Emerging Technologies

Sub Wet-Bulb Evaporative Chiller

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Prepared by:

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Acknowledgments

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EXECUTIVE SUMMARY

Cooling loads constitute approximately 13% of the total demand for the United States, and in California, the hot dry summers drive cooling loads and peak demand throughout the season. Currently, the market is driven by compressor-based systems, which are inherently limited in efficiency. In California, because the climate is hot and dry, there is potential to expand the market to incorporate evaporative cooling. Most ordinary evaporative systems, such as cooling towers, are limited to cooling to the ambient wet-bulb, which limits their cooling capacity, and their ability to be used in a radiant system. The sub wet-bulb evaporative cooling (SWEC) technology has a significant advantage over other evaporative technologies because of its ability to cool below the ambient wet-bulb.

Several unique designs exist that are considered sub wet-bulb evaporative chillers, and this report focuses on one such design. The SWEC utilizes a two stage evaporative cooling system to chill water below the web-bulb temperature of the outdoor air. The theoretical limit for the supply water temperature is the dew point of the outdoor air.

The SWEC design tested in this evaluation offers the following potential benefits:

Chilling of supply water to lower temperatures than conventional cooling towers

Cooling efficiencies higher than a conventional mechanical chiller

No introduction of humidity to the building

Potential concerns of the technology include whether or not the chilled water temperatures are suitable for building cooling systems and the water consumption required for cooling.

The objectives of this assessment are to:

- Build an analytical model of the SWEC;
- Evaluate the performance of the SWEC in the laboratory under a range of environmental conditions and operating modes; and
- Validate the model using the laboratory data.

The third-party laboratory (aka lab) tested the SWEC in an environmental control chamber in order to map its performance characteristics. The SWEC was instrumented such that the load, energy consumption, and water-use can be determined. Outdoor air conditions and return water temperature were held at a steady state. The lab performed a comparative analysis of the SWEC tested in this evaluation to another SWEC design subsequently tested and highlighted strengths and weaknesses of each system.

The SWEC chiller tested for this report was able to consistently provide cooling loads efficiently, while operating under a variety of environmental, operational, and load-based conditions. The unit consistently provided anywhere from 0.7 to 2.6 tons of cooling, with 1-2 tons being consistently typical and average. The variation in coefficient of performance (COP) ranged from 8.5 to 33, with 15 to 25 being an average and typical value. The unit was able to consistently provide chilled water as low as 58°F, which is low enough to be used in a radiant system.

The results of the lab evaluation of the SWEC technology show that it has great potential to reduce energy use in hot dry climates. Although the technology shows great potential, there are also some significant barriers that need to be overcome. In general, radiant panels add significant costs to any installation, and don't have great market penetration. A fan coil unit can potentially replace a traditional evaporator coil, and therefore be more cost-effective than a radiant system.

It is recommended that further research be done in order to determine the cost-effectiveness of a SWEC with a thermal storage system and thermal distribution system using fan coil units in a residential building. The analysis should determine if a fan coil thermal distribution system can meet the load in a residential building in California climate zones. The analysis should also determine whether the addition of a thermal storage system makes economic sense as a peak demand reduction strategy.

ABBREVIATIONS AND ACRONYMS

BTU	British Thermal Unit
CFM	Cubic Feet per Minute
COP	Coefficient of Performance
DB	Dry Bulb
DP	Dew Point
GPM	Gallon per Minute
Hz	Hertz
NI	National Instruments
RTD	Resistance Temperature Device
SWEC	Sub Wet-bulb Evaporative Chiller
WB	Wet-bulb

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INTRODUCTION

This project seeks to complete a laboratory evaluation of a sub wet-bulb evaporative chiller (SWEC). The SWEC is an evaporative cooling technology that chills water in order to cool a building in combination with a radiant or fan coil distribution system. The unit tested in the laboratory was designed to provide 1-3 tons of cooling with variable capacity.

BACKGROUND

Cooling loads constitute approximately 13% of the total demand for the United States, and in California, the hot dry summers drive cooling loads and peak demand throughout the season. Currently, the market is driven by compressor-based systems, which are inherently limited in efficiency. In California, because the climate is hot and dry, there is potential to expand the market to incorporate evaporative cooling. Most ordinary evaporative systems, such as cooling towers, are limited to cooling to the ambient wet-bulb temperature, which limits their cooling capacity, and their ability to be used in a radiant system. The SWEC technology has a significant advantage over other evaporative technologies because of its ability to cool below the ambient wet-bulb.

While sub wet-bulb cooling does exist on the market, the SWEC technology offers the potential to advance beyond some of the challenges that the existing technology faces in the market. The SWEC has the potential to be used as a stand-alone chiller, with no backup refrigerant-based system, meaning that it can reduce some of the initial expense associated with evaporative cooling. Because evaporative coolers are often paired with a backup cooling unit, this offers a significant advantage to the SWEC. Other market barriers to the SWEC include the fact that retrofits can be cost-prohibitive, and that there is additional maintenance required for an evaporative cooler.

The goal of this project is to test the SWEC under various outdoor air conditions, and analyze the performance. The analysis provides insight to the potential for the SWEC to replace traditional compressor-based systems, and the reduction of peak demand associated with using a SWEC. The test points for the SWEC were based on expected conditions in the hot, dry climate zones of California. The outdoor air conditions covered common temperature and humidity conditions in California summers, and the return water temperatures covered expected return water temperatures for a residential or light commercial building with a fan coil distribution or radiant system installed.

ASSESSMENT OBJECTIVES

- The objectives of this assessment are to:
- Build an analytical model of the SWEC,
- Evaluate the performance of the SWEC in the laboratory under a range of environmental conditions and operating modes, and
- Validate the model using the laboratory data.

TECHNOLOGY DESCRIPTION

The sub wet-bulb evaporative chiller (SWEC) uses an evaporative cooling process to chill water for use in building cooling systems. The SWEC utilizes a two-stage evaporative cooling system to chill water below the web-bulb temperature of the outdoor air. The theoretical limit for the supply water temperature is the dew point of the outdoor air.

The SWEC has four independent air streams that each pass through a heat exchanger, an evaporative media, and a second heat exchanger (Figure 1). As air passes through the first heat exchanger it is sensibly cooled by the previous air stream exiting the unit. The result is a reduction in both the dry bulb temperature and wet-bulb temperature of the air stream. The chilled air then passes through an evaporative media which evaporatively cools the air and chills the water. After the air exits the evaporative media it precools the next air stream in the second heat exchanger and is exhausted.

The water used in the SWEC is returned from the building and is distributed over an evaporative media and flows into a sump on the outer perimeter of the unit. The water is collected in the outer sump and is distributed over the inner evaporative media by a pump in the SWEC. The flow rate from the pump is balanced by means of a valve inside the SWEC that is adjusted to match the supply flow rate. The water is then pumped from the inner sump to the building. This two stage process is designed to provide a lower supply water temperature than would be achieved with a one stage process.

The SWEC design offers the following potential benefits:

- Chilling of supply water to lower temperatures than conventional cooling towers.
- Cooling efficiencies much higher than a conventional mechanical chiller.
- No introduction of humidity to the building. Because the air in the SWEC does not interact with the conditioned building space, no humidity is introduced to the building.

Potential concerns of the technology include whether or not the supply water temperatures are suitable for building cooling systems, and the impact of consumed water to provide for cooling.

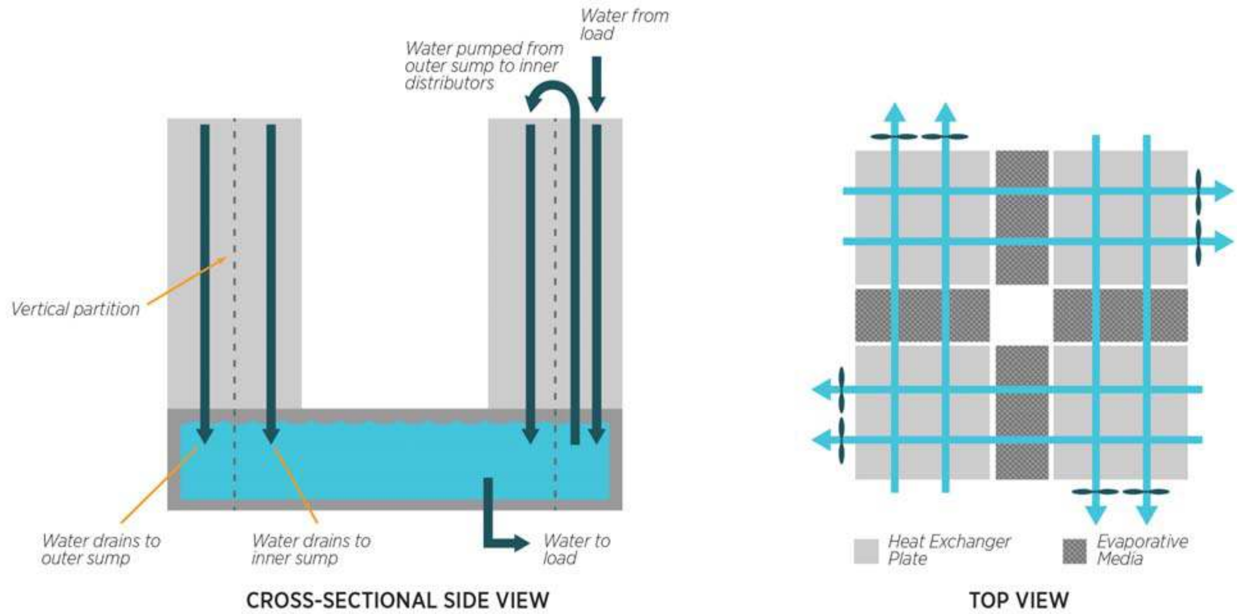

FIGURE 1. DIAGRAM OF SWEC CONFIGURATION

TECHNICAL APPROACH/TEST METHODOLOGY

OVERVIEW

The prototype SWEC was built by the inventor and proprietor of the technology and shipped to a third-party for testing in their laboratory. The unit was instrumented by the testing lab to collect data to determine power consumption, water consumption, cooling capacity, and efficiency for a selection of environmental conditions and equipment modes of operation. The test data was collected, analyzed, and compared to a model developed by WCEC to predict the unit's performance over a wide-range of environmental conditions.

