Sub Wet-Bulb Evaporative Chiller

ET15SCE7040

Prepared by:

Emerging Products
Customer Service
Southern California Edison

June 2015

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Acknowledgments

Southern California Edison's Emerging Products (EP) group is responsible for this project. It was developed as part of Southern California Edison's Emerging Technologies Program under internal project number ET15SCE7040, by University of California Davis's (UC Davis) Western Cooling Efficiency Center (WCEC). Jay Madden conducted this technology evaluation with overall guidance and management from Jerine Ahmed. Contact Jay.Madden@SCE.com for more information on this project.

EXECUTIVE SUMMARY

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 and constrain the electric infrastructure. 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 applicability in chilled water cooling systems. The sub wet bulb evaporative chiller (SWEC) technology has a significant advantage over other evaporative technologies because of its ability to cool below the ambient wet bulb. Chilled water below the ambient wet-bulb could be utilized in a radiant floor or ceiling cooling system, or a fan coil system. In light commercial buildings, this type of cooling system could replace typical roof top units with air duct systems.

Several unique designs exist that are considered sub wet bulb evaporative chillers, and this report focuses on one such design. The SWEC cools an outdoor air stream using an indirect evaporative cooling process. Part of this cooled, dry, outdoor air is delivered to the building as ventilation air; the rest is exhausted as part of the evaporative cooling process, which also chills a water supply used to cool the building.

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
- Ventilation air flow

The objective of this assessment is to evaluate the performance of the SWEC in the laboratory under a range of environmental conditions and operating modes. The supply water temperatures, system efficiency, and the water consumption required for cooling would be evaluated. The analysis provides insight to the potential for the SWEC to replace traditional compressor based systems, including the potential for peak demand and total energy savings.

The SWEC was tested 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 could be determined. Outdoor air conditions and return water temperature were held at a steady state.

The performance of the SWEC chiller illustrates a large energy savings potential in hot, dry climates (daytime temperatures above 90°F with relative humidity <30%). The results also show that, under a wide range of weather conditions, the SWEC produced chilled water at temperatures between 60 to 66°F, which is desirable for serving a radiant cooling system. Ventilation air temperatures supplied to the building were between 63°F to 75°F. The coefficient of performance (COP) of the SWEC was 5 to 8 during normal operation under a range of weather conditions. For comparison, the US Department of Energy requires commercial packaged air conditioners with a capacity between 5-20 tons of cooling to have a

minimum rated energy efficiency ratio of at least 11 at 95°F outdoor air temperature, which equates to a COP of 3.22 [1]. Therefore, the tested technology could roughly double the cooling efficiency delivered by standard packaged air conditioning equipment. The test results showed that the water consumption of the SWEC under normal operating conditions was 1.0 to 2.5 gallons per ton-hour of cooling delivered.

The results of the lab evaluation of the SWEC technology show that it has great potential to reduce energy use by 50% 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. One or multiple fan coil units can potentially replace a traditional evaporator coil, and also 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 using fan coil units in a residential or multi-family building. The analysis should determine if a fan coil thermal distribution system can meet the load in a residential building in California climate zones, or if a small backup chiller would be needed to supplement the SWEC cooling in some cases. 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
COP	Coefficient of Performance
CS	Cold Side
CFM	Cubic Feet per Minute
DB	Dry Bulb
DP	Dew Point
EA	Exhaust Air
GPM	Gallon per Minute
HS	Hot Side
Hz	Hertz
NI	National Instruments
OA	Outside Air
RA	Recirculated or Return Air
RTD	Resistance Temperature Device
SA	Supply Air
SCE	Southern California Edison
SWEC	Sub Wet bulb Evaporative Chiller
WB	Wet bulb

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INTRODUCTION

This purpose of this project is to complete a laboratory evaluation of a sub wet-bulb evaporative chiller (SWEC). The SWEC is an evaporative cooling technology that chills water to cool a building in combination with a radiant or fan coil distribution system. In addition to chilled water, the SWEC also provides cool ventilation air using indirect evaporative cooling technology. (This is done so no humidity is added to the ventilation air). The SWEC technology can be manufactured in different sizes and configurations. The unit tested in the laboratory was designed to provide approximately 3 tons of cooling, including the 600 cubic feet per minute (cfm) of ventilation air.

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, 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 could reduce some of the initial expense associated with evaporative cooling and eliminate the hazards associated with refrigerants in terms of greenhouse gas emissions, ozone depletion, and global warming potential. 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 objective of this assessment is to evaluate the performance of the SWEC in the laboratory under a range of environmental conditions and operating modes. The supply water temperatures, system efficiency, and the water consumption required for cooling would be evaluated. The analysis provides insight to the potential for the SWEC to replace traditional compressor based systems, including the potential for peak demand and total energy savings.

TECHNOLOGY DESCRIPTION

The sub wet-bulb SWEC chills water through an evaporative cooling process for use in building cooling systems Figure 1. Several unique designs exist that are considered chillers, and this report focuses on one such design. The SWEC cools an outdoor air stream using an indirect evaporative cooling process. Part of this cooled outdoor air is delivered to the building as ventilation air; the rest is exhausted as part of the evaporative cooling process, which also chills a water supply used to cool the building. The ratio of exhaust air to ventilation air is controlled with a damper. Three water loops consist of an air-to-water heat exchanger and an evaporative media. One water loop consists of a water-to-water heat exchanger to chill return water from the building and an evaporative media.

The arrangement of the SWEC is such that the water loops are used to sensibly precool the incoming air before it is used to evaporatively cool the water. Because the sensible cooling reduces the wetbulb temperature of the air, the evaporative cooling process can chill air and water below the ambient wet bulb temperature. The theoretical limit of the supply water provided by the SWEC is the ambient dew point.

The water loops consist of a pump, an evaporative media, and a coil. Water is pumped to the evaporative media, where the evaporative process chills the water. The water is then passed through the coil, where it precools the inlet air, and then returns to the pump. Upon startup, the process successively cools the water loops, and then the inlet air, until steady state is reached. The water loops act independently, however there is a pipe that connects all of the sumps to a water makeup valve. This pipe allows for sump balancing and some mixing takes place as a result. The chiller has a built-in control system and an interface where the user can change the fan speed and turn specific pumps on and off.

