air conditioners can reduce peak electrical demand from cooling by as much as 30–60% [Woolley 2014, Modera 2014, Pistochini 2014, Davis, 2015]. However, the savings achieved depends significantly on technology and application.

Many climate-appropriate technologies and system design strategies are available that use far less energy than the "one-size-fits-all" approach. Designed and tuned specifically for local climate conditions and occupant comfort needs, climate-appropriate air conditioning systems and controls provide an equal (or better) quality of service with less energy input.

Many cooling strategies are appropriate for California climates:

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

EFFICIENCY RATING ISSUES

The industry uses a single-number efficiency metric to rate air conditioning equipment. Optimizing for this metric, manufacturers have mainly sold a single type of air conditioner that functions reliably in any climate, but is also inefficient in every climate.

It is important to note that these ratings can be misleading. The limited range of conditions for the standard tests done to generate the ratings does not exactly represent every application in California. Moreover, the standard methods of test, such as American National Standards Institute/Air Conditioning, Heating, and Refrigeration Institute [ANSI/AHRI) Standards 210/240 and 340/360, can actually portray climate-optimized products as less efficient than traditional air conditioners [Woolley 2011]. In many circumstances, climate-appropriate strategies cannot even be tested by industry standard methods because the strategies call for configurations fundamentally different than those for which the standards were designed. This shortcoming is especially true for whole-building integrated design practices.

TECHNOLOGY DESCRIPTION

The sections below provide an overview of the technology and discuss design considerations for integrating an IEC into an HVAC system.

OVERVIEW

The project reported here studies an indirect evaporative air conditioner and an evaporative condenser air pre-cooler added as retrofit to a rooftop air conditioner that had been in service at a cafeteria from 1991–2013 and removed for this laboratory test. The project builds on a body of research and technology evaluations supported by Southern California Edison, Pacific Gas and Electric Company (PG&E) and other California entities to advance the understanding and market introduction of climate-appropriate HVAC solutions. In particular, this laboratory study builds on a recent field evaluation of the same technology installed in a big-box retail store in Bakersfield, CA [Woolley 2014].

Indirect evaporative air conditioning uses water evaporation in one air stream to impart sensible cooling to another air stream without adding moisture to the conditioned space. There are a number indirect evaporative heat exchanger designs and many different ways to apply indirect evaporative systems as an efficient part of a building HVAC system. The technology studied here uses a specially designed polymer plate heat exchanger that operates partly as counter-flow and partly as cross-flow.

The conceptual process is illustrated in Figure 1. The heat exchanger is integrated into a small packaged air handler with an integral sump and variable speed fan. At full speed, the single fan moves approximately 5,000 cubic feet per minute (cfm) of outside air that is cooled sensibly as it passes through the primary side of the heat exchanger. At this point, the air stream is split; roughly half of the airflow passes back through the secondary side of the heat exchanger, and roughly half is delivered to the building as the product air stream. The secondary side of the heat exchanger is wetted, and evaporation here drives the cooling process. The cascading flow design for this heat exchanger allows this manufacturer’s product, uniquely, to cool below the outside air wet bulb temperature during regular operation.

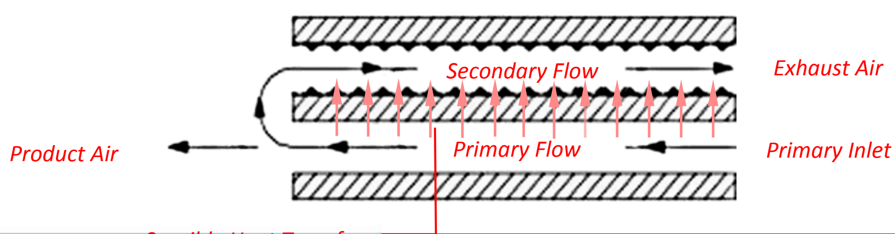


FIGURE 1. CONCEPTUAL SCHEMATIC FOR INDIRECT EVAPORATIVE COOLING

The IEC is designed to operate as a stand-alone cooling unit and incorporates a proprietary thermostat controller that adjusts the system's fan speed to vary cooling capacity in response to room conditions. However, in many commercial buildings, IED may not be adequate to cover all room cooling requirements at all times. Most important, although the system cooling capacity and efficiency increase as outside temperature increases, the product air temperature also rises. As a result, IEC in many applications must be combined with other cooling strategies. This laboratory study investigates one of the many possible approaches for such combinations, the integration of IEC as retrofit for a conventional vapor compression air conditioner.

DESIGN CONSIDERATIONS

Design engineers must consider many factors when applying indirect evaporative cooling as one element in a whole-building HVAC system, including the following:

- Controls should give priority to indirect evaporative cooling over less-efficient cooling modes.
- Controls should give priority to economizer cooling over indirect evaporative cooling.
- Systems must maintain ventilation requirements without overcooling the zone.
- Systems must maintain ventilation requirements even when heating is required.
- If the indirect evaporative system is used in combination with vapor compression equipment (as in this study), the combination must maintain adequate evaporator coil air-flow.
- Care must be taken to maintain appropriate downstream static pressure.

Balancing all of these requirements can be challenging. For example, when an IEC is the primary source for ventilation air, a separate system may be needed to provide ventilation air during the heating season. Because on-board controls do not direct this type of coordination, some sort of global controller is often required.

The sections below discuss some of the most critical design factors.

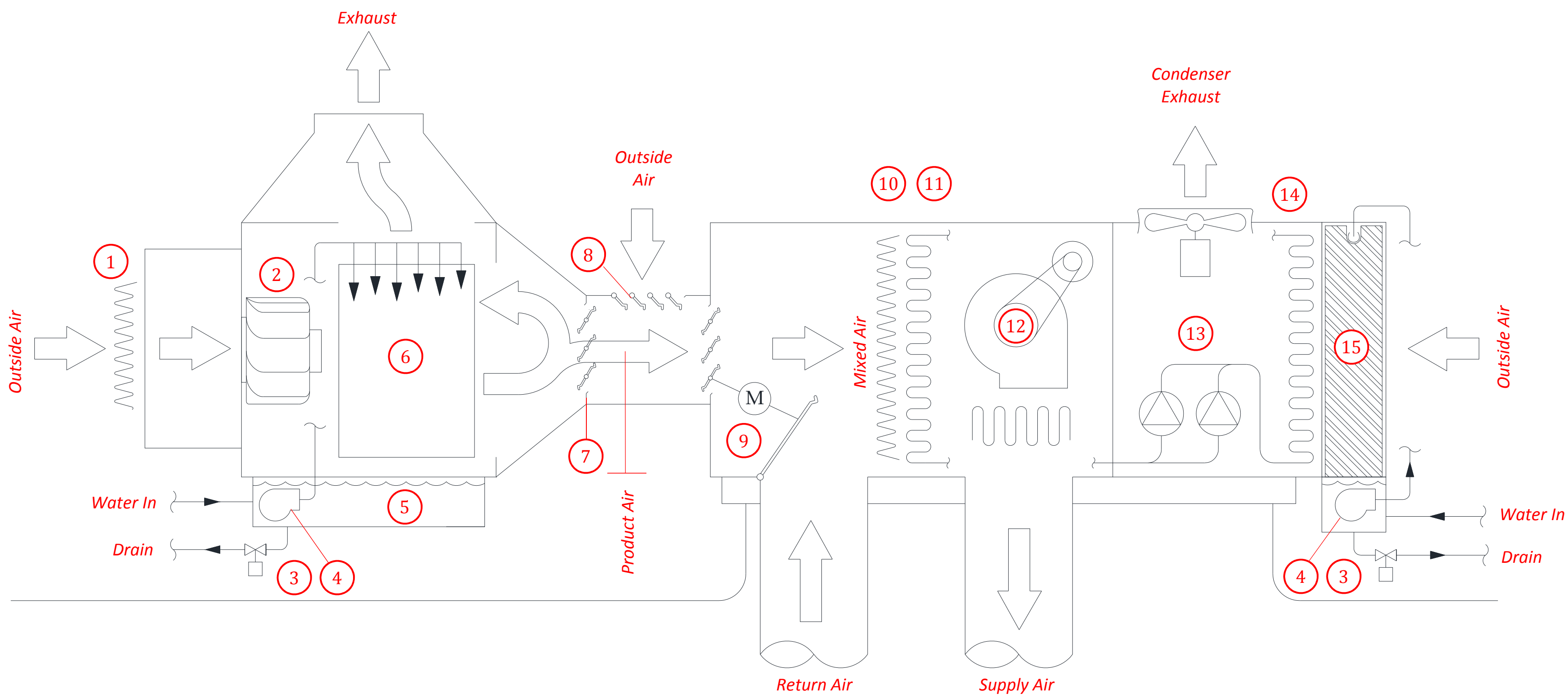
MAINTAINING PRIMARY–SECONDARY BALANCE

Maintaining appropriate static pressure conditions is especially important to avoid primary–secondary imbalance for the heat exchanger, which will degrade performance. The product tested uses a damper in the product air stream that is manually adjusted to provide the resistance needed to create the proper balance—a strategy that is sufficient in many applications. However, when the downstream resistance changes, other pressure management strategies are needed. Resistance change can occur, for example, when the indirect evaporative system is coupled with a conventional air conditioner. In such cases, the rooftop unit fan operation can disrupt the pressure balance.

In this project, the manual damper was adjusted to the proper resistance while the RTU fan was operating, then remained fixed for all tests. Consequently, the resistance to airflow was larger than design for indirect evaporative modes when the RTU fan was turned off. The added resistance decreased product airflow, increased secondary airflow, and resulted in a lower product air temperature.

SYSTEM INTEGRATION

The physical integration of systems can be complicated. A field study with the same IEC technology at a big box retail store [Woolley 2014] demonstrated a rather elaborate method to accommodate all of these needs without revising the building's existing energy management and control system. The laboratory test reported here set out to explore a simpler approach, illustrated in Figure 2, for adding the IEC unit to an existing rooftop air conditioner.



1	*Air Filters*	6	*Ind. Evap. Heat Exchanger*	11	*Evaporator Coil*
2	*Variable Speed Fan*	7	*Balancing Damper*	12	*Supply Air Fan*
3	*Drain Solenoid*	8	*Barometric Relief*	13	*2 Compressors, 1 Circuit*
4	*Pumps*	9	*Outside Air and Return Air Dampers*	14	*Condenser Coil*
5	*Sump*	10	*Air Filters*	15	*Evap. Condenser Pre Cooler*

FIGURE 2. SCHEMATIC OF ROOFTOP UNIT WITH INDIRECT EVAPORATIVE COOLING AND EVAPORATIVE CONDENSER PRE-COOLING

The IEC unit was set up to deliver product air through the outside air intake for a conventional rooftop air conditioner. In this configuration, the IEC can utilize existing ductwork distribution systems, and it can operate either independently from or in cooperation with the RTU. However, the size of the RTU to which the IEC is matched may change the optimal control strategy. If the airflow resistance through the rooftop unit is small enough, the IEC can operate without the RTU fan. In fact, if the product airflow rate were equal to the rooftop unit's supply airflow rate, the RTU fan could remain off even when vapor compression cooling is need.

In this application, the RTU fan was disabled for the "indirect evaporative only" modes, but it was enabled for vapor compression cooling. In the former case, all airflow through the RTU came from the IEC, while in the latter case, some additional airflow was drawn from the return air path.

To ensure that the RTU has access to outside air when indirect evaporative cooling is not needed, a barometric relief was added in the product air duct. The relief remains closed as long as the plenum is positively pressurized, but can open inward when the IEC fan is off and the RTU fan is on. This would occur, for example, for economizer cooling or for minimum ventilation in heating mode.

For laboratory testing, the outside air damper signal for the RTU was overridden to keep the outside air damper fully open any time the IEC was enabled. In the real world, this effect can be achieved in many ways. In most RTUs, the normal operating position for the outside air damper is adjusted to allow the rooftop unit to supply an appropriate ventilation rate while the IEC is disabled. In such cases, the IEC's damper could be adjusted to achieve an appropriate static pressure while the RTU fan is enabled.

In any case, the unfortunate result is that without active damper controls to adjust the operating pressure for the IEC, the change in downstream resistance associated with RTU fan operation will disrupt the ratio of primary–secondary airflow, and performance for the system will suffer.

Concurrent with this laboratory test, the manufacturer introduced a design for a new IEC that should address some of the challenges discussed here. In particular, the system provides the following:

- Automatically manages an appropriate primary–secondary airflow ratio regardless of changes in the downstream airflow resistance
- Incorporates a heat exchanger bypass so that the IEC fan can serve economizer cooling or serve supply ventilation when cooling is not required

Table 1 describes the six distinct modes of operation that would occur for the design described and tested.

An evaporative condenser pre-cooler manufactured by the same company was also added to the RTU to enhance capacity and improve vapor compression efficiency. This retrofit is illustrated in Figure 2, and operation of the component in each mode is described in Table 1. The pre-cooler is constructed of 12-inch deep cellulose evaporative media with overhead water distribution. The media is mounted in a stainless steel frame designed to fit the condenser inlet face of the conventional rooftop unit. A single pump is used to circulate water from the sump to the top of the media.

The characteristic performance for this condenser pre-cooler as an independent measure is evaluated in detail through a separate study: *Laboratory Testing of*

Performance Enhancements for Rooftop Packaged Air Conditioners [Davis 2015]. There, the pre-cooler was tested according to a methodology recently proposed by American Society of Heating Refrigeration and Air Conditioning Engineers (ASHRAE) SPC 212 [ASHRAE 2015], and characteristic performance metrics for the device are compared directly to other condenser pre-coolers. The laboratory results presented in this report focus on the condenser pre-cooler's net impacts to capacity and efficiency for the integrated retrofit package.

TABLE 1. MODES OF OPERATION FOR THE INDIRECT EVAPORATIVE AND ROOFTOP UNIT COMBINATION

MODE	RTU INDOOR FAN	COMPRESSORS	IEC FAN SPEED	BAROMETRIC OUTSIDE AIR INLET	RETURN AIRFLOW	IEC PUMPS	CONDENSER PRE-COOLER PUMP[3]
Off	OFF	OFF	OFF	NA	NA	OFF	OFF
Ventilation or Economizer	ON	OFF	OFF	Open	Yes	OFF	OFF
IEC	OFF[1]	OFF	MIN-100%[2]	Closed	No	ON	OFF
IEC and direct expansion unit 1 (DX1)	ON[1]	1	100%	Closed	Yes	ON	ON
IEC and DX2	ON[1]	2	100%	Closed	Yes	ON	ON
Heating	ON	OFF	OFF	Open	Yes	OFF	OFF

1. If airflow resistance through the rooftop unit and ductwork system is low enough, the IEC can operate without the RTU indoor fan. If the IEC product flow rate is large enough, the compressors could operate without the RTU indoor fan. This determination must be made by careful design.
2. IEC fan speed usually varies with the cooling load. Sequence of operations should seek to achieve the following:
    - When IEC is the most efficient mode, ensure that the fan reaches full speed before compressors are enabled
    - Prevent the IEC fan speed from dropping below the minimum ventilation rate
3. Condenser pre-cooling should only operate above an outside air temperature threshold where it is most useful.

ASSESSMENT OBJECTIVES

The main objectives of this project were as follows:

- Provide reliable performance data that can serve as the basis for annual energy modeling and design of efficiency programs
- Introduce system functions and operation to utility program managers, design engineers, and end users who may benefit from its application
- Advance CA-EESP goals for the broad market application of climate-appropriate commercial cooling technologies
- Describe important attributes of the technology that may not be readily apparent, but should be considered in application
- Scrutinize equipment design to understand the best applications to derive energy and peak savings and identify any scenario or mode of operation where the equipment may not achieve savings over the incumbent technology
- Identify opportunities for additional improvements for the technology

To accomplish these objectives, the project team from UC Davis Western Cooling Efficiency Center (WCEC) worked closely with the manufacturer to develop a strong technical understanding of the equipment from the inside-out and create a design for systems integration that would resemble field application of the technology. The team then designed the laboratory test experiments to evaluate the performance of the system as a whole and characterize performance of all major subcomponents in each mode of operation.

This testing was part of a larger series of laboratory evaluations to study retrofit efficiency measures for commercial air conditioners, *Laboratory Testing of Performance Enhancements for Rooftop Packaged Air Conditioners* [Davis 2015].

TEST METHODOLOGY

This project conducted laboratory measurements of an existing RTU (nominal capacity of 8½ tons) with and without the suite of climate-appropriate retrofit discussed above. The test team conducted thermal tests at eight outdoor air conditions for several different operating modes to develop a thorough performance map for the system with and without each retrofit. The team also measured airflow characteristics through a series of separate tests that evaluated fan power and static pressure across a range of fan speeds, internal damper positions, and external resistance conditions.