TEST PLAN

The SWEC was installed in the test lab environmental chamber to simulate desired outdoor air conditions. A load rig was constructed to re-heat and deliver a constant return water temperature to the SWEC.

The SWEC was tested at two outdoor air conditions with a variety of air flow rates, return water flow rates, and return water temperatures. Air flow rates were varied between 2250 to 900 cubic feet per minute (CFM) in 450 CFM increments. Return water temperatures varied from 65°F to 83°F in increments of 3°F, while return water flow rates varied between 3 to 6 gallons per minute (GPM) in 1 GPM increments. The test matrix is described in Table 1, and Table 2 shows how the test numbers were grouped for analysis to evaluate the effect of each independent variable.

TABLE 1. CHILLER TEST POINTS

TEST	AMBIENT TEMPERATURES (°F DB/0F WB)	RETURN WATER TEMPERATURE (°F)	AIR FLOW (CFM)	WATER FLOW (GPM)
1	90/64	71	1800	4
2	90/64	71	1350	4
3	90/64	71	900	4
4	90/64	71	2250	4
5	90/64	71	2250	5
6	90/64	71	2250	6
7	90/64	71	2250	3
8	90/64	74	2250	3
9	90/64	74	2250	4
10	90/64	74	2250	5
11	90/64	74	2250	6
12	105/73	71	1800	4
13	90/64	77	1800	4
14	105/73	74	1800	4
15	105/73	77	1800	4
16	105/73	80	1800	4
17	105/73	83	1800	4
18	90/64	65	1800	4
19	90/64	68	1800	4
20	90/64	74	1800	4

TABLE 2. VARIATION OF PARAMETERS

PARAMETER VARIED	TEST NUMBER	PARAMETER VARIED	TEST NUMBER
Air Flow Variation (90/64) Tw = 71	1	Water Temperature Variation (105/73)	12
	2		14
	3		15
	4		16
Water Flow Variation (90/64) Tw = 71	4		17
	5	Water Temperature Variation (90/64)	13
	6		18
	7		19
Water Flow Variation (90/64) Tw = 74	8		20
	9		
	10		
	11		

INSTRUMENTATION PLAN

A load cart capable of producing over 5 tons of load was placed into the chamber for all tests (Figure 2). The load cart provided a constant return water temperature to the SWEC. The set-point was maintained by a controller that modulated a proportional valve controlling the flow of hot water from a boiler to one side of a water-to-water heat exchanger. The supply water from the SWEC was pumped to the other side of the water-to-water heat exchanger. The flow rate of the water through the SWEC was measured by a pulse-output paddle wheel in-line flow meter.

The SWEC was installed into the environmental chamber (Figure 3). During the testing, the exhaust from each fan was ducted to a plenum on top of the SWEC which was then ducted to the central chamber exhaust. The intake to the fans was not restricted. The fans on the SWEC were powered by an Extech® (Model 382275) direct current power supply. The procedure to set the flow rate for test was to:

1. Measure the current and pressure drop across the fans at several fan speeds without the exhaust ducting attached
2. Attach the exhaust ducting
3. Set the speed of the environmental chamber fan and the SWEC fans such that the desired flow rate was achieved while matching the current and pressure drop across the SWEC fans as measured in step 1.

The water was pumped through the SWEC using both an internal pump and an external pump. The procedure to set the flow rates of the two loops was to:

1. Set the external loop to desired flow rate and roughly match the internal pump to this rate by setting a flow control valve.
2. Observe changes in the sump levels over a 5-minute time period.
3. Readjust internal SWEC valve according to observation in step 2.
4. Repeat steps 2 and 3 until sump levels stabilize within accepted limit (+/- ½ inch change over five minutes).

The SWEC was instrumented with temperature sensors in the supply and return water, in the sump water, and in the distributor lines (Figure 4). The temperature and absolute humidity air entering and leaving the environmental chamber was measured. The fan power consumed by the SWEC was measured using a true-power meter and current draw of the pump was measured with a current transducer. The water flow rate through the SWEC and the makeup water flow rate was measured using a pulse-output paddle wheel in-line flow meter. The airflow rate through the SWEC was measured using calibrated flow nozzles.

