Western Cooling Efficiency Center

First Annual Report on Cooling in the West

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California Energy Commission

A Report of the Western Cooling Efficiency Center

RESEARCH • INNOVATION • PARTNERSHIP

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Introduction

The hot climates of California's high growth areas are causing increased energy use for cooling. Most California buildings experience large afternoon "load spikes" that are the major cause of electric load peaks. Conventional cooling systems are sub-optimal for California; for example, they usually dehumidify unnecessarily, increasing loads and operating costs by as much as 15%. Conventional practice also suffers from inattention to integrated design strategies, application of oversized system designs, and a general failure to take advantage of natural cooling alternatives such as flushing buildings with cool night air. The WCEC catalogs and supports a range of cooling strategies that in concert can significantly and cost-effectively reduce the impact of cooling systems on California's electricity grid.

The Western Cooling Efficiency Center, in partnership with the California Energy Commission, has identified several projects that hold significant potential to increase the efficiency of cooling related technologies. This report describes these projects and their respective advances over the previous year. These projects are:

Cooling Research & Development

Combined Heat and Mass Transfer

The aim of this project is to develop the tools needed to analyze and optimize the thermal performance of an indirect evaporative cooling (IEC) heat exchanger through mathematical modeling and experimental exploration at both the macroscopic and microscopic levels. Theoretical analysis and modeling for heat and mass transfer of wetted surfaces is crucial to understanding the mechanism of evaporative cooling, as well as to optimize the process with respect to energy efficiency, water consumption, and cost. Therefore, the WCEC has been working to understand the behavior of wetted evaporative surfaces and heat exchangers.

Year 1 Accomplishments

The first step was to review the available literature to investigate evaporative heat exchangers and their respective medias in order to understand the variables impacting overall performance. The literature on this topic is limited, and tends to present individual generalized experimental studies rather than in depth theoretical analysis. Some of the materials discussed included fiber, paper, membrane, fabric, sponge, plastic, and metal; similarly, several geometric arrangements were discussed including plates, tubes, and porous structures. The literature revealed that the evaporative media material, coatings, and geometric configuration of an evaporative heat exchanger can significantly affect cooling capacity and water use. Therefore, the WCEC has developed a mathematical model of simultaneous heat and mass transfer to conduct a comprehensive analysis of the parameters that significantly affect the performance of an evaporative heat exchanger.

The WCEC's simultaneous heat and mass transfer model is unique in that it does not focus solely on evaporative effectiveness, whereas existing models seldom address anything other than evaporative effectiveness. Also, most existing models for evaporative heat exchangers have analyzed direct evaporative cooling devices, such as cooling towers, and this model is intended to investigate indirect evaporative cooling. The model currently addresses a counter flow heat exchanger, which is believed to hold the greatest potential for high-efficiency heat transfer, but which has generally been difficult to construct cost-effectively.

Though the simultaneous heat and mass transfer model is still under development, some initial observations have been made regarding the performance of the indirect evaporative counter-flow heat exchanger. Unsurprisingly, the cooling capacity is most strongly correlated to the water evaporation rate at the wetted surface. Other, less obvious, influences on performance include aerodynamic factors such as air velocity, channel height, and channel length. Conversely, the inlet water temperature and water flow rate appear to have little impact on the performance of the heat exchanger, provided the water flow rate on the wetted surface is sufficient for evaporation. All of these considerations will be accounted for in optimizing the heat exchanger to promote evaporation, which is a primary goal of the next phase of the study.

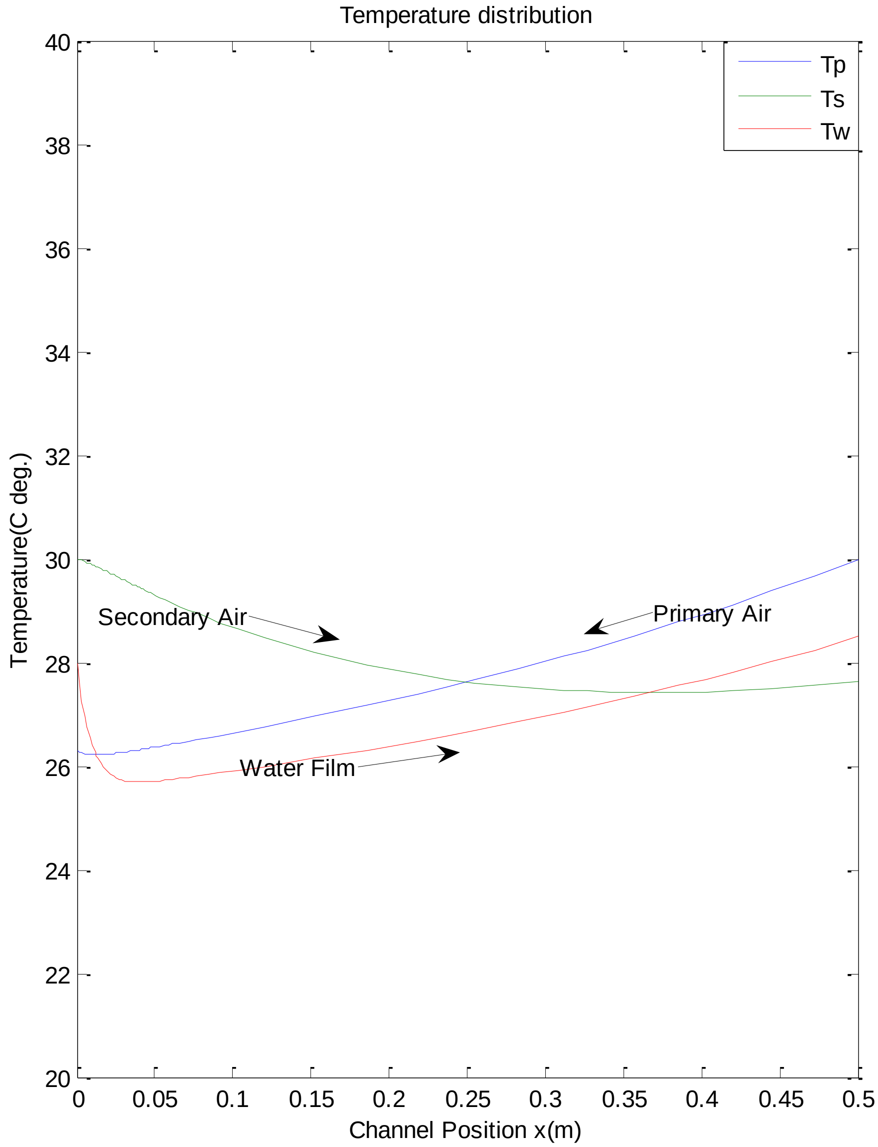

Figure 1: An example of simulation results showing the distribution of air and water temperatures in the heat exchanger. The water and secondary air enter from the left at x=0, the primary air enters from the right at x=0.5m.

Water-use Efficiency

One benchmark for optimizing hybrid evaporative/vapor-compression designs is on the basis of energy efficiency. Another parameter that should be considered is water-use efficiency. Simply comparing water-use on site by the cooling equipment is not sufficient because the generation of electricity consumes water. While compressor-based systems do not use water on-site, they do consume water through their use of electricity, which consumes water through evaporation from the lakes behind the dams at hydroelectric power plants and cooling at thermal power plants.

Theresa Pistochini and Mark Modera from the WCEC authored a paper on water-use efficiency which has been accepted for publication in *Energy and Buildings* (Appendix A). This paper defines a water-use efficiency metric and a methodology for assessing the water use of various cooling technologies. The water-use efficiencies of several example cooling technologies are compared, including direct evaporative, indirect evaporative in two different configurations, compressor-based systems, compressor-based systems with evaporative pre-cooling of condenser inlet air, and hybrid systems that consist of an indirect evaporative module combined with a compressor-based module.

The results vary depending on the method of electric power generation – compressor-based systems are more favorable from a water-use standpoint when the water consumed to produce electric power is low, and evaporative systems are more favorable when water consumed to produce electric power is high (Figure 2). It should be noted that the results in Figure 2 are based upon readily-available high-quality laboratory data, and may not be representative of newer IDEC/vapor-compression hybrid designs. The analyses presented in this paper suggest that evaporative systems that significantly reduce peak electricity demand and annual energy consumption need not consume any more water than conventional systems. The paper also stresses that designing cooling systems for arid climates is entwined in the close relationship between water and energy, and the scarcity of both resources.

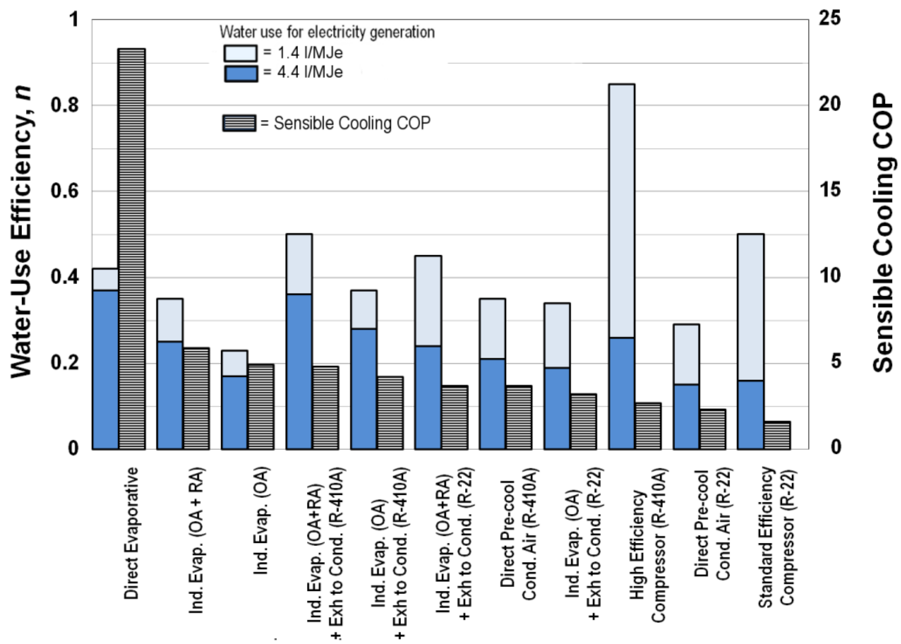

Figure 2 – Snapshot of water-use efficiency analysis for compressor-based and evaporative systems as a function of water required for electricity generation

Thermal Energy Storage

Thermal Energy Storage can strongly influence the magnitude of peak energy demand by shifting the time of day when energy is consumed. Two notable uses of thermal energy storage are of interest; the first is cooling at night to reduce or eliminate the daytime peak electricity demand associated with cooling, and the second is shifting the daytime cooling demand profile to better match on-site power generation from photovoltaic arrays. In addition to reducing the day-time peak electricity demand, other benefits of cooling at night include the greater potential to make use of non-compressor cooling. This includes the lower wet-bulb temperature at night, and radiative exchange with the cold night sky, both of which mean lower temperatures for the cooling media that cannot be achieved in the daytime. Similarly, the benefits of coupling thermal energy storage with photovoltaic arrays may allow for completely removing cooling related loads from the grid, which leads to Zero-Peak-Cooling buildings.

Year 1 Accomplishments

The initial efforts of investigating Thermal Energy Storage systems consisted of researching technical journals, and academic publications, and research articles to understand current trends and discover potential new avenues for investigation. This research led to analyzing the potential benefits of evaporative cooling at night, which takes advantage of the inherently lower wet bulb temperature (T_{wb}). The preliminary results of this analysis have shown that the advantage gained by evaporatively cooling at night is that T_{wb} is approximately 5˚C lower than T_{wb} during the day.

After communicating with various retailers that had expressed an interest in low-peak-demand cooling, Wal-Mart identified a radiantly cooled store as a candidate for taking advantage of a low-cost chilled water storage system. The initial concept was to store water beneath the slab, but, unfortunately, this concept was not well received. Furthermore, any storage above ground was deemed too expensive in terms of cost as well as real estate consumed. In addition, it was determined that Chilled Water Storage was not the cheapest, nor was it the quickest, potential solution to thermal energy storage. Therefore, the current Thermal Energy Storage system under investigation uses the mass of the concrete slab, plus the mass of the internal building contents, such as merchandise in large retail stores, to reduce cooling during peak hours.

Swimming Pools as Heat Sinks for Unitary Air Conditioners

The basic premise of this effort is that simultaneously rejecting heat from condensers to the atmosphere, while burning natural gas to heat swimming pools is imprudent. Therefore, by rejecting condenser heat to a swimming pool instead of ambient air, the energy is transferred instead of wasted. Furthermore, the reduction in sink temperatures seen by the condenser reduces compressor energy consumption during most hours of the day that require cooling. The savings realized during peak conditions will be most significant since ambient air temperatures often exceed 100°F while pool temperatures stay relatively constant between 80-85°F.

Another advantage arises out of the improved heat transfer properties of water relative to air, which allows refrigerant temperatures to be only 20°F higher than the sink temperatures. By comparison, an analogous air cooled condenser requires the refrigerant temperature to be 35°F higher than the sink temperature. Therefore, rejecting heat to a swimming pool can reduce condensing refrigerant temperatures by 30-35°F during peak conditions. The efficiency of R-410A increases by about 2% for every degree reduction in refrigerant temperatures which means these systems have the potential to improve overall vapor-compression efficiency by more than 50% during peak conditions.

Year 1 Accomplishments

To better understand the thermal interaction between a pool and its environment a model was created to calculate all heat transfer to and from a pool. The model was validated with an experiment at a pool, in Davis, CA, that was passively exchanging heat with its surroundings. Local weather data was measured on-site and used as the input for the model, along with the physical characteristics of the pool and site. The hourly predicted pool temperatures were then compared to measured pool temperatures over a 56-day period. The comparison of predicted and observed pool temperature for all hours showed an R-squared of 0.967 with the maximum error being 1.1°C (Figure 3). A paper discussing the model and the above mentioned experimental validation has been accepted for publication in *Building and Environment* Journal (Appendix B).

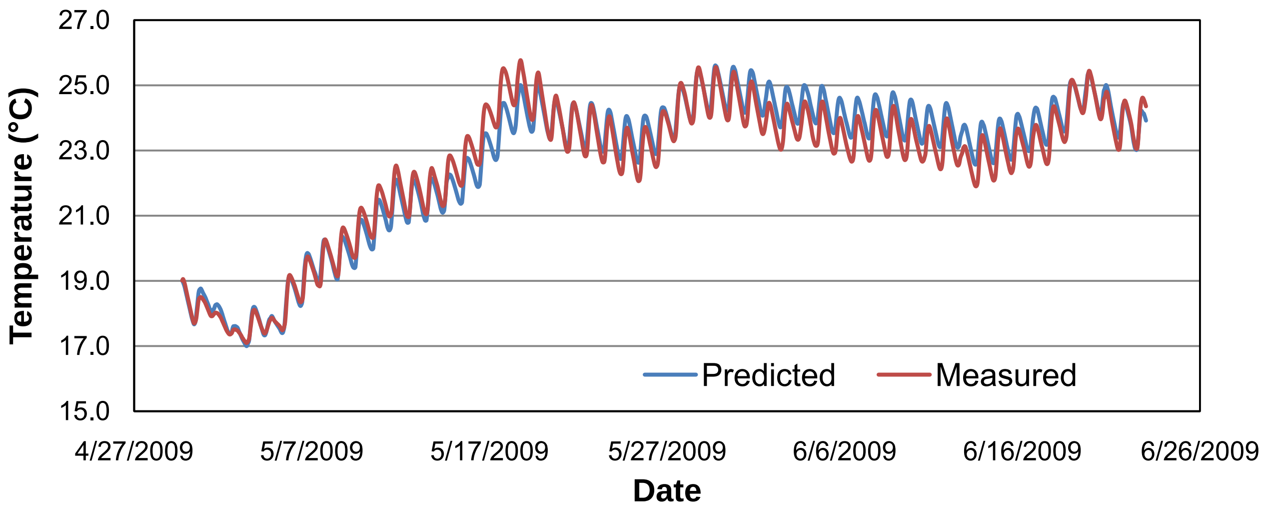

Figure 3: Comparison of predicted and measured pool temperatures (observed 4/29/2009-6/22/2009 in Davis, CA)

To further validate the pool model (including the influence of HVAC heat rejection or removal), an experiment was set up to monitor a residential swimming pool that is serving as both a heat sink and source for a heat pump. The system was designed and installed by Geremia Pools, a local pool installer in Sacramento. Data has been collected for several months during both the cooling and heating seasons. In addition to the heat exchange between the pool and the heat pump, we are also monitoring solar thermal panels for rejecting heat at night, and geothermal panels for exchanging heat with the ground.

Again, the model was able to accurately predict pool temperatures. The hourly predicted and observed pool temperatures are plotted over the 95 day test period, and had an R-squared of 0.982 with the maximum error being 1.9°C (Figure 4).

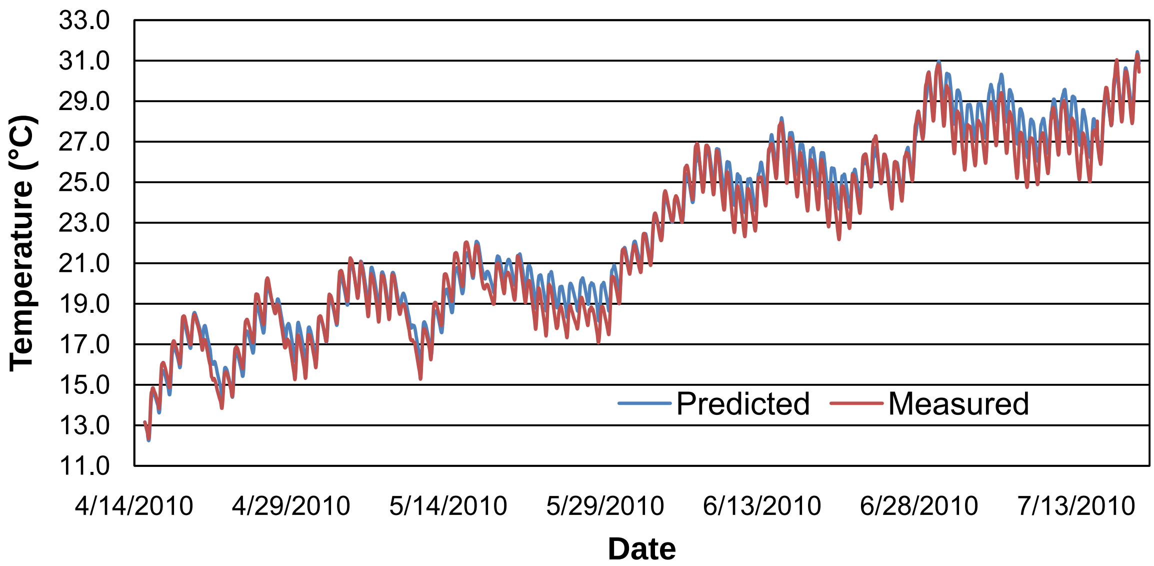

Figure 4: Comparison of predicted and measured pool temperatures (observed 4/15/2010-7/18/2010 in Sacramento, CA)

During this test period, there was 4480 kBtu of cooling applied to the pool (between the beginning of June and the end of the test period) and 2322 kBtu of heat extracted from the pool (between the beginning of the test and the middle of May). This test affirms that the model can be used to accurately predict the thermal behavior of a swimming pool that is used as a heat sink/source for a heat pump. This implies that it can be used for performance prediction, and therefore for design purposes. One key issue moving forward will be the quantity/quality of input data required.

Western Cooling Challenge – Next Generation Rooftop Units

The Western Cooling Challenge (WCC) is a multiple winner competition hosted by WCEC that encourages HVAC manufactures to develop the next generation of rooftop air conditioning equipment for western climates. The units in design, testing, and demonstration are all some variation of a hybrid unit that couples indirect evaporative cooling and high-efficiency vapor compression. WCC Certification requires that laboratory testing of a unit indicate at least 40% demand and energy savings compared to DOE 2010 standards at WCC annual and peak weather conditions. Coolerado's H80 is the first certified equipment; according to National Renewable Energy Laboratory tests, the 5-ton RTU could reduce annual energy use by almost 80% and achieve over 60% peak demand savings.

Year 1 Accomplishments

Work to date on the WCC has produced a number of important milestones, including a public kickoff of the Challenge, clear definition of the test criteria and performance requirements, successful laboratory testing of one challenge entry, initiation of field demonstration of that entry, and coordination with California utilities to begin development of financial incentive programs for WCC equipment. The following sections summarize activities, findings, and information related to several different aspects of the project.

WCC requirements, test points, and performance criteria

The Western Cooling Challenge criteria were developed in such a way that incremental improvements to a conventional vapor compression cycle would be unlikely to allow such a unit to meet the criteria in a practical manner. However, the Challenge was also designed such that conventional HVAC equipment could qualify with the addition of commercially-available add-on evaporative technologies. The intent was to encourage manufacturers to develop and commercialize hybrid units that integrate these efficiency improving components into a single package without requiring a ground-up re-design, which would likely discourage major manufacturers. Thus, partnerships between manufacturers to submit combinations of add-on evaporative components with high-efficiency conventional rooftop units are encouraged. The original Western Cooling Challenge requirements can be found in Appendix C.

In summary, the Challenge invites manufacturers to design and commercialize rooftop packaged units that meet the following key criteria:

- Minimum sensible EER of 14.0 at full capacity operation, with 120cfm/nominal-ton ventilation rate, under WCC Nominal Peak Conditions
- Minimum sensible EER of 17 at full capacity operation, with 120cfm/nominal ton ventilation rate, under WCC Surrogate Annual Conditions
- Provide some dehumidification, $\Delta\omega=0.000363$ lb/lb
- Maximum water use of 4gal/ton-hr
- Demonstrated minimum manufacturing capacity of 500 units/year
- Ability to detect and communicate performance degradation

Other points that would improve cooling and ventilation efficiency were considered, such as requiring variable speed supply fans, but the criteria were limited in part to allow standard packaged units to compete by focusing on inclusion of evaporative components.

Table 1: Test conditions and performance requirements for the Western Cooling Challenge

	ARI 340/360	Peak	Surrogate Annual	Units
Outside Air Dry Bulb	95	105	90	°F
Outside Air Wet Bulb	75	73	64	°F
Return Air Dry Bulb	80	78	78	°F
Return Air Wet Bulb	67	64	64	°F
Outdoor Ventilation	0	120	120	cfm/ton
External Static	0.2-0.75	0.7	0.7	Inches WC
Minimum Sensible EER	NA	14	17	Kbtu/h/kW
Maximum Water Use	NA	4	4	gal/ton-hr
Max Supply Humidity	NA	0.0092	0.0092	lb/lb

Table 1 outlines the laboratory test conditions, and key performance requirements for Western Cooling Challenge certification. The Nominal Peak and Surrogate Annual test conditions chosen are not indicative of any particular climate zone or set of climate zones, but were set as generalized conditions that are indicative of summertime conditions in cooling-intensive western regions. Minimum energy performance criteria was developed by estimating the savings that could be achieved under these conditions by a high-efficiency conventional rooftop unit fitted with evaporative condenser-air pre-cooling and indirect evaporative cooling of ventilation air, without a significant increase in auxiliary loads.

WCEC also recognized that evaporative equipment generally requires the addition of outdoor air to a space, but that that space may or may not require that much outdoor air. Thus, a "reasonable" amount of outdoor air requirement had to be chosen. The value chosen was 120 cfm/nominal-ton of cooling capacity, which corresponds to 30% of the "nominal" flow for typical vapor-compression equipment. In terms of laboratory testing, they must be conducted at a minimum of 120 cfm/ton outdoor air. Units can be operated at larger outdoor air rates, how the additional capacity for cooling more than 120 cfm/nominal-ton outdoor air from outdoor conditions to indoor conditions is not counted in calculations of capacity and energy efficiency.