The SWEC design 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
- Ventilation air flow

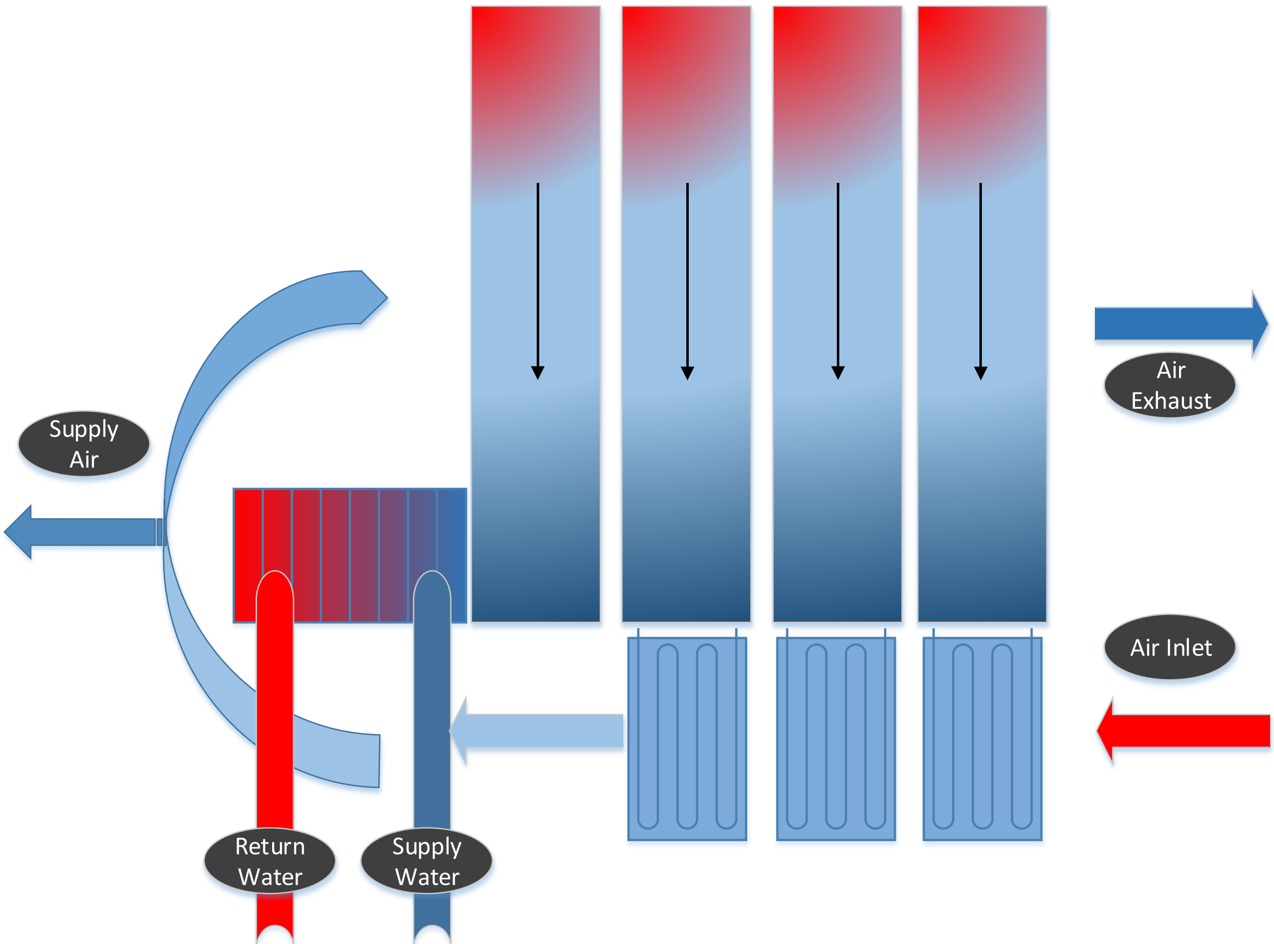

FIGURE 1. SIMPLE SCHEMATIC OF SWEC OPERATION

TECHNICAL APPROACH/TEST METHODOLOGY

OVERVIEW

The SWEC was manufactured in China and shipped to the laboratory for testing. The unit was instrumented by WCEC to collect data in order 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 reported.

TEST PLAN

The SWEC was installed in the WCEC environmental chamber to simulate desired outdoor air conditions. A load rig designed to supply a controlled water temperature was plumbed to the return water connection. Measurements for the inlet, ventilation supply, and exhaust air conditions were recorded with dew point and temperature sensors installed in the chamber. Measurements of the supply air flow rate were measured using Tracer Gas Measurements.

The test plan was designed to measure the performance of the chiller under varying operating loads, air, and water conditions.

In order to meet the testing goals the SWEC was tested with varying parameters (Table 1) including:

- Total air flow (1200-3300 CFM)
- Ventilation air fraction (0-67%). Most tests were conducted at 33% ventilation air fraction which was estimated to be the design ventilation air flow rate for a building served by the SWEC.
- Inlet air conditions (varying dry bulb and humidity)
- Return water flow rate (9-13 Gallons per Minute (GPM)). Most tests were conducted at 9 GPM which was the design rate for the chiller. One test was completed at 13 GPM in order to measure the maximum capacity that could be delivered by the chiller with the available pump.
- Return water temperature (64°F to 74°F)

TABLE 1.SWEC TEST POINTS

TEST	AMBIENT TEMPERATURES (°F DB/0F WB)	RETURN WATER TEMPERATURE (°F)	AIR FLOW (CFM)	WATER FLOW (GPM)	NOMINAL VENTILATION AIR FRACTION
1	90/64	68	1652	9.3	33%
2	90/64	68	3265	12.9	33%
3	90/64	68	2732	9.3	33%
4	90/64	68	1202	9.3	33%
5	90/64	68	1716	9.2	0%
6	90/64	68	1770	9.3	50%
7	90/64	68	1684	9.3	67%
8	90/64	64	1700	9.3	33%
9	90/64	66	1701	9.3	33%
10	90/64	68	1714	9.5	33%
11	90/64	71	1694	9.3	33%
12	105/68.8	68	1660	9.3	33%
13	95/65.7	68	1660	9.3	33%
14	85/62.3	68	1646	9.4	33%
15	75/58.7	68	1613	9.3	33%
16	65/54.8	68	1598	9.4	33%
17	105/73	70	1744	9.2	33%
18	95/70.1	70	1654	9.3	33%
19	85/67.1	70	1638	9.3	33%
20	90/64	68	3309	9.2	0%
21	105/73	74	3315	9.2	0%

TABLE 2. VARIATION OF PARAMETERS

VARIED PARAMETER	TESTS
Varied Air Flow; (90/64) Tw = 68, Vent. Air= 33%	1,3,4
Single test, max capacity (max air and water flow)	2
Varied Vent. Air Fraction; (90/64) Tw = 68	5,6,7
Varied Return Water Temp, (90/64), Vent. Air= 33%	1,8,9,10,11
Varied DB, DP = 47, Tw = 68, Vent Air = 33%	12,13,14,15,16
Varied DB, DP = 47, Tw = 70, Vent Air = 33%	17,18,19
Capacity, 0% Vent Air	20,21

INSTRUMENTATION PLAN

In order to simulate a building load, a load rig capable of producing over 5 tons of heating was connected to the chiller (Figure 2). The temperature of the chiller inlet water was controlled by means of a proportional, integral, differential controller. The load rig is capable of changing load by independently changing either temperature or water flow rate. The load cart includes a Hall effect pulse meter to measure water flow rate and two resistance temperature detectors (RTDs) to measure temperature that were connected directly to the chiller inlet and outlet. The four internal water loops of the chiller were instrumented with RTDs to provide additional information on chiller performance.