The team analyzed laboratory measurements to compare performance in each operating mode and describe the efficiency benefits of the retrofit across a range of outside air conditions. The team used these findings to evaluate the efficiency measure, identify technical challenges, and recommend technical opportunities for future projects. Finally, the team imposed the maps of system performance with and without the climate-appropriate retrofit on a standard annual system load profile to estimate the integrated energy efficiency ratio (IEER) for the system as a means of estimating the overall potential for cooling energy savings.

The sections below provide details on specific aspects of the evaluation.

LABORATORY FACILITY

All testing was performed in Pacific Gas and Electric Company's Advanced Technology Performance Lab in San Ramon, California. The test apparatus consists of two side-by-side environmental chambers designed in accordance with ASHRAE Standard 37 [ASHRAE 2009]. Each chamber has an independent conditioning system controlled to maintain the chamber's target temperature and humidity conditions.

The facility has two code tester airflow measurement stations. These systems measure differential static pressure across an array of nozzles and can determine airflow with less than 1% uncertainty. Each code tester is paired with a variable speed fan adjusted to compensate for the added resistance from the flow measurement system. For these tests, the first code tester was used to measure supply airflow rate, while the second was used to measure condenser exhaust airflow rate. Figure 3 illustrates the laboratory facility layout.

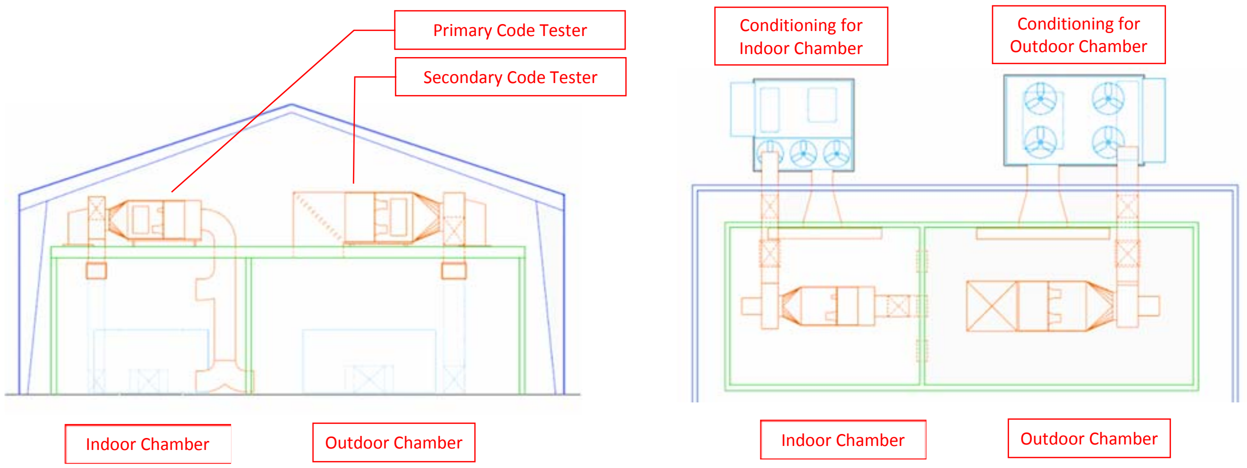


FIGURE 3. LABORATORY LAYOUT – SECTION VIEW

The indirect evaporative air conditioner, and the rooftop unit with condenser air pre-cooler were placed entirely inside the "outdoor" chamber, as illustrated in Figure 4. Air at the prescribed test conditions is drawn directly from the outdoor chamber into the condenser inlet, the economizer inlet, and the indirect evaporative cooler inlet. Supply air from the rooftop unit was ducted to one code tester, and condenser exhaust airflow was ducted to the other code tester. Return air for the rooftop unit was drawn from the indoor chamber. Air volume that is drawn out of each chamber by the test equipment is made up by supply from the conditioning unit for each chamber. This air volume originates partly from outside and partly from the outlet of each code tester.

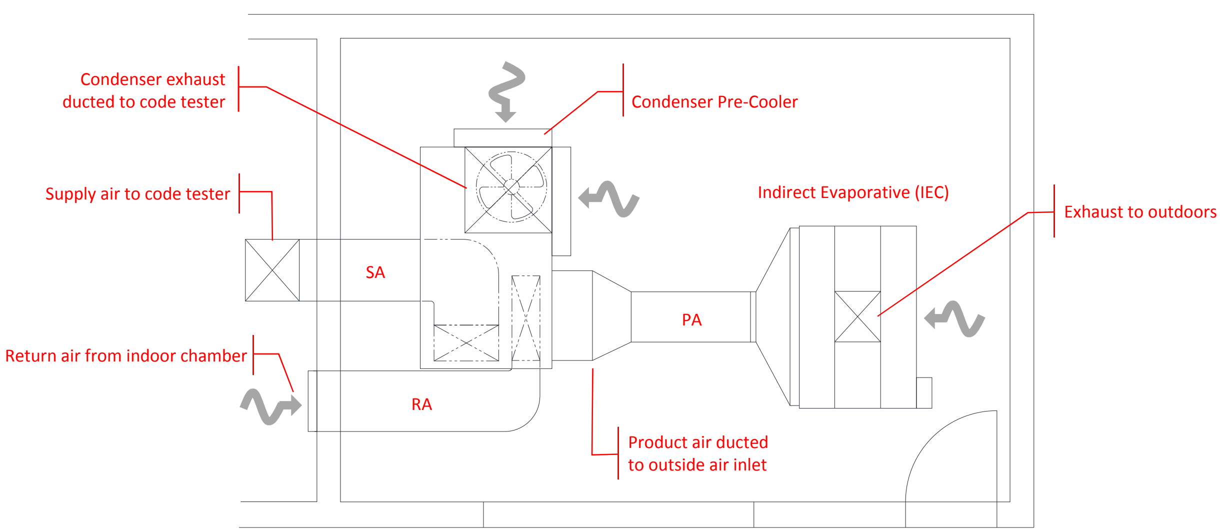


FIGURE 4. PHYSICAL LAYOUT OF ROOFTOP UNIT AND INDIRECT EVAPORATIVE UNIT IN OUTDOOR CHAMBER

INSTRUMENTATION SCHEME

Measurements for this laboratory evaluation were designed mainly to capture the net thermodynamic performance for the rooftop unit and indirect evaporative air conditioner. Accordingly, the majority of instrumentation was located in the external ductwork or at the open inlets to each machine. In addition to taking psychrometric conditions and air flow rates, laboratory measurements captured whole-system electrical-use characteristics, as well as makeup water consumption for each system. The laboratory tests captured a number of internal variables as well, such as refrigerant circuit temperatures and pressures, sub-component electric current uses, and some internal temperatures.

Table 2 documents the instrumentation used for each point of measurement in this laboratory study. Figure 5 illustrates the location for each measurement on a system schematic.

Temperature and pressure transmitters were calibrated against laboratory standards through the data acquisition system prior to testing. For the temperatures, the calibration included a low point using an ice bath (32°F) and a high point using a hot block calibrator (120°F). The raw measurements were adjusted to match the reading from a secondary temperature standard placed in the same environment. The four dew-point sensors had received a factory calibration in December 2012.

All of the instruments were connected to signal conditioning modules based on the National Instruments C-series architecture, connected to six CompactRIO chassis. The modules included different units for resistance temperature detectors (RTDs), thermocouples, voltage, current, and pulse counting, plus both analog and digital output modules for the room conditioning systems. Two of the CompactRIO chassis were programmed to read from the weather station and the condensate scale through their RS-232 connections and create network-shared variables from the text streams. The default chassis internal scan rate for reading the module inputs is 10 Hz; the weather station and scale are updated once every second.

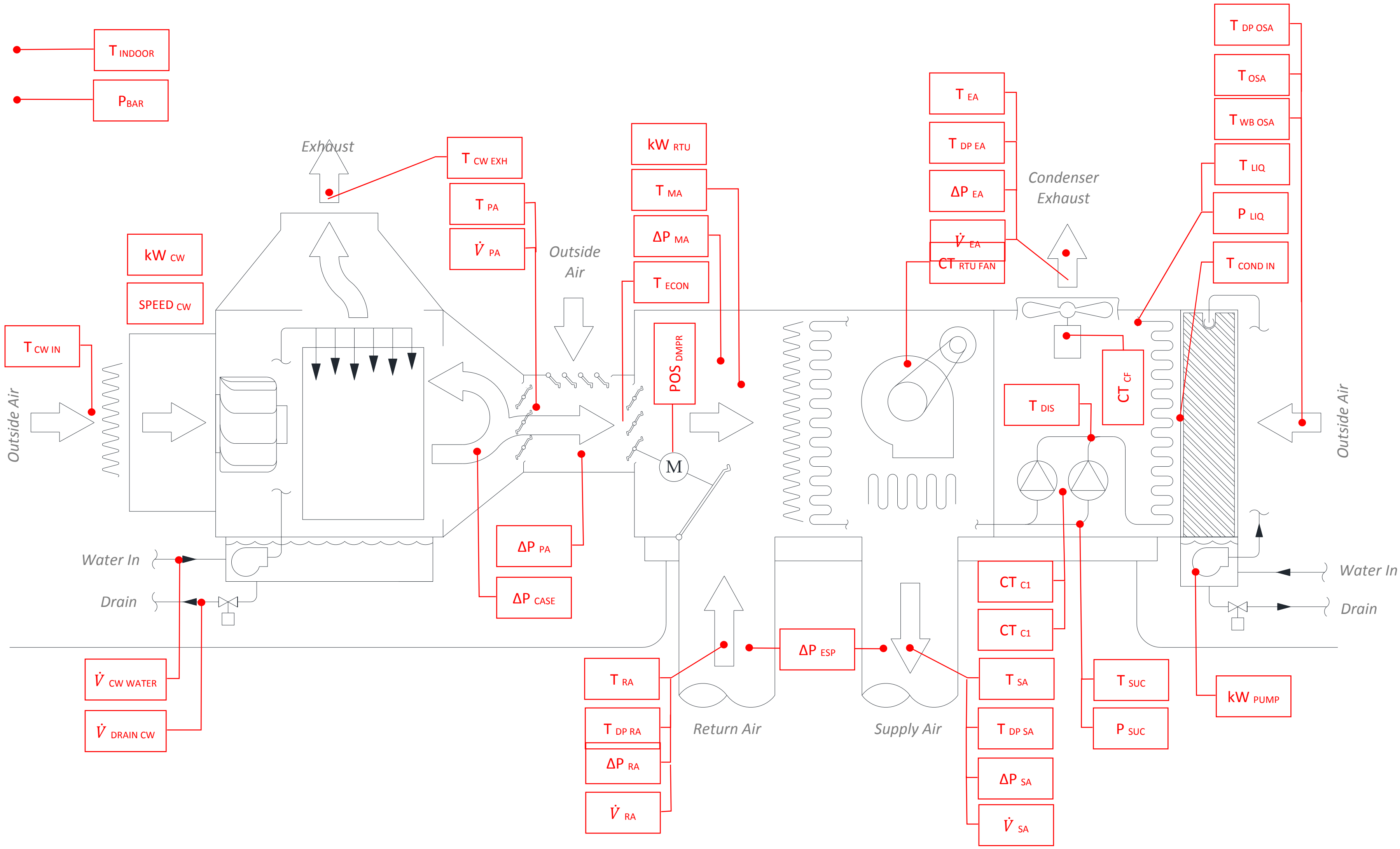


FIGURE 5. INSTRUMENTATION SCHEMATIC FOR LABORATORY TEST

The six Compact-RIO chassis communicate over an Ethernet network to a central host computer, which ran a custom data acquisition and control program developed with National Instruments LabVIEW™ graphical programming language. The program acquired readings from the chassis at a rate of 2 Hz; applied calibration scaling and maintained a running average for each measurement; and logged the averages to a file every 15 seconds. The scaled values and other calculated values, displayed on screen in text and graphical form, were used to generate feedback control signals to the space conditioning systems.

The logged data was saved in an ASCII text format that is easily imported into Microsoft Excel for analysis. A macro was run on the raw data file to apply formatting, calculate statistics, and create trend charts. The result was then analyzed to isolate a period of stable operation. For most of the tests, the target period duration was 30 minutes, although shorter duration periods were accepted when thermal stability was not critical (e.g., fan performance mapping) or on rare occasions when some operating anomaly reduced the acceptable data set.

Once the steady-state period was identified, the statistics (average, standard deviation, range) were isolated to just this period and then copied over to another spreadsheet with one row per test. Operating performance metrics were then calculated from these values, according to equations described in section *Definition and Calculation of Performance Metrics* [Davis 2015].

TABLE 2. MEASUREMENTS AND INSTRUMENTATION FOR LABORATORY TESTS

TAG	MEASUREMENT	INSTRUMENT	MAKE	ACCURACY
P_{BAR}	Barometric pressure	Multi-function weather station on roof of building	Vaisala WTX520	±0.007 PSIA (±0.5 hPa)
T_{INDOOR}	Indoor room dry-bulb temperature	Average of 4 fast-response resistance temperature detectors (RTDs) in room free air	Burns Engineering	±0.2°F
T_{RA}	Return air dry-bulb temperature	Average of 4 fast-response RTDs inserted through wall of duct attached to test unit return	Burns Engineering	±0.2°F
$T_{DP\ RA}$	Return air dew point temperature	Chilled mirror dew-point sensor in return air duct near inlet of unit	General Eastern Hygro-M2+	±0.36°F
ΔP_{RA}	Return air static pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the duct entering the unit	Rosemount 3051C	±0.04% span (-2–2 in)
$\dot{V}_{RA}$	Return airflow station	Full duct averaging pitot tube array (air monitor fan-E) with low range differential pressure sensor	Air Monitor Veltron II	±0.1% span (0 to 0.1 in)
T_{ECON}	Air temperature at economizer inlet	Average of four type-T thermocouples mounted on units outside air damper intake	Therm-X	±0.5°F
T_{MA}	Mixed air temperature	Average of 12 Type-T thermocouples averaging RTD	Therm-X Minco S457PE	±0.5°F 0.25% read
ΔP_{MA}	Mixed air static pressure	Pressure transmitter attached to port through side of RTU into mixed air plenum.	Rosemount 3051C	±0.04% of span (-2–1 in)
T_{SA}	Supply air discharge dry-bulb temperature	Average of 6 fast-response RTDs inserted through wall of duct attached to test unit return	Burns Engineering	±0.2°F

Tag	Measurement	Instrument	Make	Accuracy
$T_{DP\ SA}$	Supply air discharge dew point temperature	Chilled mirror dew-point sensor	General Electric Optica	±0.36°F
ΔP_{SA}	Supply air static pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the duct leaving the unit	Rosemount 3051C	±0.04% span (-1–2 in)
$\dot{V}_{SA}$	**Supply airflow station differential pressure**	**Pressure transmitter attached to manifolded pressure taps at center of each side of the flow box on both sides of the nozzle partition**	**Rosemount 3051C**	**±0.04% span (0–3 in)**
	Supply airflow station upstream static pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the flow box upstream of the nozzle partition	Rosemount 3051C	±0.04% of span (-3–3 in)
	Supply airflow station dry bulb temperature	Single fast-response RTD upstream of nozzles	Burns Engineering	±0.2°F
T_{OSA}	Outside air intake dry-bulb temperature	Average of 8 fast-response RTDs arrayed across the outside air intake.	Burns Engineering	±0.2°F
$T_{WB\ OSA}$	Outside air wet-bulb temperature	Average of 4 fast-response RTDs each enclosed in a wetted wick and with a fan for air movement	Burns Engineering	±0.2°F
$T_{DP\ OSA}$	Outside air dew point temperature	Chilled mirror dew-point sensor	General Eastern Hygro-M4	±0.36°F
ΔP_{ESP}	Unit pressure drop	Pressure transmitter connected between the supply and return manifolds	Rosemount 3051C	±0.04% span (-4–4 in)
$T_{COND\ IN}$	Pre-cooler outlet temperature	Single RTD inserted between the pre-cooler and condenser coil.	Burns Engineering	±0.2°F
kW_{PUMP}	Pre-cooler pump power, voltage, and current	3-element true-RMS power meter with digital outputs of power, voltage, and current for each phase	Yokogawa WT330	±(0.1% read +0.1% range)
$\dot{V}_{CW\ WATER}$	IEC makeup water	Positive displacement water meter with analog output for flow rate and pulse output for totalization	Badger M25	±1.5% f.s.
$\dot{V}_{CW\ DRAIN}$	IEC weigh tank	Catch basin on electronic scale	Measuretek	±0.2 lb
$T_{CW\ IN}$	IEC outdoor air temperature	Single RTD attached to air intake	Burns Engineering	±0.2°F
T_{PA}	IEC product air temperature	Single RTD inserted into duct at unit discharge. (Normally matched 4 T/Cs at economizer intake)	Burns Engineering	±0.2°F
$T_{CW\ EXH}$	IEC exhaust air temperature	Single RTD attached to air discharge	Burns Engineering	±0.2°F
ΔP_{CASE}	IEC case pressure	Pressure transmitter attached to case tap	Rosemount 3051C	±0.04% span (-1–2 in)
$\dot{V}_{PA}$	IEC product airflow monitor	Full duct averaging pitot tube array (Air Monitor Fan-E) with low range differential pressure sensor	Air Monitor Veltron II	±0.1% span (0–0.1 in)