Revised WCC requirements, test points, and performance criteria; Determination of Nominal Capacity for WCC Equipment

The original WCC criteria required one test at AHRI conditions to define a nominal capacity that would be comparable to conventional rooftop packaged units (used for calculating the flow associated with 120 cfm/ton). However, upon submission of Coolerado's H80 it was recognized that an AHRI nominal capacity cannot be determined unless the unit can operate with 0% outdoor air; Coolerado's minimum outdoor air fraction is approximately 45% for the conditions and operating modes tested. In an effort to define a nominal capacity that is nearly comparable to an AHRI determination, the WCC criteria was amended to use measured data from full capacity operation under WCC Nominal Peak conditions to calculate a nominal capacity using the following equation:

$$\dot{H}_{nominal} = \dot{V}_{SA} \cdot v_{SA} \cdot (31.5 - h_{SA})$$

where 31.5 is the specific enthalpy of return air for AHRI nominal capacity tests (Btu/lbm), h_{SA} is the specific enthalpy of the supply air (Btu/lbm), $\dot{V}_{SA}$ is the volumetric flow rate of supply air (cfm), and v_{SA} is the density of the supply air (lbm/ft^3). This 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 ARI test scenario. However, it does not represent space cooling capacity under ARI outdoor air conditions, nor does it represent an actual space cooling capacity that would be achieved under any particular condition since the measurements are taken during full capacity operation at WCC peak conditions and the results are mingled after the fact with the enthalpy value of AHRI return air.

The space cooling capacity of WCC equipment would be significantly lower if tested with AHRI outdoor air and return air conditions, but such a metric would not provide a fair basis for determination of what size conventional equipment could be replaced by a hybrid machine in Western climates. For example, a conventional machine that has a nominal space cooling capacity of 60 kbtu/h at AHRI standard rating conditions might only have 43 kbtu/h sensible space cooling capacity, and this would slip to less than 30 kbtu/h with 30% outdoor air at WCC peak conditions. The trend would be opposite for WCC equipment; in fact the sensible space cooling capacity of the Coolerado H80 is nearly 40 kbtu/h at WCC peak conditions. Since the H80 provides as much sensible space cooling as a 6-8 ton conventional unit under WCC peak conditions, it would not be appropriate to report a nominal capacity determined at AHRI conditions.

For further information, reference the revised Western Cooling Challenge requirements in Appendix D.

WCC Participant Manufacturers

The roster of participant manufacturers and their respective progress is continuously evolving. Originally 12 manufacturers enrolled in the WCC, though only Coolerado has qualified for certification through WCEC-observed laboratory testing, and some manufacturers have backed out of the Challenge completely. Several manufacturers have shown promise as future participants, and the WCEC will publish more information about these manufacturers and their respective systems as more details become publicly available.

Description of Western Cooling Challenge Technologies in Design, Development, and Commercialization

Though the Coolerado H80 is the only complete entry thus far, several other manufacturers are in the process of developing entries, which WCEC aims to have laboratory tested and field demonstrated as soon as the products meet WCC requirements for commercial availability. Interestingly, every system design that has been proposed is different, though each uses one or another form of indirect evaporative cooling in conjunction with vapor compression, and all evaporatively cool process air for the condenser. A description of each system design concept proposed follows:

1. Maisotsenko Cycle Indirect Evaporative Cooler (IEC) applied in series with a vapor-compression cooler (DX) to cool mixed air, where the IEC secondary air exhaust is used as DX condenser air.
2. Counterflow IEC applied in series with DX to cool mixed air, where a fraction of the primary air outlet from the IEC is used as the secondary air; and a separate direct evaporative cooler pre-cools DX condenser air.
3. Evaporative pre-cooling of DX condenser air using outdoor air, where the evaporatively-cooled sump water circulates through a water coil to cool ventilation air, and DX is applied to the mixed air stream.
4. Evaporative cooling of DX condenser air using return air, where the evaporatively-cooled sump water circulates through a water coil applied in series with DX and a Direct Evaporative Cooler (DEC) to cool supply air.

Laboratory Testing of Western Cooling Challenge Equipment

The National Renewable Energy Laboratory (NREL) conducted performance testing of the Coolerado H80 for the Western Cooling Challenge. The complete technical report, published September 2009, is attached as Appendix E. The H80 was tested at both WCC psychrometric conditions, and under three different operating modes. These tests showed this unit to be extremely energy efficient.

Table 2 summarizes the key laboratory results under the WCC conditions tested. Operating with indirect evaporative cooling only at WCC surrogate annual conditions, the equipment produced an EER of 52.2 at 1827 SCFM while providing a net total cooling capacity of 50.4 kbtu/h. Note that “net total cooling” is the metric by which capacity for cooling more than 120cfm/rated ton outdoor air is discounted as discussed previously.

Table 2: Key results from NREL Laboratory Test of the Coolerado H80. [1]

	Outside Air Conditions Tdb°F/Twb°F	Return Air Conditions Tdb°F/Twb°F	Mode*	Supply Air Flow SCFM	Supply Air Tdb°F/Twb°F	Total Space Cooling kBTU/hr	Total Ventilation Cooling kBTU/hr	Net Total Cooling kBTU/hr	Sensible EER
Peak Conditions	105/73	78/64	0	1822	71.5	10.44	18.95	29.39	40.8
			1	1834	60.5	33.78	19.30	53.08	26.4
			2	1810	58.4	42.28	19.39	61.67	20.1
Surrogate Annual Conditions	90/64	78/64	0	1827	67.5	31.28	19.15	50.43	52.3
			1	1826	56.7	53.32	19.51	72.83	31.5
			2	1806	54.2	62.17	19.61	81.78	24.2

Field Demonstration of Western Cooling Challenge Equipment

WCEC began field demonstration of WCC equipment during the 2010 cooling season. Two demonstrations were organized; one at the Los Angeles Community College District's LA Trade Technical College campus, and another at the University of California, Davis. The demonstration at UC Davis has proceeded well and the H80 is set to be installed as a retrofit at a small office building. The existing RTU was monitored for several weeks prior to retrofit and monitoring of the H80 will proceed at least through the end of the 2011 cooling season. Organization of the demonstration at LA Trade Technical College ran into a number of technical and administrative issues, and has yet to proceed. However, the unit has been purchased and other locations within the LA Community College District are being considered for the demonstration. Additionally, WCEC is in the midst of organizing scaled field placement programs through California utilities, the goal of which is to place numerous WCC units in the field for monitoring over the 2011 cooling season.

Rebates & Incentives for WCC Equipment

In order to advance the market introduction of these next generation high efficiency RTUs, WCEC is collaborating with California electric utilities to develop incentive programs specifically for WCC-certified equipment. The intent would be to provide incentives that would make WCC certified equipment immediately cost-competitive with other energy efficiency investment alternatives. .

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[1] *Mode 0 = Indirect evaporative cooling only, Mode 1 = Indirect Evaporative + First Stage Compressor Cooling, Mode 2 = Indirect Evaporative + Second Stage Compressor Cooling.*

Collaboration thus far with California utilities has identified the need for more solid estimates of savings in various building applications and climate zones before incentive programs can be structured.

Modeling Tools

The need for accurate estimates of energy savings to be achieved by WCC equipment in different building types, vintages, and climate zones is important to deploy incentive programs through California Utilities and to understand the potential impact on energy use for cooling across the state and other western climates. However, there is no building energy modeling tool currently available that can simulate the energy impact of WCC equipment. WCEC has considered a number of different approaches using simulation tools such as ePlus, but until these programs are developed further, there is no straightforward method to simulate system performance in a variety of different scenarios.

NREL, LBNL, and other groups familiar with development of simulation tools, have expressed interest in working with WCEC to develop ePlus solution to model WCC equipment. However, until a reliable scheme exists, equipment performance modeling relies on post-processing of the outputs from building simulations with typical vapor compression systems, or estimates of savings based on results from the NREL tests and information from the Database for Energy Efficient Resources (DEER), California Commercial End-Use Survey, EIA's Commercial Building Energy Consumption Survey, and assumptions about fractional cooling energy and demand savings.

Condenser-Air Pre-Cooling

This project involves designing, monitoring, and analyzing energy savings from, a condenser-air pre-cooling retrofit of a Big-Box retail store (Target) in Davis, CA. The project includes evaporative pre-cooling equipment supplied by three manufacturers:

1. Two DualCool units by Integrated Comfort, which pre-cool roof top unit (RTU) ventilation air and condenser air for energy and peak power savings. Our modeling predicted 20% energy and peak power savings.
2. Ten WicKool units by Octus Energy, which passively use available condensate for RTU condenser air pre-cooling. Modeling estimates a modest 3% energy savings, but the low cost of the product is more than paid for by the cost savings associated with eliminating condensate drain piping.
3. One FlashCool unit by Beutler Corporation, which pre-cools condenser air for the 50-ton refrigeration system and adds a variable frequency drive (VFD) control to the condenser fans.

Year 1 Accomplishments

In Year 1 an energy saving analysis was completed and a retrofit plan was recommended to Target, an energy monitoring plan was devised, and all the retrofits were installed. In Year 2 the equipment will be monitored, the data analyzed, and the results documented and presented.

Energy Saving Analysis

An energy savings analysis was completed for the retrofit plan, and the analysis methods are available in a separate report to the energy commission[2]. Contact WCEC for a draft of this report if desired. The results of the analysis are presented here (Table 3) to allow the reader to evaluate the potential impact of this project.

[2] Pistochini, Theresa, et al. Western Cooling Efficiency Center, UC Davis. 2010. Evaporative Cooling Retrofits for Retail Buildings. California Energy Commission. Publication number: Pending

Table 3: Energy Savings Estimates for Evaporative Cooling Retrofits on Davis Target Store

Technology and Equipment Retrofitted	Cooling Tons Retrofitted[3]	Retrofit Cost[4]	One Time Savings		Yearly Savings				Simple Payback (years)
			Condensate Drain Savings[5]	Rebate/ Incentive[6]	kWh Energy Savings[7]	Peak Demand Savings[8]	Energy Cost Savings[9]	PDP Cost Savings[10]	
WICKOOL RTU-01 Guest Service RTU-02 Entry-Vestibule RTU-03 Entry-FS Seating RTU-04 Food Service RTU-05 Office-Pharm-Lounge RTU-07 Sales RTU-10 Sales RTU-12 Marking RTU-13 Stock RTU-14 Control Room	126 tons	$5,010	$8,570	$739	4,929 kWh	NA	$707	NA	Immediate
DUALCOOL RTU - 06 Sales RTU - 09 Sales	48 tons	$18,043	$1,714	$4,168	17,535 kWh	15.4 kW	$3,629	$567	2.9 years
FLASHCOOL Refrigeration	50 tons	$21,938	NA	$7,530	38,201 kWh	18 kW	$6,583	$1,958	1.7 years
TOTAL PROJECT SUMMARY	224 tons	$44,991	$10,284	$12,760	60,671 kWh	38.6 kW	$10,919	$2,525	1.7 years

[3] Nominal specification on RTU or refrigeration Equipment, not all condenser air will have pre-cooling
[4] Based on estimates from Beutler, Integrated Comfort, and Octus Energy. Includes materials and installation.
[5] Todd Udenberg estimated condensate piping cost $12,000 per store. Estimated savings of $12,000/14 = $857 per RTU
[6] Estimated from typical PGE Non-residential retrofit (NRR) incentive of $0.15/kWh and $100/kW.
[7] Estimated yearly energy savings
[8] Estimated peak demand savings
[9] Based on energy savings and PGE A-10 Rate schedule
[10] Based on PG&E Peak Day Pricing (PDP) adder of $1.20/kWh, 12 event days

Equipment Installation

Installation of all three retrofit technologies commenced July 19, 2010. Installation of the FlashCool on the 50-ton refrigeration condenser was completed by Beutler. The installation consisted of:

1. Mounting the nine cells with high pressure spray nozzles and a "drift eliminator" to catch droplets to the condenser rack (Figure 5)
2. Supplying water from the roof source using ¾" PVC piping to the high pressure pump package installed near the rack.
3. Installing the variable frequency drive (VFD) for the condenser fans.
4. Supplying electricity to the VFD from the electrical room approximately 150ft away.
5. Supplying electricity from the electrical room approximately 150ft away to the high pressure pump package which provides water to the spay nozzles at 125-150psi at ten incremental flow rates through four solenoid valves.
6. Integrating sensor information with the VFD, including outdoor air temperature, humidity, and head pressure in the refrigeration loop. The VFD speed is selected to maintain a target head pressure in the refrigeration loop. The outdoor air conditions and VFD speed are used to select water flow rate. The VFD speed and alarms are sent back to Target's energy management system for monitoring.

Two DualCool units were installed on the first stages of two 24-ton Lennox Strategos SGB288H4M units. The installation consisted of:

1. Mounting the condenser direct pre-cooler assembly in front of the RTU condenser coil (Figure 6), which consists of the steel frame and sump assembly, circulation pump, 8" deep Munters CelDek evaporative media, and controller.
2. Mounting the ventilation air indirect pre-cooler to the outdoor air intake and connecting the supply water from the cold sump and the return water to the top of the Munters media.
3. Supplying water from the roof source to the sump using ¾" PVC piping.
4. Supplying electricity to the pump and controller from the RTU 120VAC source.
5. Collecting the evaporator condensate from the RTU and piping it to the sump.

The ten WicKool units were installed on the first stages of ten RTUs varying in size from 4-24 tons. The installation consisted of:

1. Mounting the WicKool tray and evaporative media in front of the first stage condenser coil (Figure 7).
2. Piping the evaporator condensate into the Wickool tray.

Figure 5 - FlashCool Installation on Davis Target store

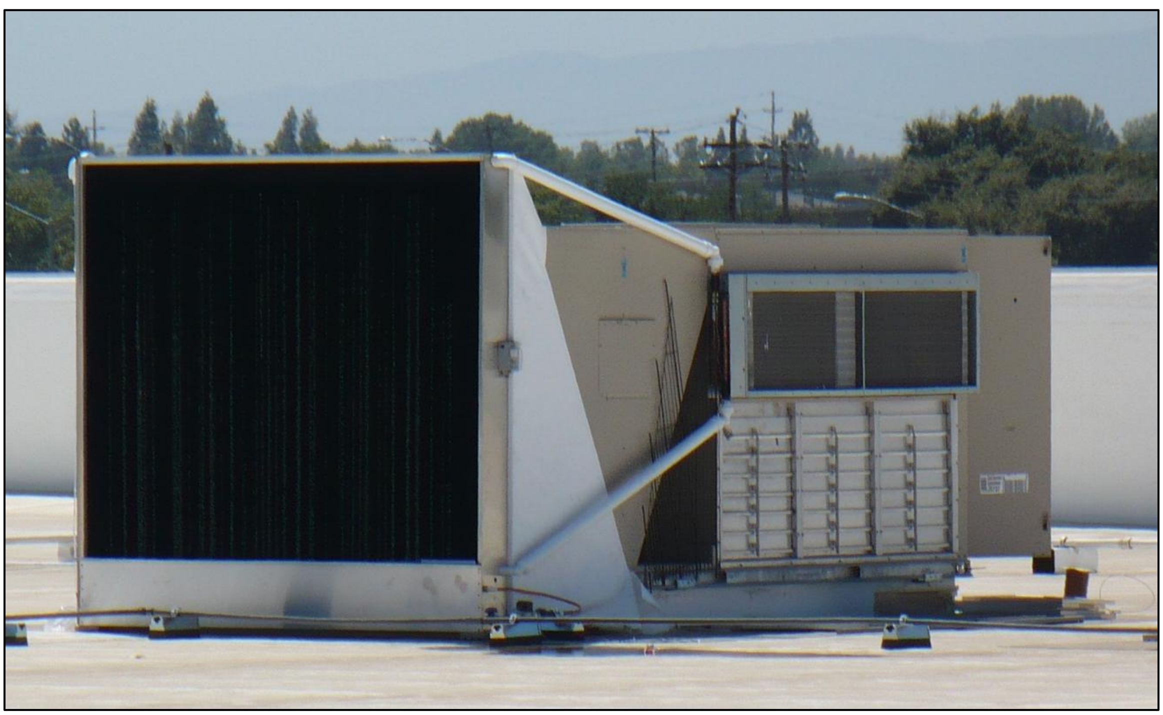

Figure 6 - DualCool Installation on Davis Target store

Figure 7 - WicKool installation on Davis Target store

Monitoring Strategy

Outdoor air conditions (temperature and relative humidity) on the roof of Target are monitored using a radiation-shielded Hobo micro station mounted on the north side of an RTU. Energy consumption and efficiency of baseline and retrofit equipment will be quantified in relation to outdoor temperature for estimation of annual energy savings using typical weather files. The store is sub-metered with each RTU and the refrigeration system having its own power meter. In addition, the store energy management system (EMS) provides useful information on the status of each RTU, such as cooling stage, fan speed, and outdoor/return air damper position. Additional instrumentation was installed by WCEC as needed for each technology. The strategy for evaluating each technology is described here and instrumentation is detailed in Table 4.

FlashCool - The monitoring strategy for Flashcool is to log power and energy consumption for the refrigeration compressor/condenser rack before and after retrofit. Because the water pump pressurizing the spray water is on a separate electrical circuit, this post-retrofit load will be monitored separately and added to the total power and energy consumption. Temperatures and pressures throughout the refrigeration system will be monitored to assure that the system is working properly. Water consumption will also be monitored.

DualCool - The RTU ventilation schedule will be modified as part of this demonstration so that all outdoor ventilation air for the sales units is supplied through the units retrofitted with Dual Cool. Therefore monitoring will be done in three phases:

1. Baseline energy use and efficiency for all sales floor RTUs, original ventilation schedule
2. Retrofit energy use and efficiency for all sales floor RTUs, original ventilation schedule
3. Retrofit energy use and efficiency for all sales floor RTUs, all sales floor ventilation through Dual Cool

The challenge with monitoring the Dual Cool system is calculating capacity, which is a function of the flow rates and conditions of ventilation outdoor air (OA), return air (RA) and supply air (SA), so that EER can be calculated from the power measurements. The EER of the RTU is:

$$EER = \left[1.08 \times \left(\dot{V}_{OA} \times \left(T_{DB,OA} - T_{DB,SA}\right) + \dot{V}_{RA} \times \left(T_{DB,RA} - T_{DB,SA}\right)\right) + L_v \dot{m}_{condensate}\right] / \mathrm{P}$$

where 1.08 is the conversion factor to Btu/hr from CFM·Δ°F for air with density 0.75 lb/ft^3 and specific heat 0.24 BTU/lb·°F, $\dot{V}$ is the volumetric flow rate of air in cubic feet per minute (CFM), T is temperature in °F for the air location specified, L_v is the latent heat of vaporization in Btu/lb, $\dot{m}_{condensate}$ is the condensate generated by the evaporator coil in lb/hr, and P is power in watts.

In order to calculate supply air flow rate, a curve for fan speed versus flow rate for the variable speed blower will be generated at the beginning of the experiment using manufacturer's blower data. The volume of air flowing through the ventilation coil is calculated by measuring the temperature drop of the water across the coil along with the known water flow rate. Based on manufacturer data, at a water flow rate of 10.2 GPM for the ventilation coil, the outdoor air flow rate in CFM is:

$$\dot{V}_{OA} = 403.7 e^{4.95\left[\frac{\Delta T_{water}}{T_{DB,OA} - T_{water,in}}\right]}$$

The return air volume flow is then the difference between supply air and outdoor air:

$$\dot{V}_{RA} = \dot{V}_{SA} - \dot{V}_{\mathrm{OA}}$$

Wickool – Because the energy efficiency improvement with WicKool is anticipated to be small (~3%), it is difficult to measure and verify in the field. Power consumption versus outdoor air temperature for the 10 RTUs with WicKool will be analyzed before and after the retrofit, but the results may not be statistically significant. However, the condensate generation will be measured before it reaches the WicKool tray. A float switch will be placed in the WicKool tray to determine if the tray overflows and under what conditions. The condensate evaporated can be used to estimate energy savings by using manufacturer data on efficiency versus condenser inlet air temperature for the RTU.

Table 4: Monitoring methods and instrumentation, pictures shown in Figures 8 - 13.

Measurement	Retrofit Tech	Log Interval	Sensor Info	Accuracy
Outdoor air temperature	All	2 min	Onset S-THB-M002	±0.4°F
Outdoor air relative humidity	All	2 min	Onset S-THB-M002	±2.5% of reading
Unit power consumption	All	15 min (average)	ADM-3612	Meets or exceeds ANSI C12.1-2001
VFD Speed	Refrigeration	2 min	Danfoss VFD output	NA
Pump status	Refrigeration	Open/close contact	CSV-A8	NA
Water consumption	Refrigeration, Dual Cool (2)	Pulse counter	DLJSJ75C	±0.5%
Supply air Temperature (averaging TC, 12 nodes)	Dual Cool (2)	2 min	Onset U12	±1.0°F
Return air temperature/RH	Dual Cool (2)	2 min	Onset U23-002	Temp: 0.4°F, RH: 2.5% of reading
Vent air coil water temp in	Dual Cool (2)	2 min	Onset U23-003	± 0.4°F
Vent air coil ΔT (averaging thermopile)	Dual Cool (2)	2 min	Onset U12	± 0.5°F
Condensate generation rate	All RTUs	Open/close contact	Condensate pump/run time logged by state logger	
VFD speed	All RTUs	5 min	Reported by RTU to Target EMS	
OA/RA Damper position	All RTUs	5 min	Reported by RTU to Target EMS	
Cooling Stage	All RTUs	1 min	Reported by RTU to Target EMS	

Figure 8: WCEC instrumentation, weather station.

Figure 9: WCEC instrumentation, averaging thermocouples for supply air.

Figure 10: WCEC instrumentation, Dual Cool water temperature sensors.

Figure 11: WCEC instrumentation, water meter for makeup water.

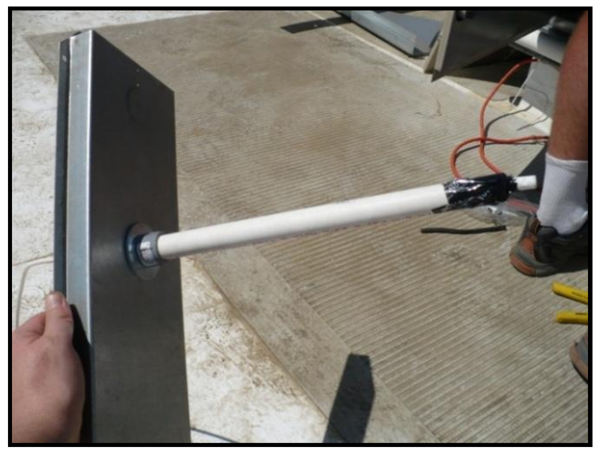

Figure 12: WCEC instrumentation, return air temperature probe.

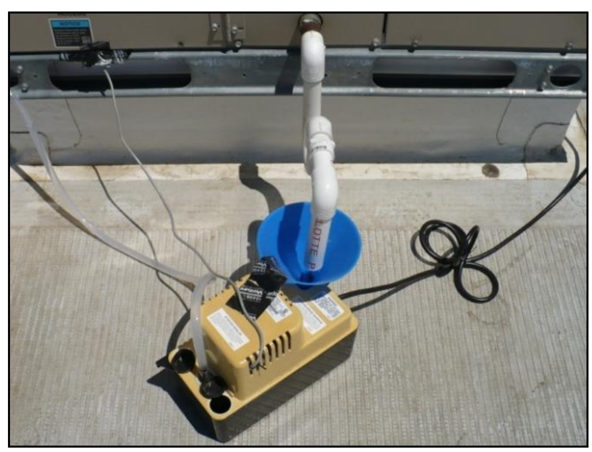

Figure 13: WCEC instrumentation, evaporator condensate meter.