The chiller was installed in the environmental chamber, with the exhaust and ventilation supply air ducted (Figure 3). The total airflow was measured by a calibrated air flow measurement system using calibrated flow nozzles. The temperature and dew point of the inlet, ventilation supply, and exhaust air streams were measured with RTDs and chilled mirror dew point sensors. The exhaust air was ducted outside of the chamber, and the ventilation supply air was ducted to a second chamber, where one-time air flow measurement was recorded using the Tracer Gas Air Flow Measurement System. The ventilation supply air was controlled by means of a manual damper, that was adjusted to allow a certain percentage of the total air to be diverted for ventilation.

In order to ensure the airflows in the chiller were unaltered by attaching an exhaust duct, a baseline air flow measurement was obtained for each operating condition to be tested. Two curves were developed relating the un-ducted pressure drop across the air inlet of the chiller to the air flow rate. For each test the air flow rate was determined by matching the pressure drop across the air intake according to the operational mode. The operational parameters which were varied and could affect air flow were chiller fan speed and ventilation damper position.

The power consumption of the chiller, fan, and cumulative pump power was monitored with three phase true power meters. One of the four pumps arrived damaged and had to be replaced. The replacement pump power reading was measured one time and added to the total power consumption of the chiller. The water consumption was monitored by comparing the absolute humidity of the inlet and exit air streams. While a water flow meter was installed on the make-up water line to the chiller, a float valve controlled the make-up water flow and introduced significant noise into the makeup water flow signal. Therefore, water consumption calculated from a change in humidity of the air stream proved to be more reliable.

