PROJECT REPORT

WESTERN COOLING CHALLENGE LABORATORY RESULTS: TRANE VOYAGER DC HYBRID ROOFTOP UNIT

Western Cooling Efficiency Center-UC Davis

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WESTERN COOLING EFFICIENCY CENTER

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ABOUT THE WESTERN COOLING EFFICIENCY CENTER

The Western Cooling Efficiency Center was established along side the UC Davis Energy Efficiency Center in 2007 through a grant from the California Clean Energy Fund and in partnership with California Energy Commission Public Interest Energy Research Program. The Center partners with industry stakeholders to advance cooling-technology innovation by applying technologies and programs that reduce energy, water consumption and peak electricity demand associated with cooling in the Western United States.

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1. EXECUTIVE SUMMARY

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The Western Cooling Challenge is a program focused on advancing commercial development and market introduction of rooftop packaged air conditioners optimized for the hot-dry climates of the Western United States. The Challenge requires 40% energy savings while operating at peak design conditions, compared to equipment that meets current federal minimum equipment efficiency requirements. Trane's Voyager DC shows a savings of 43%. The explicit goal of the effort reported here was to laboratory test the Trane Voyager DC according to Western Cooling Challenge test protocol. The protocol evaluates equipment performance in hot-dry climate conditions, operating under realistic airflow resistances and with an outside air ventilation rate that would be encountered in typical commercial retail applications. A secondary goal was to laboratory test the equipment across a broad range of operating conditions in order to characterize performance in various scenarios, and to evaluate function of each major system sub-component and operating mode.

The Trane Voyager DC is a hybrid rooftop air conditioner that integrates the DualCool™ with an otherwise conventional vapor compression system. The DualCool™ is a unique indirect evaporative cooling strategy that uses an evaporative condenser–air pre-cooler to reduce the refrigerant condensing temperature of a vapor compression system, then cycles the water chilled by evaporation through a heat exchanger to cool the hot outside air required for building ventilation.

While the Western Cooling Challenge performance requirement is well beyond the reach of conventional vapor compression systems, the laboratory results documented herein demonstrate that 40% peak energy savings is achievable with savvy incorporation of various efficiency enhancing technologies. The Trane Voyager DC equipment met the Western Cooling Challenge criteria on the mark.

The Western Cooling Efficiency Center (WCEC) directed the laboratory tests, and contracted with Intertek to utilize the largest independent environmental test chamber in the United States, located at their HVAC & Electrical test facility in Plano, TX. Trane, and Integrated Comfort provided in kind support for planning, coordination, setup and commissioning of the laboratory test.
The Trane Voyager DC was tested in each operating mode across a range of psychrometric conditions. Unfortunately, the environmental chamber was not able to reach all of the humidity conditions that were prescribed by the original design of experiments. Intertek's facility is not equipped to handle the moisture load generated by Trane's equipment.

Dehumidification capacity for the laboratory facility allowed for testing at the Western Cooling Challenge "Peak" condition (Tdb=105°F, Twb=73°F), but did not allow for evaluation at the drier "Annual" condition (Tdb=90°F, Twb=64°F). Thus determination of Western Cooling Challenge certification in this case is based on the performance measured at the "Peak" operating conditions. Despite the humidity limitations of the facility utilized, the range of laboratory tests covered enough operating conditions to develop general characterizations of system component performance, and provided great insight into opportunities for additional improvements.

Final results from the range of tests indicate the technology provides substantial energy savings for cooling, especially during peak demand periods when the electrical grid is most strained. We recommend that public interest programs and efforts designed to apply this technology consider its value compared to the alternative cost of new peak electrical generation capacity.

Scrutiny of the laboratory observations also indicates there is still room for moderate performance improvements for the equipment. Some of the possible measures for added efficiency are already manufacturer options for the equipment and will be evaluated through various pilot field demonstrations currently in progress. These measures include variable speed supply fan operation for savings at part capacity operation and during continuous ventilation periods, and microchannel heat exchangers for improved condenser heat transfer effectiveness.

Others potential enhancements, discussed herein will require further research and innovation in system design and control. These recommended measures include optimization for economizer control, improved condenser air cooling, and the potential for increased cooling capacity for the ventilation air cooling coil.

2. INTRODUCTION

The Western Cooling Challenge is an ongoing program that encourages HVAC manufactures to develop and commercialize climate-appropriate rooftop packaged air conditioning equipment that will reduce electrical demand and energy use for cooling in Western climates by at least 40% compared to DOE 2010 standards. The Challenge was developed at the behest of commercial building owners, investor-owned utilities, and HVAC industry stakeholders who recognize the economic value of efficient cooling technologies, and are motivated by state and corporate goals for energy and sustainability. For example, the California Public Utility Commission's Energy Efficiency Strategic Plan gives specific priority to the application of climate-appropriate cooling technologies, such as those advanced by the Challenge. The Western Cooling Efficiency Center (WCEC) developed the Western Cooling Challenge test protocol and minimum performance criteria in order to provide a standard basis for evaluating advanced rooftop unit cooling technologies that are designed especially for application in hot-dry climates such as California. The Challenge does not require a particular type of system design; rather, it sets ambitious yet achievable thresholds for energy and water-use efficiency. Each of the technologies currently in consideration for the Challenge employ a hybrid cooling strategy that couples various indirect -evaporative cooling technologies with conventional vapor compression equipment.

In partnership with Southern California Edison, and other sponsors, WCEC collaborates with manufacturers to advance the development of these technologies, and conducts laboratory and field evaluation of commercially available equipment. In 2012, Trane Inc. submitted the Voyager DC hybrid rooftop unit as an entry to the Challenge, and WCEC arranged to conduct rigorous laboratory testing of the system at the Intertek psychrometric test facility in Plano, TX. Intertek operates a large test facility that is regularly used to evaluate performance of unitary heating and cooling equipment. Laboratory tests were conducted in June – July 2012.

This report reviews the design and operation of the Trane Voyager DC, describes the laboratory test facility and experimental approach then documents performance results across a range of operating conditions. The laboratory facility could not maintain all of the intended psychrometric test conditions, and was not able to measure certain variables that would have been helpful for performance evaluation. Notwithstanding, the performance results recorded qualify the Trane Voyager DC for Western Cooling Challenge certification.

Beyond testing for Western Cooling Challenge certification, the observations are evaluated with great scrutiny to characterize behavior of each major sub–component in the system. This analysis highlights a number of enlightening facts about the equipment function, and reveals some opportunities for further improvement. For example, results indicate that energy savings at peak could be improved further by increasing the wet–bulb effectiveness for the condenser–air pre–cooler, and by increasing heat transfer effectiveness of the condenser coil to allow for a lower condensing temperature.

3. OVERVIEW OF TRANE VOYAGER DC OPERATION

Trane's Voyager DC couples a conventional rooftop packaged air conditioner with the DualCool™, an innovative indirect evaporative cooling strategy that increases cooling capacity and unloads compressor power by reducing the air temperature at the inlet of both the condenser and evaporator coils. The system utilizes a direct evaporative cooler to pre-cool condenser–air, then circulates the water that has been chilled by evaporation through a heat exchanger that cools incoming ventilation air.

The commercially available system incorporates staged compressor operation, variable speed control for the supply blower and condenser fans, thermostatic expansion valves, micro-channel condenser heat exchangers, integrated comparative economizer controls, and demand control ventilation. For the purposes of determining Western Cooling Challenge certification, not all of these options were included or evaluated through the laboratory tests presented here. Most importantly, the configuration tested used a constant speed supply air blower.

Figure 1 illustrates the conceptual air flow, water flow, and refrigerant paths for the Trane Voyager DC. An overview the equipment configuration corresponding to Figure 1 is described below.

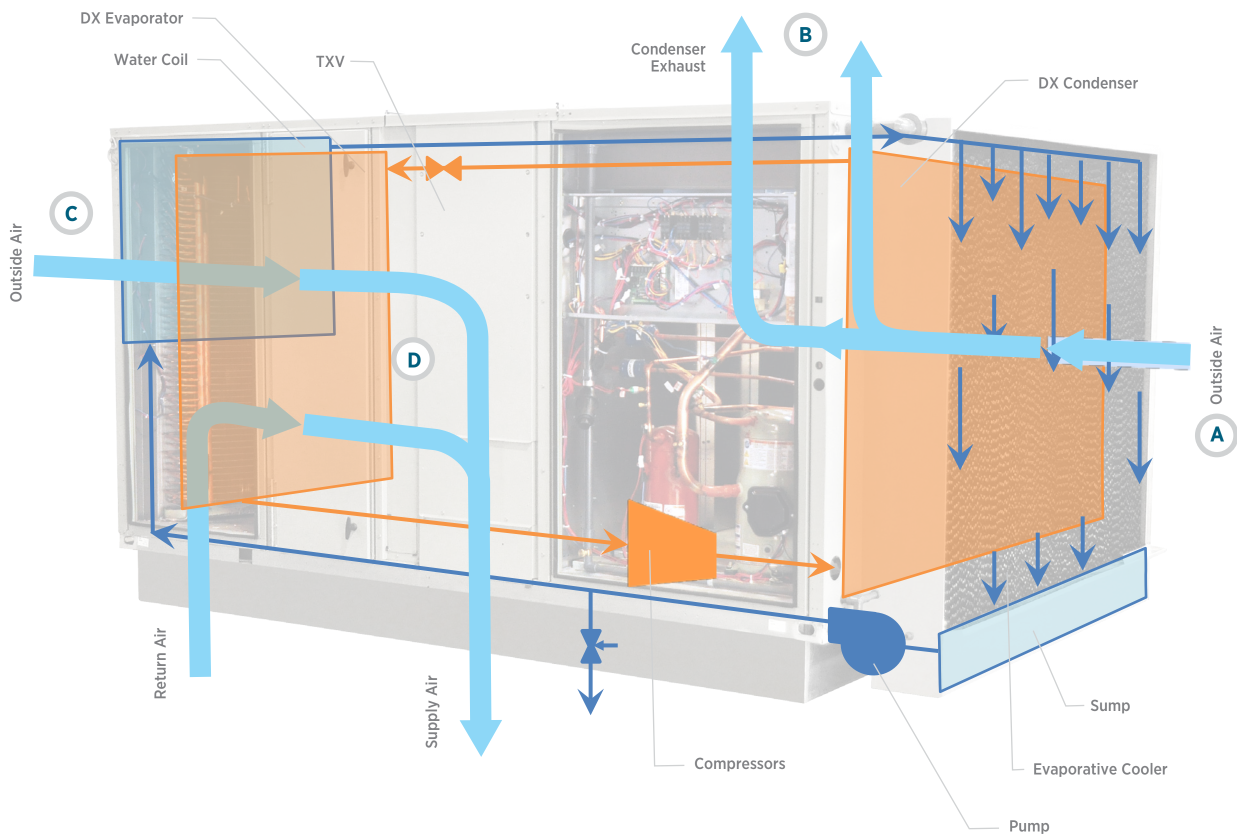
A. Hot dry outside air is drawn through a fluted cellulose media evaporative cooler located at the inlet of the vapor compression condenser coil. Water is delivered through a manifold at the top of the media, and flows through the fluted channels in contact with airflow. The air and water are both cooled by evaporation, and excess water drains by gravity to a stainless steel sump.

B. Cool moist air is drawn across the condenser coils for two separate refrigerant circuits, and afterward exhausted from the equipment through two condenser fans. When operating in a vapor compression mode, heat is rejected to this airstream, but the fans can also operate independent of compressors to cool water. The condenser fans draw from a single plenum, so both fans must operate together to draw airflow appropriately. The condenser fans are variable speed, and controlled to draw a different airflow rate for each mode of operation.

C. Water that drains from the evaporative cooler is collected in a sump, then circulated through a water coil located at the ventilation air inlet to cool fresh air for the building before it crosses the vapor compression evaporator. When the pump is activated, water flows at a constant speed and warms through the heat exchanger before returning to the evaporative cooler.

D. The ventilation air flow path is physically separated from the return air path until after the evaporator coil. When the system operates in an economizer mode with 100% outside air, flow is restricted to only pass across the upper portion of the evaporator coil. Similarly, when the system operates without ventilation, flow is restricted to pass across only the lower portion of the evaporator coil. The two separate refrigerant circuits are interlaced at the evaporator coil, so that both circuits are presented to each airflow path, regardless of the operating mode.

FIGURE 1: CONCEPTUAL SCHEMATIC FOR TRANE VOYAGER DC



There are four general modes of operation for the equipment:

1. **Ventilation:** Similar to typical rooftop packaged systems, the supply air blower operates to deliver a mixture of fresh outside air and return air to the space. No cooling is active in this mode of operation.

2. **Enhanced Economizer:** When the outside air temperature is appropriate, the condenser fans operate at part speed to chill water in the evaporative cooler. Cool water is circulated through the water coil, the supply air blower is active, and dampers actuate to provide 100% outside air.

3. **Indirect & Stage 1 DX:** Condenser fans operate at 60% speed to cool water and condenser–air. The first stage compressor operates, though the compressor power is reduced because the vapor compression circuit operates with a lower condensing temperature, and reduced load on the evaporator. When outside air temperature is below a factory selected changeover set point, dampers will actuate to provide 100% outside air; otherwise the systems will operate to deliver the minimum ventilation requirement.

4. **Indirect & Stage 2 DX:** Condenser fans operate at 90% speed to cool water and condenser inlet air. Both compressors operate, though the power draw is reduced because of a lower condensing temperature, and reduced load on the evaporator. When outside air temperature is below a factory selected changeover set point, dampers will actuate to provide 100% outside air; otherwise the systems will operate to deliver the minimum ventilation requirement.

It should be noted that when installed in an application where the unit can be allowed to operate as recirculation-only at times, such as during unoccupied periods, the indirect evaporative circuit will not operate in the last two cooling modes described. Instead, the system will shift to 0% outside air, though the water pump will still cycle to provide direct evaporative condenser–air pre-cooling. Table 1 details the complete sequence of operation for the Trane system as it was commissioned for Western Cooling Challenge laboratory testing.

TABLE 1: SEQUENCE OF OPERATION FOR TRANE VOYAGER DC

Mode	Independent Conditions			Component Operations				
	TOSA	Scheduled Occupancy	TROOM-TSP	Indoor Blower	OSA Fraction	Condenser Fans Speed	Compressor(s)	Water Pump
Off	NA	NO	≤0	OFF	0%	OFF	OFF	OFF
Ventilation Only	NA	YES	≤0	ON	MIN	OFF	OFF	OFF
Indirect & Stage 1 DX	>TSP	Yes	>0	ON	MIN	60%	1	ON
Indirect & Stage 2 DX	>TSP	Yes	>2	ON	MIN	90%	1&2	ON
Unoccupied Stage 1	>TSP	NO	>0	ON	0%	60%	1	OFF
Unoccupied Stage 2	>TSP	NO	>2	ON	0%	90%	1&2	OFF
Enhanced Economizer	<TSP	NA	>0	ON	100%	30%	OFF	ON
Indirect & Stage 1 DX	<TSP	NA	>1	ON	100%	60%	1	ON
Indirect & Stage 2 DX	<TSP	NA	>2	ON	100%	90%	1&2	ON

4. PERFORMANCE REQUIREMENTS

The Western Cooling Challenge performance rating centers on steady-state sensible energy efficiency at full capacity operation, under two outdoor psychrometric conditions, with 120 cfm/nominal ton ventilation rate, and external resistance that would produce 0.7 " WC external static pressure at 350 cfm/nominal ton. The test conditions were designed roughly around typical design specifications for a large retail facility in a hot dry climate1. The minimum performance required at these conditions achieves 40% energy savings compared to standard efficiency systems operating under similar conditions.

Table 2 details the Western Cooling Challenge test conditions and performance requirements for the two psychrometric conditions at which system efficiency is evaluated. Note that a number test conditions for the Challenge performance tests are defined as a function of nominal capacity. Therefore, the laboratory procedure focuses on determination of a nominal capacity before executing the rating tests. The procedure for determining nominal capacity is described later.

It should be noted that the two-point rating test for the Western Cooling Challenge does not fully describe performance for a system across all operating conditions. For this fact, the laboratory tests expanded from the few Western Cooling Challenge rating tests to map equipment performance in a variety of scenarios. The results presented in this report center mostly on performance at the Western Cooling Challenge rating conditions, though a complete summary of test results is included in Appendix 1.

TABLE 2: WESTERN COOLING CHALLENGE TEST CONDITIONS AND PERFORMANCE REQUIREMENTS [1,2]

	WCC Peak Conditions	WCC Annual Conditions
Outside Air Condition Tdb°F/Twb°F	105/73	90/64
Return Air Condition Tdb°F/Twb°F	78/64	78/64
Minimum Outdoor Ventilation cfm/nominal-ton	120	120
External Resistance In WC at 350 cfm/nominal-ton	0.7	0.7
Min Filtration	MERV 7	MERV 7
Operating Mode	Full Capacity	Full Capacity or Part Capacity
Min Sensible Credited Capacity (% sensible credited cooling at peak conditions)	NA	80%
Min Sensible Credited EER (kbtu/kWh)	14	17
Max Supply Air Humidity (lb/lb)	.0092	.0092
Max Water Use (gal/ ton-h)	NA	4

[1]Performance criteria are described in more detail in the "Western Cooling Challenge Program Requirements"

[2]Development of test protocol and performance requirements is described fully in an ASHRAE publication Advancing Development of Hybrid Rooftop Packaged Air Conditioners: Test Protocol and Performance Criteria for the Western Cooling Challenge.

5. FACILITY AND LABORATORY SETUP

Laboratory tests for the Trane Voyager DC utilized the 35-ton psychrometric test chamber at the Intertek HVAC/R test facility in Plano, TX. Intertek is the world's largest independent HVAC/R testing company, and the Plano, TX facility is the largest independent performance and safety testing laboratory in the Southwest United States. This facility is regularly used for a wide range of performance rating and safety tests for a variety of HVAC/R equipment.

The facility maintains desired temperature and humidity conditions in separate "indoor" and "outdoor" environmental chambers, and manages airflow resistance for the equipment examined. Figure 3 illustrates the airflow scheme for the 35-ton psychrometric test chamber as it was configured for these tests. The purpose and operation of each component in this setup is described here.

The Trane Voyager DC was positioned in the "outdoor" environmental chamber, and supply and return airflows were ducted through an insulated wall to connect with the "indoor" environmental chamber. The return air duct was positioned to draw air from a single location in the "indoor" environmental chamber, and the chamber was controlled to maintain desired return air conditions measured at the inlet of the return air ductwork.

Supply air from the Trane Voyager DC was ducted to a nozzle airflow measurement station located in the "indoor" environmental chamber where static pressure drop across a calibrated nozzle configuration was correlated to airflow according to ANSI/AMCA 210-2007 and ANSI/ASHRAE 51-2007. The Trane supply air blower was operated normally during tests, and an variable speed fan downstream of the nozzle airflow measurement station was adjusted to maintain the desired external static pressure (ESP) for the Trane Voyager DC. The supply airflow was ultimately delivered to the "indoor" environmental chamber, which was managed to maintain a desired return air condition.

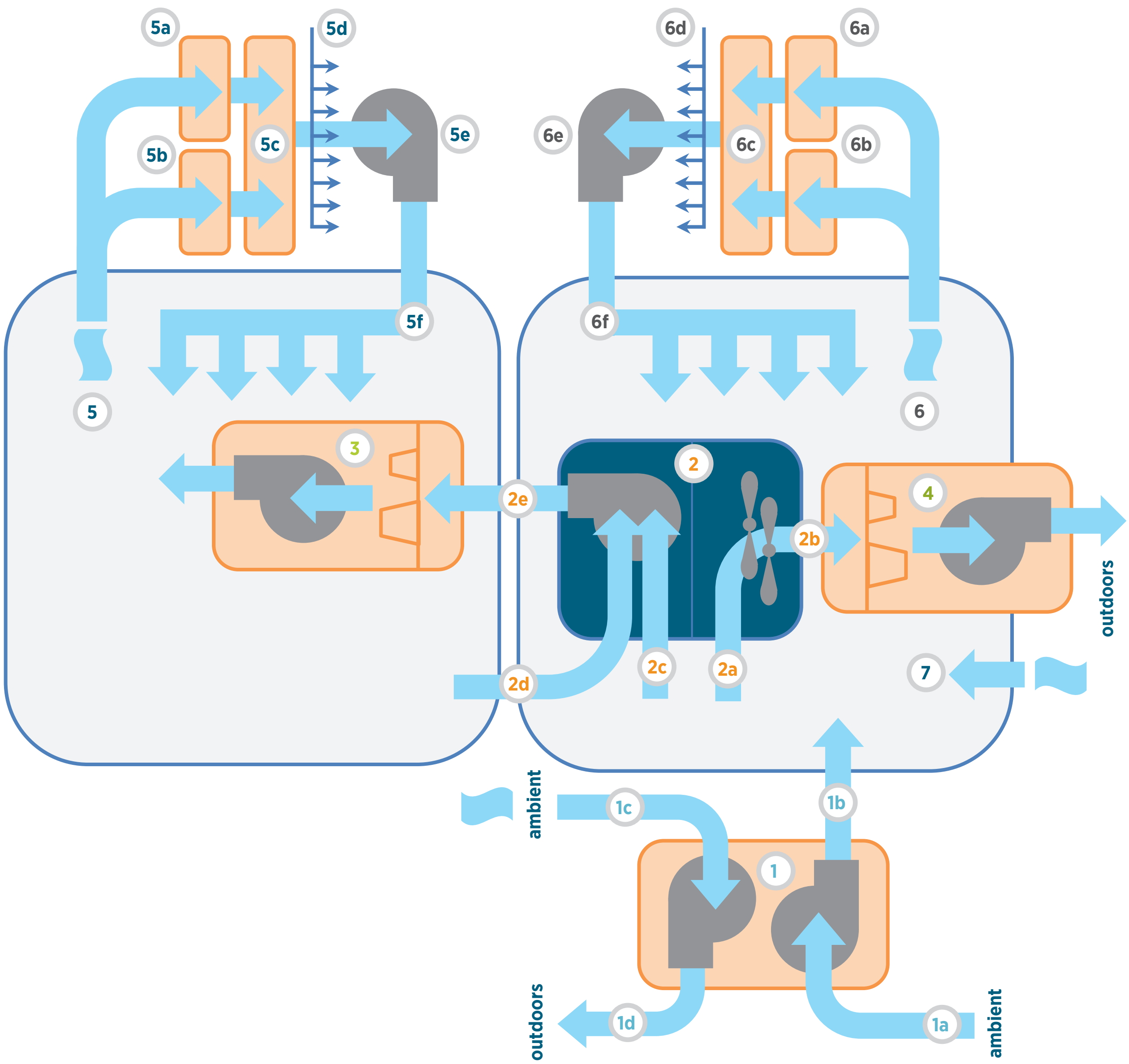
Ventilation air was drawn freely from the "outdoor" environmental chamber into the Trane's outside air hood; and the outside air damper and return air damper were adjusted to achieve the desired outside air fraction (OSAF). Due to the relative size of the outside air and return air openings, any OSAF greater than

Trane Voyager DC testing at Interek Labs in Plano, Texas

approximately 30% required that the outside air damper remain fully open, while the return air damper was adjusted to restrict return airflow. The OSAF for the Challenge rating tests was chosen in parallel with selection of the ESP, as part of the process for determination of a nominal capacity value. The procedure for determining nominal capacity is described later.

While most of the tests conducted maintained an OSAF in accordance with the ventilation rate used for Challenge certification, several tests operated the equipment with 100% outside air in order to characterize performance for Trane's enhanced economizer operating mode. For these later tests, the return air damper was fully closed and the outside air damper remained fully open. Since there was no return airflow, ESP measured between the return and supply air plenum could not be used as a target for adjusting the laboratory's variable speed fan that maintains the appropriate resistance to supply airflow. Instead, the fan speed and nozzle airflow measurement station configuration were fixed to the same settings used for Challenge certification in order to provide the same external resistance to flow imposed on other tests. Various real world scenarios may differ from the conditions imposed for these tests, particularly if the total external resistance to flow differs between regular operation and operation in economizer mode.

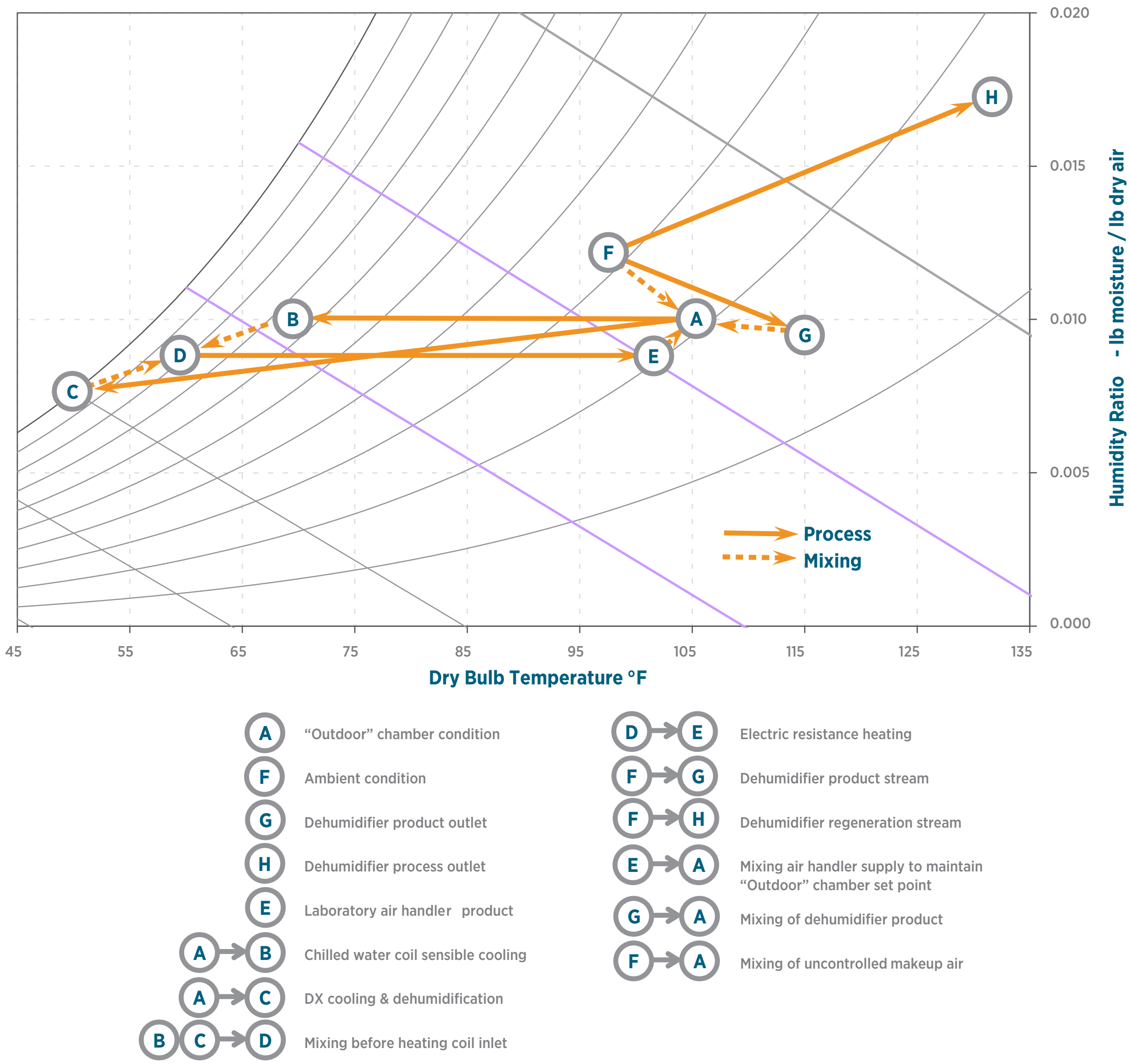
FIGURE 3: SCHEMATIC CONFIGURATION OF EXPERIMENTAL SETUP AT INTERTEK LABORATORY FACILITY



1	Dehumidifier	2	Trane Voyager DC	3	Nozzle airflow meter & VFD fan	5	"Indoor" environmental chamber	6	"Outdoor" environmental chamber
1a	Product inlet	2a	Condenser inlet	4	Nozzle airflow meter & VFD fan	5a	DX cooling & dehumidification	6a	DX cooling & dehumidification
1b	Product outlet	2b	Condenser outlet			5b	Chilled water cooling	6b	Chilled water cooling
1c	Regen. inlet	2c	Vent. air inlet			5c	Resistance heat	6c	Resistance heat
1d	Regen. Outlet	2d	Return air inlet			5d	Steam humidification	6d	Steam humidification
		2e	Supply air			5e	Blower	6e	Blower
						5f	Product air	6f	Product air

Generally, condenser–airflow for laboratory testing of a rooftop unit is drawn freely from the "outdoor" environmental chamber, and exhausted without restriction back to the same chamber. In this way, the condenser–airstream experiences zero external resistance as it would in field application. Heat is rejected to the outdoor room, but laboratory conditioning systems operate to maintain "outdoor" environmental chamber psychrometric conditions. This "outdoor" condition is measured as a space average across all outside air inlets to the unit, in accordance with ANSI/ASHRAE Standard 37-2009. However, the Intertek "35-ton" psychrometric test facility did not have adequate capacity to remove the humidity generated by the Trane Voyager DC evaporative condenser–air pre cooler, so the setup was configured to capture and exhaust the condenser outlet air stream (see Figure 3, (2b)).

Figure 4: Psychrometric Process for Control of "Outdoor" Chamber Set Point Condition



A plenum was constructed at the condenser outlet, from which condenser outlet air was ducted to a second nozzle airflow measurement station and variable speed fan (see Figure 3, (3)). The variable speed fan and nozzle configuration were adjusted to maintain zero static pressure between the "outdoor" chamber and condenser outlet plenum. Ultimately, the hot moist condenser outlet air was exhausted to outdoors. This unique condenser–airflow arrangement circumvented a massive addition of moisture to the "outdoor" chamber that would have saturated the laboratory facility's dehumidification capacity. The setup also provided a calibrated measurement of condenser–airflow, which is typically a very difficult measurement to capture and is often ignored in laboratory evaluations.

The majority of makeup airflow for the "outdoor" environmental chamber was provided through a 10,000 *cfm* (nominal), electric resistance regenerated, silica-gel wheel dehumidifier (see Figure 3,(1)). Ambient air was drawn from an unconditioned warehouse space for both the product and regeneration air streams. Dehumidified air was delivered to the "outdoor" chamber, and moist regeneration exhaust was ducted to outdoors. For full capacity tests, the condenser–airflow was measured at nearly 13,000 *cfm*, in which circumstance the balance of makeup air for the "outdoor" environmental chamber was drawn freely from the ambient unconditioned warehouse space through an open door (see Figure 3, (7)).