Industry Application of Advanced Cooling Technologies

Cooling Industry Support

Over the course of the past year, the WCEC has engaged in numerous industry association activities, including founding and supporting a new industry association (Western HVAC Performance Alliance), in addition to supporting the industry through the usual ASHRAE (American Society of Heating Refrigerating and Air-conditioning Engineers) channel. Representatives of the WCEC attended the two ASHRAE national meetings during the past year, participating actively in several committees, including SPC 152 (Method of test for residential thermal energy distribution), GPC 1.2 (Commissioning of existing buildings), TC 5.7 (evaporative cooling), TC 6.5 (radiant cooling), and TC 6.9 (thermal energy storage). In addition, Mark Modera served on the ASHRAE Handbook committee, supporting the revision process for the 2011 Applications Handbook.

Western HVAC Performance Alliance

Throughout the year, the WCEC has provided significant programmatic support to the Investor Owned Utilities (IOUs) in California for the administration of the Western HVAC Performance Alliance. To lay the groundwork for this Alliance, in 2009 the WCEC co-hosted an "HVAC Energy Efficiency Roundtable" in San Francisco. This event brought together, for the first time ever, the leaders of the HVAC industry—including individuals and organizations representing contractors, distributors, manufacturers, labor, educators, code bodies, building inspectors, regulators, researchers, municipal and investor owned utilities (including managers of integrated demand-side management, work force training, codes and standards, and emerging technologies programs)—to chart a course to achieve the ambitious greenhouse-gas reduction goals set by California's legislature and regulators.

The Roundtable was one of the first steps in the utilities' plan to partner with the industry to achieve the goals set forth in the California Long Term Energy Efficiency Strategic Plan (www.californiaenergyefficiency.com). The Strategic Plan calls for a "transformation" of the HVAC industry in order to meet specific targets:

Goal		Goal Results
1. Consistent and effective compliance, enforcement and verification of HVAC-related building and appliance standards.	→	HVAC-related permits are obtained for 50 percent of installations by 2015 and 90+ percent by 2020.
2. Quality installation and maintenance becomes the industry and market norm.	→	By 2020, 100 percent of systems are installed to quality standards and optimally maintained throughout their useful life.
3. Whole building design and construction practices fully integrate building performance objectives to reduce cooling and heating loads.	→	Integrated design and construction practices are standard practice by 2020.
4. New climate-appropriate HVAC technologies (equipment and controls, including system diagnostics) are developed with accelerated market penetration.	→	At least 15 percent of equipment shipments are optimized for California's climate by 2015 and 70 percent by 2020.

Figure 14: California Long Term Energy Efficiency Strategic Plan for transforming HVAC industry.

Table 5 shows some of the salient issues that were brought up in this Roundtable. In light of the unprecedented teamwork that is needed to achieve the goals of the strategic plan, the WCEC was asked to help to create an industry-wide alliance to address some of the near-term and longer-term issues. This Alliance is to provide ongoing guidance to the utilities, policymakers, and other stakeholders to ensure that California's bold greenhouse gas reduction goals are met.

Table 5: Findings from 2009 HVAC Roundtable

"Enforcement!" An HVAC industry mantra was created: "Enforcement, enforcement, enforcement!". A group of participants felt that if only the laws on the books were enforced, (re-obtaining building permits and complying with Title 24), then a lot of the other issues that were discussed would take care of themselves.

Investing in the Workforce. The kind of workforce needed to conduct all the Quality Installation and Quality Maintenance services will require some investment. This includes:

- providing more and better designed training opportunities,
- finding ways to recognize the status of a technician who is capable of superior work,
- differentiating between "quality" firms and their low-price competitors,
- finding ways to make the industry "sexier" and to reach out to the new workforce (emphasizing the use of innovative technology and environmental protection), and
- providing the "soft" skills needed to develop trusting partnerships with customers and influence buying decisions.

Low Barrier to Entry. One of the reasons why there are so many contractors who are providing low-quality work at a low price is that there is a very small barrier to entry in this field. Becoming an HVAC contractor can take relatively little training and relatively little startup capital. This can be addressed through efforts to improve training, require certification of providers, increasing the market value of high-quality work, and requiring contractors licensing exams more frequently.

The Importance of Customer Awareness. So many of the elements needed to provide high quality HVAC technology, equipment, installation, and maintenance come back to one critical element: customers' recognition of the value of this quality. HVAC is seen as a low-cost "commodity" rather than a part of the complex system that creates comfort and uses (or conserves) natural resources. As long as customers will always go for the lowest bid, without weighing alternatives on quality, it will be an uphill battle to change the industry.

Timing is Critical in this Industry. HVAC installation and servicing is an extremely seasonal industry. On the first very hot and on the first very cold day of the season, customers suddenly find out that their equipment is not working, and need it fixed or replaced urgently. Those seasons can be incredibly busy. Conversely, during the "off season", it is sometimes difficult to keep employees busy. This must be taken into account in the design and implementation of energy efficiency programs.

"Run to Fail" is a Flawed Model. Most residential and small commercial customers are not aware that there may be an optimal time to replace a piece of HVAC equipment, *before* it fails. They often do not pay any attention to the unit until it fails to provide comfort on an extreme day. At that point, they need their unit replaced immediately, and they have few options. Ideally, a customer will be made aware that their system is not performing efficiently or effectively anymore, or that it may have to be replaced shortly. At that point, the customer has ample time to research the lowest life-cycle cost solutions, order something that may not already be in the dealer's lot, and get the best contractor to install it. Ongoing service contracts, remote diagnostics, and analysis software may help to provide this "warning" that now is the time.

Table 5: Findings from 2009 HVAC Roundtable (cont.)

<u>Service Contracts Provide Opportunities</u>. Since HVAC is considered a low-cost commodity, customers who resort to using the Yellow Pages to find a servicing contractor will be understandably distrustful of that person's recommendations. Once a sense of trust has been established between a provider and a customer, the opportunity is created for things like early equipment retirement, "upselling" a more efficient and more expensive unit, and providing all the necessary maintenance. Bundling quality maintenance with a quality installation offering will provide the most persistent savings.

<u>Remote System Diagnostics</u>. Diagnostics technology which is starting to come out will have a great benefit for improving service productivity and providing an ongoing benefit that will go a long way in establishing a long term relationship with a service provider—crucial to keeping the performance of HVAC equipment up.

<u>Demonstrating the Value</u>. Closely related to Customer Awareness is the ability to demonstrate value of improved HVAC equipment, installation, and maintenance. It is extremely difficult to sell improved efficiency to a customer if you can't tell them what it will save, and you can't tell them what it is savings while it is operating. Tracking HVAC energy use and benchmarking performance are key to providing the reassurance that something real and of value will be provided. This is as true for the industry as a whole as it is for individual customers: there are not sufficient data to state with certainty the value of a quality installation or quality maintenance. A study should be done to establish this.

<u>Larger Incentives are Needed</u>. It is currently a reality in the market that customers seldom consider life-cycle costs when making HVAC purchases, and in fact will not often choose a higher first-cost model that is more efficient. On the other hand, if an efficient option is equal or lower in cost, it is quite likely that a customer will make the more efficient choice. Incentive payments that cover only a part of the incremental cost for efficient equipment are not sufficient: incentives must cover the entire incremental cost, or more, to motivate buyers.

<u>The Gap between Technologists and Providers</u>. There is a gap between:

- technologists, who have developed excellent technologies and processes that are capable of saving a lot of energy if they were employed, but have little exposure to the realities of the field and little visibility to the market, and
- field providers, who will be the people, after all, who actually deliver the savings, and who understand their markets fairly well, but might not have the big picture, or the time to keep track of the latest and greatest technologies.

It is important to bridge this gap, because no one of the individuals groups, on its own, can provide the solutions needed. By collaborating and forming a team, we may have a shot at achieving the goals set by public policy.

With an original list of 35 volunteers and with the assistance of the WCEC, the Western HVAC Performance Alliance has formed into a viable organization, recognized within the industry. Its defined mission is as follows:

> *The Western HVAC Performance Alliance represents the heating, ventilation and air conditioning (HVAC) industry. It will partner with the government, utilities and the energy efficiency industry in order to maximize the many benefits of cooling, heating, indoor air quality, and energy efficiency services to consumers, minimize the use of gas and electricity via sustainable practice and programs, and benefit the individuals and organizations that ably deliver the above to consumers and society. Through this collaboration, the HVAC industry will be transformed, and will ensure that technology, equipment, installation and maintenance are of the highest quality to promote energy efficiency and peak load reduction.*

In a little over a year of existence, the WCEC has helped the Alliance to accomplish many of its objectives.

- An Interim Steering Committee was formed, with 26 voting members, and held an inaugural meeting three business days after the Roundtable. Since that time, the Steering Committee has held regularly scheduled monthly teleconferences with an average attendance of over 20. On the agenda for these meetings have been items related to structure and operation of the Alliance, as well as starting to tackle questions for design of programs such as compliance and quality maintenance.
- A Chartering Committee was formed to draft a charter for Steering Committee approval. This small group of 13 reflected the range of industry perspectives. This group defined the structure of the Alliance, and drafted a formal Charter for the organization. An interim Charter was adopted by the Steering Committee unanimously on October 14, 2009 and a full Charter was adopted unanimously on November 10, 2009.
- In order to gain some early successes, the Task Force decided to dive right in, and tackle one of the most obvious recommendations that came out of the Roundtable, that is, increasing the rate of compliance with Title 24. WCEC staff is chairing a Compliance Committee, which meets bi-weekly. The group consists of 29 members, just over half of whom attended the Roundtable. The additional members are primarily utility Codes &Standards representatives, additional local code officials, and enforcement program managers from the CEC and California State Licensing Board. The members of this committee are helping the IOUs to design and deliver trainings for contractors and code officials. The unique industry perspective they bring ensures the maximum effectiveness for these trainings. This group is helping the CEC to create streamlined compliance processes and effective enforcement mechanisms. We have already successfully worked with CEC to develop a simplified residential compliance form (the new “CF1R-ALT-HVAC” form) that will greatly aid contractors in complying with the code (and hopefully resulting in higher compliance rates),

and we are now starting to tackle the myriad commercial forms. We have conducted a survey of contractors, and are now planning a pilot test of online-permitting. We are also discussing compliance issues with the CSLB to ensure that these are as effective and fair as possible.

- Additional committees were formed on Quality Maintenance (one residential and one commercial), Quality Installation (residential), Marketing, and Administration.
- In June 2010 the utility program managers decided to reorganize the committees, maintaining the Compliance and Marketing Committees, but putting the other committees into a holding pattern until reorganization is complete.

In the upcoming three years, WCEC will continue to play a major role in supporting the Alliance.

Other Interactions with Industry

The WCEC had a number of additional industry support activities this year, including in-person meetings with many different companies, serving as reviewers for energy efficiency proposals and projects, and working proactively with California utilities. We met with a number of Venture Capitalists looking to invest in the HVAC space, including CalCEF, Vantage Point Ventures, New World Capital, Claremont Creek Ventures, Good Energies, and Polaris Ventures.

Mark Modera served as a reviewer for a study by the University of Michigan to model the optimal replacement intervals for residential air conditioners according to different criteria (energy use, carbon impacts, cost), and as a reviewer for numerous energy-efficiency proposals to the US Department of Energy (DOE) and the National Science Foundation (NSF). In addition, WCEC staff held numerous meetings with industrial partners to promote energy-efficient HVAC. Some of the partners in this list included: Chevron Energy Solutions (relative to the incorporation of PIER HVAC technologies into their portfolio of energy-efficiency measures), Walmart (including high-level strategy development for energy efficiency), Southern California Edison (numerous visits to discuss HVAC Performance Alliance and HVAC Technology Advocacy efforts, and serving on the panel at the Emerging Technologies Open Forum on Energy Efficiency Innovations), and Pacific Gas and Electric (monthly HVAC-program support meetings).

Over the course of the year, a constant stream of companies met with WCEC staff, looking for advice, or for assistance with getting wider acceptance of their efficiency technologies. Our visitors included large companies, such as Armstrong Industries (phase-change sheetrock), Microsoft, and Hewlett Packard, as well as small start-up companies, such as Evaporcool, and AC Research Labs. The WCEC held its third Affiliates Forum on March 31st.

Tool Development

The WCEC worked with industry and academics to identify tools that would be helpful in advancing the application of advanced cooling systems, and to coordinate the development of the necessary tools. The WCEC continued to invest in the development of simulation capabilities. Efforts included collaborating with LBNL to construct a model of hybrid evaporative/DX equipment.

The WCEC has been working with LBNL to investigate potential methodologies for creating an EnergyPlus module that represents the Coolerado hybrid evaporative/DX equipment. Some of the possibilities for modeling include simulation using first principles, modifying an existing Energy Plus model of an indirect evaporative cooler, and creating an empirical performance map. Creating a model from first principles is by far the most complicated and ambitious choice. Conversely, a lookup table based on empirical performance data appears to be the most straightforward approach, though the testing required to generate such a performance map is exhaustive and expensive. Lastly, modifying the module of an existing indirect evaporative cooler may provide useful results in a timely and cost effective manner, but this approach is difficult to validate as it does not represent the actual principles at work nor does it pull from actual data.

Codes & Standards

Over the course of the past year, the WCEC participated in the development of codes and standards, including California Title 24 and Title 20, to advance energy efficiency for cooling. Such Codes and Standards efforts are critical to the success of many emerging technologies, including the cooling technologies being investigated by the WCEC.

California Title 24 and Title 20 Energy Efficiency Standards

Participation in PG&E CASE Studies

WCEC has provided assistance to the PG&E CASE study process, working with the Heschong Mahone Group on the following T-24 CASE Studies:

Non Residential Radiant Cooling

This CASE will develop an alternative method for T-24 compliance for a radiant cooling system that takes into account occupant thermal comfort, and evaluate the cost effectiveness and energy savings for these systems. This will be done through modifications to the nonresidential ACM rules. Energy savings and thermal comfort will be established through a combination of building energy simulation using state-of-the-art building energy simulation tools, a review of published data on radiant system performance, and industry stakeholder participation. The WCEC will conduct a literature review and market assessment regarding system designs for radiant cooling systems.

Cool Ducts

Outdoor ducts experience significant unwanted heat gains during the cooling season. Current Title 24 duct insulation requirements reduce conductive heat transfer through the duct wall, but do not address radiative heat transfer from the duct to ambient and solar gain by the ducts. Unwanted duct heat gains can be minimized through the use of high-reflectivity, high-emissivity "cool duct" coatings, which have the potential to reduce the solar heat gains by ~80%. Cool ducts do however reduce beneficial wintertime solar heat gains, and high-emissivity surfaces increase radiative heat losses to cool surroundings and the night sky relative to low-emissivity surfaces. WCEC efforts in this area will include efforts to quantify the prevalence of exterior ductwork, as well as working on modeling the impacts of that ductwork.

To quantify the prevalence of exposed ductwork in California, a methodology was developed that uses County Assessor's parcel data to randomly select buildings for duct prevalence searches. 500 parcels will be selected from a particular Climate Zone to be surveyed. The surveys will be conducted in Google Earth or Bing maps to get a high resolution picture of each rooftop for measuring duct area, orientation of primary run, and color of duct. Assuming a normal distribution, 500 surveys will tell us the area of exposed ductwork across an entire Climate Zone with 95% confidence to within +-4.5%. To reduce the error to +-3% would require more than twice the amount of surveys and will not work within the budget. Two initial surveys of 100 parcels in Sacramento County found that on average 11% of the non-residential parcels had exposed ductwork (Note: a parcel normally contains one building). If the initial surveys are representative of the entire Climate Zone then we can say with 95% confidence that 6.5%-15.5% of parcels in Climate Zone 12 have exposed ductwork.

We are working with ParcelQuest to obtain the necessary Assessor data, and are currently waiting on the first data set for Climate Zone 12. A student will conduct most of the aerial searches and WCEC engineer Curtis Harrington will be working part-time on this effort.

Residential Zonal Air Conditioning

The goal of this project is to develop requirements that would increase the energy efficiency of zonal air conditioning systems and get those requirements adopted into the 2011 Residential Standard. Mark Modera developed a simplified model for these systems, and participated in field tests of zone conditioning systems. The clearest result from the field tests to date is that the delivered energy efficiency of two different systems was increased considerably when the bypass damper was closed.

DOE Regional AC Standards Advancement

Over the course of the past year, the WCEC has participated in the process of revising the Federal Appliance Efficiency standards for air conditioners. Although we did not achieve all that we would have liked within that process, the standards slated for approval do include regional considerations, including the designation of a hot, dry region. Recent discussions have focused on the issue of fan power in the standards, and a proposal to have an efficiency standard for air handlers.

Education and Outreach

UC Davis Cooling & Heating Efficiency Workgroup

WCEC has built a close connection with UC Davis Facilities & Operations and Architects & Engineers. WCEC staff participates in a monthly Cooling & Heating Efficiency workgroup meeting, which has established opportunities for multiple technology demonstration projects at the University.

City of Davis Sustainability

WCEC WCEC continues to collaborate with the City of Davis Sustainability Program to provide public education about energy efficient cooling for homes through ongoing public forum sustainability.

Commercial Building Energy Alliance

Mark Modera attended the Commercial Building Energy Alliance Supplier Summit at the end of the ASHRAE meeting in Orlando in January. WCEC met with PNNL at that meeting to discuss including Western Cooling Challenge analysis capabilities in the DOE RTU analysis tool for the Commercial Building Energy Alliance.

Outreach to US Department of Energy

Mark Modera devoted a considerable amount of time this year working with Lawrence Berkeley National Laboratory, UC Berkeley, Oak Ridge National Laboratory, and the National Renewable Energy Laboratory developing a joint Building Energy Efficiency Hub proposal. Dr. Modera is the lead for all Physical Systems research within the proposal, as well as the Principal Investigator for all of the research that would be performed at UC Davis.

The development of the Efficiency Hub proposal included the advancement of increasingly intimate working relationships with all of the other institutions. These relationships should prove quite valuable to the WCEC. One of the outcomes of these increased associations is that Paul Torcellini of NREL became a member of the WCEC Steering Committee. In addition, we found that ORNL has facilities capable of testing large Western Cooling Challenge entries, and that those facilities could be made available for that purpose. Subsequently, Jonathan Wooley made a trip to ORNL to tour their facilities and to provide input on their new user facility.

Outreach and Advising to Architects & Engineers

WCEC has maintained contact with and provided general advice to architects and engineers interested to include advanced cooling technologies in their building projects. Part of the outreach involved attending the Center for the Built Environment Advisory Board Meeting.

Outreach to Community Colleges

Although limited, WCEC has invited community college courses focused on sustainability and low energy building design to visit the Center and consider the recent advances and current research on cooling technologies. This included a WCEC lecture on cooling in hot/dry climates to a Sustainability Class from American River College, including a facilities tour.

Appendix A

Water-Use Efficiency for Alternative Cooling Technologies in Arid Climates

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Abstract

In arid climates, evaporative cooling technologies are generally valued for their reduced energy consumption in comparison to compressor-based air conditioning systems. However, two concerns that are often raised with respect to evaporative cooling equipment are their on-site water use and the impact of poor water quality on their performance. While compressor-based systems do not use water on-site, they do consume water through their use of electricity, which consumes water through evaporation at hydroelectric power plants and cooling at thermal power plants. This paper defines a water-use efficiency metric and a methodology for assessing the water use of various cooling technologies. The water-use efficiencies of several example cooling technologies are compared, including direct evaporative, indirect evaporative in two different configurations, compressor-based systems, compressor-based systems with evaporative pre-cooling of condenser inlet air, and hybrid systems that consist of an indirect evaporative module combined with a compressor-based module. Designing cooling systems for arid climates is entwined in the close relationship between water and energy and the scarcity of both resources. The analyses presented in this paper suggest that evaporative systems that significantly reduce peak electricity demand and annual energy consumption need not consume any more water than conventional systems.

Nomenclature

Variable	Description	Metric Unit
COP	Coefficient of performance	
C_p	Specific heat capacity of air at constant pressure	J/g·K
EER	Energy efficiency ratio	
H_{total}	Total sensible cooling	MJ_c
ΔH_{vap}	Heat of vaporization of water	MJ_c/l
$\dot{m}_{condenser}$	Mass flow rate across condenser	g/s
$\dot{m}_{supply-air}$	Mass flow rate of supply air	g/s
$m_{supply-air}$	Mass of supply air	g
n	Water-use efficiency	-
P	Fan Power	W
Q	Capacity required to pre-cool condenser air	W
T_{out}	Temperature of outside air	°C
T_{room}	Temperature of room air	°C
T_{supply}	Temperature of supply air	°C

$V_{on-site}$	Volume of water-use on-site for delivered cooling	l
w_e	Water-use rate for electricity generation	l/MJ_e
$w_{off-site}$	Water-use rate off-site per unit on-site cooling	l/MJ_c
$w_{on-site}$	Water-use rate on-site	l/MJ_c
w_{total}	Total water-use rate for cooling equipment	l/MJ_c

1.0 Introduction

Residential and commercial cooling are the top two contributors to peak electricity demand for many electric utilities in the US, particularly in the more-arid western states. In California, these two end uses comprise 30% of the summer peak electricity demand [1]. The vast majority of the systems used to provide this cooling are small compressor-based air conditioners. For example, the California Residential Appliance Survey of 2004 found that 94% of homes with air conditioning had compressor-based systems [2]. Only 6% of homes employed evaporation of water for cooling, despite the fact that the various evaporative systems have a large potential to reduce both the peak electricity demand and the energy use associated with both residential and light-commercial cooling.

Evaporative cooling is an alternative or augmentation to compressor-based air conditioning that utilizes the cooling potential of evaporating water to reduce electricity consumption. Because these systems consume water, when evaluating the energy savings potential of evaporative cooling systems, it is imperative to consider not just their impacts on electricity use, but also their impacts on water consumption as well. However, it is also necessary to consider the water use associated with the electricity consumed by these systems, and the higher electricity consumption associated with compressor-based cooling systems [3, 4]. The objectives of this paper are: 1) to explore the overall water-use impacts of various small-scale cooling systems, 2) to develop an appropriate metric for water-use efficiency and 3) to use that metric to compare, through simplified models, compressor-based air conditioning and various evaporative technologies that are applicable to arid and semi-arid climates.

1.1 Defining Water-Use Efficiency

In order to compare water consumption for different cooling alternatives, it is first necessary to define a common yardstick for measuring and normalizing that consumption. The chosen metric for this paper is liters of water consumed per megajoule of indoor cooling capacity delivered, including both on-site water consumption and the off-site water consumption associated with on-site electricity use. In evaluating the total water use of cooling equipment, it important to recognize that there is water consumption associated with the off-site electricity generation and transmission required to power the fans and compressors used for residential and commercial cooling, and that that off-site water consumption is strongly dependent on the means by which the electricity was generated [3, 4].

1.1.1 Off-Site Water Consumption for Electricity Generation

Two sources that analyzed the water consumption associated with electricity production in the Southwest United States were identified. The first source, a 2003 report by National Energy Renewable Laboratory (NREL), separately analyzed water consumption for thermoelectric power generation and for hydroelectric power generation, the two main types of electricity generation [3]. The water consumption for thermoelectric power generation was based on water withdrawal data from the United States Geological Survey (USGS) and a coefficient of water loss by evaporation approximated by the power plant cooling design. The water consumption for hydroelectric power generation was based on the free-water-surface evaporation map reported by the National Weather Service. Evaporation rates for 120 of the largest damns in the United States were analyzed. The analysis also takes into account 5%

generation losses for thermoelectric plants and 9% transmission and distribution losses for all plant types. The thermoelectric and hydroelectric water consumption rates were then applied to recent electricity generation mix data for 2007 from the Energy Information Administration (EIA) [5]. It is assumed that solar and wind power sources do not consume fresh water. The results calculated from the NREL study are summarized for Arizona, California, and New Mexico (Table 1). The weighted average water consumption result is different than reported in the NREL study, which used 1999 EIA data for the electricity generation mix. Between 1999 and 2007, the percentage of electricity generated by hydroelectric power has decreased from 12% to 6% in Arizona, 21% to 13% for California, and remained flat at 1% for New Mexico.