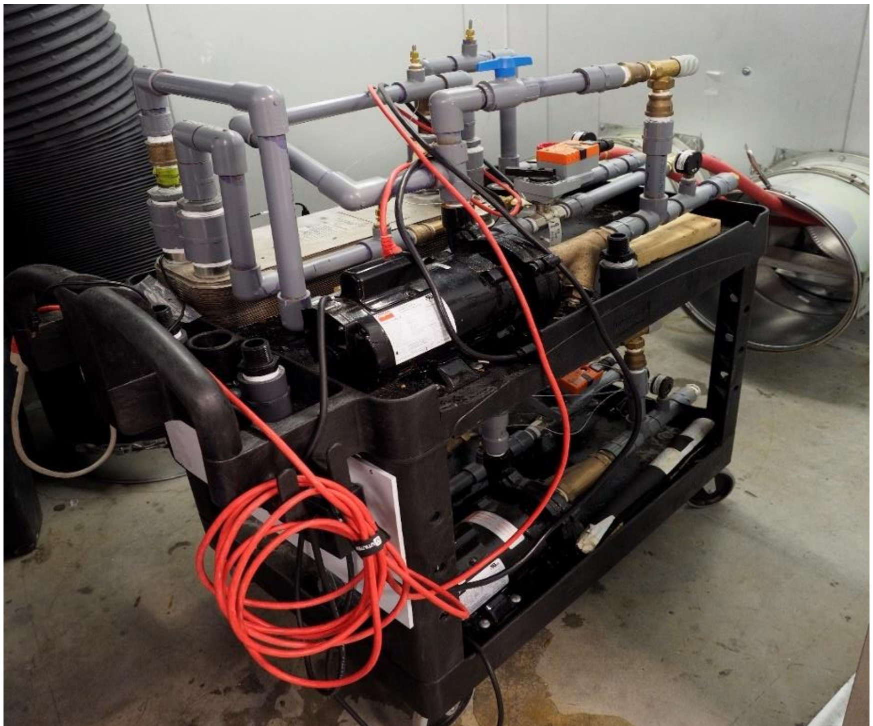

FIGURE 2. LOAD RIG USED TO PROVIDE BUILDING LOAD TO SWEC

FIGURE 3. SWEC BEING MOVED INTO THE CHAMBER

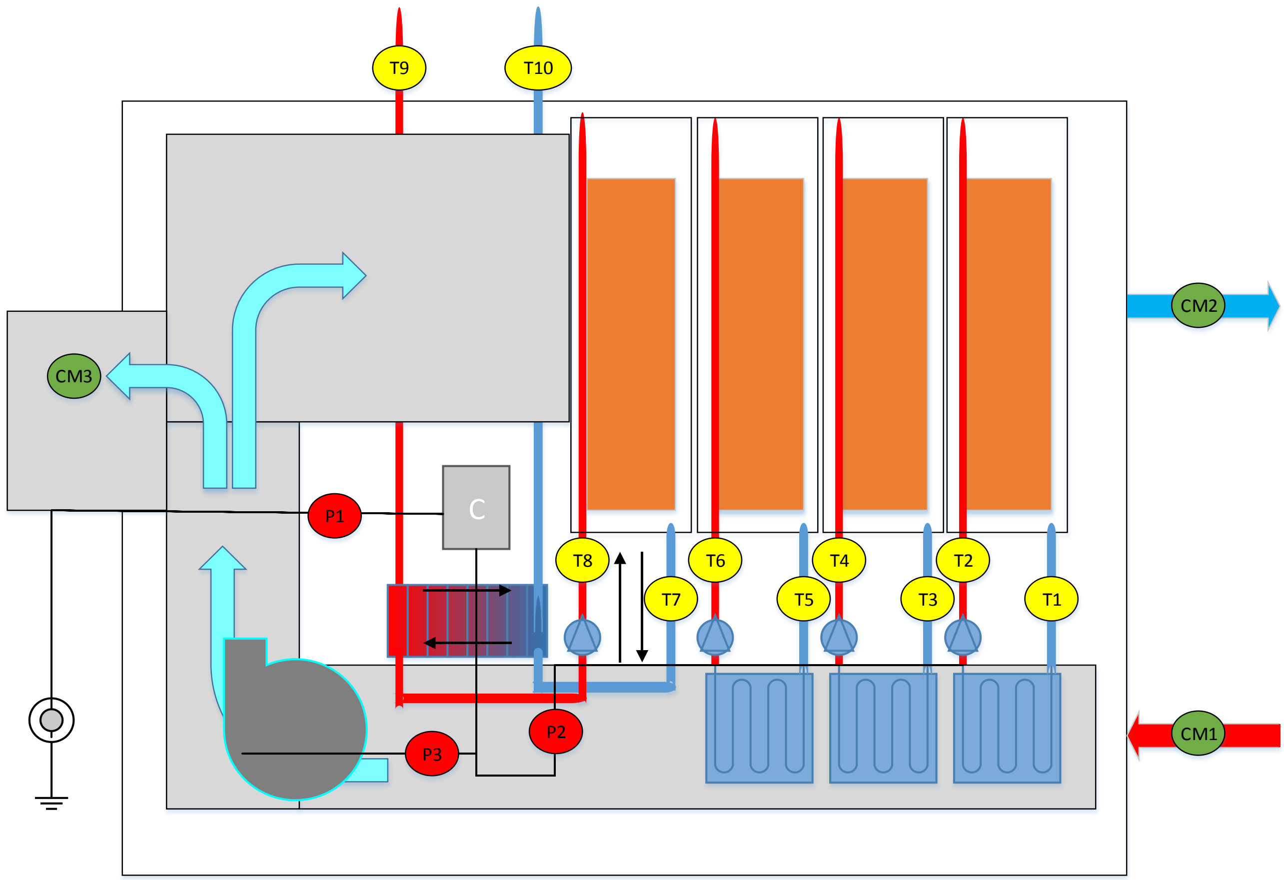

FIGURE 4. DIAGRAM OF SWEC INSTRUMENTATION LOCATIONS (INSTRUMENTATION LABELS CORRESPOND TO TABLE 4.)

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	(65 to 105)/ (45 to 60) (0F)	(+/- 0.26)/ (+/- 0.36) (0F)	03/2012	RS-232
CM2	Hot Chamber Exit Dry Bulb/Dew Point	GE Optisonde 2-1-1-1-1-0-0-0	(75 to 85)/ (60 to 85) (0F)	(+/- 0.26)/ (+/- 0.36) (0F)	01/2014	RS-232
CM3	Ventilation Air Dry Bulb/Dew Point	GE Optisonde 2-1-1-1-1-0-0-0	(60 to 85)/ (45 to 60) (°F)	(+/- 0.26)/ (+/- 0.36) (0F)	01/2014	RS-232
T1	Sump 1 Inlet Temperature	Omega RTD-NPT-72-E	60 to 85 (0F)	+/- 0.3 to 0.4 (0F)		NI 9217
T2	Sump 1 Outlet Temperature	Omega RTD-NPT-72-E	60 to 80 (0F)	+/- 0.3 to 0.4 (0F)		NI 9217
T3	Sump 2 Inlet Temperature	Omega RTD-NPT-72-E	60 to 85 (0F)	+/- 0.3 to 0.4 (0F)		NI 9217
T4	Sump 2 Outlet Temperature	Omega RTD-NPT-72-E	60 to 80 (0F)	+/- 0.3 to 0.4 (0F)		NI 9217
T5	Sump 3 Inlet Temperature	Omega RTD-NPT-72-E	60 to 75 (0F)	+/- 0.3 to 0.4 (0F)		NI 9217
T6	Sump 3 Outlet Temperature	Omega RTD-NPT-72-E	60 to 70 (0F)	+/- 0.3 to 0.35 (0F)		NI 9217
T7	Sump 4 Inlet Temperature	Omega RTD-NPT-72-E	55 to 70 (0F)	+/- 0.3 to 0.35 (0F)		NI 9217
T8	Sump 4 Outlet Temperature	Omega RTD-NPT-72-E	55 to 65 (0F)	+/- 0.3 to 0.35 (0F)		NI 9217
T9	Chiller Inlet Temperature	Omega RTD-NPT-72-E	60 to 75 (0F)	+/- 0.3 to 0.4 (0F)		NI 9217
T10	Chiller Outlet Temperature	Omega RTD-NPT-72-E	55 to 70 (0F)	+/- 0.3 to 0.35 (0F)		NI 9217
Q1	Load Cart Flow Rate	Omega FTB 4607	9 to 13 (gpm)	+/-1.5% of reading		NI-USB-6009
P1	System Power	Dent Power Scout 3+	1000 to 4000 Watts	+/- 1% of reading		RS-485
P2	Pumps 1,3,4 Power	Dent Power Scout 3+	800 to 1000 Watts	+/- 1% of reading		RS-485

P3	Fan Power	Dent Power Scout 3+	300 to 3000 Watts	+/- 1% of reading		RS-485
P4	Pump 2 Power (One Time)	Fluke 1735	65 Watts	+/- 1.5% of reading		None
DP1	Air Inlet Differential Pressure	DG 500	0-300 Pa	+/-1% of reading		RS-232
DP2	Flow Nozzle Differential Pressure	Energy Conservatory APT 8	100-500 (Pa)	+/- 1% of reading		RS-232

Water Temperature Measurements

Within the chiller, the temperature was measured at the inlet and outlet of each sump at the locations labeled “Tx” in Figure 4. The RTDs were threaded into adapters which were placed in the inlet pipes; the taps were made in the return pipes in which the RTDs were threaded directly. The temperature conditions were used to check the system performance against that predicted by the chiller designer to ensure the system was operating as intended. The temperature of the water to the chiller to the inlet and outlet were used to monitor the real-time capacity of the chiller.

Evaporative Water Consumption Measurements

The water consumed by the chiller was measured using the dew point readings from the chilled mirrors and the measured flow rates of the inlet and exhaust air streams. The chilled mirrors work by cooling a mirrored surface until the instrument detects the presence of dew which begins to appear at the dew point. Makeup water was supplied to the chiller by means of a connection to a tap water source which was controlled via a float valve. The sumps were internally balanced so that a single float valve acted to fill all four sumps.

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 intake air nozzle box and the static pressure upstream of the nozzles to calculate airflow for the chamber according to ANSI/ASHRAE 41.2-1987 [2]. The differential pressure across the chiller air inlet was measured using an Energy Conservatory DG-500.

Tracer Gas Measurements

The ventilation supply air flow rate was determined using a tracer gas airflow measurement, conducted according to ASTM E2029 Standard Test Method for Volumetric and Mass Flow Rate Measurement in a Duct Using Tracer Gas Dilution (ASTM 2011). This method mixes a measured mass flow rate of CO2 into the supply air stream then measures the corresponding rise in CO_2 concentration downstream. The volume flow of air into which the tracer is mixed can be calculated by the following equation:

EQUATION 1. TRACER GAS EQUATION

$$\dot{V}_{Airflow} = \frac{\dot{V}_{CO2}}{C_{CO2\ downstream} - C_{CO2\ upstream}}$$

This method has many advantages compared to conventional air balance techniques, the most significant of which is accuracy. The tracer gas airflow tools can measure with a calculated uncertainty of less than ±2%.

The measurement was taken once the test reached steady state, but before any data was collected. The target airflow rate was obtained by adjusting the damper position according to the previous tracer gas measurement. Once the target air flow was reached the system was allowed to reach steady state before beginning the test.

