Western Cooling Challenge Laboratory Performance Results: Munters EPX 5000 Hybrid DOAS

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The project core team is grateful for the support of Munters engineering staff and leadership in supporting the organization and execution of the laboratory evaluation. The project would not have been successful without their regular correspondence and technical support.

LEGAL NOTICE

Abbreviations and Acronyms

ATS	PG&E Applied Technology Services
AHRI	Air-Conditioning Heating & Refrigeration Institute
ANSI	American National Standards Institute
Btu	British Thermal Unit - the energy required to raise 1 pound of water by 1°F
CFM	Cubic Feet per Minute
DOAS	Dedicated Outside Air System
DX	Direct eXpansion, as a descriptor for vapor compression air conditioning
EER	Energy Efficiency Ratio (defined by ANSI/AHRI 340/360)
EPX	Munters' proprietary indirect evaporative heat exchanger
HVAC	Heating, Ventilation and Air Conditioning
IEC	Indirect Evaporative Cooling (or Cooler)
IEER	Integrated Energy Efficiency Ratio (defined by ANSI/AHRI 340/360)
PG&E	Pacific Gas and Electric Company
RTD	Resistance Temperature Detector or Resistance Thermometer
Therm	A unit of energy equal to 100,000 Btu
WBD	Wet-bulb depression
WBE	Wet-bulb effectiveness
WCEC	UC Davis Western Cooling Efficiency Center
"WC	Inches of Water Column, A unit of pressure

FIGURES

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EXECUTIVE SUMMARY

The research reported herein directly supports California Energy Efficiency Strategic Plan goals to accelerate marketplace penetration of climate appropriate air conditioning technologies. The work was executed by the UC Davis Western Cooling Efficiency Center and PG&E Applied Technology Services, with leadership, vision, project management, and funding provided by the PG&E Emerging Technologies program. This report records results of a detailed laboratory evaluation of the Munters EPX 5000, a dedicated outdoor air system that uses both indirect evaporative cooling and vapor compression to cool ventilation air for commercial buildings.

Munters submitted the technology to UC Davis for evaluation as part of the Western Cooling Challenge, a program that encourages manufacturers to develop and commercialize climate appropriate unitary air conditioning equipment. The Western Cooling Challenge works to characterize and compare the performance of these technologies in order to better inform utility program planning, customer investments in energy efficiency, and industry planning related to low energy mechanical design strategies optimized for western climates. Following this laboratory evaluation, the project team will conduct a pilot field evaluation of the Munters EPX 5000 installed at a grocery store in San Ramon, California.

PROJECT GOALS

The primary goal for this project was to characterize energy efficiency of the Munters EPX 5000 in all modes of operation, in all possible configurations, and across a full range of operating conditions. Laboratory test results were carefully analyzed to consider the technical opportunities and challenges related to the equipment and to identify opportunities for additional improvements to the technology. These results provide the basis for recommendations about how utility efficiency programs, design engineers, and customers might proceed to apply this type of technology for management of indoor environmental quality in commercial buildings while simultaneously reducing energy consumption and peak demand.

TECHNOLOGY REVIEW & RESULTS

The Munters EPX 5000 employs an innovative design that combines indirect evaporative cooling with vapor compression cooling in a packaged unitary air handler intended to operate as a dedicated outdoor air system. The equipment can be configured to use stale room air as the process air for indirect evaporative cooling; an arrangement that could be thought of as an evaporatively-enhanced heat recovery ventilation system. This configuration also allows the equipment to provide heat recovery ventilation in the winter time. However, one downside is that the arrangement forces heat recovery even when it is unwanted.

Since the equipment packages indirect evaporative cooling together with a vapor compression system, Munters has smartly located the condenser in the indirect evaporative exhaust air stream, where spent moist air generally exits at a much cooler temperature than ambient. This reduces the refrigerant condensing temperature which reduces power draw for the compressor.

Analysis indicates that addition of the Munters EPX 5000 to a big-box commercial building where fresh air ventilation is currently provided by individual rooftop packaged air conditioners could reduce peak demand from air conditioning for the entire building by 20 percent. Annual energy savings will vary by climate and by application, but the performance characteristics presented herein should provide a solid basis for analysis by simulation.

RECOMMENDATIONS

We strongly recommend that the technology be adopted by utility energy efficiency programs. We are particularly impressed by the promise of such substantial peak demand reduction from a single machine that can be straightforwardly integrated into either new or existing buildings. As is the case for most air conditioning technologies, we believe that some applications will be more well-suited than others, and suggest that programs adopting this technology be designed to avoid scenarios where overall energy performance is more limited. A simulation study should be conducted to assess the savings potential for target customers and to identify applications that ought to be avoided. This should happen in parallel with a first generation program that draws on customer smart meter data and limited field monitoring to assess the actual savings achieved by installation of the equipment.

INTRODUCTION

This study documents the laboratory measured performance for the Munters EPX 5000, an indirect evaporative–vapor compression hybrid dedicated outdoor air system, tested by PG&E Applied Technology Services and the UC Davis Western Cooling Efficiency Center as part of the Western Cooling Challenge program. This work is directly in support of PG&E Emerging Technologies efforts related to the California Energy Efficiency Strategic Plan goals to accelerate marketplace penetration of new climate-appropriate HVAC technologies.

Cooling and ventilation account for more than 25 percent of the annual electricity consumption for commercial buildings in California. Air conditioners can account for more than 50 percent of the peak electric load for commercial buildings during the hottest summer afternoons. California's electric grid experiences the highest 20 percent of demand for less than 200 hours each year, and this can be attributed almost exclusively to cooling throughout the state.

In terms of source energy consumption or greenhouse gas emissions, heating, cooilng and ventilation can account for more than 50% of the annual footprint for a commercial building. As efficiency for lighting and electronics progresses rapidly, HVAC is currently projected to become the largest annual energy end use in most commercial buildings.

Rooftop air conditioners are often the single largest individually connected load in a building. Therefore, these systems stand as a substantial opportunity for energy efficiency improvements. Since rooftop air conditioners have a very low turnover rate, the efficiency of a system installed will very likely persist in operation for more than 20 years. While efficiency programs for these systems can be complicated and costly, it must be emphasized that near term efforts to influence design of commercial air conditioning systems will likely have a longer lasting impact on our energy infrastructure than will efforts to influence energy end uses in more rapidly evolving fields.

Vapor compression air conditioners have traditionally not varied much in design according to climate zone. However, there are substantial opportunities to reduce energy consumption for cooling and ventilation when a system is designed thoughtfully to work in concert with the meteorological conditions of the region in which it is installed. Therefore, in an effort to strike substantial reduction in energy consumption, to improve management of electrical demand, and to ease integration of renewable and environmentally responsible energy resources, the California Energy Efficiency Strategic Plan has established aggressive goals to advance broader market adoption of climate appropriate HVAC technologies.

California's climate zones are all relatively dry, so one climate-appropriate method to improve efficiency for cooling is the efficient use of modern and sophisticated evaporative cooling technologies. There are a variety of technologies in this category. This study focuses on an innovative commercial product that uses indirect evaporative cooling, a strategy that separates the processes of heat and mass transfer associated with water evaporation, such that one air stream can be cooled without the addition of moisture, while a secondary air stream carries the moist air and absorbs sensible heat.

While there are a number of ways to apply indirect evaporative cooling, the system studied here is designed as a dedicated outside air system (DOAS), and intended to serve the complete ventilation needs for a building throughout the course of all seasons. DOAS is becoming a more common way to provide ventilation to larger buildings; the approach separates the conditioning for room loads from the management conditioning and distribution of fresh ventilation air. By dividing responsibilities, each equipment may be optimized for the specific role that it plays in a building. The approach offers a number of opportunities for efficiency improvements that were heretofore not easily applied with conventional rooftop units:

1. Heat recovery between incoming ventilation air room temperature exhaust can occur at a central location.
2. Low energy mechanical systems that focus on sensible cooling can be applied to manage room conditioning loads. This includes radiant cooling, chilled beams, ductless split systems, or forced air cooling with high sensible heat ratios.
3. Advanced evaporative cooling strategies can be applied for ventilation cooling.

DOAS equipment is especially well suited to employ indirect evaporative cooling. Whereas cooling capacity and efficiency of vapor compression systems generally decreases at high outdoor air temperatures, the capacity and efficiency for indirect evaporative cooling components typically increases. Subsequently, the largest incremental savings can be achieved when indirect evaporative is applied to cooling hot ventilation air. The technology can also be used to cover sensible room cooling loads, but the energy savings achieved compared to a conventional baseline is less substantial.

This report presents analysis of results from laboratory testing of the Munters EPX 5000 - one example of a commercial DOAS product that incorporates indirect evaporative cooling, vapor compression, and heat recovery. The laboratory examination was conducted at PG&E Applied Technology Services in San Ramon, California. Tests were organized in a way to measure the cooling capacity and energy consumption for the system in each mode of operation, and across a range of operating conditions. Further, performance of each major sub component was characterized 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.

Following the measurement of all intended operating modes, our research was extended to explore two opportunities that might further improve performance for the system. For example, it appears that the Munters EPX 5000 could benefit from the ability to switch to outside air for the secondary air stream when outside air is lower enthalpy than room air.

Lastly, the measured equipment performance was used to estimate the energy use for a hypothetical building that would employ the EPX 5000. The analysis was conducted for multiple steady state operating conditions, but not for any projected annual aggregate of cooling loads and environmental conditions. In order to estimate savings delivered by the EPX 5000, these steady state results were compared to the projected energy use for an equal hypothetical building under equal operating conditions but with conventional rooftop units. We believe that the results adequately represent the typical potential for peak demand reduction, and note that the annual energy savings potential for the EPX 5000 will vary significantly by climate and application.

The Munters EPX 5000 was only tested for performance in cooling conditions. The major technical advances presented by the EPX 5000 are related to cooling performance, and the objectives of the Western Cooling Challenge research program and PG&E's climate appropriate technology evaluations conducted through this ET project are focused on improvements for cooling efficiency. The Munters EPX 5000 likely offers some heating season efficiency improvements, namely through its ability to provide exhaust air heat recovery. These laboratory tests do characterize heat recovery effectiveness for the dry heat exchanger, but they do not measure heating performance.

ASSESSMENT OBJECTIVES

The main objectives of this project include:

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

In order to accomplish these objectives the project team worked closely with the manufacturer to develop a strong technical understanding of the equipment from the inside-out, then established a design of experiments for laboratory testing that would measure performance of the system as a whole, as well as operation of all major sub components in each mode of operation. Further, the laboratory evaluation was designed to test for sensitivity to system variables that could change between applications. For example, while the equipment is intended to supply 5000 *cfm* ventilation air, the research team collected data to describe the system in applications where it may not supply the full airflow potential. Also, since the equipment has various possible configurations for installation, the design of experiments was developed to describe relevant performance differences associated with these alternative arrangements.

BACKGROUND

BACKGROUND TO THE WESTERN COOLING CHALLENGE

The UC Davis Western Cooling Challenge is a program that encourages manufacturers to develop and commercialize climate appropriate unitary air conditioning equipment. The program was initiated at the behest of California utilities and various major energy end users who recognize the need for substantial HVAC energy savings. The program defines a climate appropriate test methodology for laboratory evaluation of air conditioning equipment. Unlike current industry standard test procedures for rooftop air conditioners, the Western Cooling Challenge considers both room and ventilation cooling loads.

The Challenge does not require a particular type of system design; rather, it sets ambitious minimum thresholds for energy and water-use efficiency. Western Cooling Challenge certified equipment must demonstrate a 40% savings at peak load conditions, compared to standard efficiency equipment. All of the technologies evaluated thus far for the Western Cooling Challenge employ a hybrid cooling strategy that couples various indirect-evaporative cooling technologies with conventional vapor compression equipment. The program and evaluation criteria were designed around comparison to conventional rooftop packaged air conditioners in commercial applications that operate with some minimum ventilation rate. The Western Cooling Challenge test criteria do not apply specifically to DOAS equipment like the Munters EPX 5000, but the system evaluated here was tested under similar environmental conditions, and savings were projected using the same assumptions about system operating constraints that have been used for test of other Western Cooling Challenge equipment.

The intent of the Challenge is to advance the market introduction of commercialized products, thus encouraging participants to consider the many non-performance-based design factors such as cost-effectiveness, system reliability, and non-energy code compliance. UC Davis conducts laboratory testing and monitored field demonstration of equipment to prove real-world performance and equipment reliability, and provides transparency to the industry by serving as a neutral evaluator of alternative technologies. Following the laboratory evaluation of the Munters EPX 5000 presented here, UC Davis and PG&E have arranged at least one measured field evaluation of the equipment.

OVERVIEW OF MUNTERS EPX 5000

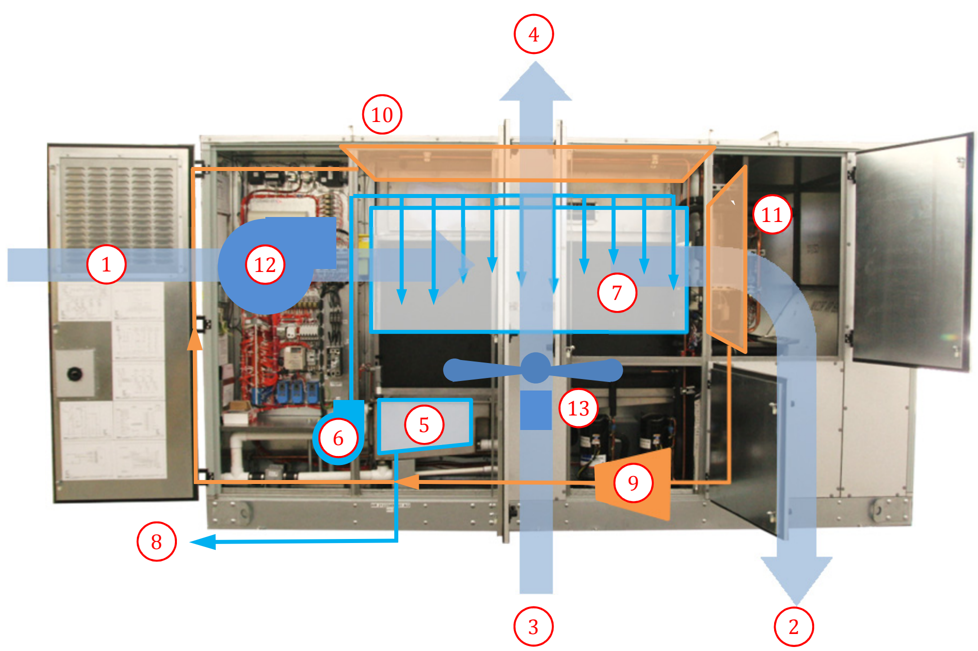
Munters' EPX 5000 is a Dedicated Outside Air System (DOAS) that utilizes an indirect evaporative heat exchanger, plus vapor compression to cool ventilation air. During the heating season, ventilation air is tempered with a gas furnace or heating hot water coil. The EPX 5000 supplies 100% fresh air, however in many instances return air can be used as the secondary air stream for the indirect evaporative heat exchanger. Regardless of the source, Munters refers to this secondary air stream as "scavenger air".

The equipment utilizes Munters' "EPX" indirect evaporative heat exchanger – a polymer construction, tube-in-flow design that is arranged in a cross-flow orientation. The secondary air stream flows upward across the outside of these tubes. During cooling hours, water is sprayed into the secondary air stream over the top of the tube array. As water flows downward over the tubes, water evaporates into the secondary air stream, and the air and water are cooled. This results in sensible heat transfer away from the primary air stream, which flows through the inside of the tube array. The heat exchanger can use outside air in the secondary air stream, or in applications where expired room air can be returned to the air handler, cooling efficiency can be improved by utilizing this lower enthalpy source as the secondary air stream. In the heating season, the return air as scavenger configuration has the added benefit of providing exhaust air heat recovery.

For many cooling hours, an indirect evaporative only mode will be adequate to maintain supply air set point temperature from the DOAS. When additional capacity is needed the EPX 5000 also includes two stages of vapor compression cooling. The condenser coil for the vapor compression system is located in the secondary air stream, downstream of the EPX heat exchanger. This location offers a thermodynamic benefit since the temperature of air at the secondary outlet can be significantly cooler than outdoors.

The EPX 5000 can be installed in a variety of different applications. For some buildings it may be desirable to supply room-neutral ventilation air, and to rely on parallel mechanical systems to condition sensible room loads. In other installations, it may be most efficient to allow the DOAS to cover a portion of the room loads, as long as the zone to which ventilation air is supplied is not overcooled. In grocery stores where DOAS equipment is preferably installed above refrigerated cases, a low supply air temperature might very easily overcool the zone. For this and other reasons, the energy savings potential for this system will depend on the configuration in which it is installed.

Generally, the EPX5000 will not have adequate capacity to address all room cooling loads in a building. However, it does have significantly more cooing capacity than is needed to deliver room-neutral ventilation air. The test results presented herein show that in western climate conditions the EPX 5000 is usually more efficient at sensible room cooling than a conventional vapor compression system operating as recirculation only. When room cooling capacity from the EPX 5000 can be used in place of less efficient cooling components in the building it will provide additional energy savings beyond addressing the ventilation cooling load. The scenario based calculated savings assessment presented in the Results section under *Assessment of Demand Savings*, assumes that the full-tilt sensible room cooling capacity of the EPX 5000 can be utilized to reduce operation of conventional vapor compression equipment.



1	*Outside Air (Primary Inlet)*	5	*Sump*	10	*DX Condenser*
2	*Supply Air*	6	*Circulation Pump*	11	*DX Evaporator*
3	*Return Air (Secondary Inlet)*	7	*EPX (Indirect Evap. Heat Exch.)*	12	*Blower (Primary Air)*
4	*Exhaust Air*	8	*Drain*	13	*Fan (Secondary Air)*
		9	*Compressors*		

FIGURE 1. CONCEPTUAL SCHEMATIC FOR MUNTERS EPX 5000

When return air is used as the source for scavenger airflow, the equipment is intended to operate with a supply airflow rate of 5000 *cfm* and a secondary airflow rate of 4000 *cfm*. In applications where the EPX 5000 must rely on outside air for the scavenger airflow, the equipment is intended to operate with supply airflow rate of 5000 *cfm* and a secondary airflow rate of 6000 *cfm*.

The EPX 5000 is generally intended to provide continuous ventilation during all building operating hours, whether for heating or for cooling. When return air is used for scavenger flow, the secondary fan is intended to operate any time the supply blower operates, and both fans are intended to run at constant speed. There is no bypass for the EPX – both the primary and secondary airflows always pass through the EPX heat exchanger and across corresponding evaporator and condenser coils. The system is programmed to shift between modes in order to maintain a supply air set point temperature which can adjust as a function of outside temperature.

In addition to measuring performance in each regular operating mode and configuration, these tests also evaluated:

1. The potential to improve performance at low ambient conditions by switching the source for scavenger airflow from return conditions to outside air conditions.
2. Performance of the EPX heat exchanger in a configuration that diverted some indirect evaporative product air for use as the scavenger air stream.

TABLE 1. MODES OF OPERATION

Mode	Primary Blower	Secondary Fan	Compressor 1	Compressor 2	Circulation Pump	Heat
Indirect Evaporative Cooling	ON	ON	OFF	OFF	ON	OFF
Indirect Evaporative & DX1	ON	ON	ON	OFF	ON	OFF
Indirect Evaporative & DX2	ON	ON	ON	ON	ON	OFF
Ventilation Only	ON	ON	OFF	OFF	OFF	OFF
Exhaust Heat Recovery	ON	ON	OFF	OFF	OFF	OFF
Heating	ON	ON	OFF	OFF	OFF	ON

TECHNICAL APPROACH & TEST METHODOLOGY

LABORATORY FACILITY

All testing was performed in the HVAC testing apparatus in the Advanced Technology Performance Lab at PG&E's San Ramon Technology Center. The apparatus consists of two side-by-side environmental chambers designed in accordance with ASHRAE Standard 37. The two chambers have independent conditioning systems for maintaining temperature and humidity, and each has its own airflow measurement station or "code tester". The airflow measurement stations follow ASHRAE standard design, and consist of a sealed box with several flow nozzles through a partition that can be opened or sealed to provide the required range of differential pressure. Variable-speed blowers on the outlets of each station can be set to maintain the desired outlet static pressures or airflow rates and compensate for the added resistance of the flow measurement system and ductwork.

The test unit was placed entirely inside the larger of the two chambers, from which it would draw its outside air at the prescribed conditions. Ductwork provided a link to the second chamber to access the return/scavenger air, and to deliver the supply air to the airflow measurement station of that chamber. The airflow measurement station of the larger chamber was used for the measurement of the exhaust airflow discharged from the top of the unit (scavenger air outlet).

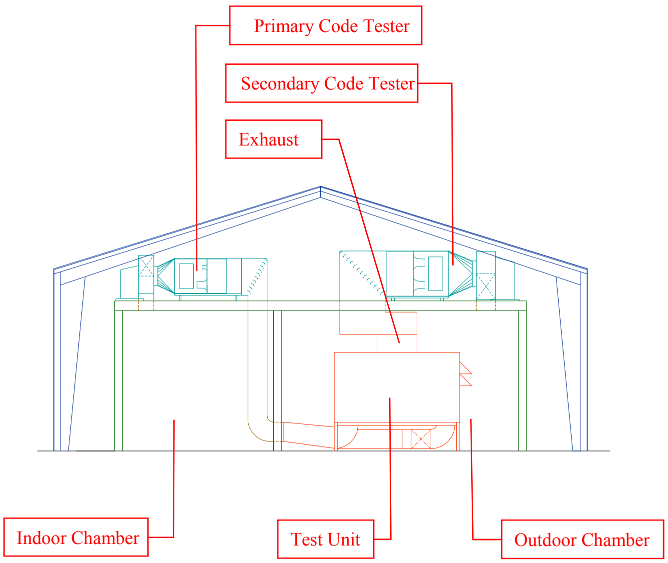


FIGURE 2. LABORATORY LAYOUT – SECTION VIEW

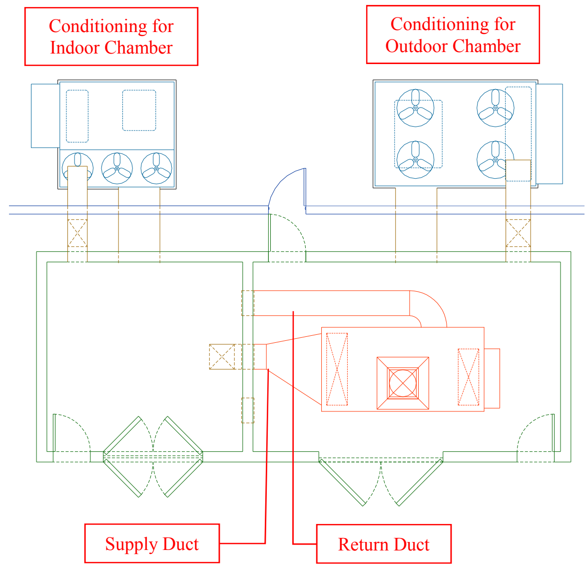


FIGURE 3. LABORATORY LAYOUT – PLAN VIEW

INSTRUMENTATION SCHEME

Table 2 documents the instrumentation and accuracy of every point of measurement for the laboratory facility. The majority of the temperature and pressure measurements were taken external to the unit, typically in the attached ductwork. Thus, the measurements do not isolate the EPX heat exchanger, and an energy balance on its performance is not possible. The 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 Compact-RIO chasses. The modules included different units for RTDs, thermocouples, voltage, current, and pulse counting, plus both analog and digital output modules for the room conditioning systems. Two of the Compact-RIO chasses were set up 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, although the weather station and scale updated once every second.

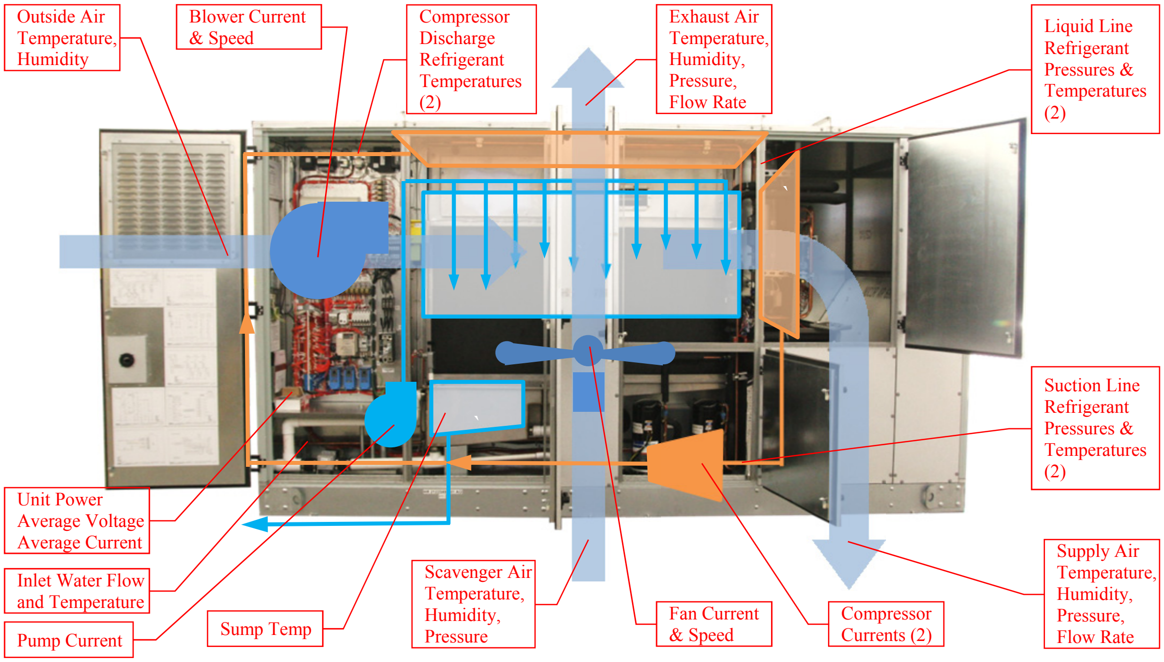


FIGURE 4. INSTRUMENTATION SCHEMATIC FOR LABORATORY TEST SETUP

TABLE 2. MEASUREMENTS AND INSTRUMENTATION FOR LABORATORY TESTS

Measurement	Instrument	Make	Accuracy
Barometric Pressure	Multi-function weather station on roof of building	Vaisala	±0.007 PSIA (±0.5 hPa)
Outside air intake dry-bulb temperature	Average of four fast-response resistance temperature detectors (RTDs) arrayed across the outside air intake.	Burns Engineering	±0.2°F
Outside air dew-point temperature	Chilled mirror dew point sensor	General Eastern Hygro-M4	±0.36°F
Supply air discharge dry-bulb temperature	Average of six fast-response RTDs inserted through wall of duct attached to test unit return.	Burns Engineering	±0.2°F
Supply air discharge dew-point temperature	Chilled mirror dew point sensor	General Eastern Hygro-M2+	±0.36°F
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% of span (-1 to 2 in. w.g.)
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 to 3 in. w.g.)
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% of span (0 to 3 in. w.g.)