The University of California Santa Barbara (UCSB) provided a second source for data specifically on California (Row 3 in Table 1) [4]. The thermoelectric power consumption reported by the UCSB study excludes nuclear power, which consumes sea water and not fresh water in California. The main difference from the NREL study is that the UCSB study referenced a report by the Pacific Institute for Studies in Development that analyzed annual evaporative losses from 100 California hydroelectric facilities [6]. This should be a more accurate assessment for California because the analysis includes 100 hydroelectric facilities in California compared to 120 nationwide in the NREL study (the fraction of the 120 dams located in California is not stated).

Table 1 – Weighted average water consumption for electricity generation in the southwestern United States

	Thermoelectric Water Consumption	Hydroelectric Water Consumption	2007 Electricity Generation Mix	Weighted Average Water Consumption (w_e)
Arizona [3, 5]	0.34 l/MJ_e	68.2 l/MJ_e	94% thermo, 6% hydro	4.4 l/MJ_e
California [3, 5]	0.05 l/MJ_e	21.9 l/MJ_e	84% thermo, 13% hydro, 3% wind and solar	2.9 l/MJ_e
California [4, 5]	0.46 l/MJ_e	7.9 l/MJ_e	67% thermo, 17% nuclear, 13% hydro, 3% wind and solar	1.4 l/MJ_e
New Mexico [3, 5]	0.66 l/MJ_e	98.8 l/MJ_e	95% thermo, 1% hydro, 4% wind	1.4 l/MJ_e

The water consumption for electricity generation differs significantly by state, with water consumption in Arizona being three times greater than that in New Mexico, two adjacent states in the arid southwestern United States. This result is driven by hydroelectric water consumption due to evaporation. Accurately quantifying this evaporation is crucial to the result, as shown by the two separate analyses for California, which yield results that differ by a factor of two. Authors of both sources agree that the water consumption for hydroelectric electricity generation is difficult to quantify and that the result may be inflated, as dams provide benefits other than electricity generation, such as flood control and recreation. In the two studies described, all evaporation is attributed to electricity generation. In evaluating cooling technologies, total water use will be calculated using both the low end and high end water consumption estimates for electricity generation in the southwestern United States.

The off-site water consumption per unit of cooling for both compressor-based air conditioning and evaporative cooling can be calculated from the efficiency of the cooling equipment, in units of either coefficient of performance (COP) or energy efficiency ratio (EER), combined with the water consumption for electricity generation (w_e) (Equation [1]).

$$w_{off-site} = w_e/COP \text{ (metric units)} \qquad [1]$$

1.1.2 On-Site Water Consumption

In order to calculate water-use efficiency, the sensible cooling delivered for the water evaporated needs to be defined. One of the trickiest parts of these calculations is the choice of an appropriate cooling metric for evaporative cooling equipment so that the result can be directly compared to compressor-based systems. The relevant difference between evaporative systems and compressor-based systems is that all evaporative systems are required to use at least some outdoor air to provide cooling, while compressor systems can run on recirculation only. Because evaporative systems use significant amounts of outdoor air, they can over-ventilate the space. The result is that an evaporative cooler system may have to provide more total cooling (to cool excess ventilation air) as compared to a compressor-based system meeting the same indoor load. In order to compare the two side by side, the equation for evaporative cooling should take credit for the temperature difference between supply air and indoor air, but should only take credit for cooling outdoor air to indoor temperature for the required building ventilation, and not any excess ventilation. For the purposes of this paper, the ventilation required is expressed as the ratio of ventilation air to the total supply air, r_{vent}. The results are calculated for $r_{vent} = 1/3$, but this variable can be changed to generate results for any ratio of ventilation. In this case, the delivered sensible cooling for evaporative equipment is calculated from Equation [2] and the efficiency for evaporative equipment, required for Equation [1], is obtained using a similar methodology (Equation [3]).

$$H_{total} = m_{supply-air} \times C_p \times \left[r_{vent}\left[T_{out} - T_{supply}\right] + [1 - r_{vent}] \times \left[T_{room} - T_{supply}\right]\right] \qquad [2]$$

$$COP = \dot{m}_{supply-air} \times C_p \times \left[r_{vent}\left[T_{out} - T_{supply}\right] + [1 - r_{vent}] \times \left[T_{room} - T_{supply}\right]\right]/P \qquad [3]$$

The on-site water-use rate is the volume of water evaporated on-site divided by the sensible cooling delivered (Equation [4]).

$$w_{on-site} = \frac{V_{on-site}}{H_{total}} \qquad [4]$$

The total water-use rate for evaporative equipment is the sum of the off-site water-use rate from electricity generation and the on-site water-use rate (Equation [5]).

$$w_{total} = w_{off-site} + w_{on-site} \qquad [5]$$

The water-use efficiency, n, is defined as the actual sensible cooling delivered, divided by the maximum cooling that can be obtained from evaporating a given mass of water (equal to the heat of vaporization, ΔH_{vap}) (Equation [6]).

$$n = \frac{1/w_{total}}{\Delta H_{vap}} \qquad [6]$$

For direct evaporative systems, using Equation [2] as the yardstick for delivered cooling, water-use efficiency is always below one. Because outdoor air is required to produce the cooling, at least some of the heat of vaporization goes into cooling the outdoor air (over and above that required for ventilation) from outdoor temperature to room temperature. Moreover, the fans in these systems all consume electricity, which adds off-site water consumption. Conventional indirect evaporative systems, which utilize only outdoor air, are also limited to efficiencies less than one when utilizing Equation [2]. However some indirect evaporative systems use return indoor air on one or both sides of the heat exchanger, which allows the equipment to re-capture a portion of the temperature difference between indoor and outdoor air. This distinguishes these systems from direct evaporative systems, for which indoor air cannot be re-circulated, as they introduce humidity as well as sensible cooling to the indoor air. For compressor-based air conditioning systems, the water-use efficiency can be greater than one and is based solely on the water consumption needed to generate electricity. An extreme example would be when a solar photovoltaic system, which consumes no water[1], powers a compressor-based air conditioner. In this case, the water-use efficiency would be infinite, regardless of the energy efficiency of the compressor system.

2.0 Methodology and Results

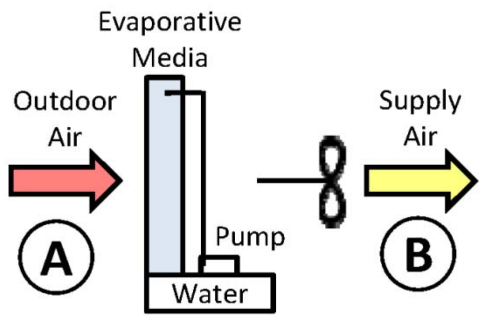
The cooling methods evaluated in this paper for water-use efficiency are:

1. direct evaporative cooling for supply air;
2. indirect evaporative cooling for supply air;
3. compressor-based air conditioning;
4. direct evaporative pre-cooling of condenser air for compressor-based air conditioning; and
5. hybrid systems that combine indirect evaporative and compressor-based systems.

For numerical analyses, all cooling approaches are evaluated using outdoor air with 37.8°C dry bulb (DB)/ 20.7°C wet bulb (WB) temperatures, and indoor air with 25.6°C DB/17.8°C WB temperatures.

2.1 Direct evaporative cooling

In direct evaporative cooling equipment, a fan pulls outdoor air through a wet media, which is generally a corrugated cellulose-based structure that distributes the water to the air (



[1] Although some water was most certainly used in the process of manufacturing the system, a photovoltaic system does not consume any water on an ongoing basis, which is the metric being used for this paper (i.e. we are not calculating “embedded-water”).

Figure **1**), and blows that cooler, wetter air into the conditioned space. The performance of these systems is generally quantified by the evaporative effectiveness (Equation [7]).

[7]

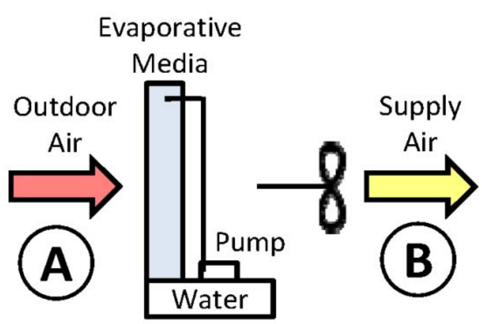
Referring to

Figure **2**, as water is evaporated, the air originally in state (A) increases in humidity and decreases in dry-bulb temperature, while the wet-bulb temperature remains constant, until point (B) is reached. The water-use efficiency was analyzed for a commercially available direct evaporative cooler that had been independently tested by a laboratory [7] with the following results:

1. Intake Air: 37.7°C DB/21.1°C WB, Airflow = 966 l/s
2. Supply Air: 22.9°C DB/21.2°C WB, Airflow = 927 l/s
3. Wet Bulb Evaporative Effectiveness = 89%
4. External Static Pressure = 75 Pa, Fan Power = 321 W
5. COP (=1/3) = 23.3

The inputs for water-use efficiency (Equation [5]) are determined by 1) calculating the sensible cooling delivered (Equation [2]) using the supply air temperature from the test results, 2) calculating the on-site water use from the humidity ratio increase of the supply air stream using the psychometric chart (

Figure **2**), and 3) calculating the off-site water use from the calculated COP (Equation [3]). The resulting water-use efficiency of the tested indirect evaporative cooler is 0.37-0.42 (range attributed to water consumption for electricity generation range of 1.4-4.4 l/MJ_e). Additional water for maintenance of the evaporative media is not included[2]. While direct evaporative cooling systems may be suitable for environments where sensible cooling is required and higher humidity is desirable (e.g. wineries, agriculture), it is often considered a lower-performance solution for residential and commercial environments (these systems do not reduce the enthalpy of the supply air, but rather trade off sensible



enthalpy for latent enthalpy).

[2] Note that these calculations do not include water used to maintain the usable lifetime of the evaporative media. In regions with good water quality, this water use can be modest (~5%), however in regions with hard water, maintenance water use can increase on-site water use by 50%.

Figure 1 – Schematic: Direct evaporative cooling of supply air

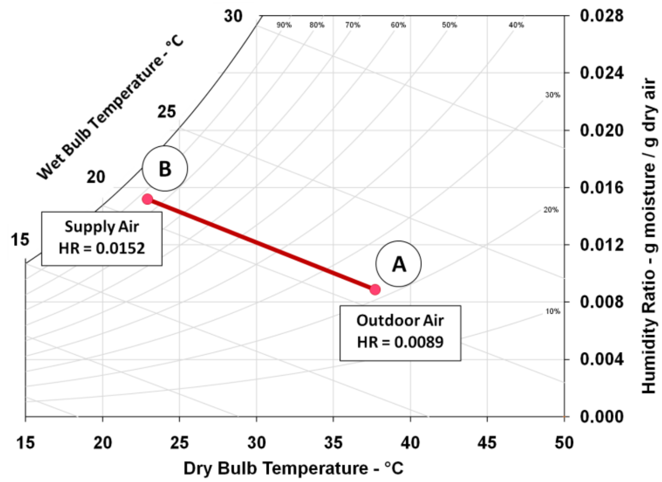


Figure 2 – Psychometric Chart: Direct evaporative cooling of supply air

2.2 Indirect evaporative cooling

Indirect evaporative cooling utilizes a heat exchanger in which one side is a wet-air passage and the other side is a dry-air passage (Figure 3). In the conventional configuration, all of the air entering the equipment is outdoor air. The outdoor air enters the dry side of the heat exchanger at (A), exchanges heat with the wet side air, and exits the heat exchanger at (B) without changing its humidity (Figure 4). A portion of the air at (B) is delivered to the building as the supply air, while the rest of the air is directed through the wet-side of the heat exchanger and exits at (C). On the wet-side, the air stream increases in enthalpy as it absorbs heat from the dry-side air and evaporates water from the water supply. As for direct evaporative coolers, the metric typically used to quantify the performance of the indirect evaporative heat exchanger is its evaporative effectiveness (Equation [7]).

An evaporative effectiveness greater than one is achievable in an indirect evaporative unit. The physical limitation for the supply air temperature is the dew-point temperature of the incoming outdoor air. Increasing heat exchanger surface area increases effectiveness, however this generally results in additional fan power, and/or increased size and materials requirements. Reducing air flow rates increases effectiveness, but reduces capacity. Water-use efficiency was analyzed for a commercially available indirect evaporative cooler that had been independently tested by a laboratory [8] with the following results:

1. Intake Air: 37.8°C DB/20.7°C WB, Airflow = 1,307 l/s
2. Supply Air: 22.4°C DB/15.6°C WB, Airflow = 703 l/s
3. Exhaust Air: 25.7°C DB/25.6°C WB, Airflow = 604 l/s
4. Wet Bulb Evaporative Effectiveness = 90%
5. External static pressure = 75 Pa, Fan Power = 1,260 W
6. COP (=1/3) = 4.9

The inputs for the water-use efficiency (Equation [5]) are determined by 1) calculating the sensible cooling delivered (Equation [2]) using the supply air temperature from the test results, 2) calculating the on-site water use from the humidity ratio increase of the exhaust air stream using the psychometric chart (Figure 4), and 3) calculating the off-site water use from the calculated COP (Equation [3]). The resulting water-use efficiency of the tested indirect evaporative cooler is 0.17-0.23 (range attributed to water consumption for electricity generation range of (1.4-4.4 l/MJ$_e$)). The on-site water-use calculation includes all water evaporated for cooling, of which ~95% cools the supply air stream and ~5% removes the additional heat generated by the fan. Additional water for maintenance of the evaporative media is not included.

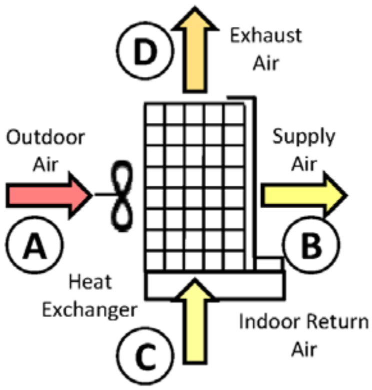


Figure 3 – Schematic: Indirect evaporative cooling using only outdoor air for intake air

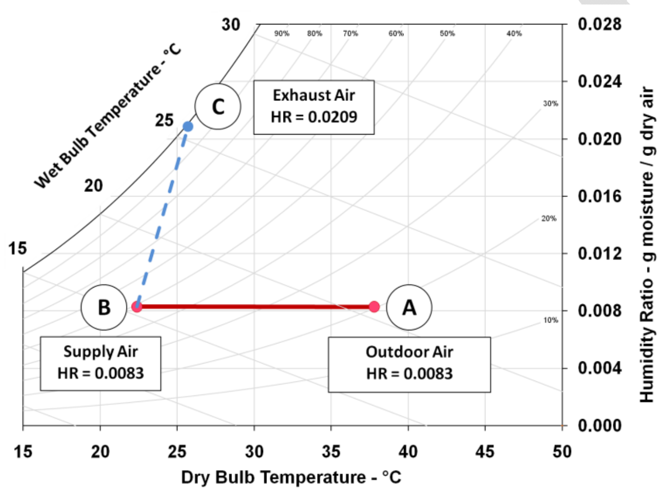


Figure 4 – Psychometric Chart: Indirect evaporative cooling using only outdoor air for intake air

2.3 Indirect evaporative cooling using exhaust air

In order to reduce unnecessary pressurization (above that needed to eliminate infiltration), indoor air can be returned and mixed with outdoor air at the entry to the heat exchanger (Figure 5). The other advantage of this configuration is that it incorporates the ability to capture the cooling embodied in the indoor air (i.e. if there weren't any water evaporation, it would act like an air-to-air heat exchanger). This strategy was recently employed by a manufacturer in designing their hybrid indirect/vapor-compression rooftop unit for the Western Cooling Challenge initiated by the Western Cooling Efficiency Center [9].

In order to investigate the impact of utilizing indoor air on water-use efficiency, the indirect evaporative cooler (without any use of compressor-based cooling) is analyzed again assuming that the outdoor air intake flow exceeds the exhaust air flow by 127 l/s to avoid any infiltration load, and that the return air from the building is mixed with the outdoor intake air to create a mixed-air stream that is used in both the wet and dry passages of the cooler:

1. Outdoor Air: 37.8°C DB/20.7°C WB, Airflow = 746 l/s

2. Return Indoor Air: 25.6°C DB/17.8°C WB, Airflow = 585 l/s
3. Mixed Intake Air: 32.4°C DB/19.4°C WB, Airflow = 1,331 l/s

In order to assess the performance and water-use efficiency implications associated with utilizing mixed outdoor and indoor air, the laboratory test results [8] for the closest available intake-air test point are used: 32.3°C DB/18.1°C WB. The results for this test point were:

1. Intake Air: 32.3°C DB/18.1°C WB, Airflow = 1,331 l/s
2. Supply Air: 20.4°C DB/13.7°C WB, Airflow = 713 l/s
3. Exhaust Air: 23.1°C DB/22.6°C WB, Airflow = 618 l/s
4. Wet Bulb Evaporative Effectiveness = 83%
5. External static pressure = 75 Pa, Fan Power = 1,357 W
6. COP (=1/3) = 5.9

The inputs for the water-use efficiency (Equation [5]) are determined by 1) calculating the sensible cooling delivered (Equation [2]) using the supply air temperature from the test results and the actual outdoor air temperature (37.8°C), 2) calculating the on-site water use from the humidity ratio increase of the exhaust air stream using the psychometric chart (Figure 6), and 3) calculating the off-site water use from the calculated COP (Equation [3]). The resulting water-use efficiency of the tested indirect evaporative cooler is 0.25-0.35 (range attributed to water consumption for electricity generation range of 1.4-4.4 l/MJ_e). The on-site water-use calculation includes all water evaporated for cooling, of which ~91% cools the supply air stream and ~9% removes the additional heat generated by the fan. Additional water for maintenance of the evaporative media is not included.

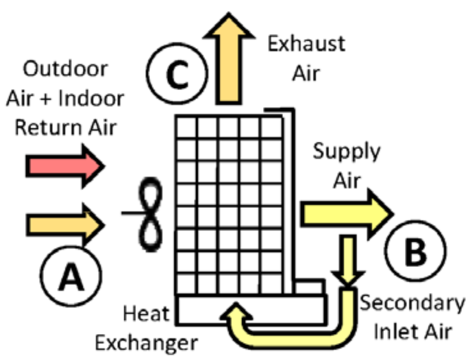


Figure 5 – Schematic: Indirect evaporative cooling using mixed outdoor air and building return air as intake air

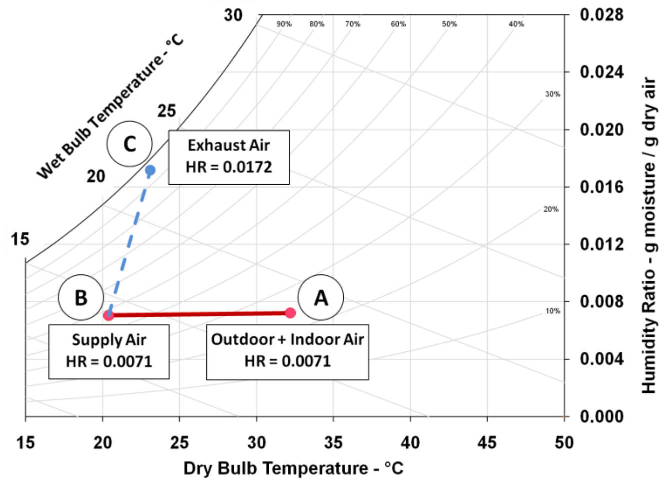


Figure 6 – Psychometric Chart: Indirect evaporative cooling using mixed outdoor air and building return air as intake air

2.4 Compressor-based air conditioning

Compressor-based air conditioning systems do not consume any water on-site. However, the electricity needed to run these systems is generated at power plants that consume water. The efficiency of the equipment can be used to calculate off-site water use. For this analysis, the efficiency is calculated as the total cooling capacity delivered when the outdoor air temperature is 37.8°C divided by the total electricity consumption for compressors, condenser fans, and the blower motor at an external static pressure of 125 Pa. While a wide variety of compressor-based systems are available, two “representative” systems were selected for this analysis. The first is a “standard-efficiency” unit manufactured by York (model DM240) with refrigerant R-22, a refrigerant that has been utilized in the United States for several decades. The second is a higher-efficiency unit manufactured by Lennox (model Strategos SGB288H4M) with refrigerant R-410A, which is replacing R-22 as it is phased out of new equipment in the United States by January 1st, 2010 to meet environmental standards[10]. While the choice of refrigerant is not the reason for increased efficiency, new high efficiency designs developed in recent years used R-410A early on in anticipation of the phase out of R-22.

In arid climates, compressor-based systems often produce unnecessary dehumidification, and therefore a better yardstick is the sensible cooling delivered for these regions. Manufacturer’s literature for both systems provides detailed test data to calculate operating efficiency for a matrix of specific outdoor temperatures and indoor temperature/humidity combinations. For the units described, delivered sensible cooling for indoor air conditions of 25.6°C DB/17.8°C WB is approximately 82% of the total cooling for the R-22 system and 80% of total cooling for the higher efficiency R-410A system.

A 70 $kW_{cooling}$ commercial compressor-based air conditioning system utilizing refrigerant R-22 with a sensible COP of 1.6 (calculated for the York DM240 at 37.8°C outdoor air temperature)[11] yields a water-use efficiency of 0.16-0.50 (range attributed to water consumption for electricity generation range of 1.4-4.4 l/MJ_e). Similarly, a 70 $kW_{cooling}$ higher-efficiency system utilizing refrigerant R-410A with a sensible COP of 2.7 (calculated for the Strategos SGB288H4M at 37.8°C outdoor air temperature)[12] yields a water-use efficiency of 0.26-0.85 (range attributed to water consumption for electricity generation range of 1.4-4.4 l/MJ_e)).

2.5 Compressor-based air conditioning with evaporative pre-cooling of condenser air

Another type of evaporative cooling uses direct evaporative cooling to decrease the temperature of the outdoor air delivered to the condenser coil of a compressor-based air conditioning system (Figure 7). The direct evaporative pre-cooler can be applied to any condenser, and the units described in section 2.4 are considered as examples. The first is a standard efficiency unit manufactured by York (model DM240) [11] utilizing refrigerant R-22. The second is a high efficiency unit manufactured by Lennox (model Strategos SGB288H4M) [12] utilizing refrigerant R-410A. Manufacturer's literature for both systems provides detailed test data to calculate operating efficiency for a matrix of specific outdoor temperatures and indoor temperature/humidity combinations. The sensible cooling efficiency of the York system improves approximately 0.034 COP per °C of pre-cooling provided, and the efficiency of the Lennox system improves approximately 0.067 COP per °C of pre-cooling provided (Figure 8). The steeper slope of the Lennox system is related to the performance of R-410A refrigerant, as it is more sensitive to outdoor air temperature compared to R-22. The reason for the difference is that R-410A has a critical temperature of 70°C while R-22 has a critical temperature of 96°C [13]. The higher critical temperature of R-22 results in a lower relative efficiency drop as outdoor air temperature increases. Therefore, condenser air pre-cooling provides a greater benefit for R-410A systems than for R-22 systems.

The energy required to pre-cool the condenser air for either system is a function of the mass flow rate of air across the condenser and the specific heat capacity of air (Equation [8]).

[8]

A direct evaporative pre-cooler with an evaporative effectiveness of 80% at outdoor conditions 37.8°C DB/20.7°C WB pre-cools the air to 23.9°C. For either example compressor-based system moving 8,490 g/s across the condenser coil, 117 $kW_{cooling}$ is needed to pre-cool the condenser air by 14°C.

With pre-cooling delivering 23.9°C air to the condenser, the York system delivers a sensible capacity of 57 $kW_{cooling}$ at a sensible COP of 2.3. Assuming all water used is evaporated, the York system consumes 0.84 liters of water on-site per megajoule sensible cooling delivered by the evaporator coil. The off-site water consumption is 0.59-1.92 $l/MJ_{cooling}$. The range is attributed to water consumption for electricity generation range of 1.4-4.4 l/MJ_e. Combining on-site and off-site water use yields a water-use efficiency metric of 0.15-0.29.