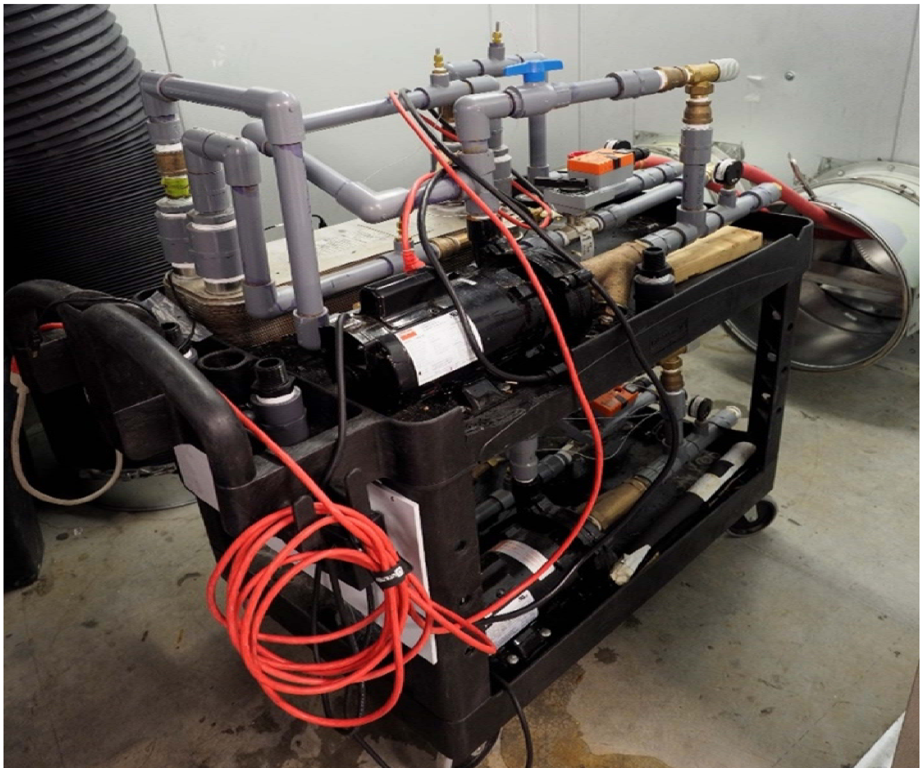

FIGURE 2. LOAD CART INSTALLED IN ENVIRONMENTAL CHAMBER

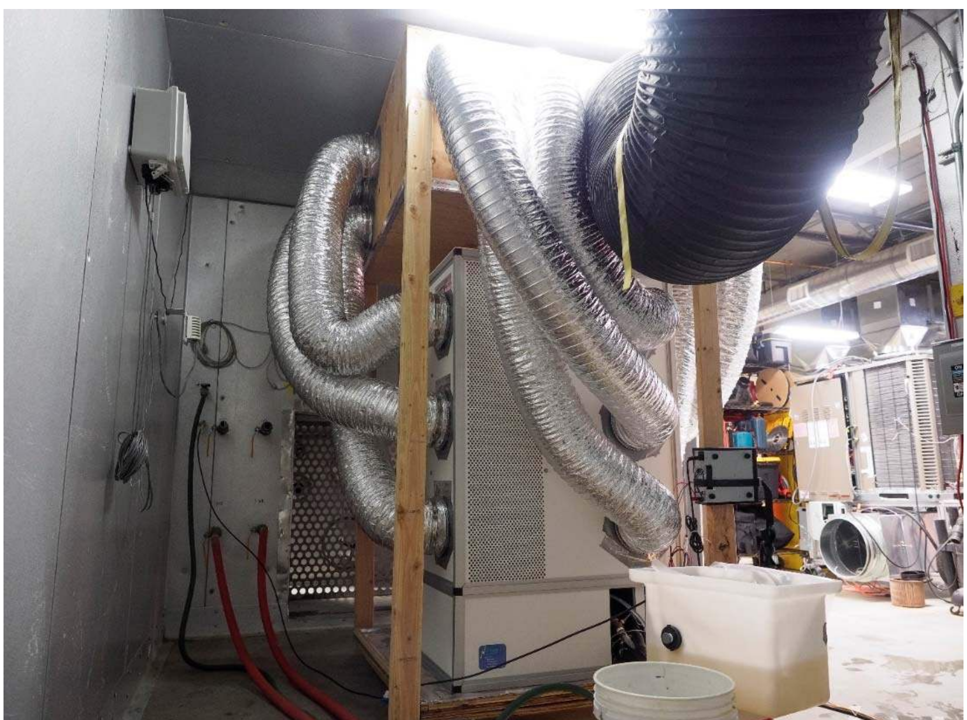

FIGURE 3. SWEC INSTALLED IN THE ENVIRONMENTAL CHAMBER

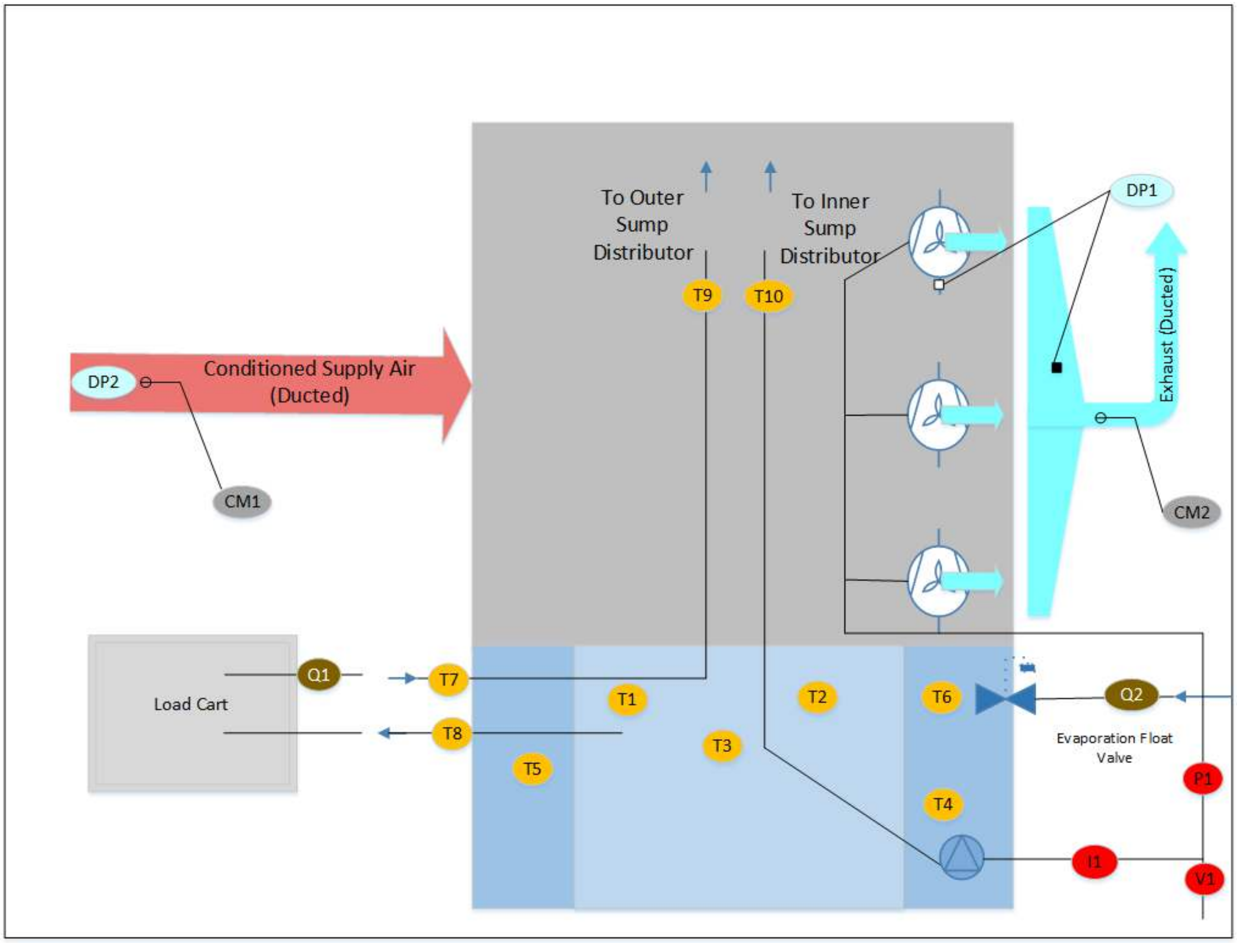

FIGURE 4. DIAGRAM OF SWEC INSTRUMENTATION LOCATIONS. INSTRUMENTATION LABELS CORRESPOND TO TABLE 3.

Table 3. Instrumentation models and accuracy

Item #	Measurement Type	Manufacturer Model #	Target Range	Accuracy	Calibration Date	DAQ Information
CM1	Hot Chamber Inlet Dry Bulb/Dew Point	GE Optisonde 2-1-1-1-1-0-0-0	(90 to 105)/ (46 to 57) (^{0}F)	(+/- 0.26)/ (+/- 0.36) (^{0}F)	03/2012	RS-232
CM2	Hot Chamber Exit Dry Bulb/Dew Point	GE Optisonde 2-1-1-1-1-0-0-0	(75 to 83)/ (60 to 67) (^{0}F)	(+/- 0.26)/ (+/- 0.36) (^{0}F)	01/2014	RS-232
T1	Inner Sump Temperature	Omega HSRTD-3-100-A-120-E	55 to 70 (^{0}F)	+/- 0.35 ^{0}F		NI 9217
T2	Inner Sump Temperature	Omega HSRTD-3-100-A-120-E	55 to 70 (^{0}F)	+/- 0.35 ^{0}F		NI 9217
T3	Inner Sump Temperature	Omega HSRTD-3-100-A-120-E	55 to 70 (^{0}F)	+/- 0.35 ^{0}F		NI 9217
T4	Outer Sump Temperature	Omega HSRTD-3-100-A-120-E	55 to 85 (^{0}F)	+/- 0.4 ^{0}F		NI 9217
T5	Outer Sump Temperature	Omega HSRTD-3-100-A-120-E	55 to 85 (^{0}F)	+/- 0.4 ^{0}F		NI 9217
T6	Outer Sump Temperature	Omega HSRTD-3-100-A-120-E	55 to 85 (^{0}F)	+/- 0.4 ^{0}F		NI 9217
T7	SWEC Inlet Temperature	Omega RTD-NPT-72-E	55 to 90(^{0}F)	+/- 0.4 ^{0}F		NI 9217
T8	SWEC Outlet Temperature	Omega RTD-NPT-72-E	55 to 70 (^{0}F)	+/- 0.35 ^{0}F		NI 9217
T9	Outer Sump Distributor Temperature	Omega RTD-NPT-72-E	55 to 90 (^{0}F)	+/- 0.4 ^{0}F		NI 9217
T10	Inner Sump Distributor Temperature	Omega RTD-NPT-72-E	60 to 75 (^{0}F)	+/- 0.35 ^{0}F		NI 9217
Q1	Load Cart Flow Rate	Omega FTB 4607	3 to 6 (gpm)	+/-1.5% of reading		NI-USB-6009
DP1	SWEC Fan Differential Pressure	Energy Conservatory DG-500	0 to 45 (Pa)	+/- 1% of reading		RS-232
Q2	Evaporation Water Flow Rate	Omega FTB 4705	0.05 to 0.15 (gpm)	+/- 1% of rdg (from 0.2 – 10 (gpm))	07/2013	NI PCIe 6321
I1	System Current	Dent Power Scout 3+	0 to 5 Amps	+/- 1% of reading		RS-485
V1	System Voltage	Dent Power Scout 3+	~120 Volts	+/- 1% of reading		RS-485

P1	System Power	Dent Power Scout 3+	50 – 300 Watts	+/-1% of reading		RS-485
DP2	Flow Nozzle Differential Pressure	Energy Conservatory APT 8	100-500 (Pa)	+/- 1% of reading		RS-232

Water Temperature Measurements

The temperatures in the sumps and the inlet and outlet pipes and the distributor manifolds were measured and recorded. Resistance temperature detectors (RTDs) were installed in-line for the inlet and outlet pipes as well as in the distributor manifolds. The distributor manifolds supply the water to the evaporative media on both the inner and outer sumps. There are four distributors for each sump so the temperature probe was placed right before the manifold for each sump. Each sump was equipped with three hermetically sealed RTDs and the reported sump temperature was the average temperature of the three RTDs in each sump.

Evaporative Water Supply Measurements

The water source available at the laboratory was purified using a reverse osmosis system prior to use to prevent scale accumulation in the sump or evaporative media. In order to ensure a consistent sump level the makeup water was controlled by a float valve. The flow rate of the makeup water was measured, but not controlled, as it was a function of the SWEC operation.

The flow rate of the evaporation water was measured using an inline low flow meter and validated by measuring the differential in the dew points between the inlet and exhaust air measurements. The flowmeter was calibrated to be within 1% of the flow reading from 0.2 to 10 GPM. Since the makeup water rate was typically only about 0.1 GPM, the actual accuracy may be different than the calibrated accuracy.

The water flow meters used were a paddle wheel, pulse output design, where the flow rate is proportional to the frequency of the pulsed signal. Pulses were counted, converted to flow rate using the manufacturer-reported conversion factor, and recorded.

