SPEAKMAN HYBRID ROOFTOP UNIT PERFORMANCE: WESTERN COOLING CHALLENGE LABORATORY TEST RESULTS

HT.10.SCE.232 Report

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Southern California Edison extends thanks to Speakman Cooling Solutions for their contribution of the Air$_2$0 Quattro HCRS2500 hybrid rooftop unit used for evaluation, and is grateful to the US Department of Energy Office of Building Technology for supporting the laboratory testing of Western Cooling Challenge equipment at the National Renewable Energy Laboratory.

Disclaimer

ABBREVIATIONS AND ACRONYMS

COP	Coefficient of Performance, *unitless*
C_p	Specific Heat Capacity
DEC	Direct Evaporative Cooling
DOE	Department of Energy
DX	Vapor Compression
EA	Exhaust Air
EER	Energy Efficiency Ratio, *btu/W*
ESP	External Static Pressure, *"WC*
$\dot{H}$	Cooling Capacity, *kbtu/h*
$\dot{h}$	Specific Enthalpy, *btu/lb*
HVAC	Heating, Ventilation, & Air Conditioning
IDEC	Indirect-Direct Evaporative Cooling
IEC	Indirect Evaporative Cooling
$\dot{m}$	Mass Flow Rate, *lb/min*
NREL	National Renewable Energy Laboratory
OSA	Outside Air
OSAF	Outside Air Fraction
RA	Return Air
SA	Supply Air
T	Temperature
T_{db}	Dry Bulb Temperature
T_{wb}	Wet Bulb Temperature
$\dot{V}$	Volume Flow Rate
WCEC	Western Cooling Efficiency Center

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EXECUTIVE SUMMARY

The Western Cooling Challenge is a program with the express aim of advancing commercial development and market introduction of rooftop packaged air conditioners optimized for the hot-dry climates of the Western United States. The Challenge defines an ambitious energy and water-use efficiency criteria that is well beyond the reach of typical vapor compression air conditioners, but which is achievable with savvy application of existing technologies. One manufacturer has met the performance criteria thus far, showing energy savings on the order of 65% when compared to current federal standards.

This report documents the results of Western Cooling Challenge laboratory tests of the Speakman Hybrid, a rooftop air conditioner that uses indirect evaporative cooling, direct evaporative cooling, and conventional vapor compression cooling. The Western Cooling Efficiency Center (WCEC) orchestrated the laboratory tests, which were conducted by the National Renewable Energy Laboratory under a commitment from the US Department of Energy Office of Building Technology to provide laboratory testing for advanced cooling technologies submitted to the Western Cooling Challenge.

During the progress of testing, ownership of the technology changed hands. Speakman Cooling Solutions submitted the original entry for the Challenge in February 2011. In November 2011, all product rights were sold to AirMax Industries LLC, who has discontinued the product. To our knowledge, Speakman Company is no longer involved in the cooling industry.

The laboratory testing aimed to evaluate performance of the hybrid across a range of psychrometric conditions and operating scenarios. The results documented in this report focus on the rating criteria for the Western Cooling Challenge. A more thorough explanation of the laboratory test methods, hybrid operating scheme, and performance for the system in a wider variety of scenarios is available in a parallel publication: *Laboratory Testing of a Hybrid Rooftop Air Conditioner Utilizing Vapor Compression, Direct Evaporative, and Indirect Evaporative Cooling*.

Performance results for the Speakman hybrid show the potential for significant energy savings compared to conventional rooftop air conditioners in certain scenarios. However, the system has some shortcomings which make the design only as efficient as current conventional equipment at peak conditions, and which will significantly reduce the potential for annualized energy savings in most commercial applications. The system did not meet Western Cooling Challenge minimum performance criteria, but does prove the functionally of a unique conceptual design that has ample potential for improvement.

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 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 2011, Speakman Cooling Solutions submitted the Air$_2$O Quattro HCRS2500 hybrid rooftop unit as an entry to the Challenge, and WCEC coordinated rigorous laboratory testing of the system at NREL, in Golden CO. NREL operates a heating, ventilation, and air-conditioning (HVAC) test facility which is uniquely suited to accurately measure the cooling performance, energy, and water use of advanced cooling systems. NREL conducted tests of the equipment during May – October 2011. During this time, Speakman opted to leave the cooling industry, and sold all existing equipment, product lines, technology rights, manufacturing and distribution relations to AirMax Industries, which has discontinued the product.