	IEC product airflow monitor	Full duct averaging pitot tube array (Air Monitor Fan-E) with low range differential pressure sensor	Ashcroft XLDP	±0.25% span (0–1.0 in)
ΔP_{PA}	IEC product air pressure	Pressure transmitter attached to static pressure tap of pitot tube array	Ashcroft XLDP	±0.25% span (-1–1 in)
kW_{CW}	IEC supply power, voltage, and current	3-element true-RMS power meter with digital outputs of power, voltage, and current for each phase	Yokogawa WT330	±(0.1% read +0.1% range)
T_{EA}	Exhaust air dry-bulb temperature	Average of 4 fast-response RTDs inserted through wall of duct attached to test unit exhaust	Burns Engineering	±0.2°F
$T_{DP\ EA}$	Exhaust air dew point temperature	Chilled mirror dew point sensor	General Eastern Hygro-M2	±0.36°F
ΔP_{EA}	Exhaust air static pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the duct leaving the unit	Rosemount 3051C	±0.04% span (-2–2 in)
$\dot{V}_{EA}$	Exhaust airflow station differential pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the flow box on both sides of the nozzle partition	Rosemount 3051C	±0.04% span (0–3 in)
	Exhaust airflow station upstream static pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the flow box upstream of the nozzle partition	Rosemount 3051C	±0.04% span (-3–3 in)
	Exhaust airflow station dry bulb temperature	Single fast-response RTD upstream of nozzles	Burns Engineering	±0.2°F
P_{SUC}	Compressor suction pressures (2 circuits)	Pressure transmitter attached to compressor suction (vapor) line Schrader valve	Rosemount 3051C	±0.04% span (0–300 psig)
P_{LIQ}	Condenser outlet pressures (2 circuits)	Pressure transmitter attached to liquid line Schrader valve	Rosemount 3051C	±0.04% span (0–400 psig)
T_{SUC} T_{DIS} T_{LIQ}	Refrigerant temperatures (compressor suction and discharge, condensed liquid)	Type-T thermocouples (6 total) clamped to outside of refrigerant tubing with thermal paste and wrapped in insulation	Therm-X	±0.5°F
kW_{RTU}	Unit supply power, voltage, and current	3-element true-RMS power meter with outputs for total power, 3-phase voltage, and 3-phase current	Yokogawa 2533	±0.2% read ±0.1% f.s.
$SPEED_{CW}$	IEC Fan Speed	Speed signal from IEC controller		
POS_{DMPR}	RTU outside air damper position	DC voltage signal from damper actuator		
CT_{C1} CT_{C2} CT_{CF} $CT_{RTU\ FAN}$	Sub-components line current	Clamp-on current transmitter on one leg of the power feeding each of two compressors, two fans (fan, cond. fan, compressors 1 and 2)	NK Technologies ATR1	±1% of f.s. (±0.2 A)

Design of Experiments

To characterize performance for the test unit, laboratory tests were arranged into two groups:

- Airflow-only tests to map performance for the RTU and IEC fans
- Thermal performance tests to map performance of the combined system in all modes of operation and across a range of environmental operating conditions

The combination was tested across the range of psychrometric conditions recorded in Table 3 and illustrated on a psychrometric chart in Figure 6. Table 4 describes the different modes of operation evaluated.

Table 3. Range of Test Conditions Used for Design of Experiments

	Dry-Bulb Temperature (T_{DB}) (°F)	Wet-Bulb Temperature (T_{WB}) (°F)	Corresponds to
Return Air	78	64	Western Return Air
Outside Air Conditions	105	73	Western "Peak"
	95	75	ANSI/AHRI 340/360 "Nominal" Rating & IEER 100% Capacity
	90	64	Western "Annual"
	82	73	"Warm Humid"
	81.5	66.3	ANSI/AHRI 340/360 IEER 75% Capacity
	78	58.5	"Warm Dry"
	68	57.5	ANSI/AHRI 340/360 IEER 50% Capacity
	65	52.8	ANSI/AHRI 340/360 IEER 25% Capacity

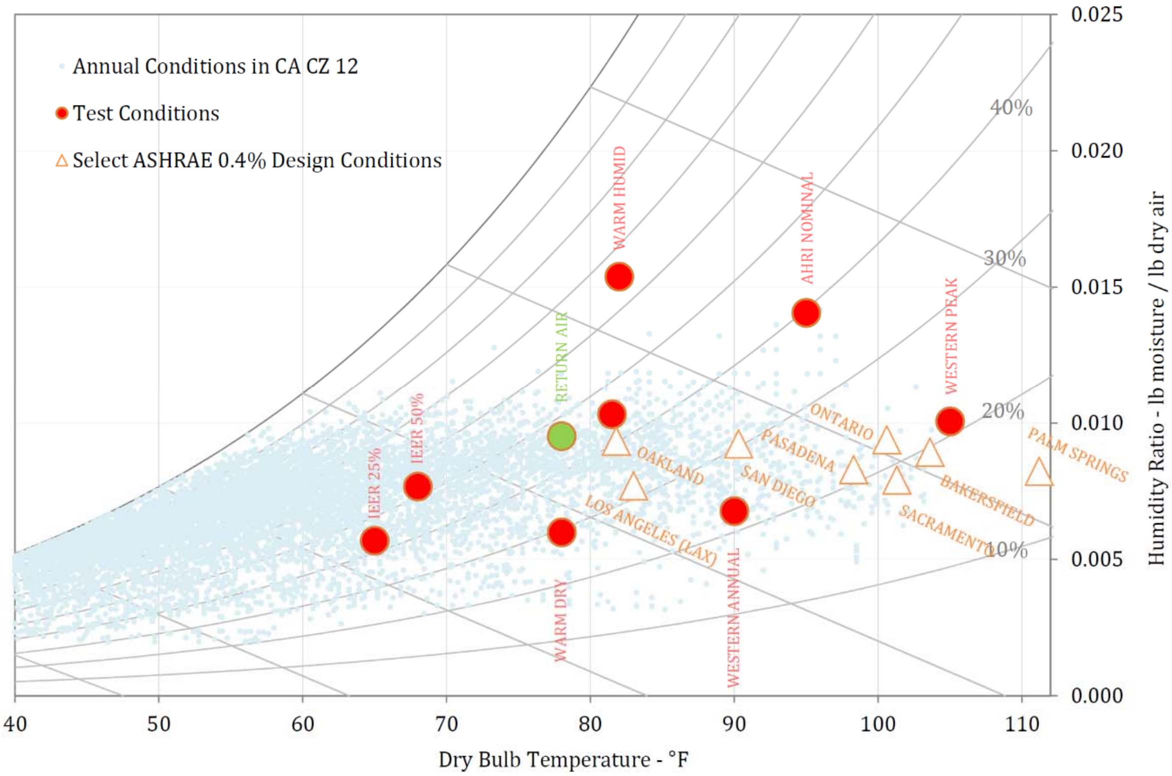


FIGURE 6. PSYCHROMETRIC CHART – RANGE OF TARGET TEST AND DESIGN CONDITIONS FOR SEVERAL CITIES

Table 4. Range of test conditions used for design of experiments

Scenario		IEC	Compressor 1	Compressor 2	RTU Indoor Fan	Open OSA Damper	Condenser Pre Cooler	Ventilation Airflow (CFM)[1]	IEC Airflow (CFM)[1]
Second stage vapor compression									
DX2	No Vent.	☐	☑	☑	☑	☐	☐	0	NA
	Ventilation	☐	☑	☑	☑	☑	☐	900	NA
Second stage vapor compression with condenser pre-cooler									
DX2 + PC	No Vent.	☐	☑	☑	☑	☐	☑	0	NA
	Ventilation	☐	☑	☑	☑	☑	☑	900	NA
Indirect evaporative cooling									
IEC	Low Speed	☑	☐	☐	☐	☑	☐	1350	
	Mid Speed	☑	☐	☐	☐	☑	☐	2100	
	High Speed	☑	☐	☐	☐	☑	☐	2600	
Indirect evaporative cooling with second-stage vapor compression									
IEC + DX2	Low Speed	☑	☑	☑	☑	☑	☐	1400	
	High Speed	☑	☑	☑	☑	☑	☐	2950	
Indirect evaporative cooling, first-stage vapor compression (no RTU fan), with condenser pre-cooler									
IEC + DX1 + PC	High Speed	☑	☑	☐	☐	☑	☑	2600	
Indirect evaporative cooling, second stage vapor compression, with condenser pre-cooler									
IEC + DX2 +PC	Low Speed	☑	☑	☑	☑	☑	☑	1400	
	High Speed	☑	☑	☑	☑	☑	☑	2950	

1. For all tests that include IEC, all ventilation comes from the IEC. For tests without IEC, ventilation rate come from rooftop unit outside air damper.

Definition and Calculation of Performance Metrics

The following sections defines performance metrics and presents the calculations used to measure these metrics.

Calculating Wet Bulb Effectiveness

Wet bulb effectiveness (WBE) describes the extent to which an evaporative cooler is able to cool toward the wet bulb temperature (T_{WB}) of the inlet air as shown in Equation 1. The metric tends to remain steady for a given system configuration across a wide range of meteorological conditions. WBE is the most common metric to describe performance of an evaporative system and is used as an input for most building energy simulation tools.

EQUATION 1: WET BULB EFFECTIVENESS

$$WBE = \frac{T_{DB\ 1st\ inlet} - T_{DB\ 1st\ out}}{WBD_{1st\ inlet}} = \frac{T_{DB\ 1st\ inlet} - T_{DB\ 1st\ out}}{T_{DB\ 1st\ inlet} - T_{WB\ 1st\ inlet}}$$

This metric, which has traditionally been used to describe performance of direct evaporative coolers, 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.

Describing performance in terms of WBE offers good conceptual comparison against conventional evaporative coolers. However, since the metric does not align with the physical heat transfer mechanisms active in the indirect evaporative heat exchanger, it usually does not provide a useful empirical correlation for use in system modeling. Alternative metrics use the wet bulb temperature of the secondary air stream or the dew-point temperature of the primary air stream as the theoretical potential for an effectiveness ratio.

The conventional WBE metric is used to describe equipment performance here, as well as an indirect WBE that accounts for the wet bulb potential in the secondary air stream, as in Equation 2:

EQUATION 2: INDIRECT WET BULB EFFECTIVENESS

$$Ind.WBE = \frac{T_{DB\ 1st\ inlet} - T_{DB\ 1st\ out}}{Ind.WBD} = \frac{T_{DB\ 1st\ inlet} - T_{DB\ 1st\ out}}{T_{DB\ 1st\ inlet} - T_{WB\ 2nd\ inlet}}$$

CALCULATING COOLING CAPACITY

The system cooling capacity for the 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, including what is lost due to fan heat.

EQUATION 3: COOLING EFFECTIVENESS

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

In this equation, $h^{\star}_{MA}$ is the specific enthalpy of the 'virtual' mixed air, a parameter that does not physically exist. Generally, the system cooling capacity for a conventional rooftop unit is measured by the difference between the mixed air enthalpy and the supply air enthalpy. However, for the hybrid machine tested here, the ventilation air stream is cooled before it mixes with return air. The 'virtual' mixed air condition represents the combined enthalpy from all inlets to the equipment and so incorporates of the cooling generated by the indirect evaporative cooler. Equation **Error! Reference source not found.** calculates the specific enthalpy for the 'virtual' mixed air condition.

EQUATION 4: SPECIFIC ENTHALPY (MIXED AIR)

$$h_{MA}^{\star} = h_{RA} + \mathrm{OSAF} \cdot (h_{OSA} - h_{RA})$$

Ambient humidity in most western climates is sufficiently low that dehumidification is not necessary to maintain occupant comfort [ASHRAE 2010] in most commercial buildings. Therefore, the assessment presented here focuses on the system's ability to produce sensible cooling. Further, since thermostat controls typically only respond to temperature and not humidity, it is not appropriate to credit any latent cooling when considering of the performance advantages for technology studied here. The net sensible system cooling capacity is determined according to Equation **Error! Reference source not found.**.

EQUATION 5: SENSIBLE COOLING CAPACITY

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

Concomitantly, the latent system cooling is determined as shown in Equation **Error! Reference source not found.**:

EQUATION 6: LATENT COOLING CAPACITY

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

The system cooling metric represents the net cooling that is actively produced by the whole equipment combination. 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 because 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 **Error! Reference source not found.**, describes the net thermal impact to the room. When outside air is cooler than the return air, the room cooling capacity may be greater than the system capacity.

EQUATION 7: ROOM COOLING CAPACITY

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

CALCULATING ENERGY EFFICIENCY RATIO

Energy efficiency for the equipment at any given operating condition is expressed as the ratio of useful thermal capacity delivered (in units of *kbtu/h*) to electrical power consumed by the system (in units of *kW*), as shown in Equation **Error! Reference source not found.**.

EQUATION 8: ENERGY EFFICIENCY RATIO

$$EER = \frac{Thermal\ Energy\ Delivered}{Electrical\ Energy\ Consumed} = \frac{\dot{H}}{\dot{E}_{system}} = \left\{\frac{kbtu}{kWh}\right\}$$

The energy efficiency ratio (EER) values presented in this report should not be confused with or compared directly to the AHRI nominal EER, seasonal energy efficiency ratio (SEER), or IEER values for conventional air conditioners because those metrics describe performance at specific conditions that cannot be replicated for the systems evaluated in this study. For this report, EER is presented as a generic metric that varies with conditions and with the frame of reference.

The efficiency assessment presented here only credits sensible cooling capacity, since dehumidification is usually not required or controlled for in commercial cooling applications in California. The sensible EER can be expressed as in Equation **Error! Reference source not found.**:

EQUATION 9: SENSIBLE ENERGY EFFICIENCY RATIO

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

Performance results are also described in terms of the sensible system EER (Equation **Error! Reference source not found.**) and the sensible room EER (Equation **Error! Reference source not found.**). The first metric considers how much energy is consumed to generate a cooling across the machine; the second considers the ratio of that energy consumption to the cooling effect on the room:

EQUATION 10: SENSIBLE ENERGY EFFICIENCY RATIO (SYSTEM)

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

and

EQUATION 11: SENSIBLE ENERGY EFFICIENCY RATIO (ROOM)

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

RESULTS AND DISCUSSION

Laboratory testing of the IEC and condenser pre-cooler retrofit package was conducted at 8 climate conditions and in 11 different operating modes. The results presented here focus on airflow characteristics, sensible cooling capacity, total system energy use, and water use for evaporation. The performance and efficiency in each mode is compared to the measured performance of the rooftop unit operating without the evaporative retrofits. These results also describe characteristics such as WBE for the IEC alone.

FAN MAPPING

Figure 7 plots the results from IEC airflow testing across a range of fan speeds, with different balancing damper positions, and with the RTU fan ON. Results are also shown for a range of fan speeds with the damper fully open and the RTU fan OFF. The IEC fan speed was managed by the manufacturer-supplied digital room controller, set to ventilation mode, and adjusted from settings 2–10.