Differential Pressure and Airflow Measurements

The differential and static pressures for the environmental chamber were recorded using an Energy Conservatory APT-8 pressure transducer with eight differential pressure channels. The device was used to measure differential pressure across the nozzle box and the static pressure upstream of the nozzles to calculate airflow for the chamber according to ANSI/ASHRAE 41.2-1987 [1]. The differential pressure across the SWEC exhaust fans was measured using an Energy Conservatory DG-500.

Chamber Conditions Measurements

During all tests the inlet and exit conditions of the chamber were monitored with two GE OptiSonde™ chilled mirror hygrometers. These sensors use an RTD to measure dry bulb temperature and air from a chilled mirror hygrometer to measure dew point. Wet-bulb temperature is then calculated from the dry bulb temperature and dew point. Data is digitally output via serial interface every second.

Power Measurements

Measurements for the fan power, pump current, and pump/fan voltage were recorded using a PowerScout 3 Plus with a serial interface and Modbus® protocol. It digitally outputs data every three seconds. Additionally, the current draw of the fans was controlled by an Extech 382275 switching mode power supply. The current set-point was controlled to achieve the desired flow rate for the test.

Data Acquisition System

All signals were acquired using National Instruments (NI) hardware at 0.3 hertz (Hz) or greater, averaged every 30 seconds using LabVIEW software, and logged to a text file.

Test Tolerances

For each test, all of the testing condition parameters, including outdoor air dry bulb and wet-bulb temperatures, inlet water temperatures, water flow rate, and air flow rate were controlled to stay within the required range (Table 4). The set-point had to stay within the required range tolerance for the entire 30-minute test. The last 15 minutes of data were then averaged. The mean for each test point condition had to fall within the mean tolerance to be considered a valid test. For example, for a test condition at 90°F outdoor air dry bulb temperature, the temperature was required to be between 88°F-92°F for the 30-minute test, and the average of the last 15 minutes had to be between 79.5°F-80.5°F.

Table 4. Test tolerances

Test Condition	Range Tolerance	Mean Tolerance
Outdoor Air Dry Bulb Temp	±2°F	±1/2°F
Outdoor Air Wet-bulb Temp	±2°F	±1/2°F
Inlet Water Temp	±2°F	±1.5°F
Water Flow Rate	+/-.35 GPM	+/-.25 GPM
Air Flow Rate	+/-100 CFM	+/-30 CFM

Data Analysis

The data was analyzed to calculate the cooling capacity and the coefficient of performance (COP) of the SWEC. Both metrics are important because they describe the performance of the unit as well as its efficiency. The uncertainties of the calculation of cooling capacity and COP are also reported.

CAPACITY

The capacity of the SWEC was calculated for each test using Equation 1:

EQUATION 1. CAPACITY

$$\dot{q} = \dot{m} \times c_p \times (T_{w,in} - T_{w,out})$$

where $T_{w,in}$ and $T_{w,out}$ are the inlet and outlet water temperatures respectively in degrees Fahrenheit, $\dot{m}$ is the mass flow rate of water in lb/hr, and c_p is the specific heat of water in BTU/(lb °F).

COEFFICIENT OF PERFORMANCE

The coefficient of performance of the unit was calculated for each test using Equation 2.

EQUATION 2. COEFFICIENT OF PERFORMANCE

$$COP = \frac{\dot{q}}{P_{Fans} + P_{Pump}}$$

where $\dot{q}$ is the capacity of the SWEC in BTU/hr and the total power consumption is the sum of the fan power, P_{Fans} , and the pump power,P_{Pump}, converted from Watts to BTU/hr.

UNCERTAINTY ANALYSIS

The uncertainty of the capacity, power draw, and coefficient of performance was calculated using the sequential perturbations method[1] which is a widely accepted numerical method in which a finite difference is used to approximate the sensitivity of the value to the possible error in its dependent measurements. The method involves first calculating a desired value, R_o, and perturbing the value one independent variable at a time. First the perturbations are performed by adding the greatest possible uncertainty to each variable to find R_i^+ for each independent variable. Next, the perturbation is performed by subtracting the uncertainty from each independent variable to find R_i^-. The average perturbation value, δR_i, for each independent variable is found by averaging the absolute difference between R_o and the perturbation values, R_i^+ and R_i^-. Finally, using Equation 3the total uncertainty is calculated for each value.

EQUATION 3. SEQUENTIAL PERTURBATION

$$U_R = \pm \left[\sum_{i=1}^{L} \left(\delta {R_i}^2 \right) \right]^{1/2}$$

NUMERICAL MODEL

In order to further advance the technology development a numerical model was developed in MATLAB using an iterative scheme and basic heat transfer equations. The model was developed in order to calculate COP, capacity, water use, and supply water temperature. The model was refined and validated through the testing of the SWEC as well as individual testing of the evaporative media and the heat exchanger.

MODEL METHODS

In order to develop the model, separate tests were conducted to determine the water-cooling effectiveness of the evaporative media and the heat transfer effectiveness of the air-to-air heat exchangers. Two individual laboratory tests were conducted to determine that:

1. The plate heat exchanger effectiveness (ε) is a function of flow rate and can be modeled as the function $\varepsilon_{DRY} = -0.0012 \times V_{air} + 0.72$ where V_{air} is the velocity of the air in feet/min [2].
2. In the evaporative media, the potential for transferring heat from the water to the air is the difference between the wet-bulb temperature of the air and the temperature of the water. Laboratory testing showed that the evaporative media transferred 90% of this heat ($\varepsilon_{WET} = 0.90$) [3]. Furthermore, the evaporative cooling process was completed with 80% evaporative effectiveness ($\varepsilon_{EVAP} = 0.80$).

Because each quadrant of the SWEC (Figure 1) is identical in materials, air flows, and water flows; for simplicity, only one quadrant is modeled. However, each quadrant actually consists of four quadrants, due to the fact the chilled water sump has a colder "inner" section and a warmer "outer" section. The warm water from the building is returned to the outer section, which is then cooled and is the source of water for the inner section. The result is four distinct air flows of different temperatures through the four quadrants of each air-to-air heat exchanger labeled "A" "B" "C" and "D" (Figure 5).

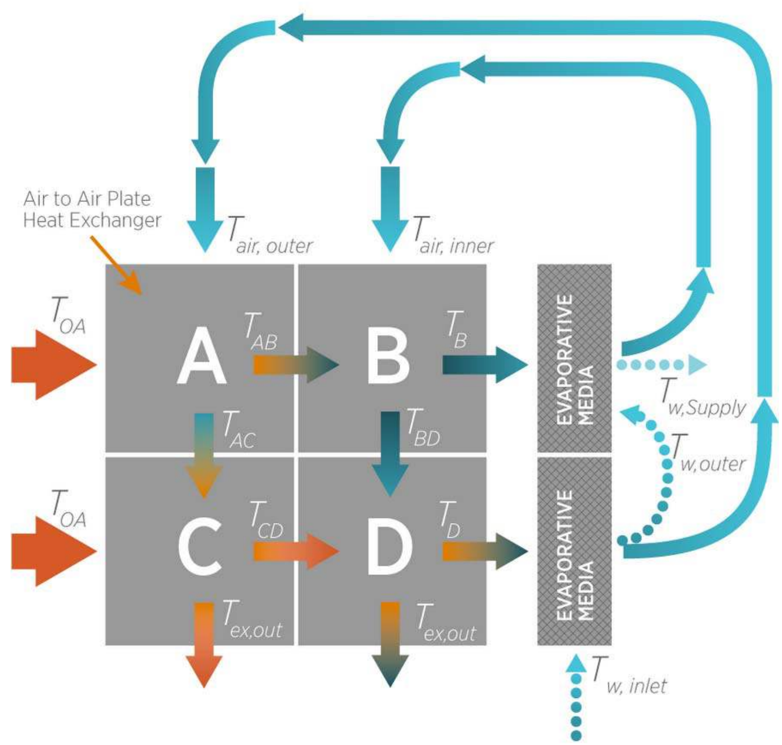

FIGURE 5. ONE QUADRANT OF THE SWEC AS MODELED

Because the calculation of some variables are dependent on future calculations, the solution must be iterated. The outdoor air temperature entering the hot side of the heat exchanger, T_{OA}, is known. Initial conditions for the air entering the cold side of the heat exchanger $T_{air,inner(i)}$ and $T_{air,outer(i)}$, are set for the first iteration as shown in Equation 4. Also, the temperature of the water in the outer sump is initially set to equal the air temperature leaving the outer evaporative media.

EQUATION 4. INITIAL CONDITIONS

$$T_{air,outer(i)} = Ambient\ Wet\ Bulb\ Temperature - 5°F$$

$$T_{w,outer(i)} = T_{air,outer(i)}$$

$$T_{air,inner(i)} = Ambient\ Wet\ Bulb\ Temperature - 10°F$$

First the air temperatures leaving quadrant A and B are calculated using Equation .