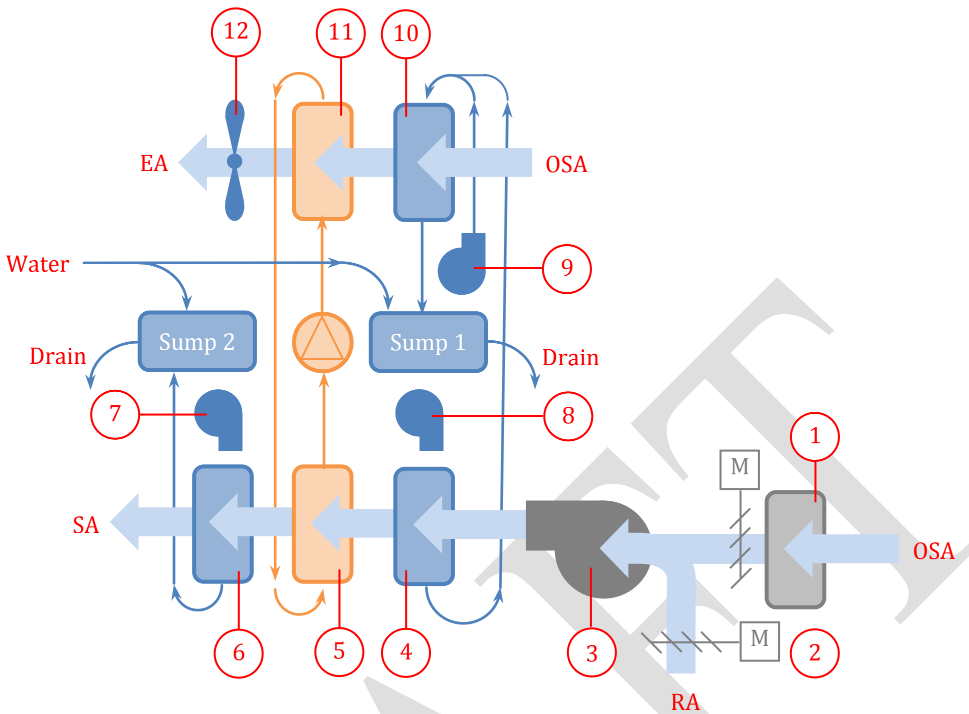
This report reviews the design of the Speakman hybrid system, describes the laboratory test methodology, and documents performance results at the Western Cooling Challenge test conditions. Laboratory results reveal that the equipment does not qualify for certification, even though some of the part-load operating modes are much more efficient than conventional cooling equipment. The results suggest that the conceptual hybrid system architecture applied is a compelling and worthwhile strategy, but also that there are many specific design characteristics which could be changed to gain large efficiency improvements.

A parallel publication, "*Laboratory Testing of a Hybrid Rooftop Air Conditioner Utilizing Vapor Compression, Direct Evaporative, and Indirect Evaporative Cooling*", documents the test methodologies and complete set of laboratory results in further detail. The publication includes characterization of performance for each sub-component, maps of cooling capacity and efficiency across a range of psychrometric conditions and airflow rate scenarios, and provides an in depth evaluation of potential design improvements for the equipment.

OVERVIEW OF SPEAKMAN HYBRID SYSTEM OPERATION

The Speakman hybrid system couples indirect evaporative cooling, direct evaporative cooling, and vapor-compression. Each cooling component may operate independently or in concert with others, as determined by the equipment control scheme which chooses operating mode based on outside air conditions and commissioned settings.

Figure 1illustrates the system air flow and water flow paths as a conceptual schematic. Outside air and return air are mixed to various ratios with motorized dampers, depending on operating mode and the commissioned setpoint for minimum outside air rate. The mixed air stream is driven through a chilled water coil, a vapor compression evaporator coil, and a direct evaporative media for cooling. The condenser side of the vapor compression system is air cooled with a direct evaporative pre-cooler, and the sump water from this condenser air cooler is circulated through the chilled water coil to provide indirect evaporative cooling. The vapor compression circuit includes a reversing valve so that this system can also provide space heating as an electric heat pump.



1. Filtration	5. DX Evaporator	9. Condenser Air Cooler Pump
2. Motorized Dampers	6. Direct Evaporative Media	10. Direct Evaporative Media
3. Supply Air Blower	7. Direct Evaporative Pump	11. DX Condenser
4. Water Coil	8. Indirect Evaporative Pump	12. Condenser Fan

FIGURE 1: CONCEPTUAL SCHEMATIC FOR THE HYBRID

Not all components in the system operate concurrently. There are four modes of operation for cooling, which cycle as a function of the measured outside air conditions (temperature and humidity) and the commissioned minimum outside air fraction. The cooling modes include:

1. Economizer. The outside air temperature is cool enough to meet a factory selected supply air set point directly. The supply fan operates with 100% outside air, but no mechanical cooling components are enabled.
2. Direct Evaporative. The outside air conditions are such that direct evaporative cooling alone can achieve the supply air set point. The supply fan operates with 100% outside air, and pump for the direct evaporative cooler is enabled, but all other components remain off.
3. Indirect & Direct Evaporative. The supply and condenser fans operate, and the indirect evaporative and direct evaporative pumps are enabled, but the compressor remains off. The system operates with 100% outside air. Water is cycled over the evaporative condenser air cooler, chilled evaporatively, and circulated through the water coil to provide sensible cooling for supply air stream. After the water coil, the supply air stream is cooled further by the direct evaporative system.
4. Indirect Evaporative & DX: Outside air temperatures are hot enough or humid enough, that direct and indirect evaporative cooling modes cannot meet the supply air set point. In this case, the indirect evaporative circuit, and the vapor compression circuit both operate. The direct evaporative system is disabled in this mode.