Supply airflow station dry bulb temperature	Single fast-response RTD upstream of nozzles	Burns Engineering	±0.2°F
Return (scavenger) air dry-bulb temperature	Average of four fast-response RTDs inserted through wall of duct attached to test unit return	Burns Engineering	±0.2°F
Return (scavenger) air dew-point temperature	Chilled mirror dew point sensor	General Eastern Hygro-M2	±0.36°F
Return (scavenger) 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% of span (-2 to 2 in. w.g.)
Return (scavenger) air dry-bulb temperature at fan	Average of eight Type-T thermocouples attached to fan cage	Therm-X	±0.5°F
Exhaust air dry-bulb temperature	Average of eight fast-response RTDs inserted through wall of duct attached to test unit exhaust	Burns Engineering	±0.2°F
Exhaust air dew-point temperature	Chilled mirror dew point sensor	General Electric Optica	±0.36°F
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% of span (-2 to 2 in. w.g.)
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% of span (-3 to 3 in. w.g.)
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% of span (0 to 3 in. w.g.)
Exhaust airflow station dry bulb temperature	Single fast-response RTD upstream of nozzles	Burns Engineering	±0.2°F
Compressor suction pressures (2 circuits)	Pressure transmitter attached to compressor suction (vapor) line Schrader valve	Rosemount 3051C	±0.04% of span (0 to 400 psig)
Condenser outlet pressures (2 circuits)	Pressure transmitter attached to liquid line Schrader valve	Rosemount 3051C	±0.04% of span (0 to 600 psig)
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
Water supply flow	Positive displacement water meter with analog output for flow rate and pulse output for totalization	Badger M25	±1.5% of scale
Water supply temperature	Single fast-response RTD inserted into supply line	Burns Engineering	±0.2°F
Water basin temperature	Single fast-response RTD in recirculation basin near pump intake	Burns Engineering	±0.2°F
Water discharge	Catch basin on electronic scale	Measuretek	

Unit Supply Power, Voltage and Current	Clamp-on 3-element true-RMS power meter with outputs for total power, average voltage and average current	Hyoki HiTester	±0.2% of reading ±0.1% f.s.
Sub-components Line Current	Clamp-on current transmitter on one leg of the power feeding each of two compressors, two fans and one pump	NK Technologies ATR1	±1% of f.s. (±0.2 A)

The six Compact-RIO chasses 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 chasses 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 were also displayed on screen in both text and graphical form, and 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 is run on the raw data file to apply formatting, calculate statistics, and create trend charts. The result is 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 this period is identified, the statistics (average, standard deviation, range) are isolated to just this period and then copied over to another spreadsheet with one row per test. Operating performance metrics are then calculated from these values, and the results are checked for the test tolerances specified in ASHRAE Standard 37.

TEST UNIT

The Munters EPX 5000 unit supplied for testing did not have the full controls package that would be supplied with a unit for field installation. Instead, the unit had a bank of toggle switches to activate the various system components manually. A toggle switch labeled "Unit Enable" activated the primary air blower. A secondary rotary switch labeled "803 Selector" would activate the scavenger air fan as well if it were set to "Space Air", while setting it to "Outside Air" would leave it off. Toggle switches labeled "Cool #1", "Cool #2", and "Cool #3" would activate the water spray pump, and the two DX compressors, respectively. Two other toggle switches, "Heat #1" and "Heat #2", could be used to activate two stages of gas furnace heating, but these were not used in the course of these tests. The electronically commutated motor for the primary air blower had a potentiometer knob to control its speed, while the speed of the scavenger air fan could be controlled by setting the output frequency of its variable speed drive.

The test unit was configured to use return air for scavenger air, without an outside air economizer. A number of tests were done to simulate its operation with all outside air, and these were done with the return air condition set the same as the outside air, and often with the access panel to the scavenger fan opened due to the large pressure loss in the return duct. A secondary temperature measurement was added for these tests in the form of an array of eight thermocouples attached to the fan cage.

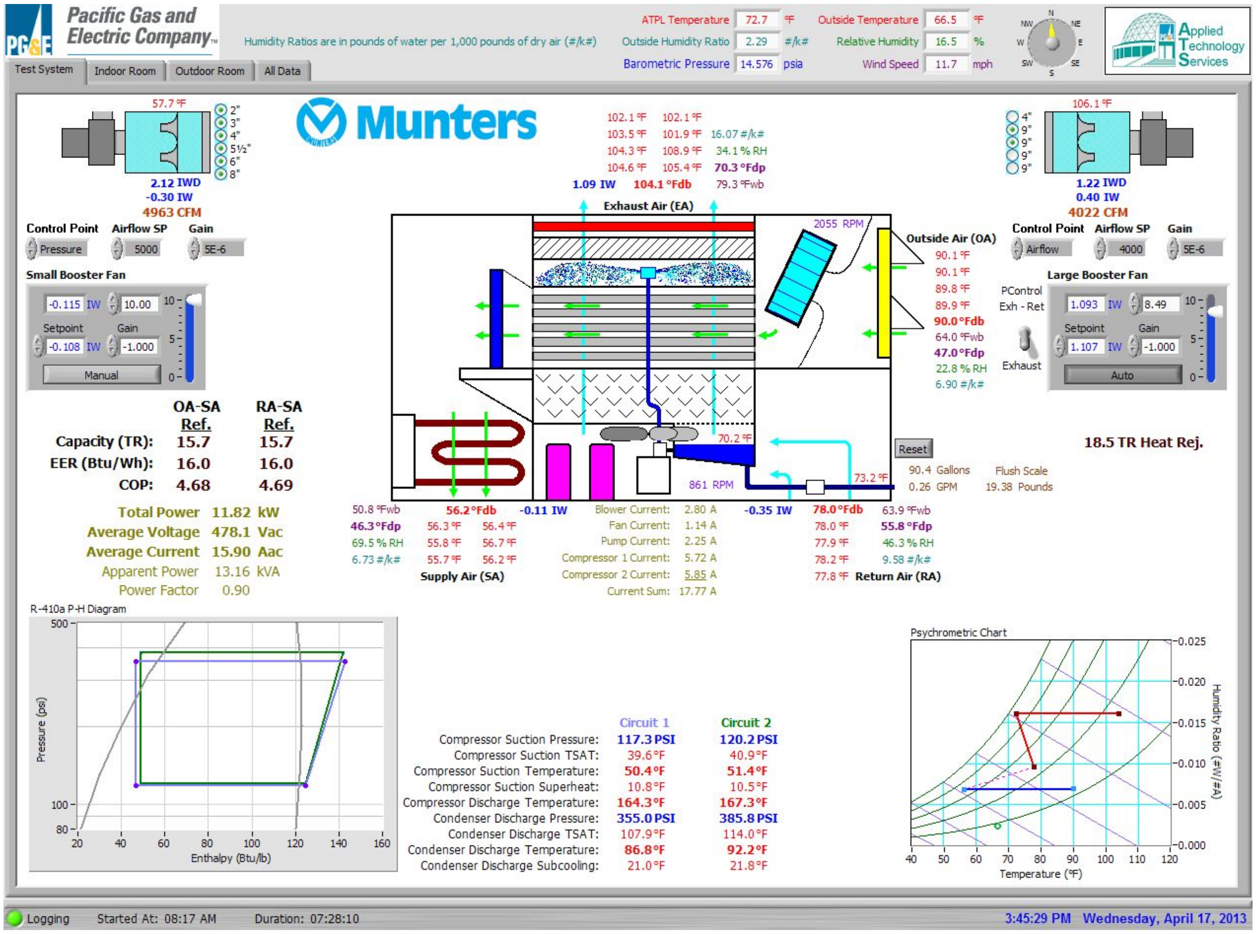


FIGURE 5. EXAMPLE SCREEN SNAPSHOT OF LABVIEW™ DATA ACQUISITION

DESIGN OF EXPERIMENTS

In order to characterize performance for all components of the test unit, in every mode of operation, and across a range of environmental conditions, laboratory tests were arranged into several groups; each designed to explore a different aspect of system operation. These included:

1. Airflow-only tests to map performance for the primary (supply air) blower, and secondary (scavenger) fan
2. System performance tests with design airflow rates (primary = 5000 *cfm*, secondary = 4000 *cfm*), using return conditions for scavenger airflow:
    - In each mode of cooling operation (1) IEC-Only (2) IEC+DX1 & (3) IEC+DX2
    - Across a range of outside air temperature and humidity conditions
3. System performance tests with reduced airflow rates, and return conditions for scavenger airflow, while maintaining a primary:secondary ratio of 5:4:
    - In indirect evaporative only mode
    - At three outside air temperature and humidity conditions
4. System performance tests with mixture of outside conditions and return conditions for scavenger airflow, and a primary:secondary ratio of 5:6:
    - In indirect evaporative only mode
    - At three outside air temperature and humidity conditions
5. System performance tests with low temperature outside air, using outside air conditions for scavenger airflow, and a primary:secondary ratio of 5:6, to test the benefit of a hypothetical "economizer" mode that could switch from return air to outside air for scavenger airflow:

- In indirect evaporative only mode
- At four low temperature outside air conditions

6. System performance tests that simulate diverting a portion of IEC product air for use as scavenger airflow:
   - In indirect evaporative only mode
   - At one outside air condition
   - For three ratios of primary:secondary airflow
7. Thermal tests to characterize dry heat exchanger performance for wintertime heat recovery ventilation

Except for testing heat recovery effectiveness for the EPX, heating performance was not evaluated. The EPX 5000 tested did include a two stage gas furnace, but the performance of this system was not assessed. Simulation and prediction of annual performance for a the system should consider whether or not the field application requires heating, and must make appropriate adjustments to fan power and supply airflow if heating components are not included. There is no reason to believe that the heating options available for the EPX 5000 would function differently from conventional systems, except that the indirect evaporative heat exchanger may provide heat recovery when the system is equipped to utilize return air for scavenger airflow.

The complete test matrix consisted of 53 separate thermal tests to characterize system performance, and 60 airflow-only tests to map fan performance. All thermal tests were conducted with return air conditions as specified by the UC Davis Western Cooling Challenge (T_{DB}=78°F, T_{WB}=64°F). System performance was not measured for sensitivity to return air conditions independently, however the sensitivity to secondary inlet conditions was measured by operation with mixtures of outside conditions and return conditions for scavenger airflow.

For test groups designed to measure sensitivity to outside air conditions, eight psychrometric points were selected to span the range of typical cooling conditions in the western United States. Where appropriate, the specific points were chosen to align with current industry standards and emerging test protocol for hot-dry 'climate appropriate' cooling equipment. The four outside air conditions defined by ANSI/AHRI 340/360 for determination of the Energy Efficiency Ratio (EER) and Integrated Energy Efficiency Ratio (IEER) were used, in addition to conditions defined by the UC Davis Western Cooling Challenge. Several of these points tested also correspond to conditions defined for U.S. DOE EERE's High Performance Rooftop Unit Specification. The system was not tested at extreme operating conditions for either temperature or humidity.
Table 3 lists each outside air conditions used for tests, and Figure 6 marks these conditions on a psychrometric chart.

TABLE 3. RANGE OF TEST CONDITIONS USED FOR DESIGN OF EXPERIMENTS

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

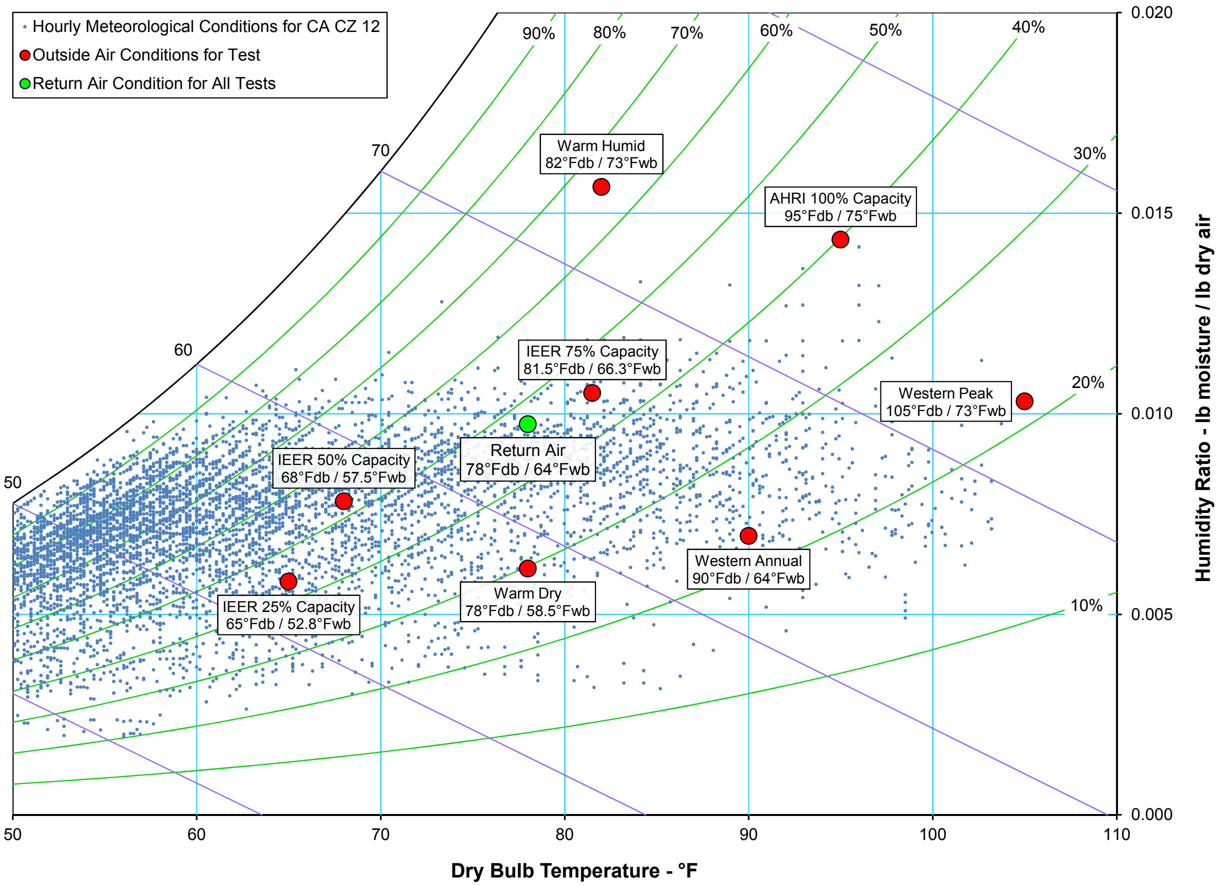


FIGURE 6. PSYCHROMETRIC CHART – RANGE OF TEST CONDITIONS

ASSESSING DEMAND SAVINGS

The energy efficiency of a DOAS cannot be directly compared to the nominal efficiency of a standard machine as determined by ANSI/AHRI 340/360. This is mainly because DOAS operate with 100% outside air, and standard rooftop air conditioners are rated whilst operating without any outside air whatsoever. Further, savings cannot be measured by simply comparing energy use for the Munters EPX 5000 against that of a standard RTU operating with 100% outside air because the application of a DOAS system also impacts the operation and performance of other rooftop units in a building that would usually operate to provide ventilation.

A complete assessment of energy savings potential must be made by comparing specific scenarios. Peak reduction and annual energy savings will therefore depend significantly on application and climate. For the analysis presented here, the savings potential was determined by comparing a scenario where the whole building ventilation is provided by the EPX 5000 to a baseline scenario where ventilation for the same building is provided by individual rooftop packaged air conditioners. The operating conditions and room cooling load for these scenarios were determined to align with the constraints developed for the UC Davis Western Cooling Challenge. The ventilation rate for both scenarios was fixed to 5000 *cfm*, corresponding to the intended application for the Munters EPX 5000.

The Western Cooling Challenge requires that individual rooftop units operate with 120 *cfm per ton* nominal cooling capacity; therefore the baseline scenario was defined as a building with five conventional 8.33 *ton* (nominal-net) rooftop units, each operating with an outside air fraction adequate to provide 5000 *cfm* ventilation for the whole building. Performance for these rooftop units was based on publicly available manufacturer stated data for the Lennox Landmark Series standard efficiency rooftop unit (KGA120S4B).

External resistance was applied to these systems according to a system curve that produces 0.7"WC external static pressure at 350 *cfm* supply air per ton (nominal). At peak conditions (T_{DB}=105°F, T_{WB}=73°F) electrical demand in the baseline scenario was calculated for all systems operating at full capacity. At "Annual" conditions (T_{DB}=90°F, T_{WB}=64°F) electrical demand was calculated for the load factor that would result in sensible room cooling of 80% the what was delivered at peak conditions. The approach assumes that the quasi steady state power draw for the systems scales equally with the sensible load factor, and disregards any gains from continuous ventilation whilst the compressors are off.

The alternative scenario applied the Munters EPX 5000 to serve all ventilation for the building. The analysis assumes that the EPX 5000 uses return air conditions for the scavenger airflow. Analysis of this scenario reassessed the number of conventional rooftop units that would be needed in order for the combination to provide the same sensible room cooling capacity as the baseline scenario at "Peak" conditions. At Annual conditions, the required room cooling was again made equal to 80% of the sensible room cooling delivered at "Peak" conditions.

The two scenarios were tested and compared at the two outside air conditions defined by the Western Cooling Challenge, plus at a lower temperature ambient condition (T_{OSA}=65°F_{WB}/52.8°F_{DB}) when conventional rooftop units would operate in an economizer mode. The low temperature comparison modeled the Munters EPX 5000 operating alone in IEC-only mode, and modeled the baseline as the number of conventional rooftop units that would be required to deliver an equal sensible room cooling in economizer mode. The sensible room cooling delivered by both scenarios at this low temperature condition was approximately 25% the sensible room cooling delivered at peak. The analysis, presented in the *Results* section under *Assessment of Demand Savings,* should give a fair indication of the potential for reduced demand at peak conditions in hot-dry climates. The analysis does not represent an annual savings estimate since this cumulative savings will vary significantly by application and climate.

DATA ANALYSIS

CALCULATING WET BULB EFFECTIVENESS

Wet bulb effectiveness (WBE) measures the extent to which an evaporative cooler is able to cool toward the wet bulb temperature of the inlet air. This metric tends to remain steady for a given system configuration even while meteorological conditions and system cooling capacity vary. 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.

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

The metric has traditionally been used to describe performance of direct evaporative coolers, but it can also be applied to indirect evaporative equipment. Since indirect evaporative heat exchangers use a secondary air stream that can have an inlet wet bulb temperature that is lower than that of the primary inlet, it is possible to achieve better than 100% effectiveness.

Describing performance in terms of wet bulb effectiveness offers good conceptual comparison against conventional evaporative coolers, but since the metric does not align with the physical heat transfer mechanisms active in the indirect evaporative heat exchanger, it does not give rise to a useful empirical correlation that can used for system and building modeling. Alternative metrics use 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 wet bulb effectiveness is used to describe equipment performance here, as well as an indirect wet bulb effectiveness that accounts for the wet bulb potential in the secondary air stream:

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

CALCULATING COOLING CAPACITY

The system cooling capacity for the equipment at any given condition is determined according to the supply airflow rate and the specific enthalpy difference between the outside air inlet and supply air, as described by equation 5; this is the net cooling produced by the system, including what is lost due to fan heat.

$$\dot{H}_{System} = \dot{m}_{SA} \cdot (h_{OSA} - h_{SA}) \qquad 3$$

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

The room cooling capacity, given by equation 5, is the cooling that is actually serviced to the room, accounting for the portion of the system cooling capacity that goes toward cooling ventilation air to the room air condition. In the case when outside air is cooler than return air, room cooling may be greater than the system cooling.

$$\dot{H}_{Room} = \dot{m}_{SA} \cdot (h_{RA} - h_{SA}) \qquad 4$$

The Western Cooling Challenge is generally concerned with a system's ability to produce sensible cooling; since ambient humidity in hot-dry climates doesn't typically require dehumidification for comfort. Thus the sensible room cooling is determined according to:

$$\dot{H}_{Room}^{sensible} = \dot{m}_{SA} \cdot C_p \cdot (T_{RA} - T_{SA}) \qquad 5$$

And the latent room cooling is determined as:

$$\dot{H}_{Room}^{latent} = \dot{H}_{Room} - \dot{H}_{Room}^{Sensible} \qquad 6$$

The ventilation cooling capacity is the difference between the system cooling and room cooling, and it can also be calculated according to equation 8.

$$\dot{H}_{ventilation} = \dot{m}_{SA} \cdot (h_{OSA} - h_{RA}) \qquad 7$$

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

CALCULATING HEAT RECOVERY EFFECTIVENESS

Air-to-air heat recovery effectiveness is defined by ASHRAE Standard 84 as the ratio of actual heat transferred between participating air streams, and the maximum theoretical heat transfer potential:

$$\varepsilon = \frac{Actual\ Heat\ Transfer}{Theoretical\ Heat\ Transfer\ Potential} \qquad 8$$

The heat recovery effectiveness is only appropriate to describe thermodynamic performance of the Munters EPX heat exchanger when operating dry. This should occur in practice during the heating season, and during any period when supply air temperature and room conditioning can be satisfied without active cooling. In any scenario when outside air is cooler than return air the dry heat exchanger will transfer heat from return air to supply air.

The Munters EPX does not transfer moisture between air streams, and does not allow cross stream transfer, except potentially by leakage between cabinets, or external recirculation between the exhaust and fresh air inlet. Therefore, sensible effectiveness is sufficient to describe heat recovery performance:

$$\varepsilon_{sensible} = \frac{\dot{m}_{1st} \cdot c_{p\,1st} \cdot (T_{OSA} - T_{PA})}{C_{min} \cdot (T_{OSA} - T_{RA})} \qquad 9$$

where

$\dot{m}_{1st}$ = mass flow rate of the primary air stream

$c_{p\,1st}$ = specific heat capacity for primary air stream

T_{OSA} = dry bulb temperature of outside air (primary stream inlet)

T_{PA} = dry bulb temperature of product air (primary stream outlet)

C_{min} = smaller of $\dot{m}_{1st} \cdot c_{p\,1st}$ and $\dot{m}_{2nd} \cdot c_{p\,2nd}$

T_{RA} = dry bulb temperature of return air (secondary stream inlet)

Calculating Energy Efficiency Ratio

Energy efficiency for the Munters EPX 5000 at any given operating condition is expressed as ratio of useful thermal capacity delivered (in units of *kbtu/h*) to electrical power consumed (in units of *kW*) by the system – the Energy Efficiency Ratio (in units of *kbtu/kWh*):

$$EER = \frac{Thermal\ Energy\ Delivered}{Electrical\ Energy\ Consumed} = \frac{\dot{H}}{\dot{E}_{system}} = \left\{\frac{kbtu}{kWh}\right\} \qquad 10$$

The EER numbers presented here should not be confused with or compared directly to the AHRI nominal EER, SEER, or IEER values for conventional air conditioners, since those metrics are developed to describe performance at very specific conditions which are not appropriate for the Munters EPX 5000. For this report, EER is presented as a generic metric that varies with conditions, and with frame of reference.

For rooftop packaged air conditioners, the Western Cooling Challenge performance criteria only credits a portion of the total cooling delivered by a system. Cooling of any excess ventilation air is not counted as useful thermal energy service, since the conventional alternative would not have to carry the increased load associated with this excess outside air. However, since the Munters EPX 5000 is intended to cover the entire ventilation requirement for a building, the performance results presented here do credit the entirety of the ventilation cooling.

The analysis presented here only credits sensible cooling capacity, since dehumidification is usually not required for commercial cooling applications in California. This assumption is in line with other evaluations for the Western Cooling Challenge. The Sensible Energy Efficiency Ratio can be expressed as:

$$EER_{sensible} = \frac{\dot{H}_{sensible}}{\dot{E}_{system}} \qquad 11$$

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

$$EER^{sensible}_{system} = \frac{\dot{H}^{sensible}_{system}}{\dot{E}_{system}} \qquad 12$$

and

$$EER^{sensible}_{room} = \frac{\dot{H}^{sensible}_{room}}{\dot{E}_{system}} \qquad 13$$

RESULTS

CHARACTERIZATION OF SYSTEM PERFORMANCE

FAN MAPPING

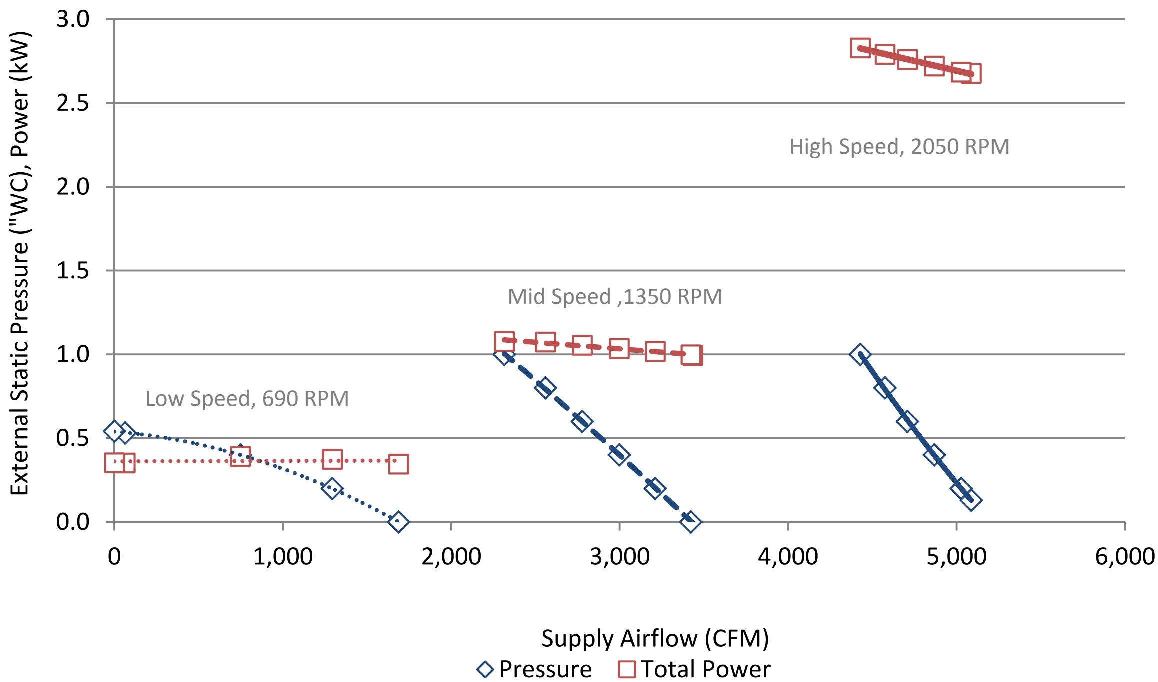


FIGURE 7: PRIMARY BLOWER AIRFLOW, POWER CONSUMPTION & EXTERNAL STATIC PRESSURE AT THREE OPERATING SPEEDS

Figure 7 charts the results from airflow only testing of the primary blower at various levels of external resistance for three fan speeds. 2050 RPM corresponds to the highest speed allowable by the motor and controls on-board the test unit. The measurements at each fan speed start at external static pressure of 1.0*"WC* and work toward zero external resistance. As a result, the map appears truncated at 1.0*"WC*. The map references external static pressure for the unit, which is measured as the difference between static pressure at the outside air inlet and static pressure in the supply duct. Thus, this is not a true fan mapping because it includes all of the internal resistances within the unit (EPX, evaporator coil).