Similarly, with pre-cooling delivering 24°C air to the condenser, the Lennox system delivers a sensible capacity of 60 $kW_{cooling}$ at a sensible COP of 3.7. Assuming all water used is evaporated, the York system consumes 0.80 liters of water on-site per megajoule sensible cooling delivered by the evaporator coil. The off-site water consumption is 0.36-1.18 $l/MJ_{cooling}$. The range is attributed to water consumption for electricity generation range of 1.4-4.4 l/MJ_e. Combining on-site and off-site water use yields a water-use efficiency metric of 0.21-0.35.

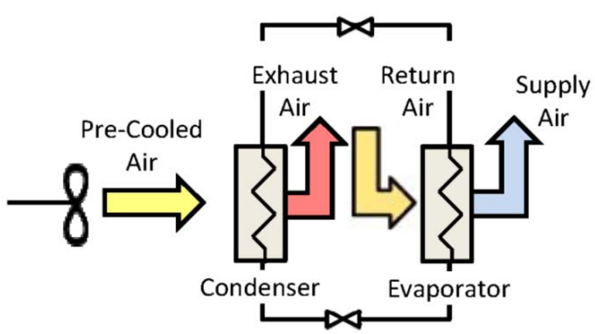


Figure 7 – Pre-cooled air to condenser of a compressor-based system.

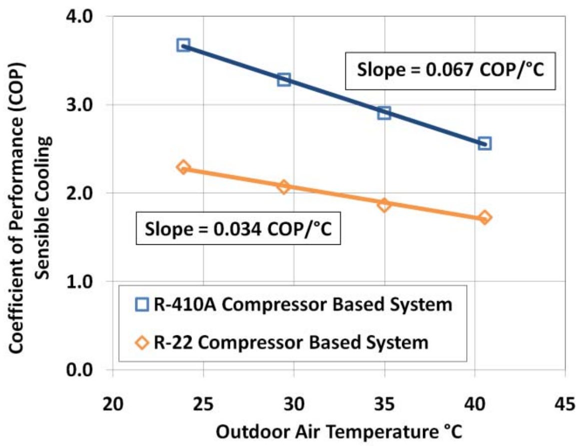


Figure 8 - Relationship between condenser air temperature and efficiency for R-22 and higher efficiency R-410A compressor based systems [11, 12]

2.6 Hybrid vapor-compression/indirect-evaporative systems

Although it can prove to be a cost-effective retrofit to improve the efficiency and capacity of existing compressor-based air conditioners, evaporative pre-cooling of condenser air is not the most energy and water efficient design possible. An option with greater energy and water-use efficiency than direct pre-cooling is a multiple component system where the wet-side exhaust of an indirect evaporative system (Figure 3 and Figure 5, flow C) is used as the intake air flow for the condenser (Figure 7). The exhaust air from an indirect evaporative system generally has a dry bulb temperature that is cooler than outdoor air temperature because it is impractical to capture all the cooling available from the exhaust air stream in the indirect heat exchanger (doing so would require an impractically large heat exchanger or an impractically slow air flow rate). The cool, wet exhaust can thus be used to improve the efficiency of the compressor-based air conditioner. To calculate the water-use efficiency for this configuration, the additional cooling provided by the exhaust air is added to the cooling provided by the indirect evaporative unit. The major benefit of this hybrid system is that the cooler wet-side air being exhausted from the indirect heat exchanger is being re-used for condenser pre-cooling instead of being wasted. The specific efficiency improvement obtainable for the hybrid system described depends on the design of the indirect evaporative system, the water used off-site for electricity generation, and the refrigerant used in the compressor-based system (Table 2). In addition, the sizing of the compressor-based system relative to the indirect evaporative system is important. If the exhaust air flow from the indirect module is larger than the flow rate across the condenser of the compressor-based system, then some of the cooled exhaust air is wasted. When the exhaust air flow matches the condenser air flow the largest gain in energy efficiency is achieved. When the exhaust air flow is less than the condenser air flow, the energy efficiency and water-use efficiency are a function of the exhaust air fraction.

For this analysis, the exhaust air flow of the indirect module is combined with a condenser of matching air flow. The indirect module used in this analysis is the same as that analyzed in the previous section where the exhaust air is 12-15°C cooler than the outdoor air temperature. For a correctly-sized compressor-based system of the same efficiency as that described in sections 2.4 and 2.5, the improvement in water-use efficiency over the standalone indirect module can be significant, with an improvement of up to 65% (Table 2). This increase in water-use efficiency for the hybrid system is greatest for the higher efficiency system using refrigerant R-410A, for which the performance is more strongly related to condenser air temperature. Also, because they are more energy efficient, the R-410A systems use less water, which increases the overall system efficiency. Combining the indirect module with the standard efficiency R-22 system actually decreases water-use efficiency in the case where water needs for electricity generation are high. This is because the water-use efficiency of the compressor system at the pre-cooled outdoor air temperature is lower than the indirect evaporative system. It should be noted that there exist a multitude of alternative hybrid system designs other than the one analyzed in this paper, and there may exist designs that have even higher water-use efficiencies.

Table 2 - Water-use efficiency improvement of using the exhaust of an indirect evaporative system for condenser-air pre-cooling

	Water-use Efficiency		
Indirect system and water-use rate for electricity generation	Indirect Only	+Indirect Exhaust to R-22 Compressor System	+Indirect Exhaust to R-410A Compressor System
Outdoor Air Only w_e= 1.4 l/MJ_e	0.23	0.34 (+48%)	0.37 (+61%)
Outdoor Air Only w_e= 4.46 l/MJ_e	0.17	0.19 (+12%)	0.28 (+65%)
Outdoor Air + Return Indoor Air w_e= 1.4 l/MJ_e	0.35	0.45 (+29%)	0.50 (+43%)
Outdoor Air + Return Indoor Air w_e= 4.4 l/MJ_e	0.25	0.24 (-04%)	0.36 (+44%)

3.0 Discussion

The water-use efficiencies for all types of cooling equipment analyzed are summarized for side-by-side comparison, sorted by energy efficiency (Figure 9). This comparison indicates that direct evaporative cooling has a competitive water-use efficiency and the highest energy efficiency. However, the value of this efficiency is offset by the fact that the applicability of this technology is limited due to the elevated indoor humidity that it produces. Evaporative pre-cooling of condenser air has a water-use efficiency of 0.15-0.35. This strategy is shown to be more advantageous for R-410A systems than for R-22 systems as R-410A performance is more strongly related to condenser air temperature. Even though pre-cooling condenser air is not the most efficient water-use option, it has several advantages, namely that it is a relatively inexpensive retrofit that provides significant electricity savings and increased cooling capacity without impacting indoor humidity levels.

The most interesting results in Figure 9 involve indirect evaporative cooling, as it does not add any indoor moisture, yet its water-use efficiency can be elevated above that of direct evaporative cooling by appropriate equipment design. For the options that were analyzed, the water-use efficiency of indirect evaporative cooling is maximized when indoor return air is incorporated into the intake air stream and the indirect-section exhaust is used as pre-cooled condenser inlet air for a properly-sized R-410A compressor-based system. When water requirements for electricity generation are high, 4.4 l/MJ_e, an indirect evaporative cooler that recycles indoor air and recovers exhaust for a R-410A compressor-based system has a water-use efficiency of n=0.36, which is actually more water efficient than the standard efficiency R-22 compressor-based system (n=0.16) and the high efficiency R-410A compressor-based system (η=0.26). When water requirements for electricity generation are low, 1.4 l/MJ_e, the high efficiency R-410A system has a water-use efficiency nearly two times greater than the most water-efficient evaporative system analyzed. In either case, the evaporative system significantly reduces peak electricity demand and annual energy consumption. It is clear that pinpointing the quantitative water-use efficiency results relies very heavily on the water use for hydroelectric electricity generation, which varies by state. Without knowing the "exact" answer, it is clear that evaporative technologies that are superior from an energy efficiency standpoint can be competitive from a water-use efficiency standpoint as well.

The comparative results shown assume that the required ventilation for the building is 33% of the total supply air. However, the direct and indirect systems with no recovery of indoor air actually provide 100% ventilation air. The indirect system with recovery of indoor air analyzed provides a supply air stream that is 56% ventilation air. The sensible cooling provided by the evaporative cooler (Equation [2]) only receives credit for the required 33% ventilation. If the required ventilation is higher, the water-use efficiencies for the evaporative cooling systems will increase. If the required ventilation is lower, the water-use efficiencies for the evaporative cooling systems will decrease. The water-use efficiency for compressor-based systems is not a function of ventilation rate. Therefore, evaporative systems are even more attractive for buildings with high ventilation requirements, or in buildings with multiple cooling units, where particular units can be dedicated to providing additional ventilation.

Another issue that will need to be addressed for evaporative coolers in some regions is the effect of hard water on the maintenance of the system. Hard water can cause mineral buildup on wet-side heat exchange surfaces. Initial experiments indicate that the mineral build up does not appear to reduce evaporative effectiveness, but that it does increase flow resistance and, therefore, capacity and efficiency. Typically, manufacturers drain sump water and/or use extra water to wash the media on a regular basis to prevent and remove mineral buildup. Use of extra maintenance water is not considered in the analysis and will reduce water-use efficiency. The amount of maintenance water required based on mineral content is not well understood, but "rule of thumbs" are in the range of 5-50% of the evaporated water. Potential options to reduce maintenance water consumption include pre-treating the water supply to remove minerals or changing evaporative media on a periodic basis when mineral buildup reaches an unacceptable level.

One other consideration when comparing the water-use efficiencies of various cooling-equipment alternatives is the difference between localized water use and power-plant water use. On the negative side for evaporative equipment is the fact that it takes energy to transport water to the local cooling equipment. On the positive side for evaporative equipment is that there are localized water sources that are suitable for evaporative cooling purposes, but which currently cannot be used for drinking. These include air-conditioner condensate, captured rainwater, and potentially grey water. Along a similar line, purge water from evaporative coolers can potentially be used for gardens, thereby eliminating maintenance-water use from the equation. Finally, the reader is cautioned that the results in Figure 9 were obtained based upon a sample of convenience. The performance data for particular equipment types (e.g. indirect evaporative systems) was not based upon the most current designs, but rather on equipment for which published laboratory data was conveniently available. That said, the authors are not aware of any current developments that will alter the results in Figure 9 dramatically.

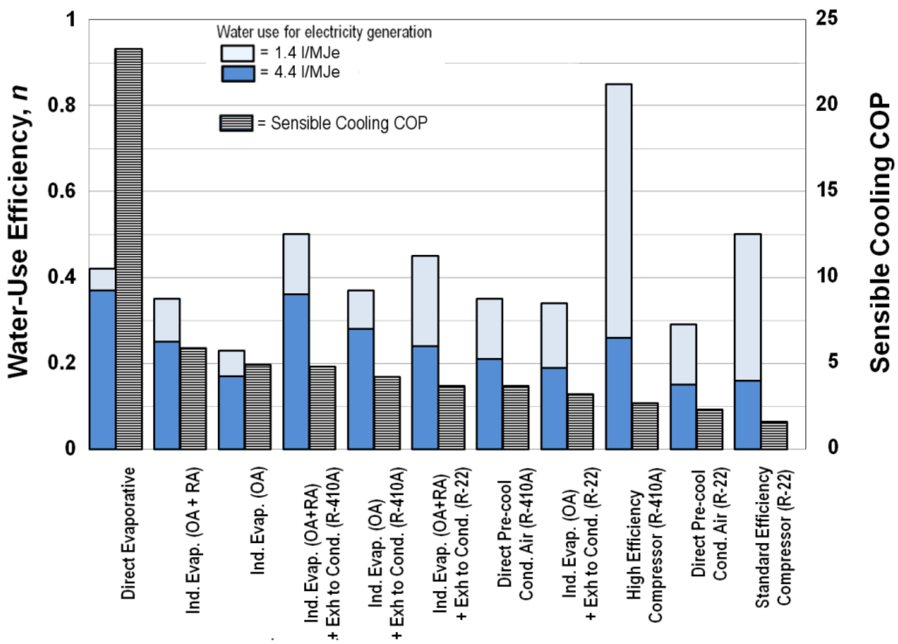


Figure 9 – Summary of water-use efficiencies for various cooling technologies (ranges are based upon high and low values for water consumption for electricity generation in the southwestern United States).

4.0 Conclusions

Designing cooling systems for California's climate is entwined in the close relationship between water and energy and the relative scarcity of both resources on both peak and annual bases. It is clear that a rational basis is needed for comparing cooling-system alternatives. This paper has presented a possible framework for such comparisons, as well as example applications of that framework to a number of cooling alternatives.

The work presented in this paper suggests that there exist viable alternatives for reducing energy consumption and peak electricity demand that do not significantly increase overall water use. One such solution may be in designing hybrid evaporative-plus-compressor systems that significantly reduce peak and annual electricity demand while making efficient use of on-site water. This paper also suggests that additional research on water quality impacts and local water management strategies could prove to be valuable.

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Appendix B

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Swimming pools as heat sinks for air conditioners: Model design and experimental validation for natural thermal behavior of the pool

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ABSTRACT

Swimming pools as thermal sinks for air conditioners could save approximately 40% on peak cooling power and 30% of overall cooling energy, compared to standard residential air conditioning. Heat dissipation from pools in semi-arid climates with large diurnal temperature shifts is such that pool heating and space cooling may occur concurrently; in which case heat rejected from cooling equipment could directly displace pool heating energy, while also improving space cooling efficiency. The performance of such a system relies on the natural temperature regulation of swimming pools governed by evaporative and convective heat exchange with the air, radiative heat exchange with the sky, and conductive heat exchange with the ground. This paper describes and validates a model that uses meteorological data to accurately predict the hourly temperature of a swimming pool to within 1.1 °C maximum error over the period of observation. A thorough review of literature guided our choice of the most appropriate set of equations to describe the natural mass and energy exchange between a swimming pool and the environment. Monitoring of a pool in Davis, CA, was used to confirm the resulting simulations. Comparison of predicted and observed pool temperature for all hours over a 56 day experimental period shows an R-squared relatedness of 0.967.

1. Introduction

In California, where all the large electric utilities experience their peak power demand in the summer, space cooling accounts for 29% of the total peak power demand and approximately 40% of the residential peak demand [1]. This occurs in part because the COP for traditional air-cooled vapor-compression cooling equipment diminishes significantly at high outdoor temperatures, such that equipment efficiency can be at its worst when cooling demand is greatest. Thermodynamics for heat pumps dictates that the work required to transfer heat from a cooler source to a warmer sink increases with the temperature difference between the two. In practice, for a vapor-compression system, since heat exchange with the refrigerant at the condenser and evaporator is driven by the temperature differences between the refrigerant and the sink and source respectively, the overall temperature difference experienced by the refrigerant is significantly larger than the temperature difference between the sink and source. For this reason, a large fraction of cooling efficiency research has focused on techniques to reduce heat sink temperatures, and reduce the required temperature differences between the refrigerant and the source and sink. For example, rejecting condenser heat to water instead of air reduces the temperature difference that is needed for adequate heat transfer; air-cooled condensers typically require a refrigerant temperature that is 20 °C higher than condenser inlet air, while exchange with water only needs a 10 °C temperature difference.

The research presented in this paper provides a foundation for the design of cooling systems that reject condenser heat to swimming pools, a strategy that has been deployed successfully in many installations [2,3], but that has not been widely adopted. One reason for the lack of application is the lack of research, documentation and standardization. Our thesis is that a better understanding of the mechanisms that drive performance and savings could inform the development of guidelines for appropriate design of these systems, and could lead to more prevalent adoption, resulting in cost-effective energy and peak demand savings. The savings should come from three mechanisms:

1. Lower sink temperature since pool water is cooler than outdoor air during most cooling periods.
2. Improved heat transfer at the condenser since exchange with water is more effective than exchange with air.
3. Reduction of energy consumption for pool heating.

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Nomenclature

A_{cond}	surface area of conduction to ground (m^2)
A_s	surface area used for shape factor calculations (m^2)
A	surface area of pool at air–water interface (m^2)
C_{Bowen}	Bowen coefficient[3] (61.3 Pa/°C)
$d_{poolavg}$	average pool depth (m)
e_a	vapor pressure in ambient air (Pa)
e_s	saturation vapor pressure of air at the pool temperature (Pa)
E_{sky}	emissivity of sky (–)
E_w	emissivity of water (–)
g	acceleration of gravity (m/s^2)
Gr_L	Grashof number (–)
$\bar{h}$	average convection coefficient (W/m^2 °C)
h_{evap}	wind speed function for evaporation (W/m^2 Pa)
HR	humidity ratio (kg/kg)
k_{air}	thermal conductivity of air (W/m °C)
k_{soil}	thermal conductivity of soil (W/m °C)
L	average length of pool (m)
L_c	characteristic length of pool used for shape factor calculations (m)
$\overline{Nu_L}$	average Nusselt number (–)
O_{sky}	opaque sky cover (tenths)
P	perimeter of pool (m)
p_a	ambient pressure (Pa)
p_o	reference pressure (Pa)
Pr	Prandtl number (–)
$\dot{q}_{ss}$	dimensionless conduction heat rate (–)
q_{cond}	conduction heat flux (W/m^2)
q_{conv}	convection heat flux (W/m^2)
q_{evap}	evaporation heat flux (W/m^2)
q_{rad}	radiation heat flux (W/m^2)
Q_{solar}	solar heat gain (W)
R_{Bowen}	Bowen ratio (–)
Ra_L	Rayleigh number (–)
S	solar input (W/m^2)
T_a	ambient air temperature (°C)
T_{dew}	dew point temperature (°C)
T_w	swimming pool temperature (°C)
T_{sky}	effective sky temperature (°C)
T_{soil}	soil temperature (°C)
V	wind speed (m/s)
α	absorptivity of water (–)
β_a	thermal expansion coefficient of air (1/°C)
β_w	thermal expansion coefficient of water (1/°C)
ρ	density of water (kg/m^3)
σ	Stefan-Boltzmann constant (5.67 E^{-8} W/m^2 K^4)
ν	kinematic viscosity of air (m^2/s)
w	average width of pool (m)

The practical use of condenser heat rejection to swimming pools relies critically on the natural temperature regulation of pools by conductive heat exchange with the ground, convective and evaporative heat exchange with the air, and radiative heat exchange with the sky. The key is to balance heat rejection from the space cooling system with heating demand for the pool, such that pool temperature is maintained in a desirable range. We expect that this balance will be easiest to maintain in climate regions of the western United States, or other semi-arid regions with low ambient humidity and relatively low nighttime temperatures. In these regions, heat dissipation from swimming pools is increased by high evaporation rates in low humidity environments, and by longwave radiative cooling which increases with low ambient humidity and clear skies. Anecdotal evidence suggests that heat dissipation from pools in these climates is such that pool heating is often required to maintain desired water temperature, even when space cooling is required to maintain desired indoor temperature. In this case, heat rejected from cooling equipment could directly displace energy consumed to heat a pool, while concurrently improving the COP of the cooling system.

The objective of this paper is to document and discuss the development of a model to simulate the energy and mass balance of a swimming pool in natural interaction with its local environment; subsequent research will validate the model for simulation of a swimming pool used as a heat sink for vapor-compression air conditioning. Since there is no standardized approach to modeling the thermal behavior of swimming pools, this research draws from the conclusions of many authors to develop a clear and generalized method, and validates model predictions against long-term experimental measurements from a pool in Davis, CA.

[3] The Bowen Coefficient is 6.13 Pa/°C for the case when evaporation from a water surface does not significantly impact absolute humidity of the air.

2. Methodology and results

2.1. Model development

An analytical model to determine the heat and mass transfer for a swimming pool was developed to calculate the transient thermal behavior of a pool based on hourly weather data. The model relies on detailed information about the site and the operating characteristics of the pool. Based on meteorological inputs and system conditions, at each hourly time step (t), the calculations draw on empirical and theoretical heat transfer correlations to estimate the steady state heat transfer rates for conductive, convective, radiative, and evaporative heat exchange mechanisms. Rates are integrated across the hour, and energy and mass storage terms are calculated to determine the average pool temperature at the beginning of the next hour ($t+1$). Meteorological inputs and system conditions at ($t+1$) are then used to solve for system conditions at the following hour ($t+2$). The following sections describe the basis for calculating heat transfer rates for each mechanism considered in the model.

2.1.1. Insolation

The heat gain (W) due to solar radiation is found by multiplying the solar insolation at the pool surface by the absorptivity and area of the pool.

$$Q_{solar} = S\cdot\alpha\cdot A \tag{1}$$

The concept is simple, but determining the solar insolation and absorptivity are challenging prospects. Insolation at the pool surface is comprised of both direct and diffuse radiation, so when a pool is partly shaded by nearby objects, raw meteorological data for the global horizontal insolation is not representative of actual conditions. To compensate, shading of the pool must be described for each hour by inspection of the site and analysis of solar pathways for the latitude, season, and time period of the simulation. Diffuse insolation is used as the solar input for shaded periods and global insolation is used for un-shaded periods; periods of fractional shading receive a corresponding fraction of global and diffuse insolation.

Absorptivity is even more complicated due to the phenomenon of multiplicative reflection and absorption in transparent materials. Only a portion of the solar radiation available at the pool surface is accumulated as heat in the water volume; a fraction is reflected by the water surface, and of the portion that passes unabsorbed through to the pool, a fraction is absorbed by the pool bottom and a fraction is reflected back into the pool. Water clarity affects the absorption rate per unit depth, and the pool construction, especially color, affects absorption at the pool bottom. Moreover, the fraction of radiation reflected from the water surface changes with solar incident angle, so the net absorptivity of the pool varies by time and season. However, for the model presented here, an annual average absorptivity was calculated using a method presented by Wu [4], which uses latitude, longitude, refractive index of water and air, pool bottom absorptivity, and average depth. The approach divides solar insolation into separate spectral bands to account for the fact that energy content and extinction rates vary as a function of wavelength; it considers multiplicative reflection and absorption, and the impact of incident angle. According to validation by Wu, the method predicts absorptivity to within 3.67% of experimental observations over the course of a day.

2.1.2. Conduction

Conduction between the swimming pool and ground is simpler to model, and in most circumstances accounts for less than 1% of the total energy loss from the pool [5–7]. Since the time constant of thermal response for the earth is very large, pool temperature only affects ground temperature very near the pool walls, and transient pool temperature has very little effect on the daily temperature distribution in the ground. Therefore, most authors assume that soil temperature remains constant, and since the temperature difference is small and conduction is minimal compared to other heat transfer mechanisms, many authors ignore this heat transfer component altogether. A review of the literature identified several different models to estimate conduction effects. All authors rely on a standard one-dimensional conduction equation; some use the thermal conductivity of the wall and a constant ground temperature, while others approximate a total thermal resistance for the pool wall plus soil in a temperature transition zone. Govaer [6] accounts for seasonal changes in ground temperature, but few account for the effect of a vertical temperature profile in the ground. Hull developed a semi-empirical method which uses the distance from the bottom of the pool to a constant temperature sink, the ground conductivity and pool dimensions [8]. The model presented here includes an analysis of conduction, but assumes a constant ground temperature, even across seasons, and a soil conductivity of 0.52 W/m-K [9].