It should be noted that when installed in an application where the unit can be allowed to operate as recirculation-only at times, the indirect evaporative circuit will not operate in the last cooling mode described. Instead, the DX system will operate alone, with 0% outside air. In this case, while water isn't circulated through the water coil, a secondary pump will still cycle water over the evaporative condenser air cooler.

WESTERN COOLING CHALLENGE PERFORMANCE CRITERIA

Western Cooling Challenge rating centers on steady-state sensible energy efficiency at full capacity operation, with 120 *cfm/nominal ton* ventilation rate, and external resistance that would produce 0.7 *" WC* external static pressure at 350 *cfm/nominal ton*, at two different outdoor psychrometric conditions. The laboratory test protocol, described in Table 1, was designed roughly around conditions in a large retail facility. The two outdoor conditions were chosen to represent peak-day design conditions and average cooling-season conditions for cooling intensive regions in the Western United States.

Since multiple system operating conditions for the Challenge tests are dependent on the nominal capacity of the equipment, the laboratory procedure focused on determining 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, and does not provide enough information to offer a comparison of the total annual energy savings that a system would offer in comparison to a baseline conventional rooftop unit. The Challenge tests stand as hurdles to prove performance a couple key operating points. The complete energy footprint of a system must consider performance in all modes, at all operating conditions, and be based on a model that integrates performance across the operating scenarios that would be encountered annually in a particular climate zone. 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 from these tests are documented in a parallel publication, "*Laboratory Testing of a Hybrid Rooftop Air Conditioner Utilizing Vapor Compression, Direct Evaporative, and Indirect Evaporative Cooling*".

TABLE 1: WESTERN COOLING CHALLENGE PERFORMANCE CRITERIA[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

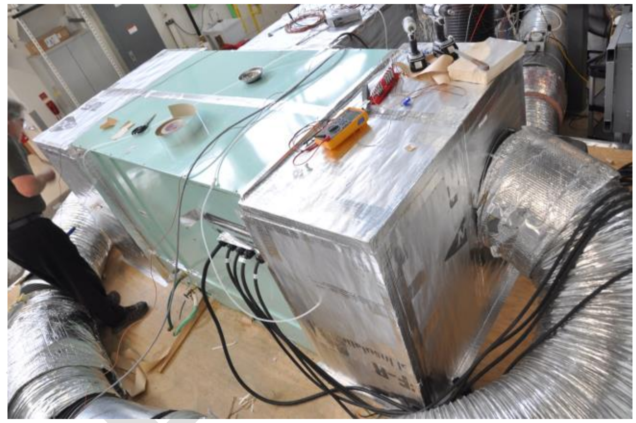
[1] Performance criteria are described in more detail in the *Western Cooling Challenge Program Requirements*

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

Max Water Use (*gal/ ton-h*)	NA	4

METHODS

LABORATORY TEST FACILITY



The NREL HVAC equipment test laboratory manages airflow rate, inlet psychrometric conditions, and unit external static pressure for two separate air streams. The primary air stream feeds the supply air side of the unit, while the secondary air stream feeds the outdoor section. The primary stream was given a single air flow through the outside-air inlet at the desired mixed air conditions to simulate the psychrometric mixing of return air and outside air at desired conditions and a desired outside air fraction. Likewise, the secondary air stream was managed to provide desired outside air conditions at an air flow rate appropriate for normal function at standard conditions.

Since psychrometric relationships change as barometric pressure shifts, great care is taken to convert from measured conditions in Golden Colorado at 5,675 *feet* elevation, to standard operating conditions at sea level. Generally the conversions follow a process of calculation that:

1. maintains equal mass flow rates
2. maintains equal sensible heat transfer potential for each heat exchange component
3. maintains equal latent heat transfer potential for each heat exchange components,

Due to decreased air density at elevation, and thus different relationships between mass flow rate, fan differential pressure, fan power consumption, and fan temperature rise, performance for the fan is determined separately from thermodynamic tests, and metrics such as EER are calculated with values combine from each tests. The detailed protocol for of these process is documented in a parallel publication: "*Laboratory Testing of Hybrid Rooftop Air Conditioner Utilizing Vapor Compression, Direct Evaporative, and Indirect Evaporative Cooling.*" All values presented in this report are for standard sea level operation.