Temperature and humidity conditions in each environmental chamber were controlled with laboratory integrated air handler systems that recirculated air within each chamber (see Figure 3, (5)&(6)). Both air handlers included DX cooling coils, chilled water cooling coils, electric resistance heat, steam humidification, and a variable speed blower. The operation of components in each system was controlled with a PID control algorithm that targeted a user-selected chamber set point condition. In the case that the chamber required some cooling and dehumidification, the chilled water coil would provide a significant amount of sensible cooling while the DX evaporator was set to operate with a low airflow and very low temperature in order to provide as much latent cooling as possible. After cooling and dehumidification, airflow was heated to produce an appropriate supply air condition to maintain chamber set point conditions after mixing with the bulk air volume (Figure 4).

Data Confidence

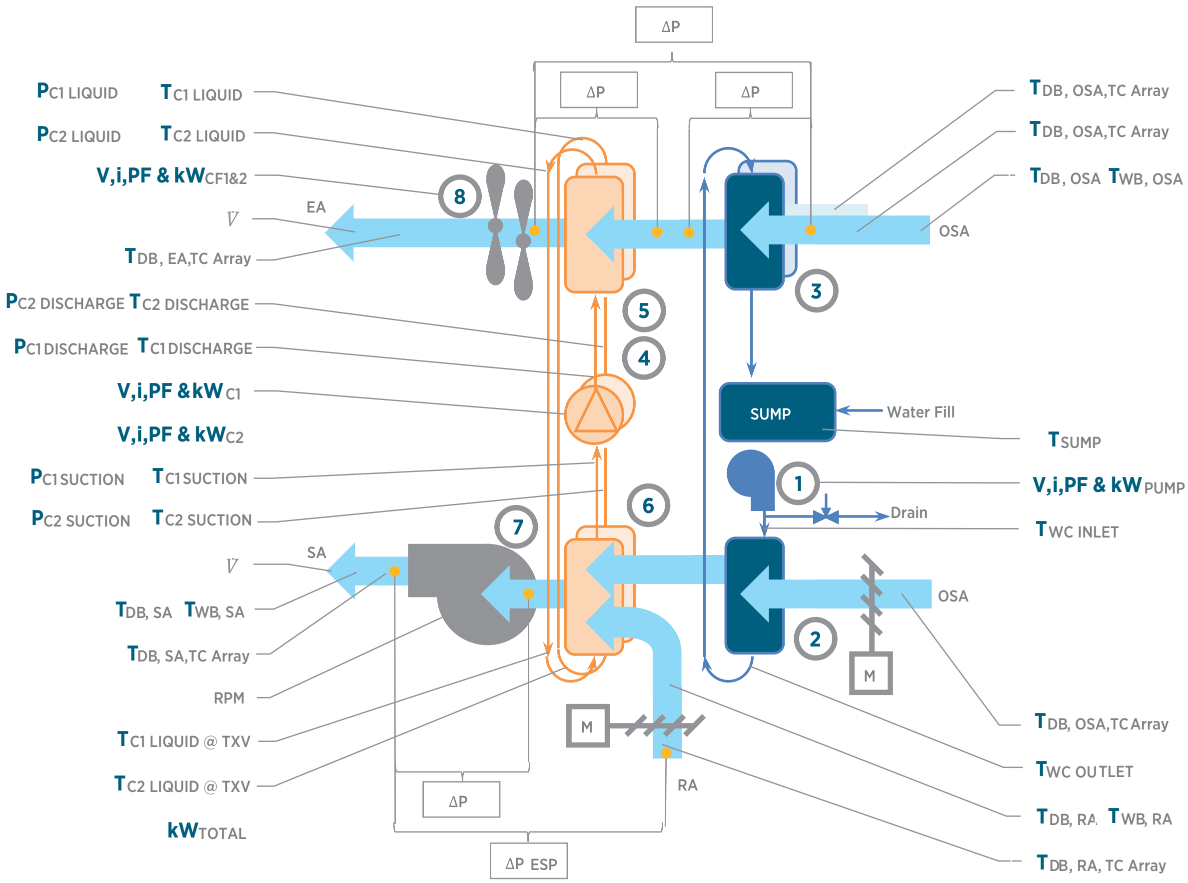
Accuracy the variables directly measured in the Intertek psychrometric test facility, and results from an uncertainty analysis for the key metrics used to describe performance of the equipment are presented in Table 3. The values here are derived from the documented accuracy for sensors types used in the laboratory. Rigorous laboratory measurement techniques, and industry standard test methods are followed to avoid instrument installation errors, environmental effects, and uncertainty due phenomena such as spatial and temporal variation. Such sources of methodological uncertainty are not calculated here.

TABLE 3: DATA CONFIDENCE

Measured Variable	Uncertainty	Calculated Metric	Uncertainty [1]
Temperature (TC, single pt.)	+/- 1.8 °F	Temperature (TC, 9 pt. avg.)	+/- 0.6 °F
Temperature (RTD)	+/- 0.27 °F	Outside Air Fraction	+/- 0.028 (–)
Airflow	+/- 2 %	Sensible Capacity	+/- 6.36 kbtu/h
Static Pressure	+/- 0.025 inWC	Coefficient of Performance	+/- 0.405
Electric Power	+/- 150 W	External Static Pressure	+/- 0.0354 inWC

[1]Uncertainty for derived metrics is calculated for supply air temperature of 60°F, return air temperature of 78°F, outside air temperature of 105°F, outside air fraction of 43%, supply air volume of 6000 cfm, sensible capacity of 224 kbtu/h, and power draw of 16.5 kW.

FIGURE 5: INSTRUMENTATION SCHEMATIC FOR TRANE VOYAGER DC



1. Pump
2. Water Coil
3. Evaporative Cooler
4. Compressors 1 & 2
5. DX Condenser Coils
6. DX Evaporator Coils
7. Supply Air Blower
8. Condenser Fan

Figure 5 details the layout of instrumentation for the laboratory test. It also provides a detailed schematic locating all key components in the Trane Voyager DC system. All measurements denoted were made with Intertek laboratory equipment.

Dry bulb and wet–bulb temperature for the outside air, return air, and supply air streams were measured with Class-A accuracy platinum RTDs. The wet–bulb temperature for each air stream was measured with a wicking psychrometer; this method yields the wet–bulb condition directly instead of relying on calculation from a relative humidity measurement. Space–average temperature measurements for each air stream were achieved with an aspirated sampler that spanned the cross–section of each flow and extracted a de minimis portion of each for measurement. The outside air condition recorded for each test was a measurement of the physical mix from aspirated samplers at the ventilation air inlet and both condenser–air inlets. All of these aspirated temperature measurements were corroborated with nine point averaging thermocouple arrays that spanned the same airflow cross–sections.

The condenser outlet temperature was measured as an average of eight separate point thermocouple measurements located downstream of the condenser fans and mounted to the fan guards. Water temperatures and refrigerant temperatures were measured with single point, surface mounted, insulated thermocouples. The unit ESP was recorded continuously through each experiment, while all other differential pressures were recorded manually at a single time for each test. Supply blower RPM was also recorded m annually for each test.

In order to acquire component-by-component electric power consumption, each major electrical device was powered separately, instead of through the equipment disconnect. Power was supplied by laboratory transformers that provided an appropriate three-phase source, while recording voltage, amperage, and power factor. In order to reduce uncertainty in calculations for overall equipment efficiency, the total equipment power draw was also measured directly, instead of relying on the sum of power draw by each component. Disaggregating the equipment power consumption was also used to calculate the presumed temperature rise across the supply blower, and to correct for inconsistencies in the condenser fan power draw due to multiple motor malfunctions during some tests.

As described previously, supply airflow and condenser–airflow for the Trane Voyager DC were determined in nozzle airflow measurement stations, according to ANSI/AMCA 210-2007 and ANSI/ASHRAE 51-2007.

It should be noted that some measurements that would have been very useful to this study were not possible. Temperature of the condenser–air flow in between the evaporative cooler and the condenser coils would be suspect to error due to the potential for moisture deposition on the sensors. Also, the physical space between the evaporative media and condenser coil is not amenable to sensor placement. Likewise, while the air temperature between the water coil in the ventilation air stream and the evaporator coil would provide useful information about the equipment's indirect evaporative cooling performance, there was no straightforward way to place instrumentation due to space limitations and physical access. It is possible that theses discontinuities in data could be addressed with direct measurements in future laboratory and field studies, though for this study they are non-essential variables that can be reverse-calculated using data from other system measurements, as described later. Lastly, and unfortunately, the Intertek facility was not able to measure supply water consumption, drain flow, or volume flow rate through the circulation pump.

6. DESIGN OF EXPERIMENTS

Western Cooling Challenge test criteria and performance requirements prescribe the return air condition and two outside air psychrometric conditions at which performance is evaluated for certification. These requirements focus primarily on system efficiency at full capacity operation during daily high temperature periods in hot dry climates. In addition to evaluating performance at these few conditions, the range of laboratory experiments was designed to characterize equipment performance in various operating modes and across a broad range of temperature and humidity scenarios.

Many of the tests conducted were outside the intended operating envelope for the Trane Voyager DC, but measurement under such circumstances allowed for analysis of equipment performance sensitivity. The broad mapping of system operation in each mode also allowed for evaluation of component performance characteristics in response to a range of environmental conditions. For example, even though the equipment sequence of operation constrains "enhanced economizer" mode to periods when outside air temperature is below the indoor comfort set point, experiments tested operation in this mode across a range of outside air temperatures between 65 $°F$ and 105 $°F$. Results from these tests helped to isolate performance of the indirect evaporative ventilation air cooling coil by measuring it's impact while in an operating mode where it is the only component to provide cooling.

For each of the three distinct modes of cooling operation, eight different outside air conditions were tested. Four of these psychrometric conditions replicate standard test conditions defined by ANSI/AHRI 340/360-2010 for EER and IEER rating of commercial unitary air conditioning equipment. Two are the Western Cooling Challenge rating conditions, one is a warm-humid condition used to test performance sensitivity to humidity, and one is a mild temperature condition with absolute humidity that is representative of semi-arid climates such as California. These lower temperature conditions constitutes a significant portion of the cooling hours for commercial buildings, and strategies that extend the envelope for very high efficiency economizer-type cooling modes would have great energy savings potential.

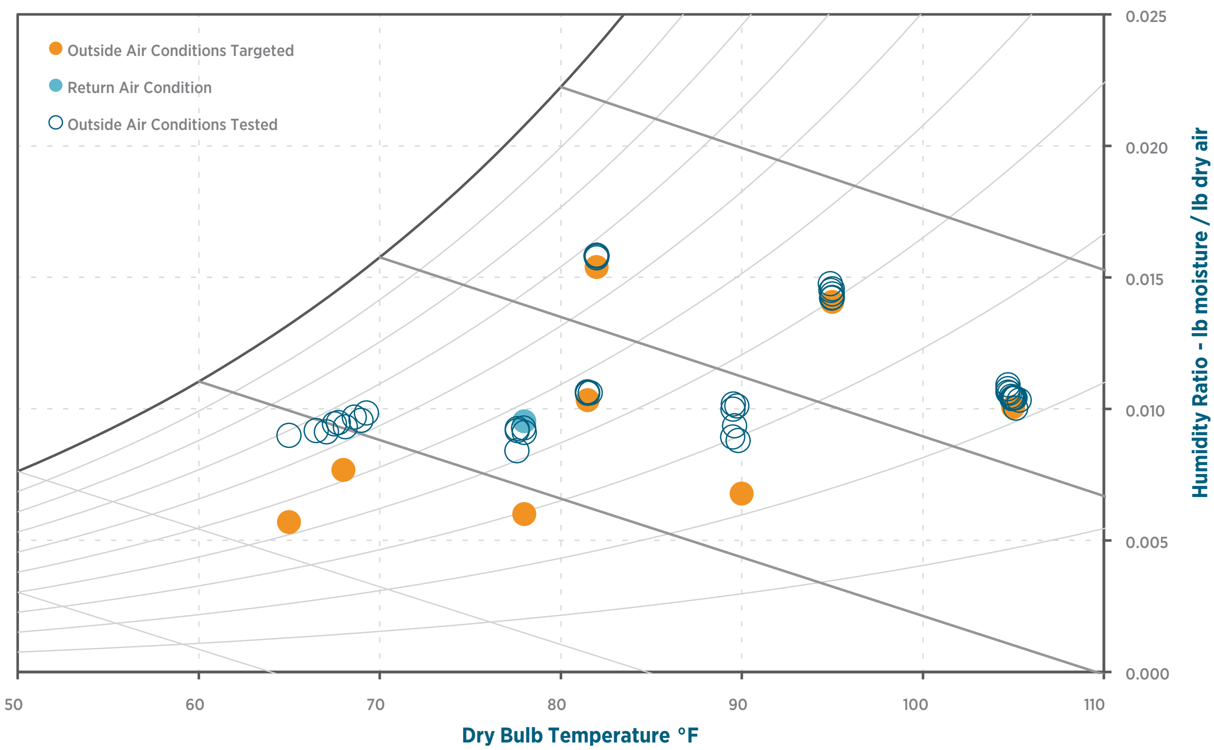
Table 4 details the design of experiments. Each target condition and combination of component operations described was tested a single time, with no replication of tests. Prior to data collection, each scenario was set to operate for at least thirty minutes and up to two hours to allow the equipment, psychrometric chambers, and laboratory air handler systems to reach steady state. Once the entire apparatus had found equilibrium, second-by-second data was recorded from every instrument for a period of at least thirty minutes.

Figure 6 (next page) illustrates the range of outside air psychrometric conditions targeted for test in each operating mode. The chart also indicates the conditions that were actually achieved in the "outdoor" environmental chamber. Note that the intended range of psychrometric conditions was not realized. Despite the complete removal of humid condenser exhaust air, and the addition of a dehumidifier for makeup air to the chamber, the laboratory facility was not capable of maintaining absolute humidity levels below roughly 0.0085 lb/lb.

TABLE 4: DESIGN OF EXPERIMENTS

		CONDITION							
		WCC "Peak" Condition	WCC "Annual" Condition	EER & IEER 100% Load	IEER 75% Load	IEER 50% Load	IEER 25% Load	Mild Dry	Warm Humid
		$T_{DB, OSA}$ (°F)							
		105	90	95	81.5	68	65	78	82
		$T_{WB, OSA}$ (°F)							
MODE	**OSAF**	73	64	75	66.3	57.5	52.8	58.5	73
Indirect & Stage 2 DX	MIN	✓	✓	✓	✓	✓	✓	✓	✓
Indirect & Stage 2 DX	100%					✓	✓	✓	✓
Indirect & Stage 1 DX	MIN	✓	✓	✓	✓	✓	✓	✓	✓
Indirect & Stage 1 DX	100%					✓	✓	✓	✓
Indirect Only	MIN		✓						
Indirect Only	100%	✓		✓	✓	✓	✓	✓	✓

FIGURE 6: MAP OF OUTSIDE AIR PSYCHROMETRIC CONDITIONS TARGETED & CONDITIONS ULTIMATELY TESTED



Nominal Capacity & Ventilation Requirements

Nominal capacity for a rooftop packaged air conditioner is typically determined at standard rating conditions according to ANSI/AHRI 340/360–2007. However, since the standard test protocol is not designed to rate equipment operating with ventilation air, it would not fairly describe a comparable nominal capacity for hybrid air conditioners designed especially to capture energy savings in cooling code–required ventilation air. Therefore, the Challenge protocol uses an alternate method to define a nominal capacity that is based on equipment performance at peak conditions while operating with outside air. Once determined, the value is used to set the ventilation rate and the external resistance for Western Cooling Challenge tests, and to determine the sensible credited EER by which a unit qualifies for certification. This alternate nominal capacity is determined by:

$$\dot{H}_{nominal} = \dot{m}_{SA} \cdot (31.5 - h_{SA}^{WCC\ Peak})$$

where 31.5 is the specific enthalpy of return air for AHRI nominal capacity tests. The method uses the enthalpy difference between return air and supply air to discount the capacity for cooling ventilation air and count only the space cooling delivered. This effectively scales the capacity measured under WCC peak conditions to a value that represents operation with 0% outdoor air, as in an AHRI test scenario. However, it does not represent space cooling capacity under AHRI outdoor air conditions, nor does it represent an actual space cooling capacity that would be achieved under any particular condition. This value is determined in parallel with figuring the ventilation rate and external static pressure at which the system will be tested.

External static pressure is measured as differential static pressure between supply and return plenum, with MERV 7 filtration in place. The Challenge requires the system operate with an external static resistance that would develop 0.7 “WC external static pressure at 350 cfm/nominal-ton. Thus, for systems that supply more or less than 350 cfm/nominal-ton, the external static pressure for tests is adjusted to match the same external resistance according to:

$$ESP_{test}\,\{InWC\} = \left(\frac{V_{SA}\left\{\frac{cfm}{nominalton}\right\}}{350\left\{\frac{cfm}{nominalton}\right\}}\right)^2 \cdot 0.7\,\{InWC\} \qquad \text{<<eq. 1>>}$$

The Challenge tests equipment performance while supplying ventilation air, as is generally the case for rooftop packaged equipment in commercial spaces. The protocol requires 120 *cfm* ventilation per nominal ton:

$$\dot{V}_{ventilation} = 120\left\{\frac{cfm}{ton}\right\} \cdot \dot{H}_{nominal}\{tons\}$$ <<eq. 2>>

Since the nominal capacity is impacted by the required external static pressure and ventilation rate, and since the ventilation rate and external static pressure, these values must be determined through iterative tests. The external resistance effects the supply airflow, so the supply airflow is determined at the same time, and the outside air fraction can be determined according to:

$$OSAF = \frac{\dot{V}_{OA}}{\dot{V}_{SA}}$$ <<eq. 3>>

The iterative nominal-capacity test resulted in the set of system operating conditions described in Table 5, which were held constant for all subsequent Western Cooling Challenge certification tests. The same external resistance and outside air fraction conditions determined through this process were also used for most other tests in the design of experiments, except those tests operating with 100% OSA. The 100% OSA tests used the external resistance determined here, and allowed the supply airflow to change in response to the change in damper arrangement.

Operating Condition	Value for Tests
Exsternal Static Pressure ("WC)	0.45
Supply Airflow (scfm)	6012
OSAF	43%
Nominal Capacity (tons)	21

Western Cooling Challenge Performance Metrics

The system cooling capacity for the equipment at any given condition is determined according to the airflow rate and the specific enthalpy difference between the mixed air and supply air, as described by equation 4; this is the net cooling produced by the system, including what is lost due to fan heat.

$$\dot{H}_{System} = \dot{m}_{SA} \cdot (h_{MA} - h_{SA})$$ <<eq. 4>>

Note that for the Trane Voyager DC, h_{MA} is a hypothetical condition that does not exist in physical reality. For a conventional air conditioner, "mixed air" is the average temperature and humidity condition entering the vapor compression evaporator coil after ventilation air flow has mixed with return airflow. With addition of the DualCool, ventilation air is cooled before it mixes with return air. In fact, for the Trane Voyager DC configuration tested here, the ventilation air and return air were physically separated until after they'd passed through the evaporator coil. Thus, for Equation 4, h_{MA} is determined as the hypothetical mixture of return air and outside air.

The space cooling capacity (also called recirculation cooling, or room cooling), 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.

$$\dot{H}_{Space} = \dot{m}_{SA} \cdot (h_{RA} - h_{SA})$$ <<eq. 5>>

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 demand dehumidification for comfort. Thus the sensible space cooling is determined according to:

$$\dot{H}_{Space}^{Sensible} = \dot{m}_{SA} \cdot C_p \cdot (T_{RA} - T_{SA})$$ <<eq. 6>>

And the latent space cooling is determined as:

$$\dot{H}_{space}^{latent} = \dot{H}_{Space} - \dot{H}_{Space}^{Sensible}$$ <<eq. 7>>

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

$$\dot{H}_{ventilation} = \dot{m}_{SA} \cdot (h_{MA} - h_{RA})$$ <<eq. 8>>

Since the Western Cooling Challenge rates performance for operation at a particular ventilation rate, if the ventilation rate for operation in a particular mode is greater than the minimum requirement, the excess ventilation air cooling is not counted toward system efficiency. In these circumstances, evaluation of performance for the Challenge only credits a portion of the total ventilation rate, equal to the minimum requirement.

$$\dot{V}_{credited\ ventilation} = 120 \cdot \dot{H}_{nominal}$$

<<eq. 9>>

The credited ventilation rate translates to a credited ventilation cooling capacity as described in equation 10:

$$\dot{H}_{credited\ ventilation} = \dot{m}_{SA} \cdot \left(\left(\frac{\dot{V}_{credited\ ventilation}}{\dot{V}_{SA}} \cdot h_{OA} + \left(1 - \frac{\dot{V}_{credited\ ventilation}}{\dot{V}_{SA}} \right) \cdot h_{RA} \right) - h_{RA} \right)$$

<<eq. 10>>

And the sensible credited ventilation cooling capacity is the portion associated with temperature change:

$$\dot{H}^{sensible}_{credited\ ventilation} = \dot{m}_{SA} \cdot C_p \cdot \left(\left(\frac{\dot{V}_{credited\ ventilation}}{\dot{V}_{SA}} \cdot T_{OA} + \left(1 - \frac{\dot{V}_{credited\ ventilation}}{\dot{V}_{SA}} \right) \cdot T_{RA} \right) - T_{RA} \right)$$

<<eq. 11>>

The sensible credited cooling is the capacity used to rate equipment performance for the Challenge, and is calculated as the sum of sensible space cooling and sensible credited ventilation cooling.

$$\dot{H}^{sensible}_{credited} = \dot{H}^{Sensible}_{Space} + \dot{H}^{sensible}_{credited\ ventilation}$$

<<eq. 12>>

The minimum efficiency requirements for the Challenge are given as sensible credited EER, calculated by:

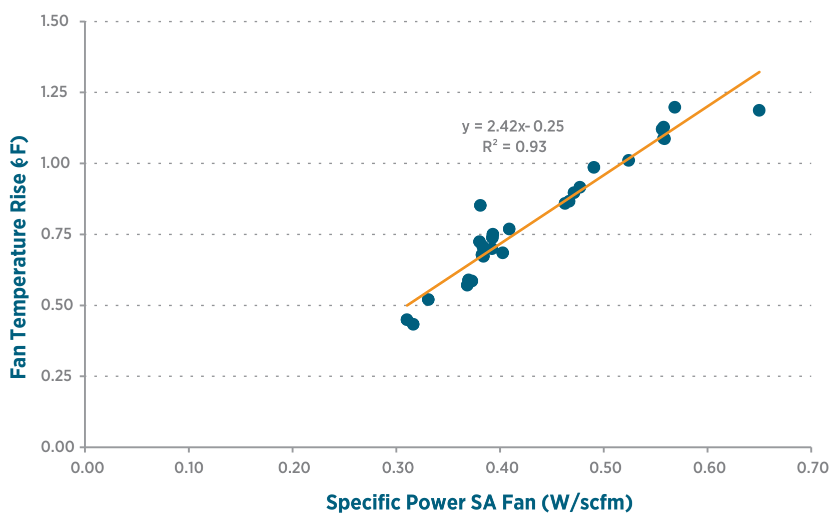
$$EER^{sensible}_{credited} = \frac{\dot{H}^{sensible}_{credited}}{\dot{W}} \left\{ \frac{kbtu/hr}{kW} \right\}$$

<<eq. 13>>

It is important to note that the “sensible credited EER” values presented in this report are not directly comparable to common “EER” values determined according ANSI/AHRI 340/360–2007 standard protocol, which operates equipment without outside air, and gives credit for latent cooling. A conventional system rated with an EER of 12 according to ANSI/AHRI 340/360 will have a “sensible credited EER” nearer 9 according to Western Cooling Challenge test conditions.

FIGURE 7: MAP OF SUPPLY FAN PERFORMANCE AS A FUNCTION OF AIRFLOW

FIGURE 8: TEMPERATURE RISE ACROSS THE SUPPLY FAN AS A FUNCTION OF AIRFLOW–SPECIFIC FAN POWER



Mapping Supply Fan Performance

In order to characterize airflow behavior for the Trane Voyager DC, apart from evaluation of thermal performance, the equipment was run through a battery of airflow–only tests at various fan speeds and external resistances. Figure 7 charts the results, describing fan differential pressure, temperature rise across the fan, ESP, and electric power draw as a function of supply airflow.

Fan Temperature Rise

Fan temperature rise was determined as part of the fan mapping tests, where the supply blower was run on its own while the "indoor" and "outdoor" environmental chambers were maintained at equal conditions. In this scenario, the difference between supply air temperature and return air temperature was used to calculate the sensible heat imparted by the fan. Since the temperature rise changes as a function of motor load and airflow, the measured results for fan temperature rise over 35 separate airflow–only tests were evaluated as a function of the flow–specific fan power ($W/scfm$) to develop a mathematical relationship that could be applied to other tests. Figure 8 charts the results. With this relationship, the presumed fan temperature rise could be calculated for thermal tests where airflow and supply fan power were known.

Calculating Outside Air Fraction

The Intertek laboratory facility was not capable of measuring supply air flow rate and ventilation air flow rate simultaneously; determining the outside air fraction for the experiments required an innovative but laborious method. For any combination of return air damper position, outside air damper position, external resistance, and supply fan shiv setting, a separate test was run for the sole purpose of determining outside air flow rate. This additional test operated all fans and dampers as they would be run in the experiment of interest, but all thermal components remained off. The "outdoor" environmental chamber and "indoor" environmental chambers were maintained at conditions with a 40°F temperature difference such that when airflow from each chamber mixed through the unit, the resulting supply air temperature would indicate the fraction of flow originating as outside air. Using this method the OSAF is calculated by:

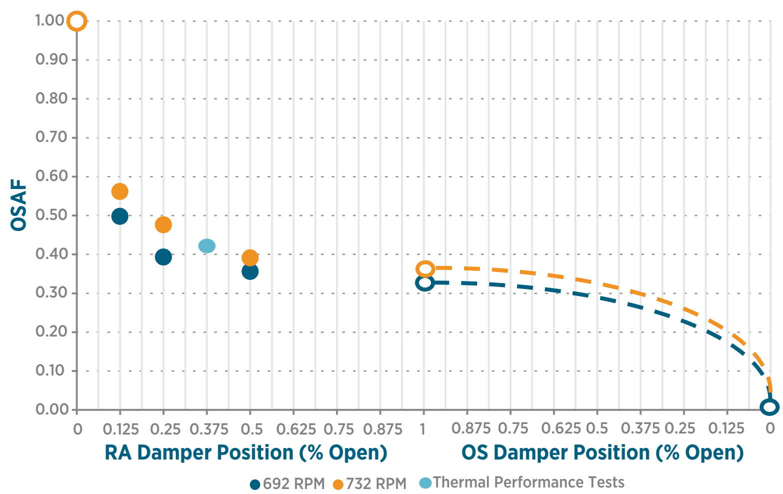
$$OSAF = \frac{(T_{db,SA} - \Delta T_{fan} - T_{db,RA})}{(T_{db,OSA} - T_{db,RA})} \qquad \text{<<eq. 14>>}$$

where ΔT_{fan} is the air temperature rise across the supply blower.

Tests to determine outside air fraction illuminated that for the physical configuration of the Trane Voyager DC examined, any outside air fraction above roughly 30% required partial closure of the return air damper to restrict return airflow. As will be discussed in the results section, this flow restriction has some consequences on total supply airflow when operating at high outside air fractions.

Outside air fraction measurement tests yielded the results charted in Figure 9. The figure indicates outside air fraction as a function of damper position for two different fan speeds operating against the same external resistance. It also plots the damper position and outside air fraction measurement used for all thermal performance tests. The closed points chart actual laboratory measurements, while the open points and dashed lines chart reasonably presumed trends that were not measured. The horizontal axis indicates the combination of outside air and return air damper positions. The left extreme indicates the return damper as fully closed and the outside air damper as fully open. The right extreme indicates the outside damper as fully closed and the return damper as fully open. The center point marked "1" indicates that both dampers are fully open. It should be noted that these tests were for a very high external resistance that yielded external static pressures from 1.2 – 1.5 $"WC$. The OSAF measured for the thermal performance tests was for 0.45$"WC$ ESP.

FIGURE 9: OUTSIDE AIR FRACTION AS A FUNCTION OF DAMPER POSITIONS FOR VARIOUS FAN SPEEDS



Calculating Water Circuit Flow Rate

Water flow rate through the pump and water circuit was not directly measured. Calculation of the over-arching equipment efficiency does not require this value, but it is a useful metric to tease apart the performance of system sub–components. Therefore, the value was reverse–calculated using an energy–balance for the ventilation air cooling coil. This calculation was exercised for tests where the system was operated in the enhanced economizer mode. In this scenario, the equipment operated as 100% outside air and the indirect evaporative ventilation air cooling coil was the only component to provide cooling. The water circuit flow rate was calculated by:

$$\dot{V}_{pump} = \dot{V}_{osa} \cdot \frac{\rho_{air}}{\rho_{water}} \cdot \frac{c_{air}}{c_{water}} \cdot \frac{(T_{OSA} - T_{SA} + \Delta T_{fan})}{(T_{water\ out} - T_{water\ in})}$$ <<eq. 15>>

where $T_{water\ out}$ and $T_{water\ in}$ are the water temperature at the outlet and inlet of the ventilation air cooling coil. Of the seven enhanced economizer tests with 100% outside air, this calculation was limited to the tests with outside air temperature T_{db}=105°F, 95°F and 81.5°F. Tests with cooler outside air conditions yielded such small temperature shift across the water coil that uncertainty in the resulting energy balance yielded very high uncertainty for the resulting water flow rate. Since there were not physical alterations to the water circuit from test to test, the water flow rate was assumed to remain constant for every experiment.