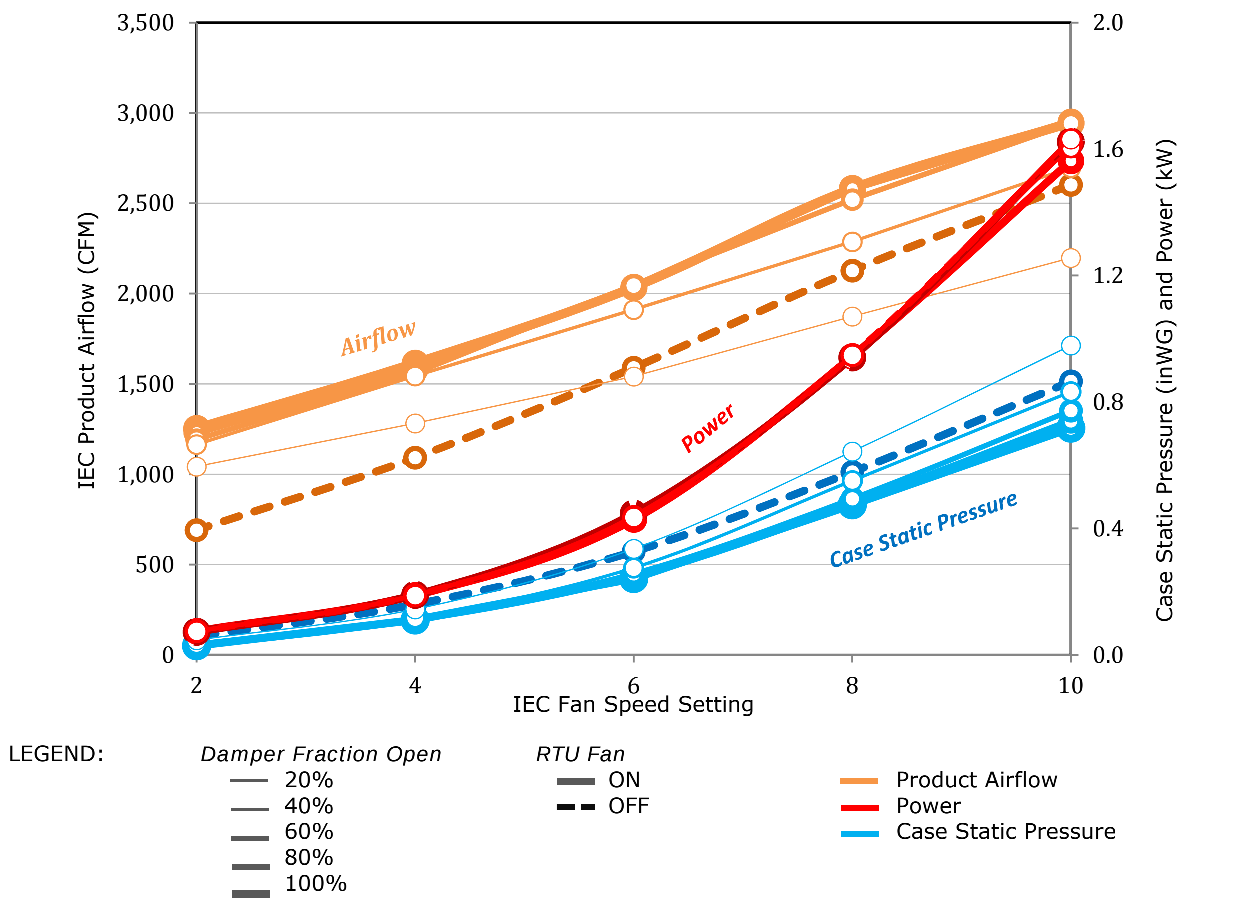


FIGURE 7. MAP OF IEC FAN AIRFLOW, POWER CONSUMPTION, AND CASE STATIC PRESSURE ACROSS A RANGE OF FAN SPEEDS, WITH DIFFERENT DAMPER OPENINGS AND WITH THE RTU FAN ON AND OFF

The map presents the product airflow rate, the system power consumption, and the static pressure measured at the outlet of the indirect evaporative heat exchanger, upstream of the product air balancing damper. (See Figure 5 for the case static pressure measurement location.) The values presented for power represent to the real power consumption for the IEC, including ancillary power for controls. The values for airflow represent the product airflow rate measured by the pitot tube airflow measurement station in the duct connecting the IEC and the RTU (not the IEC fan flow rate).

As discussed previously, IEC performance is sensitive to the downstream airflow resistance because the static pressure at the primary outlet of the heat exchanger drives flow through the secondary passages of the heat exchanger. The downstream airflow resistance can be adjusted with a balancing damper located at the IEC product air outlet, as indicated in Figure 2. Manufacturer documentation indicates that the appropriate balance between primary and secondary airflow should be achieved when the dampers are adjusted so that the case static pressure operates at 0.72 inch water gauge (inWG) while the IEC is at full speed. As long as the external resistance remains steady, the ratio of primary to secondary airflow should remain the same even while the IEC changes fan speed.

The results from airflow testing indicate that the RTU and distribution ductwork provide more downstream resistance than is needed for the IEC. At full speed, with the balancing damper fully open and the RTU fan OFF, the case static pressure reaches 0.865 inWG. Therefore, to reach the targeted primary–secondary balance the RTU fan would need to operate in combination with the IEC fan at all times. However, since the RTU fan consumes almost as much electricity as the IEC fan, the research team decided that the indirect evaporative only mode would operate without the RTU fan. This results in a lower-than-intended product airflow in this mode and in a lower product air temperature because the primary–secondary airflow ratio decreases.

Figure 7 shows the IEC product airflow rates only and does not represent the supply airflow to the building. The supply air flow rates were different for each thermal test, varying mainly according to the IEC fan speed and whether or not the RTU fan was enabled. For some operating modes, the combination was tested at multiple airflow rates, with the IEC adjusted to speed settings 10 (high), 8 (mid), and 5 (low).

When the RTU fan operates in series with the IEC fan, the supply air to the building is a mixture of product air and return air. When the IEC operates without the RTU fan, some airflow is driven through the supply ductwork, and some passes backward through the return air ductwork. Inevitably, some of the product air also leaks out of the rooftop unit cabinet. For all of the results presented here, cooling capacity and efficiency are calculated according to two assumptions: no air volume is lost to leakage and no thermal losses occur as product air passes through the RTU.

Therefore, for operation in indirect evaporative only mode with the rooftop unit fan OFF, Equation **Error! Reference source not found.** applies:

EQUATION 12: SENSIBLE COOLING CAPACITY, INDIRECT EVAPORATIVE ONLY MODE

$$\dot{H}_{system}^{sensible} = \dot{m}_{PA} \cdot C_p \cdot \left(T_{OSA} - T_{PA}\right)$$

For combined modes with indirect evaporative and vapor compression with the RTU fan ON, Equation **Error! Reference source not found.**:

EQUATION 13: SENSIBLE COOLING CAPACITY, COMBINED MODES (FAN ON)

$$\dot{H}_{system}^{sensible} = \dot{m}_{SA} \cdot C_p \cdot \left(T_{RA} + \frac{\dot{m}_{PA}}{\dot{m}_{SA}} \cdot (T_{OSA} - T_{RA}) - T_{SA}\right)$$

Finally, for indirect evaporative cooling and vapor compression with the RTU fan OFF, Equation **Error! Reference source not found.** applies:

EQUATION 14: SENSIBLE COOLING CAPACITY, COMBINED MODES (FAN OFF)

$$\dot{H}_{system}^{sensible} = \dot{m}_{SA} \cdot C_p \cdot (T_{OSA} - T_{SA}) + (\dot{m}_{PA} - \dot{m}_{SA}) \cdot C_p \cdot (T_{OSA} - T_{PA})$$

COOLING CAPACITY

Figure 8 charts the sensible cooling capacity in each mode of operation for all eight outside air conditions tested. In each plot, the absolute magnitude of each bar represents the system sensible cooling capacity, while the vertical placement of each bar indicates what portion of the system capacity cools ventilation air to room-neutral conditions, and what portion provides sensible room cooling. Zero on the vertical axis corresponds to room-neutral temperature, and the value at the top of each bar indicates the sensible room cooling capacity delivered. The bottom of each blue bar indicates the sensible ventilation cooling load (recorded as negative room cooling). If the outside air is already cooler than the return air, the associated "free" room cooling capacity is recorded in Figure 8 as a green hatch, and additional system cooling begins from that point.

One advantage for the IEC is that system sensible cooling capacity increases as outside air temperature increases. However, the supply air increases concurrently, so the room cooling capacity is lower at high outside air temperatures. This trend is shown in Figure 8: the IEC generates 6.4 tons of cooling at 105°F_{DB}/73°F_{WB} in mode "IEC High," yet the net impact to the room is only 1.3 tons. By comparison, at 78°F_{DB}/58.5°F_{WB}, the sensible system capacity and sensible room cooling capacity are both 3.4 tons.

For any given environmental condition, the system combination tested would need to satisfy the sensible room cooling load and provide adequate ventilation. When multiple modes of operation would satisfy these requirements, controls should choose the most efficient option. Figure 8 allows for direct comparison of the room cooling capacity in each mode. In addition, each chart also overlays a hypothetical room cooling load for the corresponding condition, shown as a red area in the background of each chart. The room cooling load at peak was chosen to match the measured room cooling capacity for the baseline RTU operating with a 20% outside air fraction. The hypothetic room cooling load for other conditions was adjusted to correspond roughly with industry standard methodology for determination of the IEER [AHRI 2011].

Figure 8 indicates that at 105°F_{DB}/73°F_{WB}, the preferred operating mode should be "IEC-Low + DX2 + PC." Indirect evaporative at full speed does not provide enough cooling at this point, even with the addition of the first-stage compressor. Further, although the sensible system capacity for IEC at this condition increases with fan speed, the room cooling capacity remains about the same regardless of speed. This situation occurs because the product air temperature rises as the product airflow rate increases. Thus, for the hybrid operating modes, operating the IEC at part speed in combination with both compressors appears to offer the greatest benefit. The condenser pre-cooler further enhances cooling capacity and efficiency for any mode with vapor compression cooling.

At 90°F_{DB}/64°F_{WB}, the capacity for mode "IEC-High+DX1+PC" matches the hypothetical room load almost exactly. In this mode, the IEC operates at full speed with a single compressor, but no rooftop unit blower. The product airflow rate supplied by the IEC is sufficient to sustain first-stage compressor operation, but probably not large enough to sustain second-stage compressor operation without freezing on the evaporator coil.

Although the capacity for mode "IEC-High+DX1+PC" matches the load at 90°F_{DB}/64°F_{WB}, this may not be the best operating scenario for the condition. Additional benefits can be gained by cycling between higher and lower capacity stages. In fact, if "IEC-High" ran with a 80% runtime fraction in combination with "IEC-Low+DX2+PC" at 20% runtime fraction, the aggregate capacity would match the load and the average room cooling efficiency would be 50% higher than for mode "IEC-High+DX1+PC." The resulting efficiency would allow saving 54% on hourly energy use for cooling compared to the baseline RTU with 20% outside air.

At part-load conditions, the IEC offers outstanding benefits because it allows for operation without compressors altogether. For the hypothetical scenario presented, compressor cooling would not be needed at all below about 80°F. This effect is especially valuable because the largest cumulative portion of annual cooling requirements for commercial buildings occurs in these mid-range conditions [AHRI 2011].

Below about 70°F, the IEC should be able to cover all room cooling loads at low speed, where it operates with EER>45.

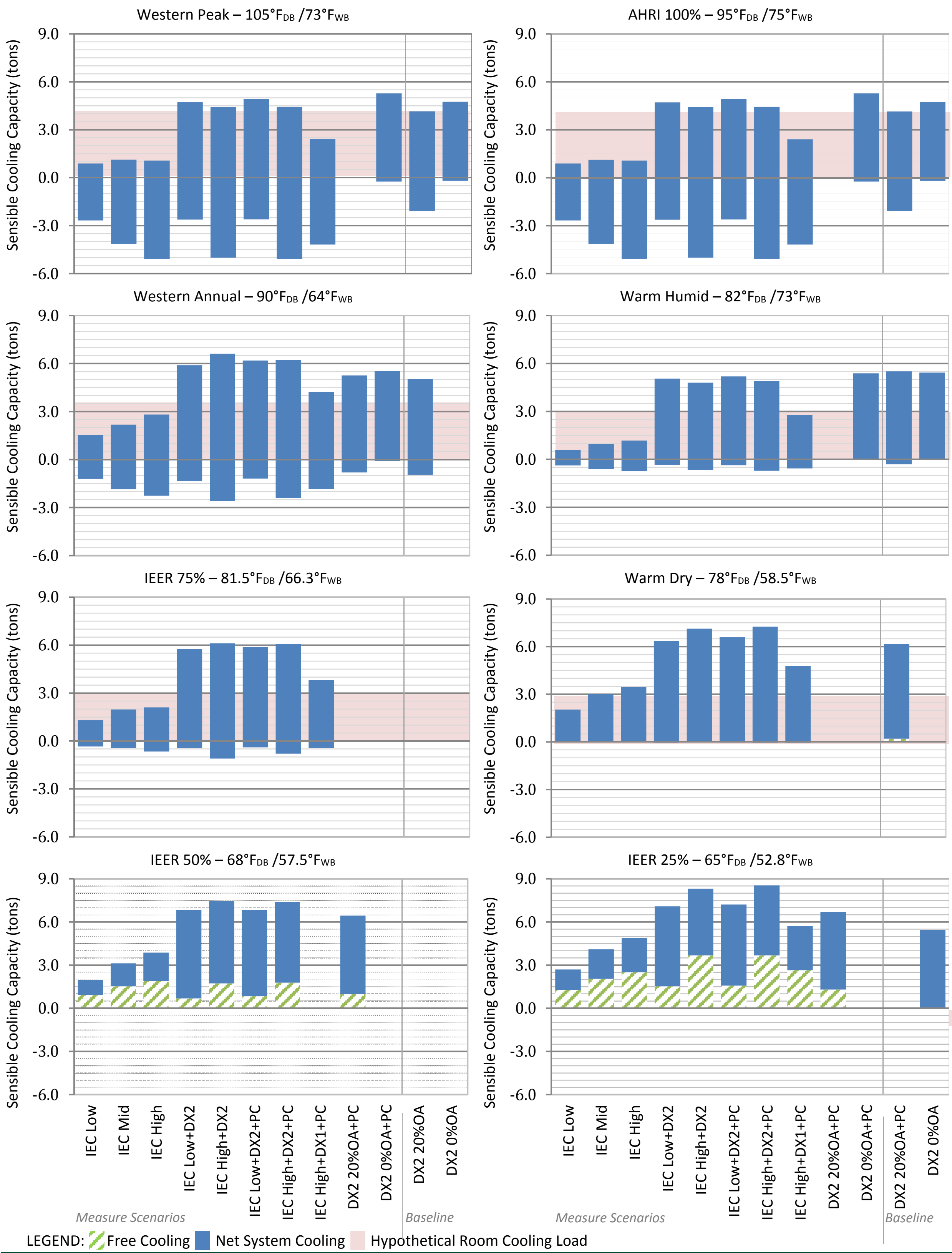


FIGURE 8. SENSIBLE COOLING CAPACITY IN EACH MODE OF OPERATION FOR EIGHT CLIMATE CONDITIONS

SYSTEM POWER AND EFFICIENCY

Figure 9 charts the power draw from each component in each mode of operation for the eight outside air conditions tested. Figure 10 charts the sensible system EER and the sensible room EER for the same tests. Table 5 summarizes test data for each operating mode and climate condition.

Power draw for the IEC is much smaller than that for compressor cooling. As a result, the IEC modes achieve very high efficiency. At 105°F_{DB}/73°F_{WB}, mode "IEC-Low" reaches EER=79.2. That said, an appropriate comparison must consider the room cooling load and the ventilation requirements associated with each climate condition for a particular application. In the scenario considered here, mode "IEC-Low" would provide adequate ventilation, but not enough sensible room cooling to cover loads at 105°F_{DB}/73°F_{WB}.

In every climate condition tested, the IEC and condenser pre-cooler retrofit can increase cooling capacity and improve efficiency. This is even true for the "Warm Humid" scenario 82°F_{DB}/73°F_{WB}, where an aggregate combination of modes "IEC-High" and "DX2–0% OSA" would increase sensible room cooling efficiency by 5%.

For peak conditions of 105°F_{DB}/73°F_{WB}, an aggregate combination of modes "IEC_{LOW}+DX2+PC" and "IEC_{LOW}" would reduce energy use for cooling by 3.1 kilowatt-hours per hour (kWh/hr) compared to the baseline RTU with 20% outside air. This equates to a savings of more than 25%. At 90°F_{DB}/64°F_{WB}, the system can spend less time operating with compressor cooling, so an aggregate combination of "IEC $_{HIGH}$" and "IEC $_{LOW}$+DX2+PC" would reduce energy use for cooling by 4.3 kWh/hr, a 54% savings.

At milder conditions where IEC can carry the entire room cooling load, the fractional savings potential increases substantially. At 78°F_{DB}/58.5°F_{WB}, the IEC could reduce energy consumption by 75%—or 2.7 kWh/hr—by operating in mode "IEC_{HIGH}" for 20% of the time then in mode "IEC_{LOW}" for the remainder of the time. In this scenario, the aggregate sensible room cooling would be 75% of the room cooling load at peak.

Of course, the savings achieved at any given condition depends on the room cooling load profile for the actual application in question. For a building where sensible room cooling load at 78°F_{DB} is only half the sensible room cooling load at peak, the IEC could operate in mode "$IEC_{LOW,}$" reducing energy use for cooling by 85%. The potential for savings increases in scenarios with higher ventilation needs.

Energy saving potential notwithstanding, the hybrid combination evaluated here results in higher electrical power demand than that for the baseline RTU in almost all circumstances because the connected load increases. Thus, electric demand for the IEC, RTU, and pre-cooler operating concurrently will be larger than for the RTU operating alone. This trend is evident in Figure 9: at 105°F_{DB}/73°F_{WB}, mode "DX 2 – 20% OSA" draws 11.2 kW and mode "IEC High + DX2 + PC" draws 12.8 kW.