Results for power refer to the whole unit power at the described airflow condition, which includes ancillary power for electronics and crankcase heaters. The pump, compressors, and scavenger fan were off.

Notably, the design supply airflow for this unit was only achieved at full speed for external static pressure of less than 0.25*"WC*. Adjusted for standard conditions the fan achieved maximum airflow of 4,925 *scfm* at 0.13 *"WC* external static. The design airflow of 5000 *cfm* is clearly at the highest end of the fan operating space. At 1.0*"WC* external static pressure the primary airflow dropped to 4,429 *cfm*.

For all thermal tests at design flow rates, the system was run with primary airflow of 5,000 *cfm*, which corresponded to a range of roughly 4,800 *scfm* – 5,100 *scfm*, depending on supply air conditions. Reaching this primary airflow set point required operating with external static pressure between -0.12 *"WC* and +0.18*"WC*. It seems very likely that in-field conditions will impose greater external resistance than this and that the actual delivered supply airflow will vary accordingly.

Similarly, the fan power reported for each thermal test reflects the flow and resistance conditions that were imposed to achieve the desired mass flow rate. Therefore, power draw for the primary blower in field application may vary from these measurements. In fact, the fan power recorded for some tests at design airflow is roughly 25% lower than what would be expected from the fan performance map shown in Figure 7. The energy required to boost the airflow through the system was applied external to the unit and not measured, and thus not accounted for in the energy analysis.

Munters later disclosed that the fan used in this early model (an EBM Pabst 2730 Watt fan) is different than what has been selected for their production units (a Rosenberg 3300 Watt fan). The higher power fan is expected to deliver the design airflow.

Further, upon completion of these laboratory tests, Munters revised design of the primary blower for the EPX 5000 system, replacing the primary blower tested here for a different centrifugal fan with high power draw and larger static-flow capability. The equipment laboratory tested utilized the EBM Papst R3G4560 (2730 W nominal), while equipment now in production utilize the Rosenberg GKH CIA450 (3300 Watt nominal). The manufacturer published full speed fan curves for each of these fans is presented in Figure 8. It appears that the new fan should better serve the full 5000 *cfm* design airflow under realistic operating scenarios. However, the power draw for this new fan will also be larger than the performance results published herein. Modeled predictions of equipment performance should correct for these differences.

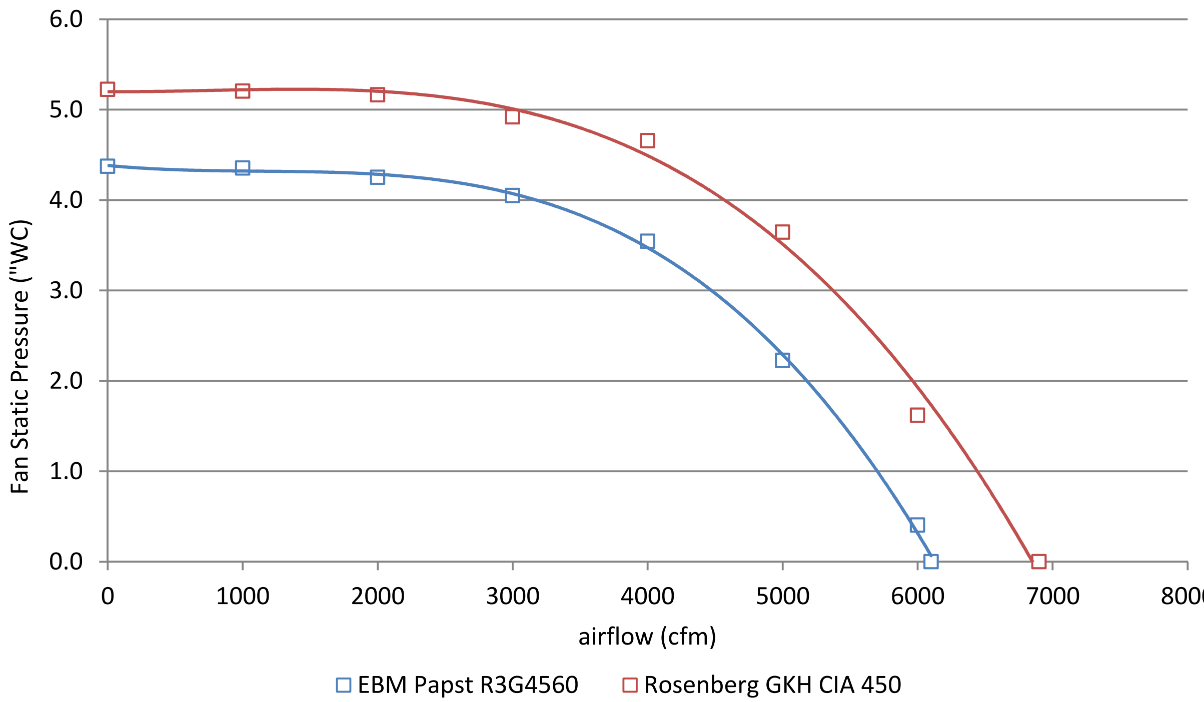


FIGURE 8: MANUFACTURER STATED FAN CURVES FOR PRIMARY BLOWER TESTED, AND FOR NEW SELECTION FOR EPX 5000

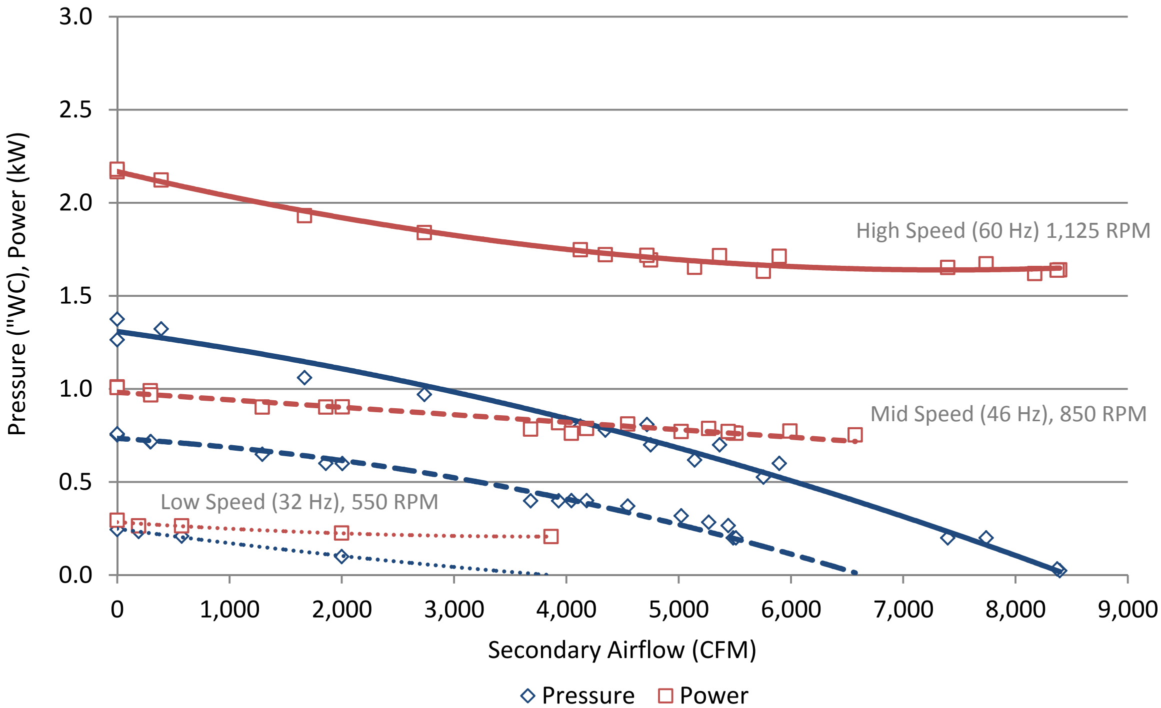


FIGURE 9: SCAVENGER FAN AIRFLOW, POWER CONSUMPTION & DIFFERENTIAL PRESSURE AT THREE OPERATING SPEEDS

Airflow only tests for the scavenger fan ran at three different speeds and adjusted external resistance to several points in order to map the entire fan curve between zero flow and zero resistance. 1,125 RPM corresponds to the highest speed allowed by the VFD control. Since the primary and secondary fans were both in operation for this test, power for the scavenger fan was determined by taking the total unit power measurement and subtracting out the power for the primary fan and ancillary components. The primary fan was held at a fixed combination of speed (full) and airflow (5,000 cfm) throughout these tests, and the power to be subtracted was that determined from the previous test phase.

At high speed (1,125 *RPM*) the fan was able to deliver the design secondary airflow of 4,000 *cfm* at roughly 0.8*" WC* external static pressure. When outside air is used as the source for scavenger air the design secondary airflow rate is intended to shift to 6,000 *cfm*, which is achievable at full speed up to roughly 0.5*"WC* external static pressure. It is unlikely that most field applications will experience external resistance in the return airflow as high as 0.8*"WC*, which means that most applications would likely run the secondary fan at part speed.

Subsequent thermal tests were conducted in a way that would operate with the design secondary airflow rate, or the part flow rate intended for test. This was accomplished by adjusting both the external resistance and the secondary fan speed. External static pressure for the secondary airflow for these tests ranged between 0.25*"WC* and 0.4*"WC*, The scavenger fan power reported for each test reflects the flow and resistance conditions that were imposed for each test. Therefore, power draw for the scavenger fan may vary from these measurements depending on field application.

Performance with Design Airflows

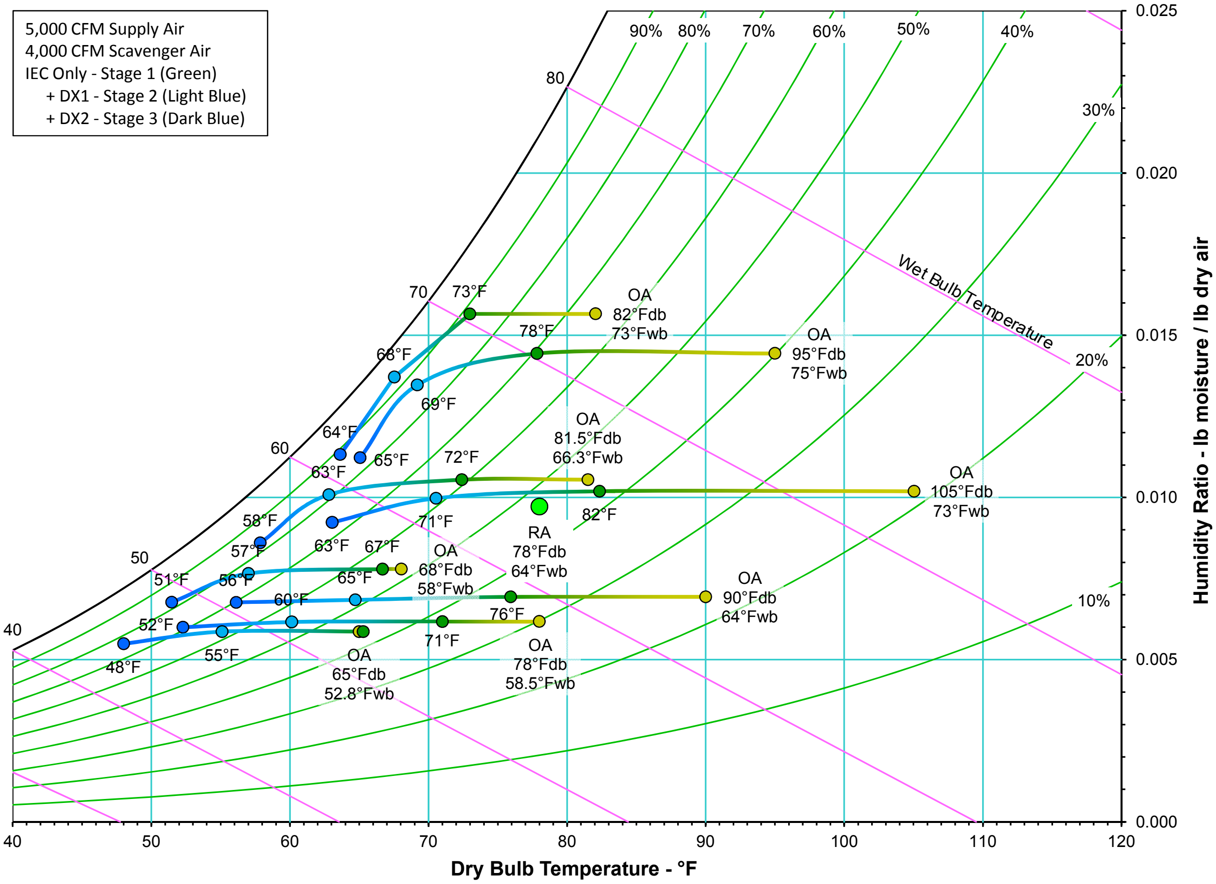


Figure 10. Psychrometric Chart – Performance in Each Mode of Operation for Eight Outside E Air Conditions

When configured to use return air as the source for scavenger flow the EPX 5000 is intended to supply 5000 *cfm* fresh air ventilation, and to draw 4000 *cfm* return air through the scavenger fan. Since this is the intended design configuration, it was used as the basis for mapping system performance across a range of outside air conditions.

Laboratory tests were conducted for the system in all three modes of cooling operation, at design airflows, and at eight separate outside air conditions. For all tests, return air conditions were maintained at 78°F_{DB}/64°F_{WB} as specified by the UC Davis Western Cooling Challenge.

Figure 10 marks the eight outside air conditions tested, as well as the corresponding measured supply air conditions in each mode of operation. The return air condition used for all tests is also indicated for reference.

Figure 11 charts the sensible cooling capacity in each mode of operation for all eight outside air conditions tested. The absolute size of each blue bar represents the sensible system cooling capacity, while the location of each bar relative to the vertical axis indicates what portion of the system cooling capacity goes toward cooling ventilation air to room-neutral conditions, and what portion provides sensible room cooling. Zero on the vertical axis represents room-neutral supply air conditions. The value corresponding to the top of each blue 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 11 as a green hash, and any additional net cooling generated by the system starts from that point. If the system cooling capacity is not adequate to overcome the sensible ventilation cooling load, the shortfall will result in a sensible gain to the room. This is indicated by a red hash below the room neutral condition, and the top of the blue bar corresponds to a negative sensible room cooling.

Through review of the test results we note that system cooling capacity increases at warmer outside air temperatures, and the delivered room cooling decreases. Notably, while the IEC Only mode generates 9.75 *tons* of sensible cooling at 105°F_{DB}/73°F_{WB}, the supply air condition actually results in a sensible gain to the room. For these ambient conditions, the EPX 5000 would need to operate with at least one compressor stage.

Similarly, while sensible room cooling is greatest at low ambient conditions, the system cooling effect is generally smaller. These results indicate that when outside air conditions are cooler than return air conditions, using return air for scavenger flow may no longer provide the advantage that it does when return air is much cooler than outside air. In fact, when outside air is significantly cooler than return air, the operation of indirect evaporative cooling with return air for scavenger flow appears to barely overcome the tendency for heat recovery from the return air. IEC only mode at 55°F_{DB}/53°F_{WB} delivers roughly 5.7 *tons* room cooling, but has practically zero system cooling effect.

Following this laboratory evaluation we recommend a careful comparison of projected annual performance compared to a conventional scenario. The system clearly provides benefits at peak, but does appear to be less efficient than a conventional economizer cooling, or integrated economizer cooling at low ambient temperatures. An application that is ventilation load dominant and experiences a significant portion of the total annual cooling load while at high ambient conditions, will very likely achieve substantial savings. The savings may be less significant for buildings that are internal load dominated or experience the majority of cumulative cooling load at low ambient conditions.

Related to concerns about performance for the EPX 5000 at low temperature conditions, our design of experiments constructed a test group to evaluate the impact of switching from return air to outside air for scavenger flow when outside air conditions are lower enthalpy. These results are presented in Figure 25 and Figure 26 in section: *Performance with Low Temperature Outside Air for Scavenger Air (Economizer Mode)*.

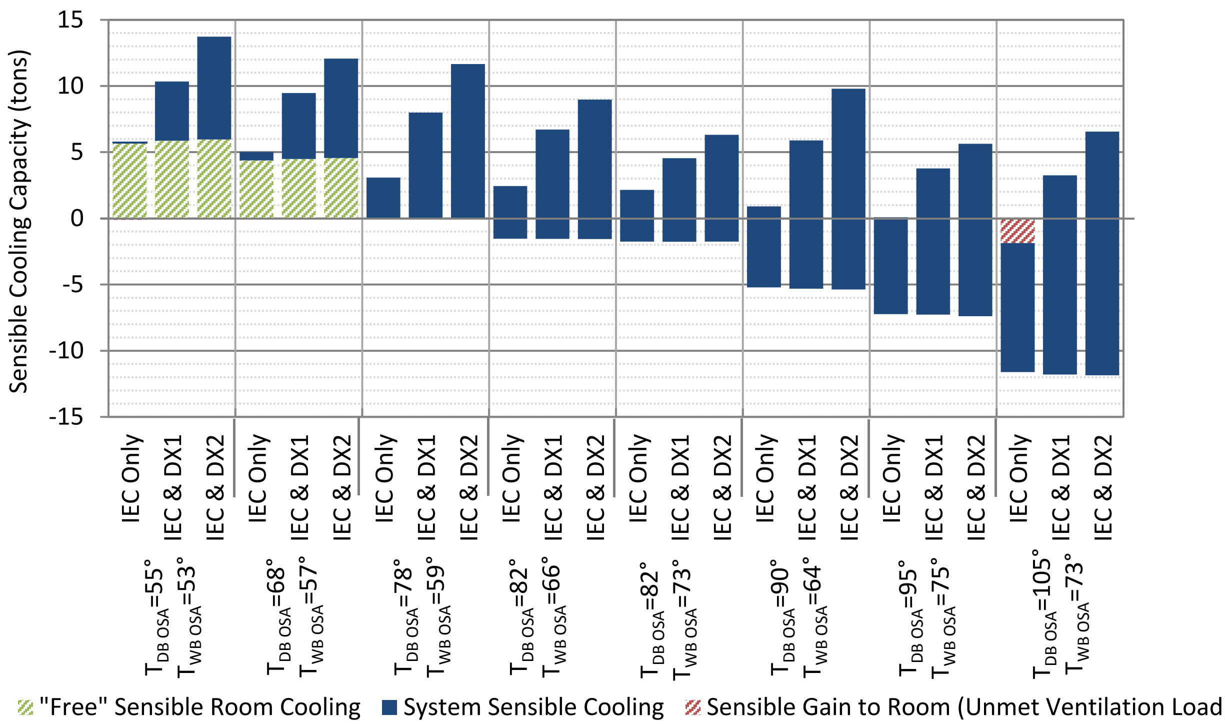


FIGURE 11: SENSIBLE COOLING CAPACITY IN EACH MODE OF OPERATION FOR EIGHT OUTSIDE AIR CONDITIONS

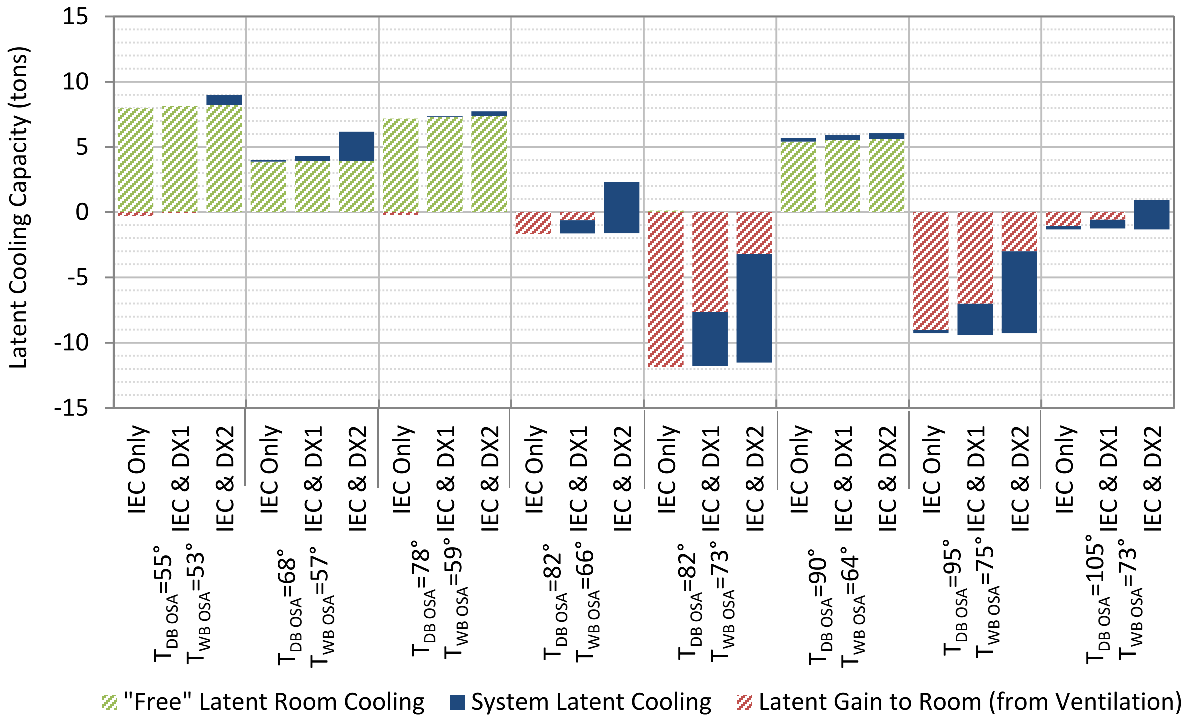


FIGURE 12: LATENT COOLING CAPACITY IN EACH MODE OF OPERATION FOR EIGHT OUTSIDE AIR CONDITIONS

Figure 12 charts the latent cooling capacity in each mode for the same eight outside air conditions. This chart is constructed in the same form as Figure 11 except that it illustrates the latent cooling effect. Not surprisingly, the latent system cooling capacity for each stage of DX is largely dependent on the inlet humidity condition.

It is advantageous that the hybrid system allows some capacity for dehumidification. The system may be somewhat less efficient when ambient conditions are more humid, but at least the equipment is not restricted to very dry climates only. We believe the system should be appropriate over a relatively broad geography, and most efficient in hotter dryer locations. Munters recommends that the system may be applied in IEEC/ASHRAE climate zones 2-6 (b) and (c).

Figure 13 charts the power draw from each component in each mode of operation for the eight separate outside air conditions tested. Figure 14 charts the sensible system efficiency and sensible room cooling efficiency for each test.

For cooling modes with DX, power draw does increase some with increasing outside air temperature. Section *Characterization of Indirect Evaporative Cooler Performance* describes the relationship between outside air temperature and condenser inlet temperature. Those results show that condenser inlet temperature is generally between 65°*F* and 75°*F*, even when outside air temperature is as high as 105°*F*. Although these tests were not able to quantify the specific efficiency benefit associated with locating the DX condenser in the cool moist exhaust air, this configuration undeniably reduces the compressor load by substantially reducing the condensing temperature.

Appendix A chart the measured psychrometric conditions for all air nodes from each of the 24 tests with design airflow rates. The indirect evaporative exhaust (condenser inlet) conditions are shown for each test; in all cases the condenser inlet temperature is actually cooler than the return air temperature. Interestingly, for high ambient temperatures, the condenser exhaust can actually be slightly cooler than outside air.

Figure 14 illustrates that energy efficiency of this equipment depends significantly on operating conditions. Annualized estimates of savings must account for theses condition and application-specific relationships. It should be noted that for high temperature operation in indirect evaporative only mode, the sensible system cooling efficiency of the Munters EPX 5000 can be between 20 and 30 *kbtu/kWh.* The system can generally cover the sensible ventilation cooling load in this mode of operation, and adding DX operation allows the system to deliver a significant amount of room cooling as well.