CALCULATING CAPACITY FOR THE VENTILATION COOLING COIL

As discussed previously, the ventilation air cooling coil and the DX coil were installed so close to one another that measurement of the average ventilation airflow temperature between the coils was not a reasonable prospect. This value is not required to evaluate the overall equipment efficiency, but it is useful to describe effectiveness of the indirect evaporative process and capacity of the ventilation cooling coil. These metrics were calculated using the previously calculated water flow rate, and an energy–balance for the ventilation air cooling coil. Equation 17 describes calculation of the wet–bulb effectiveness for the indirect evaporative cooling of ventilation air:

$$WBE_{IEC} = \frac{\frac{\dot{V}_{pump}}{OSAF \cdot \dot{V}_{SA}} \cdot \frac{\rho_{water}}{\rho_{air}} \cdot \frac{C_{water}}{C_{air}} \cdot (T_{water\ out} - T_{water\ in})}{T_{db\ OSA} - T_{wb\ OSA}}$$ <<eq. 16>>

Likewise, the sensible cooling capacity of the indirect evaporative ventilation air cooling coil is given by:

$$\dot{H}_{IEC} = OSAF \cdot \dot{V}_{SA} \cdot \rho_{air} \cdot C_{air} \cdot \Delta T_{air}$$ <<eq. 17>>

where ΔT_{air} is given by:

$$\Delta T_{air} = WBE_{IEC} \cdot (T_{db\ OSA} - T_{wb\ OSA})$$ <<eq. 18>>

Results of these calculations are presented later, along with discussion about implications to equipment performance.

Calculating Condenser Inlet and Outlet Conditions

Similar to the indirect evaporative cooling coil, the sensible cooling provided by the evaporative condenser–air pre cooler was not measured directly. This was mostly because of challenges with physically locating temperature sensors to accurately capture space average dry bulb temperature in between the evaporative media and the condenser coil. For the tests where compressors did not operate, performance of the evaporative cooler could be described by the temperature measured at the condenser outlet according to:

$$WBE_{DEC} = \frac{T_{db\ OSA} - T_{DB\ Condenser\ Outlet}}{T_{db\ OSA} - T_{wb\ OSA}}$$ <<eq. 19>>

For other tests, where the compressors were running, the condenser inlet temperature was calculated by:

<<eq. 20>>

$$T_{DB\ Condenser\ Inlet} = T_{DB\ Condenser\ Outlet} - \frac{\Delta h_{Condenser}}{C_{\ air}}$$

The enthalpy difference for air across the condenser coil is determined by an energy balance considering the condenser–air flow rate, and the condenser heat transfer rate measured on the refrigerant side:

$$\Delta h_{Condenser\ Outlet} = \frac{\dot{H}_{Condenser}}{\dot{V}_{Condenser\ Air} \cdot \rho_{Air}}$$ <<eq. 21>>

For this method, the calculated condenser inlet temperature can be applied to equation 19 in place of the measured outlet temperature to describe evaporative cooler performance.

$$WBE_{DEC} = \frac{T_{db\ OSA} - T_{DB\ Condenser\ Inlet}}{T_{db\ OSA} - T_{wb\ OSA}}$$ <<eq. 22>>

It is also possible to determine humidity for the condenser inlet and outlet conditions. The calculation relies on the fact that the condenser coil only provides sensible heat exchange, and is based on an energy balance for the direct evaporative media. Note that since water flow enters the media well above the wet–bulb temperature, a portion of the enthalpy for phase change is drawn from the water flow, and a portion is drawn from the air flow. This results in sensible cooling for both the air and water, and a net enthalpy increase for the air flow across the evaporative media.

$$\omega_{Condenser\ Outlet} = \omega_{Condenser\ Inlet} = \omega_o + \frac{h_{Condenser\ Outlet} - h_o - \frac{(T_{Condenser\ Outlet} - T_O)}{c_{air}}}{\lambda} \qquad \text{<<eq. 23>>}$$

7. RESULTS

Laboratory tests observed operation in each cooling mode and across a range of psychrometric operating conditions as described by the design of experiments. Observation of the system operating at full capacity for Western Cooling Challenge "peak" test conditions indicates that the Trane Voyager DC uses 43% less power than a minimum efficiency standard air conditioner operating at the same conditions. This satisfies the Western Cooling Challenge performance requirements and qualifies the Trane Voyager DC for certification.

Data collected across the range of other tests was collated and scrutinized to describe performance of the system each major system component in response to relevant environmental variables such as airflow, and wet–bulb depression. The major results and observations are discussed here. A complete summary table of observations for each test is included as Appendix A. Charts to illustrate air–side and refrigerant–side system behavior for every test may be referenced in Appendix B.

Performance for Evaporative Components

Sump water temperature and water temperature at the inlet of the ventilation air cooling coil were recorded for every test, and were found to vary mostly as a function of the outside air wet–bulb temperature. This behavior is to be expected given that wet–bulb is the theoretical equilibrium for water flow in contact with air. The significant observation is that water cooling maintains a regular wet–bulb approach of 1-2 °F, regardless of the wet–bulb depression, and regardless of the thermal load from the ventilation air cooling coil.

Every test also recorded a temperature rise of less than 1°F between the sump water and the inlet to the cooling coil. This small difference is likely due to heat addition through the pump, or heat exchange between the supply water plumbing and outdoors. However, the difference observed is smaller than the limits of uncertainty for the thermocouples, so measurement bias may either exaggerate or minimize this small effect.

FIGURE 10: WATER TEMPERATURE MEASURED IN THE SUMP AND AT THE INLET TO THE VENTILATION AIR COOLING COIL

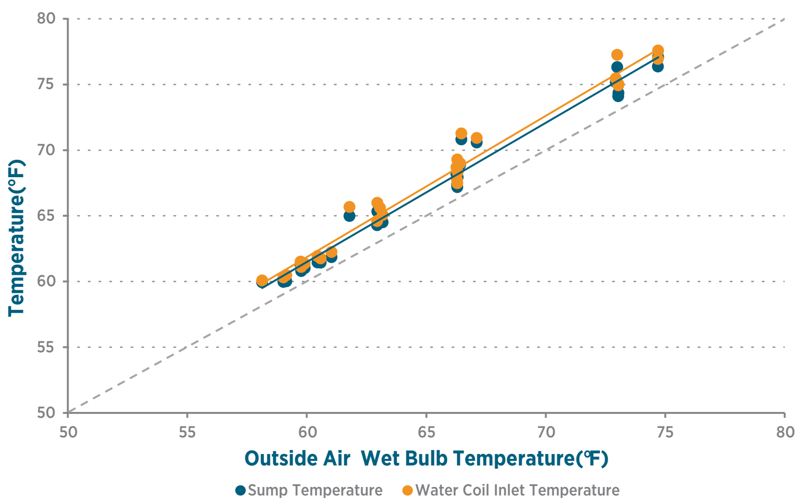


Figure 11 charts the temperature difference for water and air flows across the evaporative condenser–air pre-cooler and ventilation air cooling coil. The temperature shift for each component was found to vary directly with the outside air wet–bulb depression. Since the evaporative water cooling was observed to maintain a regular wet–bulb approach regardless of operating conditions, it is reasonable to expect that the magnitude of temperature shift across each component in this evaporative outdoor air cooling system should be proportional to the wet–bulb depression.

Data for air temperature drop across the ventilation air cooling coil is split into two series for Figure 11. Operation with 100% outside air results in a smaller air temperature difference than for operation at 43% outside. At the same time, water temperature change across the ventilation air cooling coil does not respond significantly to the ventilation airflow rate. This observation indicates, importantly, that the cooling capacity delivered by this coil does not vary with the amount of ventilation air flow. Future optimization of system control sequences should take this fact into consideration. For example, it may be preferable to use a low supply fan speed, 100% outside air, indirect–evaporative cooling mode as the first stage of space cooling, even while outside air temperature is well above the indoor control temperature set point.

Similarly, it is significant that air temperature drop across the condenser–air pre-cooler is mainly a function of outside air wet–bulb depression for all tests, and not dependent on the condenser–air flow rate. The condenser–air flow varied from roughly 6,300 to 12,500 *scfm*, depending on the mode of operation, but this did not impact the air temperature drop. The same is true for water cooling. For all outside air web-bulb temperatures tested, the condenser–airflow rate had no apparent bearing on the ultimate sump temperature.

Figure 12 charts the wet–bulb effectiveness of the condenser–air pre–cooler and indirect evaporative ventilation air cooling coil; it is a reinterpretation of the air temperature data shown in Figure 11 to illustrate performance of these components as a ratio with the theoretical limit for direct evaporative cooling. These observations indicate that that the wet-bulb effectiveness for indirect evaporative cooling depends significantly on the ventilation airflow rate. Similar to the observations from Figure 11, tests with 100% outside airflow do not cool ventilation air as far as tests with a lower ventilation airflow rate.

FIGURE 11: TEMPERATURE SHIFTS FOR WATER AND AIR FLOWS IN THE DUALCOOL COMPONENTS

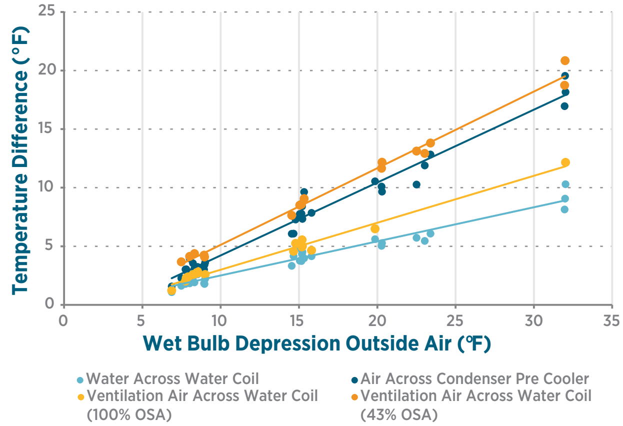
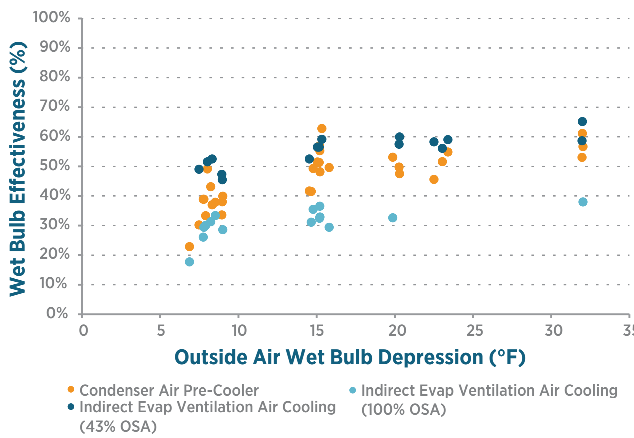


FIGURE 12: WET-BULB EFFECTIVENESS AS A FUNCTION OF WET-BULB DEPRESSION



Further scrutiny of the results presented in Figure 12 yields a few significant observations. First, wet–bulb depression appears to have some impact on the wet–bulb effectiveness for both components. This is most likely due to the fact that the rate of sensible heat transfer is driven by temperature difference and that conditions with lower wet-bulb depression yield smaller temperature difference to drive heat transfer between the water streams and air streams. Second, and more importantly, it is apparent that for lower ventilation rates, the indirect evaporative cooler actually achieves better wet-bulb effectiveness than the direct–evaporative condenser–air pre–cooler.

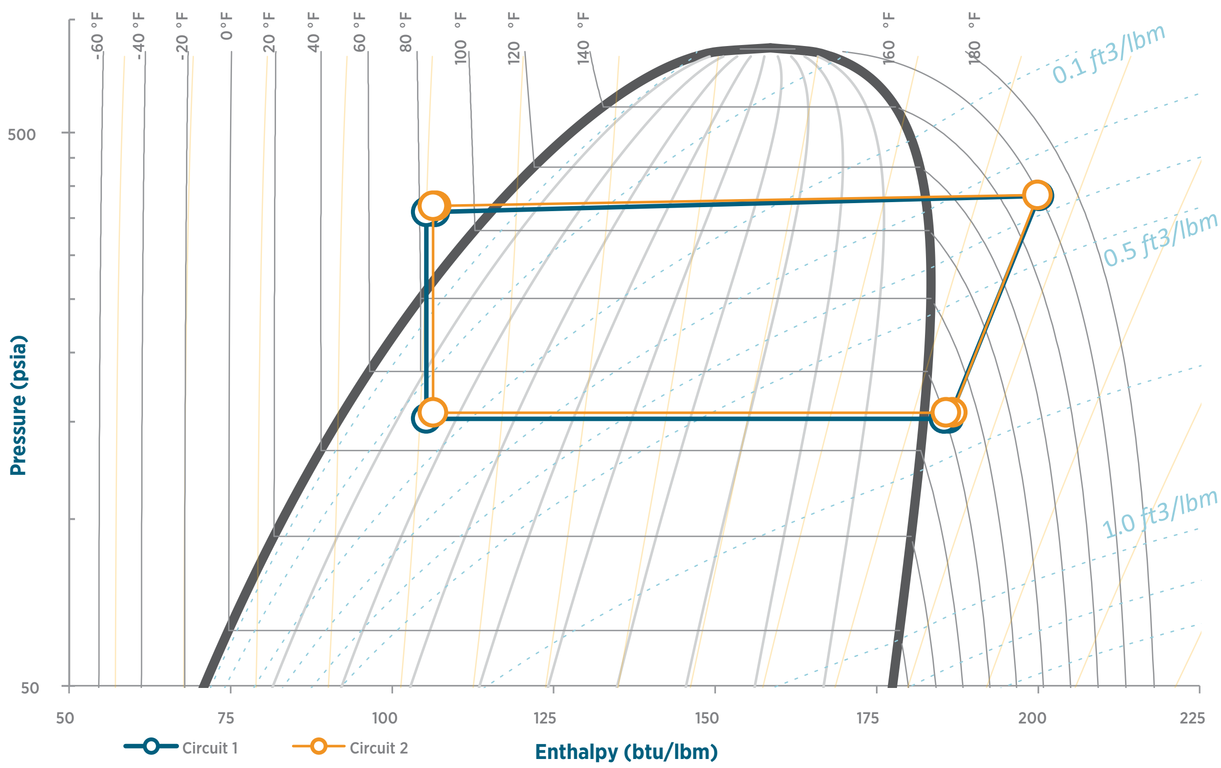
One may also reflect that the wet–bulb effectiveness for the evaporative condenser–air pre–cooler is lower than what might regularly be seen for a direct evaporative system. This is particularly important considering that the sump water is cooled so close to the wet–bulb temperature. Conclusive explanation of this result may require further investigation, but a few possible reasons deserve consideration. First, since warm water delivered to the top of the evaporative media cools as it flows down to the sump, the condenser–air flow crossing this media is presented with an unequal temperature face for sensible heat exchange. Airflow through the upper portion of the media will be presented with a smaller temperature difference to drive convective heat transfer than airflow through the bottom of the media. Second, it is possible that water distribution across the media was uneven, such that even while the water flow cooled to very near wet–bulb there were some drier sections of the media that allowed bypass without adequate opportunity for cooling. Neither of these possibilities can be substantiated from the laboratory data available. Also, high airflow across an evaporative media can result in reduced wet - bulb effectiveness, although this last mechanism doesn't seem a likely factor in this case since tests with widely varying condenser–airflows has no obvious impact on wet - bulb effectiveness of the condenser–air pre–cooler

Refrigerant Side Performance

Refrigerant temperatures and pressures were measured throughout each compressors circuit, as described earlier, and the resulting observations from each test were plotted on a pressure-enthalpy diagram for R410a. Results from the entire range of tests can be referenced in Appendix C. The most compelling observation from these refrigerant measurements is to note the liquid line temperature relative to outside air conditions. Figure 13 charts refrigerant measurements from the Western Cooling Challenge "Peak" test. Even while the outside air temperature is 105°F, the condenser is able to cool liquid refrigerant down to 86°F. The condenser inlet temperature for this test is only 160°F, at least 30°F cooler than it would need be without the DualCool components. This all amounts to significant compressor load reduction.

Despite the great performance increase due reduced compressor temperature, these refrigerant–side observations also indicate room for additional system improvements. In particular, if heat exchange effectiveness for the condenser coil were improved, the compression ratio could be controlled to avoid liquid sub-cooling, and the same cooling capacity could be achieved with much less compressor power input.

FIGURE 13: PRESSURE-ENTHALPY DIAGRAM FOR WESTERN COOLING CHALLENGE "PEAK" CONDITIONS



Psychrometric Performance

The most conceptually illustrative way to describe behavior of the integrated system is to plot air flow conditions on a psychrometric chart. Figure 14 charts results from the full capacity tests at Western Cooling Challenge "Peak" conditions, and Figure 15 through Figure 17 chart results from tests in each operating mode near Western Cooling Challenge "Annual" Conditions. All of these experiments ran the system with approximately 120 ***cfm-osa/ nominal-ton*** ventilation air, as prescribed by the Challenge test criteria. For the supply fan speed selected, this ventilation rate corresponded to roughly 43% OSAF.

For reference, water temperature at the inlet and outlet of the ventilation cooling coil are plotted along the horizontal axis; temperature of the liquid refrigerant is plotted as well. Solid markers in these figures indicate that the temperature and humidity condition were measured, while the open markers indicate that the condition was calculated as described in the ***Experimental Methods & Calculations*** section. Note that the air temperature was measured at the condenser outlet, while humidity was not.

Arrows on each chart plot the general psychrometric trajectory of each airflow stream. Recall that the ventilation airstream and return airstream were physically separated for the Trane Voyager DC configuration that was tested, thus two arrows converge on the supply air condition. Although each point plotted represents a physical measurement or calculation, they are values for space–averaged conditions. For example, the return airstream and ventilation airstream very likely cool to different conditions across the evaporator coil. Here, a single, mixed supply air condition is plotted that includes the addition of fan heat.

FIGURE 14: PSYCHROMETRIC PERFORMANCE AT CHALLENGE "PEAK" CONDITIONS FOR INDIRECT +STAGE 2

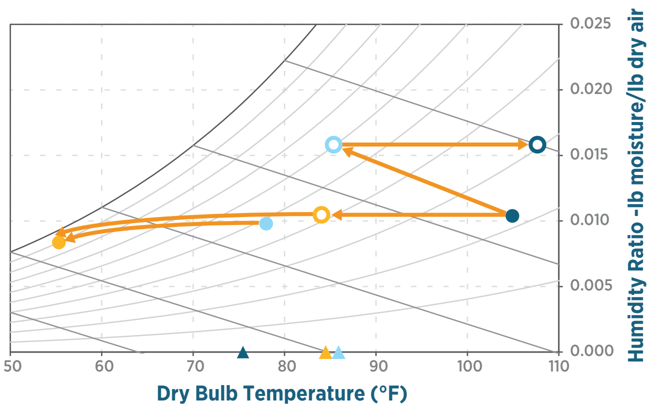


FIGURE 15: PSYCHROMETRIC PERFORMANCE AT CHALLENGE "ANNUAL" CONDITIONS, IEC + STAGE 2, 43% OSA

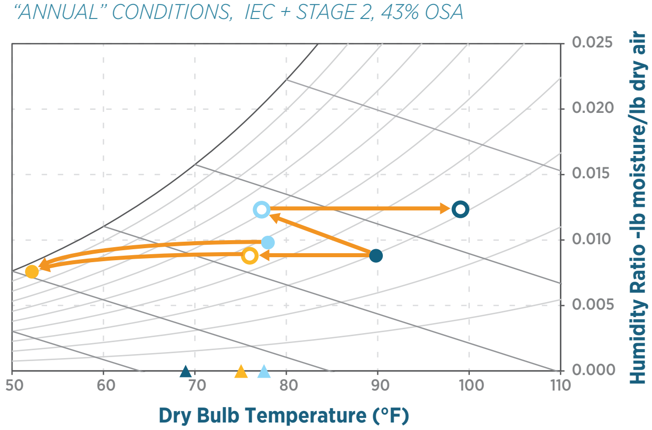


FIGURE 16: PSYCHROMETRIC PERFORMANCE FOR CHALLENGE "ANNUAL" CONDITIONS, IEC + STAGE 1, 43% OSA

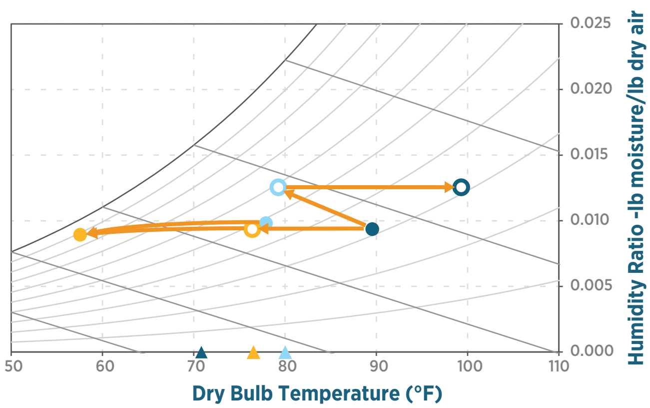
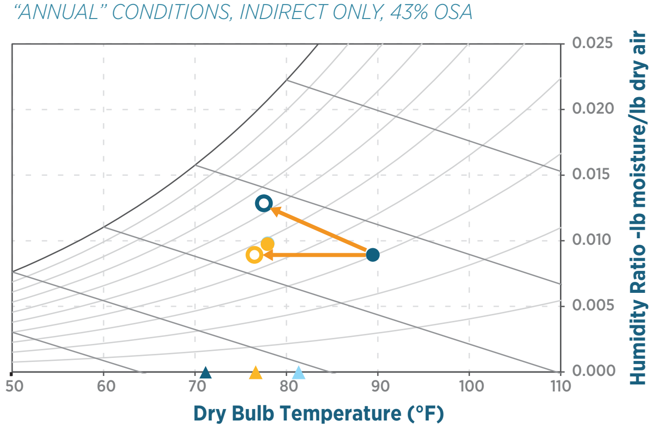


FIGURE 17: PSYCHROMETRIC PERFORMANCE FOR CHALLENGE "ANNUAL" CONDITIONS, INDIRECT ONLY, 43% OSA



For the “Peak” condition test, indirect evaporative ventilation air cooling offloads the ventilation cooling load significantly, delivering air to the vapor compression evaporator coil at approximately 84°*F*, instead of 105°*F*. For “Annual” conditions, indirect evaporative completely cuts the ventilation cooling load and actually provides a minor amount of space cooling capacity. For all cases, air and water exit the ventilation cooling coil nearly the same temperature. Similarly, the condenser liquid line temperature is always only 1-2°*F* warmer than the condenser inlet temperature after the evaporative condenser–air pre–cooler.

For Figure 14 through Figure 17, note that the condenser inlet condition is at a somewhat higher specific enthalpy than the outside air. As explained in ***Experimental Methods & Calculations*** section this difference is due to transfer of sensible enthalpy from the ventilation air flow to the condenser–air flow via the indirect evaporative cooling process. Accounting for the difference in air flow rates, the total enthalpy gain across the condenser–air pre–cooler is equal to enthalpy decrease for airflow across the ventilation air cooling coil.

Figure 17 plots performance of the system in an indirect evaporative only mode, operating with 43% outside air. While operation with minimum ventilation air and indirect evaporative cooling only is not a part of the current sequence of operations, future revisions should consider the benefit of this scenario to cover ventilation cooling load while there is no active call for cooling. In fact, operation in this mode could even provide a significant amount of space cooling for certain conditions, effectively extending the range for economizer operation.

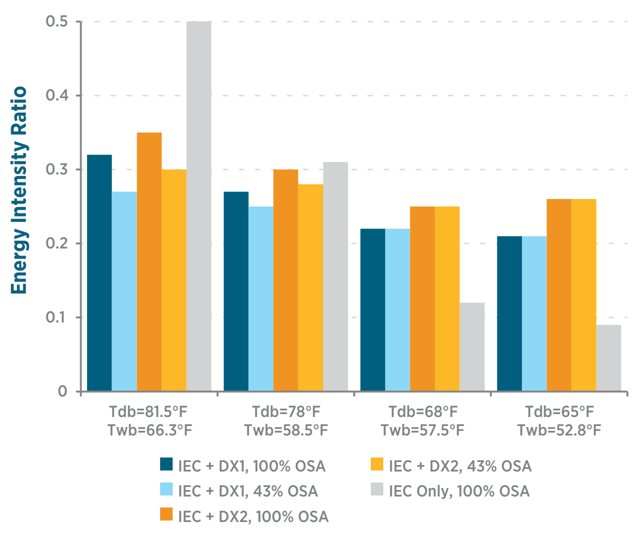
Appendix B provides similar psychrometric charts for the entire range of tests that were conducted.

Integrated Economizer Operation

For outside air conditions cooler than the room set point temperature, the Trane Voyager DC is programmed to operate in an integrated comparative economizer mode that is aided by the indirect evaporative cooler. In this mode, the system would shift to 100% OSA, the indirect evaporative cooler would operate, and the compressors would cycle as needed for additional cooling capacity. When the indirect evaporative ventilation air cooling is adequate to cover thermal loads, the compressors would remain off. Ostensibly, an integrated differential economizer mode should improve efficiency even without the indirect evaporative ventilation cooling coil. The added benefit of ventilation air cooling should improve efficiency, and increase capacity to offset the need for compressor operation during these times. To test the impact of this integrated economizer mode, experiments were run at several outside air conditions, in all three modes of operation, first with 43% outside air, then with 100% outside. Contrary to expectations, observations show that in most scenarios, it may actually make more sense to operate with the minimum ventilation rate, rather than with 100% outside air, even when outside air is cooler than the room set point temperature.

Results indicate that at for tests where outside air temperature was above the return air temperature, space cooling capacity and efficiency for operation with the two compressor stages are both hurt by a shift to 100% outside air. For outside air temperatures below the indoor set point, we observe that there is no energy efficiency improvement for a shift to 100% outside air, and that space cooling capacity decreases. These trends are illustrated in Figure 18. To be clear: at these lower outside air temperatures, it is much more efficient to operate in the so-called “enhanced economizer” mode than to operate compressors, but a switch to 100% outside air does not improve performance when compressors are operating. This observation comes as a surprise, since it should be preferable to work with cooler outside air than to work with ventilation air, especially with the added indirect evaporative capacity.

FIGURE 18: COMPARISON OF ENERGY INTENSITY RATIOS FOR ECONOMIZER OPERATION



Upon scrutiny of the results, two factors seem to contribute to the patterns observed. First, since the return airflow and outside airflows are physically separated until they pass the vapor compression evaporator coil, a switch to 100% outside air significantly reduces the coil area for heat exchange with compressor operation. Second, that supply airflow is reduced significantly with a switch to outside air. Operating at 43% outside our, the Trane Voyager DC delivered approximately 6000 *scfm*, while a switch to 100% outside air resulted in airflow nearer 5300 *cfm.* Since the fan speed and resistance to supply airflow remained the same for all tests, this airflow reduction must be attributed to undue resistance in the outside air path. It is likely that with a larger outside air pathway, and removal of the separation between outside airflow and return airflow, 100% outside airflow would be beneficial for these mild temperature compressor operating modes.

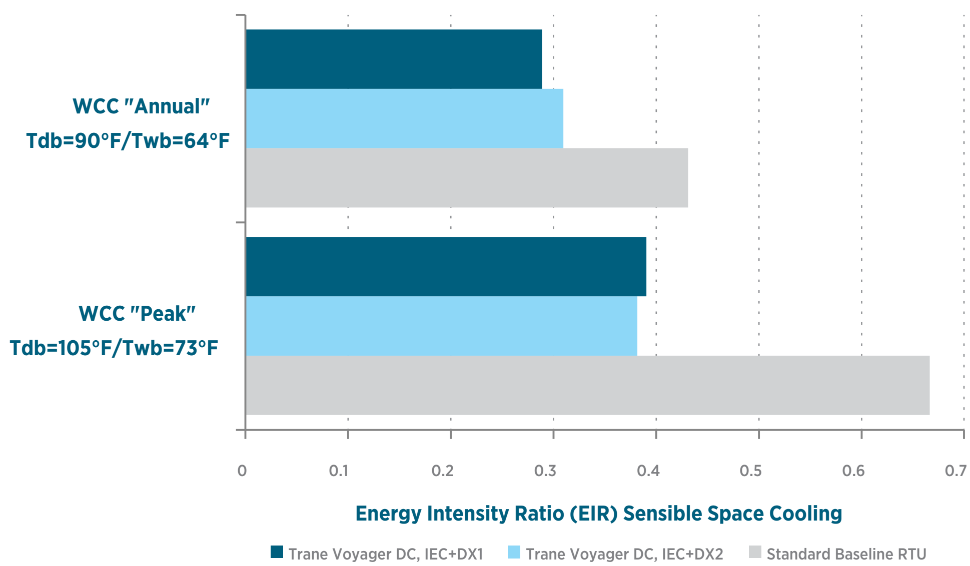
It should also be noted that for outside air conditions near 65°F, the indirect evaporative coil only provides about 3–4°F sensible cooling, and it is unclear whether or not the condenser fan and pump power are worth the small added capacity for the "enhanced economizer" operation. Although it was not evaluated by these laboratory tests, it may prove more efficient to operate in a true economizer mode for mild outside air temperatures, where the supply fan is the only operating component.

System Energy Efficiency

A summary table of observations and calculated efficiency for every laboratory test is presented in Appendix A. Since the Intertek laboratory facility was not capable of testing at the Challenge "Annual" condition, determination of Western Cooling Challenge certification is based on measured performance at the Challenge "Peak" conditions only. Performance at this condition indicates the Trane Voyager DC delivers cooling with 43% less electrical energy, as compared to equipment that meets federal minimum efficiency standards.

Figure 19 charts the Trane Voyager DC's coefficient of performance for sensible space cooling at Challenge "Peak" and "Annual" conditions compared to that of a standard federal minimum efficiency rooftop unit operating at similar conditions. The performance results for "Annual" conditions are shown for the sake of comparison, even though the laboratory did not meet the prescribed humidity conditions for the "Annual" test. It is anticipated that at the appropriate humidity, performance improvement would be even more significant than the results shown.

FIGURE 19: ENERGY INTENSITY RATIO FOR TRANE VOYAGER DC COMPARED TO STANDARD 1 STAGE CAV RTU



8. CONCLUSIONS

The Trane Voyager DC is a hybrid rooftop packaged air conditioner that couples a conventional vapor compression cooling system with a unique evaporative cooling process that cools both condenser–air flow, and ventilation air flow, without adding any moisture to the space. The system can provide space cooling with the evaporative components operating alone, and can cycle two compressor stages to provide added cooling capacity as needed. The technology achieves energy savings in two main ways. First, it allows the compressor to operate with a lower compression ratio by providing cooler air to the vapor compression condenser. Second, it reduces load on the vapor compression evaporator by cooling the system's fresh ventilation air. The sequence of operations incudes an "enhanced economizer" mode where the outside air temperature range for effective economizer operation can be extended due to the added capacity of the indirect evaporative cooling for the ventilation air.

The Trane Voyager DC was submitted for certification by the UC Davis Western Cooling Challenge. WCEC utilized the 35 *ton* psychrometric test facility at Intertek in Plano, TX, to conduct laboratory evaluation of the equipment. Although the Intertek facility was not able to test all of the conditions originally prescribed, results from a broad range of tests do highlight several important performance characteristics, and provide enough information to qualify the equipment for Western Cooling Challenge certification.