DETERMINING NOMINAL CAPACITY

Nominal capacity of a system is typically determined at AHRI standard rating conditions. For the Challenge this nominal value is used to determine the credited ventilation cooling, the sensible credited cooling capacity, and thus the sensible credited EER by which a unit qualifies for certification. However, since the AHRI test occurs with 0% outside air and some hybrid equipment will have a non-zero minimum outside air fraction, the Cooling Challenge uses an alternate method to determine nominal capacity that uses measured performance from full-capacity operation under WCC peak conditions. 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 will be adjusted to match the same external resistance according to:

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

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:

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

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:

$$OSAF = \frac{\dot{V}_{OA}}{\dot{V}_{SA}} \qquad 3$$

The iterative nominal-capacity test resulted in the set of system operating conditions described in Table 2, which were held constant for all subsequent Western Cooling Challenge tests. Note that laboratory tests were not able to isolate a complete convergence for all requirements, so the nearest achievable scenario was used; the operating conditions described in Table 2 would call for an external static pressure of 0.67 *"WC*.

TABLE 2: RESULTS FOR NOMINAL CAPACITY TEST

Operating Condition	Value for Tests
Exsternal Static Pressure (*"WC*)	0.57
Supply Airflow (*scfm*)	1925
OSAF	35%
Nominal Capacity (*tons*)	5.62

CALCULATION OF 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 $\dot{H}_{System} = \dot{m}_{SA} \cdot (h_{MA} - h_{SA})$ 4; this is the net cooling produced by the system, including what is lost due to fan heat.

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

The space cooling capacity (also called recirculation cooling, or room cooling), given by equation $\dot{H}_{Space} = \dot{m}_{SA} \cdot (h_{RA} - h_{SA})$ 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.

$$\dot{H}_{Space} = \dot{m}_{SA} \cdot (h_{RA} - h_{SA}) \qquad 5$$

The Western Cooling Challenge is generally concerned with a systems 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 $\dot{H}_{Space}^{Sensible} = \dot{m}_{SA} \cdot C_p \cdot (T_{RA} - T_{SA})$ 6,

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

And the latent space cooling is determined as:

$$\dot{H}_{space}^{latent} = \dot{H}_{Space} - \dot{H}_{Space}^{Sensible} \qquad 7$$

The ventilation cooling capacity is the difference between the system cooling and space cooling, and it can also be calculated according to equation $\dot{H}_{ventilation} = \dot{m}_{SA} \cdot (h_{MA} - h_{RA})$ 8.

$$\dot{H}_{ventilation} = \dot{m}_{SA} \cdot (h_{MA} - h_{RA}) \qquad 8$$

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.

$$\dot{V}_{credited\ ventilation} = 120 \cdot \dot{H}_{nominal} \qquad 9$$

The credited ventilation rate translates to a credited ventilation cooling capacity as described in equation $\dot{H}_{credited\ ventilation} = \dot{m}_{SA} \cdot \left(\left(\frac{\dot{V}_{credited\ ventilation}}{\dot{V}_{SA}} \cdot h_{OA} + \left(1 - \frac{\dot{V}_{credited\ ventilation}}{\dot{V}_{SA}}\right) \cdot h_{RA} \right) - h_{RA} \right)$ 10:

$$\dot{H}_{credited\ ventilation} = \dot{m}_{SA} \cdot \left(\left(\frac{\dot{V}_{credited\ ventilation}}{\dot{V}_{SA}} \cdot h_{OA} + \left(1 - \frac{\dot{V}_{credited\ ventilation}}{\dot{V}_{SA}}\right) \cdot h_{RA} \right) - h_{RA} \right) \qquad 10$$

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

$$\dot{H}_{credited\ ventilation}^{sensible} = \dot{m}_{SA} \cdot C_p \cdot \left(\left(\frac{\dot{V}_{credited\ ventilation}}{\dot{V}_{SA}} \cdot T_{OA} + \left(1 - \frac{\dot{V}_{credited\ ventilation}}{\dot{V}_{SA}}\right) \cdot T_{RA} \right) - T_{RA} \right) \qquad 11$$

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.

$$\dot{H}_{credited}^{sensible} = \dot{H}_{Space}^{Sensible} + \dot{H}_{credited\ ventilation}^{sensible} \qquad 12$$

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

$$EER_{credited}^{sensible} = \frac{\dot{H}_{credited}^{sensible}}{\dot{W}} \left\{ \frac{kbtu/hr}{kW} \right\} \qquad 13$$

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 AHRI 340/360 protocol, which operates equipment without outside air, and gives credit for latent cooling. A conventional system rated with an EER of 12 according to AHRI 340/360 protocol will have a "sensible credited EER" nearer 9 at Western Cooling Challenge test conditions.