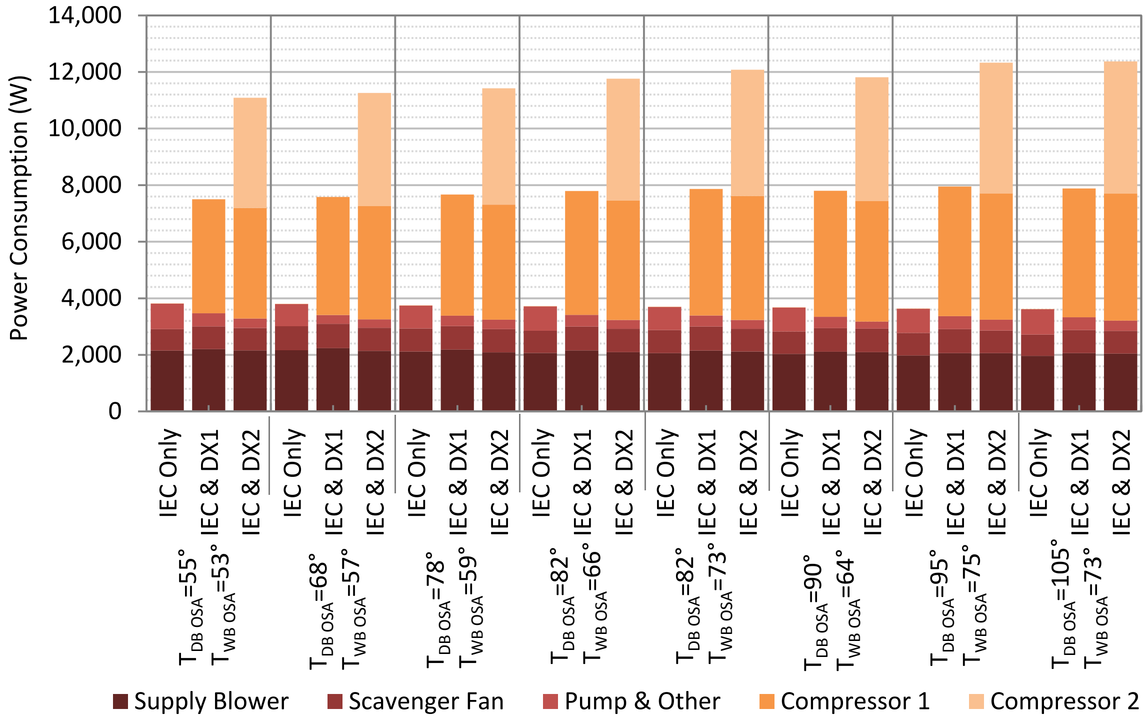


FIGURE 13: POWER CONSUMPTION FOR EACH COMPONENT IN EACH MODE OF OPERATION FOR EIGHT OUTSIDE AIR CONDIITONS

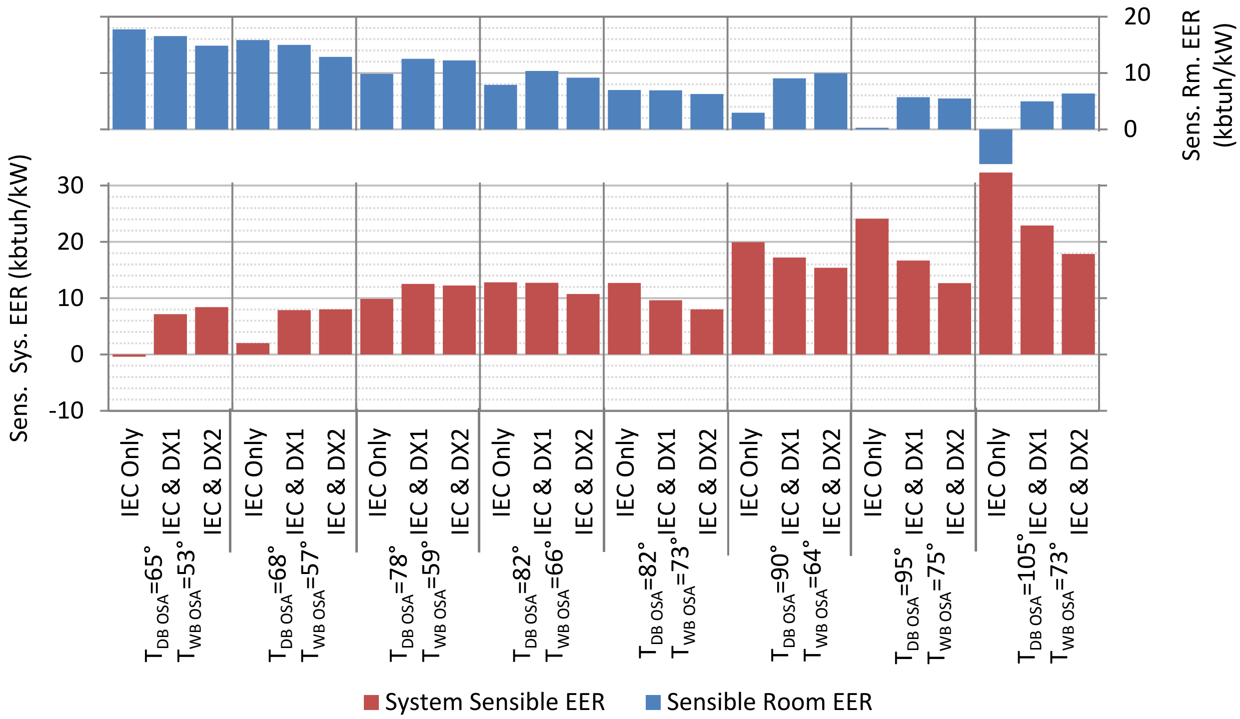


FIGURE 14: SENSIBLE COOLING EFFICIENCY IN EACH MODE OF OPERATION FOR EIGHT OUTSIDE AIR CONDIITONS

TABLE 4. SUMMARY OF PERFORMANCE IN EACH MODE OF OPERATION FOR EIGHT OUTSIDE AIR CONDITIONS WITH DESIGN AIRFLOW RATES (PRIMARY = 5,000 CFM , SECONDARY = 4,000 CFM) AND CONSISTENT SCAVENGER AIR CONDITOINS (78°FDB/64°FWB)

T_{OSA} (°*F*)		WBE		Mode	Power (*kW*)	Cooling Capacities (*tons*)					EER (*Btu/Wh*)	
DB	WB	OSA	RA			System	Room	Sensible Room	Latent Room	Vent	System	Room
105	73	71%	55%	IEC	3.6	10.0	-2.9	-1.9	-1.0	12.9	33.2	-9.7
				+ DX1	7.9	15.7	2.7	3.3	-0.6	13.0	23.9	4.1
				+ DX2	12.4	20.7	7.5	6.7	0.8	13.2	20.0	7.3
95	75	86%	55%	IEC	3.6	7.6	-8.9	0.1	-9.0	16.5	25.0	-29.5
				+ DX1	8.0	13.4	-3.2	3.8	-7.1	16.7	20.3	-4.9
				+ DX2	12.3	19.3	2.6	5.7	-3.1	16.7	18.8	2.6
90	64	54%	54%	IEC	3.7	6.4	6.6	0.9	5.7	-0.2	20.8	21.4
				+ DX1	7.8	11.6	11.8	6.0	5.8	-0.2	17.8	18.2
				+ DX2	11.8	15.6	15.9	10.0	5.9	-0.2	15.9	16.1
82	73	100%	50%	IEC	3.7	4.0	-9.5	2.3	-11.7	13.5	12.8	-30.6
				+ DX1	7.9	10.4	-3.1	4.6	-7.7	13.6	15.9	-4.8
				+ DX2	12.1	16.4	3.1	6.4	-3.3	13.3	16.3	3.1
81.5	66.3	60%	52%	IEC	3.7	4.0	0.8	2.5	-1.7	3.2	12.8	2.5
				+ DX1	7.8	9.3	6.1	6.8	-0.7	3.2	14.3	9.4
				+ DX2	11.8	14.5	11.3	9.1	2.2	3.2	14.8	11.5
78	58.5	36%	50%	IEC	3.8	2.9	10.0	3.1	6.9	-7.2	9.1	32.1

				+ DX1	7.7	8.1	15.3	8.1	7.2	-7.3	12.6	24.0
				+ DX2	11.4	12.0	19.4	14.4	4.9	-7.4	12.6	20.4
68	57.5	14%	35%	IEC	3.8	0.8	9.0	5.1	3.9	-8.3	2.4	28.4
				+ DX1	7.6	5.4	13.8	9.6	4.1	-8.4	8.5	21.8
				+ DX2	11.3	9.8	18.2	15.4	2.9	-8.5	10.4	19.4
65	52.8	-2%	-29%	IEC	3.8	-0.4	13.3	5.7	7.6	-13.7	-1.3	41.9
				+ DX1	7.5	4.4	18.4	11.2	7.2	-14.0	7.1	29.5
				+ DX2	11.1	8.5	22.7	19.2	3.5	-14.1	9.2	24.6

Performance with Reduced Airflows

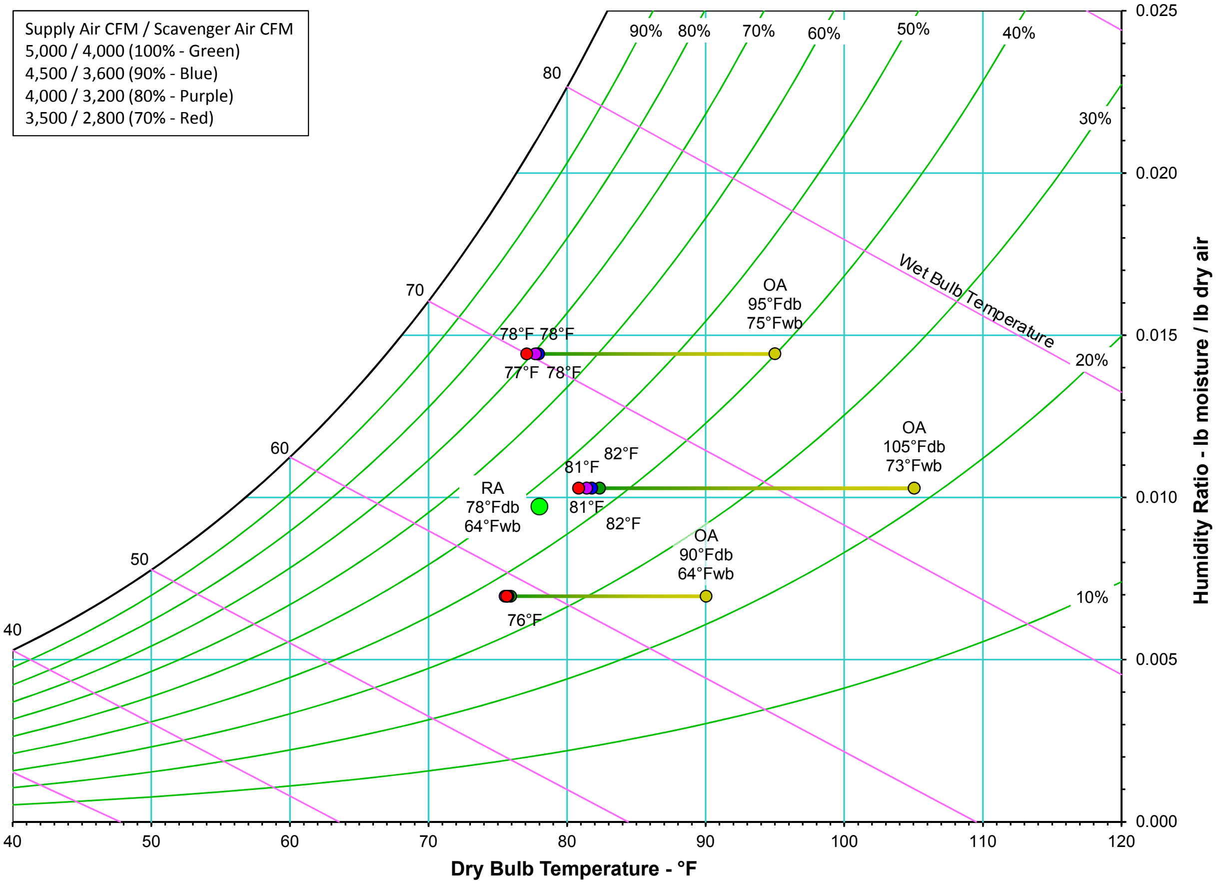


Figure 15: Psychrometric Chart – Performance in Indirect Evaporative Only Mode for Three Outside Air Conditions When Airflow is Reduced While Mainting 5:4 Primary:Secondary Flow Ratio and Consistent Secondary Air Conditions (78°F$_{DB}$/64°F$_{WB}$)

In order to assess performance sensitivity to airflow rate, the Munters EPX 5000 was tested in indirect evaporative only mode, at different airflow rates, while maintaining a 5:4 primary:secondary ratio, and using return conditions as the source for scavenger airflow. For installations that do not need 5000 *cfm* ventilation air, the EPX 5000 is designed with variable speed fans so that the flow rate on each fan can be set up to match the continuous ventilation need and an appropriate pressure balance for the space. In conducting these tests, the fan speeds were reduced proportionally to the desired reduction in airflow.

Even though the system is equipped with variable speed fans, they do not change speed as part of the regular sequence of operations – the system operates as constant volume during regularly scheduled hours. In an application that demands less than 5000 *cfm* continuous ventilation, it is intended that the machine be set up to maintain the same ratio of primary:secondary airflow as long as return air is used as the source for scavenger airflow.

Figure 15 charts the outside air conditions and supply air conditions for each reduced airflow test. Interestingly, supply air temperature and indirect evaporative effectiveness are not especially sensitive to the airflow rate. Even with primary airflow reduced to 3,500 *cfm*, the supply air temperature didn't shift by more than 1°F.

Subsequently, as Figure 16 illustrates, the sensible system cooling capacity for IEC Only mode changes proportional to supply airflow reduction. Figure 17 charts the corresponding power draw from each component for these tests. Since power consumption in the IEC Only mode is dominated by fan power, the total system draw shifts substantially with airflow reduction. At 75% airflow, the total system power is reduced by more than half.

On this account, it appears that the system may provide the most significant fractional energy savings benefits in applications where it is oversized for the continuous ventilation load. Since for certain operating conditions, this DOAS can be more efficient at sensible room cooling than a conventional rooftop unit operating as recirculation only, future development may consider a more advanced sequence of operations that operates in a very efficient part speed regime when covering the continuous ventilation demand, but which can ramp to a higher airflow to increase room cooling capacity when appropriate.

The equipment was not tested in DX modes for reduced airflow rates. While many of the observations shown here should still hold for reduced flow operation in these modes, one should note that sensible heat ratio is generally quite sensitive to airflow rate, and that reduced airflow rates would very likely increase the amount of latent cooling and decrease the sensible cooling for DX modes.

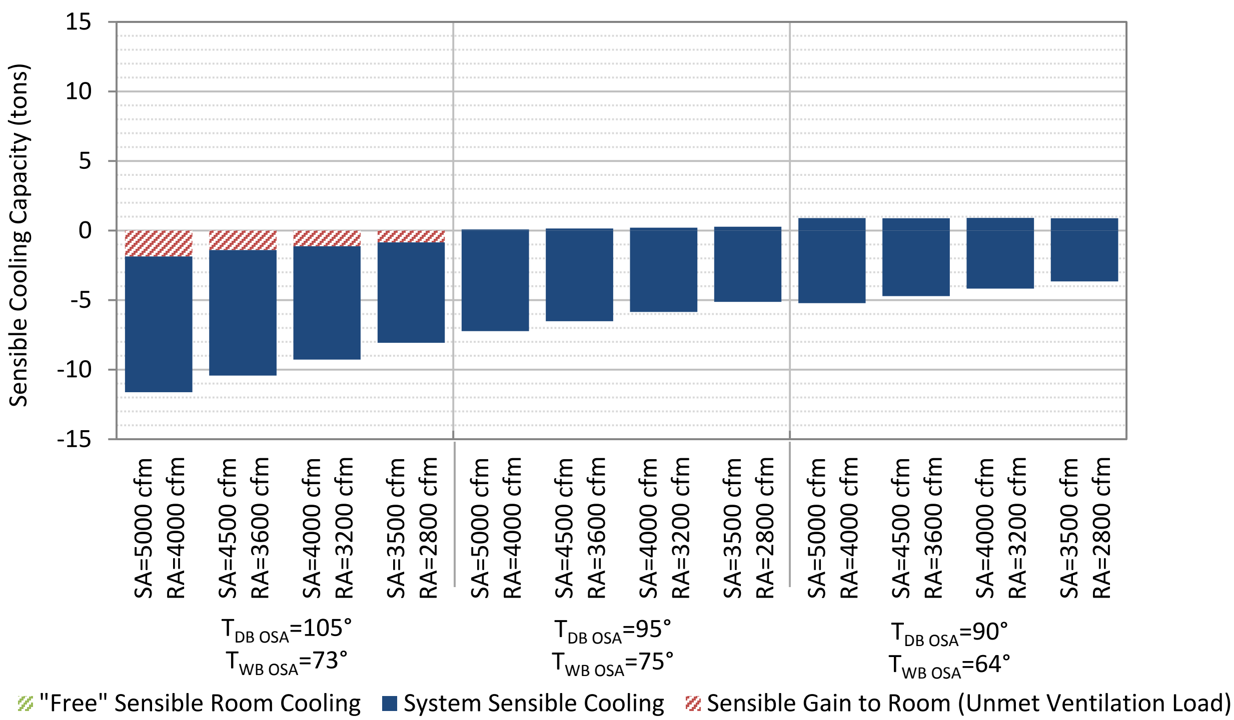


FIGURE 16: SENSIBLE COOLING CAPACITY IN INDIRECT EVAPORATIVE ONLY MODE FOR THREE OUTSIDE AIR CONDITIONS – THE EFFECT OF REDUCING AIRFLOW RATE WHILE MAINTAINING 5:4 PRIMARY:SECONDARY FLOW RATIO AND CONSISTENT SECONDARY AIR CONDITIONS (78°F_{DB}/64°F_{WB})

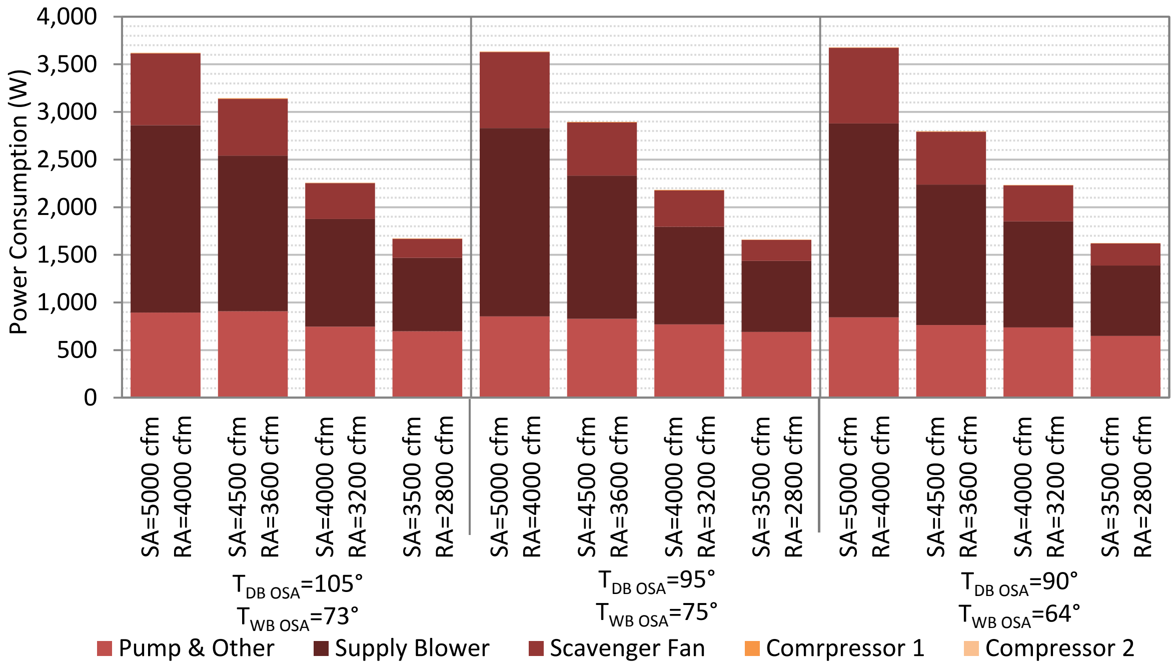


FIGURE 17: POWER CONSUMPTION IN INDIRECT EVAPORATIVE ONLY MODE FOR THREE OUTSIDE AIR CONDITIONS – THE EFFECT OF REDUCING AIRFLOW RATE WHILE MAINTAINING 5:4 PRIMARY:SECONDARY FLOW RATIO AND CONSISTENT SECONDARY AIR CONDITIONS (78°F_{DB}/64°F_{WB})

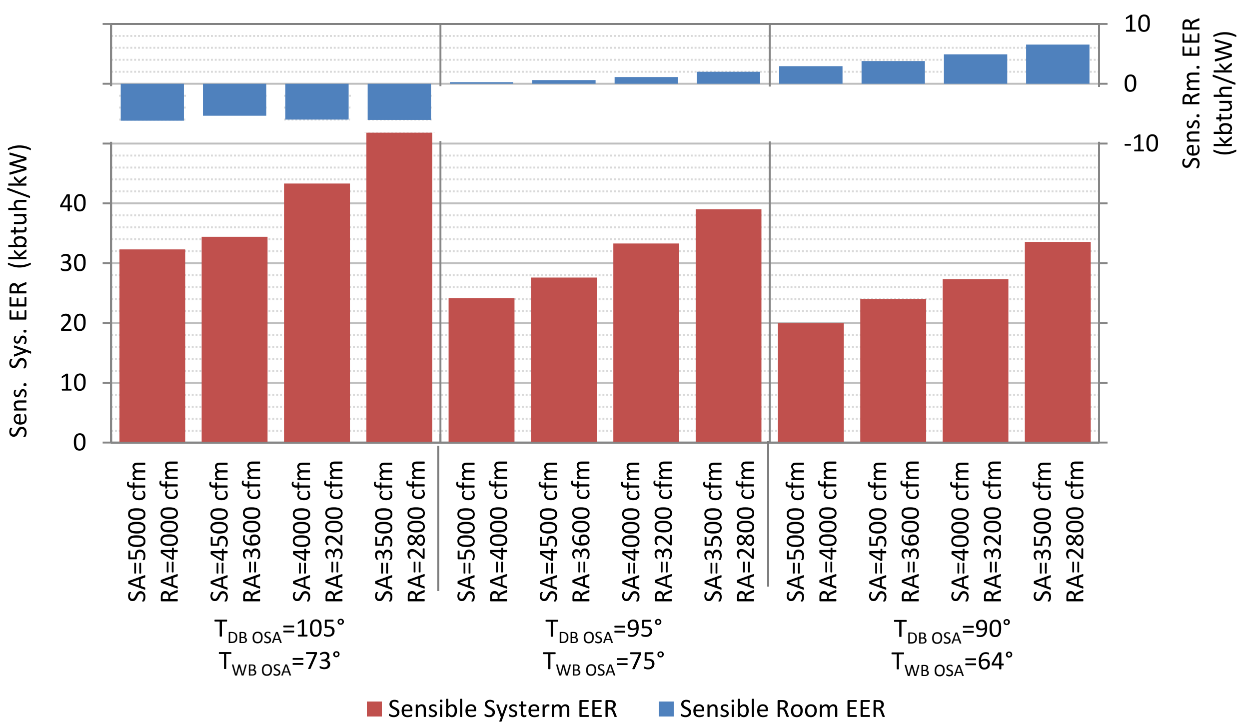


FIGURE 18: SENSIBLE EFFICIENCY IN INDIRECT EVAPORATIVE ONLY MODE FOR THREE OUTSIDE AIR CONDITIONS – THE EFFECT OF REDUCED AIRFLOW RATE WHILE MAINTAINING 5:4 PRIMARY:SECONDARY FLOW RATIO AND CONSISTENT SECONDARY AIR CONDITIONS (78°F_{DB}/64°F_{WB})

TABLE 5. SUMMARY OF PERFORMANCE METRICS IN INDIRECT EVAPORATIVE ONLY MODE FOR THREE OUTSIDE AIR CONDITIONS WITH REDUCED AIRFLOW RATES, WHILE MAINTAINING 5:4 PRIMARY:SECONDARY FLOW RATIO AND CONSISTENT SECONDARY AIR CONDITIONS (78°F_{DB}/64°F_{WB})

T_{OSA}(°F)		Airflow Rates (*cfm*)			IEC Effectiveness		Power (*kW*)	Cooling Capacities (*tons*)					EER (*Btu/Wh*)	
DB	WB	% of Design	Supply	Scavenger	OSA	RA		System	Room	Sensible Room	Latent Room	Ventilation	System	Room
105	73	100%	5,000	4,000	71%	55%	3.6	10.0	-2.9	-1.9	-1.0	12.9	33.2	-9.7
		90%	4,500	3,600	73%	57%	3.1	9.0	-2.5	-1.4	-1.1	11.6	34.5	-9.6
		80%	4,000	3,200	74%	58%	2.3	8.4	-1.9	-1.1	-0.8	10.3	44.6	-10.3
		70%	3,500	2,800	76%	59%	1.7	7.1	-2.0	-0.9	-1.1	9.1	51.1	-14.1
95	75	100%	5,000	4,000	86%	55%	3.6	7.6	-8.9	0.1	-9.0	16.5	25.0	-29.5
		90%	4,500	3,600	86%	56%	2.9	7.0	-7.9	0.1	-8.0	14.9	28.9	-32.7
		80%	4,000	3,200	88%	57%	2.2	6.0	-7.2	0.2	-7.4	13.2	32.6	-39.1
		70%	3,500	2,800	89%	58%	1.7	5.3	-6.3	0.3	-6.5	11.6	38.3	-45.3
90	64	100%	5,000	4,000	54%	54%	3.7	6.4	6.6	0.9	5.7	-0.2	20.8	21.4
		90%	4,500	3,600	55%	55%	2.8	5.5	5.5	0.9	4.6	-0.1	23.4	23.6
		80%	4,000	3,200	56%	56%	2.2	5.4	5.6	0.9	4.6	-0.2	28.9	29.9
		70%	3,500	2,800	58%	57%	1.6	4.6	4.7	0.9	3.8	-0.1	33.9	34.6

PERFORMANCE WITH MIXED RETURN AND OUTSIDE AIR FOR SCAVENGER AIR

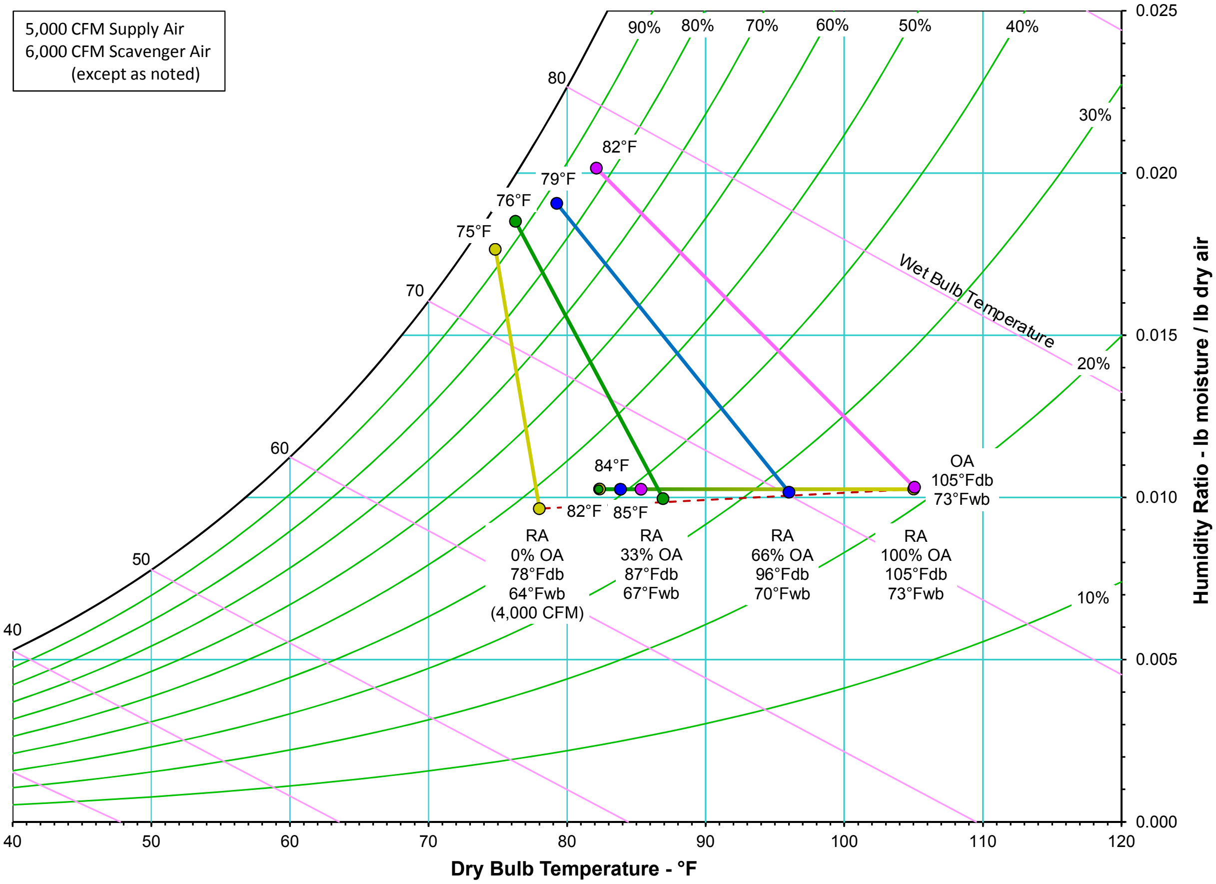


FIGURE 19: PSYCHROMETRIC CHART – PERFORMANCE IN INDIRECT EVAPORATIVE ONLY MODE WHEN OUTSIDE AIR ($105°F_{DB}/73°F_{WB}$) IS USED AS A PORTION OF THE SCAVENGER AIRFLOW INSTEAD OF RETURN AIR

In applications where adequate return air is not available as a source for scavenger airflow, the EPX 5000 is intended to use outside air instead. The system does not automatically adjust between return air and outside air, rather, the air handler must be outfit from the factory for one configuration or the other. Installations where return air may not be available include most grocery stores, or other buildings where makeup air for kitchen exhaust is drawn from the main space and typical ventilation airflow is used to maintain positive pressure for the building.