CHAMBER CONDITIONS MEASUREMENTS

During all tests the inlet air, ventilation supply air, and exhaust air were monitored with two 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 chiller power, pump power for three pumps, and fan power were recorded using PowerScout™ 3+ instruments with serial communication using a Modbus® protocol. The power difference between the component power and the total power is consumed by the electronics package on the unit. Finally, due to a malfunction of one of the original pumps, a replacement pump was installed and a one time reading was taken for the power use. This reading was taken using a Fluke 1735 Power Logger. Since the pump is single speed and resistance in the water loop is constant, this measured power reading was assumed constant for the duration of testing and added to the total chiller power consumption.

DATA ACQUISITION SYSTEM

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

TEST TOLERANCES

For each test all of the test condition parameters including outdoor air dry bulb and wet bulb temperatures, inlet water temperatures, water flow rate, and air flow rate were controlled to 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	±0.5°F
Outdoor Air Wet bulb Temp	±2°F	±0.5°F
Return 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 chiller was calculated for each test, where the capacity is the sum of cooling for both the water and the ventilation air, as described by Equation 2.

EQUATION 2. CAPACITY

$$\dot{q} = \dot{m}_{water} \times c_{p,water} \times \left(T_{water,in} - T_{water,out}\right) + \dot{m}_{air,vent} \times c_{p,air} \times \left(T_{air,in} - T_{air,vent}\right)$$

where $\dot{q}$ is capacity in British Thermal Units (BTU/hr). The subscripts “water” and “air” denote water and air qualities respectively. The subscript “in” denote the inlet outside air and the subscript “vent” denotes the supplied ventilation air. T refers to temperatures for either inlet or outlet in °F, c_p is the specific heat, and $\dot{m}$ refers to mass flow rate, which is derived from volumetric flow rate according to Equation 3 and Equation 4.

EQUATION 3. MASS FLOW RATE OF WATER

$$\dot{m}_w = Q_{water} * 60 * \rho_{water}$$

Equation 3 shows the derivation for mass flow rate of water used in Equation 2, where Q is flow rate in gallons per minute, and ρ is density in pounds per gallon. Since Equation 2 is expressed per hour 60 is used to convert from minutes to hours.

EQUATION 4. MASS FLOW RATE OF AIR

$$\dot{m}_{air,vent} = Q_{air,vent} * 60 * \rho_{air,vent}$$

Equation 4 shows the derivation of mass flow rate for ventilation air used in Equation 2. Q represents air flow in CFM and ρ is air density in pounds per cubic foot at the measured temperature and pressure. As with Equation 3, the mass flow rate is expressed in pounds per hour, so a multiplier of 60 is used to convert from minutes to hours.

Coefficient of Performance

The coefficient of performance of the unit was calculated for each test using Equation 5.

Equation 5. Coefficient Of Performance

$$COP = \frac{\dot{q}}{P_{total}}$$

where $\dot{q}$ is the capacity of the SWEC in BTU/hr and P_{total} is the total power consumption of the chiller power, 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 method1 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, Ro, 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 Ri+ for each independent variable. Next, the perturbation is performed by subtracting the uncertainty from each independent variable to find Ri-. The average perturbation value, δRi, for each independent variable is found by averaging the absolute difference between Ro and the perturbation values, Ri+ and Ri-. Finally, using Equation 3, the total uncertainty is calculated for each value.

Equation 6. Sequential Perturbation

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

RESULTS

Figure 5, Figure 6, and Figure 7 show the effects of varying parameters on the chiller efficiency and supply temperatures from the chiller. The detailed results of the tests conducted are tabulated in the Appendix (Table 5).