RESULTS

Laboratory tests observed operation in each mode and across a range of psychrometric operating conditions, in order to describe efficiency of the system in various scenarios, to characterize performance of each component, and to test sensitivity to variables such as airflow rate. The results presented here document measured system performance at Western Cooling Challenge rating conditions. Further analysis of the laboratory results, review of the detailed system design, and development of a complete model for estimating annualized performance in various climactic conditions is documented in a parallel publication: "*Laboratory Testing of Hybrid Rooftop Air Conditioner Utilizing Vapor Compression, Direct Evaporative, and Indirect Evaporative Cooling.*"

The following psychrometric charts illustrate the cooling process recorded for each Western Cooling Challenge test. Note that only the IEC-DX mode was tested at Peak conditions, while all three active cooling modes were tested at Annual conditions. Although the direct and indirect-direct evaporative cooling modes may very well be more efficient than the IEC-DX mode, their operation is precluded by the system for this outside air condition because the supply-air temperature and sensible space cooling generated would be inadequate. The system control scheme dictates that the indirect-direct evaporative mode would operate at WCC Annual conditions, but all three operating modes were tested at WCC Annual conditions in order to evaluate the differences between each scenario. Figure 3 clearly shows the progression in sensible space cooling capacity, for operation in DEC, IDEC, and then IEC&DX modes for a single set of operating conditions. Each chart shows the outside air condition, return air condition, and mixed air condition for the required 35% *OSAF*. Note that for the DEC and IDEC modes, the equipment operates as 100% *OSAF.*

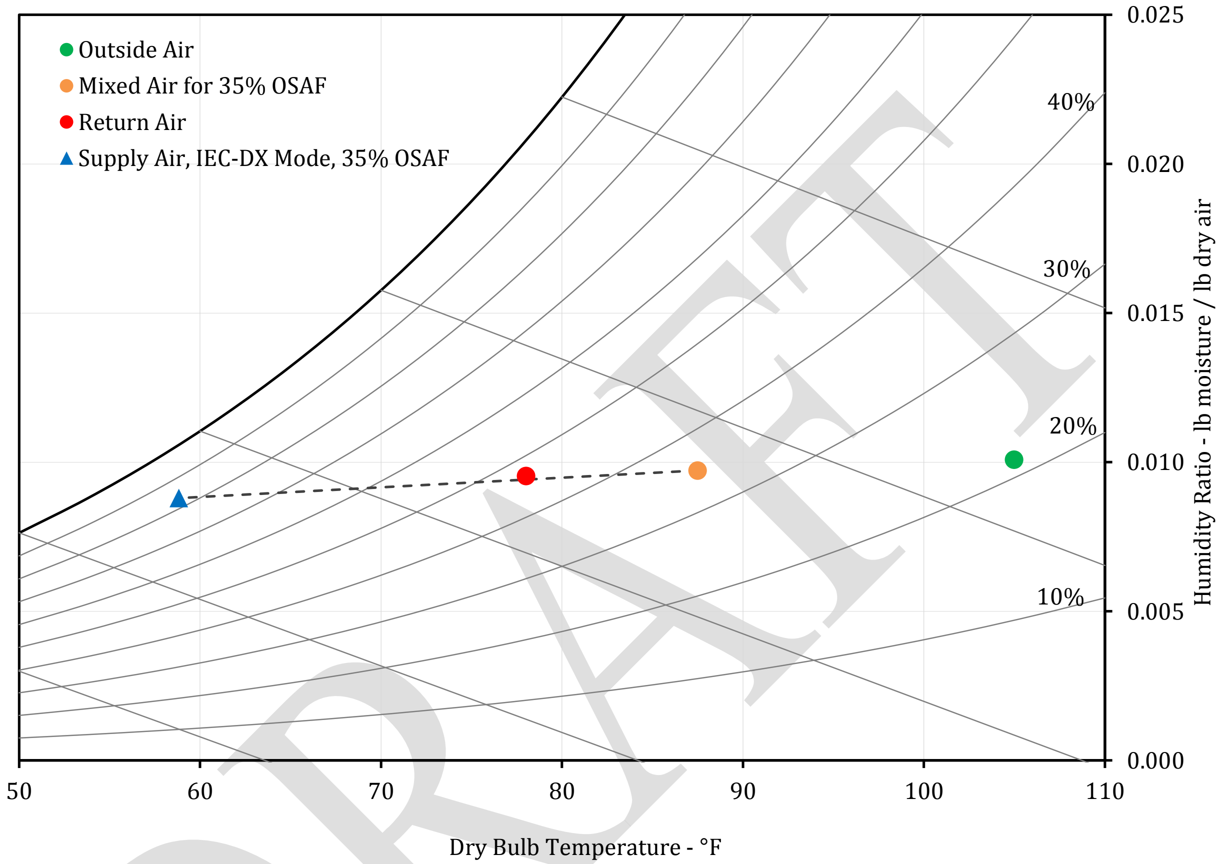


FIGURE 2: PSYCHROMETRIC CHART OF SUPPLY AIR CONDITIONS AND PAIRED INLET CONDITIONS FOR FULL CAPACITY OPERATION IN AT WCC PEAK CONDITIONS