EQUATION 5. AIR TEMPERATURES FOR QUADRANTS A AND B

$$T_{AB\ (i)} = T_{OA} - (T_{OA} - T_{air,outer(i)}) \times \varepsilon_{DRY}$$

$$T_{AC\ (i)} = T_{air,outer(i)} + (T_{OA} - T_{air,outer(i)}) \times \varepsilon_{DRY}$$

$$T_{B(i)} = T_{AB(i)} - \left(T_{AB(i)} - T_{air,inner(i)}\right) \times \varepsilon_{DRY}$$

$$T_{BD(i)} = T_{air,inner(i)} + \left(T_{AB\ (i)} - T_{air,inner(i)}\right) \times \varepsilon_{DRY}$$

Where:

T_{OA} is the temperature of the outdoor air,

$T_{AB(i)}$ is the temperature of air leaving quadrant A and entering quadrant B,

$T_{AC(i)}$ is the iterated temperature of the air leaving quadrant A and entering quadrant C,

$T_{B(i)}$ is the temperature of air leaving quadrant B,

$T_{BD(i)}$ is the temperature of air leaving quadrant B and entering quadrant D,

$T_{air,outer(i)}$ is the temperature of the outer section,

$T_{air,inner(i)}$ is the temperature of the inner section, and

ε_{DRY} is the efficiency of the air-to-air heat exchanger.

Next the air temperatures leaving quadrant C and D are calculated using Equation 5:

EQUATION 5. AIR TEMPERATURES FOR QUADRANTS C AND D

$$T_{CD(i)} = T_{OA} - \left(T_{OA} - T_{AC(i)}\right) \times \varepsilon_{DRY}$$

$$T_{D\ (i)} = T_{CD(i)} - \left(T_{CD(i)} - T_{BD(i)}\right) \times \varepsilon_{DRY}$$

where $T_{CD(i)}$ is the temperature of air leaving quadrant C and entering quadrant D and $T_{D\,(i)}$ is the temperature leaving quadrant D.

A second set of iterative equations (Equation 6) is required to solve the heat transfer process occurring inside the evaporative media. The psychrometric functions required for the calculation make an explicit solution impractical. The solution is iterated until a steady state value is reached.

EQUATION 6. AIR TEMPERATURES INSIDE THE EVAPORATIVE MEDIA

$$T^{D}_{heat\,(j)} = T_{D(i)}$$

$$T^{D}_{heat,WB\,(j)} = T_{D,WB(i)}$$

$$T^{D}_{heat\,(j+1)} = T^{D}_{heat\,(j)} + \varepsilon_{WET} \times \dot{m}_w \times c_{P,w} \times \left(T_{w,inlet} - T^{D}_{heat,WB\,(j)}\right) / (c_{P,a} \times \dot{m}_a)$$

$$T^{D}_{heat,WB\,(j+1)} = psych(P_a, T^{D}_{heat\,(j)}\,, T_{D(i),Dew\ Point})$$

$$T^{B}_{heat\,(j)} = T_{B(i)}$$

$$T^{B}_{heat,WB\,(j)} = T_{B,WB(i)}$$

$$T^{B}_{heat\,(j+1)} = T^{B}_{heat\,(j)} + \varepsilon_{WET} \times \dot{m}_w \times c_{P,w} \times \left(T_{w,outer\,(i)} - T^{B}_{heat,WB\,(j)}\right) / (c_{P,a} \times \dot{m}_a)$$

$$T^{B}_{heat,WB\,(j+1)} = psych(P_a, T^{B}_{heat\,(j)}, T_{B\,(i),Dew\ Point})$$

In the set of equation solve heat transfer inside the evaporative media, T_{heat} is the heated air temperature of the air after the heat transferred from the water to the air has been accounted for but before the evaporative cooling process has been applied. In reality, the heat transfer from water to air and the evaporative cooling processes happen concurrently, but they are separated here for ease of calculation. In the equations the subscript "WB" designates the wet-bulb temperature and the superscript "B" and "D" designate the inner and outer portions of the media. Furthermore:

ε_{WET} is the sensible heat transfer effectiveness from the water to the air,

$\dot{m}_w$ is the water flow rate,

$\dot{m}_a$ is the air flow rate, $C_{p,a}$ is the specific heat of air,

$C_{p,w}$ is the specific heat of water,

$T_{w,inlet}$ is the inlet water temperature to the SWEC,

$T_{w,outer}$ is the water temperature from the outer sump fed to the inner sump, and

psych is a psychrometric calculator solving for the wet-bulb temperature of the heated air at the inlet air dew-point at the ambient pressure P_a.

Next the air and water temperatures leaving the evaporative media are calculated from Equation 7, the result which is used in the subsequent iterations starting with Equation .

EQUATION 7. AIR TEMPERATURES INSIDE THE EVAPORATIVE MEDIA

$$T_{w,outer\ (i+1)} = T^{D}_{heat,WB} + (1 - \varepsilon_{WET}) \times \left(T_{w,inlet} - T^{D}_{heat,WB}\right)$$
$$T_{w,supply\ (i+1)} = T^{B}_{heat,WB} + (1 - \varepsilon_{WET}) \times \left(T_{w,outer\ (i+1)} - T^{B}_{heat,WB}\right)$$
$$T_{air,outer\ (i+1)} = T^{D}_{heat} - (\varepsilon_{EVAP}) \times \left(T^{D}_{heat} - T^{D}_{heat,WB}\right)$$
$$T_{air,inner\ (i+1)} = T^{B}_{heat} - (\varepsilon_{EVAP}) \times \left(T^{B}_{heat} - T^{B}_{heat,WB}\right)$$

Once the equilibrium of system has been determined, meaning that the iteration process reaches a steady state value, the performance of the system is calculated from Equation 8.

EQUATION 8. AIR TEMPERATURES INSIDE THE EVAPORATIVE MEDIA

$$Capacity = \left(T_{w,inlet} - T_{w,supply}\right) \times C_{P_w} \times \dot{m}_w \times 4$$

$$COP = Capacity\ /\ Power$$

$$Water\ Use = \left(\frac{(\emptyset_{air,outer(i)} - \emptyset_{D(i)}) + (\emptyset_{air,inner(i)} - \emptyset_{B(i)})}{2}\right) \times \dot{m}_a \times 4$$

where $Capacity$ is the system capacity, COP is the coefficient of performance, $T_{w,return}$ and $T_{w,supply}$ are the temperatures of the return and supply water respectively, and $\emptyset$ is the absolute humidity ratio of the air stream notated by the sub-script. The multiplier of 4 is included because the model is of one of four symmetric quadrants.

RESULTS

The detail results of the all tests conducted are tabulated in the Appendix

Test	Ambient Condition (DB/WB) (°F)	Air Flow (CFM)	Water Flow (GPM)	Return Water Temp °F	Supply Water Temp °F	Power (Watts)	Water Use Gal/ (Ton*hr)	Capacity (Tons)	Capacity Uncertainty (Tons)	COP	COP Uncertainty
1	90/64	1797	4.07	71	60.8	266	3.02	1.74	9.39E-02	23.1	1.26
2	90/64	1377	4.07	71	61.9	208	2.87	1.54	9.31E-02	25.9	1.59
3	90/64	896	4.07	71	64.1	168	2.77	1.19	9.19E-02	24.8	1.94
4	90/64	2234	4.00	71	60.1	388	3.41	1.80	9.27E-02	16.3	0.85
5	90/64	2239	4.95	71	61.4	384	3.49	1.95	1.13E-01	17.9	1.05
6	90/64	2234	6.12	71	61.9	394	3.20	2.29	1.40E-01	20.4	1.26
7	90/64	2262	2.83	71	58.8	382	3.80	1.43	6.63E-02	13.2	0.62
8	90/64	2260	2.80	74	58.7	383	3.31	1.79	6.77E-02	16.4	0.64
9	90/64	2248	4.12	74	61.0	349	2.97	2.25	9.74E-02	22.6	1.00
10	90/64	2241	5.02	74	61.8	322	2.90	2.54	1.18E-01	27.8	1.30
11	90/64	2256	5.87	74	62.2	383	2.75	2.91	1.37E-01	26.7	1.28
12	105/73	1793	4.04	71	66.4	315	6.11	0.76	9.03E-02	8.5	1.01
13	90/64	1797	4.13	77	62.4	256	2.53	2.41	9.85E-02	33.1	1.38
14	105/73	1801	4.02	74	67.7	254	4.90	1.02	9.03E-02	14.1	1.26
15	105/73	1802	4.10	77	68.2	252	3.66	1.45	9.33E-02	20.2	1.31
16	105/73	1800	4.07	80	68.5	252	3.12	1.91	9.46E-02	26.6	1.34
17	105/73	1800	4.03	83	69.2	251	3.01	2.11	9.47E-02	29.6	1.35
18	90/64	1793	4.00	65	60.0	268	5.65	0.77	8.94E-02	10.1	1.18
19	90/64	1793	4.17	68	60.3	269	3.22	1.50	9.52E-02	19.6	1.25
20	90/64	1812	4.15	74	61.8	267	2.21	2.09	9.71E-02	27.6	1.30

Table 6). A summary of the laboratory test results is compared to modeling results (Error! Reference source not found. - Figure 9) to characterize the performance of the SWEC and demonstrate the capabilities of the model. The analysis includes comparisons between the model and the actual results shows that the model is able to effectively predict within a reasonable range of error, the performance characteristics of the SWEC, and can therefore be used to optimize its performance.