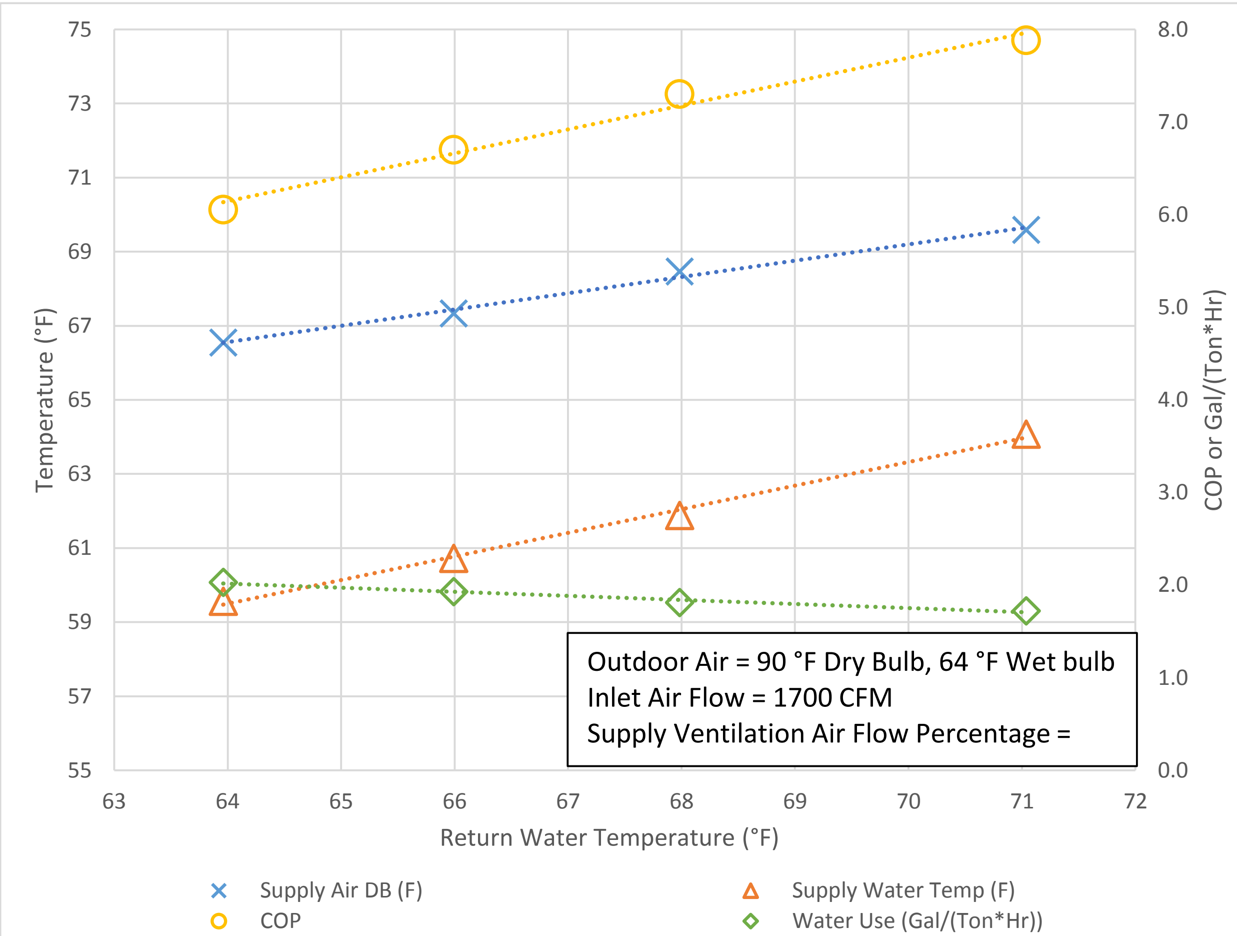

FIGURE 5.. EFFECTS OF VARYING RETURN WATER TEMPERATURE

Figure 5 shows the effects of varying the return water temperature on the performance of the chiller. The tests were conducted at constant air inlet properties, 1700 CFM total air flow, and 33% ventilation air fraction. In all cases the chiller supplied water at or below the wet bulb temperature of the air. The effect of the increased return water temperature is to raise both the ventilation air temperature air and the supply water temperature. The water use increases as the return water temperature is increased. The return temperature increase negatively impacts the supply water temperature the chiller can achieve, however the efficiency the system increases as the return water temperature increases. The temperature differential between the return water temperature and supply water temperatures increases as the return water temperature increases.

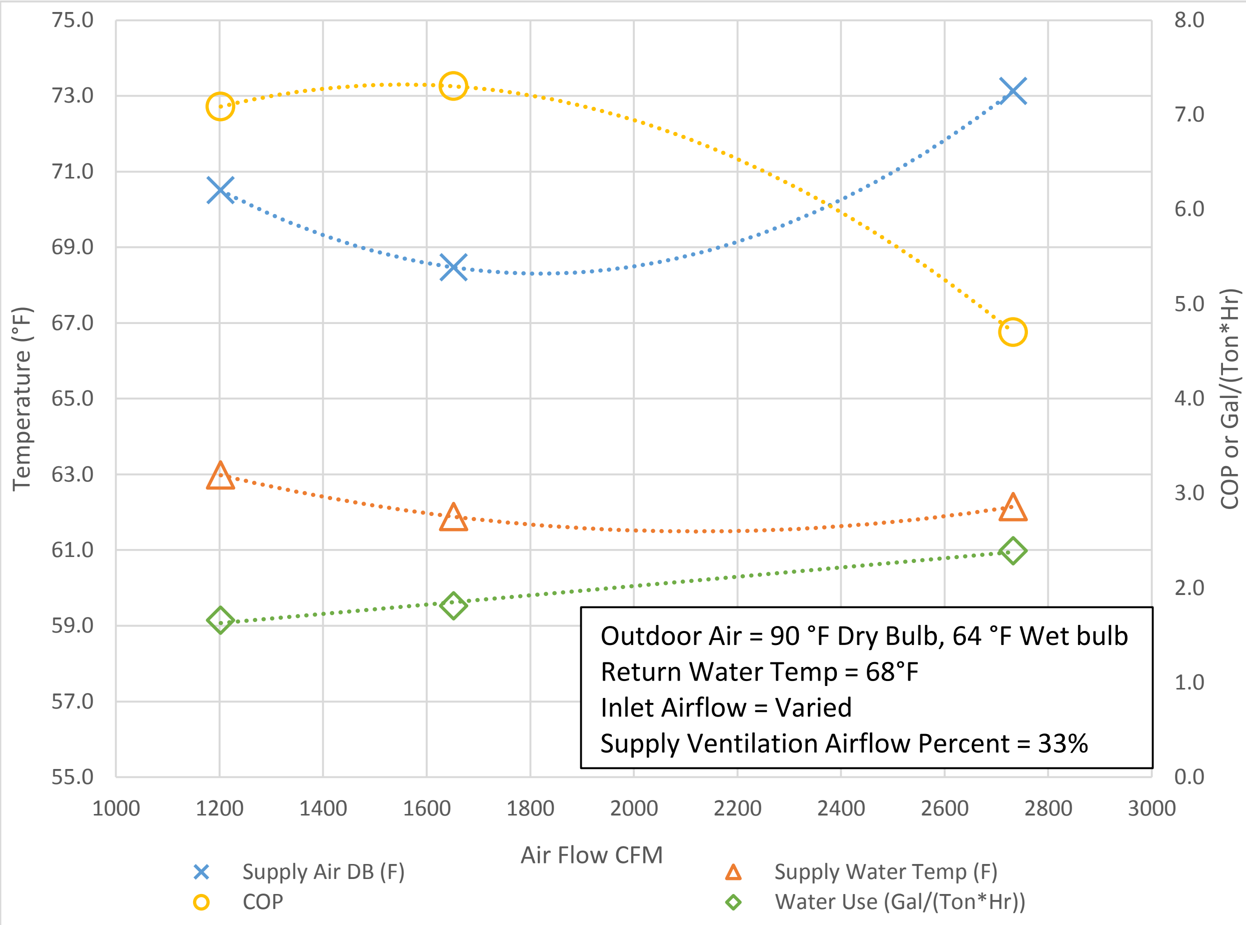

FIGURE 6. EFFECTS OF VARYING INLET AIR FLOW

Figure 6 shows the results of varying inlet air flow rate while holding the ventilation air fraction and inlet air properties constant. The flow was varied from 1200 CFM to 2700 CFM. The chart clearly illustrates that in this configuration the optimal flow rate tested was 1650 CFM. The COP is highest at that point, and the supply temperature for both the air and water are lowest. The total capacity of the system is highest at 2700 CFM, but the total COP is decreased because of the increased fan power. The water use exhibits a clear trend of increasing as the air flow rate increases.