The calculated results for all Western Cooling Challenge tests is summarized in Table 3, and compared against minimum performance criteria. A more detailed documentation of data measured at Western Cooling Challenge conditions is provided in Table 4 at the end of this report. While the equipment shows compelling efficiency improvements in certain operating modes, the performance observed does not meet all Western Cooling Challenge thresholds. Namely, the sensible credited EER for full capacity operation in IEC & DX mode at WCC Peak conditions misses the minimum by almost 40%. At this climate condition and in this operating mode, the equipment is only as efficient as a conventional standard RTU operating in the same scenario.

An interpolation of efficiency between the IDEC and DEC modes at WCC Annual conditions does reach minimum performance criteria for the climate condition. However, the added humidity in these modes surpasses the maximum humidity requirement, and the water consumption from operation in the IDEC mode is higher than allowable for the sensible capacity generated.

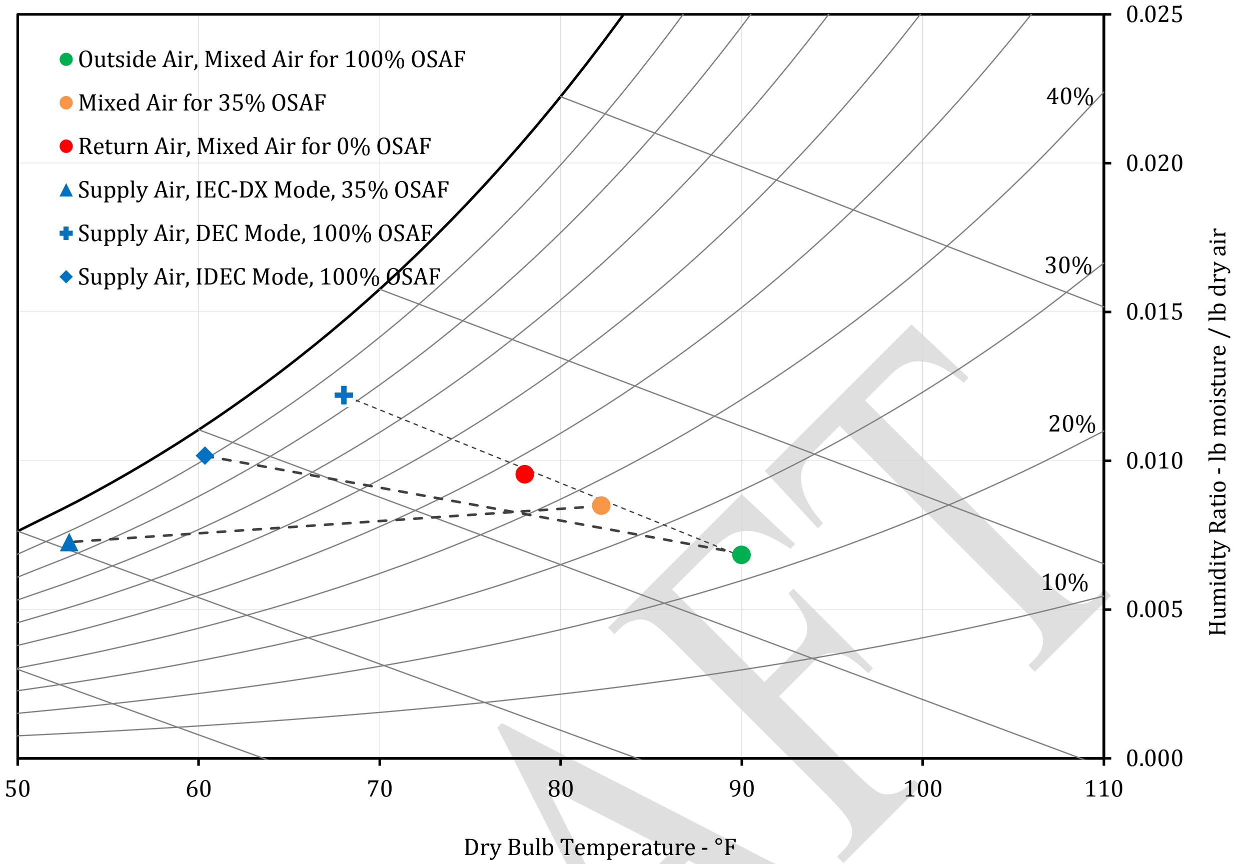


FIGURE 3: PSYCHROMETRIC CHART OF SUPPLY AIR CONDITIONS AND PAIRED INLET CONDITIONS FOR VARIOUS OPERATING MODES TESTED AT WCC ANNUAL CONDITIONS.