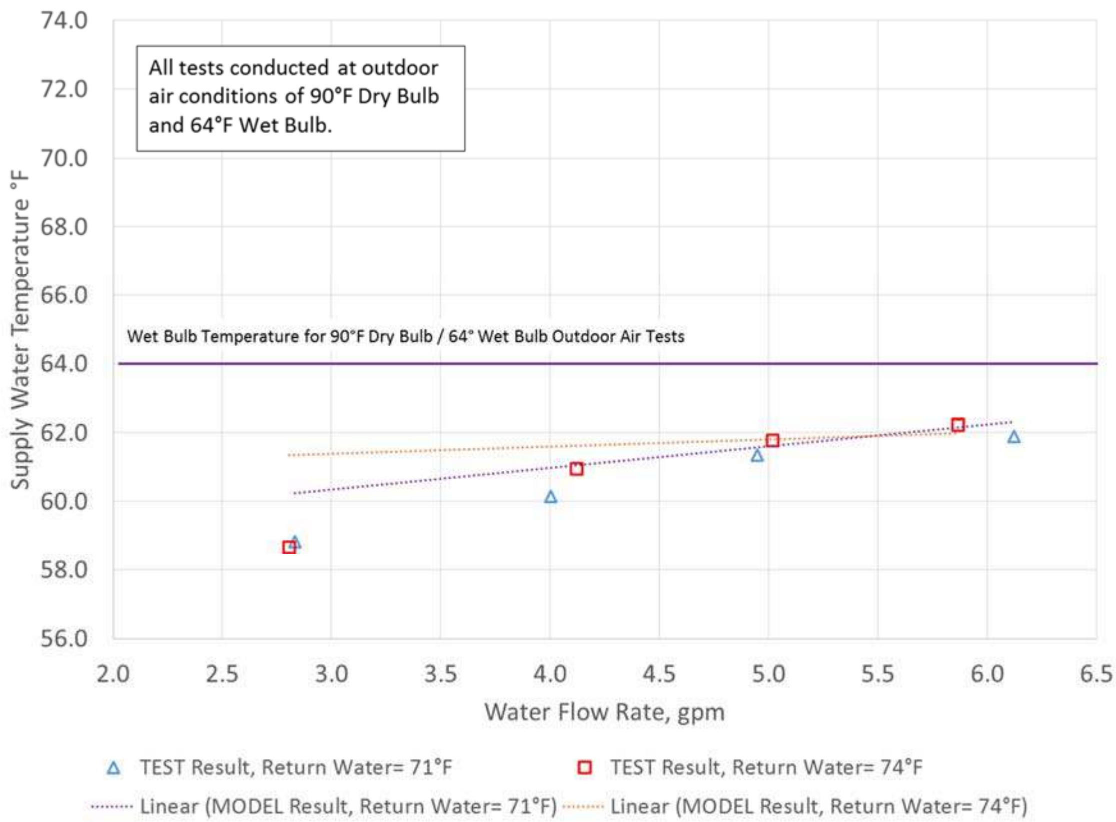

FIGURE 6. SUPPLY WATER TEMP VS WATER FLOW RATE

Error! Reference source not found. compares eight test results at different operating conditions to the model results. The outdoor air conditions for the tests are 90°F dry bulb and 64°F wet-bulb. One series of four tests is run at a return water temperature of 71°F and the other is run at 74°F. For each data series the varied parameter was the water flow rate. As the flow rate of water is increased with constant return water temperature, the temperature differential of the water is decreased, and the supply water temperature is decreased.

The standard deviation of the difference in the supply water temperature between the measured lab result and the modeled result for the 71°F and 74°F tests was found to be 1.0°F and 1.7°F, respectively.

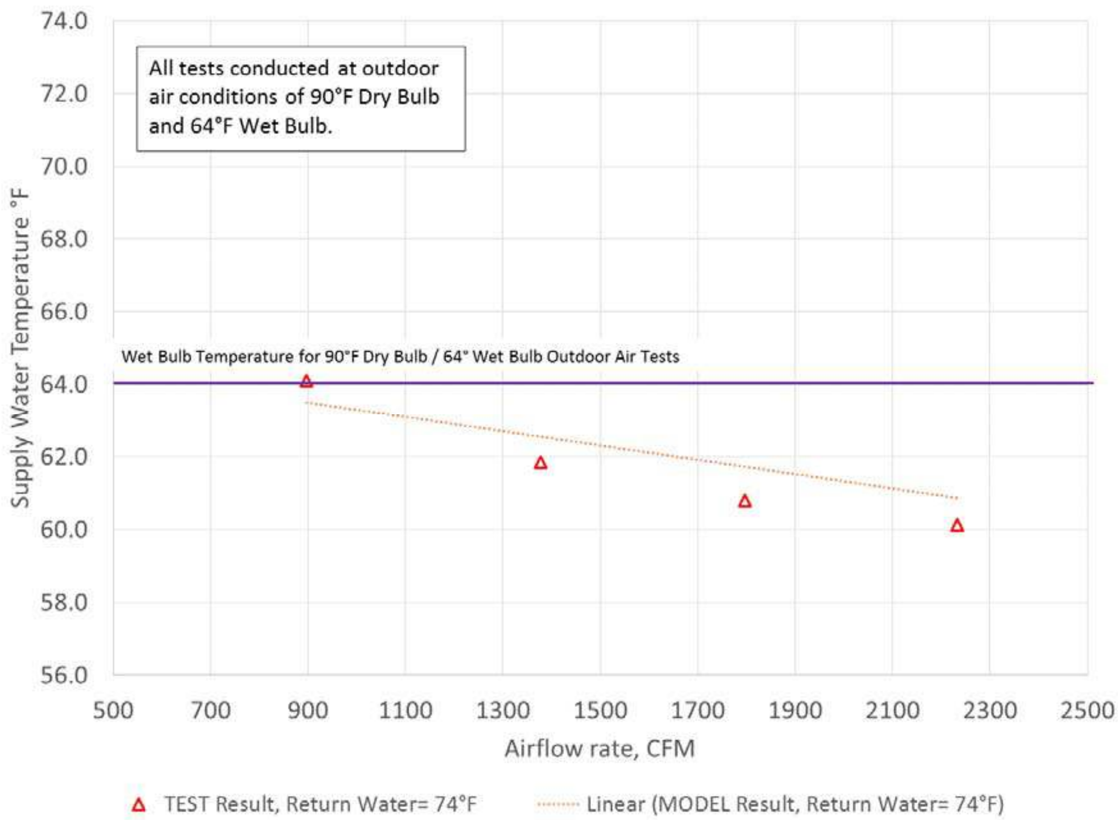

FIGURE 7. SUPPLY WATER TEMP VS AIR FLOW RATE

Figure 7illustrates the effect of air flow variation on the supply water temperature. The graph illustrates the model’s ability to predict SWEC performance as a function of air flow rates and demonstrates that a substantial change in air flow within the operating range of the SWEC produces results consistent with the results from the model. The standard deviation of the difference in the supply water temperature between the measured lab result and the modeled results for the tests shown in Figure 7 is 0.8°F.

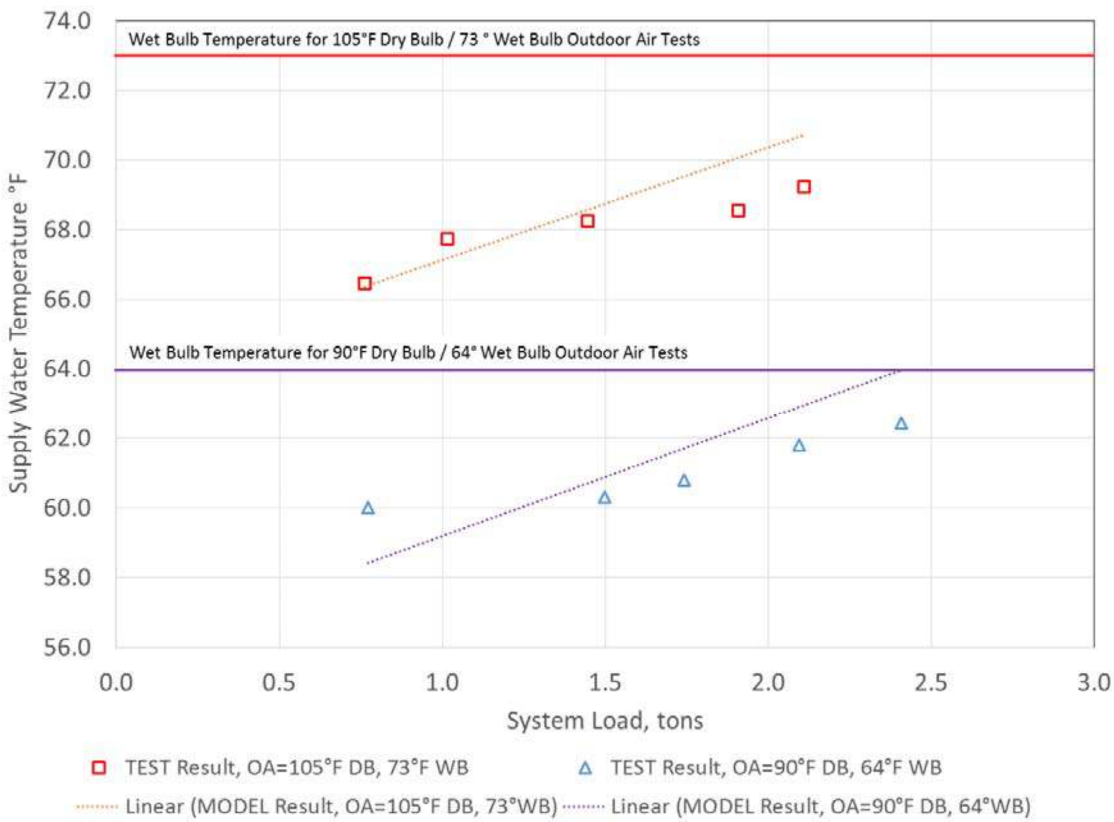

FIGURE 8. SYSTEM LOAD VS SUPPLY WATER TEMP

Figure 8shows the system load versus the supply water temperature for both the model and the experimental data. The data shows that the SWEC is able to produce a lower supply water temperature at the expense of reducing the cooling capacity, and vice versa. Additionally, the data shows that as the ambient wet-bulb temperature is increased the supply water temperature increases. The data suggests that the chilling temperature should be optimized to consider capacity in any real-world applications. The data also shows that the model is able to closely match the system load and supply water temperature for several conditions. The standard deviation of the difference in the supply water temperature between the measured lab result and the modeled results is 1.1°F for the 105°/73°F ambient conditions and 1.3°F for the 90°/64°F ambient conditions.