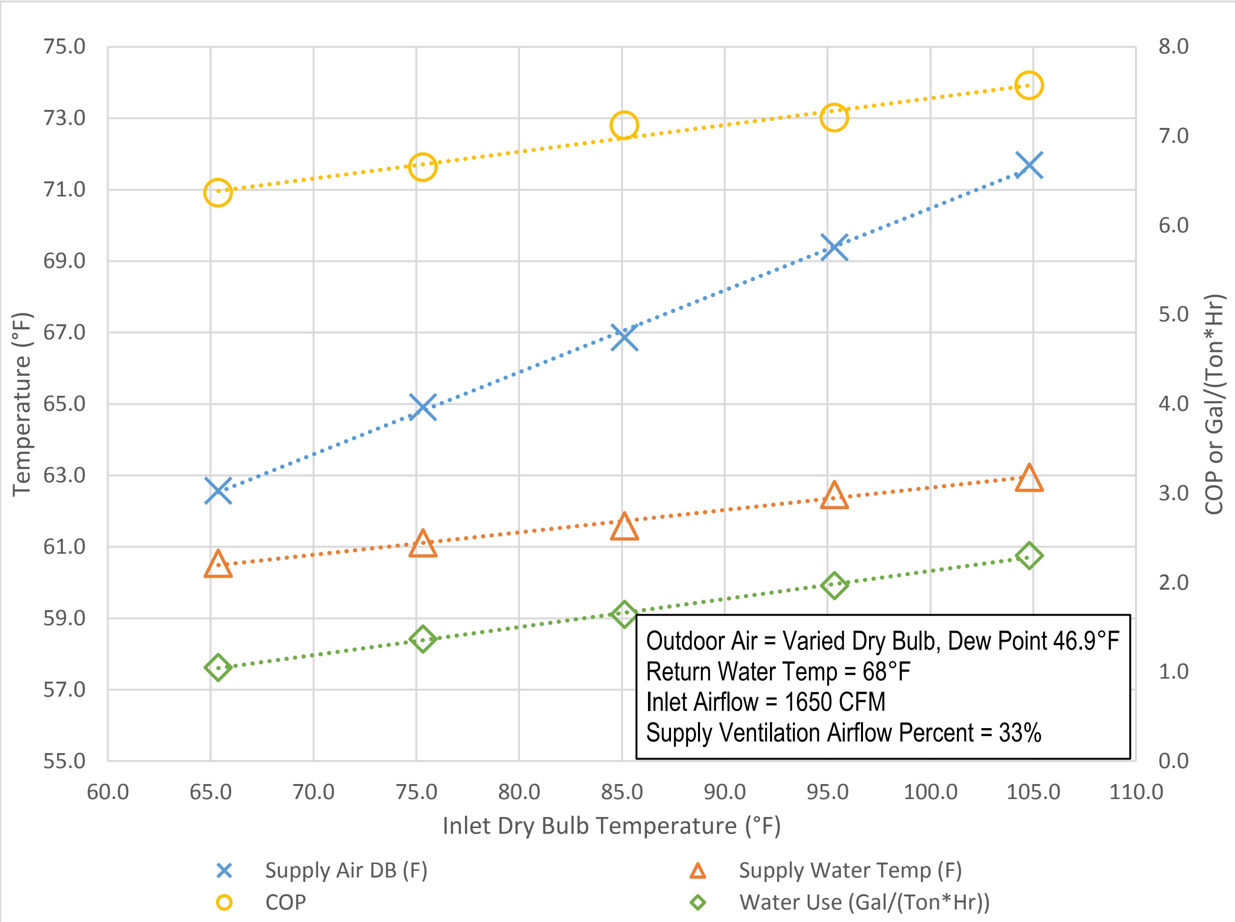

FIGURE 7. EFFECTS OF VARYING INLET DRY BULB TEMPERATURE

Figure 7 shows the effects of varying ambient dry bulb temperature while keeping the dew point (46.9°F) and return water temperature (68°F) constant. The results show that the supply ventilation air temperature is sensitive to the change in ambient dry bulb temperature. The supply water temperature is less sensitive to the change in dry bulb temperature. Increasing the outdoor air temperature from 65°F to 105°F increased the supplied water temperature from 60.5°F to 63°F. However, the supply ventilation air temperature increased from 63°F to 72°F. The chiller COP increases with the ambient dry bulb, however the water use increases substantially with dry bulb temperature (56% increase with 20°F increase in dry bulb temperature).

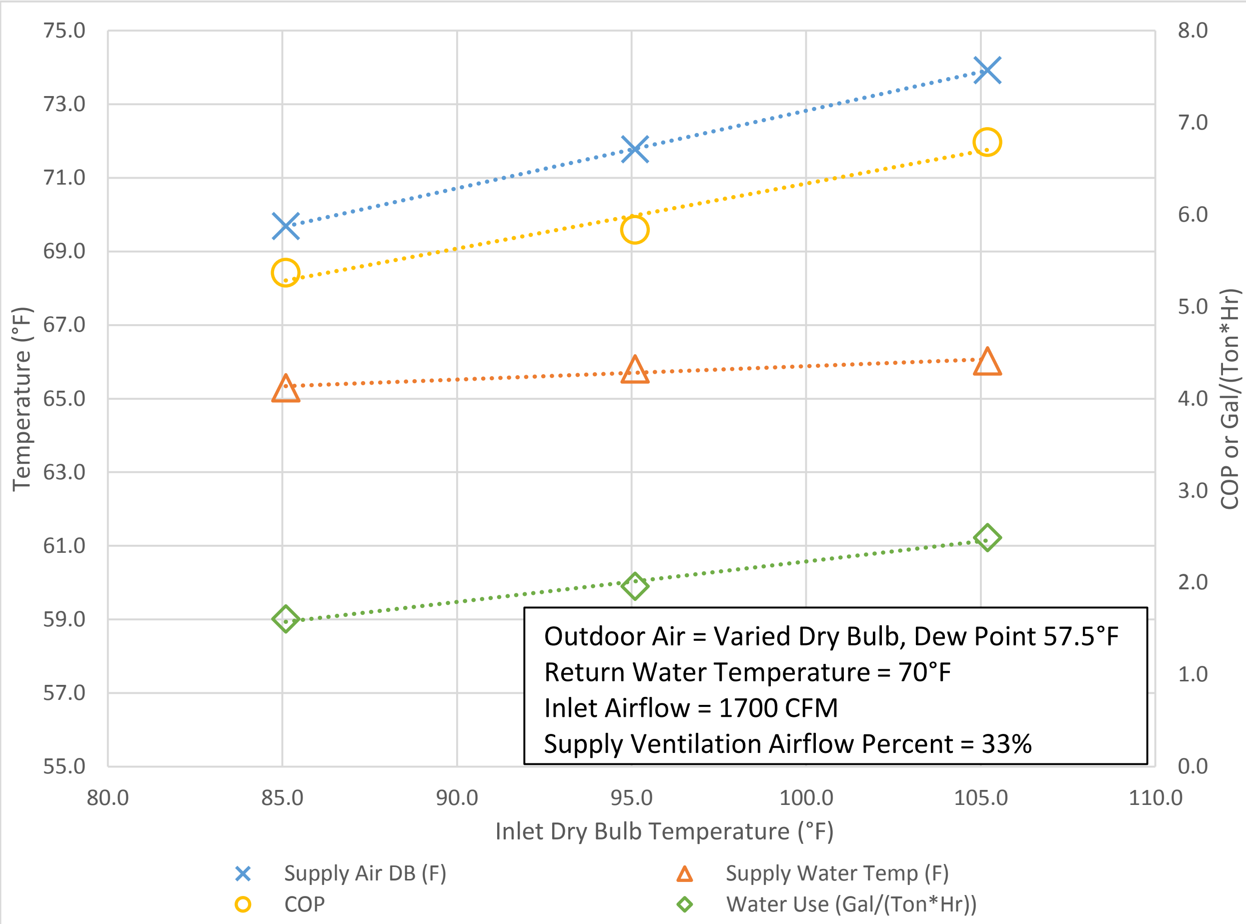

FIGURE 8. EFFECTS OF VARYING INLET DRY BULB TEMPERATURE

Figure 8 shows the effects of varying the inlet dry bulb at ambient air conditions that are more humid than Figure 7. The return water temperature (70°F) was also higher. The higher dew point (57.5°F) reduces the cooling potential of the chiller, and the higher return water temperature is more representative of excepted return water at the ambient air conditions tested. The supply air and water temperatures, COP, and water-use all increase with increasing inlet air temperature as shown in Figure 7. As expected, the ventilation supply air temperature is more sensitive to the changing inlet air temperature than the supply water temperature. Comparing the results from Figure 7 to Figure 8A shows that the total COP of the chiller decreases with increasing outdoor air dew point.