TABLE 3: SUMMARY OF WESTERN COOLING CHALLENGE RATED RESULTS

	WCC REQUIREMENTS	WCC PEAK CONDITION TEST	WCC ANNUAL CONDITION TESTS		
Operating Mode	*Peak*: Full Capacity *Annual*: All Applicable	IEC & DX	IEC & DX	DEC	IDEC
Outside Air Conditions, (T_{db}°*F*/T_{wb}°*F*)	*Peak*: 105/73 *Annual*: 90/64	105/73	90/64	90/64	90/64
Sensible Credited Capacity (*kbtu/h*)	-	59.62	61.09	29.57	45.24
Sensible Credited EER	*Peak*: min=14 *Annual*: min=17	8.6	9.3	20.9	16.4
Outdoor Ventilation (*scfm/nominal-ton*)	120	120	120	344	341
Supply Air Humidity (*lb/lb*)	max=0.0092	0.0088	.0073	.0122	.0102
Water Consumption (*gal/sensible-credited-ton-hr*)	*Peak*: NA *Annual*: max = 4.00	2.65	2.13	2.33	4.14
External Static Pressure ("WC)	0.67	0.57	0.57	0.57	0.57

Western Cooling Challenge criteria aside, these observations do show a system that could save significant amount of energy compared to the conventional alternative in certain scenarios. Figure 4 plots the sensible space cooling capacity, ventilation rate, and sensible space COP for the hybrid system at the two Challenge climate conditions, and compares performance to a similarly sized conventional RTU. The RTU selected for comparison is a single stage, constant volume, R-410a DOE 2010 standard system; and was chosen so as to generate sensible space cooling capacity equal to the Speakman Hybrid when operating at Peak conditions.

From this comparison it is clear that efficiency of the IDEC & DX mode at peak or annual conditions is only as efficient as conventional equipment. However, measurements for the IDEC mode indicate a 70% improvement in sensible space cooling efficiency; observation of DEC operation shows an 87% improvement. Thus, despite the fact that the equipment does not pass Cooling Challenge requirements, for a large portion of cooling hours in hot-dry climates the system will use more than 40% less energy.