In order to evaluate the performance impact of this configuration, the equipment was tested with various mixtures of outside air and return air for the scavenger flow. When return air is not available for scavenger it is intended that the scavenger airflow be increased relative to the supply airflow, to a primary:secondary ratio of 5:6.

Figure 19 charts the outside air condition (primary inlet), scavenger inlet condition (secondary inlet), supply air condition, and exhaust air condition for four different mixtures of outside air and return air as source for the scavenger airflow while operating in indirect evaporative only mode. The outside air condition is held consistent at $105°F_{DB}/73°F_{WB}$ for all four tests recorded in Figure 19. The three tests with mixed air use an increased scavenger air flow at a primary:secondary ratio of 5:6.

Figure 20, and Figure 21 chart the same results for tests at $T_{OSA}=95°F_{DB}/75°F_{WB}$, and $T_{OSA}=90°F_{DB}/64°F_{WB}$

The test unit was factory configured with a return air pathway, and no outside air damper so the secondary inlet conditions for these tests were achieved by adjusting the return air conditions and by partially opening an access panel on the unit to allow outside air flow directly from the outdoor air chamber. The red dotted line in Figure 19 – Figure 21 charts the mixing regime for outside air and return air; each outside air fractions tested falls on this line.

These tests results indicate a notable decrease in indirect evaporative cooling capacity with a switch to outside conditions for the scavenger airflow. At peak conditions, capacity decreases by 15% and since fan power increases with a shift to 6,000 *cfm* scavenger airflow, sensible system EER falls by 30%.

The difference depends signficantly on outside air web bulb conditions, since the supply air temperature and system sensible cooling capacity depend mostly on the the difference between outside air dry bulb temperature and scavenger air wet bulb temperature. Any mixture of 90°F_{DB}/64°F_{WB} outside air and 78°F_{DB}/64°F_{WB} return air results in practically the same supply air temperature and sensible system cooling capacity. There is a very small but measureable increase to cooling capacity with the increase in scavenger airflow. This physical relationship is discussed in more detail in section: *Characterization of Indirect Evaporative Cooler Performance*

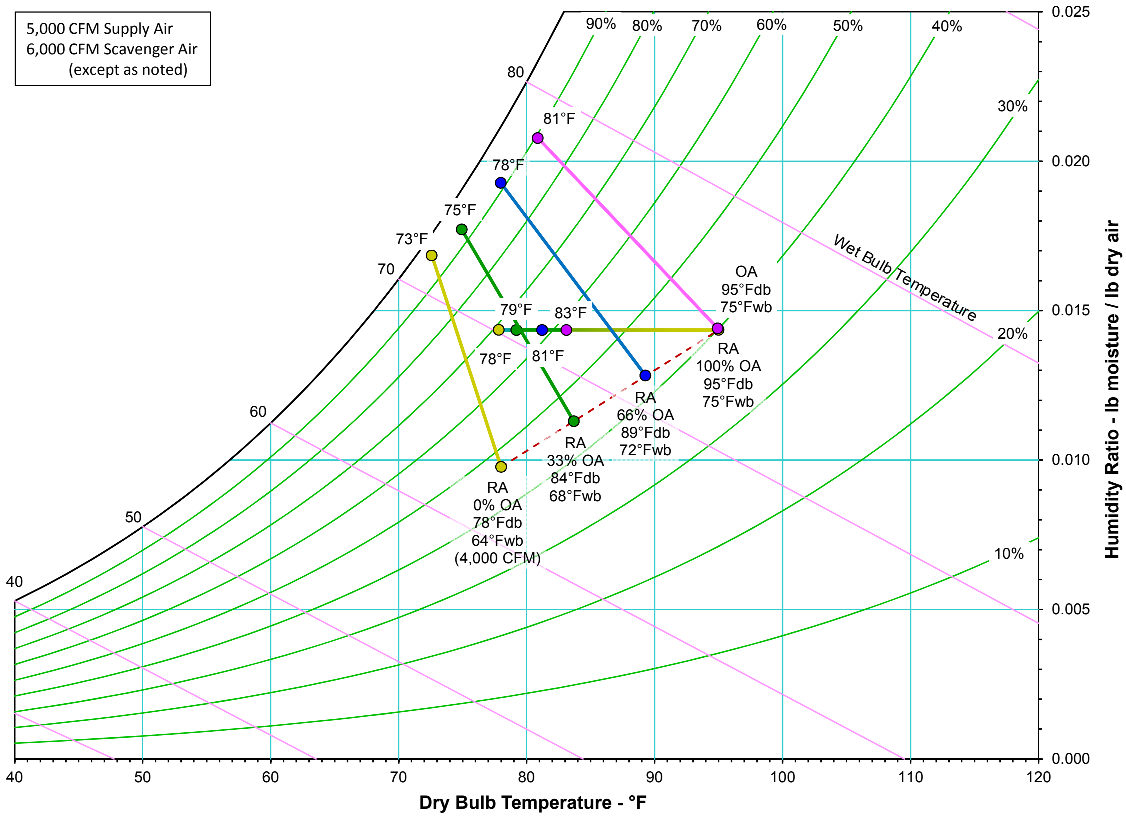


FIGURE 20: PSYCHROMETRIC CHART – PERFORMANCE IN INDIRECT EVAPORATIVE ONLY MODE WHEN OUTSIDE AIR (95°F_{DB}/T=75°F_{WB}) IS USED AS A PORTION OF THE SCAVENGER AIRFLOW INSTEAD OF RETURN AIR

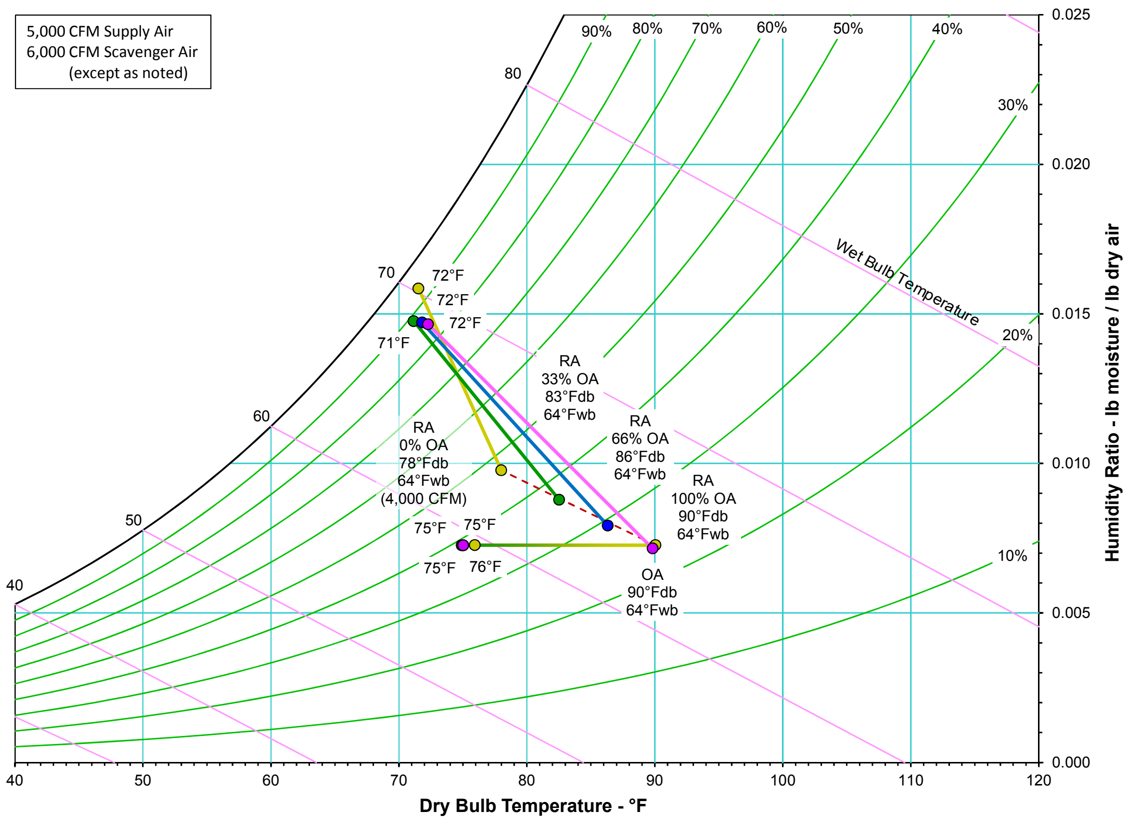


FIGURE 21: PSYCHROMETRIC CHART – PERFORMANCE IN INDIRECT EVAPORATIVE ONLY MODE WHEN OUTSIDE AIR ($90°F_{DB}$/T=$64°F_{WB}$) IS USED AS A PORTION OF THE SCAVENGER AIRFLOW INSTEAD OF RETURN AIR

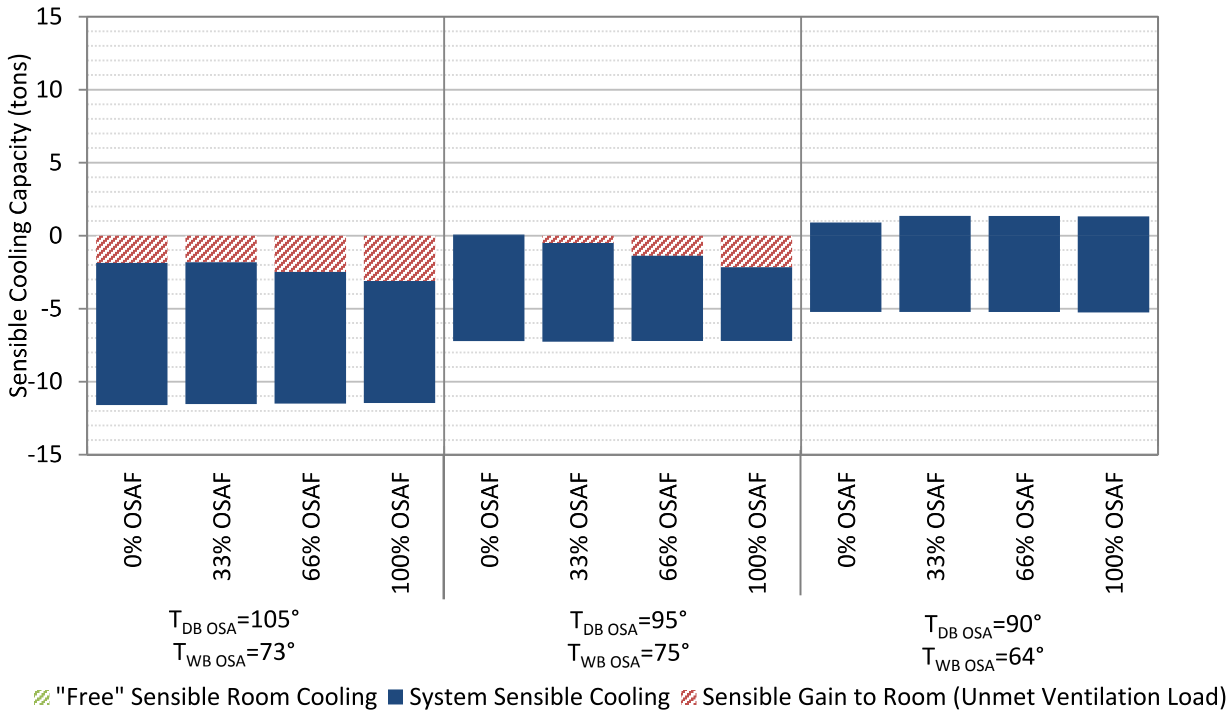


FIGURE 22: SENSIBLE COOLING CAPACITY IN INDIRECT EVAPORATIVE ONLY MODE FOR THREE OUTSIDE AIR CONDIITONS – THE EFFECT OF USING OUTSIDE AIR FOR SCAVENGER FLOW INSTEAD OF RETURN AIR.

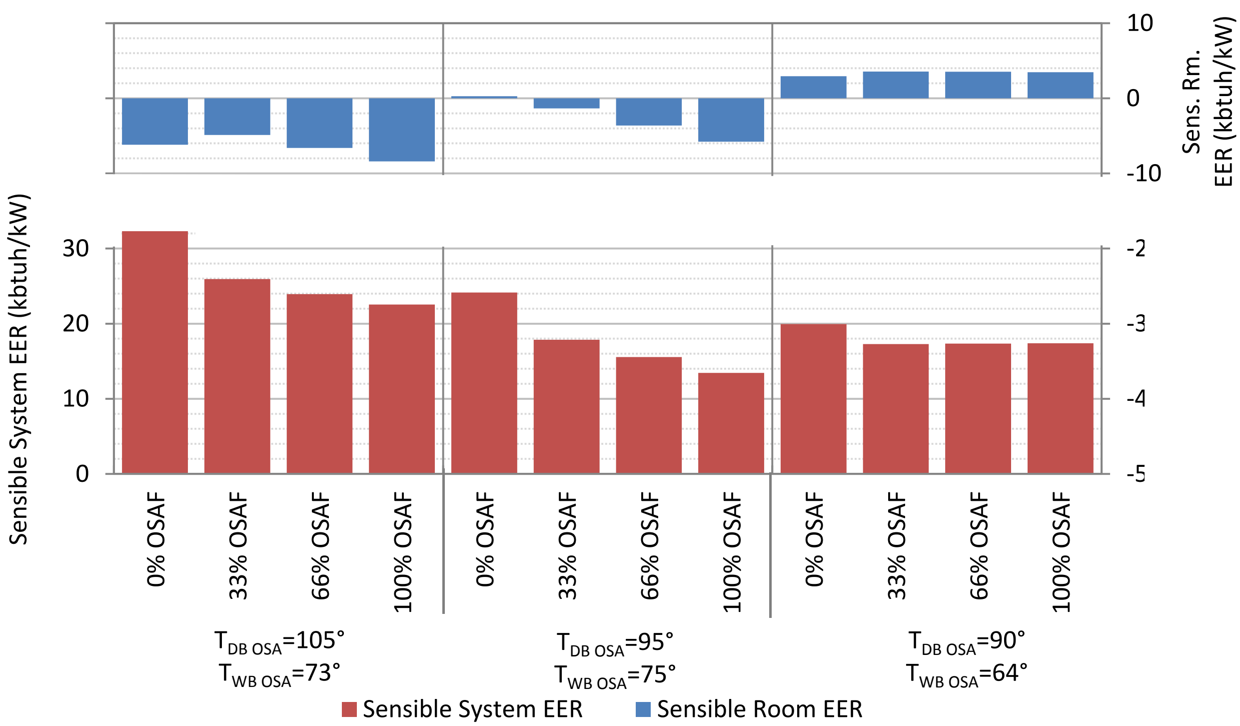


FIGURE 23: SENSIBLE COOLING EFFICIENCY IN INDIRECT EVAPORATIVE ONLY MODE FOR THREE OUTSIDE AIR CONDITIONS – THE EFFECT OF USING OUTSIDE AIR FOR SCAVENGER FLOW INSTEAD OF RETURN AIR

PERFORMANCE WITH LOW TEMPERATURE OUTSIDE AIR FOR SCAVENGER AIR (ECONOMIZER MODE)

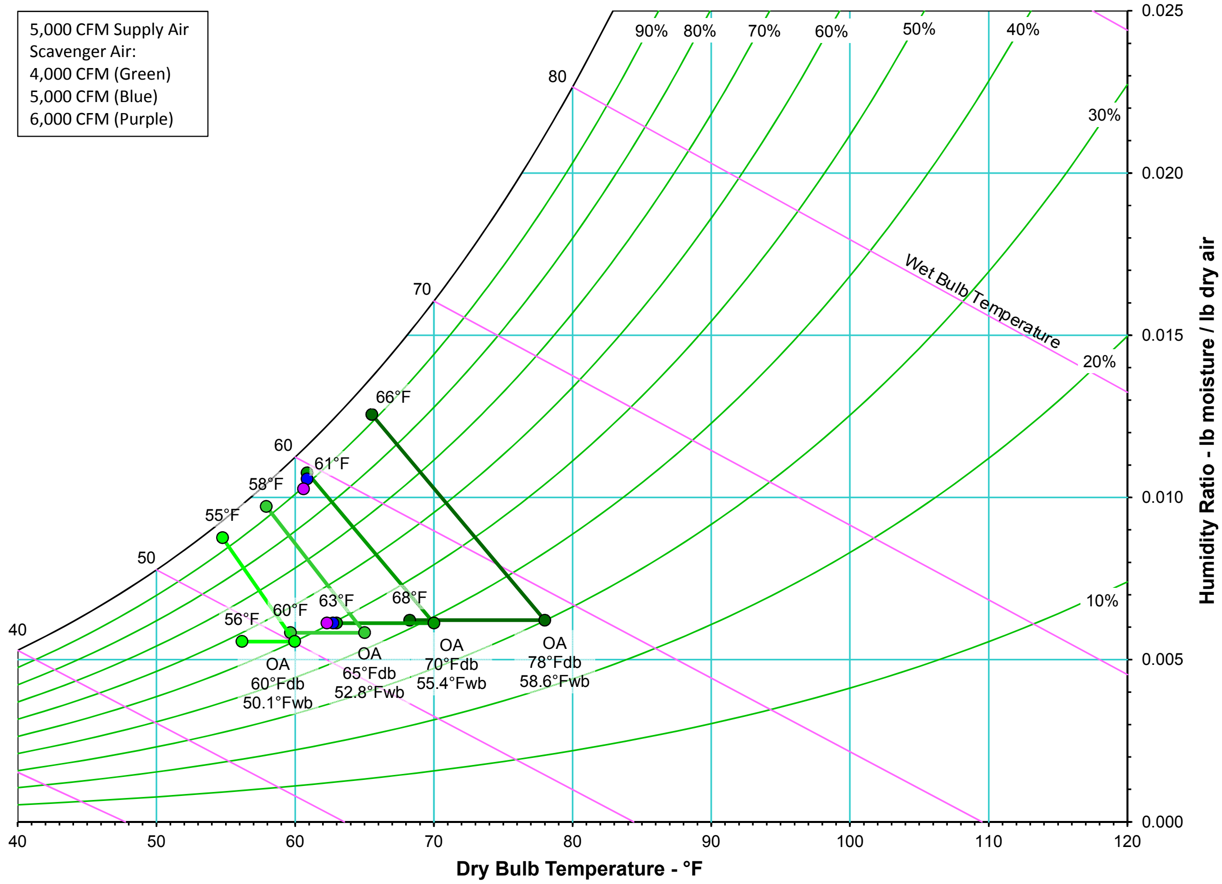


FIGURE 24: PSYCHROMETRIC CHART – PERFORMANCE IN INDIRECT EVAPORATIVE ONLY MODE FOR FOUR LOW-TEMPERATURE OUTSIDE AIR CONDITIONS WHILE USING OUTSIDE AIR AS THE SOURCE FOR SCAVENGER AIR

As discussed previously, the EPX 5000 is intended to operate with either return air, or outside air as the source for scavenger airflow. The system is not equipped to switch actively between these sources. However, recognizing that system cooling performance could likely be enhanced at lower temperature ambient conditions were there an option to use outside air for scavenger instead of return air, laboratory tests were conducted to evaluate the impact.

These tests considered four lower temperature outside conditions where commercial buildings would still have considerable internal loads to demand room cooling. Each temperature and humidity condition was tested using outside air for scavenger flow instead of return air, at 5000 *cfm* supply air, and 4000 *cfm* scavenger air. The 70°F_{DB}/55.4°F_{WB} condition was tested for three levels of secondary airflow.

First, similar to the analysis presented earlier, these tests reveal that the scavenger airflow rate has very little effect on supply air temperature, though it does have a substantial impact on fan power draw. Again, supply air temperature is most sensitive to the difference between dry bulb at the primary inlet, and wet bulb at the scavenger inlet. Therefore, when outside air wet bulb is lower than the return wet bulb an advantage can be gained in cooling capacity by using outside air as the source for scavenger airflow.

Figure 24 charts the outside air, supply air, and exhaust air condition for each test. Figure 25 charts the sensible cooling capacity for each of the four tests with scavenger airflow of 4,000 *cfm* and compares these results to the projected performance in indirect evaporative only mode using return air as the source for scavenger airflow.

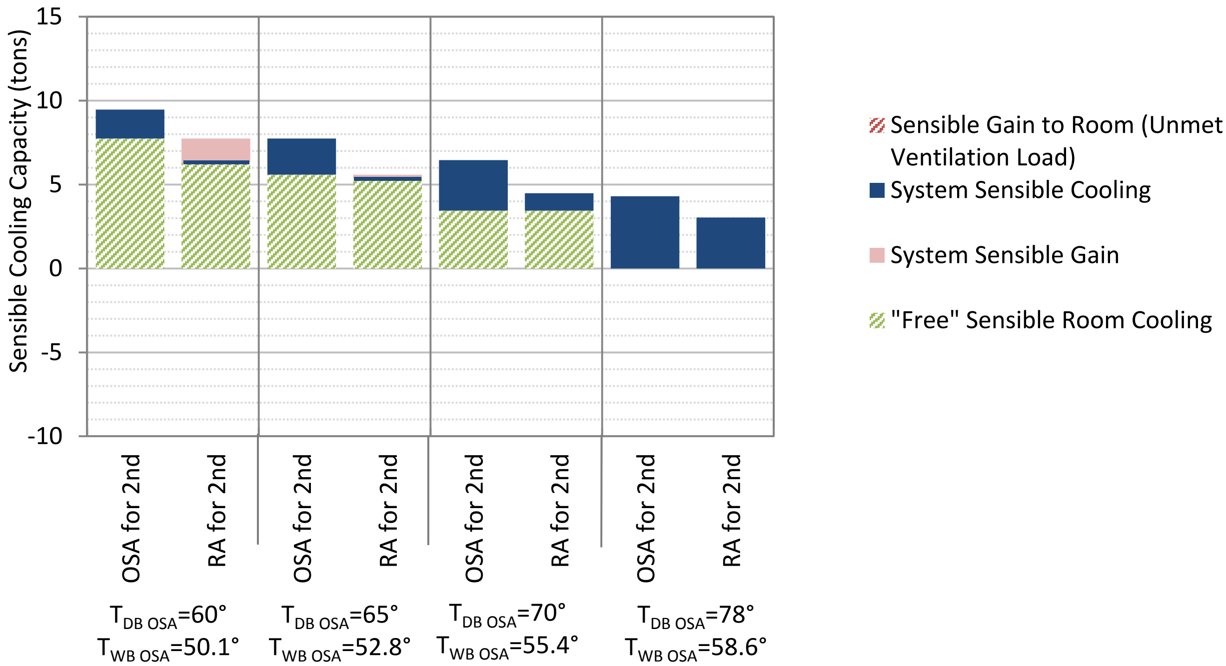


FIGURE 25: SENSIBLE COOLING CAPACITY IN INDIRECT EVAPORATIVE ONLY MODE FOR FOUR LOW-TEMPERATURE OUTSIDE AIR CONDITIONS – THE EFFECT OF SWITCHING TO OUTSIDE AIR FOR SCAVENGER FLOW (ECONOMIZER MODE)

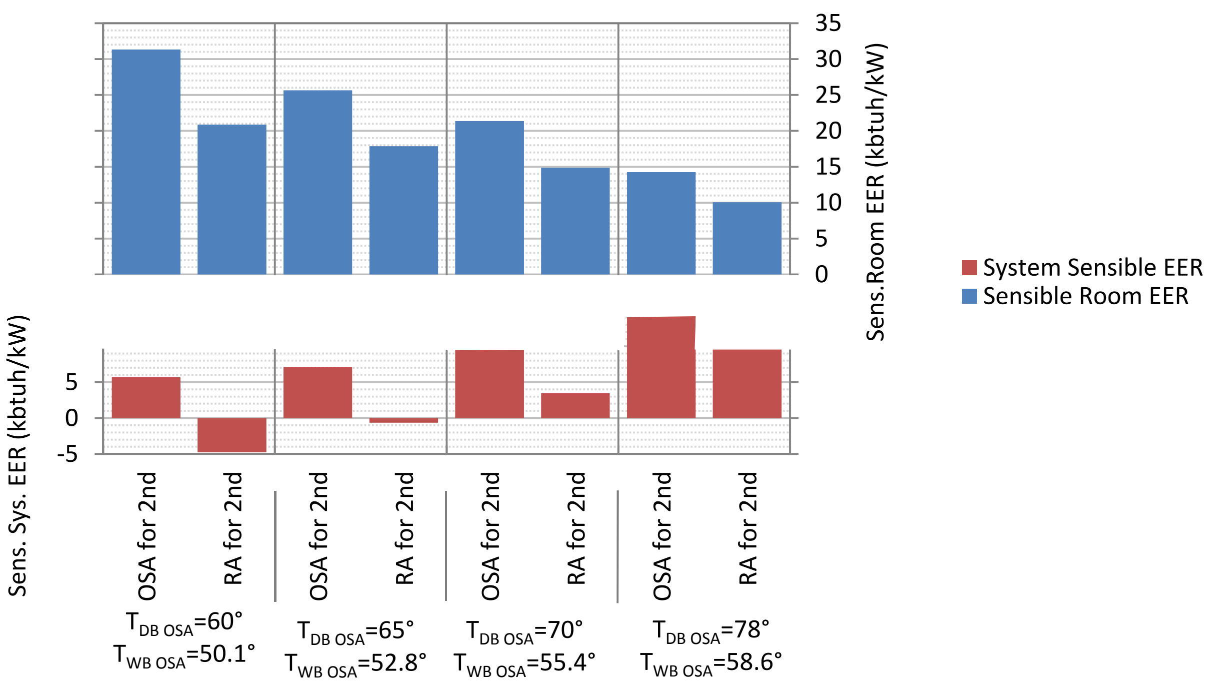


FIGURE 26: SENSIBLE COOLING EFFICIENCY IN INDIRECT EVAPORATIVE ONLY MODE FOR FOUR LOW-TEMPERATURE OUTSIDE AIR CONDITIONS – THE EFFECT OF SWITCHING TO OUTSIDE AIR FOR SCAVENGER FLOW (ECONOMIZER MODE)

These results show a marked increase in cooling capacity at low ambient conditions when using outside air for scavenger instead of return air. At T_{OSA}=60°F_{DB}/50.1°F_{WB} , sensible room cooling increases by 45%. Notably, when the outdoor conditions are cool enough and return air is used for scavenger, the indirect evaporative cooling mode actually heats the primary airflow, instead of cooling it. The cumulative annual impact of this characteristic should be considered in greater detail. For internal load dominated buildings that experience a significant amount of annual cooling hours at low ambient conditions, the potential for switching from return air to outside air could play an important role.