The laboratory facility utilized was not able to measure a number of system operating variables that would have helped to describe all aspects of system behavior with very high accuracy. Instead, analysis of some performance metrics required second–hand correlation to primary measurements. These calculations resulted in a somewhat higher, though acceptable, level of theoretical uncertainty. Certain techniques, such as the method utilized for determining outside air fraction, introduce methodological uncertainties that cannot be straightforwardly quantified.

A summary table of the measurements and key metrics calculated for each test is recorded in Appendix A. Appendixes B and C illustrate equipment behavior for each test, plotting measurements and calculated metrics on psychrometric charts and refrigerant vapor dome diagrams. Through analysis and consideration of these observations, this research has unraveled some enlightening observations and conclusions about the equipment, including.

- At Western Cooling Challenge "Peak" conditions (T_{db}=105°F, T_{wb}=73°F), and providing 43% ventilation airflow, the Trane Voyager DC operates with an energy intensity approximately 40% lower than that of conventional rooftop air conditioners designed to meet federal minimum efficiency standards and operating at similar conditions.

TABLE 6: SUMMARY OF WESTERN COOLING CHALLENGE RATED RESULTS FOR CERTIFICATION

	WCC Requirements	Measured Results
Outside Air Condition (Tdb°F/Twb°F)	105/73	104.9/72.9
Return Air Condition (Tdb°F/Twb°F)	78/64	78/64
Min Ventilation (cfm/nominal-ton)	120	125
External Static Pressure ("WC) [1]	0.7	0.45
Min Filtration	MERV 7	MERV 8
Operating Mode	Full Capacity	IEC + Stage 2 DX (Full Capacity)
Min Sensible Credited EER (kbtu/kWh)	14	13.54 ± 1.38
Max Supply Air Humidity (lb/lb)	.0092	0.0083
Max Water Use (gal/ ton-h)	NA	NA

[1] Challenge requires external resistance that would produce 0.7"WC at 350 cfm/nominal-ton. For the nominal capacity determined, this corresponds to 0.45 "WC required ESP.

- Water cooling by evaporation regularly achieves 1–3 *°F* wet–bulb approach, regardless of thermal load on the ventilation coil, and regardless of wet–bulb depression for the outside air.
- Cooling capacity delivered by the ventilation air cooling coil is not sensitive to ventilation airflow. Rather, it varies mainly as a function of wet–bulb depression.
- Sump water temperature is apparently independent of sensible load on the ventilation coil, and not impacted by condenser–air flow rate. For a wet–bulb depression above 30*°F* water passing through the evaporative media may achieve sensible cooling of up to 10*°F*, at airflow between 6,300-12,500 *scfm*.
- Depending on airflow, wet–bulb effectiveness for indirect evaporative ventilation air cooling can be greater than the direct evaporative effectiveness for condenser–air pre-cooling.

The technology presented here is one of various indirect evaporative cooling technologies for rooftop units. The Trane Voyager DC seems to be a particularly compelling approach because the components applied in the technology are already widely utilized in the industry. This fact portends good cost effectiveness for a climate-appropriate cooling technology that promises great peak energy savings over conventional rooftop packaged air conditioners.

9. RECOMMENDATIONS

According to laboratory evaluation, the Trane Voyager DC offers significant energy savings potential. Programs and efforts considering application of the equipment for this purpose should account espcially for the system's value during peak electrical demand periods, when the Trane's performance over conventional cooling equipment is most pronounced. On an electric grid forecast for continued demand growth, this peak demand reduction should be valued in contrast to the cost for new peak electric generation capacity.

Notwithstanding the considerable performance improvements offered by this equipment, the laboratory research presented here has identified a number of opportunities for further performance improvement, and has left other significant questions in need of further evaluation:

- Wet–bulb effectiveness for the direct–evaporative condenser–air pre–cooler is lower than what was anticipated. This fact should be investigated; further enhancement of the cooling effect for condenser air would yield additional savings.
- Every test with compressor operation was observed to yield roughly 20°F sub-cooling. This increases evaporator coil capacity for a given compression ratio, but the same cooling capacity could be achieved with much less power if the system operated such that the same condenser outlet temperature yielded saturated liquid refrigerant, instead of a sub-cooled condition. This would require a larger, or more effective condenser coil, and more sophisticated control of refrigerant metering.
- Since the sump water temperature seems to be inelastic to condenser airflow rate, it stands to reason that a similar wet–bulb approach for the sump water temperature would be achieved even with a higher water flow rate. Increasing the pumped water flow rate would increase the ventilation coil cooling capacity. It is possible that a higher water flow rate could result in cooler supply air temperature for 100% outside air indirect evaporative cooling.
- Economizer controls demand a closer evaluation and possible tuning. It appears there may be some advantage to a low fan speed, 100% outside air, indirect evaporative only cooling mode, even when outside air is well above the set point. It may also prove that indirect evaporative cooling for an enhanced economizer mode is not useful at low outside air temperature, but that it could be more efficient to operate in a pure economizer mode to avoid the condenser fan and pump energy investment for small gains in cooling capacity.

TRANE VOYAGER DC

The equipment is designed especially to capture energy savings in applications where the system provides a significant amount of ventilation air. If installed to replace equipment that currently operates without ventilation air, the Trane Voyager DC would not achieve the same degree of savings. The ventilation rate applied to these tests is representative of typical design practices for large retail facilities. If air balance for the building is such that this amount of ventilation is not required, the Trane Voyager DC could be applied to cover the continuous ventilation load that would be required by multiple rooftop units that serve the same general space.

It should be noted that the manufacturer offers a number of options for additional energy savings measures that were not laboratory tested and evaluated here. Ongoing investigation through various pilot field installations will evaluate the impact of variable speed fan controls, micro–channel heat exchangers, and demand controlled ventilation strategies. In summary, the laboratory evaluation inspires great confidence that the equipment provides compelling energy savings, and highlights a number of opportunities for still further improvement.

10. REFERENCES

Woolley, Modera. ***Speakman Hybrid Rooftop Unit Performance: Western Cooling Challenge Laboratory Test Results.*** Southern California Ediston, Design and Engineering Services. Report HT.10.SCE.232. 2011.

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

Western Cooling Efficiency Center. ***Western Cooling Challenge Program Requirements***. Online. http://wcec.ucdavis.edu/. 2012.

Woolley, J. Modera, M. ***Advancing Development of Hybrid Rooftop Packaged Air Conditioners: Test Protocol and Performance Criteria for the Western Cooling Challenge***. ASHRAE 2011-86098. ASHRAE Transactions, 2011, Vol. 117 Issue 1, p533-540. 2011.

Kozubal, E; Slayzak, S. ***Coolerado 5 Ton RTU Performance: Western Cooling Challenge Results***. National Renewable Energy Laboratory. Technical Report NREL/TP-5500-46524. November 2010.

APPENDIX A:
SUMMARY TABLE OF MEASUREMENTS AND RESULTS

TABLE 7: PERFORMANCE DATA FOR ALL TESTS

Test No.	Mode	TDB OSA (°F)	TWB OSA (°F)	TDB RA (°F)	TWB RA (°F)	TDB SA (°F)	TWB SA (°F)	Supply Airflow (scfm)	OSAF	ESP ("WC)	Cndsr. Airflow (SCFM)	Total Power (kW)	Capacity (kbtu/hr)	Sens. Rm. Cap. (kbtu/hr)	Vent Coil Cap. (kbtu/hr)	COP	Sensible Room COP
IV	Indirect & DX Stage 2	104.9	72.9	78.0	64.0	55.3	53.3	6013	0.43	0.45	12538	16.48	277.0	147.1	58.9	4.93	2.62
V	Indirect & DX Stage 2	89.8	66.4	78.0	64.0	52.2	50.5	6077	0.43	0.46	12538	15.38	259.5	169.3	39.4	4.94	3.23
VI	Indirect DX Stage 1	89.6	67.1	78.0	64.0	57.7	55.2	6027	0.43	0.47	9452	11.19	188.3	132.3	37.2	4.93	3.46
VII	Indirect Evap. Only	89.5	66.5	78.0	64.0	78.0	63.9	5841	0.43	0.48	6330	2.36	23.1	0.1	35.5	2.86	0.02
1	Indirect & DX Stage 2	78.0	63.0	77.8	64.0	51.0	49.6	6116	0.43	0.49	12538	14.45	244.8	177.2	24.4	4.97	3.60
2	Indirect & DX Stage 2	95.0	74.7	78.0	64.0	57.6	56.1	5983	0.43	0.44	12538	16.47	250.6	131.6	34.2	4.46	2.34
3	Indirect & DX Stage 2	81.6	66.3	77.8	64.0	52.3	50.9	6054	0.43	0.44	12379	14.71	252.4	166.6	25.8	5.03	3.32
4	Indirect & DX Stage 2	82.0	73.0	77.8	64.0	55.9	54.8	6012	0.43	0.45	12538	15.57	256.7	142.1	11.5	4.83	2.67
6	Indirect & DX Stage 2	68.6	60.6	77.8	64.0	50.1	48.9	6112	0.43	0.47	12538	13.82	235.5	183.1	11.9	4.99	3.88
7	Indirect & DX Stage 2	81.5	66.3	77.8	64.0	53.9	52.1	5394	1.00	NA	12379	14.26	232.3	139.2	32.4	4.77	2.86
8	Indirect & DX Stage 2	78.0	63.2	77.8	64.0	51.4	49.2	5485	1.00	NA	12538	13.88	217.7	156.1	31.1	4.60	3.29
9	Indirect & DX Stage 2	67.7	59.9	77.8	64.0	47.7	46.6	5453	1.00	NA	12538	13.19	196.0	177.3	13.5	4.36	3.94
10	Indirect & DX Stage 2	69.3	61.0	77.8	64.0	48.1	47.4	5485	1.00	NA	12538	13.34	205.9	175.7	15.3	4.52	3.86
11	Indirect & DX Stage 1	77.6	63.1	77.8	64.0	55.4	53.8	6013	0.43	0.43	9419	10.57	175.5	145.7	21.6	4.86	4.04
12	Indirect & DX Stage 1	105.0	73.0	78.0	64.0	61.8	58.0	6003	0.43	0.48	9419	12.05	198.0	105.2	52.9	4.82	2.56
13	Indirect & DX Stage 1	95.0	74.7	78.0	64.0	62.1	60.1	5970	0.43	0.47	9419	11.79	177.9	102.7	32.7	4.42	2.55
14	Indirect & DX Stage 1	81.5	66.3	77.8	64.0	56.9	55.3	6027	0.43	0.47	9419	10.76	179.2	135.7	24.3	4.88	3.70
15	Indirect & DX Stage 1	82.0	73.0	77.8	63.9	60.3	59.1	5976	0.43	0.47	9419	11.25	179.1	113.2	11.9	4.67	2.95
16	Indirect & DX Stage 1	68.1	59.8	77.8	64.0	53.8	52.6	6046	0.43	0.47	9419	10.05	169.6	156.6	12.4	4.94	4.57
17	Indirect & DX Stage 1	66.5	59.0	77.8	64.0	53.6	52.3	6090	0.43	0.48	9419	10.00	168.3	159.4	10.5	4.93	4.67
18	Indirect & DX Stage 1	81.5	66.3	77.8	64.0	59.1	55.9	5434	1.00	NA	9452	10.36	177.2	109.7	29.4	5.01	3.10
19	Indirect & DX Stage 1	77.6	63.0	77.8	64.0	56.2	52.7	5485	1.00	NA	9452	10.16	164.1	128.2	27.0	4.73	3.70
20	Indirect & DX Stage 1	69.0	60.5	77.8	64.0	52.8	51.2	5501	1.00	NA	9419	9.67	144.6	148.6	16.9	4.38	4.50
21	Indirect & DX Stage 1	67.1	59.1	77.8	64.0	51.4	50.0	5556	1.00	NA	9419	9.58	140.5	158.2	14.3	4.30	4.84
22	Indirect Evaporative Only	77.6	61.8	77.8	64.0	73.7	60.6	5363	1.00	NA	6330	2.21	19.9	23.9	26.9	2.64	3.18
23	Indirect Evap. Only	105.0	73.0	78.0	64.0	92.5	69.4	5086	1.00	NA	6330	2.12	70.1	-79.7	66.8	9.67	-10.99
24	Indirect Evap. Only	95.0	75.1	77.6	64.9	86.9	73.0	5189	1.00	NA	6330	2.16	46.6	-52.1	36.3	6.33	-7.08
25	Indirect Evap. Only	81.5	66.3	77.6	64.0	76.8	64.7	5283	1.00	NA	6330	2.20	28.8	4.5	28.2	3.84	0.60
26	Indirect Evap. Only	82.0	73.0	77.8	63.9	79.6	72.3	5158	1.00	NA	6330	2.15	14.0	-9.8	14.4	1.91	-1.34
27	Indirect Evap. Only	67.5	59.7	77.8	64.0	67.1	59.7	5283	1.00	NA	6330	2.19	1.4	61.1	11.6	0.18	8.19
28	Indirect Evap. Only	65.0	58.1	77.8	64.0	64.0	57.8	5283	1.00	NA	6330	2.19	5.2	78.9	7.0	0.69	10.53

APPENDIX B: PSYCHROMETRIC CHARTS

Figure B1: Psychrometric Chart test Iv

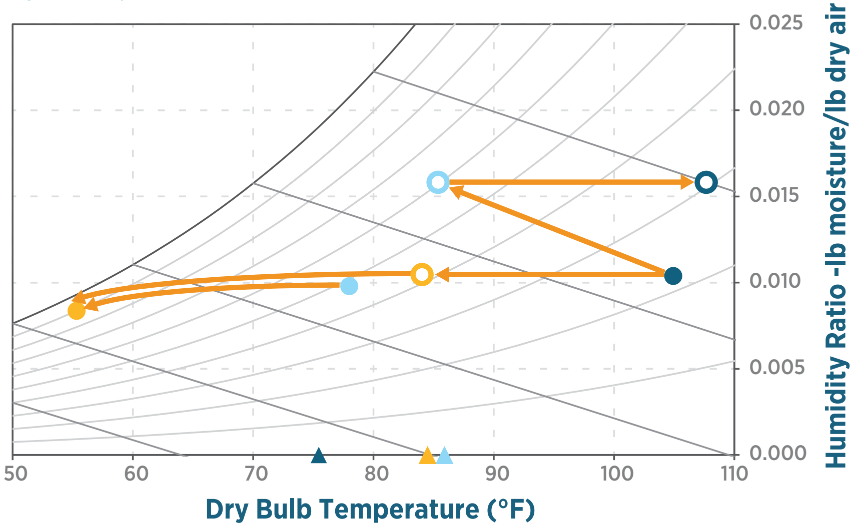


Figure B1: Psychrometric chart test v

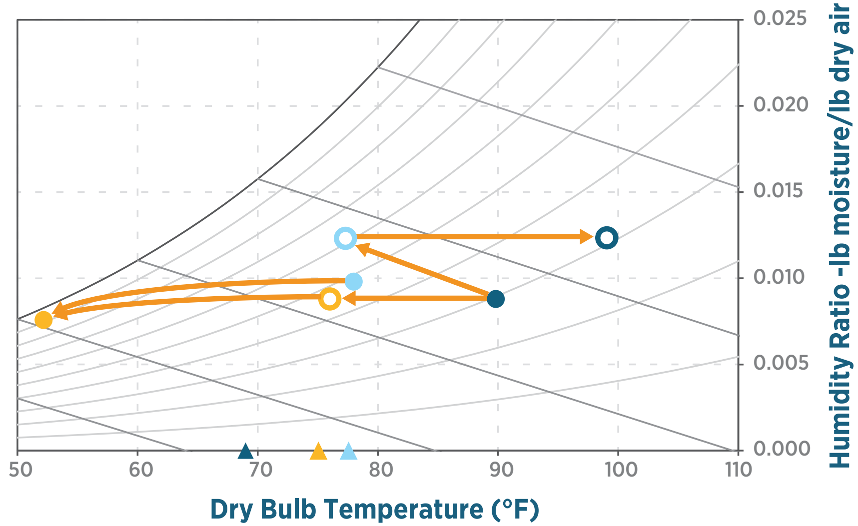


Figure B1: PSYCHROMETRIC chart test vi

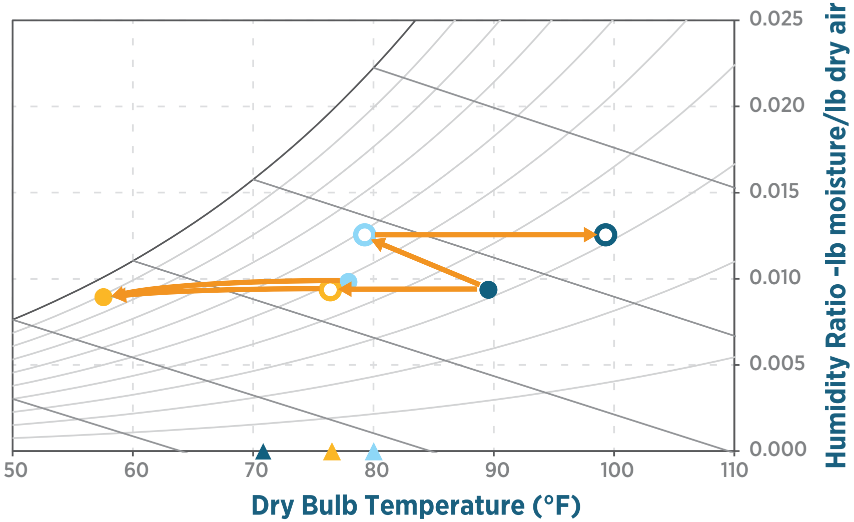
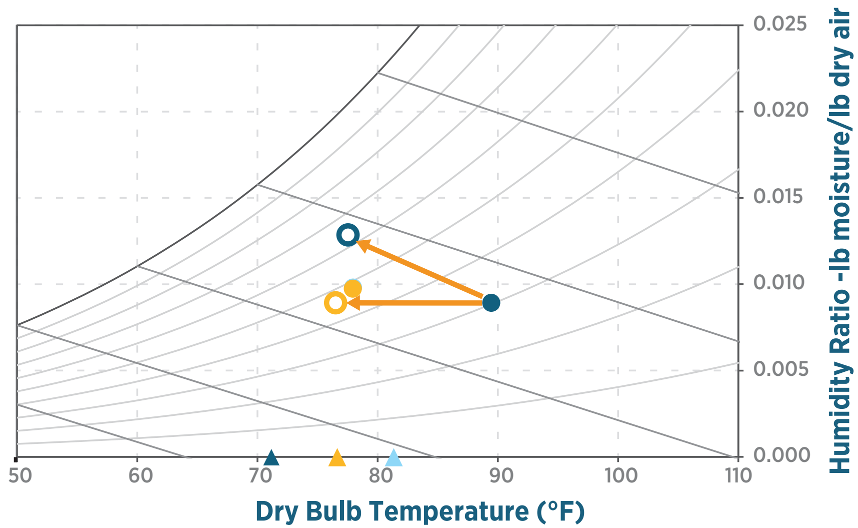


Figure B1: Psychrometric chart test vII



LEGEND

- ▲ vent coil inlet water
- ▲ vent coil outlet water
- ▲ condenser liquid
- ● outside air
- ● return air
- ● supply air
- ○ condenser inlet
- ○ condenser outlet
- ○ vent coil outlet

APPENDIX B: PSYCHROMETRIC CHARTS

Figure B1: Psychrometric Chart test 1

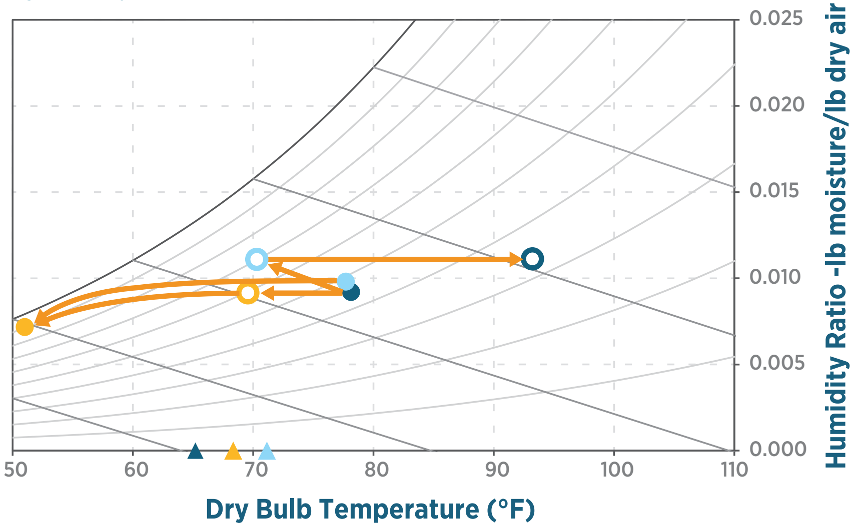


Figure B1: Psychrometric chart test 2

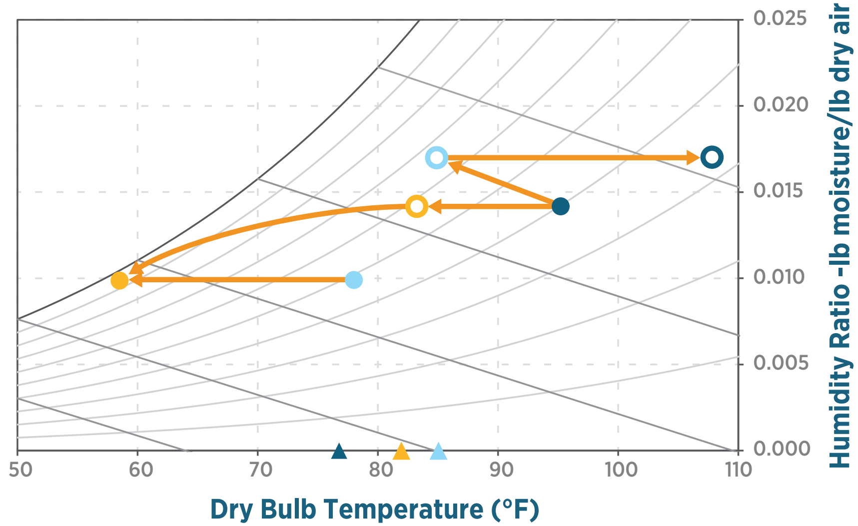


Figure B1: PSYCHROMETRIC chart test 3

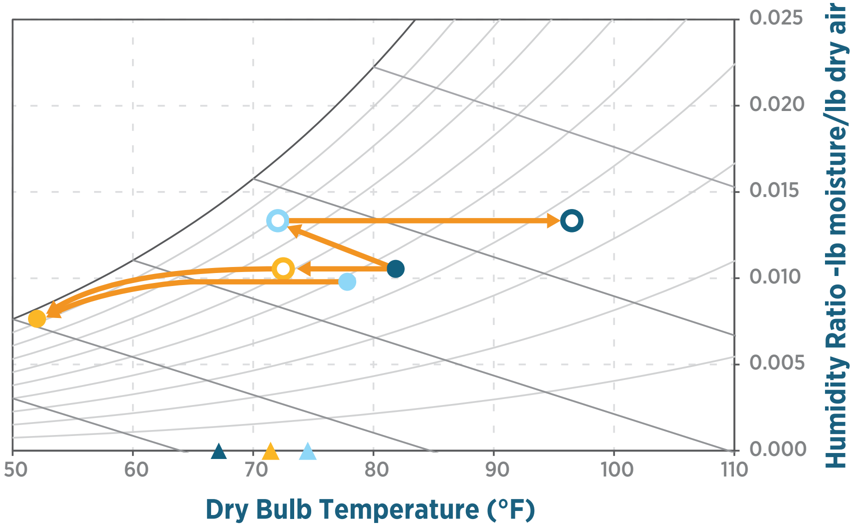
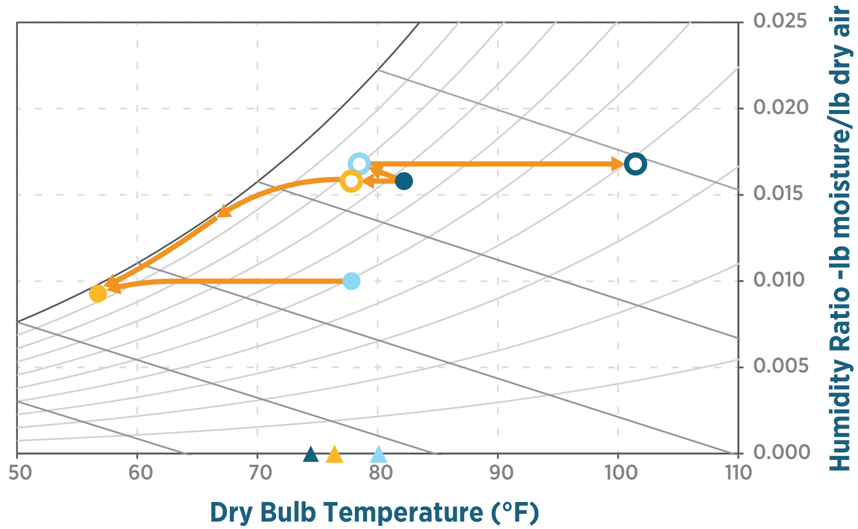


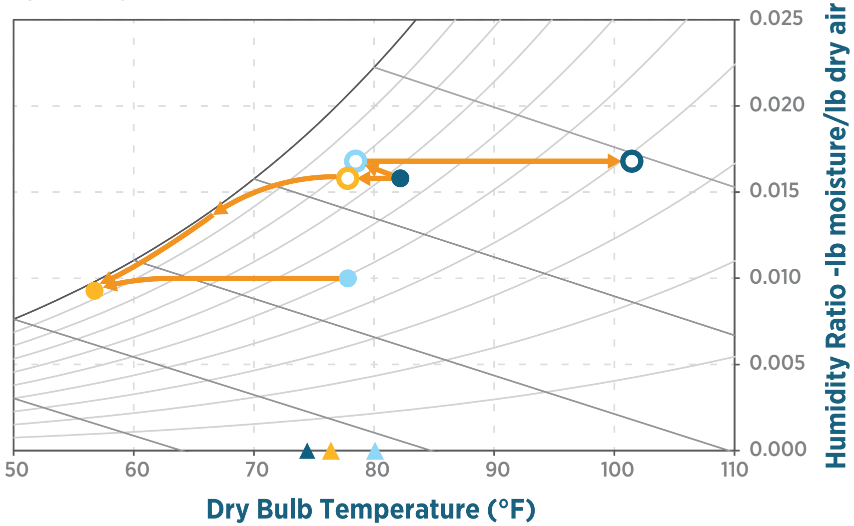
Figure B1: Psychrometric chart test 4



LEGEND

- ▲ vent coil inlet water
- ▲ vent coil outlet water
- ▲ condenser liquid
- ● outside air
- ● return air
- ● supply air
- ○ condenser inlet
- ○ condenser outlet
- ○ vent coil outlet

Figure B1: Psychrometric Chart test 5



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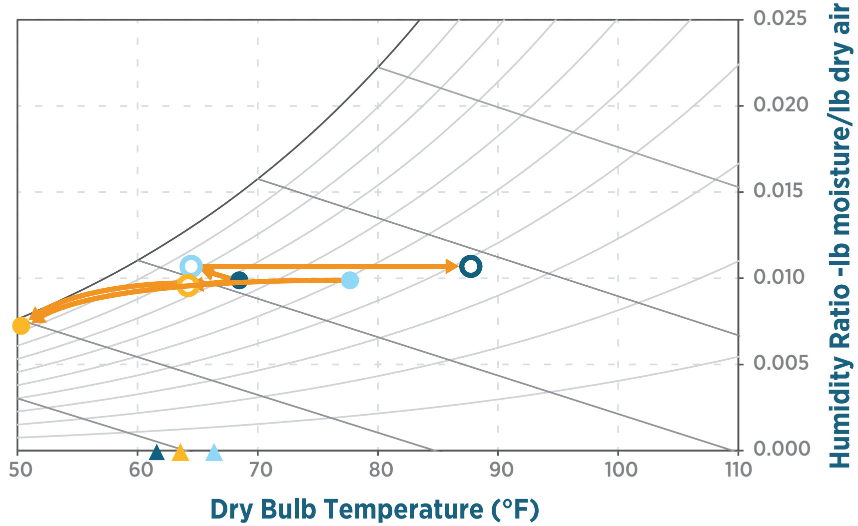


Figure B1: PSYCHROMETRIC chart test 7

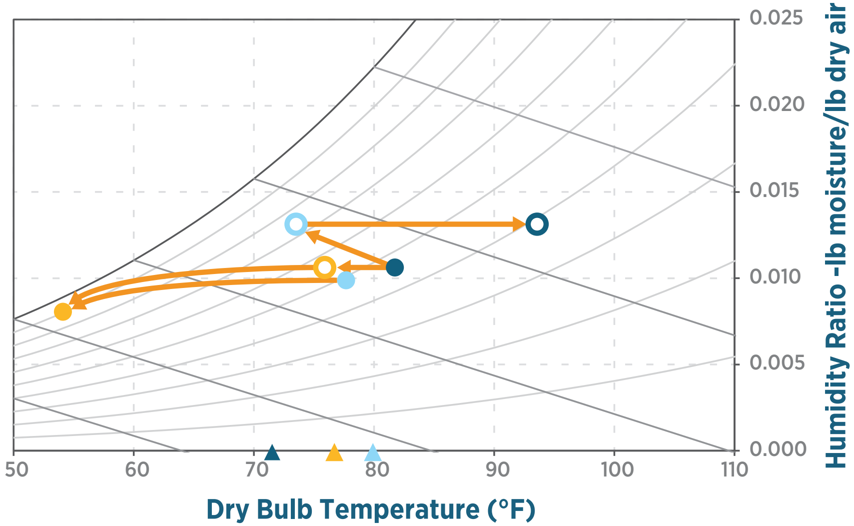
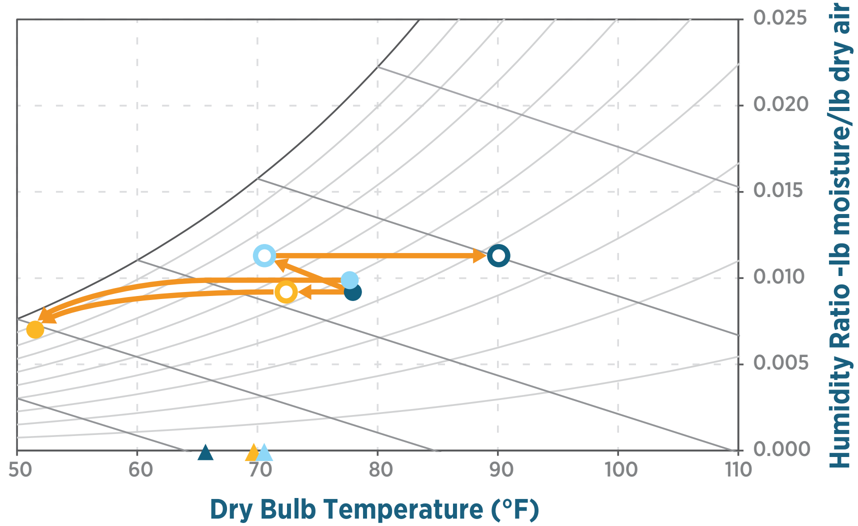