While a three dimensional conduction model could be used, since the ground temperature is assumed to remain uniform and constant, this model simplifies the conduction problem to a one-dimensional function by using a shape factor to account for geometry effects. Shape factors have been developed analytically for many different geometric cases; for this model the swimming pool is approximated as a cuboid embedded in an infinite medium, and a shape factor given by Incropera [9] is adapted to account for heat transfer through five faces of the cuboid with ground interface area similar to that of the pool.

d/D	$\dot{q}_{ss}$
0.1	0.943
1.0	0.956
2.0	0.961

$$q_{\mathrm{cond}} = \frac{1}{2L_c}\dot{q}_{ss}k_{\mathrm{soil}}\frac{A_s}{A_{\mathrm{cond}}}\left(T_{\mathrm{soil}} - T_{\mathrm{pool}}\right) \tag{2}$$

where

$$As = 2D^2 + 4Dd \tag{3}$$

$$d = 2d_{\mathrm{poolavg}} \tag{4}$$

$$L_c = \left(\frac{A_s}{4\pi}\right)^{0.5} \tag{5}$$

$$D = \left(A_{\mathrm{cond}} + d^2\right)^{0.5} - d \tag{6}$$

2.1.3. Radiation

Exchange of longwave radiation with the sky is one of the most significant pool cooling effects; it occurs continuously, separate from solar radiation. The magnitude of this heat flux is calculated using equation (7), the standard radiative heat transfer equation. The approach relies on the effective sky temperature – a value that reduces the complex phenomenon of radiant exchange between the pool, the semi-opaque atmosphere, and space beyond, to a simple radiative exchange between the pool and a much larger surface of representative temperature. Walton [10] developed two methods to determine the effective sky temperature; one is a function of infrared radiation from the sky, the other relies on the dew point temperature, sky emissivity and opaque sky cover. The latter approach was used here, as the infrared sky radiation is not generally measured by standard meteorological stations.

$$q_{\mathrm{rad}} = \sigma E_w\left[\left(T_{\mathrm{sky}} + 273\right)^4 - (T_w + 273)^4\right] \tag{7}$$

$$T_{\mathrm{sky}} = (T_a + 273)\cdot\left(E_{\mathrm{sky}}^{0.25}\right) - 273 \tag{8}$$

$$E_{\mathrm{sky}} = \left[0.787 + 0.764\cdot\log\left(\frac{T_{\mathrm{dew}} + 273}{273}\right)\right]\left[1 + 0.224\cdot O_{\mathrm{sky}} - 0.0035\cdot O_{\mathrm{sky}}^2 + 0.00028\cdot O_{\mathrm{sky}}^2\right] \tag{9}$$

This radiative heat transfer decreases with increases in dew point temperature, opaque sky cover, and ambient air temperature. It can account for up to 60% of the total thermal losses at night in arid climates with low humidity, minimal cloud cover, and low nighttime temperatures. Compared with the other two parameters, dew point temperature, an indicator of ambient humidity, has a relatively small impact on the overall longwave radiative heat transfer. However, there is an obvious correlation between low ambient humidity and a low degree of cloud cover. Fig. 1 and Fig. 2 illustrate the magnitude of longwave radiation loss (W/m^2) as a function of opaque sky cover and ambient temperature respectively. Note that heat flux into the pool is the positive convention, so negative exchange of longwave radiation represents cooling of the pool. It is critical for the model's accuracy to obtain cloud cover data for each hour of the day because, as Fig. 1 shows, the radiative heat transfer can differ significantly between clear sky and cloudy conditions.

2.1.4. Evaporation

Evaporation is an especially complicated phenomenon to model for swimming pools since:

1. There is no commonly accepted theoretical approach for estimating evaporation rates from free water surfaces [11]
2. Empirical equations may only be appropriate under the conditions for which they were developed

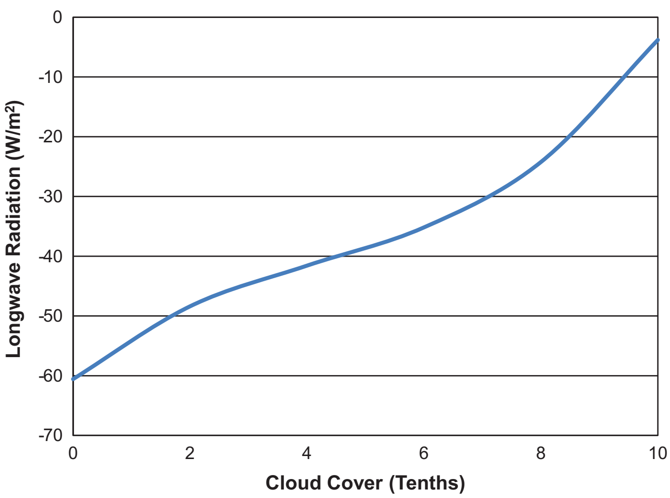

Fig. 1. Exchange of longwave radiation as a function of cloud cover (T_a = 30 °C, HR = 0.010, T_w = 25 °C).

3. Evaporation is sensitive to local environmental conditions, which differ from available meteorological data due to the proximity of obstructions such as buildings and trees, and microclimatic patterns related to neighborhood scale phenomena

The model developed herein couples empirical formulae for mass transfer from free water surface evaporation with empirical and theoretical heat transfer correlations to develop a more complete model of the evaporative heat and mass transfer mechanisms at play in a swimming pool.

Evaporation is driven by wind speed, and by the difference between the saturated vapor pressure of air at the pool surface temperature and the vapor pressure of ambient air. Thus, evaporation is greater in arid climates, and is typically the most significant heat transfer mechanism for the overall energy balance of a pool. Equations (10) and (11) were developed by McMillan [12], and confirmed by Sweers [13] and Sartori [11]. They describe the relationship between evaporative heat transfer and relevant environmental conditions.

$$q_{evap} = -h_{evap} \cdot (e_s - e_a) \quad (10)$$

$$h_{evap} = 0.0360 + 0.0250 \cdot V \quad (11)$$

where the wind velocity V is corrected to a height of 3 m.

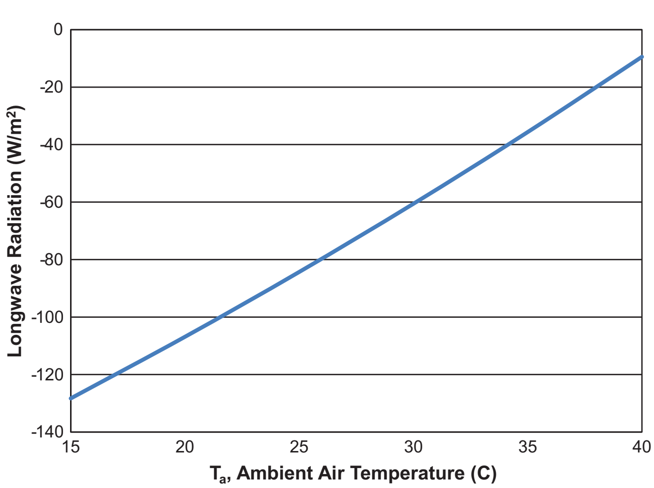

Fig. 2. Exchange of longwave radiation as a function of ambient air temperature (clear sky, HR = 0.010, T_w = 25 °C).

These equations were developed experimentally by correlating water temperature in several lakes to meteorological measurements of air temperature, relative humidity, and wind speed. A negative value for q_{evap} indicates that water is evaporated, sensible heat is lost from the water mass, and latent heat is gained in the air mass. The empirical wind speed function for evaporation, h_{evap}, accounts for the latent energy content of water vapor and the rate at which water vapor diffusion occurs under different wind conditions, while the equation for q_{evap} accounts for the evaporative potential due to the difference between vapor pressure in the ambient air and saturation conditions at the water surface temperature. It's worth noting that the evaporation rate is driven by the relationship between pool temperature and the absolute humidity of the ambient air, but it is not directly correlated to the ambient dry bulb temperature.

2.1.5. Convection

Evaporative and convective heat transfer phenomena are related; they operate by very similar mechanisms and can be reasonably conceptualized as a single process of coupled heat and mass transfer. Mass transfer and the associated transformation of sensible heat to latent heat occur by evaporation, while sensible heat transfer occurs by convection. A difference in absolute humidity, or vapor pressure, is the driving potential for evaporation, and a temperature difference is the driving potential for convection. Wind affects both by increasing the total effective interface for heat and mass transfer, and notwithstanding the role of wind, evaporation and convection are rate limited by mass diffusivity and thermal diffusivity respectively. The evaporation equations indicate that the water mass provides all sensible heat for phase change to latent heat through evaporation. However, if you consider evaporation and convection together, it is clear that as the water cools sensibly due to evaporation, convective heat transfer rates will shift, and given that the air is warmer than the water surface some sensible heat for evaporation will effectively be drawn from the air by convection. In contrast, if evaporation occurs under conditions where the air is cooler than the water, all sensible heat for evaporation must necessarily be derived from the water mass. Thus, as the two phenomena are closely related, the convective heat transfer rate can be derived theoretically as a function of the evaporative heat transfer rate. Bowen [14] expresses the relationship as a ratio:

$$\frac{q_{conv}}{q_{evap}} = R_{bowen} \quad (12)$$

which can be calculated by:

$$R_{bowen} = C_{Bowen} \cdot \frac{p_a}{p_o} \cdot \frac{(T_w - T_a)}{(e_s - e_a)} \quad (13)$$

Bowen developed this formula from first principles, based on a control volume analysis of sensible and latent heat densities in a differentially small element of air, subject to molecular and thermal diffusivity, and forced air movement. Note that the formula accounts for the impact of ambient pressure on the ratio of convective and evaporative heat transfer. Using the Bowen ratio, the convective heat transfer rate is determined simply by multiplying by the evaporative heat transfer rate:

$$q_{conv} = R_{bowen} \cdot q_{evap} \quad (14)$$

The net heat transfer by the coupled process of convection and evaporation is simply the sum of q_{conv} and q_{evap}. Fig. 3 plots R_{bowen} alongside the net heat transfer by convection plus evaporation, for several different ambient dry bulb temperature conditions, all as a function of humidity ratio. Note that the humidity ratio at low

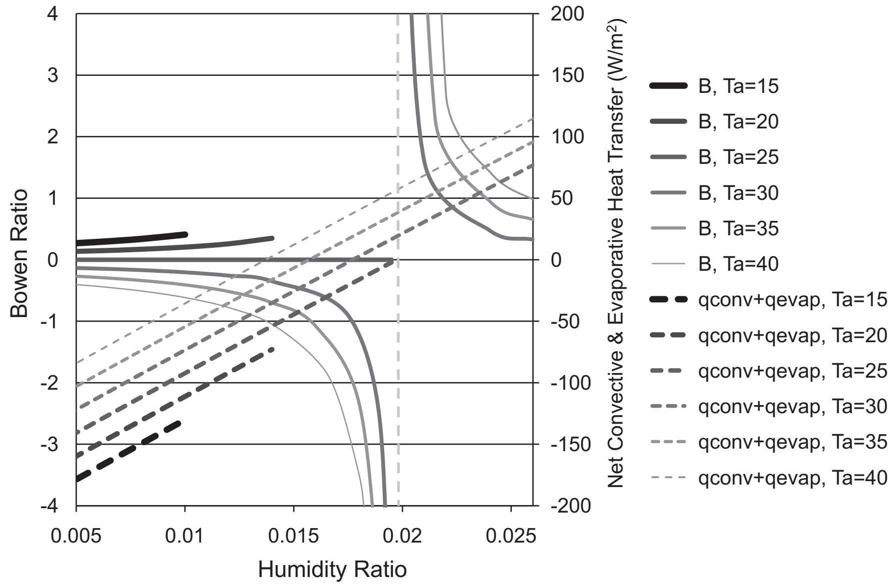

Fig. 3. Plot of Bowen ratio and net heat transfer by convection and evaporation, for several different ambient dry bulb temperatures, as a function of humidity ratio. Wind speed = 1 m/s, water temperature = 25 °C.

ambient dry bulb temperatures is limited by saturation, and that the Bowen ratio diverges asymptotically at the saturation humidity for air at the water surface temperature.

For conditions where humidity of the ambient air is less than the saturation humidity at the water surface temperature (the left side of Fig. 3), a positive Bowen ratio indicates that heat is lost from the water mass by both evaporation and convection. As illustrated by the net convective and evaporative heat transfer lines in Fig. 3, these conditions result in the maximum water cooling effect. As ambient dry bulb temperature approaches the water surface temperature the Bowen ratio approaches zero. At this point no heat is exchanged by convection, though heat may still be lost from the water by evaporation. A negative Bowen ratio indicates that evaporation and convection have opposing effects. A ratio between 0 and -1 means that the cooling effect of evaporation is dominant, a ratio of −1 indicates a net-zero balance of convection and evaporation, and a ratio beyond −1 means that convective heat gains to the pool dominate evaporative losses, resulting in a net heat gain to the water. For conditions where ambient humidity is greater than the saturation humidity at the water surface temperature (the right side of Fig. 3), condensation and convection will both contribute heat to the water mass. Note that for such conditions the Bowen ratio must be positive, since such absolute humidity conditions cannot occur for air temperatures below the water surface temperature.

At zero wind speed, the solution of equation (14) should equate to other well developed theoretical models for heat transfer. When the water surface temperature is greater than the air temperature the solution should agree with models for buoyancy driven free convection. If water surface temperature is less than the air temperature, air movement should stagnate above the water mass and equation (14) should yield similar results to models for conduction with a semi-infinite non-circulating mass. Equations (15) through (18), presented by Incropera [9], were used to validate equation 14 at zero wind speed for cases of buoyancy driven free convection; equations (19) through (21) were used similarly for conduction with a stagnant air mass.

For buoyancy driven convection:

$$q_{\text{conv}} = \overline{h}\cdot(T_w - T_a) \tag{15}$$

where:

$$\overline{h} = \frac{\overline{Nu_L}\cdot k_{\text{air}}}{L_c} \tag{16}$$

$$L_c \equiv \frac{A}{P} \tag{17}$$

$$\overline{Nu_L} = Ra_L^{1/3} \quad \left(10^7 \lesssim Ra_L \lesssim 10^{11}\right) \tag{18}$$

and for conduction with non-circulating air:

$$q_{\text{conv}} = 0.932\cdot\frac{k_a(T_w - T_a)}{L_c} \tag{19}$$

where:

$$L_c = \sqrt{\frac{A_s}{4\pi}} \tag{20}$$

$$A_s = 2wL \tag{21}$$

The result of this validation shows that at zero wind speed, deriving the average convective heat transfer rate from the evaporative heat transfer rate and the Bowen Ratio, as described by equation (10), agrees with other well developed models for buoyancy driven convection or conduction in a non-circulating air mass to within 5%.

2.1.6. Other mechanisms

Other mechanisms that affect the pool temperature include swimmers, rain, makeup water, pool covers, and the thermal effect of pumps. Accounting for swimmers is particularly difficult because of the myriad variables involved, though some authors have considered it. For example, Molineaux [15] assumes a heat addition of approximately 400 calories per swimmer per hour, which would have a measurable effect on water temperature in a pool with heavy use. The impacts of rain and makeup water may be significant in certain instances and can be calculated by accounting for the mass and temperature of the added water, though these values may be difficult to estimate. Pool covers can significantly impact the thermodynamics of a pool, mostly by eliminating evaporation and reducing longwave radiative losses. Each pool

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cover has different absorptive, emissive, and insulative properties and must be modeled accordingly. Pumps contribute heat to the pool, in part through frictional interactions with piping and dissipation of kinetic energy, and in part by way of heat generated within the pump. The design, operation, and site-specific meteorological characteristics of each pool will impact the relative importance of each of these heat additions, such that in certain cases they should be included in the model.

2.2. Model validation

To validate the calculations discussed herein, an experiment was conducted at a residential swimming pool in Davis, CA, a relatively hot and dry region in California Climate Zone 12. The thermal behavior of the pool was monitored over a 56 day period in spring 2009, and observations were compared to results from the model using hourly meteorological data for the same time period. The pool was left uncovered, no swimmers were permitted, no makeup water was added, and the filter pump was set to run continuously at a constant flow rate (Fig. 4).

2.2.1. Methodology for experimental validation

The model requires definition of an initial pool temperature, as well as several meteorological variables on an hourly basis, including: global horizontal insolation, pool shading, cloud cover, ambient dry bulb temperature, ambient humidity, wind speed, and barometric pressure. Future iterations of the model will accept the definition of an hourly thermal input from vapor-compression cooling equipment, though this study focuses on validation of the thermodynamic balance between a passive swimming pool and the environment.

Note that an accurate initial pool temperature is not absolutely necessary for the model to appropriately predict the long-term hourly temperature behavior of the pool, though it may take up to a few weeks for the model to track the actual pool temperature if the initial conditions are off by 5 °C. The results presented here use measured pool temperature as an initial condition for the simulation.

Although global horizontal solar insolation could be measured on site, in certain instances it is impossible to install a pyranometer in a completely un-shaded location; thus the model is designed to allow input from offsite meteorological measurements. The global horizontal insolation for each hour of the experiment was obtained from the California Department of Water Resources' California Irrigation Management Information System (CIMIS) [16] which reports measurements for a meteorological station in Davis, CA, as well as hundreds of other sites throughout the state. The meteorological station used to obtain hourly insolation values was close by and assumed to be representative of the test site. The direct and diffuse portions of this measurement were calculated using a quasi-physical model for converting hourly global horizontal to direct normal insolation developed by Maxwell and published by the NREL Solar Energy Research Institute [17].

Since the meteorological measurements of insolation are fully exposed, whereas pools are often surrounded by obstructions, an hourly shading factor for the pool was developed by inspection of the site and an analysis of solar pathways for the latitude, season, and time period of the experiment. The model uses diffuse insolation as the solar input for shaded periods and global insolation for un-shaded periods; periods of fractional shading receive a corresponding fraction of global and diffuse insolation.

For the experimental validation presented here, ambient temperature, relative humidity, and wind speed were measured on site. Some error is incurred due to slight variations in meteorological conditions at different points on site; however, the location of each measurement was selected to avoid significant misrepresentation of conditions at the pool surface. For example, the anemometer was placed to avoid eddies and vorticies that could occur very near a building. The wind speed measurement was corrected to a 3 m height using standard atmospheric boundary layer methods [18] since the McMillan wind speed coefficient presented in equation (11) is derived for wind speed at that height. These measurements could all be taken from regional meteorological data, or from typical meteorological year resources, though microclimatic variations between a meteorological station and an actual site introduces errors that are not associated with the mathematical model itself. Wind speed is an especially sensitive input variable. Since meteorological stations tend to be located in open, unobstructed areas, and sites of interest are often surrounded by nearby obstructions such as trees, fences, and buildings, measurements near to the ground do to not scale well using standard atmospheric boundary layer methods to correct for terrain differences. For example, over the test period presented here, CIMIS wind speed observations at 2 m in open terrain and corrected to 3 m in a highly obstructed urban area were consistently high as

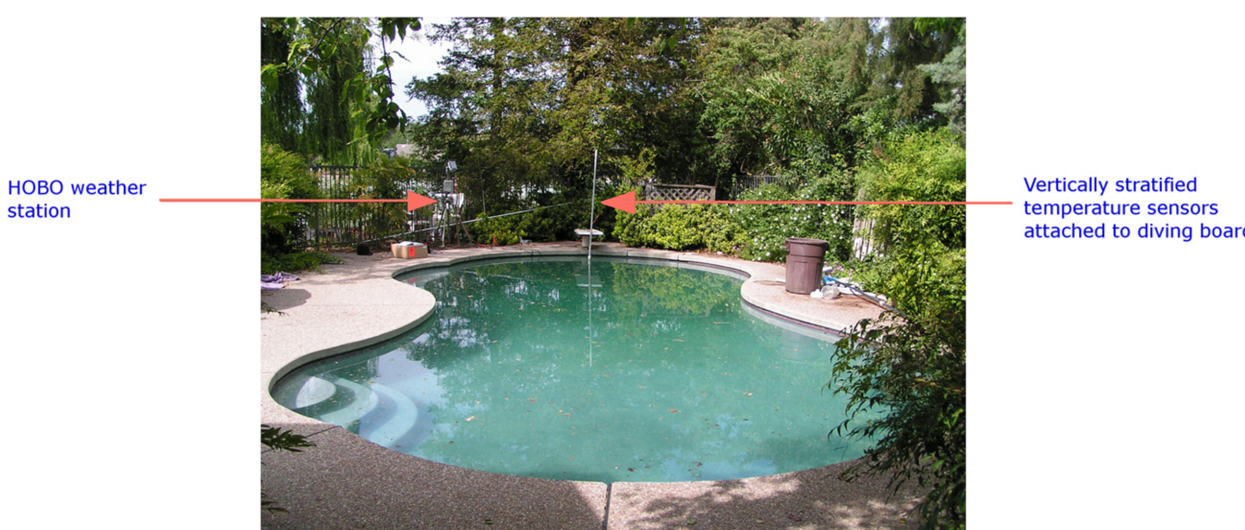

Fig. 4. Photo of pool in Davis California used for experimental validation.

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compared to actual measurements, with an RMS error of 1.6 m/s. This overestimation of wind speed would result in a consistent under estimation of pool temperature. Relative humidity and ambient temperature measurements vary as well, such that the pool temperature predictions from a simulation using CIMIS data and a simulation using site data differ with an RMS error of 3.0 °C.

Calculation of longwave radiative exchange requires information about the fractional portion of the sky dome that is obstructed by clouds, though this data is not regularly collected by all meteorological stations. For this simulation, data was obtained from the National Oceanic and Atmospheric Administration's (NOAA) Quality Controlled Local Climatological Database (QCLCD) [19] for a station in Sacramento, CA, which is about 20 miles from the experimental site.

Conduction between the pool and earth was calculated using a constant soil temperature of 15 °C. Transient effects due to diurnal and seasonal heat transfer from the pool were ignored, and the pool geometry was approximated as a cuboid as described previously by equation (6).

The temperature of the pool was measured at 1′, 4′, and 7′ from the pool bottom to develop an average pool temperature and to describe the extent of thermal stratification. For this experiment, since the filter pump ran continuously at a constant flow rate, the pool was relatively well mixed and no thermal stratification was observed.

2.2.2. Validation results

The model described herein used the initial inputs and hourly meteorological conditions to determine heat and mass exchange between the pool and environment, and to predict hourly average pool temperature. The predicted values were then compared to the observed temperature history and analyzed for accuracy, Fig. 5 illustrates the results.

The results suggest that, given input of appropriate meteorological conditions, an accurate prediction of the pool temperature can be made. The RMS error of the pool temperature prediction compared to measured values is 0.4 °C and the largest discrepancy is only 1.1 °C. The temperature sensors used in the experiment had an absolute error of ±0.2 °C, so accuracy of the model is very near sensor accuracy.

The most significant periods of error between measured and predicted pool temperature occur for approximately one week near the beginning of the test period, and for several days near the middle of the test period. The first instance is likely due to a storm that brought cloud cover and measurable precipitation. Although the model responds to data for both opaque cloud cover and global horizontal insolation, since those values were measured at offsite meteorological stations they could differ somewhat from local conditions. The error during this period can be reduced almost completely if values for cloud cover are inflated, but there are no theoretical grounds to include such adjustments in the model. The periods of error in the middle of the test are not related to any obvious meteorological event, and are not easily explained. In all instances the discrepancy rarely reaches 1.0 °C. It is noteworthy that the model recovered from poor prediction periods automatically, without any input other than the new meteorological data, which suggests that it is a robust model.

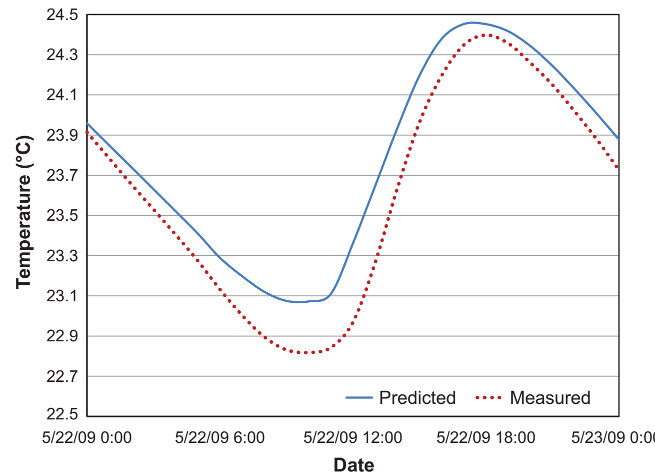

Fig. 6. Predicted and measured temperatures for a case of very good accuracy.