It should be noted that although the switch to outside air for scavenger at low ambient conditions increases cooling capacity, the operating mode is still not as efficient as a conventional economizer which delivers outside air directly, but with less airflow resistance and lower total power consumption. Analysis presented in *Assessment of Demand Savings* considers the total power that might be used for a building with the EPX plus multiple rooftop units operating in economizer mode at low ambient conditions. The findings discussed suggest the cumulative impact of performance at low ambient conditions should be carefully considered.

Performance with Product Air for Scavenger Air

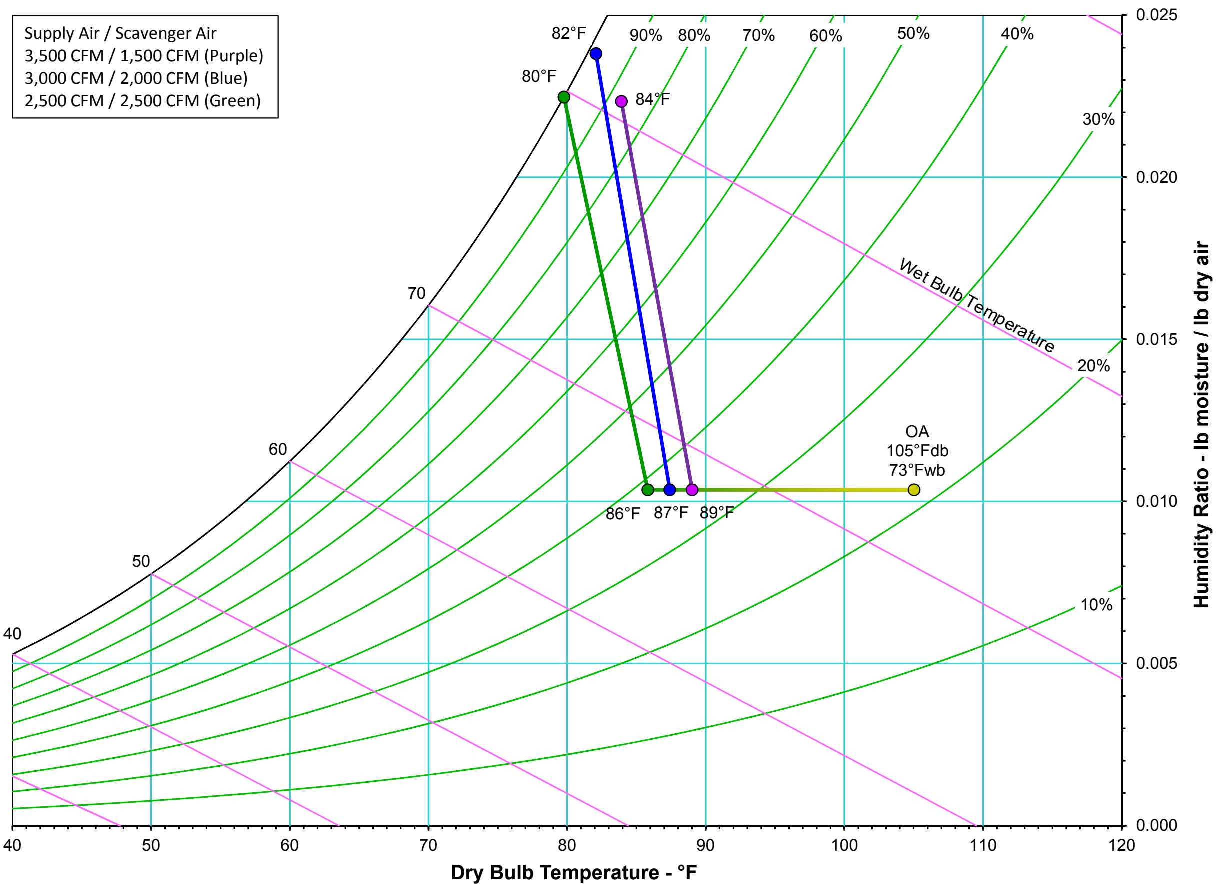


Figure 27: Psychrometric Chart – Performance in Indirect Evaporative Only Mode Under Conditions That Simulate Using a Portion of the Product Air as the Source for Scavenger Air

Some indirect evaporative coolers are designed to divert a portion of the product air stream from the primary outlet into the secondary inlet. Since the product air temperature is driven primarily by the wet bulb temperature of the scavenger inlet, the product air has potential to serve as a better source for scavenger air, as long as there is enough primary air to sacrifice some flow for use as the secondary air stream. Variations on such a strategy have been referred to as a cascading or multi-stage indirect evaporative cooler, and as the Maisotsenko cycle. Studies of such technologies have shown that the strategy can achive evaporative effectiveness well above 100%. However to some extent, this temperature reducing effect may come at the expense of cooling capacity due to reduced supply airflow.

Since the cascading indirect evaporative cooler is currently the subject of other PG&E Emerging Technologies studies, this laboratory evaluation added some tests to consider any gains that could be had for the Muntes EPX heat exchanger if the air handler were configured in this way. Since the physical architecture of the test unit doe not currently allow for connection of the supply air path and return air path, laboratory conditions were adjusted so that return air temperature, humidity, and flow rate would mimic the diversion of some indirect evaporative product air for use as the scavenger air flow. In this arrangement, the primary flow rate was held to 5,000 *cfm*, and the secondary air flow rate was adjusted to three points that would simulate diverting 30%, 40%, and 50% of the product air for use as the scavenger air flow. System capacity would then be calculated based on a supply airflow that was the difference between the primary intake airflow and the scavenger airflow. All three tests were run at the Western Cooling Challenge peak condition T_{OSA}=105°F_{DB}/73°F_{WB}.

Measurements indicate that cooling performance is not improved in any of the three scenarios tested. The results charted in Figure 27 can be compared directly to performance for the intended operating configuration shown in Figure 19. Apparently, for this outside air condition, the lower wet bulb temperature for scavenger air inlet does not overcome the detrimental effect of reduced supply airflow.

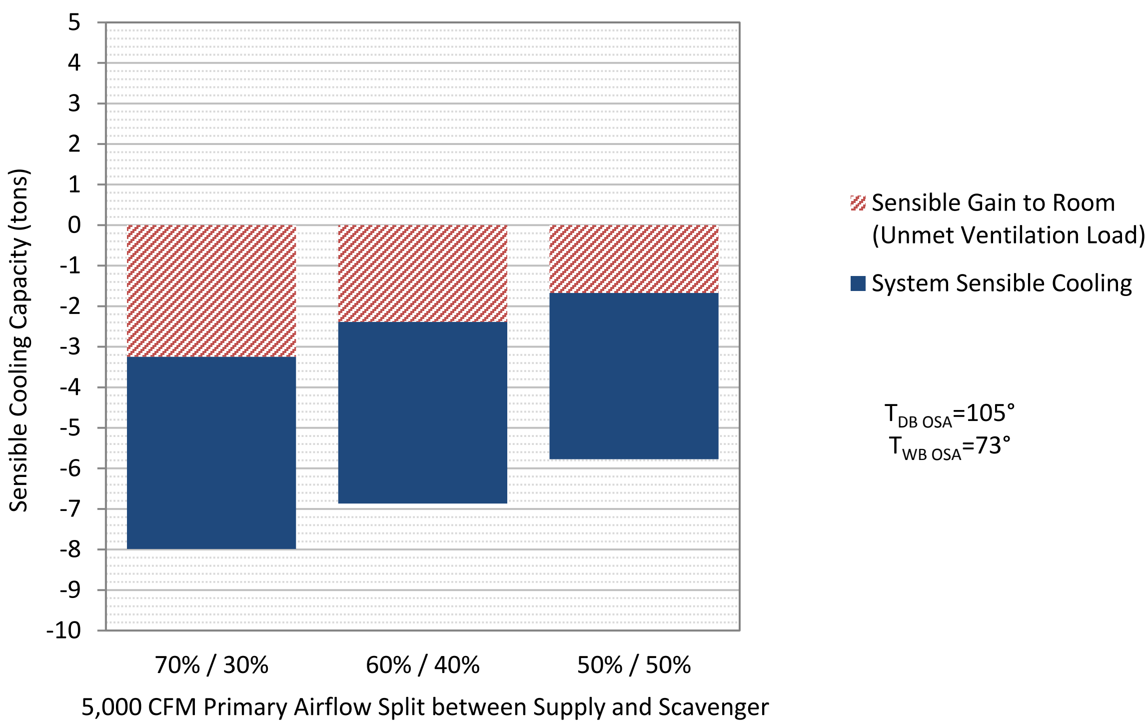


FIGURE 28: SENSIBLE COOLING CAPAITY IN INDIRECT EVAPORATIVE ONLY MODE SIMULATING SUPPLY AIR FOR SCAVENGER

PERFORMANCE AS HEAT RECOVERY VENTILATOR

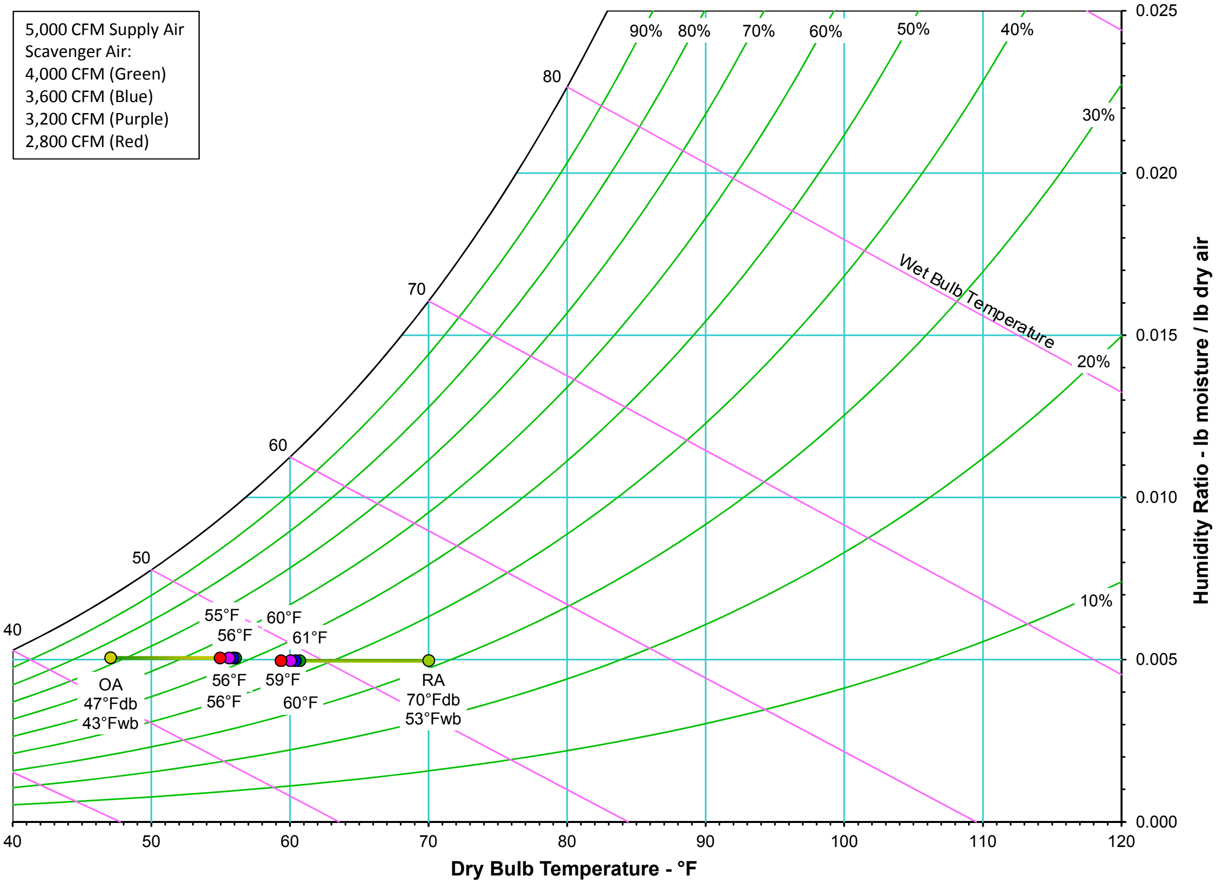


FIGURE 29: PSYCHROMETRIC CHART FOR EPX AS HEAT RECOVERY VENTILATOR AT FOUR SCAVENGER AIRFLOW RATES

One advantage of the EPX 5000 over other indirect evaporative and hybrid air conditioning systems is that the return air for scavenger configuration enables the EPX to operate dry which should provide a significant amount of winter time heat recovery for ventilation. The value of this feature will depend significantly on climate and application. Internal load dominated buildings that require cooling year round would not benefit, and buildings in mild climates may not regain a substantial amount of heat. Futher, the feature is only applicable in buildings that can utilize return air for scavenger flow. If the entire ventilation flow rate is exhausted from somewhere else in the building the EPX 5000 cannot benefit.

Even for buildings that do benefit from heat recovery ventilation, the value of heat recovery in the heating season is somewhat at odds with its detriment to cooling performance during mild conditions, as discussed previously in section: *Performance with Low Temperature Outside Air for Scavenger Air (Economizer Mode)*. While the results presented here charactersize the specific performance of each mode of operation, their cumulative value over the annual operating cycle of a building is not obvious and will require careful annual simulation.

Laboratory tests evaluated dry heat exchanger performance at one low temperature operating condition (T_{OSA}=47°F_{DB},43°F_{WB}) and one return air condition (T_{RA}=70°F_{DB},53°F_{WB}) for several scavenger air flow rates. The psychrometric results are charted in Figure 29, and performance results are described in terms of dry heat exchange effectiveness in Figure 30. For the design airflow conditions, heat recovery effectiveness was measured to be 49%, although the performance characterization does not correct for fan heat in either air stream. Fan temperature rise was not measured, but a reasonable estimate based on a portion of the fan power measurement reduces effectiveness to about 45%. Heat exchange effectiveness increases at lower scavenger air flow rates because of the reduction in the denominator (C_{MIN}) in equation 11. Although the heat exchange effectiveness is higher for lower scavenger air flow rates, less heat is actually recovered under this condition, and the resulting supply air temperature is not as warm. To demonstrate this, a temperature-only effectiveness that removes the heat capacitance rates from Equation 11 has been included in the chart. Dashed lines have also been added to show the estimated effectiveness without the contribution of fan heat for both measures.

Looking at performance through another lens, for the condition tested, the heat recovery feature reduced the heating load from ventilation by 39%. Although this effect does come at the cost of some exhaust fan energy we expect it will translate into a substantial amount of gas energy savings.

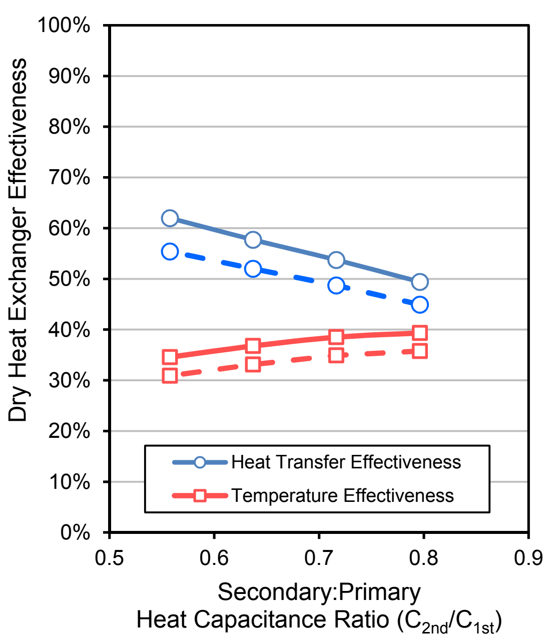


FIGURE 30: EXP DRY HEAT EXCHANGE EFFECTIVENESS

Scavenger Air Flow (CFM)	Heat Exchange Effectiveness	Temperature Effectiveness
4000	49%	39%
3600	54%	39%
3200	58%	37%
2800	62%	35%

TABLE 6. EPX DRY HEAT EXCHANGE EFFECTIVENESS

Characterization of Indirect Evaporative Cooler Performance

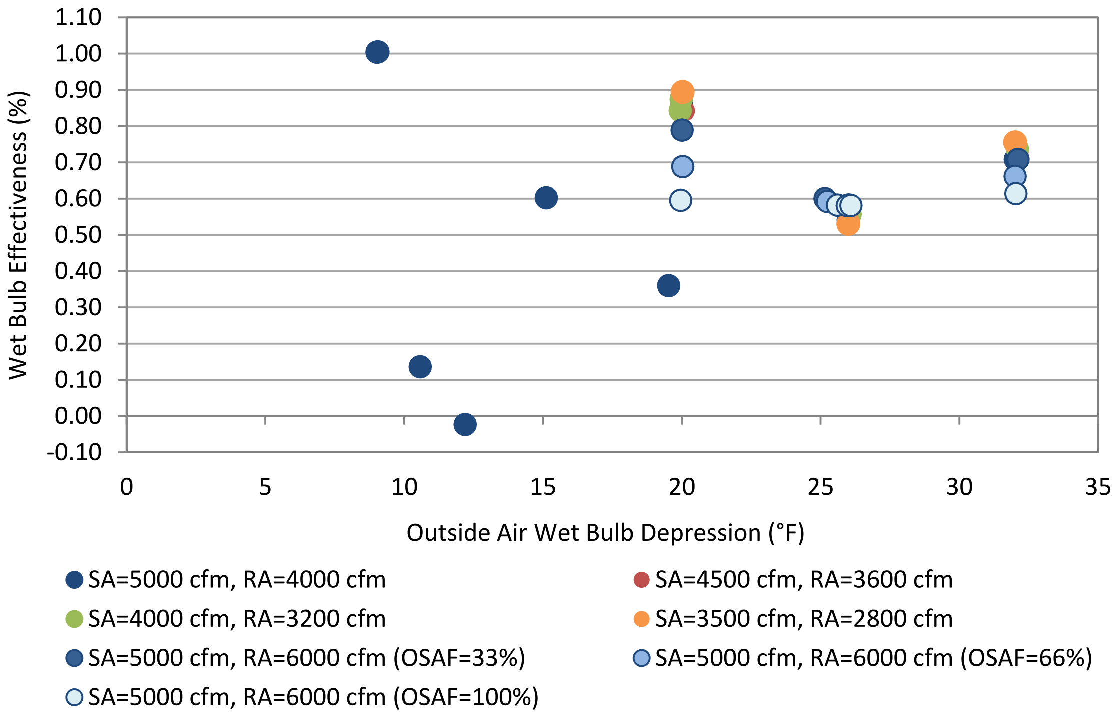


Figure 31: Wet bulb effectiveness as a function of outside air wet bulb depression

Performance for evaporative coolers is typically described in terms of evaporative effectiveness – the degree to which an evaporative process can cool air toward the wet bulb temperature. For a conventional direct evaporative cooler, this metric provides a good structure for defining an empirical model of the system. However the strength of the metric as a modeling tool breaks down when applied to more complicated systems such as Munters EPX 5000 and other indirect evaporative cooling technologies.

Figure 31 charts the wet bulb effectiveness (Equation 3) as a function of wet bulb depression for the outside air for all conditions, airflows, and outside air fractions tested in indirect evaporative only mode. Interestingly, the wet bulb effectiveness ranges from -2% to 100%. However, for modeling purposes, this system demands a metric that reflects the physical mechanisms active within the heat exchanger.

Since the heat transfer rate is driven by temperature difference between the primary flow and secondary flow, and since 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, the indirect evaporative effectiveness (Equation 4) and the indirect wet bulb depression provide a much stronger basis for modeling.

Figure 32 charts the indirect wet bulb effectiveness for all tests as a function of indirect wet bulb depression. Interestingly, the indirect wet bulb effectiveness does not appear to be sensitive to airflow rate. Beyond 10°F indirect wet bulb depression, effectiveness remains between 50 – 60 %. Below approximately 10°F, effectiveness appears to drop. This is likely a result of fan heat, which is not adjusted for in this analysis, and related to the fact that the conditions with low indirect wet bulb depression are associated with tests where outside air is cooler than return air conditions.

Figure 33 plots sensible cooling capacity in indirect evaporative only mode as a function of indirect wet bulb depression. While effectiveness appears to not be sensitive to airflow, cooling capacity is clearly a function of both airflow and temperature – each airflow exhibits a distinctly different linear relationship. Again, cooling capacity drops below zero where the indirect evaporative cooling process is not able to overcome the tendency for heat recovery from warmer return air to cooler outside air.