The implications of this increase in electrical demand will vary by application. For a small commercial building with a single rooftop unit, the measure might increase demand charges because the single largest 15-minute interval energy consumption will likely occur with all components operating. However, for larger customers with many RTUs, because all systems are far less likely to operate simultaneously, peak demand charges would not increase. The condenser pre-cooler plays a valuable role in these respects since it increases cooling capacity while also reducing instantaneous electrical demand. In any circumstance, the authors recommend that efforts to apply the measure should consider means to reduce connected load. For example, in certain circumstances, the addition of the IEC could justify permanent disconnection of one compressor.

For the grid as a whole, since the measure reduces aggregate energy consumption for each site, it will provide needed benefits for network-wide peak demand reduction.

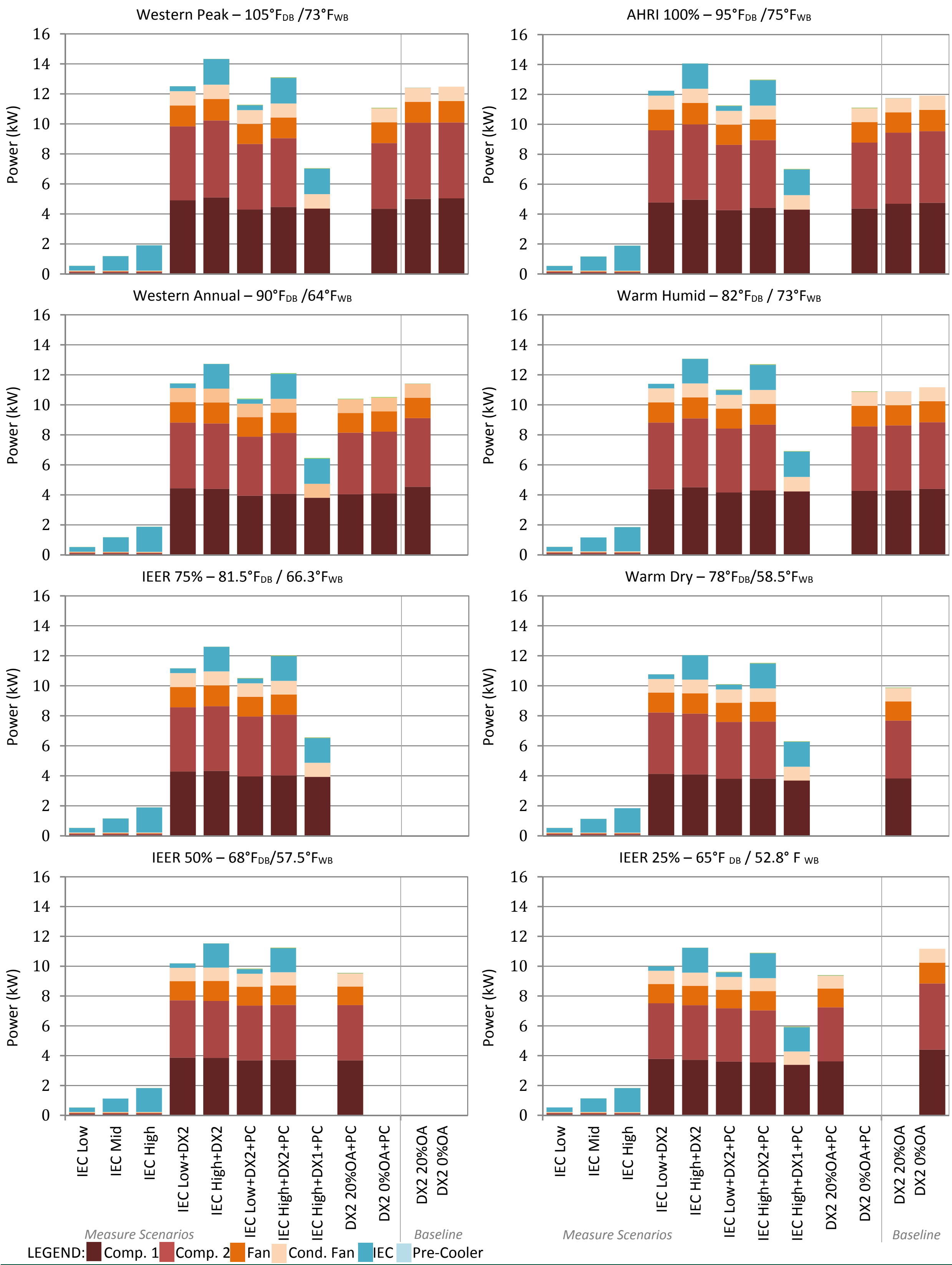


FIGURE 9. ELECTRIC POWER CONSUMPTION FROM EACH COMPONENT IN EACH MODE OF OPERATION

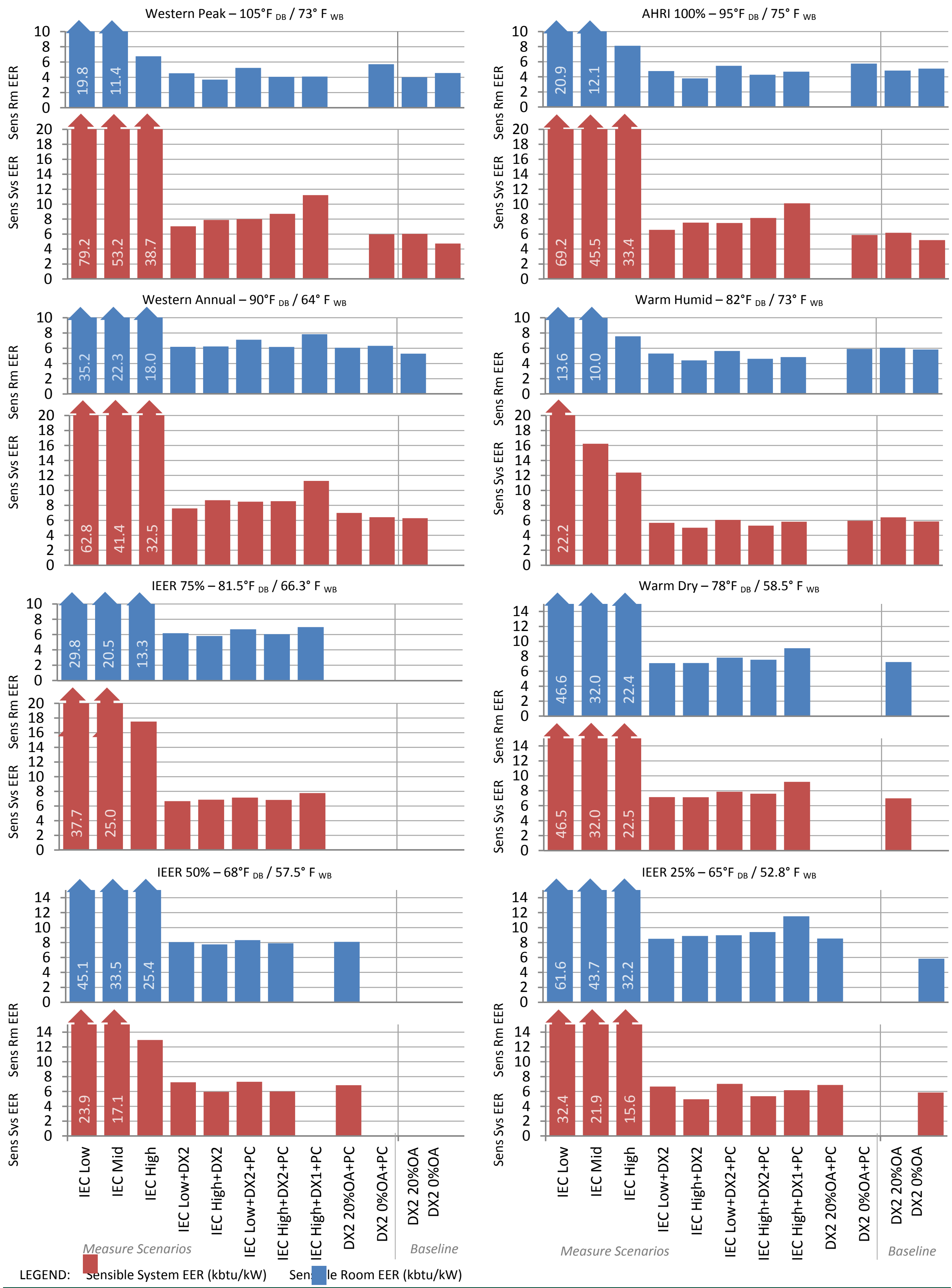


FIGURE 10. ENERGY EFFICIENCY RATIO IN EACH MODE OF OPERATION

TABLE 5. SUMMARY PERFORMANCE DATA TABLE FOR TESTS IN EACH MODE AND EACH MODE AND CLIMATE CONDITION

Test Condition	Mode	Outside Air			Supply Air			Return Air			Mixed Air			Condenser Exhaust Air			IEC Product Air				IEC Exhaust Air	
		T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	ΔP_{CASE} (inWG)	T_{DB} (°F)	ω (lb/lb)
Western Peak (105°F $_{DB}$/73°F $_{WB}$)	IEC Low	104.9	0.0105	1110													67.0	0.0105	1110	0.23	87.5	0.0247
	IEC Mid	104.8	0.0108	1730													69.4	0.0108	1730	0.55	87.2	0.0243
	IEC High	105.2	0.0110	2110													71.2	0.0110	2110	0.84	88.1	0.0239
	IEC Low+DX2	105.1	0.0105	1130	58.2	0.0089	2630	78.0	0.0097	1580	73.6	0.0101	2630	124.8	0.0102	5876	67.5	0.0105	1130	0.24	87.3	0.0241
	IEC High+DX2	105.1	0.0113	2480	60.4	0.0099	2780	78.0	0.0101	640	73.1	0.0110	2780	126.2	0.0111	5831	71.8	0.0113	2480	0.74	87.7	0.0221
	IEC Low+DX2+PC	105.0	0.0123	1130	57.4	0.0090	2640	78.0	0.0101	1610	74.8	0.0110	2640	101.8	0.0180	5806	70.3	0.0123	1130	0.24	88.4	0.0250
	IEC High+DX2+PC	105.0	0.0128	2480	60.3	0.0101	2780	78.0	0.0103	730	74.8	0.0122	2780	103.7	0.0184	5790	73.9	0.0128	2480	0.74	88.7	0.0231
	IEC High+DX1+PC (no RTU fan)	105.0	0.0129	2110	60.8	0.0111	2110	78.0	0.0103	0	72.9	0.0129	2110	91.7	0.0185	5796	72.9	0.0129	2110	0.86	87.0	0.0234
	DX2 0%OA+PC	104.9	0.0103	0	59.5	0.0092	2870	80.0	0.0114	2870	80.9	0.0114	2870	100.5	0.0162	5816						
	DX2 20%OA	105.0	0.0100	810	60.6	0.0095	2670	78.0	0.0098	1860	86.7	0.0099	2670	125.8	0.0095	5885						
	DX2 0%OA	105.0	0.0073	-10	61.6	0.0098	2850	80.0	0.0113	2860	80.7	0.0112	2850	124.2	0.0070	6503						
AHRI 100% (95°F $_{DB}$/75°F $_{WB}$)	IEC Low	95.5	0.0143	1380													69.5	0.0143	1380	0.23	83.7	0.0240
	IEC Mid	95.0	0.0143	2170													71.3	0.0143	2170	0.56	83.6	0.0234
	IEC High	95.0	0.0143	2650													72.1	0.0143	2650	0.86	83.5	0.0232
	IEC Low+DX2	96.2	0.0143	1300	58.8	0.0095	2680	78.8	0.0097	1760	75.4	0.0117	2680	117.4	0.0144	5882	71.0	0.0143	1300	0.27	83.8	0.0234
	IEC High+DX2	97.6	0.0144	3070	61.3	0.0107	2800	78.9	0.0097	990	74.6	0.0132	2800	119.9	0.0144	5872	73.1	0.0144	3070	0.74	84.7	0.0228
	IEC Low+DX2+PC	95.0	0.0144	1350	57.0	0.0093	2700	78.0	0.0097	1730	75.0	0.0118	2700	100.4	0.0183	5850	71.1	0.0144	1350	0.25	83.5	0.0229
	IEC High+DX2+PC	94.9	0.0144	3090	59.7	0.0101	2800	78.0	0.0097	890	74.1	0.0133	2800	101.7	0.0183	5832	73.0	0.0144	3090	0.75	83.9	0.0221
	IEC High+DX1+PC (no RTU fan)	95.0	0.0144	2640	60.7	0.0111	2640	78.0	0.0097	0	72.2	0.0144	2640	89.3	0.0179	5797	72.2	0.0144	2640	0.87	82.5	0.0218
	DX2 0%OA+PC	95.0	0.0145	-10	59.3	0.0093	2870	80.0	0.0115	2880	80.5	0.0116	2870	100.8	0.0169	5830						
	DX2 20%OA	95.0	0.0098	800	58.1	0.0089	2650	78.0	0.0098	1850	83.5	0.0098	2650	115.8	0.0093	5908						
	DX2 0%OA	95.0	0.0071	-20	60.5	0.0095	2850	80.1	0.0113	2870	80.5	0.0112	2850	114.6	0.0069	6523						

Test Condition	Mode	Outside Air			Supply Air			Return Air			Mixed Air			Condenser Exhaust Air			IEC Product Air				IEC Exhaust Air	
		T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	ΔP_{CASE} (inWG)	T_{DB} (°F)	ω (lb/lb)
Western Annual (90°F $_{DB}$/64°F $_{WB}$)	IEC Low	90.0	0.0084	1110													61.1	0.0084	1110	0.24	78.4	0.0188
	IEC Mid	90.0	0.0090	1740													63.1	0.0090	1740	0.56	78.2	0.0189
	IEC High	89.9	0.0085	2110													62.7	0.0085	2110	0.84	77.7	0.0186
	IEC Low+DX2	90.0	0.0084	1150	53.4	0.0076	2640	78.0	0.0098	1590	70.7	0.0092	2640	109.4	0.0080	5921	61.0	0.0084	1150	0.23	78.6	0.0192
	IEC High+DX2	90.0	0.0077	2480	51.9	0.0072	2780	78.0	0.0098	680	66.2	0.0081	2780	108.8	0.0074	5902	63.0	0.0077	2480	0.71	78.4	0.0174
	IEC Low+DX2+PC	90.0	0.0087	1130	52.2	0.0073	2630	78.0	0.0097	1610	71.3	0.0093	2630	90.6	0.0131	5865	62.0	0.0087	1130	0.22	78.7	0.0186
	IEC High+DX2+PC	90.0	0.0101	2490	53.4	0.0077	2780	78.0	0.0098	680	68.9	0.0100	2780	92.4	0.0141	5896	66.4	0.0101	2490	0.71	79.5	0.0183
	IEC High+DX1+PC (no RTU fan)	90.0	0.0089	2110	50.9	0.0075	2110	78.0	0.0097	0	64.1	0.0089	2110	79.7	0.0132	5902	64.1	0.0089	2110	0.84	77.6	0.0175
	DX2 20%OA+PC	90.0	0.0098	950	55.8	0.0080	2590	78.2	0.0098	1640	82.9	0.0098	2590	92.9	0.0138	5899						
	DX2 0%OA+PC	90.0	0.0071	0	58.4	0.0089	2860	80.0	0.0114	2860	80.4	0.0113	2860	91.1	0.0119	5848						
	DX2 20%OA	90.1	0.0097	790	56.9	0.0086	2640	78.0	0.0098	1850	81.9	0.0098	2640	110.7	0.0092	5922						
Warm Humid (82°F $_{DB}$/73°F $_{WB}$)	IEC Low	82.0	0.0157	1080													70.9	0.0157	1080	0.24	77.1	0.0199
	IEC Mid	82.0	0.0157	1730													71.3	0.0157	1730	0.57	77.2	0.0197
	IEC High	82.0	0.0157	2110													71.4	0.0157	2110	0.85	77.0	0.0197
	IEC Low+DX2	82.0	0.0157	1100	56.9	0.0090	2640	78.0	0.0098	1660	74.9	0.0122	2640	103.6	0.0154	5936	70.2	0.0157	1100	0.24	76.7	0.0199
	IEC High+DX2	82.0	0.0157	2430	58.8	0.0097	2770	78.0	0.0098	960	72.8	0.0140	2770	104.5	0.0153	5919	70.8	0.0157	2430	0.74	76.8	0.0195
	IEC Low+DX2+PC	82.0	0.0157	1120	56.3	0.0089	2640	78.0	0.0099	1630	74.8	0.0123	2640	96.3	0.0171	5900	70.2	0.0157	1120	0.25	76.9	0.0200
	IEC High+DX2+PC	82.0	0.0157	2450	58.5	0.0096	2780	78.0	0.0097	900	72.9	0.0141	2780	97.1	0.0170	5893	71.0	0.0157	2450	0.76	76.9	0.0195
	IEC High+DX1+PC (no RTU fan)	82.0	0.0157	2120	58.9	0.0106	2120	78.0	0.0097	0	70.6	0.0157	2120	85.3	0.0169	5885	70.6	0.0157	2120	0.88	76.2	0.0197
	DX2 0%OA+PC	82.0	0.0157	-40	58.8	0.0092	2820	80.0	0.0114	2860	80.1	0.0115	2820	97.0	0.0172	5884						
	DX2 20%OA	82.0	0.0097	780	55.0	0.0082	2640	78.0	0.0098	1860	79.3	0.0098	2640	102.6	0.0092	5936						
	DX2 0%OA	82.1	0.0069	-40	59.1	0.0091	2850	80.0	0.0113	2890	80.1	0.0112	2850	102.0	0.0067	6567						
Condition	Mode	Outside Air			Supply Air			Return Air			Mixed Air			Condenser Air Exhaust			IEC Product Air				IEC Exhaust Air	