Figure B1: Psychrometric chart test 8



LEGEND

- vent coil inlet water
- vent coil outlet water
- condenser liquid
- outside air
- return air
- supply air
- condenser inlet
- condenser outlet
- vent coil outlet

APPENDIX B: PSYCHROMETRIC CHARTS

Figure B1: Psychrometric Chart test 9

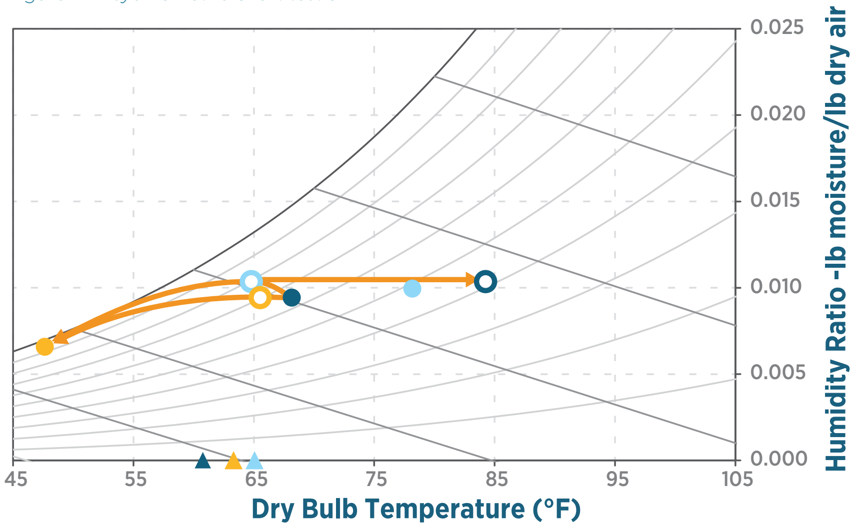


Figure B1: Psychrometric chart test 10

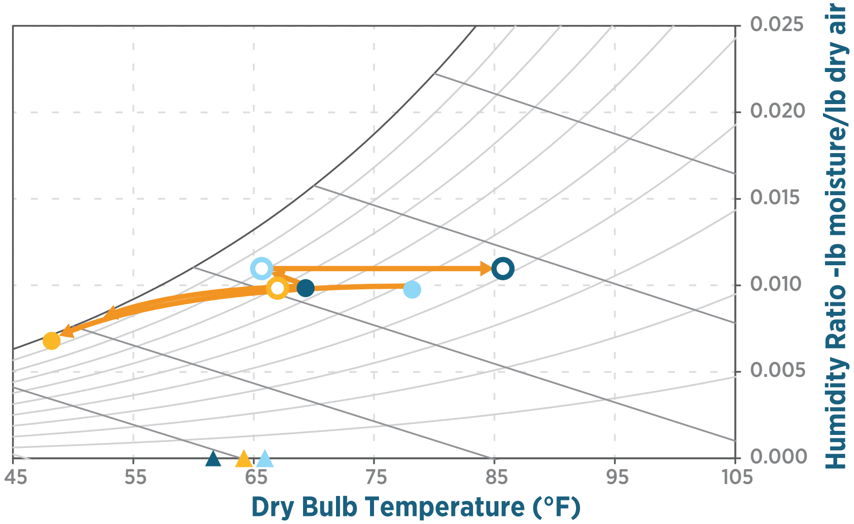


Figure B1: PSYCHROMETRIC chart test 11

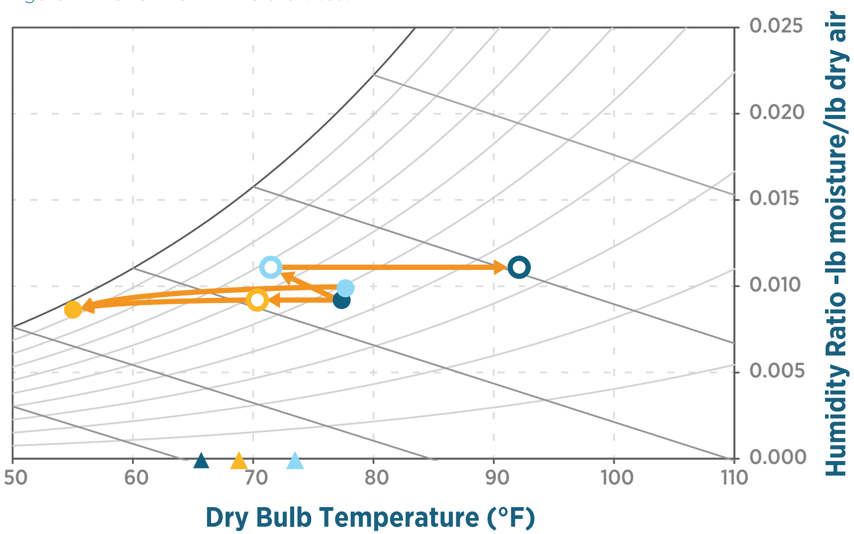
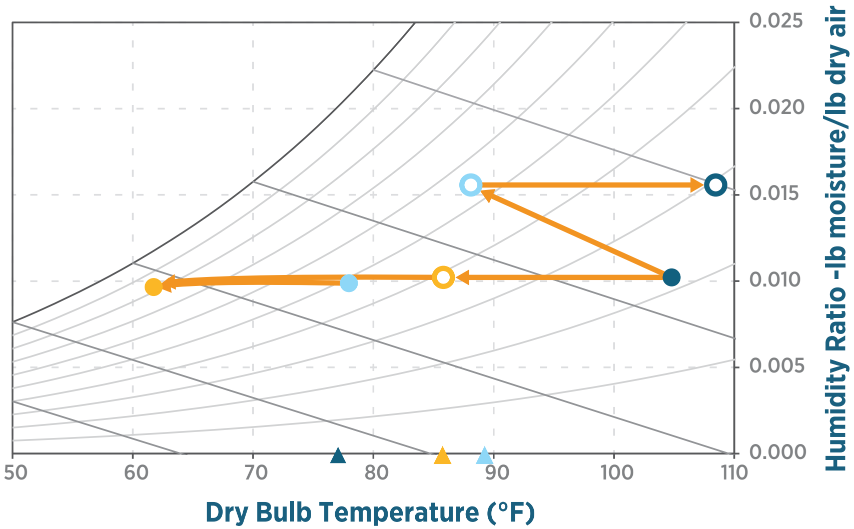


Figure B1: Psychrometric chart test 12

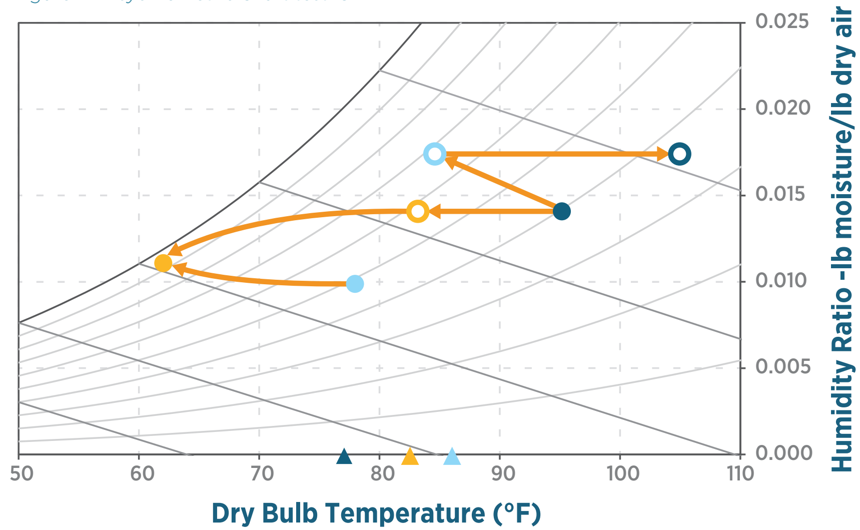


LEGEND

- ▲ vent coil inlet water
- ▲ vent coil outlet water
- ▲ condenser liquid
- ● outside air
- ● return air
- ● supply air
- ○ condenser inlet
- ○ condenser outlet
- ○ vent coil outlet

APPENDIX B: PSYCHROMETRIC CHARTS

Figure B1: Psychrometric Chart test 13



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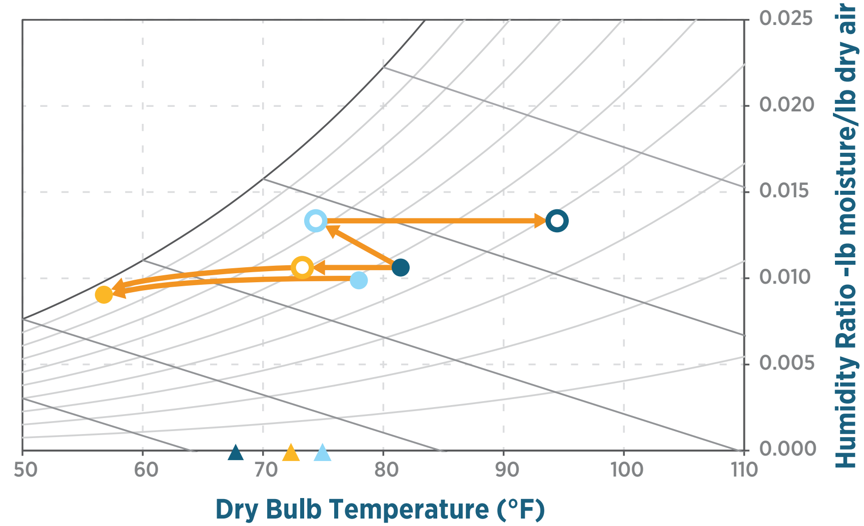


Figure B1: PSYCHROMETRIC chart test 15

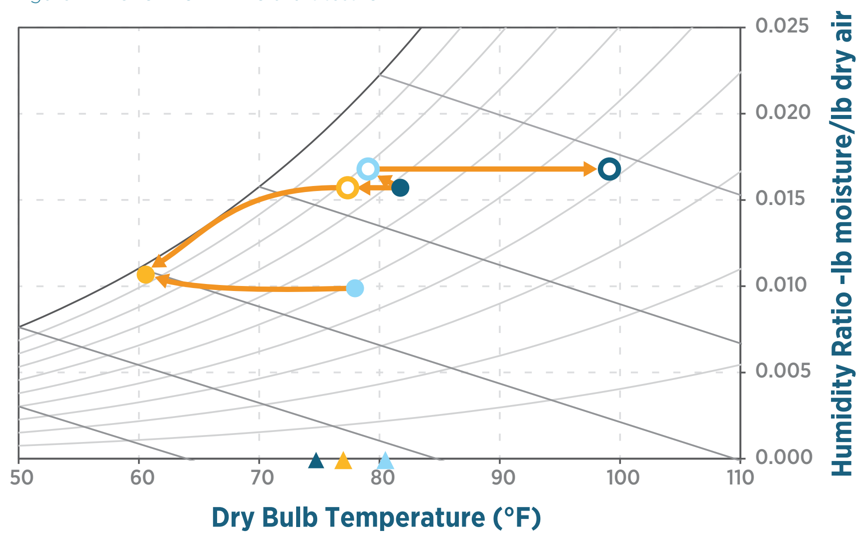
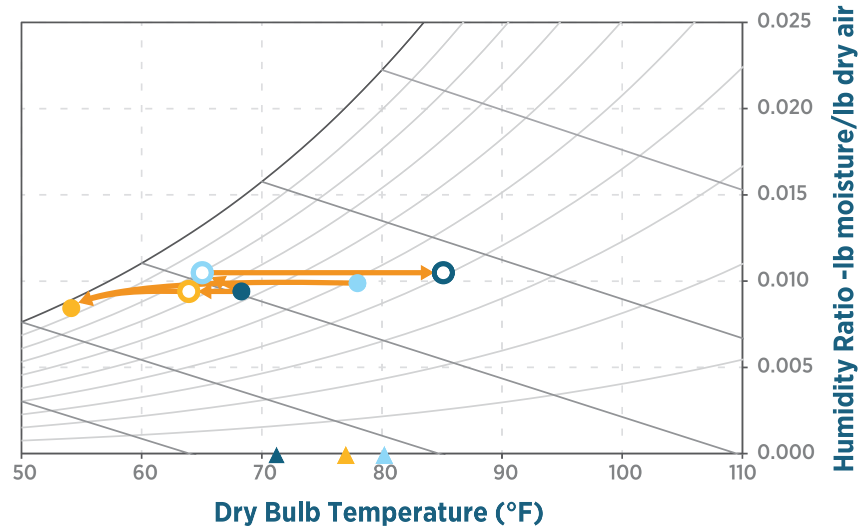


Figure B1: Psychrometric chart test 16



APPENDIX B: PSYCHROMETRIC CHARTS

Figure B1: Psychrometric Chart test 17

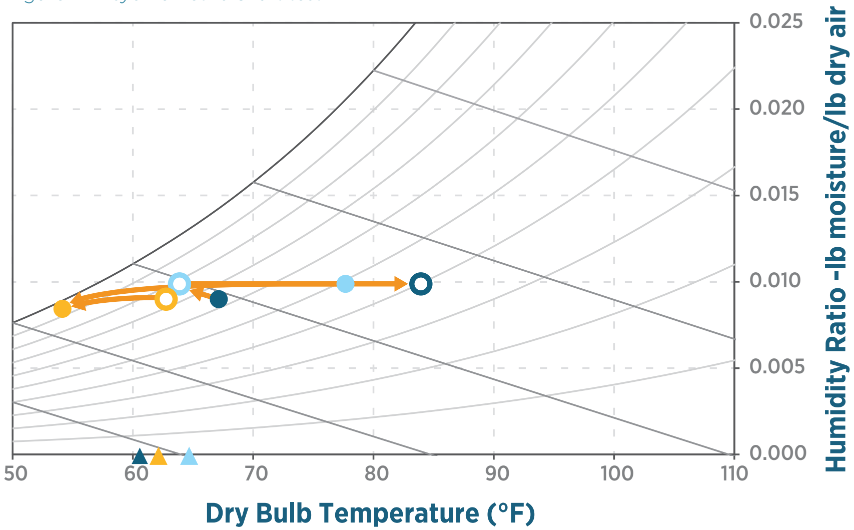


Figure B1: Psychrometric chart test 18

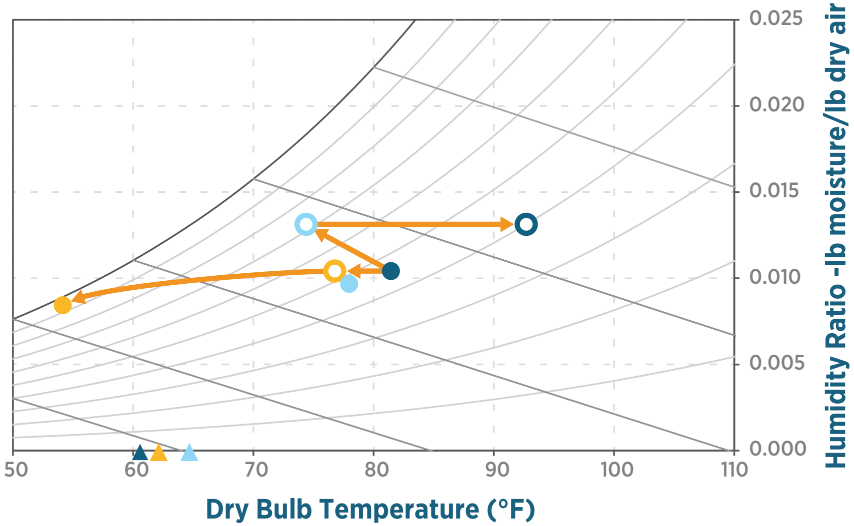


Figure B1: PSYCHROMETRIC chart test 19

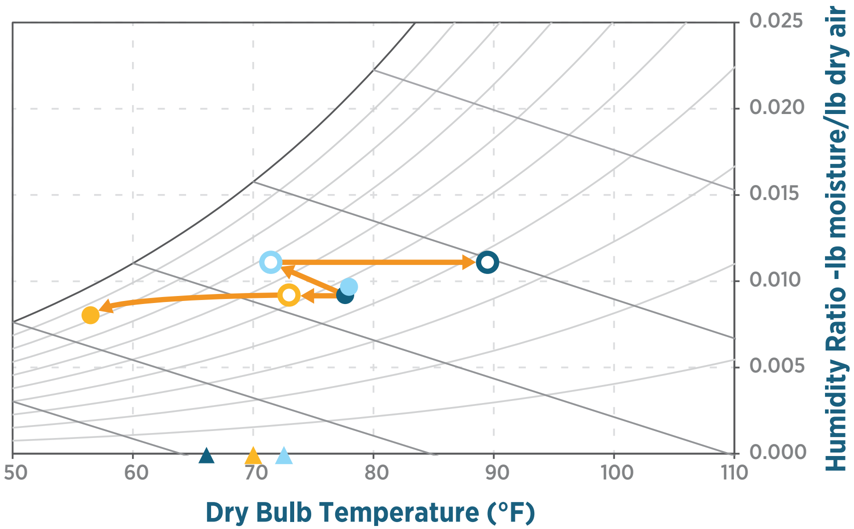
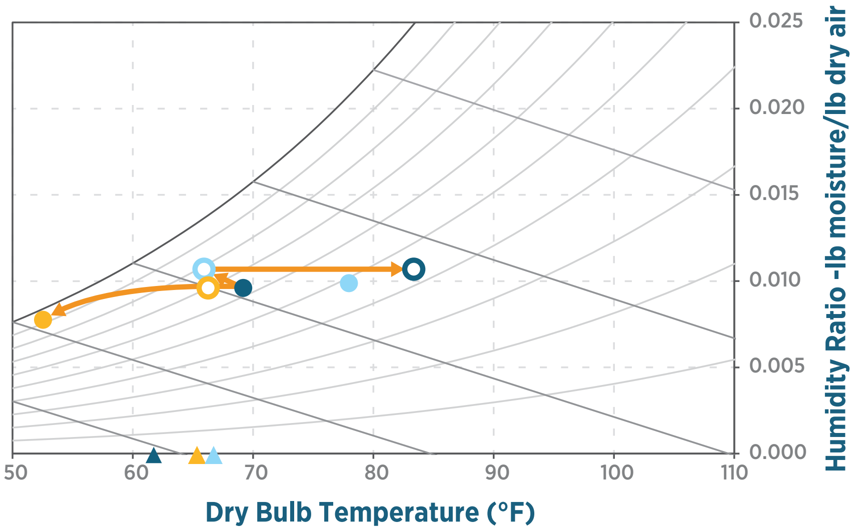


Figure B1: Psychrometric chart test 20



LEGEND

- ▲ vent coil inlet water
- ▲ vent coil outlet water
- ▲ condenser liquid
- ● outside air
- ● return air
- ● supply air
- ○ condenser inlet
- ○ condenser outlet
- ○ vent coil outlet

Figure B1: Psychrometric Chart test 21

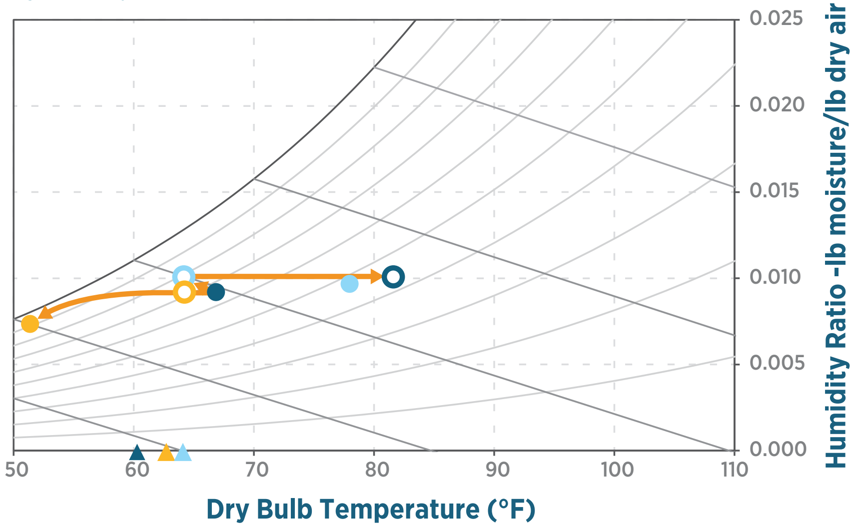


Figure B1: Psychrometric chart test 22

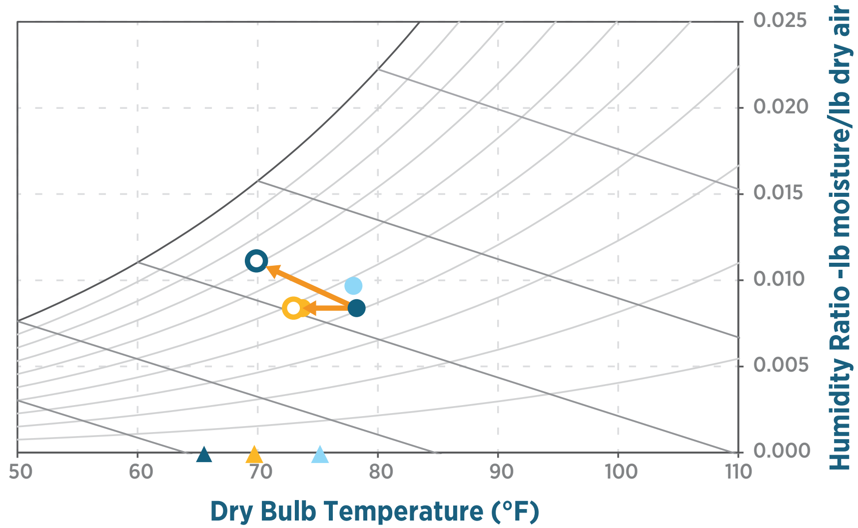


Figure B1: PSYCHROMETRIC chart test 23

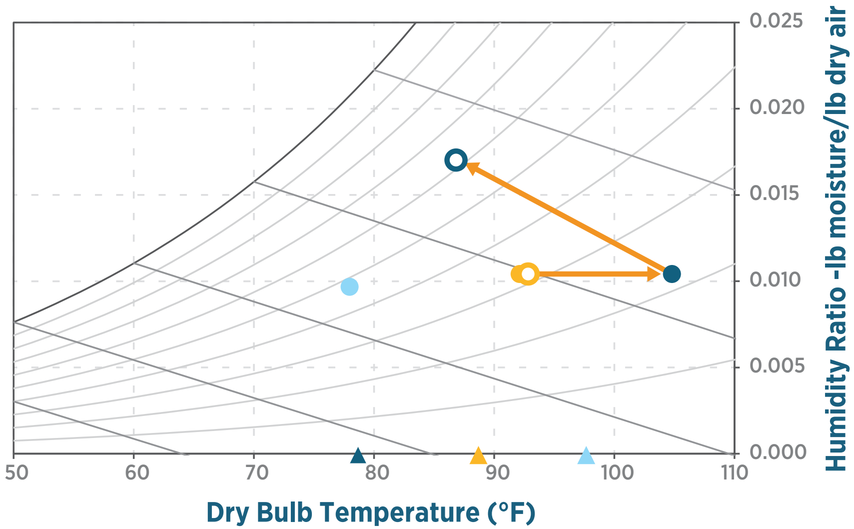
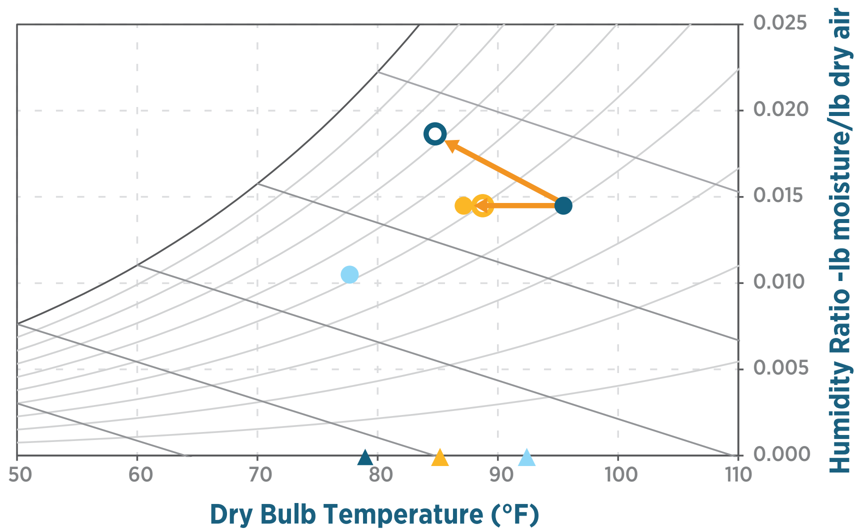


Figure B1: Psychrometric chart test 24



LEGEND

- ▲ vent coil inlet water
- ▲ vent coil outlet water
- ▲ condenser liquid
- ● outside air
- ● return air
- ● supply air
- ○ condenser inlet
- ○ condenser outlet
- ○ vent coil outlet

Figure B1: Psychrometric Chart test 25

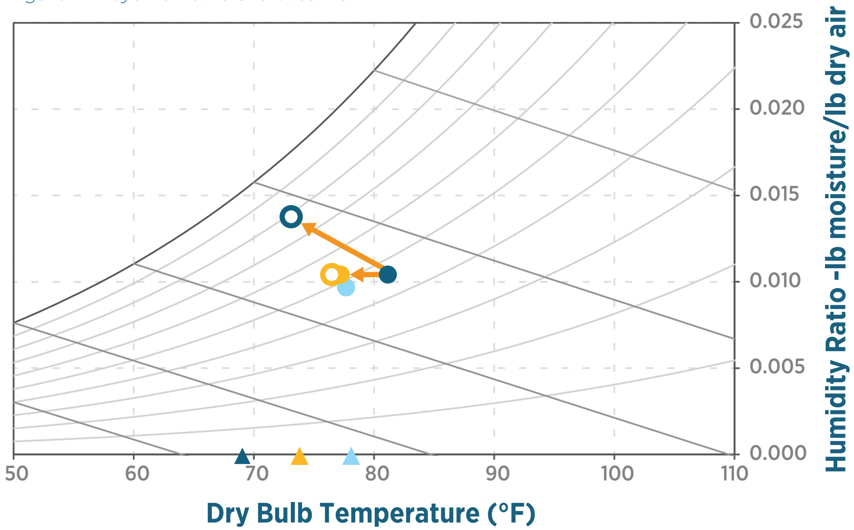


Figure B1: Psychrometric chart test 26

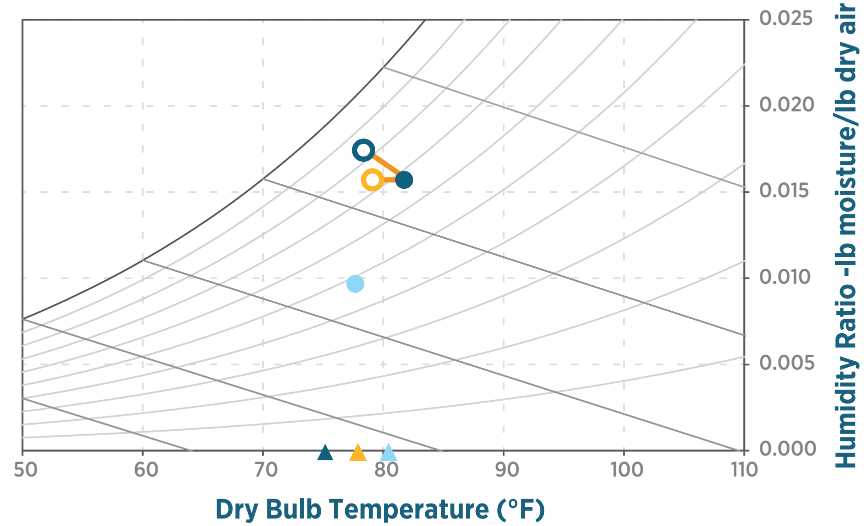


Figure B1: PSYCHROMETRIC chart test 27

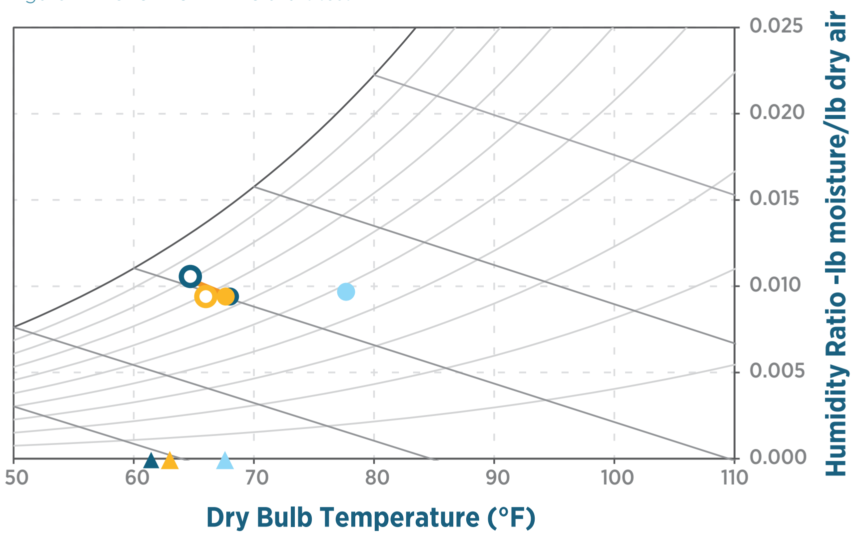
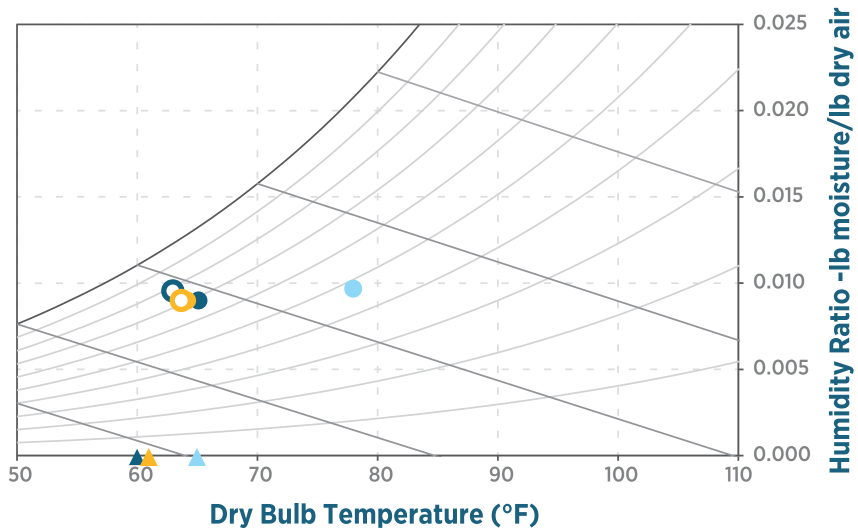