Figs. 6 and 7 illustrate typical diurnal cycles for the pool during the observed period. Fig. 6 is an example of one day for which the model gives a very accurate prediction, while Fig. 7 is for a relatively poor prediction. In both instances the simulation is in phase with the measurements; though the measured pool temperature transitions gradually between heating and cooling, while the predicted pool temperature responds more abruptly. This is likely due to the hourly time step implemented in the simulation. In a physical system meteorological conditions change continuously while the model relies on constant values for each hour. If weather data were resolved more continuously the model would respond more gradually.

Fig. 8 plots all hourly predicted pool temperatures against all hourly measured pool temperatures. A perfectly accurate model

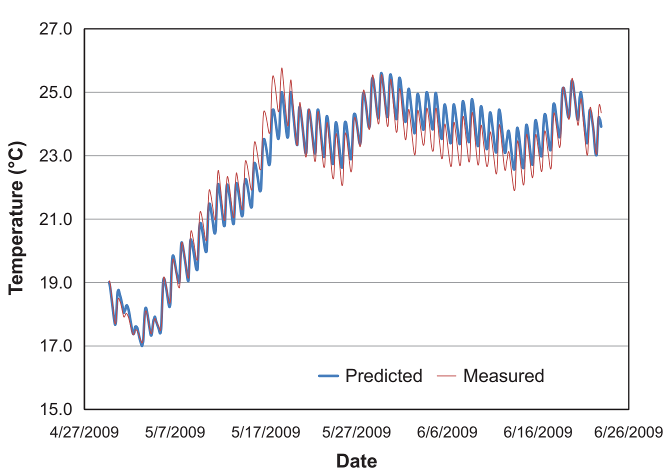

Fig. 5. Predicted and measured pool temperatures for a pool in Davis California observed 4/29/2009–6/22/2009.

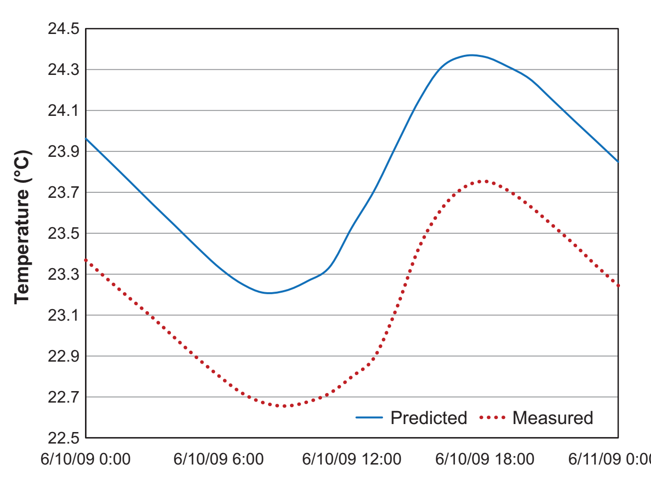

Fig. 7. Predicted and measured temperatures for a case of poor accuracy (error of 0.6°).

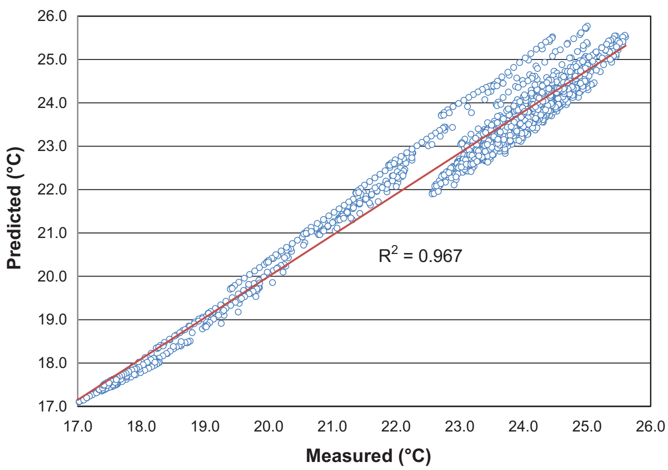

Fig. 8. Comparison of predicted and measured pool temperature for all hourly observations.

would have a one-to-one relationship between the values. For the period of validation this simulation has very good fit with an R^2 value of 0.967.

The relative impact of each heat transfer mechanism over the duration of the experiment is illustrated in Fig. 9. Note that solar insolation is the only heat gain, the sum of all heat losses balances with the solar gains, and that evaporation and emission of longwave radiation dominate over conduction and convection.

Fig. 10 plots the magnitude of each heat transfer mechanism and the total heat accumulated across two typical days from the experiment. Convection and longwave radiative exchange with the sky are affected directly by diurnal air–temperature cycles while evaporation is not. Note that heat flux into the pool is the positive convention, so negative values represent cooling of the pool. Longwave radiative exchange with the sky is consistently negative since the effective sky temperature never exceeds the pool temperature during the plotted period.

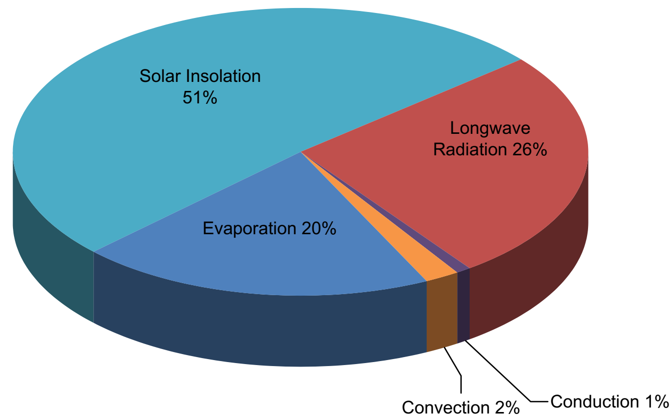

Fig. 9. Relative magnitude of each heat transfer mechanism integrated across the duration of the experiment.

3. Discussion

3.1. Experimental considerations

Although average pool temperature has been presented as the metric by which to validate simulations, mass evaporation could be used as well. The predicted mass evaporation rate can be calculated directly in the model by relating heat transfer by evaporation to the latent heat of vaporization of water; and the actual cumulative evaporation can be measured directly by monitoring the water level. However, since the mass rate of evaporation is small compared to pool volume, it is very difficult to accurately measure changes in depth on an hourly basis. The barometrically corrected water depth sensors used for our experimentation are accurate to within 0.0035 m, so for a pool with 50 m^2 surface area and 5 kg/h evaporation the hourly change in depth of 0.0001 m cannot be reliably observed, especially considering the noise associated with naturally occurring disturbances to the water surface. The issue was further complicated in this experimental validation because the

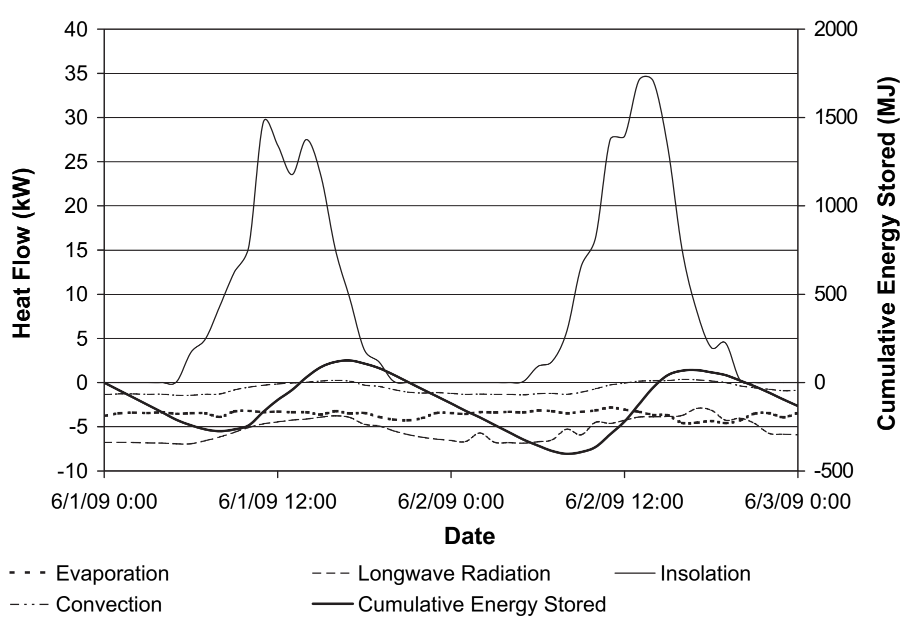

Fig. 10. The magnitude of each heat transfer mechanism and heat accumulation for two typical days of the experiment (conduction heat flow was omitted from this figure as it was effectively zero).

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depth sensor was mounted to a pole at the end of a diving board, which seemed to expand and contracted slightly with diurnal temperature cycles, causing sensor movement and misrepresentation of fluctuations in water depth. In subsequent field tests the depth sensor will be placed at a fixed point, and the water column above the sensor will be isolated from small disturbances to the pool surface.

3.2. Future work

The next phase of model development and validation involves the addition of heat from vapor-compression space cooling equipment, and the development of design guidelines for such heat pump systems in various climate zones in the western United States. A preliminary simulation was conducted for the pool studied here with the addition of heat rejected from a condenser. The condenser heat for each hour was calculated for a 3.5 ton heat pump assuming a constant COP, and was based on cooling loads generated in MICROPAS [20] for a 1764 square foot, single story home in California Climate Zone 12. Under this scenario the pool temperature never exceeded 28.5 °C. Another experiment will be conducted to compare this model with observations from a geothermal heat pump system that is coupled to a swimming pool with a gas-fired pool heater, solar thermal pool heaters, night radiative coolers, and fountains for evaporative water cooling. The intent is to account for the impact of all system components in the model in order to simulate performance under various configurations in multiple climate zones and develop guidelines to reduce energy consumption for space cooling while preventing overheating. Additionally, future work will explore the potential to offset pool heating costs during swing seasons when pool temperatures are low yet space cooling is required. Research is needed to clarify when this occurs and how much energy could be saved in various climate zones, and with different degrees of pool shading.

4. Conclusions

Predictions from the mathematical model developed match well with measured pool temperature results, suggesting that it could be used to accurately analyze the temperature response of a pool used as a thermal sink for a heat pump during the cooling season, or as a thermal source for a heat pump in the heating season. The accuracy of the model is impressive, and is due mostly to the extensive theoretical and empirical research by other authors to explain each heat transfer mechanisms at play in this scenario. It should be noted that our methodology to describe shading of the pool each hour is the only variable that was not derived from other published work or directly measured with instrumentation, and that no “correction factors” have been used to calibrate the model against the measurements. Although the model is very accurate, if used as a design tool it should be noted that meteorological conditions at a site may differ significantly from available data, and that predictions may not be as accurate as the validation results presented here. The test period allowed for validation of the model under multiple environmental conditions including clear and cloudy scenarios, as well as cool and very hot conditions. However, the model was not validated for extended cold periods, heavy rain conditions, mechanical thermal loading, or extreme climates.

5. Legal notice

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Western Cooling Challenge: Requirements and Performance Metrics

Requirements

1. Roof-top package unit (RTU) systems
2. Capacity (ARI 340/360 Standard Rating Conditions - Cooling Test), tons: 3 – 30
3. An entry need not be part of a model line that spans the entire capacity range
4. Demonstrated minimum production capacity: 500 units per year
5. Test results for one capacity will not necessarily qualify units with other capacities
6. Testing can be performed under observation and data analysis by University of California Davis Western Cooling Efficiency Center (WCEC) representative, at manufacturer's location or at a third-party facility
7. No use of R-22
8. WCEC-covered observation and data analysis costs may be limited to one unit per manufacturer, and/or a total of three units
9. WCEC will not cover the costs for observation and data analysis on a unit that fails to meet all Challenge requirements
10. Unit shall have under-voltage protection that prevents grid connection in a stalled condition
11. On-board electronics shall have the capability to communicate significant performance degradation
12. Heating capability is recommended for marketing purposes but is not required to meet the challenge

Performance Metrics

1. **Three Tests Required**
   a. ARI 340/360 Standard Rating Conditions (0% outdoor-air basis, standard external resistance)[1]
   b. WCC Nominal Peak Performance
   c. WCC Surrogate Annual Average
2. **Protocols for WCC Tests**
   a. *Indoor/Return Air:* 78 db/64 wb
   b. *External Resistance:* fixed orifice with 0.7" H_2O pressure drop at 350 cfm/ton
   c. *Filtration (Internal):* MERV 7 or higher
   d. *Outdoor Air Flowrate:* 120 cfm/rated-ton or higher (flows above 120 cfm/ton are not credited for cooling air from outdoor to indoor conditions, only for cooling beyond indoor conditions)
   e. *Max Supply Humidity Ratio:* 0.0092
3. **Nominal Peak Performance Test**
   a. *Test Condition:* full-capacity operation
   b. *Surroundings/Outdoor Air:* 105 db/73 wb
   c. *Minimum Sensible EER*[2]*:* 14.0
   d. *Minimum Sensible Capacity:* 95% of rated sensible capacity
4. **WCC Surrogate Annual Average Test**
   a. *Test Condition:* full-capacity operation plus one optional test mode[3]
   b. *Surroundings/Outdoor Air:* 90 db/64 wb
   c. *Minimum Sensible EER*[2]*:* 17.0, delivering at least 80% of rated sensible capacity
   d. *Maximum Water Use:* 4 gal/ton-hour with hardness of 200 ppm (as $CaCO_3$)

[1] For units that cannot be ARI Listed with 0% outdoor air, rated capacity will be determined at ARI indoor and outdoor temperature conditions in the operating configuration used for the WCC nominal peak performance test. Rated sensible capacity shall be calculated based on the temperature difference between indoor air and supply air.

[2] Including all parasitics - e.g. blowers, fans, pumps, controls (EER= sensible capacity/total kW)

[3] Manufacturer may specify full-capacity operation or a two-point test with 1) full-capacity operation, and 2) continuous partial-capacity operation, delivering not less than 120 cfm outdoor air per rated ton, and between 70% and 85% of ARI rated sensible capacity. Annual average test results will be either 1) full-capacity operation results or 2) a linear interpolation or extrapolation to 80% of rated sensible capacity based upon the two test points.

Draft Total Capacity Based Modifications mbh 05/21/09

Western Cooling Challenge: Requirements and Performance Metrics

Requirements

1. Roof-top package unit (RTU) systems
2. Capacity (ARI 340/360 Standard Rating Conditions - Cooling Test), tons: 3 – 30
3. An entry need not be part of a model line that spans the entire capacity range
4. Demonstrated minimum production capacity: 500 units per year
5. No use of R-22
6. Test results for one capacity will not necessarily qualify units with other capacities
7. The National Renewable Energy Laboratory (NREL) at their Golden, Colorado facility will do testing. Testing can be performed under observation and data analysis by University of California Davis Western Cooling Efficiency Center (WCEC) representative, at manufacturer's location or at a third-party facility
8. WCEC covered observation and data analysis costs may be limited to one unit per manufacturer, and/or a total of three units
9. WCEC will not cover the costs for observation and data analysis on a unit that fails to meet all Challenge Requirements and Performance Metrics
10. Unit shall have under-voltage protection that prevents grid connection in a stalled condition
11. On-board electronics shall have the capability to communicate significant performance degradation
12. Heating capability is recommended for marketing purposes but is not required to meet the challenge

Performance Metrics

1. **Three Tests Required**
    a. ARI 340/360 Standard Rating Conditions (0% outdoor air basis, standard external resistance)[1]
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    a. *Test Condition:* full-capacity operation
    b. *Surroundings/Outdoor Air:* 105 db/73 wb
    c. *Minimum Sensible EER[3]:* 14.0
    d. *Minimum Sensible Capacity:* 95% of sensible capacity at ARI 340/360 conditions[4]
4. **WCC Surrogate Annual Average Test**
    a. *Test Condition:* full-capacity operation plus one optional test mode[5]
    b. *Surroundings/Outdoor Air:* 90 db/64 wb
    c. *Minimum Sensible EER[2]:* 17.0
    d. *Minimum Sensible Capacity:* 80% of sensible capacity at ARI 340/360 conditions [6]
    e. *Maximum Water Use:* 4 gal/ton-hour with hardness of 200 ppm (as $CaCO_3$)

<u>**Schedule of Events**</u> *(updated 5/19/09)*

May 2009 Laboratory testing of WCC entries can begin
July 2009 Field testing of WCC entries can begin
January 2010 Shipments of WCC compliant products can begin (production products must be available by 1/11)

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[1]For units that cannot be ARI listed with 0% outdoor air, rated ton total capacity will be determined at WCC Peak indoor and outdoor temperature conditions in the operating configuration used for the WCC nominal peak performance test. Rated ton total capacity shall be calculated using the following equation: Rated ton total capacity = 4.45*supply cfm*(31.5 – enthalpy of supply air)..

[2]Airflows above 120 cfm/rated ton total capacity are not credited for cooling air from outdoor to indoor conditions.

[3]Including all parasitics - e.g. blowers, fans, pumps, controls (EER= sensible capacity/total kW)

[4]Not applicable to units that cannot be ARI Listed with 0% outdoor air.

[5]Manufacturer may specify full-capacity operation or a two-point test with 1) full-capacity operation, and 2) continuous partial-capacity operation, delivering not less than 120 cfm outdoor air per rated ton, and between 70% and 85% of ARI rated sensible capacity. Annual average test results will be either 1) full-capacity operation results or 2) a linear interpolation or extrapolation to 80% of rated sensible capacity based upon the two test points.

[6] For units to which footnotes 1 and 3 apply the Minimum Sensible Capacity is 85% of Sensible Capacity measured during testing to determine rated ton total capacity.

National Renewable Energy Laboratory
A national laboratory of the U.S. Department of Energy
Office of Energy Efficiency & Renewable Energy
Innovation for Our Energy Future

Coolerado 5 Ton RTU Performance: Western Cooling Challenge Results

Eric Kozubal and Steven Slayzak

Technical Report
NREL/TP-550-46524
September 2009

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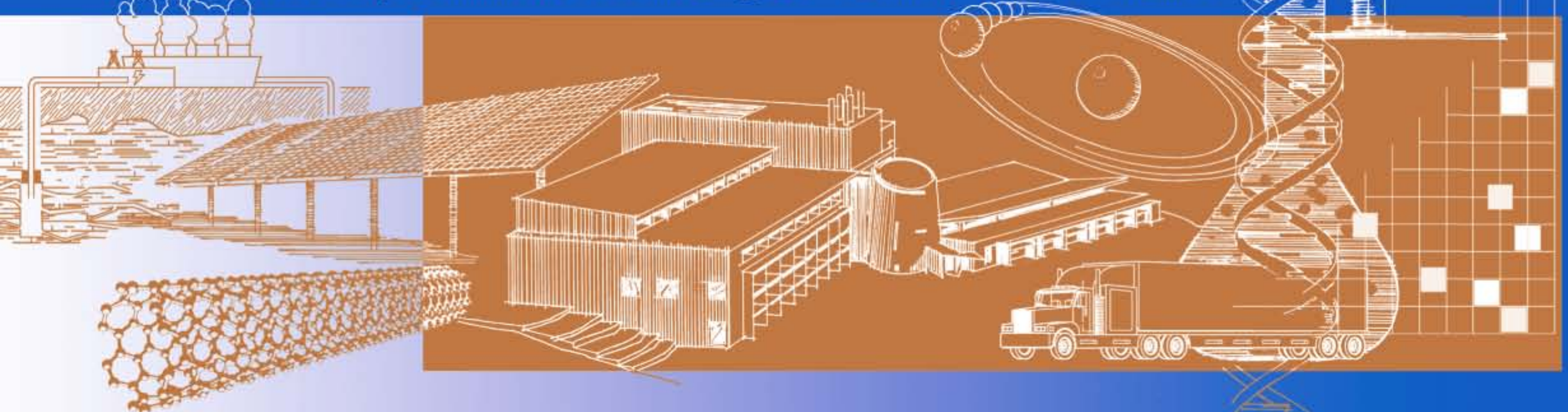

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Prepared under Task No. BEC71115

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Acronyms

DX	direct expansion
EA	exhaust air
EER	energy efficiency ratio
HVAC	heating, ventilation, and air-conditioning
NREL	National Renewable Energy Laboratory
OA	outside air
RA	return air
RTU	rooftop unit
SA	supply air
SHR	sensible heat ratio
w.c.	water column
WCC	Western Cooling Challenge
WCEC	Western Cooling Efficiency Center

Contents

Figures

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1 Introduction

The National Renewable Energy Laboratory (NREL) is tasked, through funding from the U.S. Department of Energy Office of Building Technology, to evaluate the performance of advanced cooling concepts that meet or exceed the performance criteria developed by the Western Cooling Efficiency Center (WCEC) (http://wcec.ucdavis.edu/). The WCEC has developed a set of criteria for test conditions, minimum energy, and water use performance for prototype cooling equipment. The WCEC has identified these conditions as indicative of western state climates. These criteria, named the Western Cooling Challenge (WCC), have been set forth as a challenge to manufacturers to improve the state-of-the-art space cooling products. NREL is to verify these criteria through laboratory testing at its heating, ventilation, and air-conditioning (HVAC) test facility (www.nrel.gov/dtet/lab_capabilities.html) in Golden, Colorado, which is uniquely suited to accurately measure the cooling performance, energy, and water use of advanced cooling systems. The facility provides flexibility to test prototype equipment and develop subsequent test methodology. Data are analyzed and reported to reflect performance at sea level elevation.

This report is intended for individuals with technical understanding of cooling technologies for buildings.

2 Unit Description and Test Method

NREL tested a prototype rooftop unit (RTU) manufactured by Coolerado Corporation (see Figure 1). The unit, an advanced ultra-cooler that uses the patented "M-cycle" process, is a hybrid indirect evaporative cooling and refrigeration direct expansion (DX) system. An airflow schematic of the RTU is shown in Figure 2. Return air (RA) and outdoor air (OA) are brought into the unit and cooled by an indirect evaporative media. Between 43% and 46% of this air is used as an indirect evaporative cooling stream. The balance is then passed through a refrigerant evaporator coil and supplied to the space by a high-efficiency fan. The exhaust air from the evaporative process is generally cooler than the ambient air and is therefore used for the heat sink air flow going through the refrigerant condenser coil. OA and exhaust air (EA) flow rates were matched during testing. The RA and supply air (SA) flow rates are also equal, thus there is no make-up air (to the space) supplied by the unit. The mode of operation can be described as recirculation and ventilation air cooling with no makeup air.