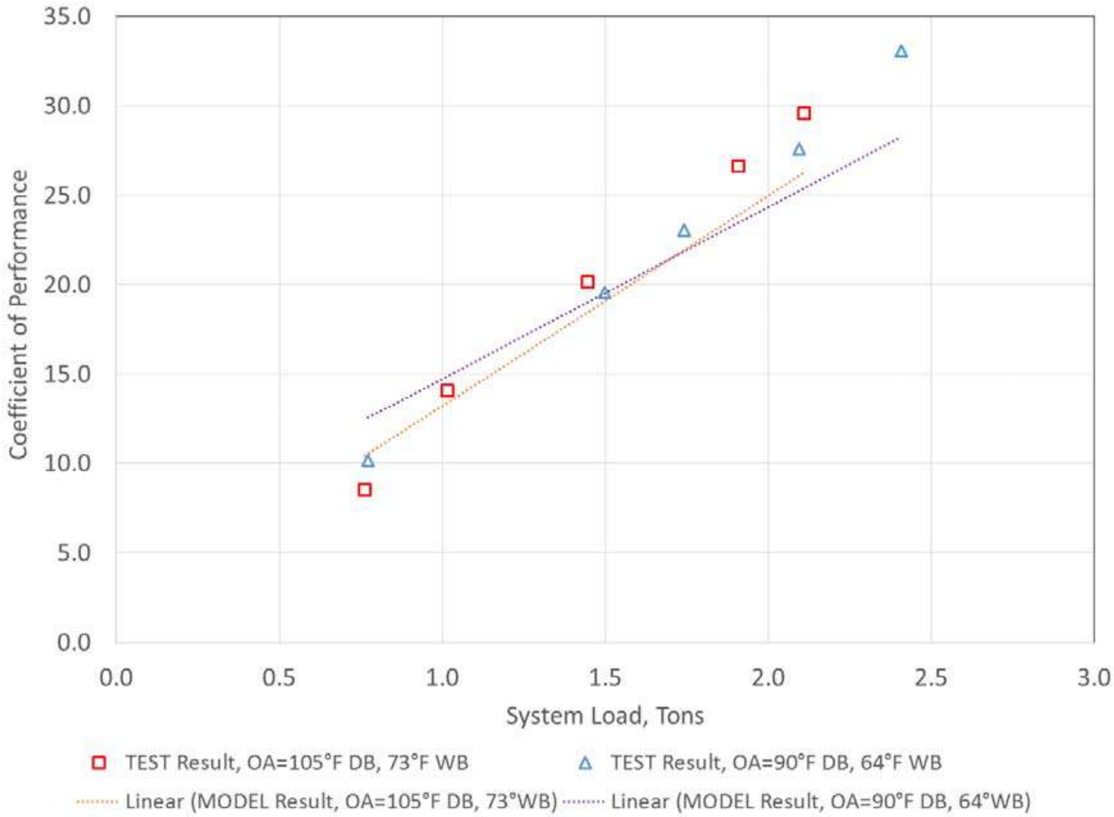

FIGURE 9. SYSTEM LOAD VS. COP

Figure 9 compares the system load to the COP of the system. Figure 9 shows that as the capacity of the unit is increased, its COP also increases. Combining the results from Figure 8 and Figure 9 show that as the supply water temperature increases both capacity and COP increase. The standard deviation of the difference in the supply water temperature between the measured lab result and the modeled results is 2.5°F for the 105°/73°F ambient conditions and 3°F for the 90°/64°F ambient conditions. Since the model uses the experimental power use to calculate the COP, the COP variation seen in Figure 9 is due completely to differences in the experimental and model load analysis.

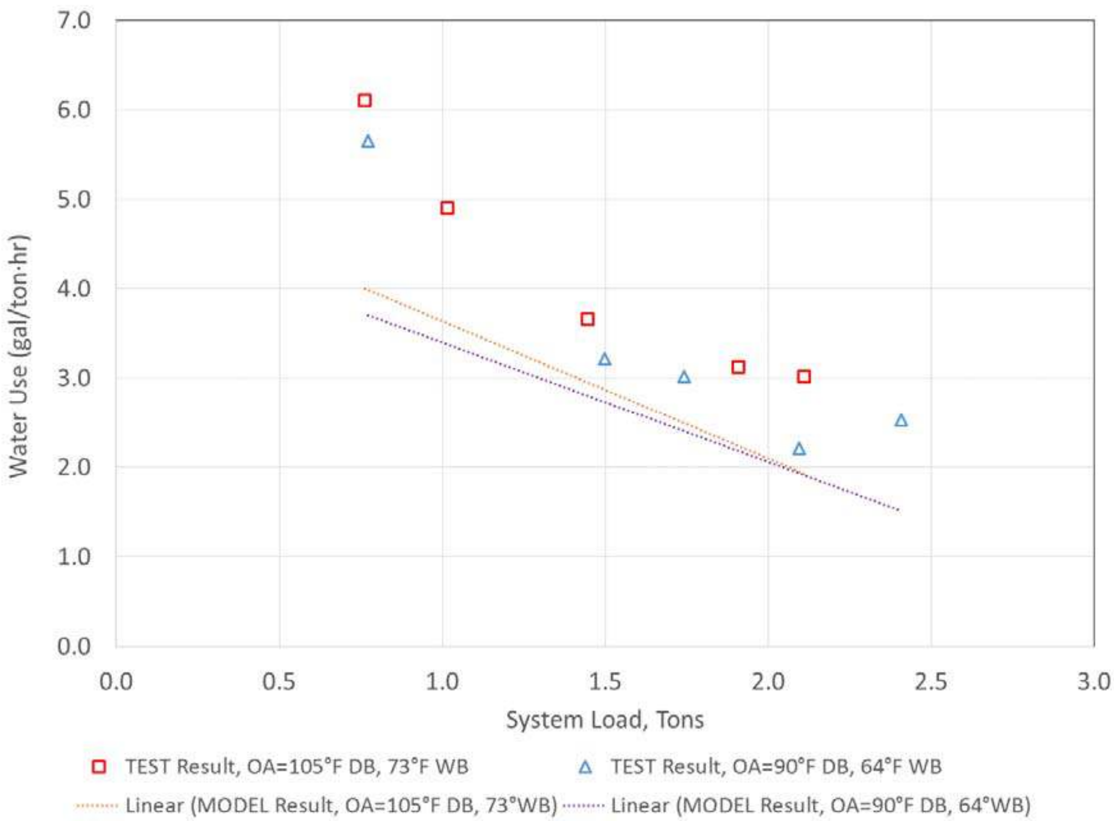

FIGURE 10. SYSTEM LOAD VERSUS WATER USE

Figure 10 illustrates the effect of system load on water-use, where water-use is presented in gallons per ton hour of cooling delivered. It is clear that the efficiency of the water-use increases significantly as the system load increases. The system load increases as return water temperature increases. The measured water-use was between 2-6 gal/ton·hr. The model prediction for the water-use was consistently below the measured value. The average difference between the modeled values at the actual values was 28%.

DISCUSSION

The performance of the SWEC chiller illustrates a large energy savings potential in hot dry climates. The results also reveal that, in hot dry weather conditions, the SWEC produced chilled water at temperatures between 58 to 70 degrees, which is desirable for serving a radiant cooling system. The COP of the SWEC was 8 to 30 during normal operation under a range of weather conditions. The test results revealed that the water consumption of the SWEC under normal operating conditions was 2.2 to 6.1 gallons per ton-hour of cooling delivered.

Table 5 compares the results of this chiller with those of another sub wet-bulb evaporative chiller, that was also tested . For the comparison, the SWEC evaluated in this report is labelled SWEC_Water, and the other chiller is labeled SWEC_Water/Air. Both technologies are more energy-efficient than a conventional air-cooled chiller, but do consume water in order to provide cooling.

In order to understand the comparison between units it is important to note the differences in operation. The major differences are the capacity, the maximum design air flow, the maximum design water flow, and the fact that the SWEC_Water only provides chilled water, whereas the SWEC_Water/Air is designed to provide chilled water and ventilation air. Given these differences in operation, Table 5 highlights the performance of the two chillers under similar external parameters, namely Inlet air conditions, and return water temperature.