		T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	ΔP_{CASE} (inWG)	T_{DB} (°F)	ω (lb/lb)
IEER 75% (81.5°F $_{DB}$/63.4°F $_{WB}$)	IEC Low	81.5	0.0107	1090													63.5	0.0107	1090	0.23	74.7	0.0171
	IEC Mid	80.8	0.0109	1720													64.6	0.0109	1720	0.56	74.0	0.0168
	IEC High	81.5	0.0116	2120													66.4	0.0116	2120	0.86	74.6	0.0172
	IEC Low+DX2	81.5	0.0107	1130	54.1	0.0086	2650	78.0	0.0097	1610	72.1	0.0101	2650	102.1	0.0107	5932	63.8	0.0107	1130	0.24	75.0	0.0174
	IEC High+DX2	81.5	0.0107	2450	53.8	0.0086	2790	78.0	0.0098	730	68.3	0.0105	2790	102.1	0.0102	5908	65.4	0.0107	2450	0.73	75.7	0.0172
	IEC Low+DX2+PC	81.5	0.0106	1130	53.3	0.0084	2620	78.0	0.0098	1600	72.3	0.0102	2620	90.7	0.0139	5876	64.5	0.0106	1130	0.24	75.0	0.0169
	IEC High+DX2+PC	81.4	0.0107	2450	53.8	0.0087	2760	78.0	0.0098	630	68.6	0.0105	2760	90.6	0.0135	5858	66.3	0.0107	2450	0.73	75.3	0.0163
	IEC High+DX1+PC (no RTU fan)	81.5	0.0106	2090	53.0	0.0090	2090	78.0	0.0098	0	65.8	0.0106	2090	79.8	0.0140	5888	65.8	0.0106	2090	0.86	73.6	0.0161
Warm Dry (78°F $_{DB}$/58.5°F $_{WB}$)	IEC Low	78.0	0.0070	1080													56.2	0.0070	1080	0.23	70.2	0.0147
	IEC Mid	78.0	0.0076	1700													57.7	0.0076	1700	0.54	70.1	0.0149
	IEC High	78.0	0.0073	2110													59.4	0.0073	2110	0.85	70.0	0.0142
	IEC Low+DX2	78.4	0.0083	1140	51.4	0.0072	2620	78.0	0.0098	1610	69.6	0.0092	2620	97.9	0.0080	5977	58.3	0.0083	1140	0.24	71.0	0.0155
	IEC High+DX2	78.0	0.0086	2460	49.9	0.0070	2770	78.0	0.0098	750	64.5	0.0089	2770	96.8	0.0083	5954	60.6	0.0086	2460	0.72	71.3	0.0149
	IEC Low+DX2+PC	78.0	0.0091	1150	50.5	0.0071	2630	78.0	0.0098	1610	69.9	0.0095	2630	85.3	0.0115	5935	59.0	0.0091	1150	0.24	71.4	0.0160
	IEC High+DX2+PC	78.0	0.0096	2470	49.4	0.0070	2780	78.0	0.0098	740	65.1	0.0096	2780	85.2	0.0117	5922	61.4	0.0096	2470	0.73	71.7	0.0156
	IEC High+DX1+PC (no RTU fan)	78.0	0.0100	2110	47.9	0.0073	2110	78.0	0.0098	0	61.4	0.0100	2110	74.9	0.0120	5933	61.4	0.0100	2110	0.85	71.1	0.0162
	DX2 20%OA+PC	78.0	0.0086	910	52.6	0.0075	2590	78.0	0.0098	1680	78.0	0.0093	2590	85.3	0.0112	5937						

Condition	Mode	Outside Air			Supply Air			Return Air			Mixed Air			Condenser Air Exhaust			IEC Product Air				IEC Exhaust Air	
		T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	T_{DB} (°F)	ω (lb/lb)	Q (CFM)	ΔP $_{CASE}$ (inWG)	T_{DB} (°F)	ω (lb/lb)
IEER 50% (68°F $_{DB}$/57.5°F $_{WB}$)	IEC Low	68.2	0.0079	1040													56.5	0.0079	1040	0.24	64.1	0.0120
	IEC Mid	68.0	0.0079	1680													57.0	0.0079	1680	0.56	64.1	0.0117
	IEC High	68.0	0.0078	2100													57.3	0.0078	2100	0.85	64.1	0.0116
	IEC Low+DX2	68.0	0.0081	1130	49.8	0.0074	2660	78.0	0.0098	1680	69.2	0.0091	2660	87.6	0.0077	5964	56.6	0.0081	1130	0.24	66.1	0.0133
	IEC High+DX2	68.0	0.0084	2450	48.6	0.0071	2770	78.0	0.0098	800	63.8	0.0088	2770	87.1	0.0082	5948	59.3	0.0084	2450	0.73	67.1	0.0121
	IEC Low+DX2+PC	68.5	0.0082	1120	49.7	0.0071	2640	78.0	0.0098	1630	69.5	0.0091	2640	80.3	0.0100	5915	57.6	0.0082	1120	0.24	65.8	0.0124
	IEC High+DX2+PC	68.0	0.0086	2460	48.8	0.0072	2780	78.0	0.0098	740	64.3	0.0089	2780	80.3	0.0102	5918	60.3	0.0086	2460	0.73	67.2	0.0118
	DX2 20%OA+PC	68.0	0.0079	860	50.6	0.0070	2570	78.0	0.0098	1710	74.3	0.0091	2570	79.9	0.0097	5905						
IEER 25% (65°F $_{DB}$/52.8°F $_{WB}$)	IEC Low	65.0	0.0062	1090													50.5	0.0062	1090	0.24	60.9	0.0110
	IEC Mid	65.0	0.0067	1710													51.5	0.0067	1710	0.56	60.4	0.0114
	IEC High	65.0	0.0069	2100													52.2	0.0069	2100	0.84	60.5	0.0113
	IEC Low+DX2	65.0	0.0080	1130	48.7	0.0066	2640	78.0	0.0097	1680	67.6	0.0090	2640	84.4	0.0073	6007	53.1	0.0080	1130	0.24	60.3	0.0114
	IEC High+DX2	65.0	0.0083	2480	45.7	0.0060	2800	78.0	0.0096	930	60.7	0.0086	2800	82.9	0.0076	5979	54.4	0.0083	2480	0.76	60.1	0.0106
	IEC Low+DX2+PC	65.0	0.0080	1160	48.0	0.0064	2620	78.0	0.0097	1630	67.5	0.0090	2620	77.3	0.0092	5925	53.5	0.0080	1160	0.24	60.8	0.0112
	IEC High+DX2+PC	65.0	0.0083	2480	44.9	0.0058	2800	78.0	0.0096	830	59.9	0.0086	2800	76.1	0.0093	5944	54.1	0.0083	2480	0.76	60.0	0.0108
	IEC High+DX1+PC (no RTU fan)	65.0	0.0090	2090	43.3	0.0064	2090	78.0	0.0096	0	55.2	0.0090	2090	66.6	0.0097	5960	55.2	0.0090	2090	0.87	61.1	0.0118
	DX2 20%OA+PC	65.0	0.0078	900	49.7	0.0068	2590	78.0	0.0098	1690	73.1	0.0090	2590	77.6	0.0091	5931						
	DX2 0%OA	82.1	0.0069	-40	59.1	0.0091	2850	80.0	0.0113	2890	80.1	0.0112	2850	102.0	0.0067	6567						

INDIRECT EVAPORATIVE COOLING PERFORMANCE

The indirect evaporative heat exchanger studied here regularly cools air beyond the outside air wet bulb temperature. This unique capability is maintained at all fan speeds and across a wide range of climate conditions. Figure 11 A charts the wet bulb effectiveness for the indirect evaporative cooler measured from every test in this laboratory study. Figure 11 B charts the average for all tests at each fan speed tested.

WBE increases with when product airflow rate decreases because the air flow is allowed more residence time for evaporation and sensible heat transfer inside the heat exchanger.

Notably, tests in indirect evaporative only mode were subjected to a higher airflow resistance than that recommended by the manufacturer. Such operation should tend to change the primary–secondary airflow balance in a way that would decrease product temperature and decrease product airflow. This assumption is supported by the WBE results recorded in Figure 11, which are somewhat higher than manufacturer-stated performance.

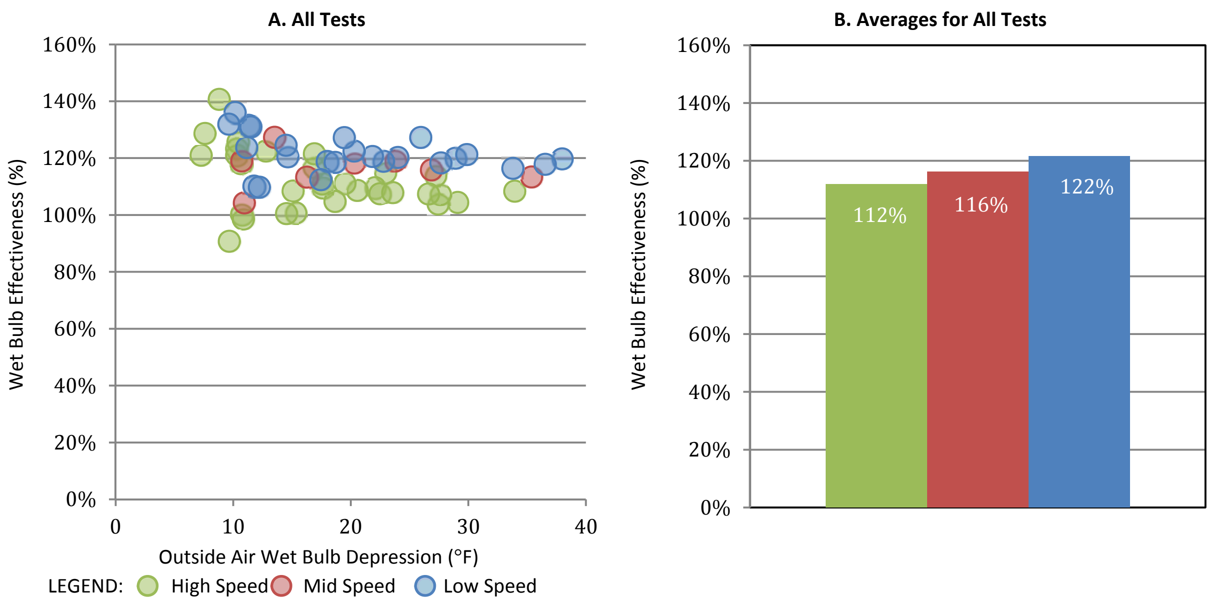


FIGURE 11. WET BULB EFFECTIVENESS AS A FUNCTION OF OUTSIDE AIR WET BULB DEPRESSION FOR THREE FAN SPEEDS (A) RESULTS FROM EVERY TEST (B) AVERAGE VALUES FOR RESULTS FROM ALL TESTS

The heat transfer rate is driven by temperature difference between the primary flow and secondary flow, and the temperature of the secondary flow is driven by the direct evaporation of water and a cooling trend that is limited by the wet bulb temperature of the secondary inlet. Therefore, describing heat exchanger performance in terms of the indirect evaporative effectiveness (given by Equation 2) can be helpful.

Figure 12 A plots the indirect wet bub effectiveness measured for each test. Figure 12 B illustrates the psychrometric process for the primary and secondary airflows. The psychrometric chart indicates that the wet bulb temperature at the secondary inlet is much lower than is the wet bulb temperature at the primary inlet. Evaporation from this point establishes a larger temperature difference for heat exchange, ultimately allowing for cooling below the outside air wet bulb temperature.

As illustrated in Figure 12 A, the indirect WBE does change some with airflow, but 87% of all tests achieved an indirect wet bulb depression effectiveness between 70–90%. In either case, WBE remains fairly consistent across a range of wet bulb depression conditions. The larger variation at lower wet bulb depressions occurs because natural variation in measurement and system performance is distorted and magnified mathematically by the wet bulb effectiveness ratio as wet bulb depression approaches zero.

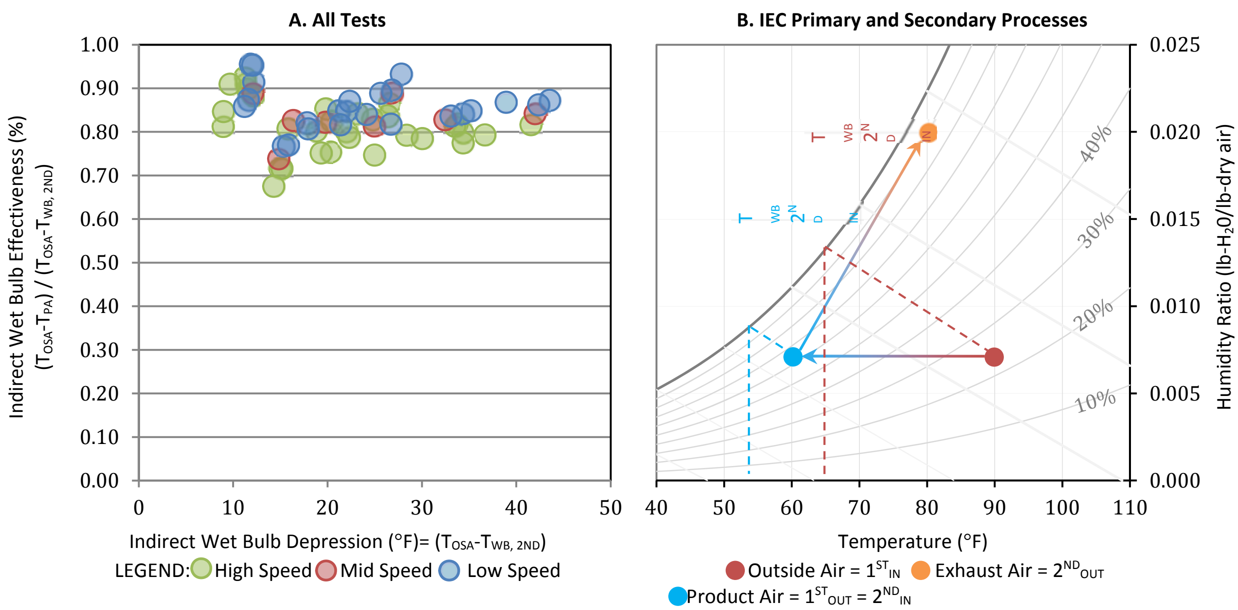


FIGURE 12. (A) INDIRECT WET BULB EFFECTIVENESS AS A FUNCTION OF INDIRECT WET BULB DEPRESSION
(B) PSYCHROMETRIC DIAGRAM OF PRIMARY AND SECONDARY PROCESSES FOR INDIRECT EVAPORATIVE

Figure 13 plots sensible cooling capacity for the indirect evaporative only mode as a function of outdoor wet bulb depression. While effectiveness is fairly consistent across a range of conditions, cooling capacity is strongly dependent on both airflow and temperature. Each airflow rate tested exhibits a separate and distinct linear relationship as a function of outside air wet bulb depression. As also underscored, outdoor wet bulb depression and airflow rate could be the best two independent parameters for regression modeling of system performance in indirect evaporative cooling mode.

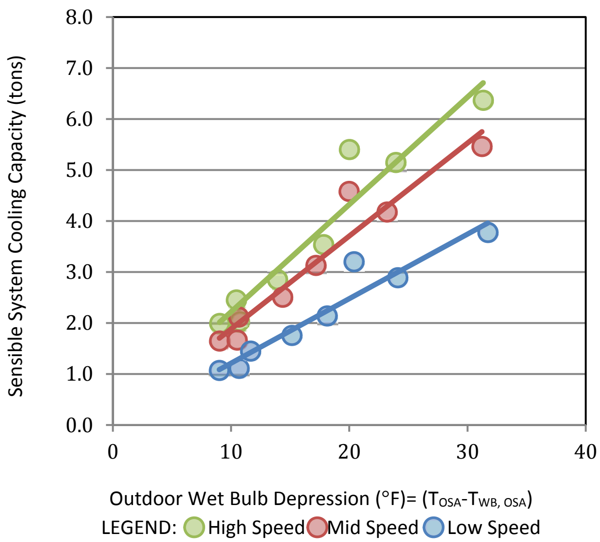


FIGURE 13. COOLING CAPACITY AS A FUNCTION OF OUTDOOR WET BULB DEPRESSION, IN INDIRECT EVAPORATIVE ONLY MODE

PSYCHROMETRIC PERFORMANCE

Figure 14 through Figure 16 illustrate the cooling characteristics for three modes tested at five outside air conditions. Figure 14 examines mode "IEC_{HIGH}," Figure 15 examines mode "IEC_{LOW}+DX2+PC," and Figure 16 examines mode "IEC_{HIGH}+DX2+PC." Measurements for temperature and humidity at key air nodes in the system are plotted on a psychrometric chart for each outside air condition. Points are included for the outside air, product air, return air, mixed air, and supply air conditions. For clarity, the points listed in Figure 14 through Figure 16 can be referenced against the diagram in Figure 2. These plots offer some insight into how the IEC behaves across a range of inlet temperatures and how the IEC interacts with the rooftop unit for the combination evaluated here.