Figure 1. The prototype RTU and the unit being tested at the NREL HVAC laboratory

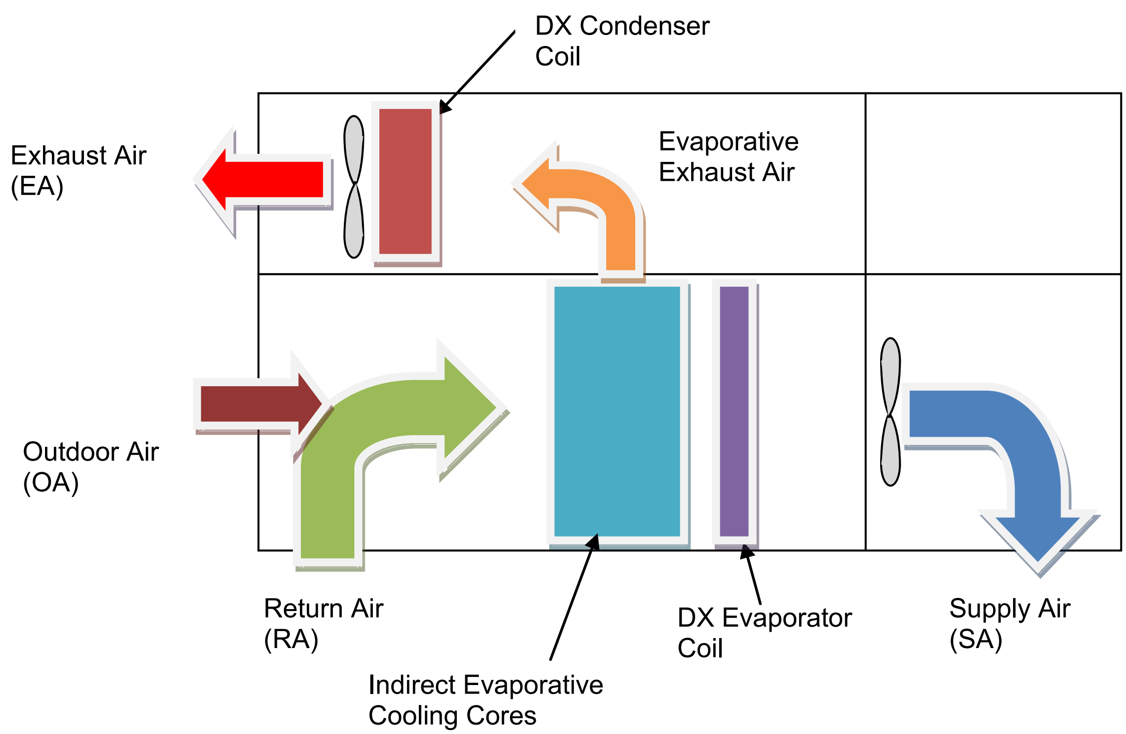

Figure 2. Coolerado RTU air flow schematic

The unit brings in OA and mixes it with RA to create a fresh air rate shown in equations (1) and (2).

$$\boldsymbol{OA\ Fraction} = \frac{\dot{V}_{OA}}{\dot{V}_{OA} + \dot{V}_{RA}} \qquad (1)$$

$$\dot{V}_{Ventilation} = \boldsymbol{OA\ Fraction} \times \dot{V}_{SA} \qquad \text{[cfm]} \qquad (2)$$

The nominal cooling capacity is given by equation (3) when tested at peak conditions. This number should not be confused with the net total cooling defined later in equation (11). Rather, this number is used as a baseline to determine ventilation cooling capacity and a nominal specific cooling rate in cfm/ton. The cfm/ton calculation is also used to determine the static pressure imposed during the test, which is set at 0.7 in. w.c. at 350 cfm/ton. (See the WCC test specification for further details.)

$$\boldsymbol{Capacity} = \dot{m}_{SA} \times (\mathbf{31.5} - h_{SA}) \qquad \text{[Btu/h]} \qquad (3)$$

The RTU utilizes a high ventilation rate to provide air to the evaporative process. The specification states that ventilation cooling credit is limited to a specified OA flow. The nominal cooling capacity is used to determine the credited ventilation cooling and is calculated with equation (4).

$$\dot{V}_{Ventilation,Credited} = \mathbf{0.01} \times \boldsymbol{Capacity} \qquad \text{[cfm]} \qquad (4)$$

Given the constraint:

$$\dot{V}_{Ventilation,Credited} \leq \dot{V}_{Ventilation}$$

The unit was given a single air flow at the OA inlet location. The RA and OA were psychrometrically mixed at the test facility rather than inside the RTU. Cooling capacity at the WCC test conditions is calculated with equations (5–13).

Space (Recirculation) Air Cooling:

$\boldsymbol{Total\ Space\ Cooling = \dot{m}_{SA} \times (h_{RA} - h_{SA})}$	[Btu/h]	(5)
$\boldsymbol{Sensible\ Space\ Cooling = \dot{m}_{SA} \times C_p \times (T_{RA} - T_{SA})}$	[Btu/h]	(6)
$\boldsymbol{Latent\ Space\ Cooling = Total\ Space\ \ Cooling - Sensible\ Space\ Cooling}$	[Btu/h]	(7)

Credited Ventilation Air Cooling:

$\boldsymbol{Total\ Ventilation\ Cooling = \dot{m}_{OA,Credited} \times (h_{OA} - h_{RA})}$	[Btu/h]	(8)
$\boldsymbol{Sensible\ Ventilation\ Cooling = \dot{m}_{OA,Credited} \times C_p \times (T_{OA} - T_{RA})}$	[Btu/h]	(9)
$\boldsymbol{Latent\ Ventilation\ Cooling = Total\ Ventilation\ Cooling - Sensible\ Ventilation\ Cooling}$	[Btu/h]	(10)

Net Cooling:

$\boldsymbol{Net\ Total\ Cooling = Total\ Space\ Cooling + Total\ Ventilation\ Cooling}$	[Btu/h]	(11)
$\boldsymbol{Net\ Sensible\ Cooling = Sensible\ Space\ Cooling + Sensible\ Ventilation\ Cooling}$	[Btu/h]	(12)
$\boldsymbol{Net\ Latent\ Cooling = Latent\ Space\ Cooling + Latent\ Ventilation\ Cooling}$	[Btu/h]	(13)

Testing was done at nominal peak and surrogate annual conditions. The psychrometric conditions for the cooling challenge are shown in Table 1. RA conditions apply to the peak and annual tests.

Table 1. WCC Psychrometric Conditions

	T_{db}	T_{wb}	Unit
Nominal Peak OA Conditions	105	73	°F
Surrogate Annual OA Conditions	90	64	°F
RA Conditions	78	64	°F

The unit has three primary modes of operation that are labeled as stages 0 to 2. Stage 2 has a higher OA fraction to provide additional air to the condensing coil.

- Stage 0: Indirect evaporative cooling only, with 43% OA fraction.
- Stage 1: Indirect evaporative cooling + low stage DX cooling, with 43% OA fraction.
- Stage 2: Indirect evaporative cooling + high stage DX cooling, with 46% OA fraction.

See the WCEC Web site (http://wcec.ucdavis.edu/) to view the complete WCC test specifications.

3 Results

The following graphs illustrate the cooling process of the Coolerado RTU on a psychrometric chart for all cooling stages. Figure 3 and Figure 4 show the mixed air and SA conditions at WCC nominal peak and surrogate annual conditions, respectively. The figures show the progression of cooling capacity and supply conditions as the unit ramps up from stage 0 to stage 2. At nominal peak conditions, the RTU provides space cooling with a sensible heat ratio (SHR) between 0.92 and 1.25. Stage 2 is used to rate the system at nominal peak conditions.

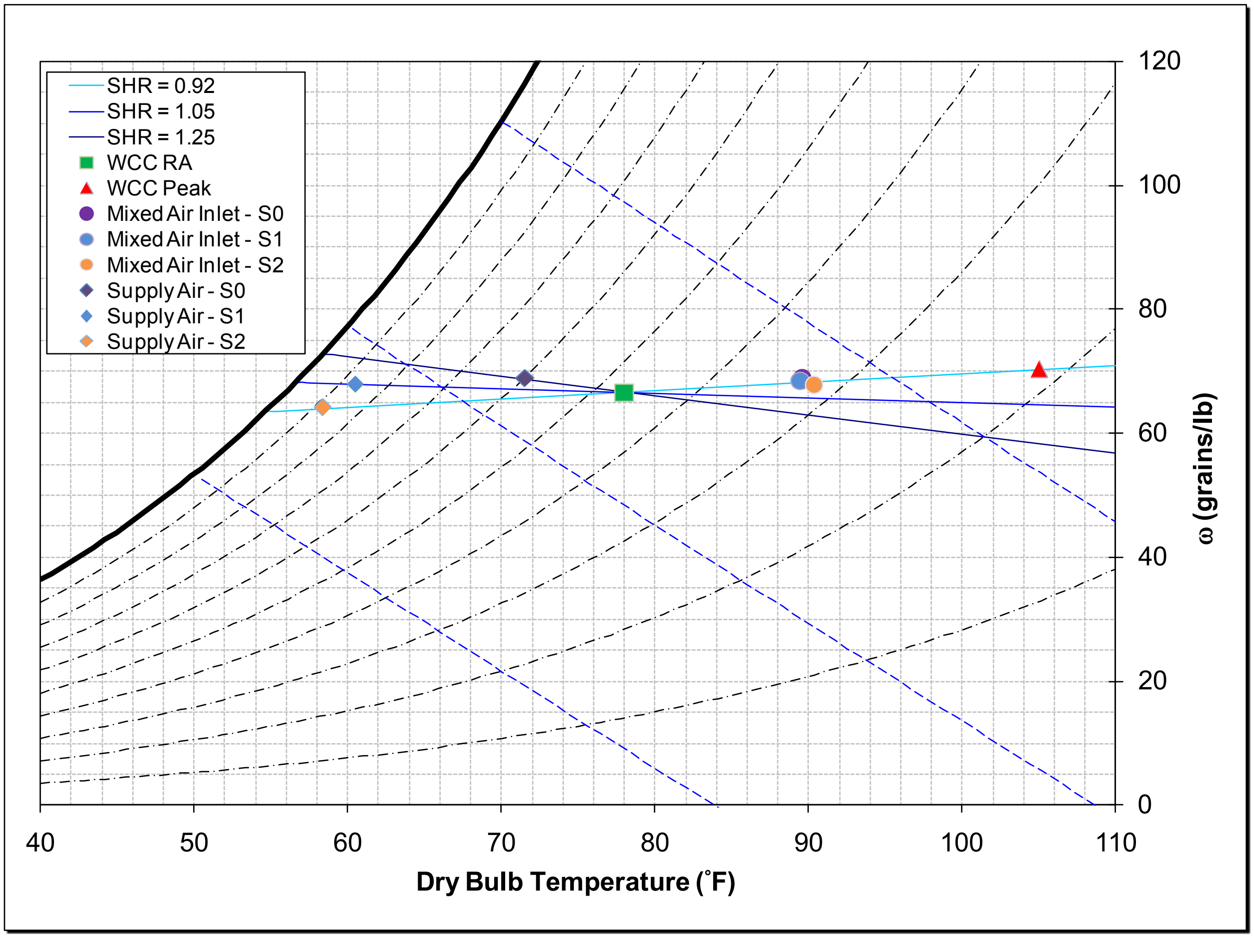

Figure 3. Psychrometric chart of RTU performance at nominal peak conditions (Shown at 0 ft elevation. S0, S1, and S2 denote stages 0, 1, and 2, respectively)

At surrogate annual test conditions, the RTU provides space cooling with an SHR between 0.68 and 0.81. The large dehumidification capacity is primarily due to the large OA flow provided by the unit. For surrogate annual conditions, only stages 0 and 1 are used to rate the system.

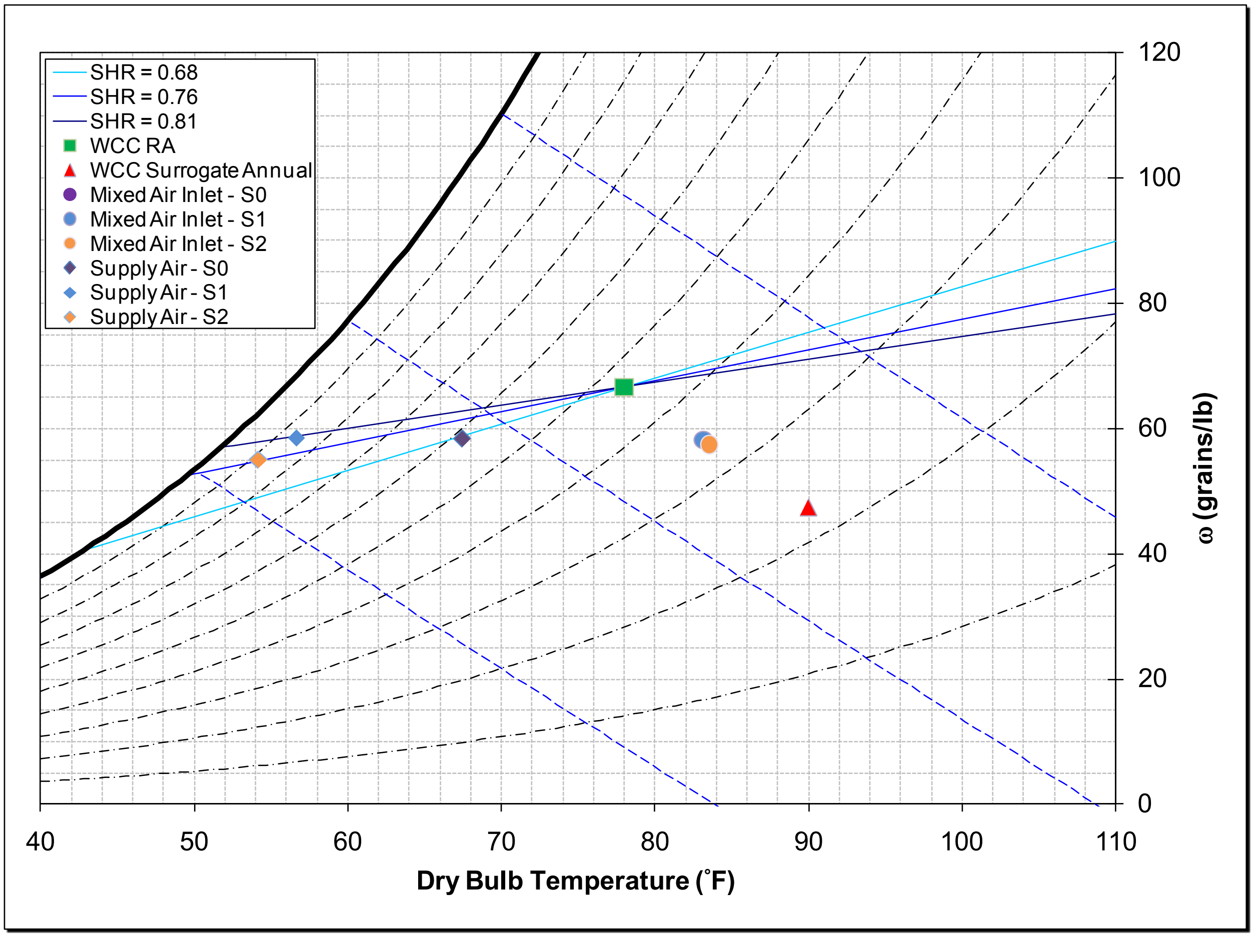

Figure 4. Psychrometric chart of RTU performance at surrogate annual conditions (Shown at 0 ft elevation. S0, S1, and S2 denote stages 0, 1, and 2, respectively)

Figure 5 shows the interpolation of cooling and power at surrogate annual conditions using stages 0 and 1. These data are used to estimate annual cooling performance, assuming that the average building load over a cooling season uses 80% of the measured sensible capacity at peak conditions. The interpolated capacity and power use are then used to calculate the surrogate annual energy efficiency ratio (EER). The same approach is used for water use. Total water use and water evaporation (in gallons per hour) are interpolated to 80% of the sensible capacity (see Figure 6). This number is then used with the surrogate cooling capacity to determine gallons per sensible ton·h.

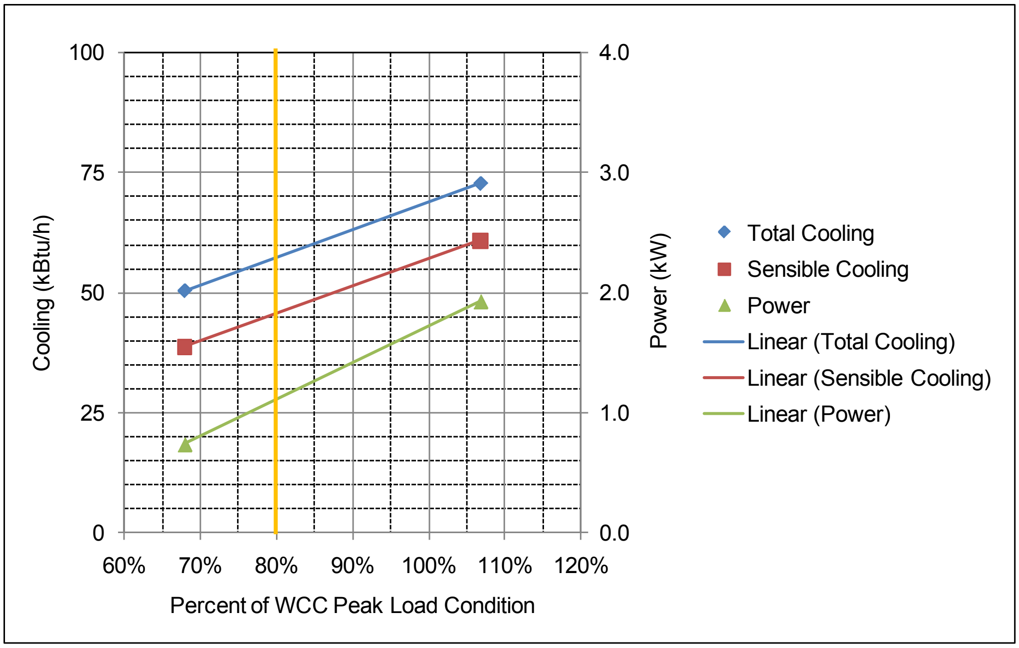

Figure 5. Data used to interpolate surrogate annual EER performance

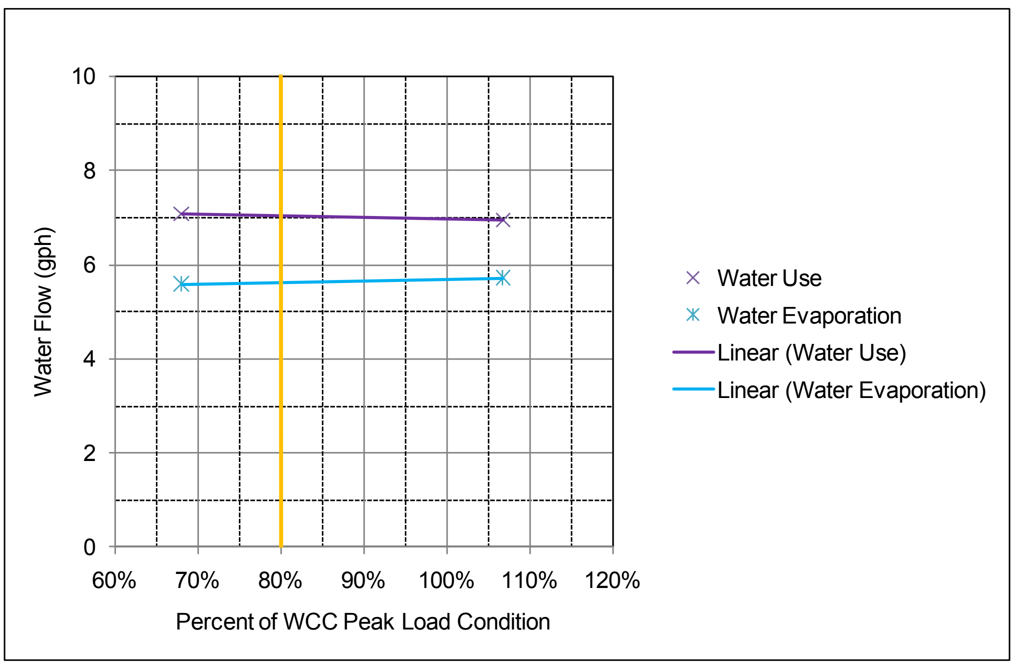

Figure 6. Data used to interpolate surrogate annual water use/evaporation performance

The nominal capacity given by equation (3) for determining credited ventilation rate was calculated to be 60.5 kBtu/h. From this, the credited ventilation rate was then taken to be 600 cfm. The actual ventilation rate was measured to be approximately 800 cfm. The static pressure applied to the unit was 0.7 in. w.c.

The calculated performance of the Coolerado RTU is shown in Table 2. Comparing the calculated performance below to the WCC specifications, the unit meets and exceeds all minimum thermodynamic and water use requirements of the challenge.

Table 2. Western Cooling Challenge Summary

		Specification	Performance	Units
Peak Conditions (105°F/73°F)	Total Cooling	36–360	61.7	kBtu/h
	Sensible Cooling	–	56.9	kBtu/h
	Power	–	2.84	kW
	Total EER	–	21.7	Btu/Wh
	Sensible EER	≥14.0	20.1	Btu/Wh
	Outlet Humidity	≤0.0092	0.00917	kg/kg
	* Water Use	–	1.83	gal/ton·h (sensible)
	Water Evaporation	–	1.50	gal/ ton·h (sensible)
Surrogate Annual Conditions (90°F/64°F)	Total Cooling	–	57.4	kBtu/h
	Sensible Cooling	–	45.6	kBtu/h
	Mean Power	–	1.11	kW
	Total EER	–	51.8	Btu/Wh
	Sensible EER	≥17.0	41.1	Btu/Wh
	* Water Use	≤4.0	1.85	gal/ton·h (sensible)
	Water Evaporation	–	1.48	gal/ton·h (sensible)

* NREL cannot verify through laboratory testing the unit's ability to withstand scaling caused by water evaporation. The measurements are made available in terms of water use and evaporation in the laboratory. Water use will vary in practice because of system adjustments for water quality.

Appendix – Measured Data Tables

	Stage	Air Flow$_{OA\text{-}RA,\ Mixed}$	Air Flow$_{SA}$	Air Flow$_{EA}$	OA Mass Fraction	$T_{OA\text{-}RA\ Mixed}$	T_{SA}	T_{EA}	$\omega_{OA\text{-}RA\ Mixed}$	ω_{SA}
	–	scfm	scfm	scfm	%	°F	°F	°F	grains/lb	grains/lb
Peak Conditions	0	3357	1822	1437	43%	89.6	71.5	76.5	68.8	68.8
	1	3354	1834	1422	42%	89.4	60.5	93.4	68.5	67.9
	2	3542	1810	1624	46%	90.4	58.4	96.8	67.9	64.2
Surrogate Annual Conditions	0	3444	1827	1482	43%	83.2	67.5	72.0	58.4	58.4
	1	3383	1826	1451	43%	83.2	56.7	88.3	58.1	58.6
	2	3591	1806	1660	46%	83.6	54.2	91.1	57.4	55.0

	Stage	Water Use	Water Evaporation	Unit Power	Total Space Cooling	Sensible Space Cooling	Latent Space Cooling
	–	gal/sensible-ton·h	gal/sensible-ton·h	kW	kBtu/h	kBtu/h	kBtu/h
Peak Conditions	0	3.17	2.51	0.75	10.44	13.06	-2.62
	1	1.82	1.43	2.01	33.78	35.45	-1.67
	2	1.83	1.50	2.84	42.28	39.19	3.09
Surrogate Annual Conditions	0	2.19	1.73	0.74	31.28	21.21	10.07
	1	1.37	1.13	1.93	53.32	42.96	10.36
	2	1.40	1.17	2.70	62.17	47.40	14.77

	Stage	Total Ventilation Cooling	Sensible Ventilation Cooling	Latent Ventilation Cooling	Net Total Cooling	Net Sensible Cooling	Net Latent Cooling	Total EER	Sensible EER
	–	kBtu/h	kBtu/h	kBtu/h	kBtu/h	kBtu/h	kBtu/h	Btu/Wh	Btu/Wh
Peak Conditions	0	18.95	17.37	1.57	29.39	30.43	-1.04	39.4	40.8
	1	19.30	17.69	1.61	53.08	53.14	-0.06	26.4	26.4
	2	19.39	17.75	1.63	61.67	56.95	4.72	21.7	20.1
Surrogate Annual Conditions	0	19.15	17.51	1.64	50.43	38.72	11.71	68.1	52.3
	1	19.51	17.84	1.67	72.83	60.80	12.03	37.8	31.5
	2	19.61	17.91	1.69	81.78	65.31	16.47	30.3	24.2

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The Western Cooling Efficiency Center (WCEC) developed a set of criteria for test conditions, minimum energy, and water use performance for prototype cooling equipment and identified these conditions as indicative of western state climates. These criteria, named the Western Cooling Challenge, have been set forth as a challenge to manufacturers to improve the state-of-the-art space cooling products. NREL is to verify these criteria through laboratory testing at its heating, ventilation, and air-conditioning test facility.

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