TABLE 5. BASELINE COMPARISONS

	COMPARISON 1		COMPARISON 2	
	SWEC_WATER/AIR	SWEC_WATER	SWEC_WATER/AIR	SWEC_WATER
Inlet DB (°F)	90.0	90.0	105.2	104.7
Inlet DP (°F)	47.2	47.2	55.2	56.2
Air Flow CFM	1694	1797	1744	1793
Ventilation Air	553	0	595	0
Water Flow GPM	9.3	4.1	9.2	4.0
Return Water Temp (°F)	71.0	71.1	70.0	71.0
Supply Water Temp (°F)	64.1	60.8	66.0	66.4
Ventilation Supply Air Temp (°F)	69.6	-	73.9	-
Capacity Tons	3.7	1.7	3.2	0.8
Evaporation Gal/(Ton*Hr)	1.7	3.7	2.5	7.4
COP	7.9	23.1	6.8	8.5
AHRI 340/360 COP	9.0	23.1	7.8	8.5

Table 5 outlines the difference in operation between the SWEC Water and the SWEC Water/Air. The first comparison is at an ambient condition of 90°F dry bulb and 64°F wet-bulb, with both units operating at design air and water flow conditions. The comparison shows that the SWEC Water/Air is able to provide a larger capacity, however the SWEC Water provides chilled water that is substantially cooler than the SWEC Water/Air. The COP of the SWEC Water/Air is much lower than the baseline unit, however the SWEC Water/Air includes the ventilation fan power to supply cool air to the building. In order to adjust for this, the power consumption of a typical air handler was considered according to AHRI standard 340/360, and subtracted from the total power-use of the SWEC Water/Air. The adjusted COP is still lower than the SWEC Water but is more representative of the actual difference that will exist in an installation. Finally, the water-use per ton-hour of cooling is significantly lower for the SWEC Water/Air than for the SWEC Water.

The second comparison is at an ambient condition of 105°F dry bulb and 73°F wet-bulb. The flow rates for the two units are near their ideal values, and the return water temperatures were closely matched. In this comparison, the SWEC Water produced a slightly larger drop in water temperature. The COP of the SWEC Water unit is slightly better. The SWEC Water/Air significantly outperforms the SWEC Water in terms of evaporation losses per ton-hour.

The trends reflected in the baseline comparisons show that both sub wet-bulb chillers have benefits and drawbacks. The SWEC Water/Air that was tested for this project is capable of a higher capacity, can provide chilled ventilation air, and accomplishes more cooling while consuming less water per unit of cooling. The SWEC Water is favorable for its higher efficiency, and slightly lower supply water temperatures. Additionally, the SWEC Water unit is smaller, lighter, and will perhaps be better adapted for smaller residential applications than the SWEC Water/Air.

As with any evaporative cooling unit, there are additional complications associated with running the SWEC, compared to a standard compressor-based system. The main concern is regular maintenance of the system in order to prevent corrosion or scale deposits. Current methods used to accomplish this include continuous water bleeds to reduce mineral concentration and replacing the evaporative media every one to three years, depending on the hardness of the water supply.

CONCLUSIONS

The SWEC chiller tested for this report was able to consistently provide cooling loads efficiently, while operating under a variety of environmental, operational, and load-based conditions. The unit consistently provided anywhere from 0.7 to 2.6 tons of cooling, with 1-2 tons being consistently typical and average. The variation in COP was seen to be from 8.5 to 33, with 15 to 25 being an average and typical value. The unit was able to consistently provide chilled water as low as 58°F, which is low enough to be used in a radiant cooling system.

Current technologies that the SWEC can functionally replace are refrigerant-based chillers, cooling towers, and residential packaged units, especially those that currently operate in California's hot, dry climate zones. The advantage the SWEC has over these units is the ability to provide most, if not all, the functionality of a refrigerant-based system, while providing energy savings. Further, the SWEC technology can replace a coupled cooling tower/refrigerant unit combination, leading to lower capital costs, and decreased complexity for energy-conscious consumers. The SWEC requires no refrigerant or compressors, which reduces complexities associated with compressors and environmental hazards associated with refrigerants.

A potential barrier to the SWEC being adopted is the requirement of a thermal distribution system for the system to work. The cost of installation for either a radiant system or for a fan coil unit will have to be considered in any cost/benefit analysis, and will add significantly to the cost of the unit. The SWEC will require yearly maintenance to reduce the impacts of scale and corrosion, and will require mineral management on a smaller timescale, which can be managed by supplying a bleed to the unit to maintain mineral concentration at safe levels.

The SWEC meets an important design criteria that is needed in the market. Barriers to the current prototype are expected due to the initial costs necessary to install the units, and lack of familiarity with the technology. The initial costs of the unit are expected to be low enough to make commercial retrofits viable. The SWEC referenced in this report will require simple maintenance, however this maintenance is different than what would typically be done with a refrigerant-based system.

RECOMMENDATIONS

The results of the lab evaluation of the SWEC technology show that it has great potential to reduce energy-use in hot dry climates. There are also some significant barriers in place that need to be overcome. In general, using the technology in a retrofit application is difficult because most current buildings use ducted forced-air systems for cooling distribution. Adding radiant panels will add significant costs to a retrofit installation. For retrofit applications, a fan coil unit can potentially replace a traditional evaporator coil in a ducted forced-air system. Because chilled water temperatures will be higher than an evaporator in a compressor-based system, the cooling capacity of the existing system will be reduced. Based on the design on the SWEC, it is expected that large scale manufacturing of the SWEC can be cost-competitive in comparison to compressor-based air conditioners. In new construction, a radiant cooling installation combined with the SWEC can be cost-competitive with a compressor-based air conditioner with a ducted forced-air system. In China, the SWEC water/air technology has been installed with radiant cooling systems to cool over one million square feet of commercial buildings.

It is recommended that further research be done in order to determine the cost-effectiveness of a SWEC with a thermal storage system and thermal distribution system using fan coil units in a residential building. The analysis should determine if a fan coil thermal distribution system can meet the load in a residential building in California climate zones, along with the expected energy and demand savings. The analysis should also determine if the addition of a thermal storage system makes economic sense as a peak demand reduction strategy.

REFERENCES

[1] ASHRAE, "Standard 41.2-1987 - Standard Methods for Laboratory Air Flow Measurement," ASHRAE, Atlanta, 1987.

[2] T. P. Yu Hou, "Measuring Performance of a Plate Air-to-Air Heat Exchanger," Western Cooling Efficiency Center, UC Davis, Davis, 2015.

[3] T. P. Yu Hou, "Measuring Performance of Evaporative Media under Load," Western Cooling Efficiency Center, UC Davis, Davis, 2015.

APPENDICES

Test	Ambient Condition (DB/WB) (°F)	Air Flow (CFM)	Water Flow (GPM)	Return Water Temp °F	Supply Water Temp °F	Power (Watts)	Water Use Gal/ (Ton*hr)	Capacity (Tons)	Capacity Uncertainty (Tons)	COP	COP Uncertainty
1	90/64	1797	4.07	71	60.8	266	3.02	1.74	9.39E-02	23.1	1.26
2	90/64	1377	4.07	71	61.9	208	2.87	1.54	9.31E-02	25.9	1.59
3	90/64	896	4.07	71	64.1	168	2.77	1.19	9.19E-02	24.8	1.94
4	90/64	2234	4.00	71	60.1	388	3.41	1.80	9.27E-02	16.3	0.85
5	90/64	2239	4.95	71	61.4	384	3.49	1.95	1.13E-01	17.9	1.05
6	90/64	2234	6.12	71	61.9	394	3.20	2.29	1.40E-01	20.4	1.26
7	90/64	2262	2.83	71	58.8	382	3.80	1.43	6.63E-02	13.2	0.62
8	90/64	2260	2.80	74	58.7	383	3.31	1.79	6.77E-02	16.4	0.64
9	90/64	2248	4.12	74	61.0	349	2.97	2.25	9.74E-02	22.6	1.00
10	90/64	2241	5.02	74	61.8	322	2.90	2.54	1.18E-01	27.8	1.30
11	90/64	2256	5.87	74	62.2	383	2.75	2.91	1.37E-01	26.7	1.28
12	105/73	1793	4.04	71	66.4	315	6.11	0.76	9.03E-02	8.5	1.01
13	90/64	1797	4.13	77	62.4	256	2.53	2.41	9.85E-02	33.1	1.38
14	105/73	1801	4.02	74	67.7	254	4.90	1.02	9.03E-02	14.1	1.26
15	105/73	1802	4.10	77	68.2	252	3.66	1.45	9.33E-02	20.2	1.31
16	105/73	1800	4.07	80	68.5	252	3.12	1.91	9.46E-02	26.6	1.34
17	105/73	1800	4.03	83	69.2	251	3.01	2.11	9.47E-02	29.6	1.35
18	90/64	1793	4.00	65	60.0	268	5.65	0.77	8.94E-02	10.1	1.18
19	90/64	1793	4.17	68	60.3	269	3.22	1.50	9.52E-02	19.6	1.25
20	90/64	1812	4.15	74	61.8	267	2.21	2.09	9.71E-02	27.6	1.30

TABLE 6: SWEC PERFORMANCE RESULTS