Figure B1: Psychrometric chart test 28



LEGEND

- vent coil inlet water
- vent coil outlet water
- condenser liquid
- outside air
- return air
- supply air
- condenser inlet
- condenser outlet
- vent coil outlet

APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST IV

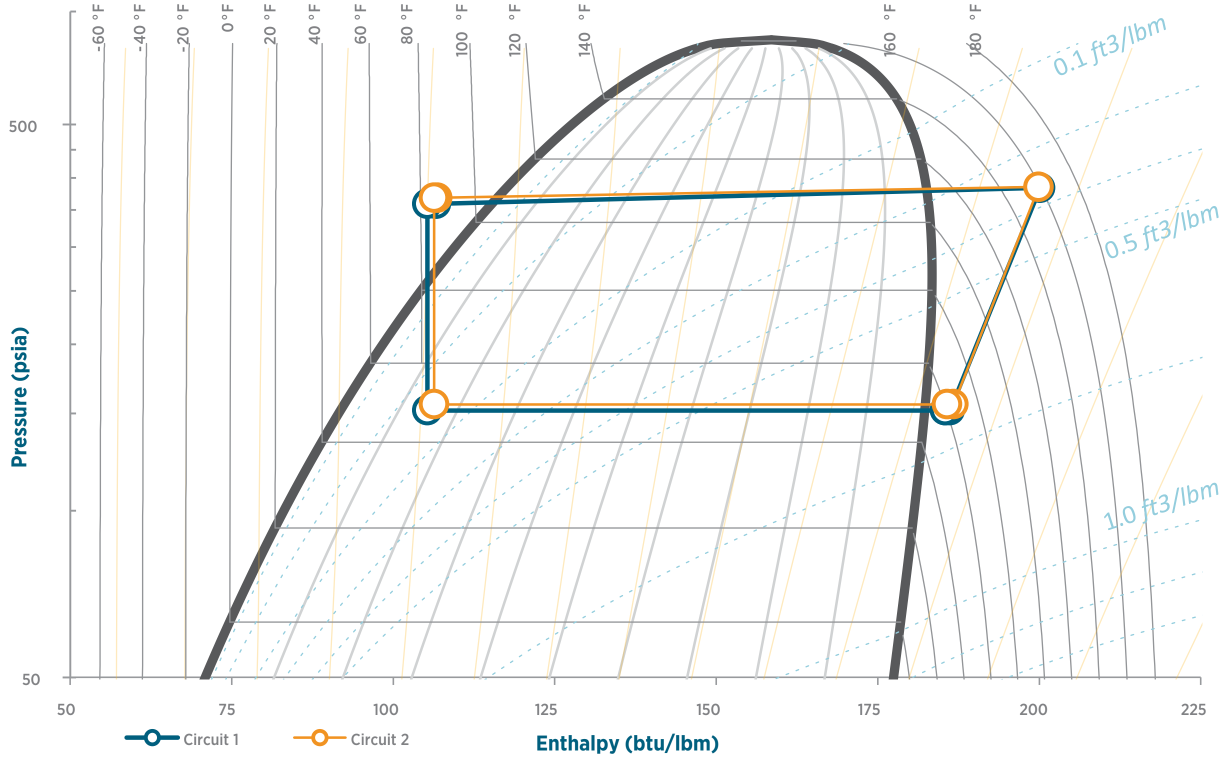
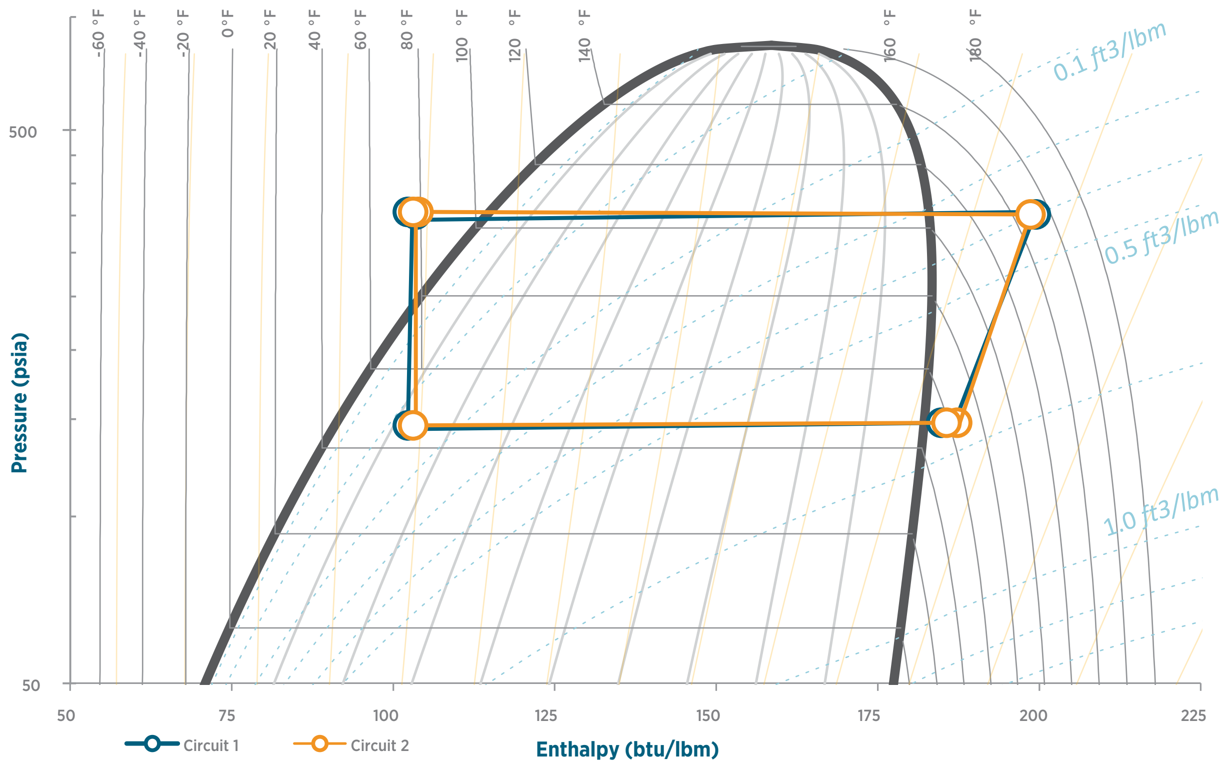


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST V



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST VI

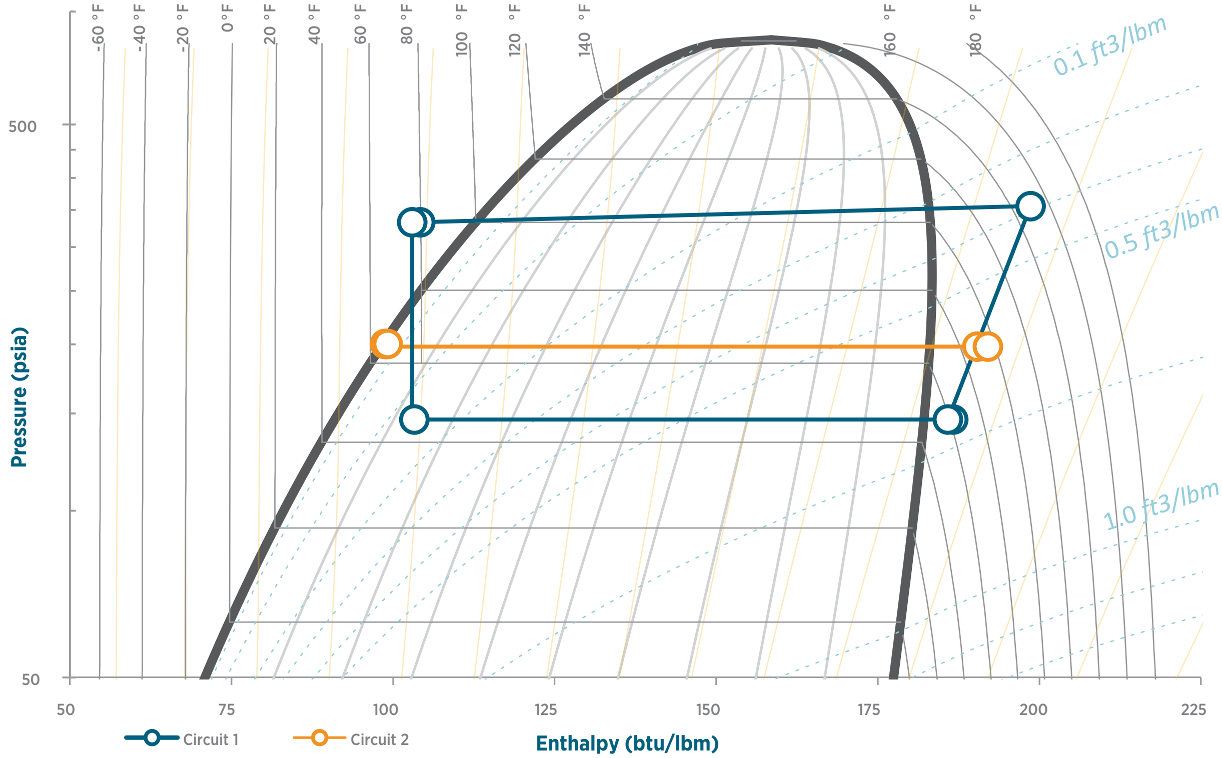
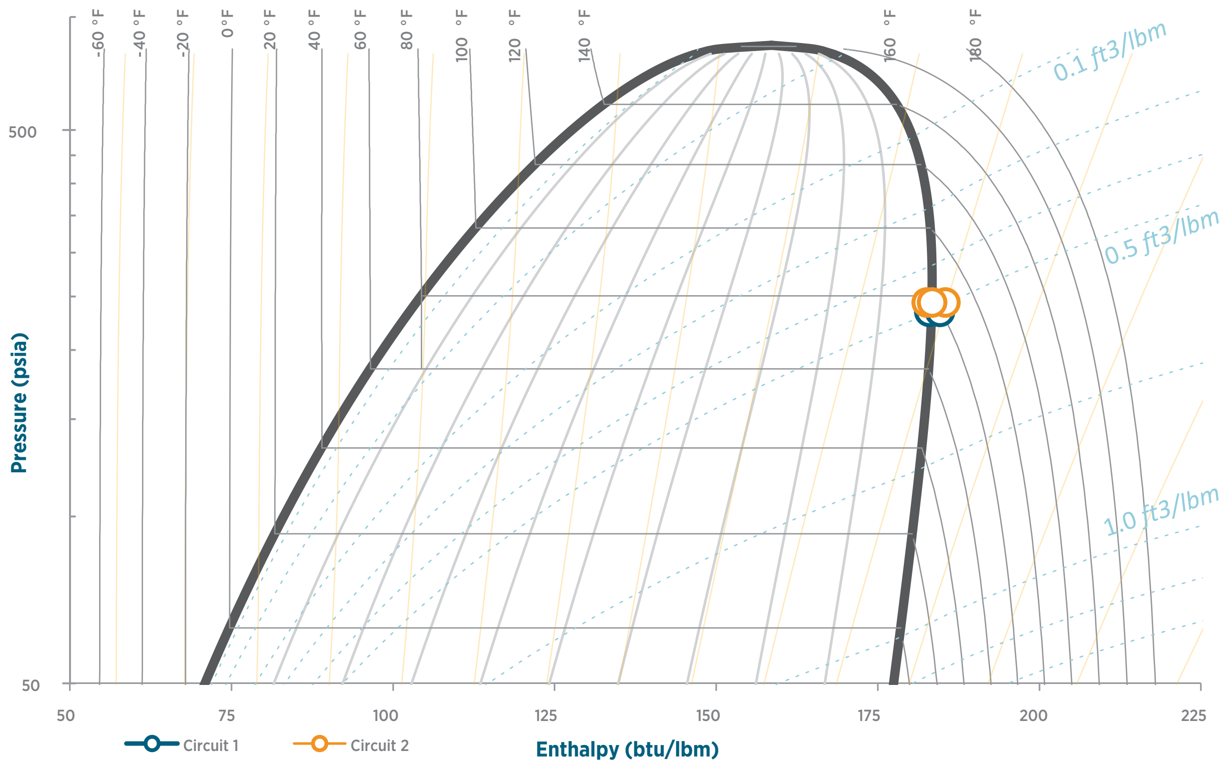


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST VII



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 1

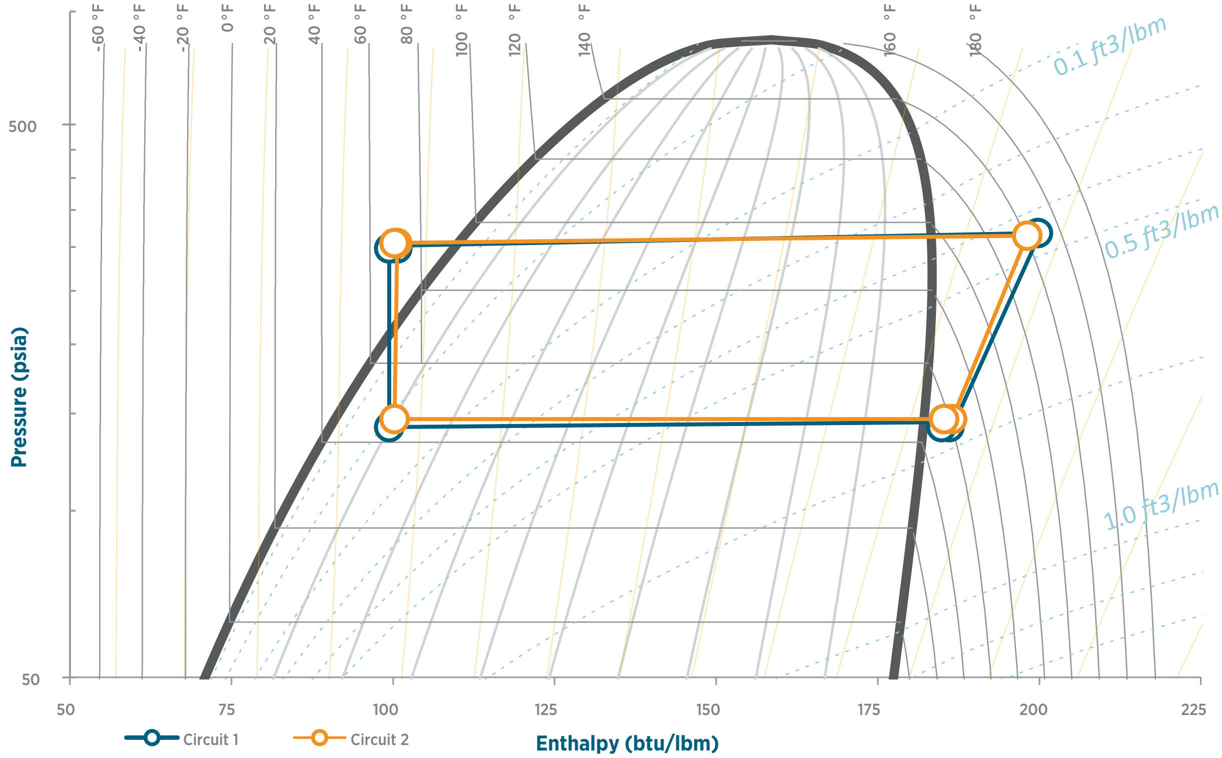
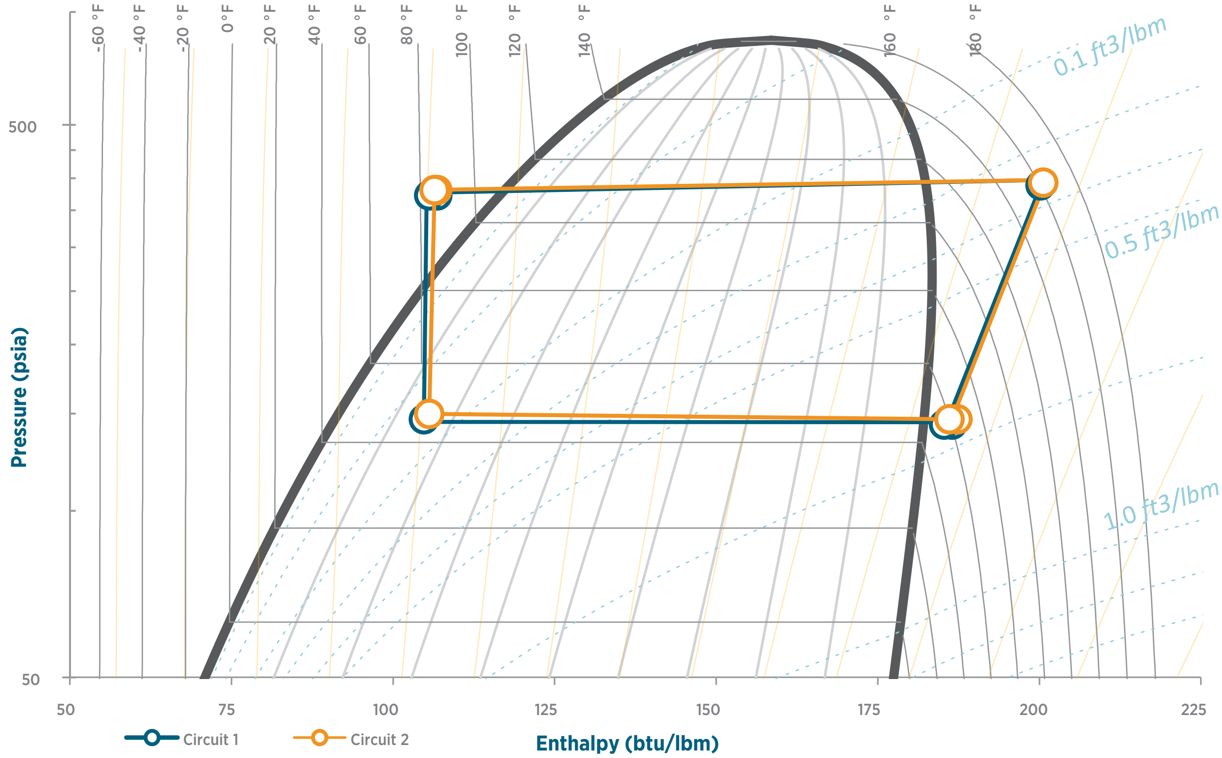


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 2



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 3

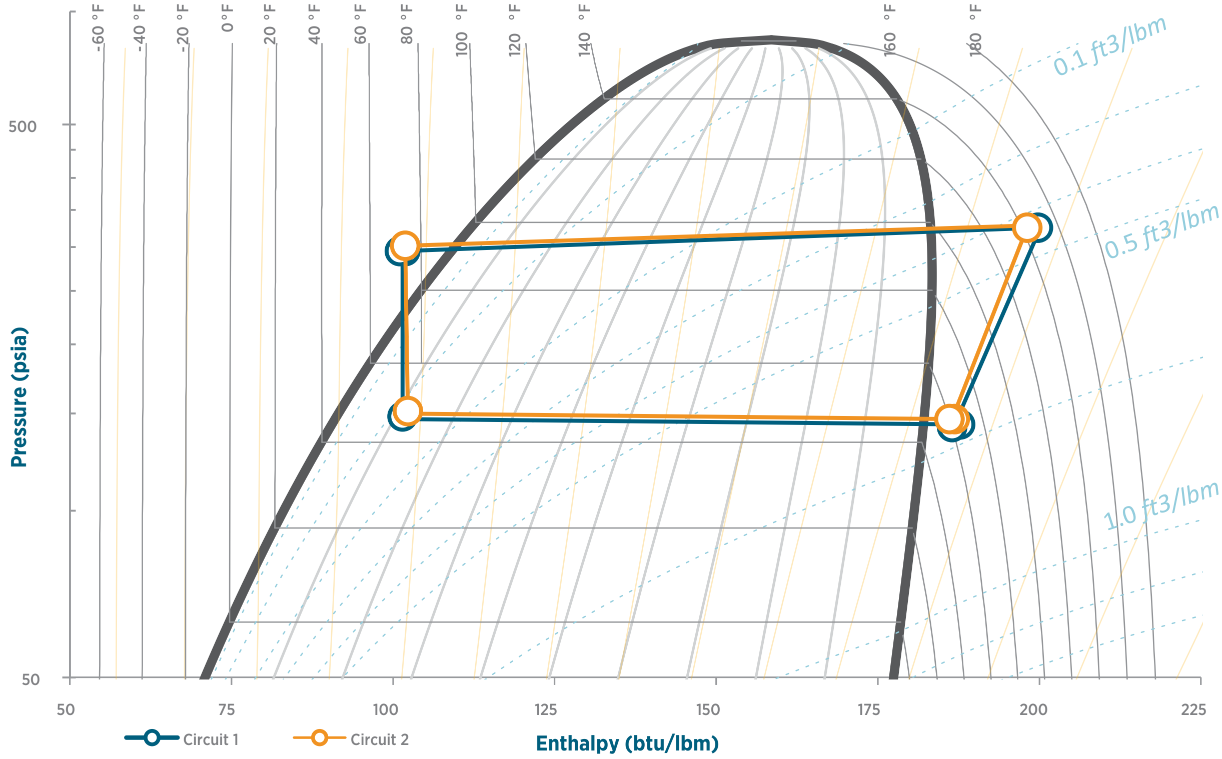
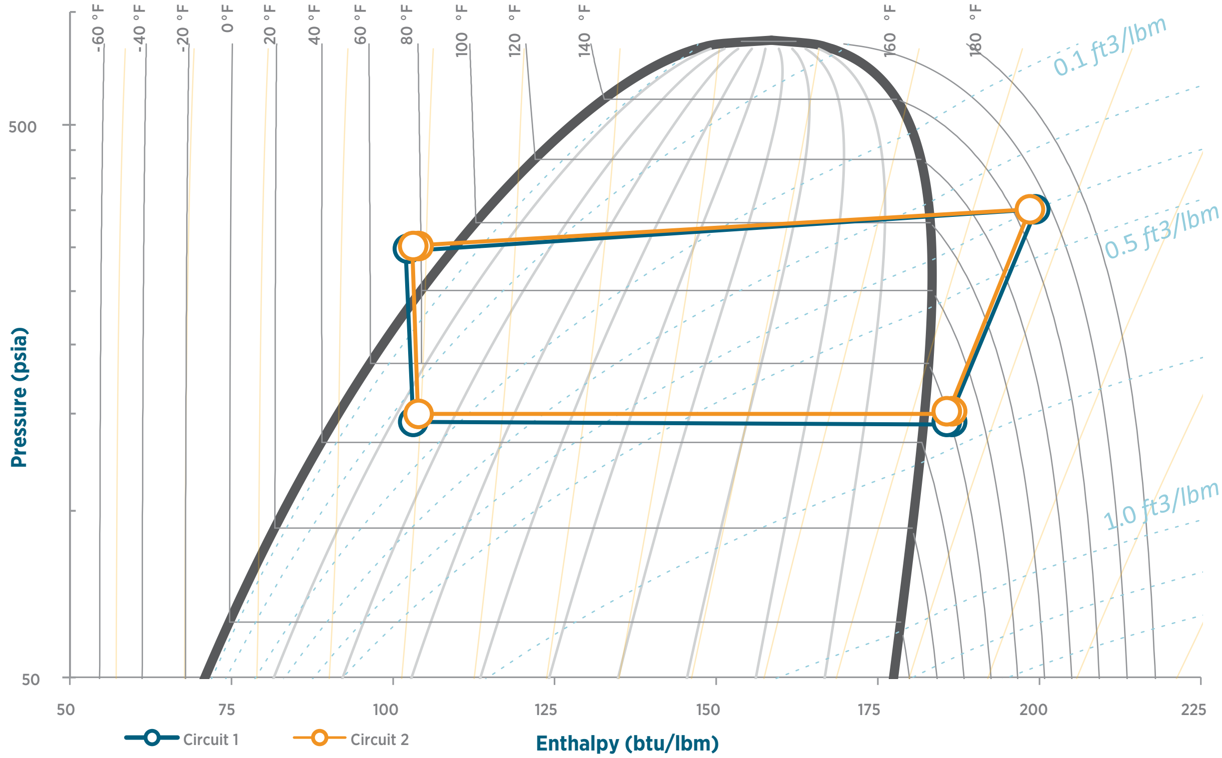


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 4



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 6

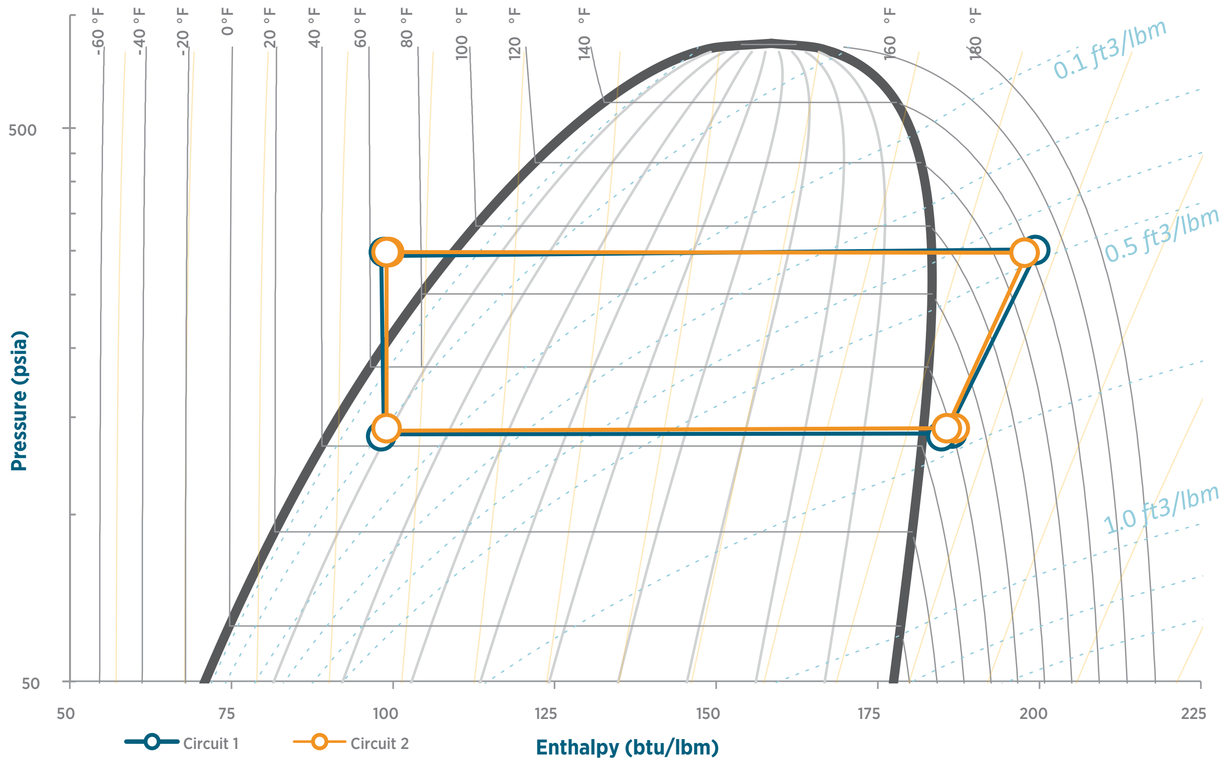
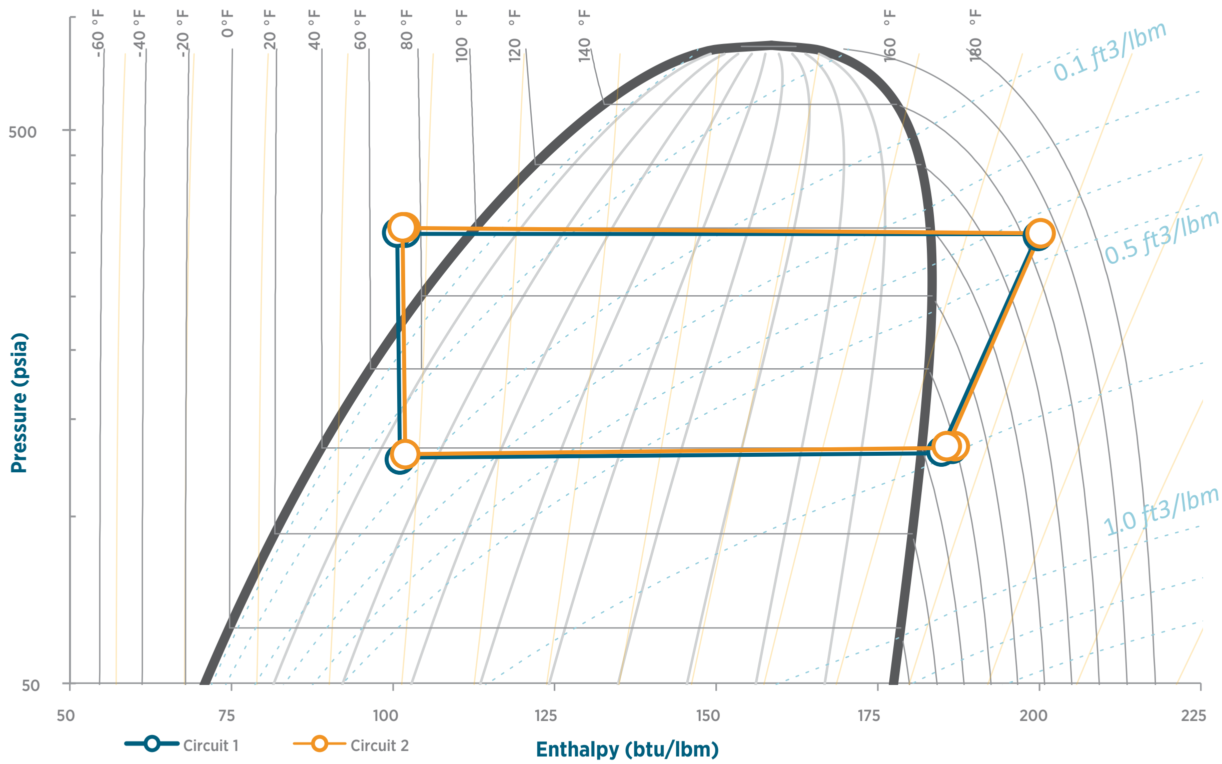