These psychrometric plots illustrate several notable characteristics:

- Although mode "IEC_{LOW}" delivered less product airflow than did mode "IEC_{LOW}+DX2+PC," the product air temperatures for each mode are very similar.
- Although mode "IEC_{LOW}+DX2+PC" generates a lower product air temperature than does mode "IEC_{HIGH}+DX2+PC," the latter mode results in a lower mixed air temperature because product air makes up a larger portion of the mixed air. Notably, the lower mixed air temperature significantly reduces vapor compression cooling capacity and decreases the sensible heat ratio. Electricity use for compressor operation decreases somewhat, but this decrease is not as great as the reduction in cooling capacity. Thus, while the IEC operates with very high efficiency, and the combination achieves substantial energy savings, integrated operation actually reduces efficiency for the vapor compression portion of the operation. This important observation suggests that there is room for alternate hybrid architecture to achieve even better savings.

- The IEC generates an outstanding temperature split from outside air to product air, suggesting great potential for operation with 100% outside air equipment, which can struggle to achieve such sensible cooling capacity.

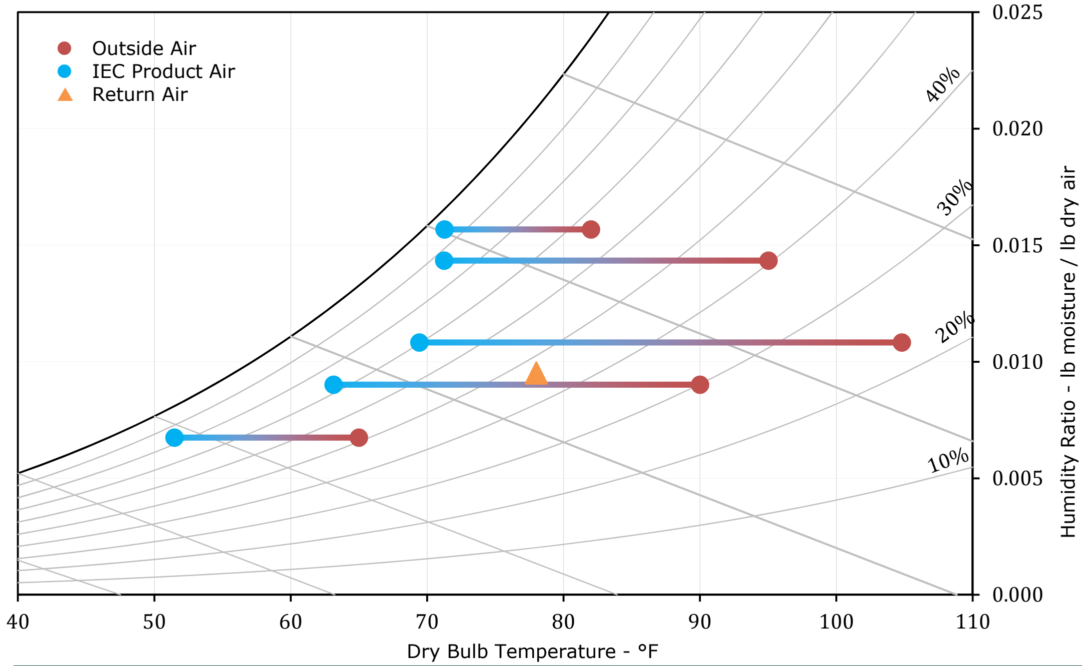


FIGURE 14. PSYCHROMETRIC PERFORMANCE MODE: "IEC HIGH" AT SEVERAL OUTSIDE AIR CONDITIONS; IEC PRODUCT AIRFLOW = 2,100 CFM

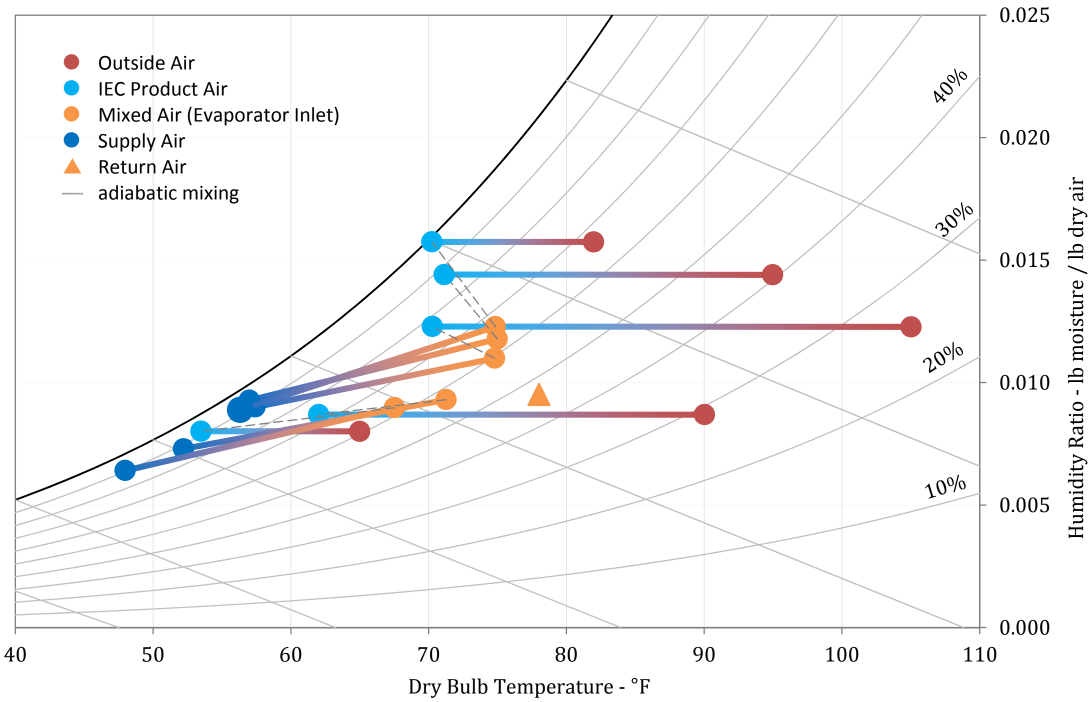


FIGURE 15. PSYCHROMETRIC PERFORMANCE MODE: "IEC LOW + DX2 + PC" AT SEVERAL OUTSIDE AIR CONDITIONS; IEC PRODUCT AIRFLOW = 1130 CFM, RETURN AIRFLOW = 1660 CFM, SUPPLY AIRFLOW = 2640 CFM

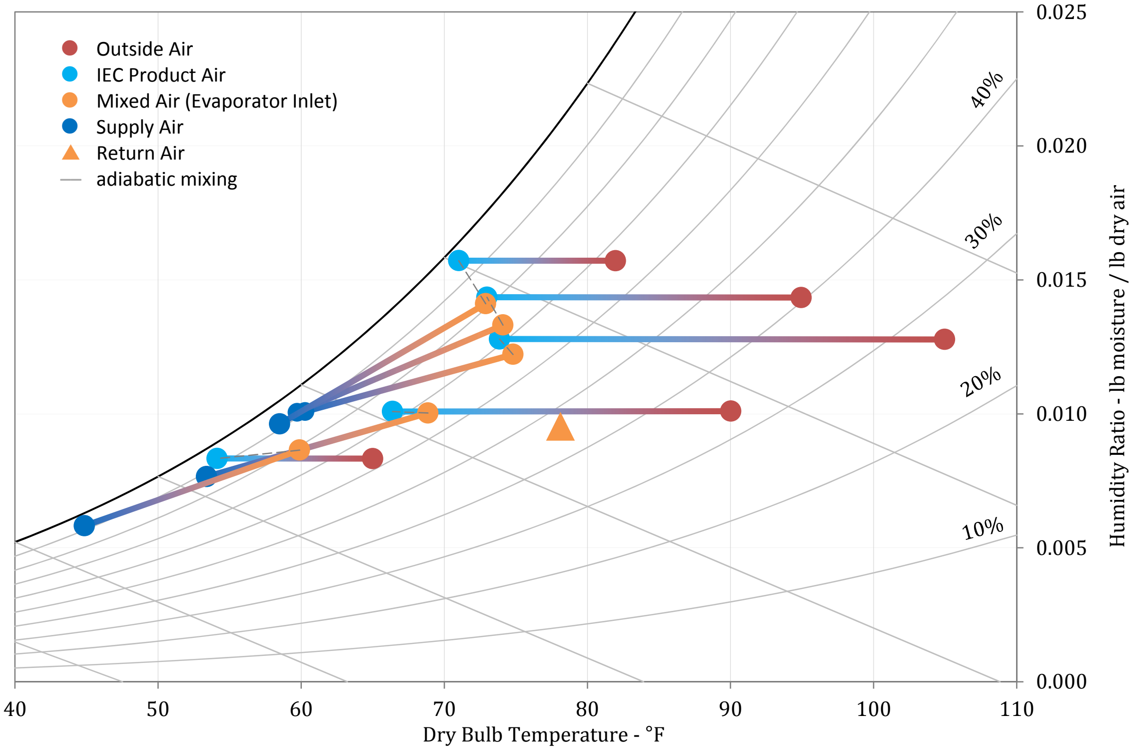


FIGURE 16. PSYCHROMETRIC PERFORMANCE MODE: "IEC HIGH + DX2+PC" AT SEVERAL OUTSIDE AIR CONDITIONS; IEC PRODUCT AIRFLOW = 2460 CFM, RETURN AIRFLOW = 800 CFM, SUPPLY AIRFLOW RATE = 2780 CFM

ESTIMATED ANNUAL SAVINGS

An estimate of annual savings must account for application-specific details, such as the baseline scenario, the amount of ventilation required, and the sequence of operations employed on the ground. For comparison, the team made a generalized estimate of the integrated annual savings using the following method:

- Compare the efficiency for the retrofit system to the efficiency of the baseline unit operating with 20% outside air at several different climate conditions.
- For each climate condition, select a combination of modes that would meet the expected sensible room load at that condition. Select the load at peak conditions to match the full sensible room capacity of the baseline unit operating with 20% outside air.
- Determine annual savings by weighting the savings at each condition by the fraction of annual cooling load at each condition.

This generalized estimate adapted the annual load profile inferred by AHRI 340-360 standards for determination of the IEER metric [AHRI 2011] as given by in Equation **Error! Reference source not found.**:

EQUATION 15: IEER METRIC

$$\mathrm{IEER} = 0.125 \cdot (\mathrm{EER}_{25\%\ \mathrm{LOAD}}) + 0.238 \cdot (\mathrm{EER}_{50\%\ \mathrm{LOAD}}) + 0.617 \cdot (\mathrm{EER}_{75\%\ \mathrm{LOAD}}) + 0.02 \cdot (\mathrm{EER}_{100\%\ \mathrm{LOAD}})$$

Figure 17 illustrates the annual distribution of cooling at each condition and each part-load fraction as specified by AHRI 340-360. The figure also shows the adapted distribution used for energy savings estimates in this study, selected to better represent California climate conditions. Real-load distributions vary somewhat by climate zone and application, but this binned approach provides a rough approximation. Notably, the large majority of cooling is needed when outside temperature is between 68–82°F, periods when the IEC has the greatest efficiency advantage. At cooler conditions, a conventional economizer mode could be the most efficient option, but these conditions reduce the overall cooling needed. At hotter conditions, energy intensive compressor cooling is needed to keep up with the load, but only a small fraction of the total cooling occurs during peak periods.

The sensible room cooling load at each condition does not always align with the capacity in each mode tested. Therefore, an evaluation of the savings potential at a particular climate condition must account the system’s ability to cycle between multiple modes as long as the aggregate cooling capacity matches the cooling load and the minimum required ventilation rate is achieved. Such operation allows for substantial savings at all climate conditions evaluated—and highlights the dynamic nature of the hybrid combination and the opportunity for the system to adjust to suit different cooling needs at different times.

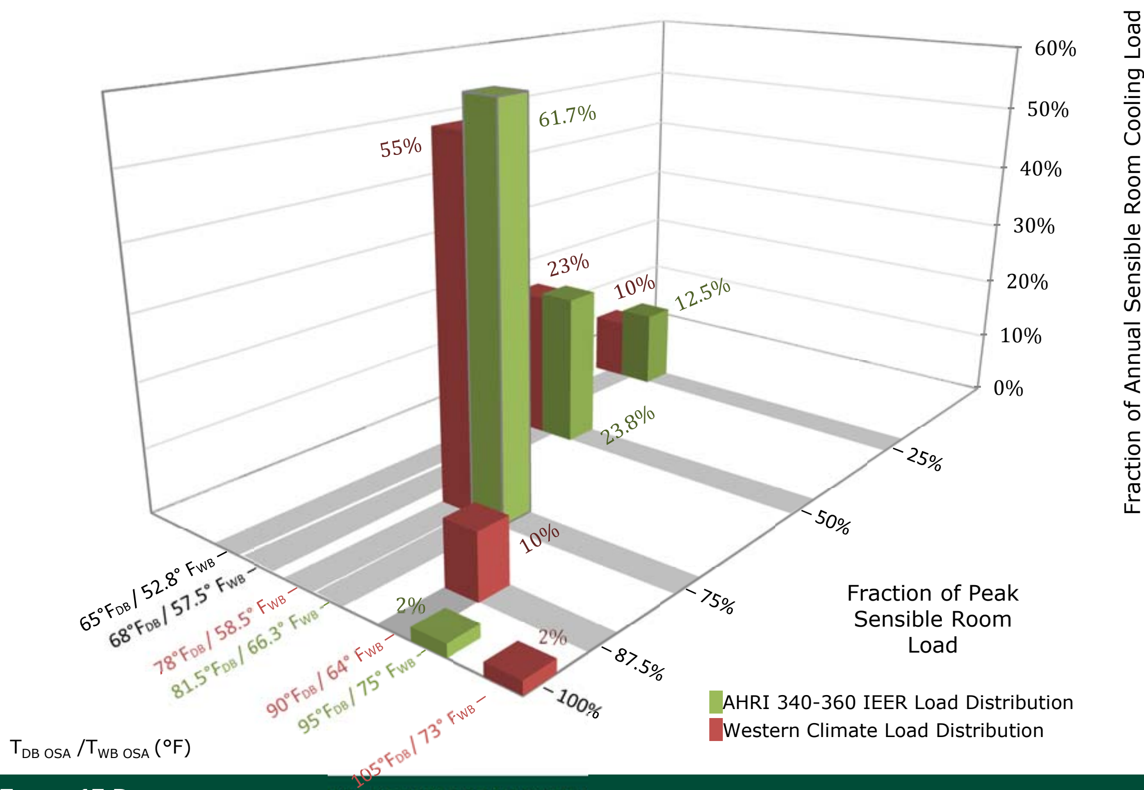


FIGURE 17: DISTRIBUTION OF ANNUAL COOLING LOAD BY OUTSIDE TEMPERATURE AND LOAD FRACTION

Table 6 describes the sensible room load, capacity, and efficiency for the retrofit scenario selected at each climate condition. The table summarizes the energy savings at each condition and the fraction of annual sensible room load assumed to occur at each condition. For the worst-case hours, the retrofit reduces energy use by 26%. For typical daily high temperature conditions, the hybrid combination achieves more than 50% savings. For milder part-load hours, the combination can reduce energy use by as much as 86%.

These estimates

- Do not consider energy use for ventilation during non-cooling hours
- Do not consider the benefits of economizer cooling in the baseline scenario
- Assume the baseline scenario is mode "DX2-20% OSA" as laboratory tested
- Do not consider transient gains or losses associated with inter-mode cycling

The aggregate load–weighted assessment indicates 68% reduction in annual energy use for cooling and ventilation.