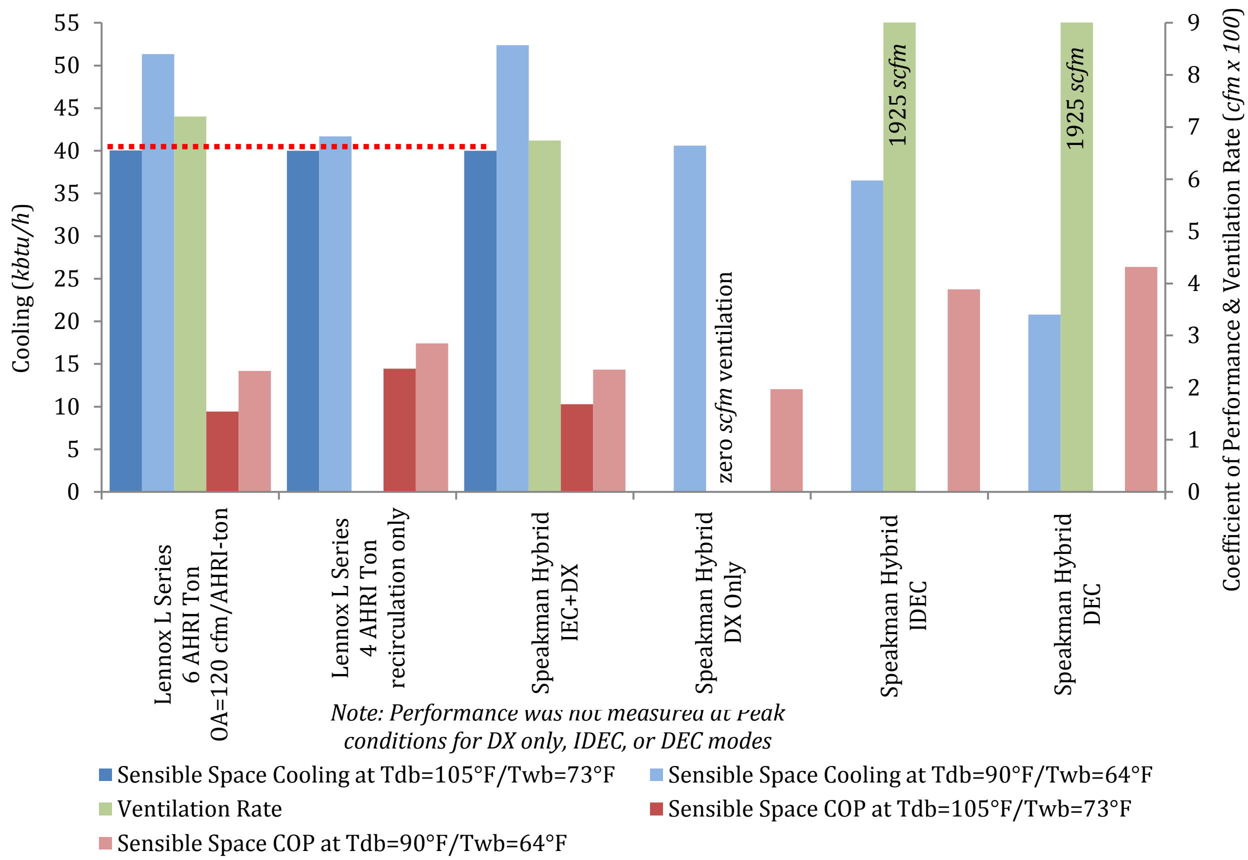


FIGURE 4: COMPARISON OF PERFORMANCE FOR THE SPEAKMAN HYBRID AND A CONVENTIONAL SINGLE STAGE CAV RTU

RECOMMENDATIONS

Even with an evaporative condenser air cooler, and indirect evaporative cooling to reduce load on the vapor compression system, the measured efficiency of the Speakman Hybrid in IEC-DX mode is barely equal to that of a similarly sized conventional system. According to our observations, it is apparent that the conceptual hybrid system architecture holds serious merit, but that there is large room for improvement to design of the system. Primarily, it seems that fan efficiency, internal resistance, and vapor compression circuit efficiency are all candidates for major design revisions.

The measured internal static pressure loss at 1,977 *acfm* is nearly 2.0*" WC*. With an external static pressure of approximately 0.5 *"WC*, the fan is pushed to operate at nearly 2.5 *"WC*. For a similar air flow rate and external resistance, a conventional RTU can operate with roughly half the total fan pressure, and half the fan power. If the hybrid is to be improved so as to operate more efficiently at peak, and reach for Western Cooling

Challenge performance thresholds, the internal resistance will have to be reduced dramatically. Given that roughly half of the annual electricity use from commercial rooftop units in California is due to fan power for continuous ventilation, the fact that the hybrid draws more than twice the power of a conventional system in ventilation-only mode is a major concern that demands improvement.

The efficiency indicated by operation in IDEC and DEC modes is compelling, even with the burden of internal static resistance and high fan power. The system could benefit significantly from some major design iterations, but the opportunity to pair the IDEC and DEC cooling with a parallel vapor compression operating deserves serious attention. AirMax currently manufacturers an IDEC system that is similar to the hybrid tested here except without the vapor compression components. In the near term, it is possible that this alternative system could be coupled effectively with a conventional RTU to deliver impactful energy savings.

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Table 4: Performance Data For Tests at Western Cooling Challenge Conditions

Column Titles	WCC Requirements	WCC Peak Condition Test	WCC Annual Condition Tests					
Operating Mode	*Peak*: Full Capacity *Annual*: All Applicable	IEC & DX	DX	IEC	IEC	IEC & DX	DEC	IDEC
Outside Air Conditions, (T_{db}°*F*/T_{wb}°*F*)	*Peak*: 105/73 *Annual*: 90/64	105/73	90/64	90/64	90/64	90/64	90/64	90/64
Return Air Conditions, (T_{db}°*F*/T_{wb}°*F*)	-	78/64	78/64	78/64	78/64	78/64	78/64	78/64
Supply Air Flowrate (*scfm*)	-	1925	1926	1919	1922	1926	1932	1915
Outside Air Fraction (%)	-	35%	0%	35%	100%	35%	100%	100%
Outside Air Flowrate (*scfm*)	-	674	0	672	1922	674	1932	1915
Sensible Space Capacity (*kbtu/h*)	-	40.00	40.61	16.47	11.59	52.37	20.80	36.50
Sensible Credited Ventilation Capacity (*kbtu/h*)	-	19.62	0	8.71	8.75	8.72	8.76	8.74
Sensible Credited Capacity (*kbtu/h*)	-	59.62	40.61	25.18	20.33	61.09	29.57	45.24
Total Power (*W*)	-	6970	6036	2729	2721	6547	1414	2750
Sensible Credited EER	*Peak*: min=14 *Annual*: min=17	8.6	6.7	9.2	7.5	9.3	20.9	16.4
Outdoor Ventilation (*scfm/nominal-ton*)	120	120	0	120	342	120	344	341
Supply Air Humidity (*lb/lb*)	max=0.0092	0.0088	.0086	.0085	.0079	.0073	.0122	.0102
Measured Water Consumption (*gpm*)	-	0.219	.139	.196	.207	.181	.096	.260
Water Consumption (*gal/sensible-credited-ton-hr*)	*Peak*: NA *Annual*: max = 4.00	2.65	2.46	5.60	7.34	2.13	2.33	4.14
External Static Pressure ("WC)	0.67	0.57	0.57	0.57	0.57	0.57	0.57	0.57