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 7



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 8

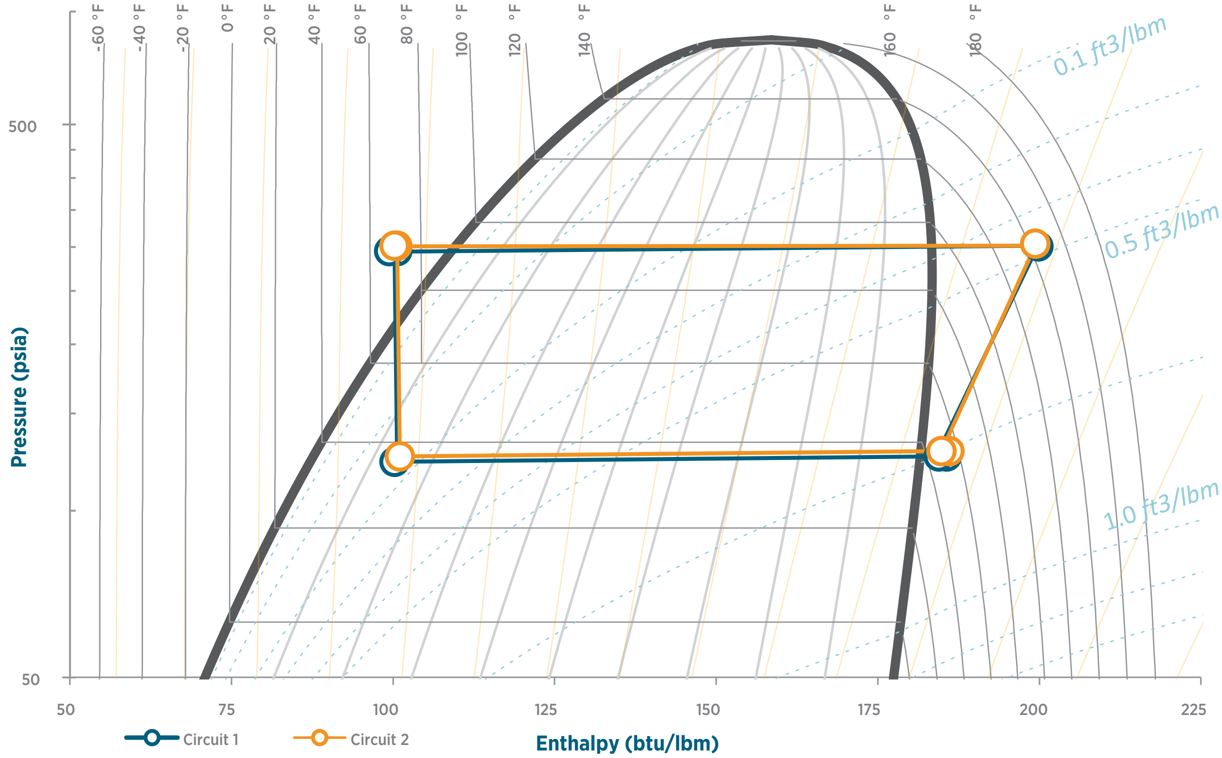
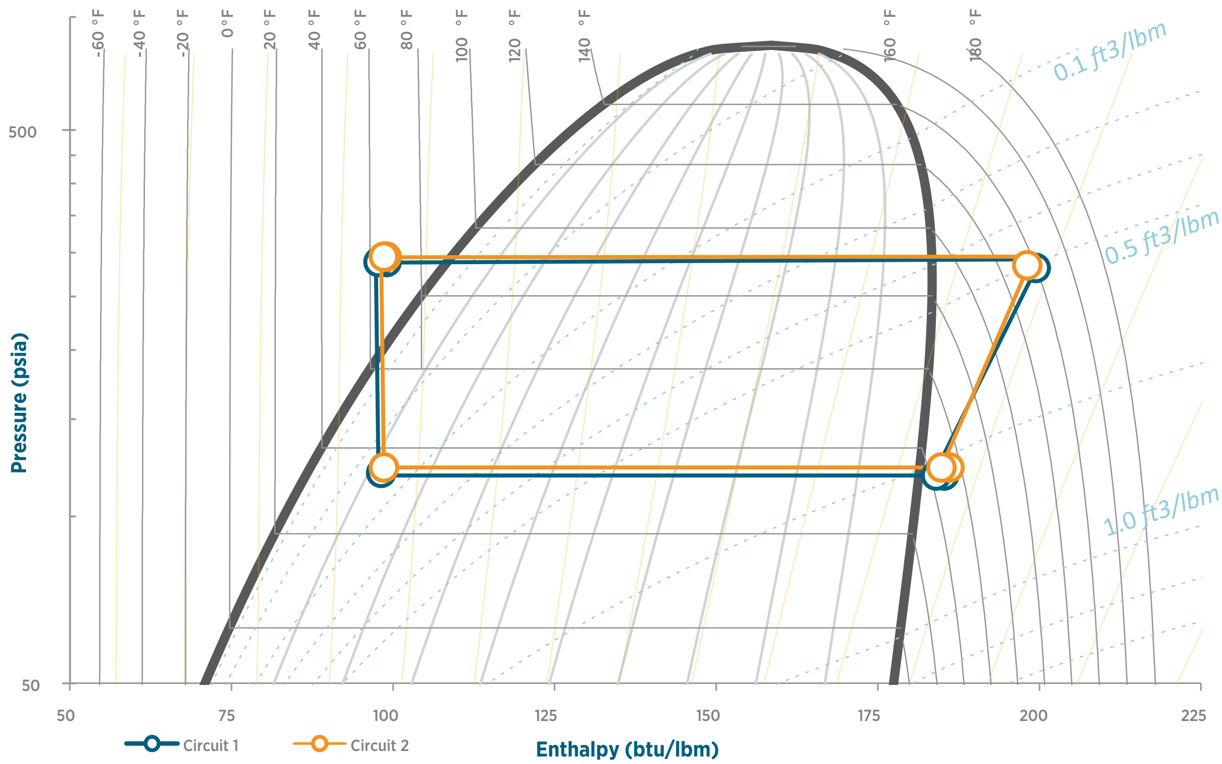


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 9



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 10

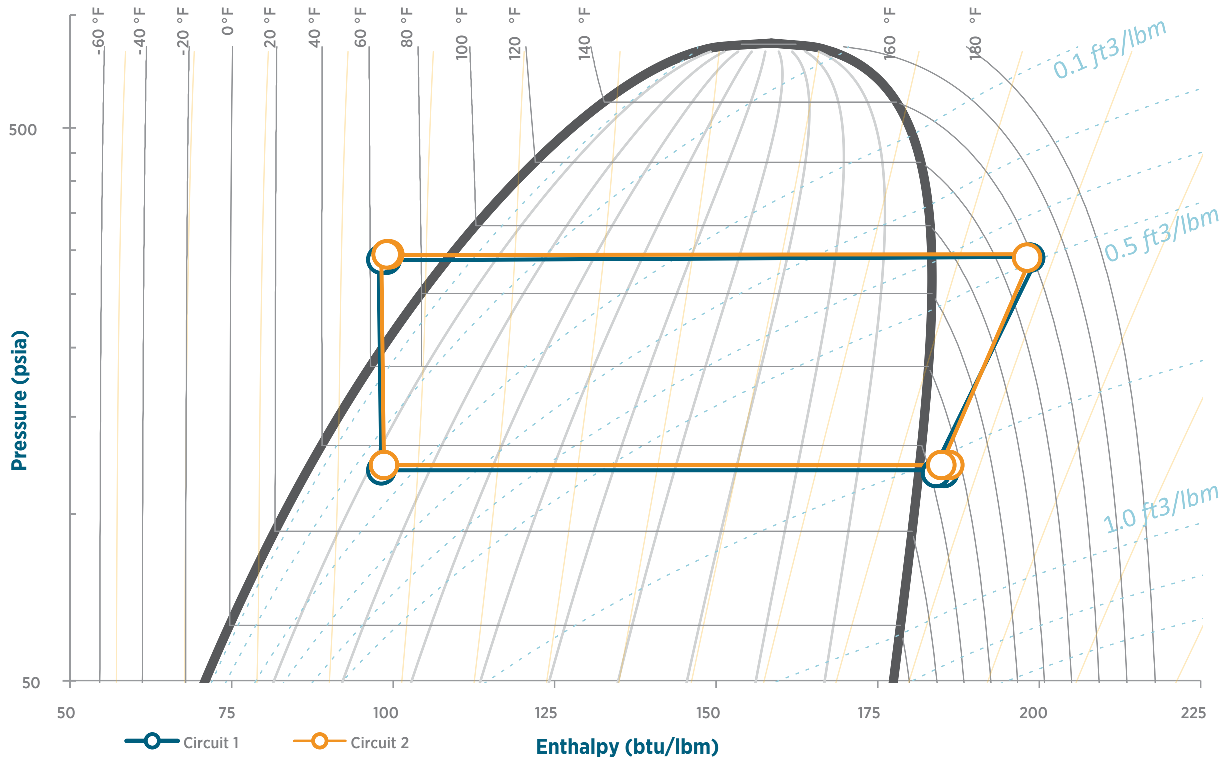
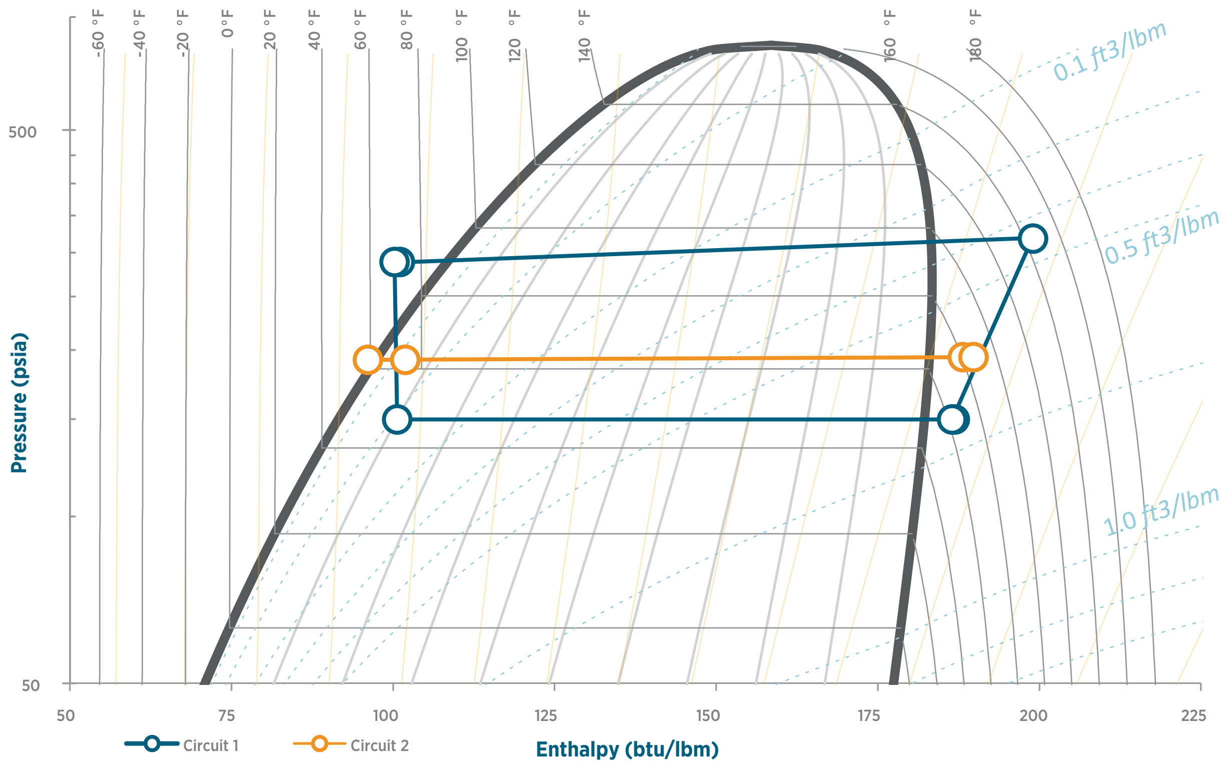


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 11



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 12

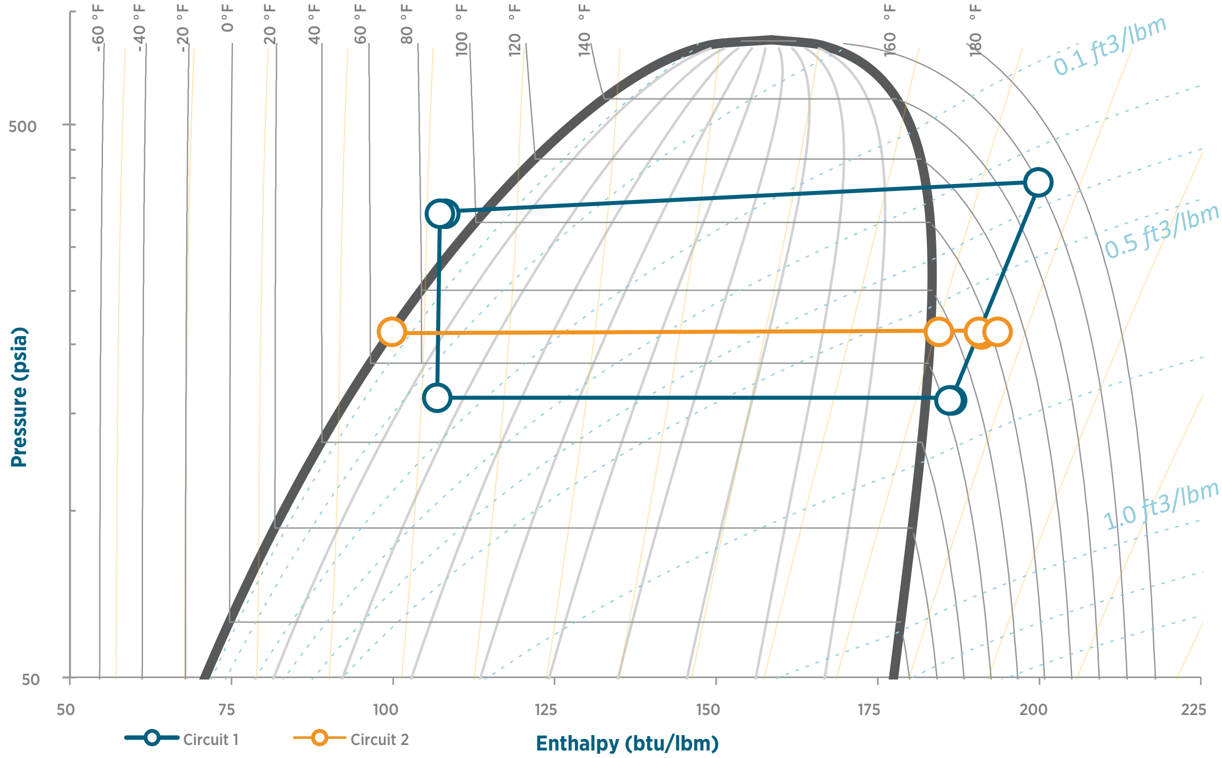
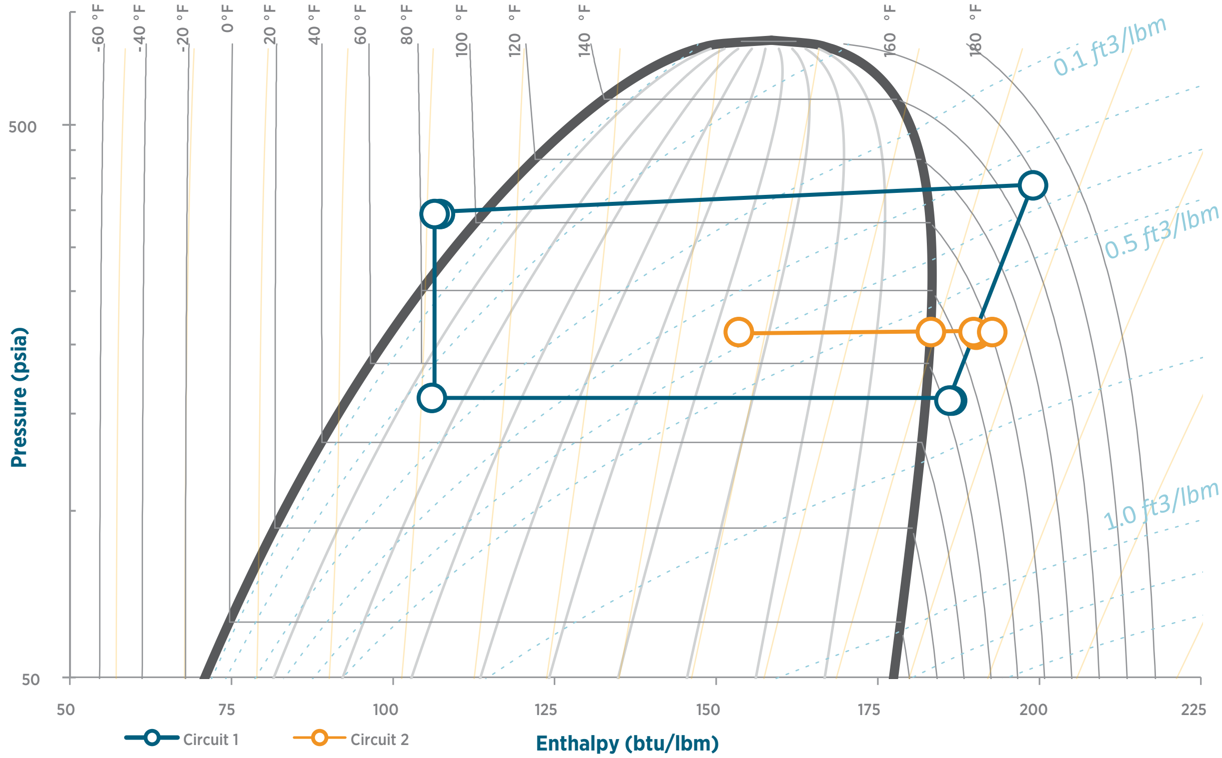


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 13



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 14

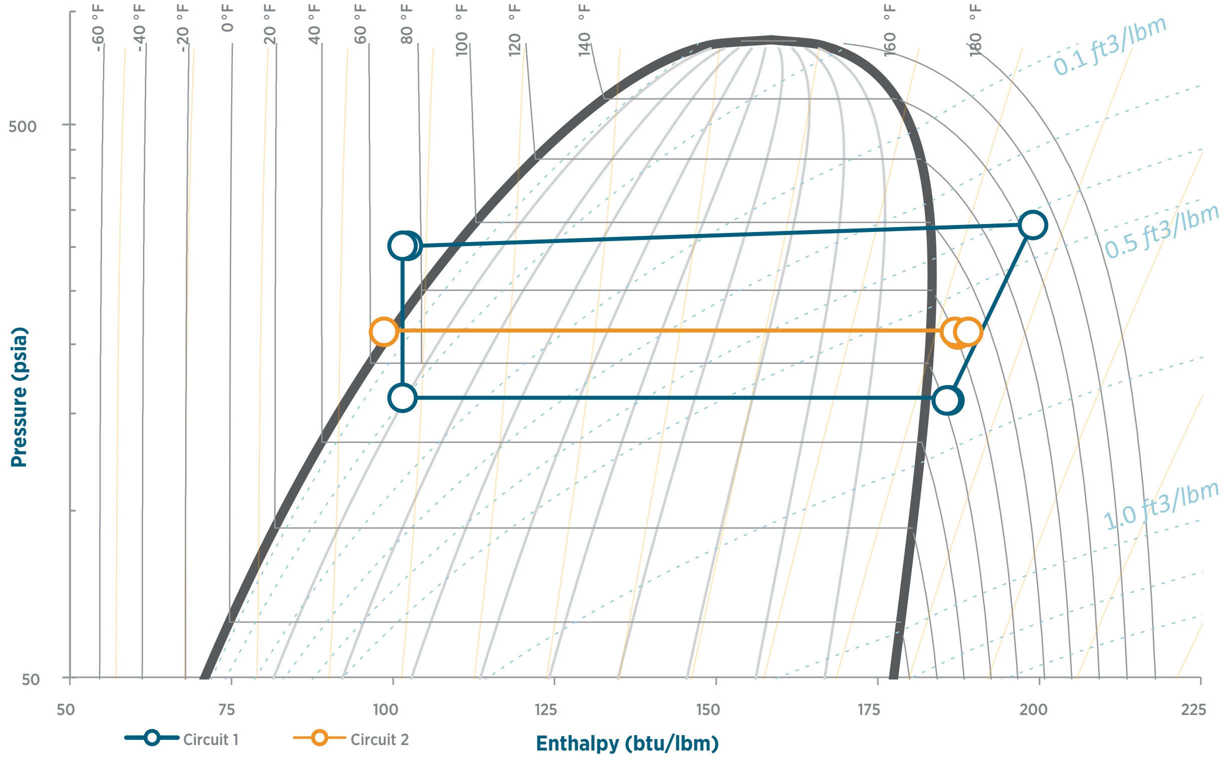
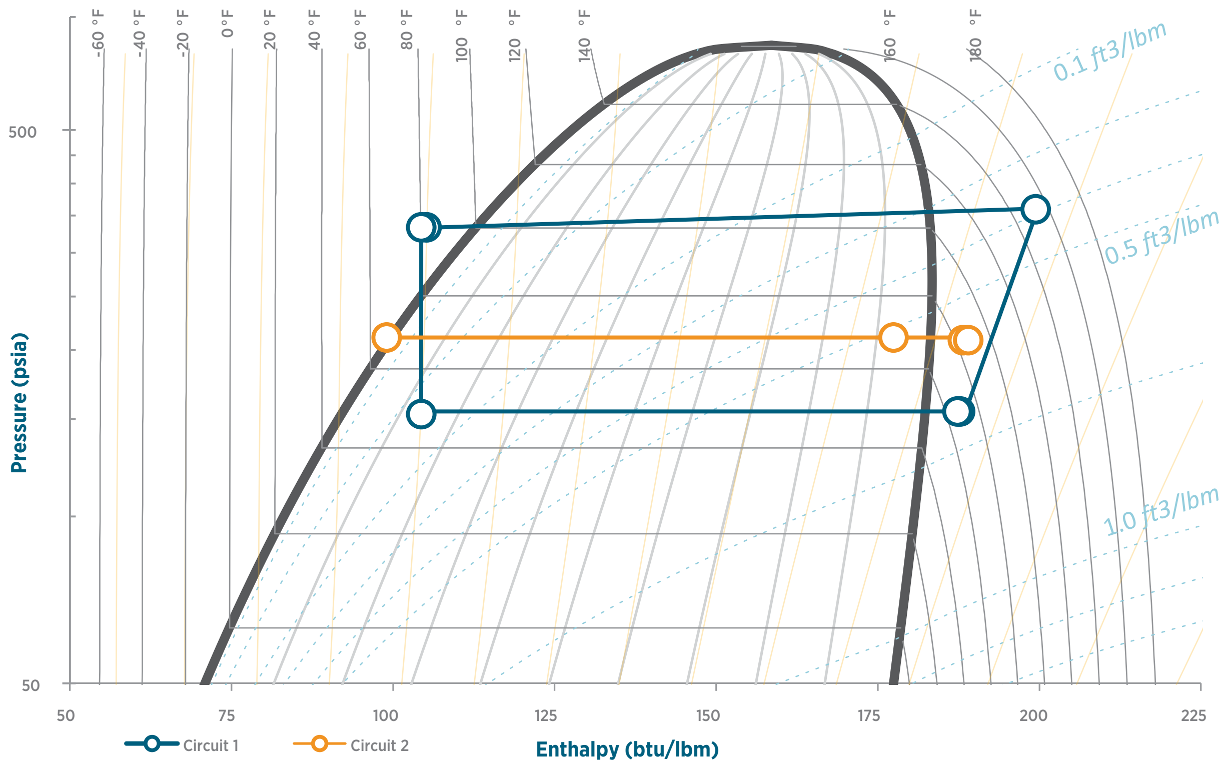


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 15



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 16

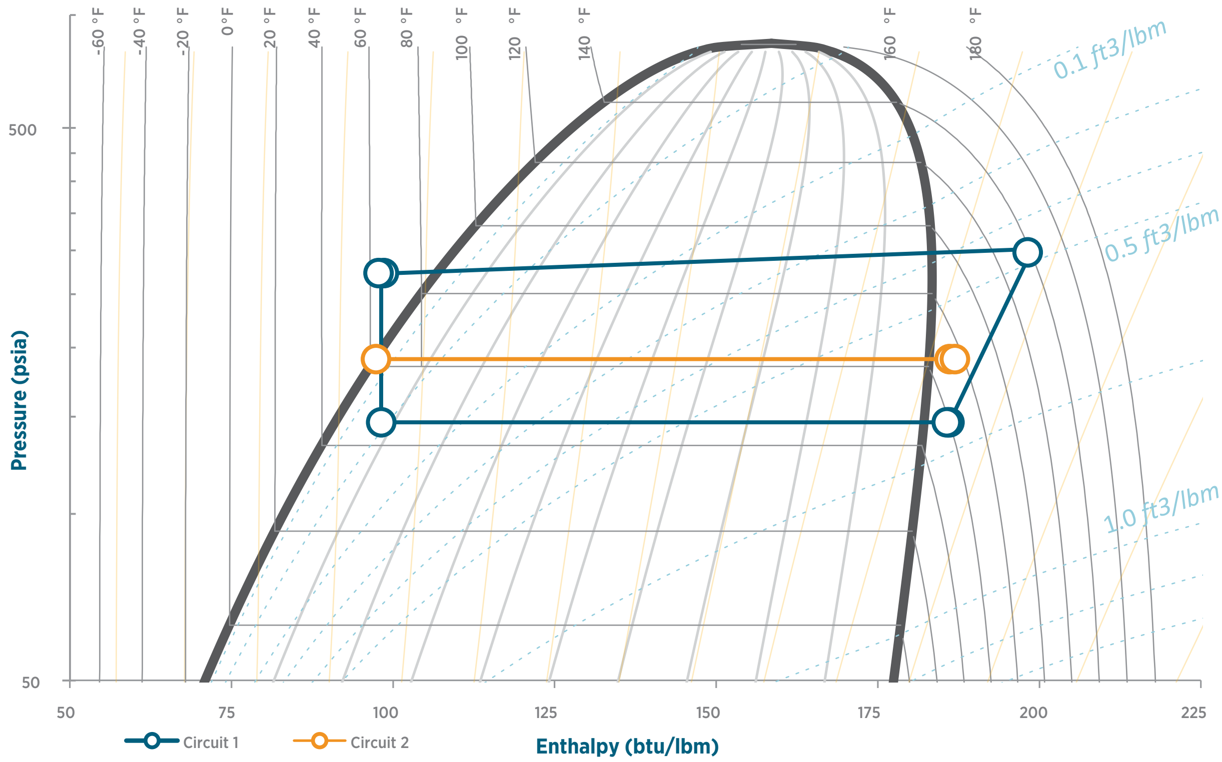
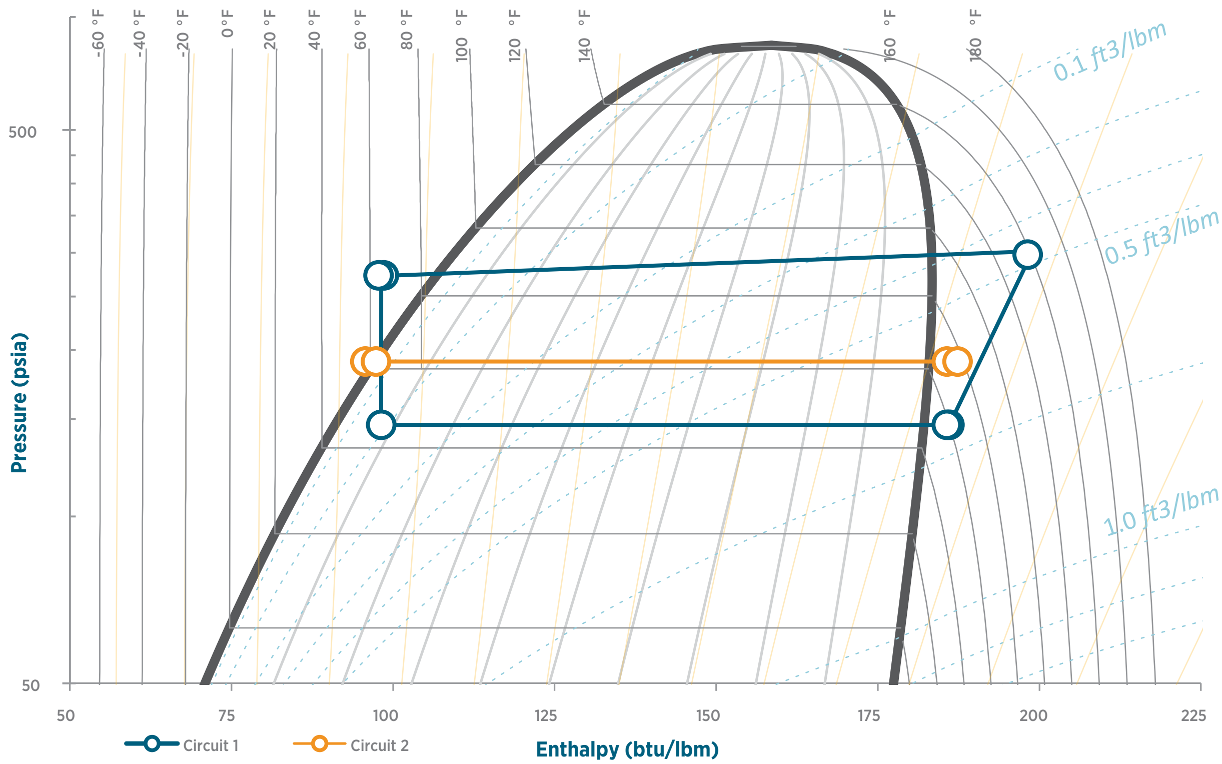