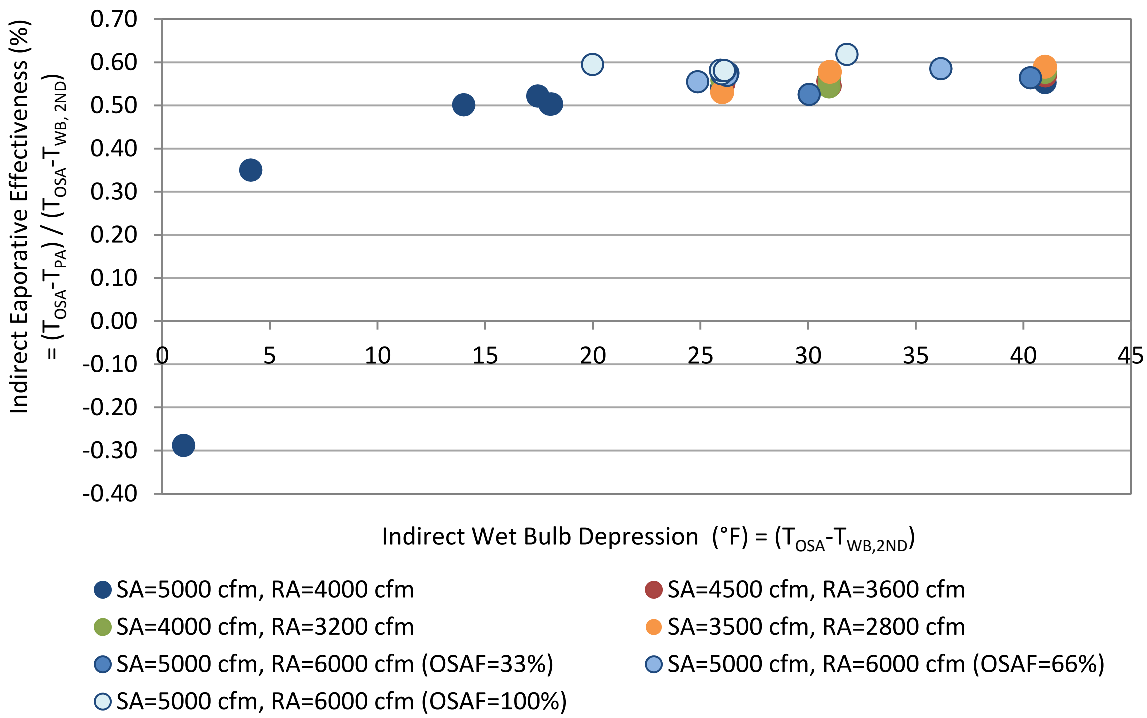


FIGURE 32: INDIRECT EVAPORATIVE EFFECTIVENESS AS A FUNCTION OF INDIRECT WET BULB DEPRESSION

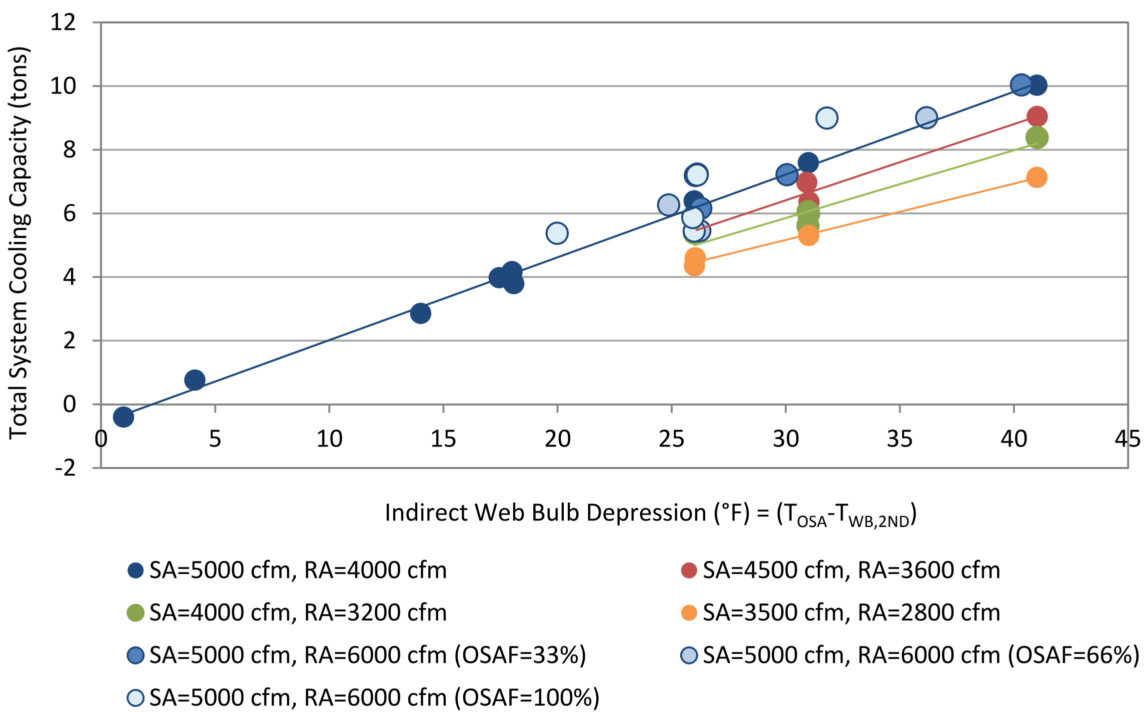


FIGURE 33: COOLING CAPACITY AS A FUNCTION OF INDIRECT WET BULB DEPRESSION, IN INDIRECT EVAPORATIVE ONLY MODE

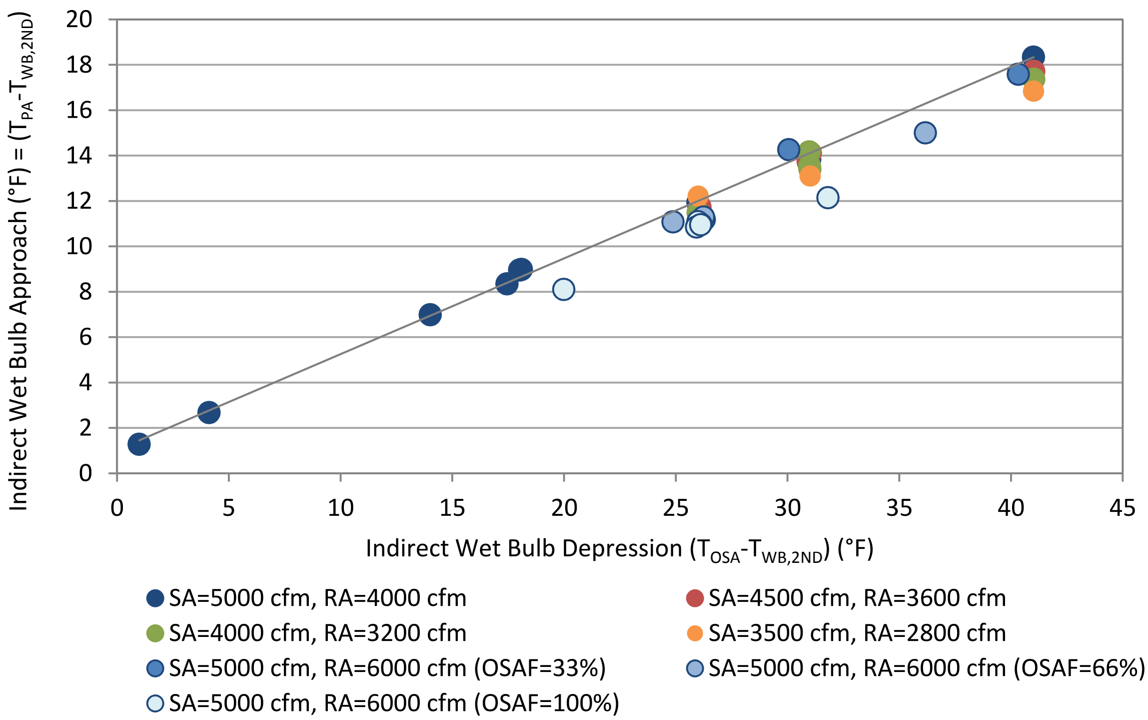


FIGURE 34: INDIRECT WET BULB APPROACH AS A FUNCTION OF INDIRECT WET BULB DEPRESSION

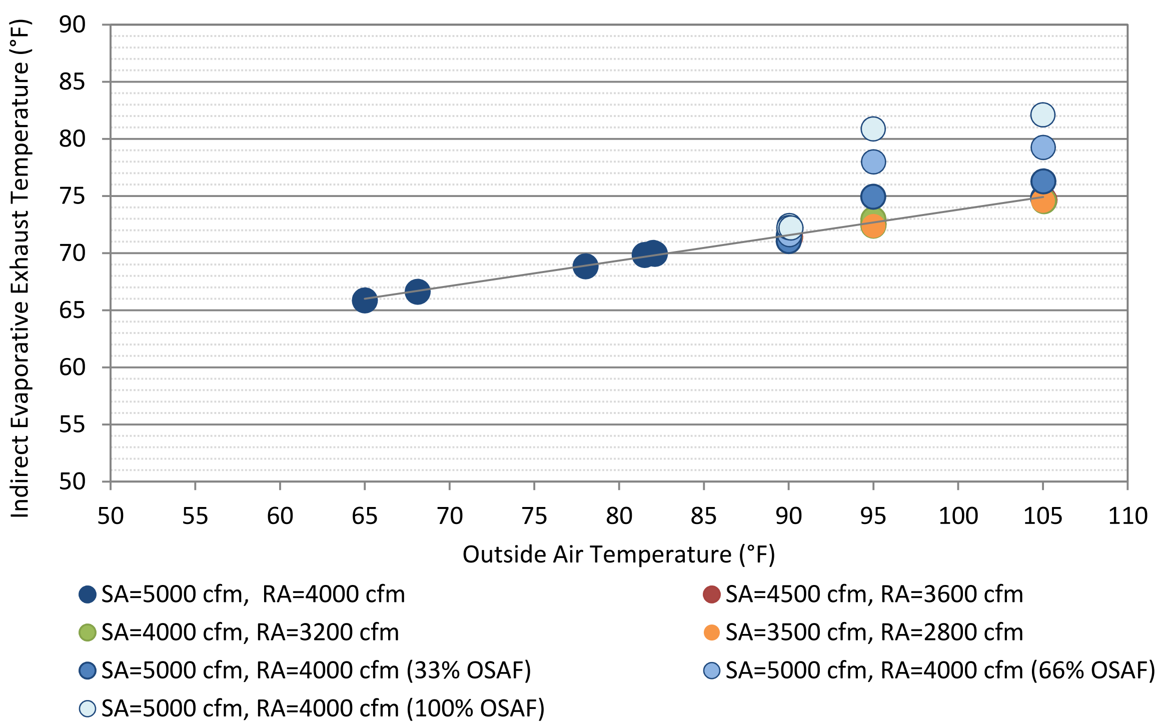


FIGURE 35: RELATIONSHIP BETWEEN INDIRECT EVAPORATIVE EXHAUST TEMPERATURE AND OUTSIDE AIR TEMPERATURE FOR VARIOUS AIRFLOWS AND INLET CONDITIONS

This analysis also indicates that indirect wet bulb approach could be a very good metric for modeling performance in the indirect evaporative cooling mode. In fact, wet bulb approach is the metric typically used to model performance for evaporative fluid coolers, which operate by the exact same principals as the heat exchanger examined here. The only difference is that the heat exchanger is used to cool air instead of water or refrigerant.

Further, characterization of the indirect evaporative exhaust air temperature is important in order to understand the potential to enhance vapor compression efficiency when the condenser is located in the indirect evaporative exhaust stream. Figure 35 charts exhaust air temperature as a function of outside air temperature for the same test groups. Notably, the indirect evaporative exhaust air can be more than 30°F cooler than outside air. The effect is most prominent at high outside air temperatures, and depends significantly on conditions at the secondary inlet. Tests that operate with a mixture of outside air and return air deviate from the linear relationship between outside air temperature and exhaust air temperature, because conditions shift for the secondary inlet. If the same data is considered in terms of the approach to wet bulb for the secondary air stream as a function of the indirect wet bulb depression a very clear linear relationship emerges. For predictive modeling purposes, we recommend using the relationship determined in Figure 36 to estimate exhaust conditions for the Munters EPX 5000.

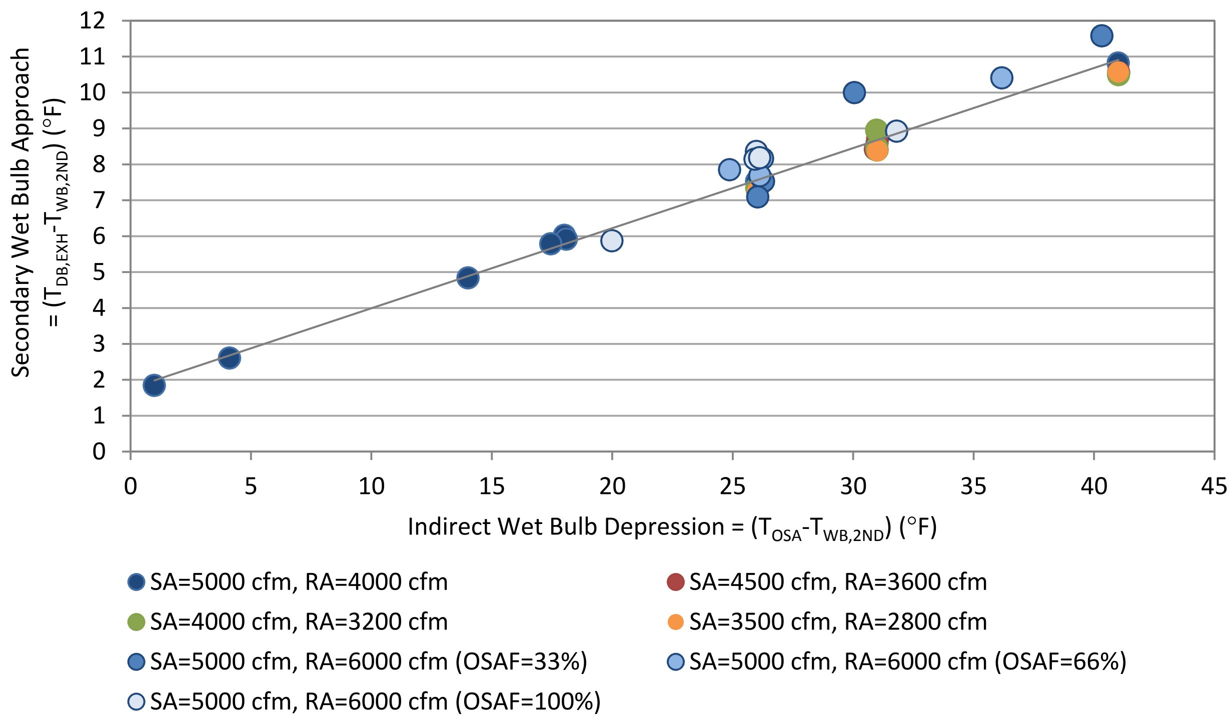


FIGURE 36: RELATIONSHIP BETWEEN WET BULB APPROACH FOR THE SECONDARY AIRFLOW AND INDIRECT WET BULB DEPRESSION FOR VARIOUS AIRFLOWS AND INLET CONDITIONS

TESTING ISSUES

Testing the Munters EPX 5000 presented several operational challenges that led to some installation and testing compromises. The initial challenge was the size of the unit, and accommodating it in the test chamber. Because of the tight quarters, the ductwork connecting to the adjacent room for the return and supply air had more bends and a smaller area than optimal. This precluded following the ASHRAE test standard requirement for the static pressure measurements, which were taken at a point only about one inch away from the test unit. The absolute pressure loss in the return duct was also a concern; and to help compensate for the loss, a booster propeller fan was added to the return duct intake.

Most of the performance testing was done with the code tester booster fan set so as to maintain a constant airflow rate through the primary and scavenger air paths. This is an untypical mode of operation, as the system would be normally operating against a fixed duct resistance, and the unit fans would provide as much airflow as they are allowed. On the supply side, most of the tests resulted in a positive static pressure, except when the DX systems were activated. All of the tests with both DX systems active and half of those with just one required pulling a negative pressure on the supply duct to achieve the required airflow. The reason for this is the production of condensate on the DX coils creating more internal resistance to flow. Thus, these points likely produce performance metrics that are better than what the system will likely achieve.

The "System Enable" switch that the unit came with caused the sump to drain when it was switched off, so the unit started each test day with a new water supply, When the unit is normally operated, there is a conductivity sensor in the water sump that initiates a blowdown when the conductivity gets too high and could lead to solids buildup. The water quality at this location and the intermittent operation of the unit for testing meant that this dump cycle was never initiated during the course of the tests. The sump discharge was captured in a holding tank, which did collect some solid particles that were shed by the heat exchanger, thus demonstrating that the system is self-cleaning.

Assessment of Demand Savings

The laboratory test results presented here were used in a calculated analysis to estimate the demand savings potential for the Munters EPX 5000 at various outside air conditions. The analysis compared the electrical demand for a hypothetical building with conventional rooftop units to a similar building that uses the EPX 5000 for all ventilation instead of requiring each individual rooftop unit to provide fresh outside air.

This analysis, described in detail in the *Technical Approach & Test Methodology* section, indicates 20% reduction in electrical demand from the whole building HVAC at Western Cooling Challenge peak conditions, and 10% reduction at Western Cooling Challenge annual conditions. Moreover, the analysis indicates that while the baseline scenario requires roughly 42 *tons* (net nominal) conventional rooftop unit cooling capacity, the EPX 5000 scenario only requires 29 *tons* (net nominal). This means that an existing building with five conventional rooftop units could downsize to three or four conventional units by shifting the ventilation cooling load over to a new 5000 *cfm* DOAS.

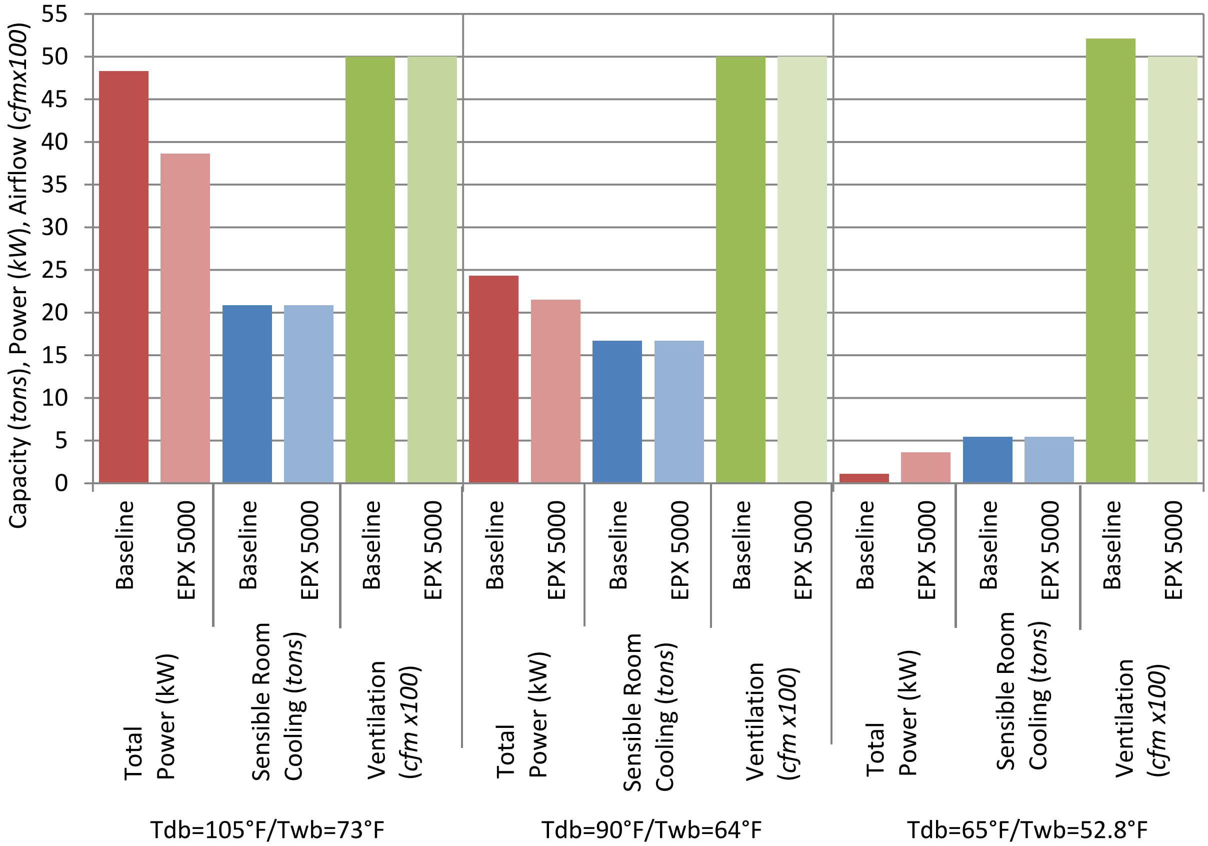


FIGURE 37: WHOLE BUILDING HVAC DEMAND SAVINGS ESTIMATES AT WESTERN COOLING CHALLENGE TEST CONDITIONS

Figure 37 charts the key results from this comparison. The plot shows electrical power, sensible room cooling capacity, and ventilation rate for the two scenarios at three outside air conditions. This illustration shows that both scenarios deliver 5000 *cfm* ventilation and a similar room cooling capacity for each outdoor condition. At peak conditions, the Munters EPX 5000 operates at full capacity and the remaining conventional rooftop units operate so that the combination provides the same amount of sensible room cooling as would be delivered by the baseline scenario. At annual conditions, both scenarios deliver 80% of the sensible room cooling capacity that was delivered at peak.

In the two high temperature scenarios the Munters EPX 5000 scenario requires considerably less power. At low ambient cooling conditions, where both scenarios deliver approximately 25% of the sensible room cooling capacity needed at peak, the EPX 5000 actually requires more power than would be needed for conventional rooftop units operating in economizer mode. For the analysis presented at this condition, the EPX 5000 provides the total sensible room cooling for the building in IEC Only mode, and no other equipment operates. Performance of the baseline scenario represents two rooftop units operating in economizer only mode.

This analysis does not evaluate all possible scenarios for operation. It is possible that at high ambient conditions the Munters EPX 5000 may not always need to operate at full capacity. In fact, at T_{OSA}=90°F_{DB}/64°F_{WB} the supply air condition for IEC&DX2 would be 56°F, which may be cooler than a desirable supply air temperature set point. When operating at part capacity, the demand savings over baseline is reduced significantly. For the low ambient conditions, it is certainly possible that conventional systems could operate in economizer or integrated economizer mode at the same time as the Munters EPX 5000 operates as IEC Only. If this were the case, a portion of the sensible room cooling would be delivered by the more efficient economizer operation. It is also quite possible that the EPX 5000 could operate in a heat recovery ventilation mode while conventional rooftop units operate as economizer cooling. Considering all of these possibilities, it may be beneficial for this type of DOAS air handler to be controlled by an overarching building management system to follows a sequence of operations that optimizes efficient operation for all cooling and heating systems in a building.

DISCUSSION AND CONCLUSIONS

The Western Cooling Challenge has been working to advance climate appropriate cooling technologies, focused mainly on hybrid air conditioners that combine indirect evaporative cooling with vapor compression cooling. These systems are particularly well suited for cooling ventilation air and reducing the peak electrical demand associated with cooling at high temperatures.

The Munters EPX 5000 packages an indirect evaporative cooler together with vapor compression in a Dedicated Outside Air System (DOAS) that is designed to provide the entire ventilation need for a building. Uniquely, the Munters EPX 5000 has the capability of using return air as the source for the secondary airflow in the indirect evaporative cooler which improves thermal performance for the system. Further, since the indirect evaporative cooler is incorporated as as part of a packaged air handler, the cool moist indirect evaporative exhaust is able to double as the condenser air for the vapor compression circuit. The configuration also allows the EPX heat exchanger to operate in a heat recovery ventilation mode during the heating season.

The analysis presented herein indicates that the EPX 5000 offers the potential for considerable electrical demand reduction during peak cooling periods. At the Western Cooling Challenge design conditions T_{OSA}=105°F_{DB}/73°F_{WB}, we estimates 20% savings for the whole building HVAC electrical demand. These results are based on comparison to a hypothetical building with modern standard efficiency rooftop packaged air conditioners. We expect that savings would be even more substantial for existing buildings with less efficient baseline systems. Considering the fact that addition of the single DOAS air handler to a building can produce such large savings at peak, with minimal reconfiguration to other equipment on the building, and without the need for a sophisticated building energy management system to integrate control of all systems, we expect that the Munters EPX 5000 offers a comparatively simple route to peak demand reduction for HVAC in hot dry climates like California, and other western states. We strongly recommend that the technology be adopted by utility energy efficiency programs.

The annual energy savings potential for this equipment will vary significantly by climate, and by application. It is important to note that the equipment is most efficient at cooling outside air in hot-dry ambient conditions. Generally, demand savings will increase as temperature increases and humidity decreases. However, the savings over a conventional baseline appears to diminish relatively quickly as ambient temperature declines. Savings at T_{OSA}=105°F_{DB}/73°F_{WB} is approximately 20%, while savings at T_{OSA}=90°F_{DB}/64°F_{WB} is only 10%.

When ambient conditions are cooler than return air it appears that the Munters EPX 5000 requires more energy for room cooling than would a conventional rooftop unit. The reduced cooling efficiency at this condition may be considered inconsequential if other convention air conditioners are allowed to operate in an economizer mode during those periods or in applications where the cumulative amount of cooling at these conditions is minimal. However, there is a growing tendency to employ a DOAS design in applications where room cooling equipment does not have access to outside air. Radiant systems, chilled beams, ductless split and VRF system can fall in this category. In these circumstances the potential savings available at low ambient conditions could be diminished.

Moreover, when conventional air conditioners are operating in economizer mode there should be ample ventilation for the building, negating the need for DOAS operation at those times. In this circumstance, operation of the Munters would amount to unneeded energy consumption. However, to put these theoretical downsides in perspective, it is well understood that economizers for conventional rooftop units rarely operate correctly. In our experience, between 60%-80% of economizers on light commercial building are mis-programmed or non-functional. In many circumstances existing buildings are not equipped with economizers at all. Therefore, it should prove highly beneficial to have a DOAS unit that provides very high efficiency cooling at low ambient conditions, even if the efficiency of this system is somewhat lower than the theoretical potential of a regular economizer. The analysis presented here does not quantify the annual impacts of these alternative scenarios, and we recommend that further simulation studies and in field evaluations be conducted to consider the cumulative effects.

Simulation and compliance tools for this type of hybrid air conditioner only exist as prototype software. The design of energy efficiency programs for the technology will require custom modeling. The quickest and most defensible route to this end could be a sequence for post processing hourly data from annual simulation of a baseline building in order to estimate the energy use of a building with the Munters EPX 5000 that maintains equal ventilation rate and hourly room cooling capacity. The characterization of performance presented in this report should provide an appropriate basis for mapping energy use of the EPX 5000.

Part of the poorer performance at low ambient conditions is related to the fact that in applications where the equipment is configured to use return air as scavenger, the indirect evaporative process becomes somewhat disadvantaged whenever outside air is cooler than return air. For some low ambient conditions, the indirect evaporative only mode actually manages to heat the primary air stream. In these circumstances it would be preferable to use outside air for the scavenger flow. We recommend that the manufacturer consider an option that would switch between return air and outside air for scavenger. Measurements conducted for this study show that room cooling capacity can be improved by 40% – 45% with no added electrical demand. Equipment that is factory configured to use outside air for scavenger at all times do not experience this performance degradation at low temperatures, but they do operate with somewhat reduced performance at high ambient temperatures.

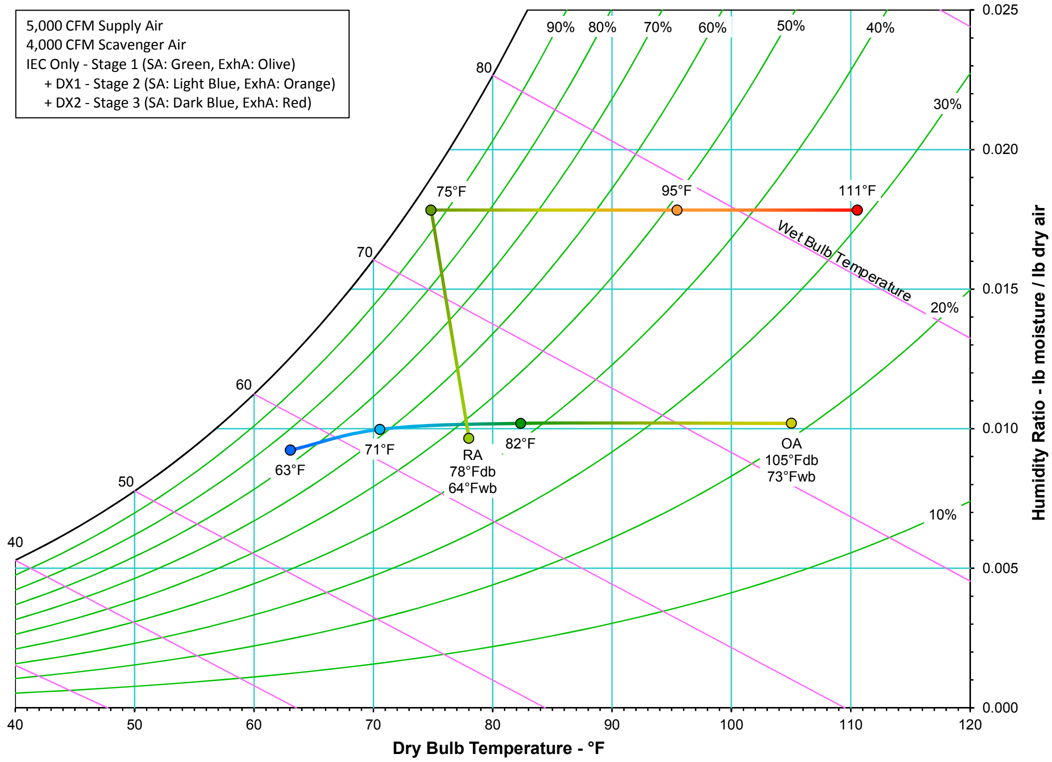
For additional savings, we recommend that all building HVAC systems be smartly controlled to operate in the most efficient combination depending on room loads and ambient conditions. For example, when a building requires cooling at low ambient conditions, this could be served by rooftop units operating in a typical economizer mode, and the EPX 5000 could shut down. In applications where less than 5000 *cfm* ventilation is required, the EPX 5000 could regularly operate at part speed to cover the appropriate amount of ventilation, then adjust to full speed when added room cooling capacity is needed. During high ambient conditions the added room cooling provided by this system in higher capacity operating modes comes with less energy cost than the room cooling by conventional systems. We expect there is room for additional savings if the equipment could also incorporate demand controlled ventilation in applications where it is allowed and appropriate. This strategy could be easily applied with the Munters EPX 5000, and would be much more cost effective than employing demand controlled ventilation for all rooftop units on a large commercial building.