TABLE 6. OPERATING SCENARIOS, CAPACITY, AND EFFICIENCY AT CLIMATE CONDITIONS USED TO ESTIMATE ANNUAL SAVINGS

Western Peak – 105°F$_{DB}$ / 73°F$_{WB}$				
Retrofit Scenario			Load Scenario	100%
			Sensible Room Load	4.0 tons
			Baseline Sensible Room EER (mode: DX2-20% OSA)	4.0
			Fraction of Annual Sensible Room Load at Condition	2%
	Mode	*Runtime Fraction*	*Effective Sens. Room Capacity (tons)*	*Sens. Room EER*
	IEC $_{LOW}$	23%	0.2	19.8
	IEC $_{LOW}$+DX2+PC	77%	3.8	5.2
	Combined Modes:	100%	4.0	5.4
			Energy Savings at Condition	26%
Western Annual – 90°F$_{DB}$ / 64°F$_{WB}$				
Retrofit Scenario			Load Scenario	87.5%
			Sensible Room Load	3.5 tons
			Baseline Sensible Room EER (mode: DX2-20% OSA)	5.3
			Fraction of Annual Sensible Room Load at Condition	10%
	Mode	*Runtime Fraction*	*Effective Sens. Room Capacity (tons)*	*Sens. Room EER*
	IEC $_{HIGH}$	80%	2.2	18
	IEC $_{LOW}$+DX2+PC	20%	1.3	7.1
	Combined Modes:	100%	3.5	11.6
			Energy Savings at Condition	54%
Warm Dry – 78°F$_{DB}$ / 58.5°F$_{WB}$				
Retrofit Sc.			Load Scenario	75%
			Sensible Room Load	3.0 tons
			Baseline Sensible Room EER (mode: DX2-20% OSA)	7.2
			Fraction of Annual Sensible Room Load at Condition	55%
	Mode	*Runtime Fraction*	*Effective Sens. Room Capacity (tons)*	*Sens. Room EER*
	IEC $_{LOW}$	100%	2.0	32
			Energy Savings at Condition	66%

IEER 50% – 68°F_{DB} / 57.5F_{WB}				
Retrofit Scenario			Load Scenario	50%
			Sensible Room Load	2.0 tons
			Baseline Sensible Room EER (mode: DX2-20% OSA)	8
			Fraction of Annual Sensible Room Load at Condition	23%
	Mode	*Runtime Fraction*	*Effective Sens. Room Capacity (tons)*	*Sens. Room EER*
	IEC $_{LOW}$	92%	1.75	45.1
	IEC $_{MID}$	8%	0.25	33.5
	Combined Modes:	100%	2.0	43.2
			Energy Savings at Condition	82%

IEER 25% – 65°F_{DB} / 52.8°F_{WB}				
Retrofit Sc.			Load Scenario	25%
			Sensible Room Load	1.0 tons
			Baseline Sensible Room EER (mode: DX2-20% OSA)	8.5
			Fraction of Annual Sensible Room Load at Condition	10%
	Mode	*Runtime Fraction*	*Effective Sens. Room Capacity (tons)*	*Sens. Room EER*
	IEC $_{LOW}$	0.37	1.0	61.6
			Energy Savings at Condition	86%

WATER USE EFFICIENCY

The electrical energy savings achieved by these evaporative retrofits comes at the expense of increased on-site water consumption. Figure 18 presents the water evaporate rate from each component for every test with either component operating. For field applications, total water consumption will include both water evaporated and water drained to maintain water quality. However, since the amount of water needed to maintain water quality will vary by location, the laboratory tests focused on water use for evaporation. The value was calculated as the difference between water supplied to each component and water drained from each component during the course of each test.

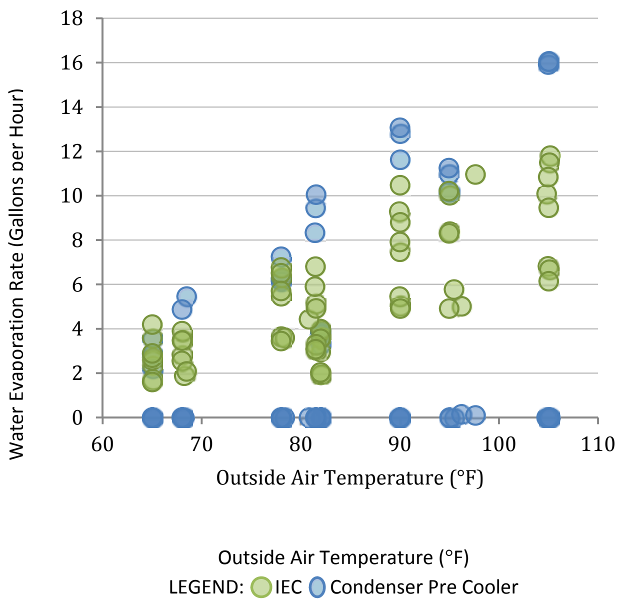


FIGURE 18. WATER EVAPORATION RATE FOR IEC AND PRE COOLER MEASURED IN EACH TEST

Water evaporation closely correlates with outside air temperature since the driving force for evaporation scales proportionally to the wet bulb depression. For the IEC studied here, the rate of water evaporation also depends on the fan speed: water use at low speed is roughly half the water use at high speed.

At 90°F$_{DB}$ /64°F$_{WB}$, the IEC evaporates approximately 10 gallons/hour at high speed, and 5 gallons/hour at low speed, while the IEC evaporates roughly 12 gallons/hour. At this condition, for the scenario described in Table 6, the coincident energy savings is roughly 4.3 kWh/hr. If the maintenance water accounts for an additional 10% water use, as measured for this technology in a previous field study [Woolley 2014], the ratio of water to energy savings equates to roughly 5.4 gallons/kWh of savings.

For context, estimates of the water use intensity for electricity generation range by more than an order of magnitude, depending on the source mix for electricity generation. However, the most well-founded research estimates an average water use intensity of 1.41 gallons/kWh for California's grid mix, including evaporative losses from reservoirs for hydroelectric generation [Pistochini 2011, Torcellini 2003, Larson 2007]. This means, from a statewide water use perspective, the local water consumption for these retrofits would be partially offset by the water savings associated with reduced electrical generation.

With these systems, water use is the inescapable tradeoff for the energy and cost savings demonstrated. The decision to use the technology should give careful consideration to the related environmental benefits and consequences.

If every commercial building in California were to use evaporative systems with water use intensity of 5 gallons/ton-hr, the cumulative statewide water consumption would only amount to approximately 87,000 acre feet (28.5 billion gallons) per year. This is only about 0.23% of California's annual urban and agricultural water consumption (see Figure 18 [DWR 2014]). At the same time, these evaporative technologies could put a dramatic dent in California's summertime electricity generation requirements and could reduce annual commercial building energy use for cooling and ventilation by more than 50%

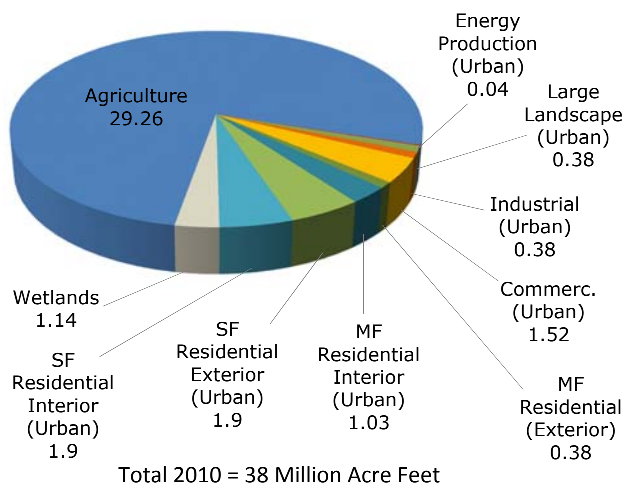


FIGURE 19. CALIFORNIA APPLIED WATER USES (MILLION ACRE FEET)

CONCLUSIONS

The study showed that IEC coupled with condenser air pre-cooling has the potential to transform new or existing RTUs into climate-appropriate equipment for hot dry climates. The IEC delivers a very efficient means to cool outside air for ventilation or modest room cooling while the condenser air pre-cooler can significantly improve the performance of the RTU vapor compression cycle. The configuration tested demonstrated it could provide these benefits:

- Reduce annual energy use for cooling and ventilation by in 68% in California commercial buildings
- At peak cooling conditions (105°FDB/73°FWB), achieve equal capacity while reducing energy use for cooling by 26%
- At milder conditions (65°FDB/ 52.8° FWB), offer savings at large as 86%

While the retrofit improves sensible efficiency at all conditions tested, it does connect new loads and subsequently increases electrical demand at every condition tested. On aggregate, the retrofit will decrease energy used during peak cooling conditions by reducing compressor operation. However, because the highest instantaneous electrical draw will be larger, customer demand charges could increase. The retrofit could reduce the total electrical demand for a building if the retrofit handled ventilation air for multiple units, thereby allowing the operation of existing units in recirculation mode or even disconnection of the existing units.

The IEC and condenser pre-cooler retrofit strategies support CA-EESP goals for climate-appropriate air conditioning technologies. The retrofit measure promises considerable energy benefits for end users and offers demand management benefit for utilities during the cooling season when it is needed most.

Given the substantial overall energy benefit, this study strongly supports the development of utility programs and other public interest efforts to advance the broader application of the technologies.

RECOMMENDATIONS

CONFIGURATION CHANGES

Three factors related to the retrofit configuration for the test may have limited the technical potential of the units tested:

- A significant amount of product air leaked out of the barometric damper and out of leaks in the rooftop unit cabinet. This reduced overall system capacity and efficiency for all modes with compressor operation.
- In all modes without the rooftop unit fan, the airflow resistance was larger than that recommended by the manufacturer. The larger resistance reduced product airflow rate and offset the ratio of primary–secondary airflow.
- The sensible capacity and efficiency for the vapor compression cycle are reduced somewhat when the evaporator coil inlet has already been cooled by indirect evaporative cooling.

The project team therefore recommends applying different configurations of the IEC and vapor compression cooling to increase the energy savings technical potential:

- Future product evolution should integrate the IEC and vapor compression into a unitary hybrid system. This system could incorporate direct access to outside air for economizer cooling and apply variable compressor control to maximize sensible cooling efficiency at any operating condition. Such a hybrid could use evaporative condenser air pre-cooling or may benefit from locating the vapor compression condenser coil in the indirect evaporative secondary exhaust air stream.
- The IEC should be installed in parallel with vapor compression cooling systems, such as rooftop units and variable refrigerant flow (VRF) equipment. This design strategy avoids many of the challenges associated with retrofit integration, would allow both systems to perform at their best, could simplify the controls required, and circumvents the factors that reduced performance for these laboratory tests.

In fact, according to these laboratory measurements, a parallel configuration for the IEC and a rooftop unit utilizing the condenser pre-cooler would achieve 41% savings at peak conditions (105°F_{DB}/73°F_{WB}), a significant increase over the 26% savings seen in the tests.

REDUCTION IN MAXIMUM INSTANTANEOUS DEMAND

To limit the maximum instantaneous electrical demand for the systems, future applications could downsize vapor compression equipment or could include variable capacity controls that would limit the maximum. In any circumstance, efforts to apply the measure should consider means to reduce connected load. For example, in certain situations, the addition of the IEC could justify permanent disconnection of one compressor.

TECHNOLOGY APPLICATION IMPROVEMENTS

Efforts to improve the technology should focus on design guidance to facilitate proper physical application and optimized controls integration. In particular, smart controls are needed that can assess the multiple combinations of IEC and compressor stages possible to select the most appropriate and beneficial operating mode for a particular situation. Surrounding resources, such as codes and standards, modeling tools, and professional education, would facilitate accelerated application of these strategies.

EFFECTIVE PROGRAM DESIGN

As is the case for most air conditioning technologies, some applications will be better suited than others, and programs that adopt this technology should be designed to avoid scenarios that limit overall energy performance. A simulation study could help assess the savings potential for target customers and to identify applications that ought to be avoided. In parallel, a first-generation program that draws on customer smart meter data and limited field monitoring could aid in assessing the actual savings achieved by installation of the equipment.

REFERENCES

American National Standards Institute and Air-Conditioning, Heating, and Refrigeration Institute. "ANSI/AHRI Standard 340/360: Standard for Performance Rating of Commercial and Industrial Unitary Air-Conditioning and Heat Pump Equipment." Standard. Air-Conditioning, Heating, and Refrigeration Institute, October 27, 2011. http://www.ahrinet.org/App_Content/ahri/files/STANDARDS/ANSI/ANSI_AHRI_Standard_340-360-2007_with_Add_1_and_2.pdf.

American National Standards Institute and Air-Conditioning, Heating, and Refrigeration Institute. "ANSI/AHRI Standard 210/240: Performance Rating of Unitary Air-Conditioning & Air-Source Heat Pump Equipment." Standard. Air-Conditioning, Heating, and Refrigeration Institute, December 2012. http://www.ahrinet.org/App_Content/ahri/files/standards%20pdfs/ANSI%20standards%20pdfs/ANSI.AHRI%20Standard%20210.240%20with%20Addenda%201%20and%202.pdf.

American Society of Heating Refrigeration and Air Conditioning Engineers. "ANSI/ASHRAE Standard 37-2009: Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment," 2009.

American Society of Heating Refrigeration and Air Conditioning Engineers. "ANSI/ASHRAE Standard 55-2010: Thermal Environmental Conditions for Human Occupancy," 2010.

American Society of Heating Refrigeration and Air Conditioning Engineers. Special Project Committee 212. "Method of Test for Determining Energy Performance and Water-Use Efficiency of Add-On Evaporative Pre-Coolers for Unitary Air Conditioning Equipment." American Society of Heating Refrigeration and Air Conditioning Engineers, 2015. https://www.ashrae.org/standards-research--technology/standards--guidelines/titles-purposes-and-scopes#SP212P.

California Department of Water Resources. "California Water Plan: Investing in Innovation & Infrastructure." California Natural Resources Agency, 2014. http://www.waterplan.water.ca.gov/cwpu2013/final/index.cfm.

California Energy Commission. "California Commercial End-Use Survey." Consultant Report. California Energy Commission, March 2006. http://www.energy.ca.gov/2006publications/CEC-400-2006-005/CEC-400-2006-005.PDF.

California Independent System Operator. Open Access Same Time Information System (OASIS). California Independent System Operator. Accessed November 6, 2015. http://oasis.caiso.com/.

California Public Utilities Commission. "California Energy Efficiency Strategic Plan, January 2011 Update." California Public Utilities Commission, January 2011. http://www.cpuc.ca.gov/NR/rdonlyres/A54B59C2-D571-440D-9477-3363726F573A/0/CAEnergyEfficiencyStrategicPlan_Jan2011.pdf.

Davis, Robert, and Marshall Hunt. "Laboratory Testing of Performance Enhancements for Rooftop Packaged Air Conditioners." Pacific Gas and Electric Company, February 5, 2015. http://www.etcc-ca.com/reports/laboratory-testing-performance-enhancements-rooftop-packaged-air-conditioners.

Larsen, Dana, Cheryl Lee, Stacy Tellinghuisen, and Arturo Keller. "California's Energy-Water Nexus: Water Use in Electricity Generation." *Southwest Hydrology* 6, no. 5 (October 2007): 20–21.

Modera, Mark, Jonathan Woolley, and Zhijun Liu. "Performance Evaluation for Dual-Evaporative Pre-Cooling Retrofit in Palmdale, California," November 2014.

Pistochini, Theresa. "Evaporative Condenser Air Pre-Coolers." Southern California Edison, 2014.

Pistochini, Theresa and Mark Modera. "Water-Use Efficiency for Alternative Cooling Technologies in Arid Climates." *Energy and Buildings* 43, no. 2–3 (February 2011): 631–38. doi:10.1016/j.enbuild.2010.11.004.

Torcellini, Paul, N. Long, and R Judkoff. "Consumptive Water Use for U.S. Power Production." Technical Report. National Renewable Energy Laboratory, December 2003.

U.S. Energy Information Administration. "Commercial Building Energy Consumption Survey," 2012. http://www.eia.gov/consumption/commercial/.

U.S. Energy Information Administration. "Electricity Data Browser," n.d. Accessed November 6, 2015. http://www.eia.gov/electricity/data/browser/.

Woolley, Jonathan and Caton Mande. "Performance Evaluation for Hybrid Rooftop Air Conditioners with Dual Evaporative Pre-Cooling." Southern California Edison, October 2014.

Woolley, Jonathan, Caton Mande, and Mark Modera. "Side-by-Side Evaluation of Two Indirect Evaporative Air Conditioners Added to Existing Packaged Rooftop Units." Pacific Gas and Electric Company, August 18, 2014. http://www.etcc-ca.com/sites/default/files/reports/et12pge3101_indirect_evap_coolers_retrofit_to_existing_rtus.pdf.

Woolley, Jonathan and Mark Modera. "Advancing Development of Hybrid Rooftop Packaged Air Conditioners: Test Protocol and Performance Criteria for the Western Cooling Challenge." *ASHRAE Transactions* 117, no. 1 (May 2011): 533–40.