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 17



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 18

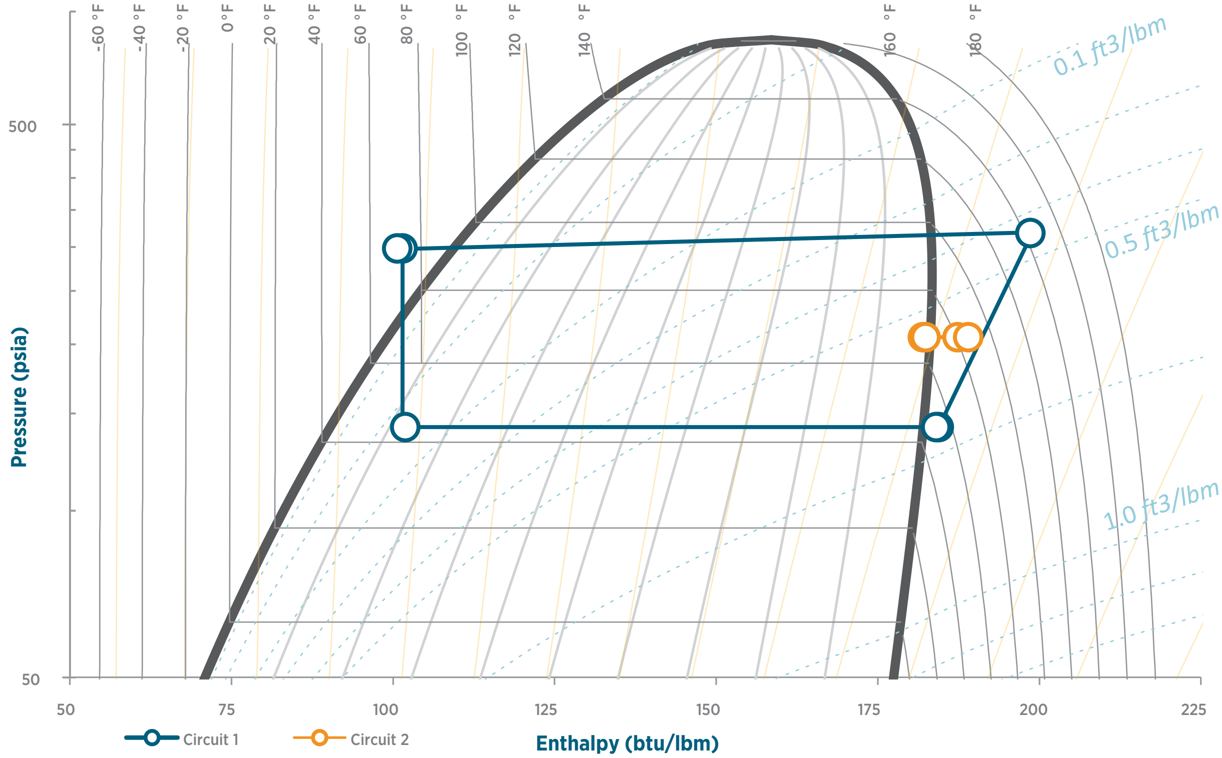
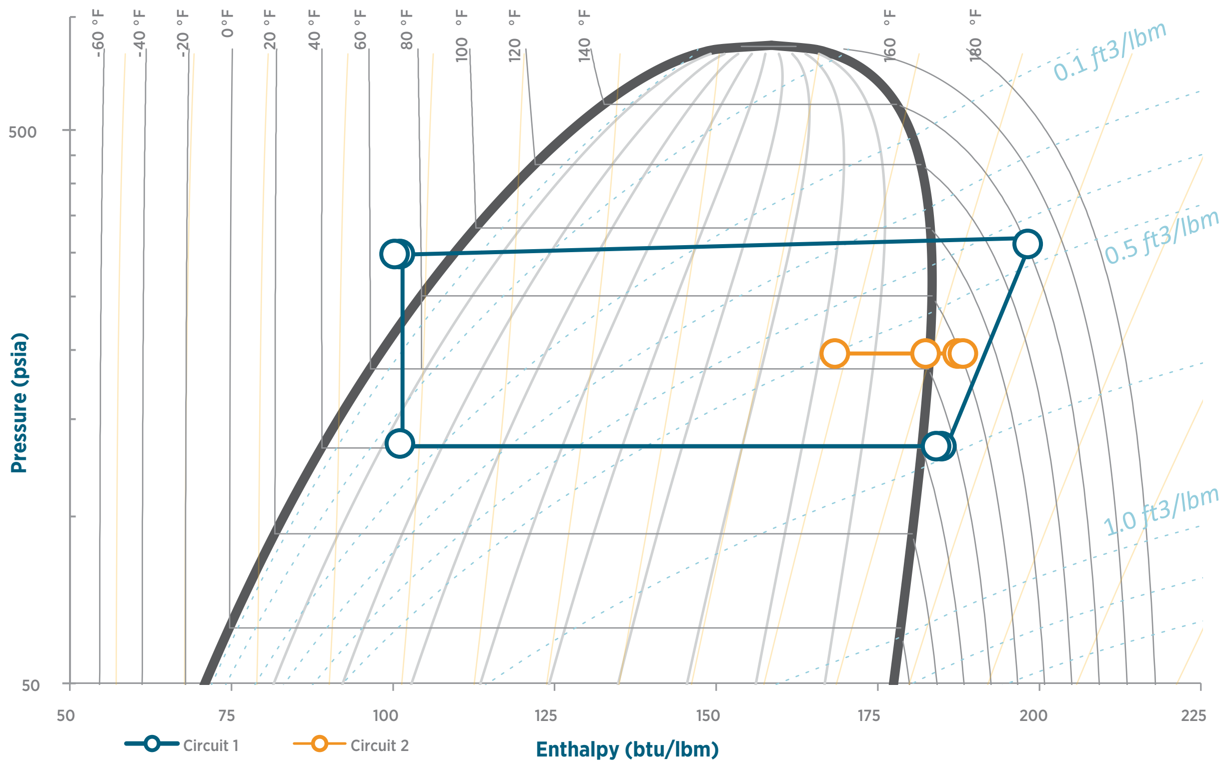


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 19



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 20

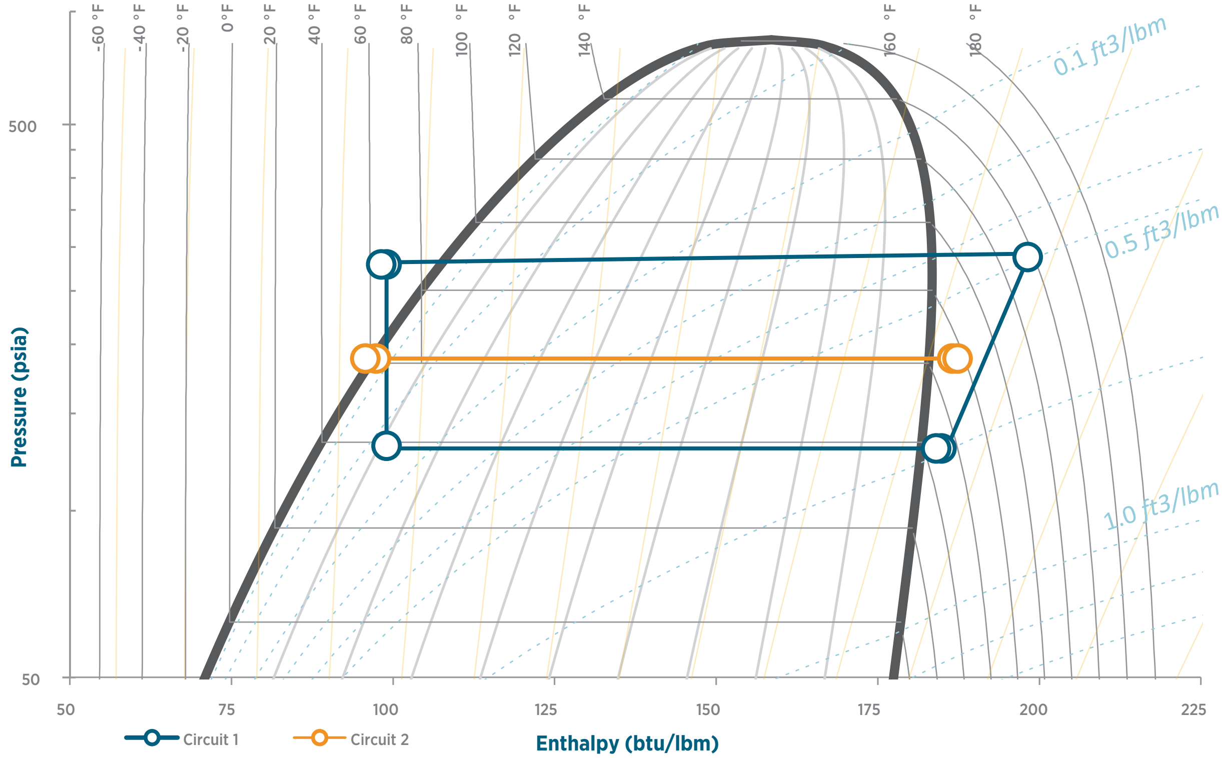
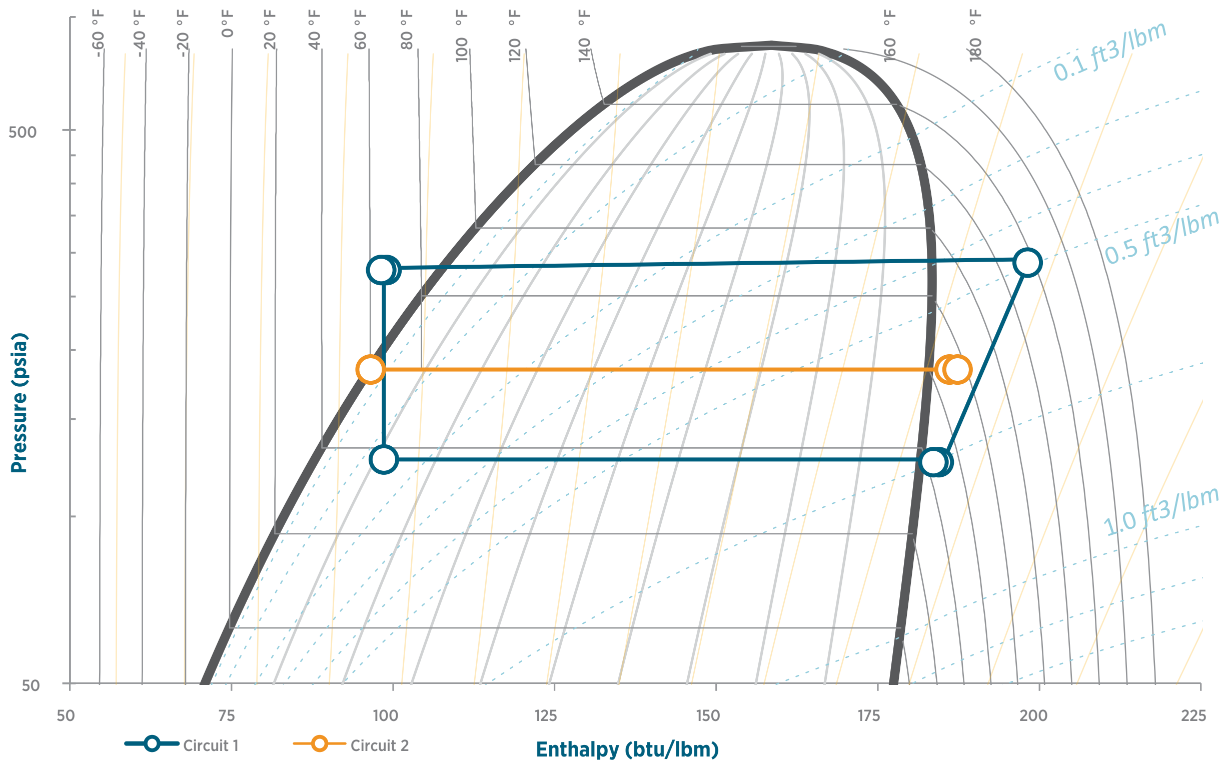


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 21



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 22

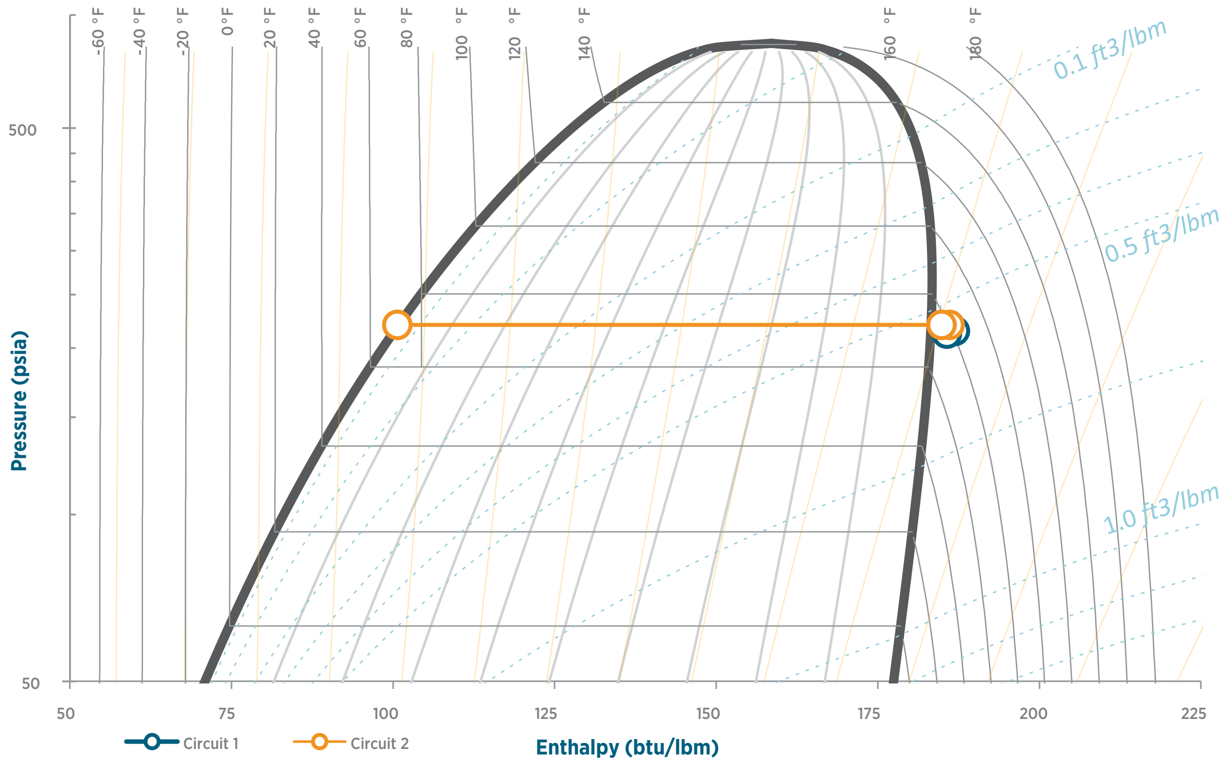
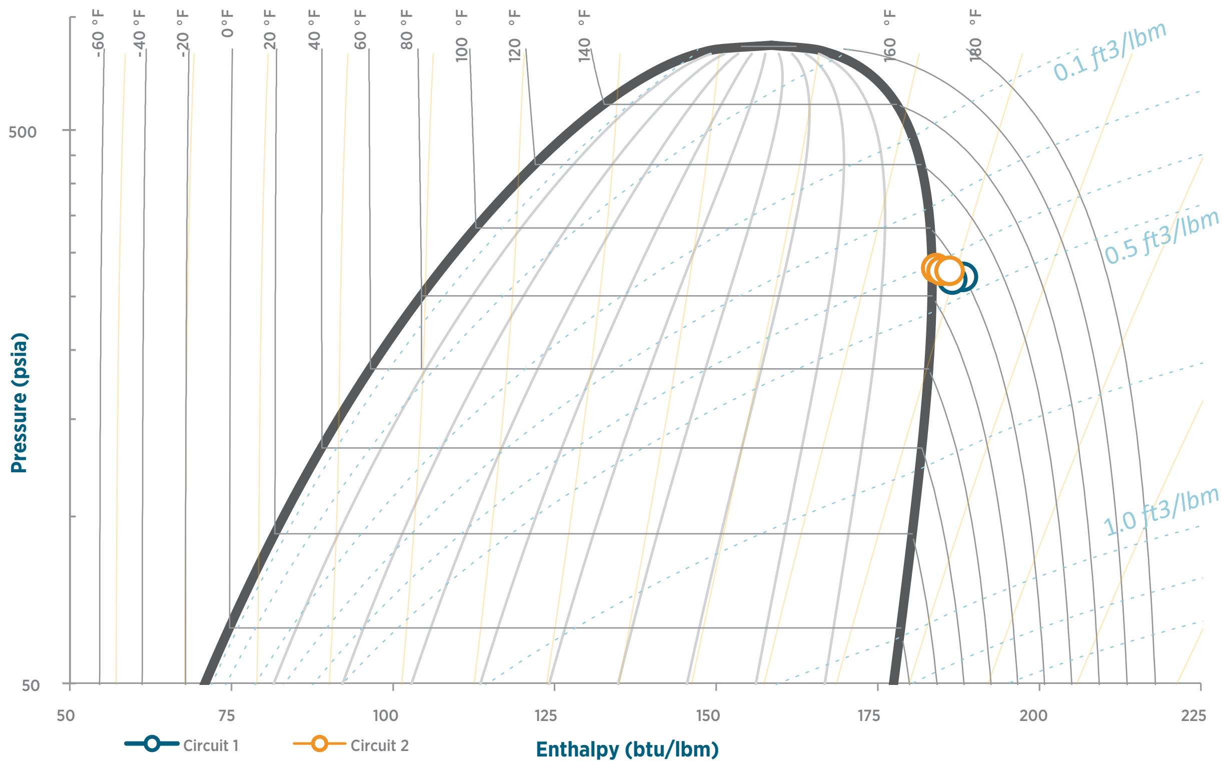


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 23



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 24

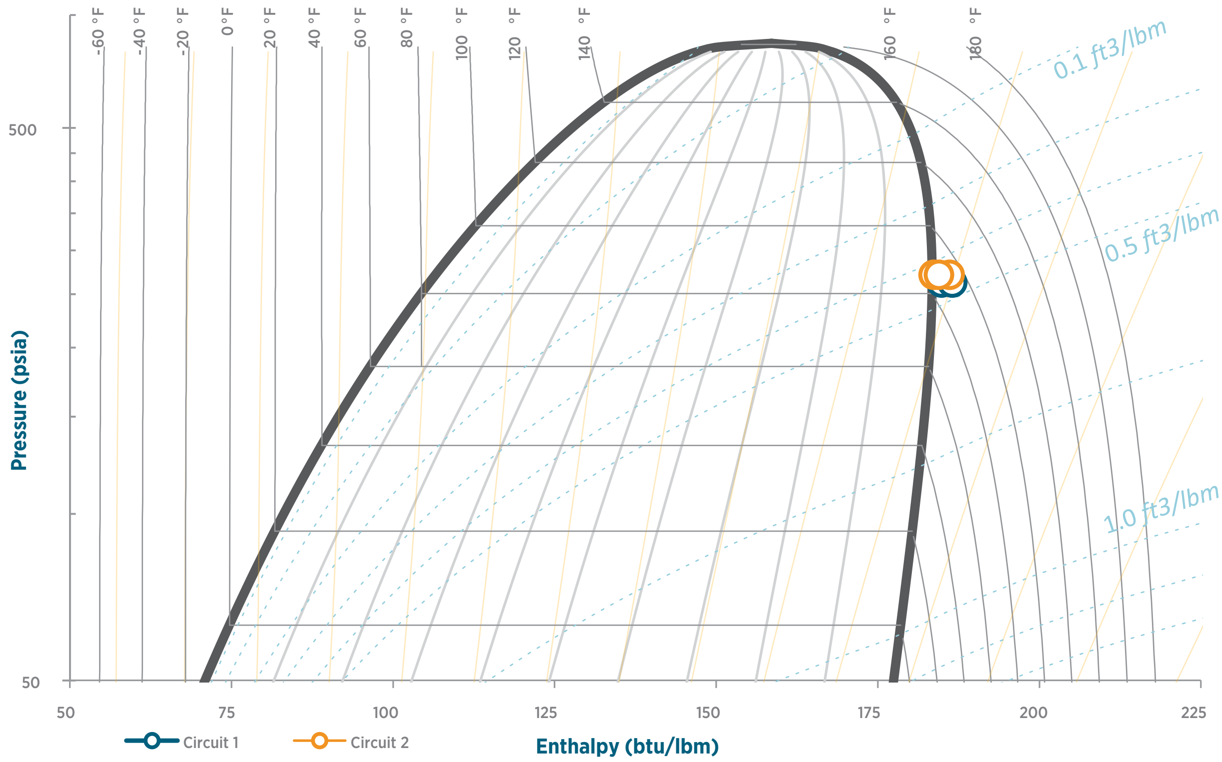
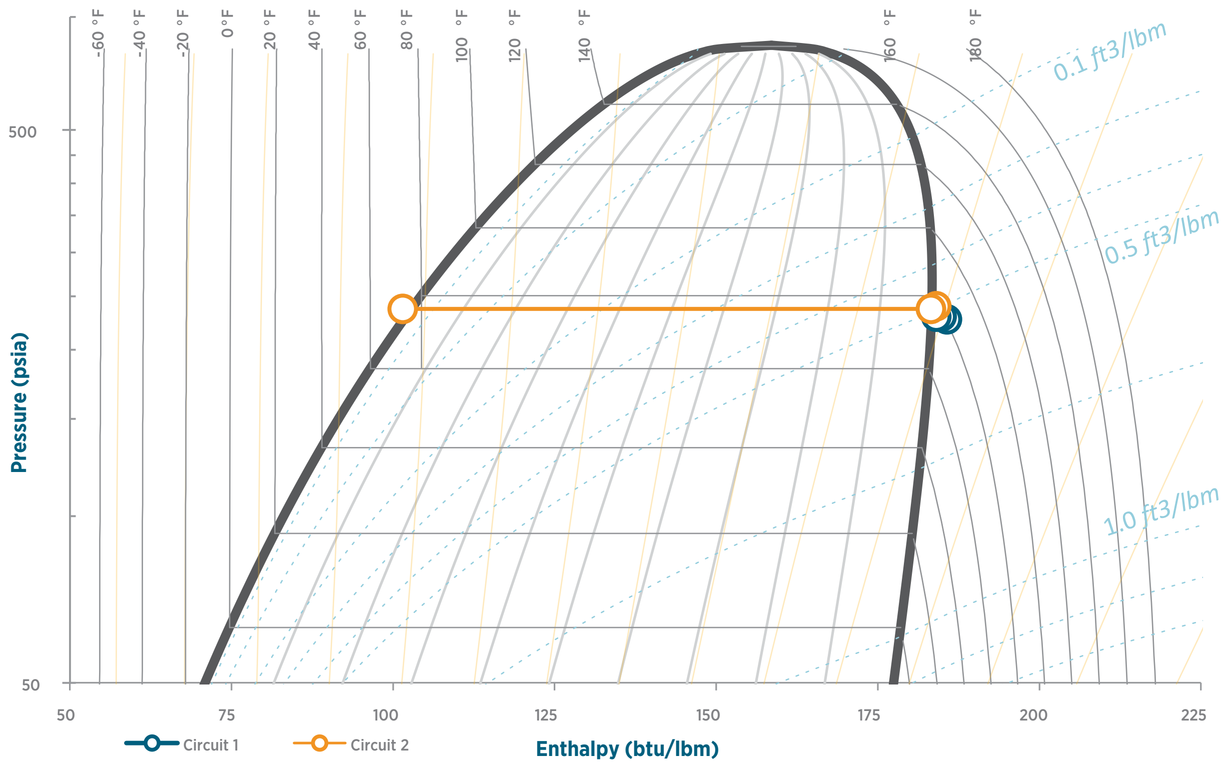


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 25



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 26

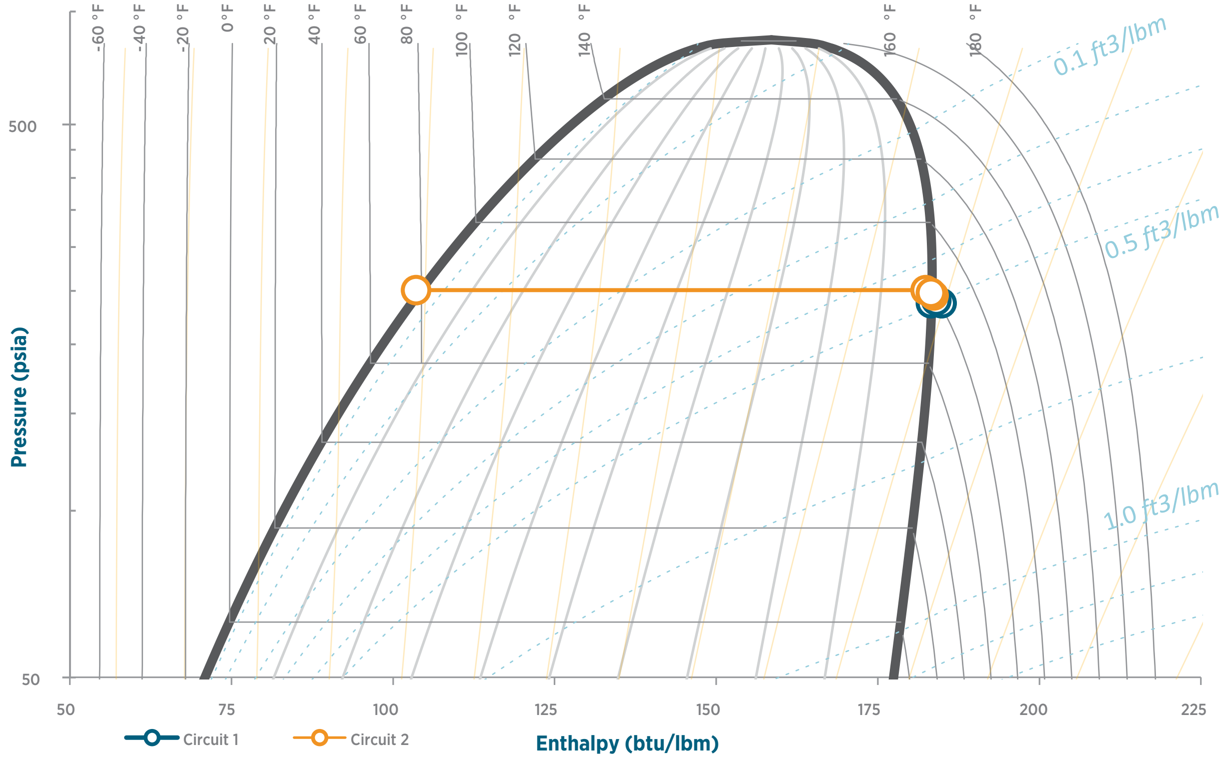
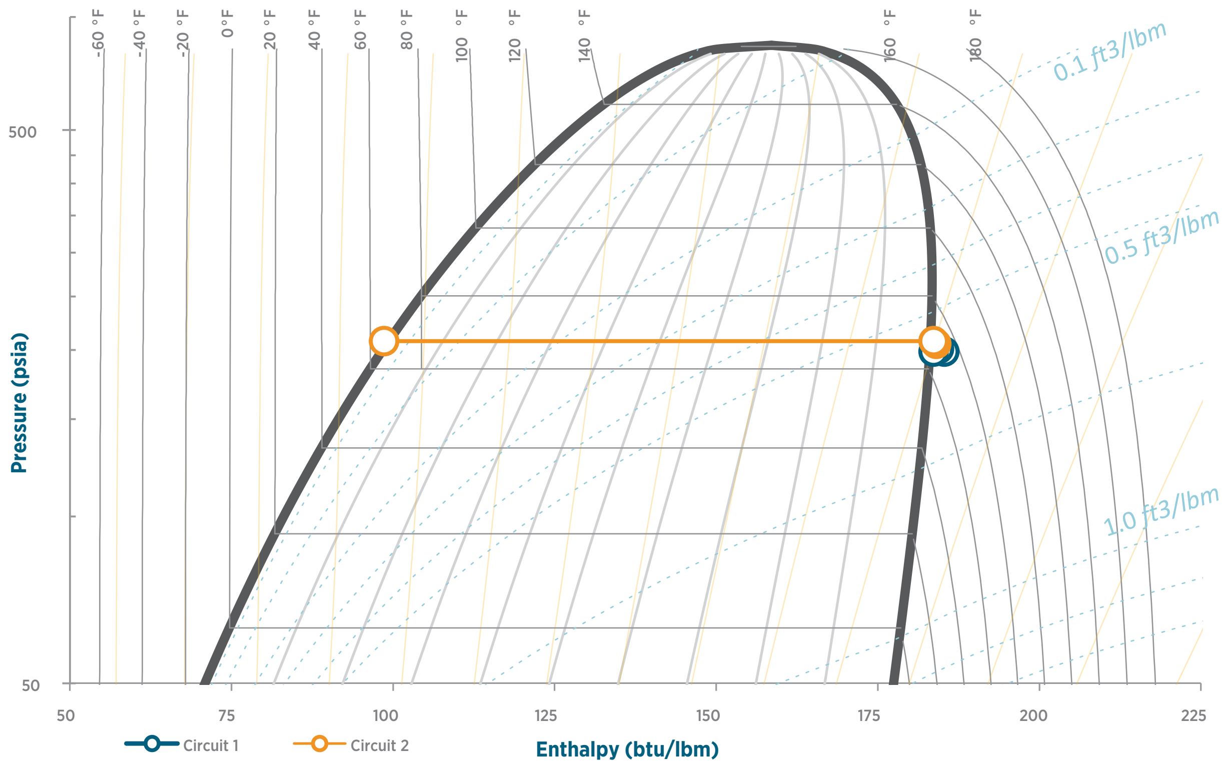


FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 27



APPENDIX C: PRESSURE ENTHALPY DIAGRAMS

FIGURE C1: PRESSURE ENTHALPY DIAGRAM TEST 28