CONCLUSION

The performance of the SWEC chiller illustrates a large energy savings potential in hot, dry climates. The results also reveal that, under a wide range of weather conditions, the SWEC produced chilled water at temperatures between 60 to 66 degrees, which is desirable for serving a radiant cooling system.

The SWEC was able to consistently provide cooling loads efficiently, while operating under a variety of climate conditions. The unit consistently provided anywhere from 2.5 to 4.5 tons of cooling, with 3-4 tons being consistently typical and average. The variation in COP was seen to be from 2 to 8, with 6 to 8 being an average and typical value. For comparison, the US Department of Energy requires commercial packaged air conditioners with a capacity between 5-20 tons of cooling to have a minimum rated energy efficiency ratio of at least 11 at 95°F outdoor air temperature, which equates to a COP of 3.22 [1]. Therefore, the tested technology could roughly double the cooling efficiency delivered by standard packaged air conditioning equipment. The test results showed that the water consumption of the SWEC under normal operating conditions was 1.0 to 2.5 gallons per ton-hour of cooling delivered.

Current technologies that the SWEC may be able to functionally replace are refrigerant based chillers, cooling towers, and residential and light commercial air conditioning 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 comparable to a cooling tower. 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. However, because the SWEC can only chill water to approximately 60°F (depending on climate), it will not work for applications that require colder chilled water temperatures.

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 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.

RECOMMENDATIONS

The results of the lab evaluation of the SWEC technology show that it has great potential to reduce energy use by 50% 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. One or multiple fan coil units can potentially replace a traditional evaporator coil, and also 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 when using fan coil units in a residential or multi-family building. The analysis should determine if a fan coil thermal distribution system can meet the load in a residential building in California climate zones, or if a small backup chiller would be needed to supplement the SWEC cooling in some cases. Additional retrofit measures such as duct sealing, insulation, and envelope sealing can be used to counter-act the reduction in capacity. Because the design of the SWEC is based on simple off the shelf components, it is expected that large scale manufacturing of the SWEC can be cost competitive in comparison to compressor-based air conditioners. The analysis should also determine whether the addition of a thermal storage system makes economic sense as a peak demand reduction strategy.

In new construction, a radiant cooling installation combined with the SWEC is likely to be cost competitive with a compressor-based air conditioner with a ducted forced-air system. In China, the SWEC technology has been installed with radiant cooling systems to cool over one million square feet of commercial buildings.

REFERENCES

[1] US Department of Energy, "Small, Large, and Very Large Commercial Package Air Conditioners and Heat Pumps," [Online]. Available: https://www1.eere.energy.gov/buildings/appliance_standards/product.aspx/productid/77#standards. [Accessed 14 10 2015].

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

APPENDICES

TABLE 5.SWEC PERFORMANCE RESULTS

TEST	AMBIENT CONDITION (DB/WB) (°F)	TOTAL AIR FLOW (CFM)	SUPPLY AIR FLOW (CFM)	WATER FLOW (GPM)	RETURN WATER TEMP °F	SUPPLY WATER TEMP °F	SUPPLY AIR TEMP °F	POWER (WATTS)	WATER USE GAL/ (TON*HR)	CAPACITY (TONS)	CAPACITY UNCERTAINTY (TONS)	COP	COP UNCERTAINTY
1	90/64	1652	555	9.3	68.0	61.9	68.5	1665	1.81	3.46	0.19	7.30	0.41
2	90/64	3265	1086	12.9	68.0	62.6	74.5	3988	2.35	4.36	0.26	3.84	0.24
3	90/64	2732	933	9.3	68.0	62.1	73.1	2768	2.39	3.70	0.19	4.70	0.25
4	90/64	1202	421	9.3	68.0	63.0	70.5	1334	1.66	2.69	0.19	7.09	0.50
5	90/64	1716	0	9.2	68.0	60.6	74.1	1655	2.72	2.83	0.19	6.02	0.40
6	90/64	1770	956	9.3	68.0	63.5	70.9	1652	1.63	3.37	0.19	7.17	0.42
7	90/64	1684	1136	9.3	68.0	64.2	73.4	1649	1.41	3.16	0.19	6.75	0.42
8	90/64	1700	552	9.3	64.0	59.6	66.5	1665	2.03	2.87	0.18	6.05	0.39
9	90/64	1701	553	9.3	66.0	60.7	67.3	1661	1.93	3.16	0.19	6.70	0.40
10	90/64	1714	601	9.5	68.0	62.5	68.9	1651	1.95	3.29	0.19	7.01	0.42
11	90/64	1694	553	9.3	71.0	64.1	69.6	1657	1.72	3.71	0.19	7.88	0.42
12	105/68.8	1660	534	9.3	68.0	62.9	71.7	1649	2.30	3.55	0.19	7.57	0.41
13	95/65.7	1660	544	9.3	68.0	62.5	69.4	1657	1.96	3.39	0.19	7.21	0.41
14	85/62.3	1646	543	9.4	68.0	61.6	66.9	1674	1.64	3.39	0.19	7.12	0.41
15	75/58.7	1613	538	9.3	68.0	61.1	64.9	1678	1.36	3.17	0.19	6.65	0.40
16	65/54.8	1598	540	9.4	68.0	60.5	62.6	1688	1.05	3.05	0.19	6.36	0.40
17	105/73	1744	595	9.2	70.0	66.0	73.9	1656	2.49	3.20	0.19	6.78	0.41
18	95/70.1	1654	552	9.3	69.9	65.8	71.8	1665	1.96	2.76	0.19	5.83	0.41
19	85/67.1	1638	520	9.3	70.0	65.3	69.7	1667	1.60	2.54	0.19	5.37	0.41
20	90/64	3309	0	9.2	68.0	60.0	74.8	3884	4.35	3.08	0.19	2.79	0.17
21	105/73	3315	0	9.2	74.0	67.2	81.2	3823	5.67	2.62	0.19	2.41	0.18