On an anecdotal note, we recommend that efficiency programs avoid installing this system in applications where supply air temperature for the DOAS system will be limited due to the potential for overcooling the zone in which it is supplied. For example, during a scouting trip to identify pilot demonstration sites for the Munters EPX 5000, we observed DX-only DOAS equipment supplying 100+ °F outside air directly to a zone overtop large refrigerated cases. DOAS equipment can integrate well with a mechanical design that takes advantage of case credits, but there is very little benefit for efficient cooling operation in equipment that rarely functions in cooling mode.

Further, we recommend efficiency programs are designed to ensure that building controls are programmed to avoid the potential for simultaneous heating and cooling. We believe there is a distinct possibility that for some periods the Munters EPX 5000 could heat outside air to achieve a desirable supply air set point whilst other systems in the building operate in a cooling mode.

In conclusion, we suggest that this equipment and other similar technologies be incorporated into utility efficiency programs. The technology aligns very well with California Energy Efficiency Strategic Plan goals related to advancing marketplace penetration for climate appropriate air conditioning technologies. Beyond these strategic goals, the equipment promises considerable demand reduction for medium to large commercial buildings, freeing valuable generating capacity for the state and offering significant financial benefits for customers.

APPENDIX A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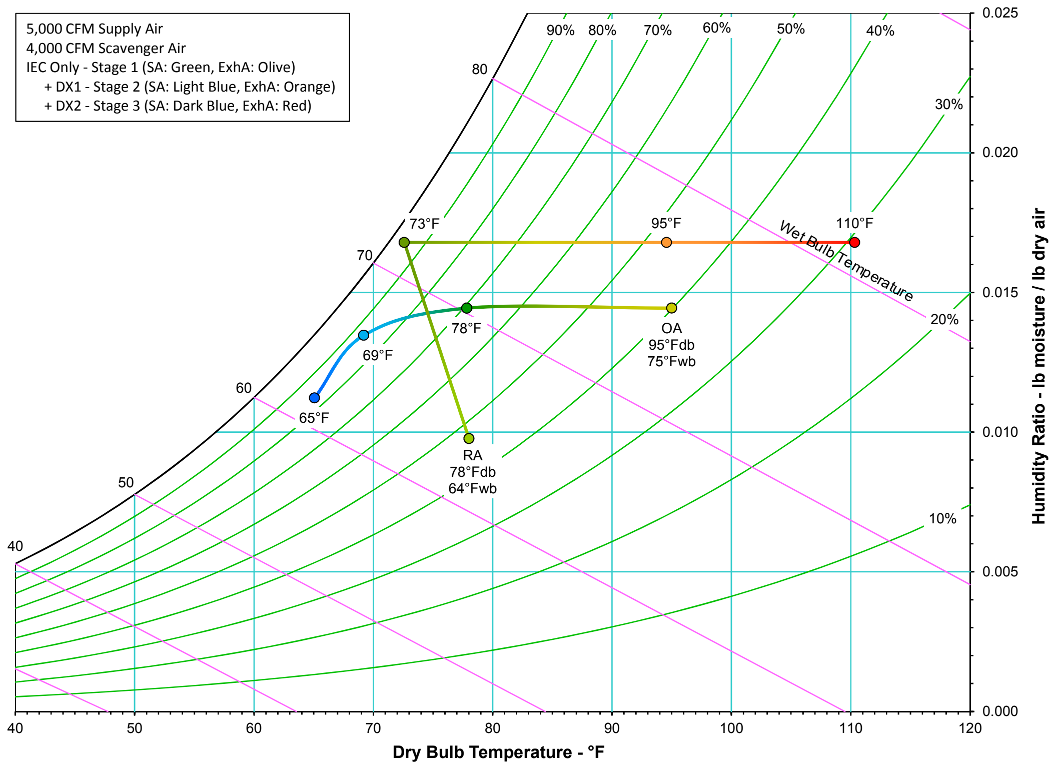
4,000 CFM Scavenger Air
IEC Only - Stage 1 (SA: Green, ExhA: Olive)
+ DX1 - Stage 2 (SA: Light Blue, ExhA: Orange)
+ DX2 - Stage 3 (SA: Dark Blue, ExhA: Red)
90% 80% 70% 60% 50% 40% 30% 20% 10%
75°F
95°F
111°F
Wet Bulb Temperature
71°F
63°F
82°F
RA
78°Fdb
64°Fwb
OA
105°Fdb
73°Fwb
Dry Bulb Temperature - °F
Humidity Ratio - lb moisture / lb dry air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4,000 CFM Scavenger Air
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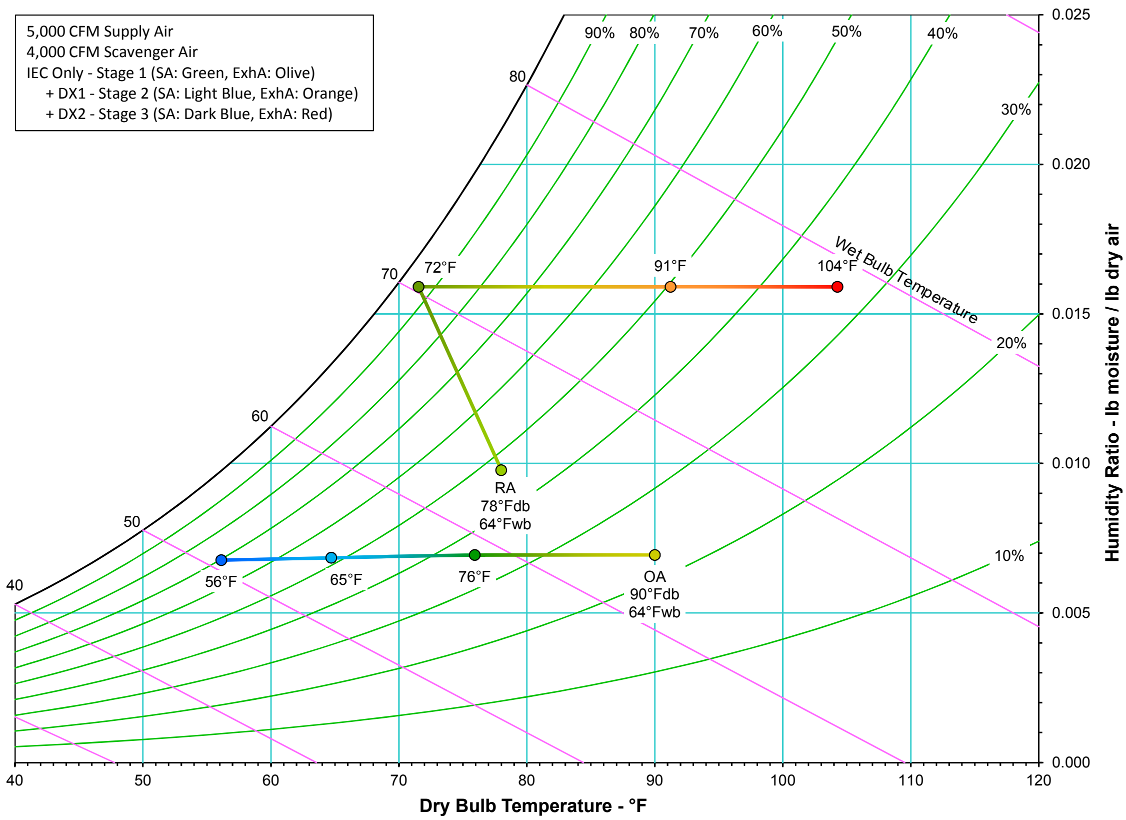
+ DX1 - Stage 2 (SA: Light Blue, ExhA: Orange)
+ DX2 - Stage 3 (SA: Dark Blue, ExhA: Red)
90% 80% 70% 60% 50% 40% 30% 20% 10%
73°F
95°F
110°F
Wet Bulb Temperature
78°F
69°F
65°F
OA
95°Fdb
75°Fwb
RA
78°Fdb
64°Fwb
Dry Bulb Temperature - °F
Humidity Ratio - lb moisture / lb dry air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4,000 CFM Scavenger Air
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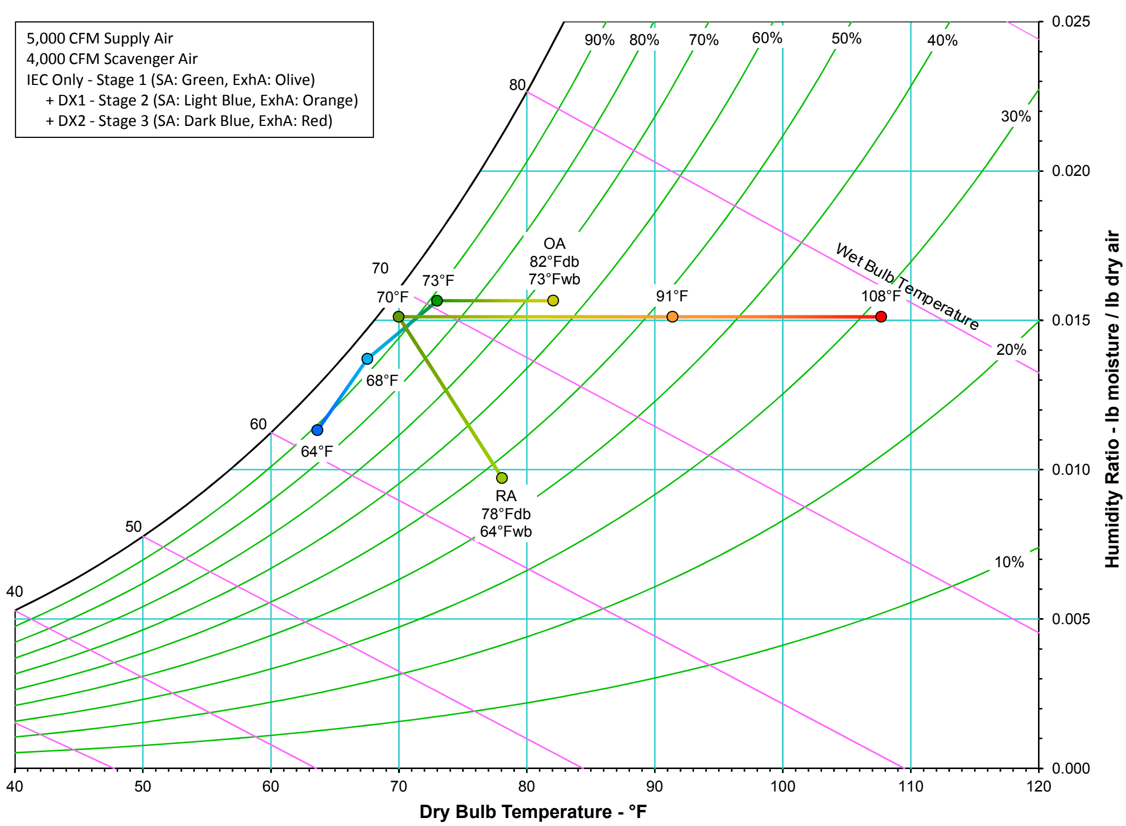
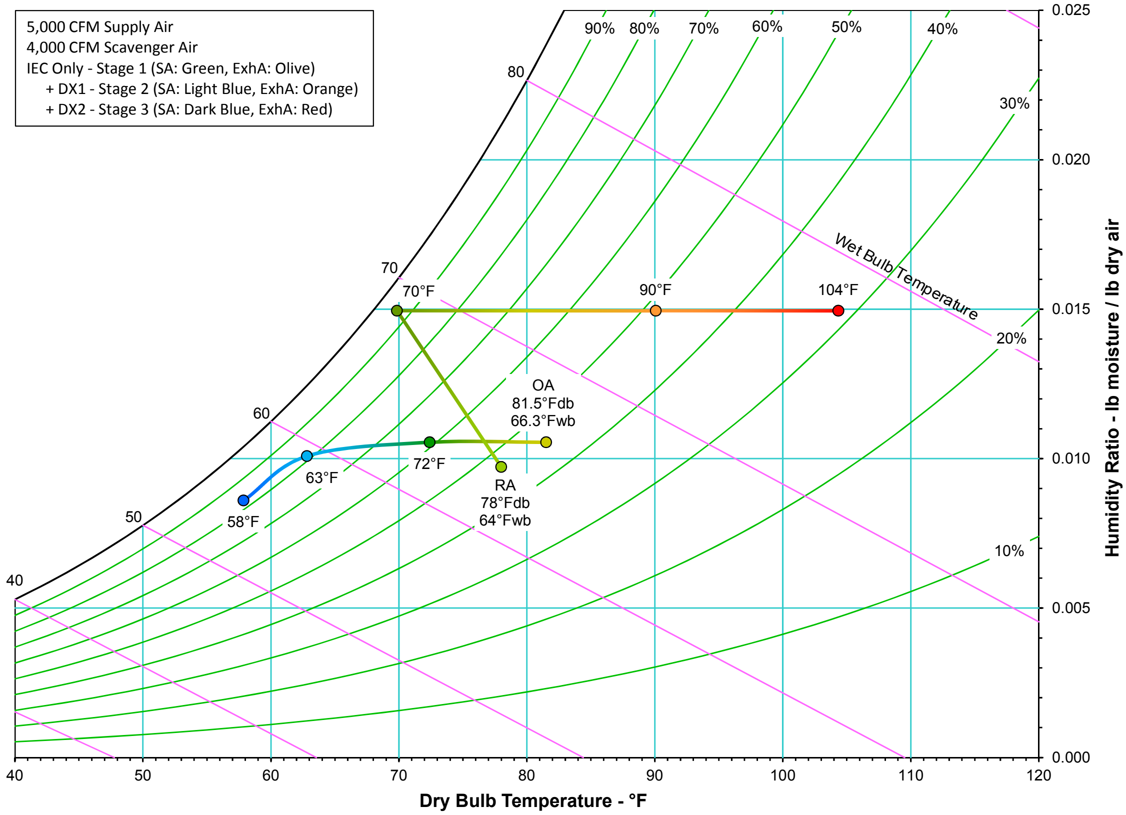
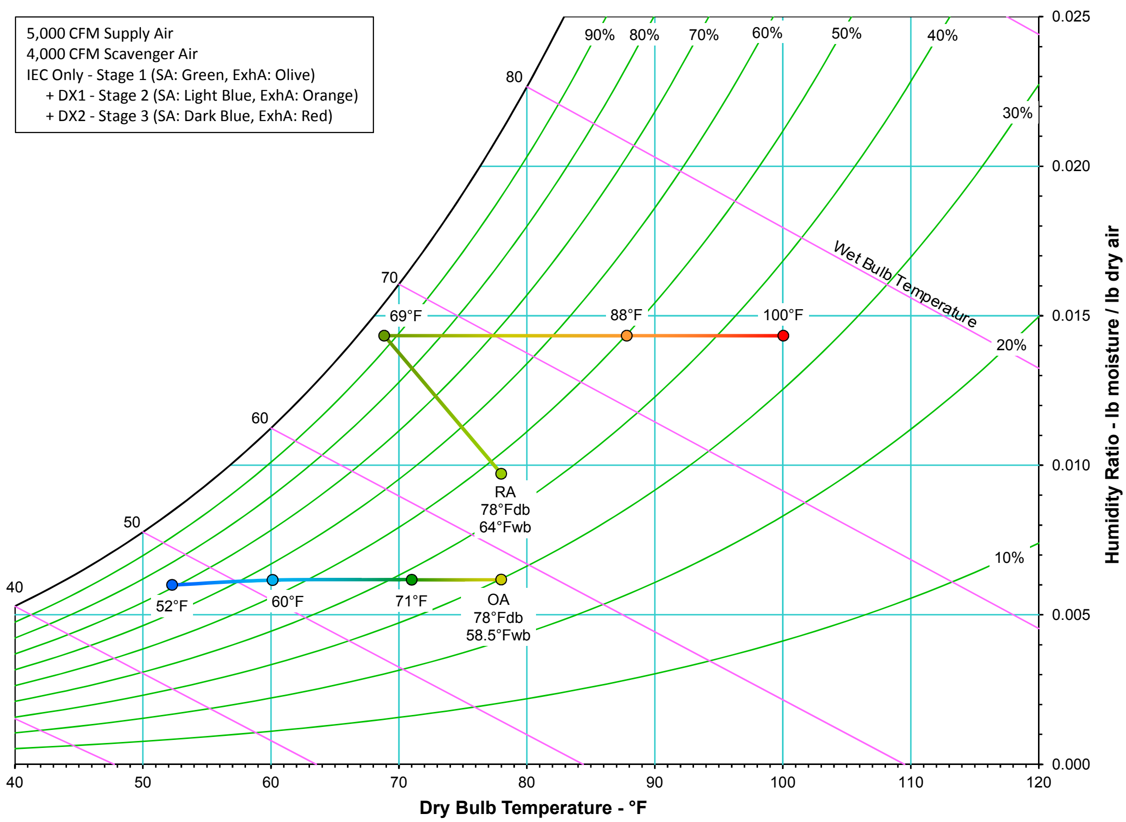
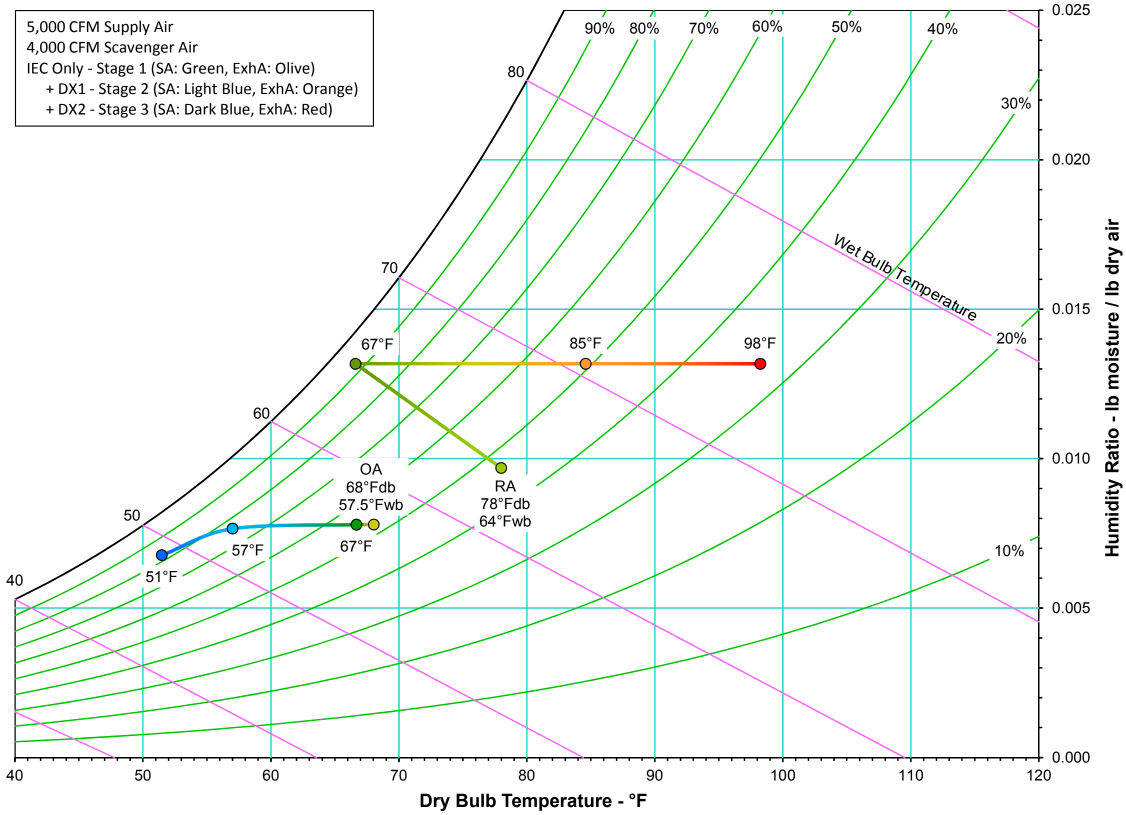
+ DX1 - Stage 2 (SA: Light Blue, ExhA: Orange)
+ DX2 - Stage 3 (SA: Dark Blue, ExhA: Red)
72°F
91°F
104°F
RA
78°Fdb
64°Fwb
56°F
65°F
76°F
OA
90°Fdb
64°Fwb
Wet Bulb Temperature
90%
80%
70%
60%
50%
40%
30%
20%
10%
Dry Bulb Temperature - °F
Humidity Ratio - lb moisture / lb dry air

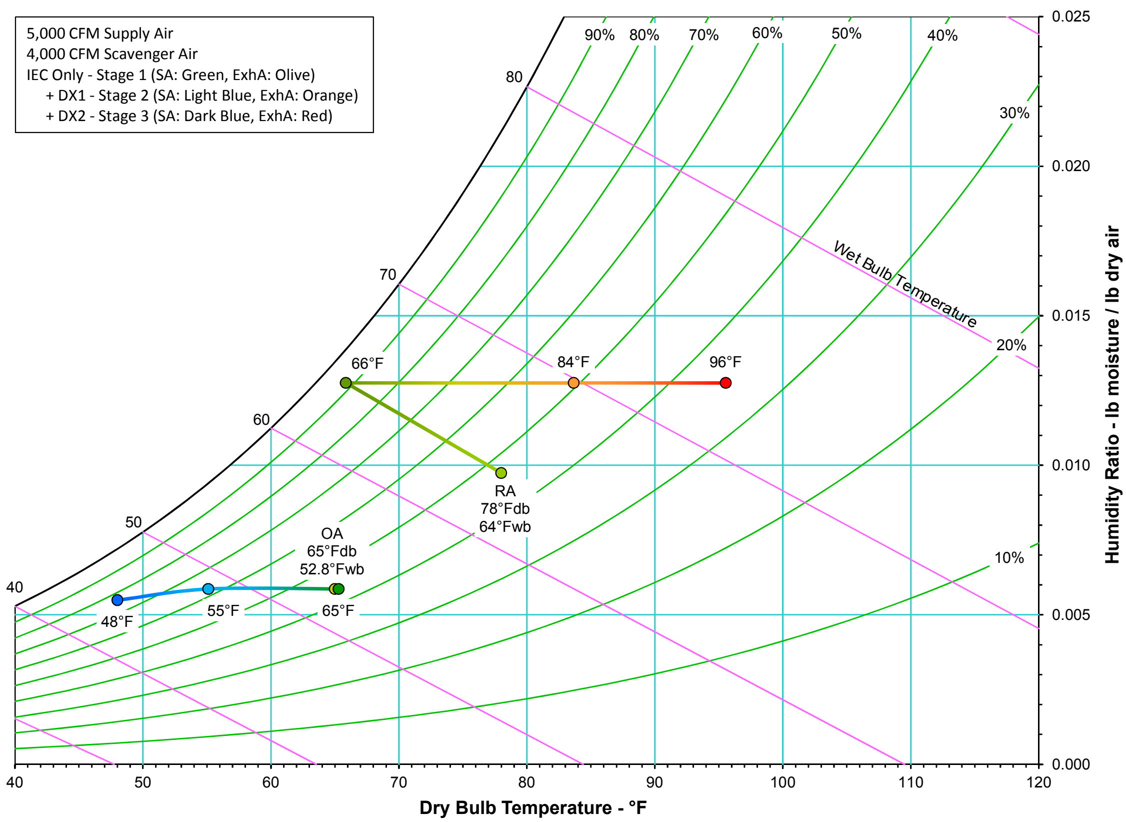

5,000 CFM Supply Air
4,000 CFM Scavenger Air
IEC Only - Stage 1 (SA: Green, ExhA: Olive)
+ DX1 - Stage 2 (SA: Light Blue, ExhA: Orange)
+ DX2 - Stage 3 (SA: Dark Blue, ExhA: Red)
OA
82°Fdb
73°Fwb
73°F
70°F
91°F
108°F
68°F
64°F
RA
78°Fdb
64°Fwb
Wet Bulb Temperature
90%
80%
70%
60%
50%
40%
30%
20%
10%
Dry Bulb Temperature - °F
Humidity Ratio - lb moisture / lb dry air


5,000 CFM Supply Air
4,000 CFM Scavenger Air
IEC Only - Stage 1 (SA: Green, ExhA: Olive)
+ DX1 - Stage 2 (SA: Light Blue, ExhA: Orange)
+ DX2 - Stage 3 (SA: Dark Blue, ExhA: Red)
70°F
90°F
104°F
OA
81.5°Fdb
66.3°Fwb
72°F
63°F
58°F
RA
78°Fdb
64°Fwb
Wet Bulb Temperature
Dry Bulb Temperature - °F
Humidity Ratio - lb moisture / lb dry air


5,000 CFM Supply Air
4,000 CFM Scavenger Air
IEC Only - Stage 1 (SA: Green, ExhA: Olive)
+ DX1 - Stage 2 (SA: Light Blue, ExhA: Orange)
+ DX2 - Stage 3 (SA: Dark Blue, ExhA: Red)
69°F
88°F
100°F
RA
78°Fdb
64°Fwb
52°F
60°F
71°F
OA
78°Fdb
58.5°Fwb
Wet Bulb Temperature
Dry Bulb Temperature - °F
Humidity Ratio - lb moisture / lb dry air



5,000 CFM Supply Air
4,000 CFM Scavenger Air
IEC Only - Stage 1 (SA: Green, ExhA: Olive)
+ DX1 - Stage 2 (SA: Light Blue, ExhA: Orange)
+ DX2 - Stage 3 (SA: Dark Blue, ExhA: Red)
90%
80%
70%
60%
50%
40%
30%
20%
10%
Wet Bulb Temperature
67°F
85°F
98°F
OA
68°Fdb
57.5°Fwb
RA
78°Fdb
64°Fwb
57°F
51°F
67°F
Dry Bulb Temperature - °F
Humidity Ratio - lb moisture / lb dry air



5,000 CFM Supply Air
4,000 CFM Scavenger Air
IEC Only - Stage 1 (SA: Green, ExhA: Olive)
+ DX1 - Stage 2 (SA: Light Blue, ExhA: Orange)
+ DX2 - Stage 3 (SA: Dark Blue, ExhA: Red)
90%
80%
70%
60%
50%
40%
30%
20%
10%
Wet Bulb Temperature
66°F
84°F
96°F
RA
78°Fdb
64°Fwb
OA
65°Fdb
52.8°Fwb
48°F
55°F
65°F
Dry Bulb Temperature - °F
Humidity Ratio - lb moisture / lb dry air

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