Proceedings of the ASME 2010 4th International Conference on Energy Sustainability
ES2010
May 17-22, 2010 Phoenix, Arizona, USA

ES2010-90248

COST AND PERFORMANCE ANALYSIS OF A SOLAR THERMAL COOLING PROJECT

Joel K. Dickinson
Salt River Project
Phoenix, AZ, U.S.A.

Robert O. Hess
Salt River Project
Phoenix, AZ, U.S.A.

Jeff Seaton
Arizona Army National Guard
Phoenix, AZ, U.S.A.

Henny van Lambalgen
Quest Energy Group
Tempe, AZ, U.S.A.

Andrea L. Burnham
Quest Energy Group
Tempe, AZ, U.S.A.

ABSTRACT
This paper presents the findings of a field study and performance analysis of a solar thermal cooling system located in Phoenix, Arizona. This system is compared to conventional air conditioning equipment, as well as conventional air conditioning equipment powered by solar electric (photovoltaics). The design of the solar cooling system, which incorporates solar-generated hot water and a single-stage absorption chiller, is discussed. Capital and maintenance cost estimates, including auxiliary electric load and water, are also provided. Operational problems are reviewed together with the design modifications that were required to resolve these issues. Performance of the system and the individual components was determined based on field data collected. Using this data the solar cooler energy savings over conventional air conditioning equipment was determined (Figure 1).

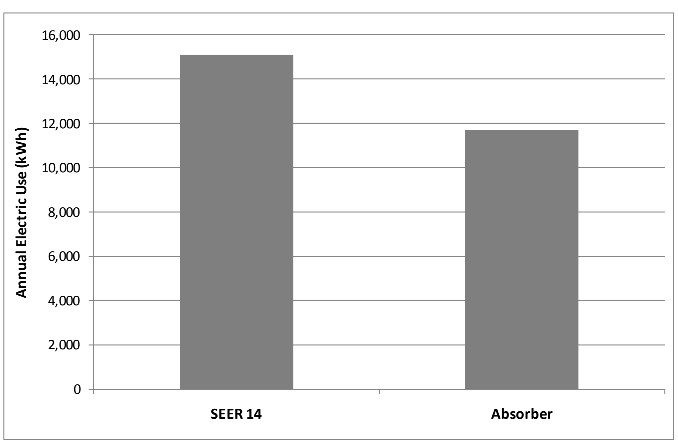

FIGURE 1: MODELED ANNUAL ENERGY USE

FIGURE 2: DEMA SOLAR COOLING SYSTEM

INTRODUCTION
Salt River Project (SRP), a public power utility that serves over 930,000 customers in the Phoenix Arizona metropolitan area, has constructed a 35 kW (10 ton) solar cooling demonstration project at a Department of Defense building site (Figure 2). This project was initiated largely in response to customer interest in solar cooling. SRP was approached for information on the technology and vendors were seeking utility incentives. From an energy provider's perspective, solar cooling is a potential renewable technology to reduce peak demand.

As a result of previous SRP consultant studies, it was determined that solar thermal cooling had potential, but given limited experience with this technology, a demonstration project was needed to better understand design and operating issues. SRP and the State of Arizona Department of Military Affairs (DEMA) agreed to participate in the solar supplemental cooling of a building in Phoenix. The project involves the procurement, installation and evaluation of a solar thermal absorption cooling system that includes a chiller, vacuum tube solar collector array, cooling tower, insulated storage tank, pumps, valves, piping and instrumentation. The project was in operation from April 2007 through August 2009.

NOMENCLATURE

m	=	mass flow (kg/s)
C_p	=	heat capacity (J/kg-K)
ΔT	=	temperature difference (°C)
Q_{array}	=	energy from evacuated tube array (W)
A_{array}	=	evacuated tube absorber plate area (m^2)
I_{rad}	=	solar irradiance (W/m^2)
η_{array}	=	evacuated tube array efficiency (%)
$Q_{heat,abs}$	=	energy to absorption chiller (W)
Q_{losses}	=	energy losses (tank and piping) (W)
COP_{abs}	=	coefficient of performance of absorption chiller
$Q_{cooling,out}$	=	cooling energy (W)
$COP_{overall}$	=	coefficient of performance of overall system
Q_{aux}	=	auxiliary load (W)
Q_{Elec}	=	energy from photovoltaics (W)
$A_{PV,array}$	=	photovoltaic array area (m^2)
$\eta_{PV,array}$	=	photovoltaic array efficiency (%)
$Q_{Elec,usable}$	=	usable energy from photovoltaics (W)
$Q_{Elec,losses}$	=	energy losses (DC-AC conversion) (W)
COP_{Elec}	=	coefficient of performance of unitary equipment
$COP_{Elec,overall}$	=	coefficient of performance of overall photovoltaic system

SYSTEM DESCRIPTION

The major components of the system consist of a 72 square meter (780 square foot) solar array of Sunda Solar vacuum heat pipes with a tilt angle of 15 degrees, a nominal 35 kW (10-ton) Yazaki water-fired single effect lithium-bromide absorption chiller, a 4,900 liter (1,225 gallon) hot water storage tank, and a 77 kW (22-ton) Marley cooling tower. A simplified schematic of the solar cooling system (hot water tank not shown) is illustrated in Figure 3.

The storage tank serves as a capacitor, storing the water until it is sufficiently hot for the chiller. In addition, there are four pumps including a solar array pump, hot water pump (that delivers water from the tank to the chiller), chilled water pump that serves the building load, a cooling tower pump and the absorption chiller has an internal solution pump. Table 1 defines the total electrical power required to operate the system at 3.3 kW. Considering the electrical input only and disregarding thermal input of the chiller operating at its full 35 kW (10-ton) capacity results in a 5.9 COP (0.33 kW/ton).

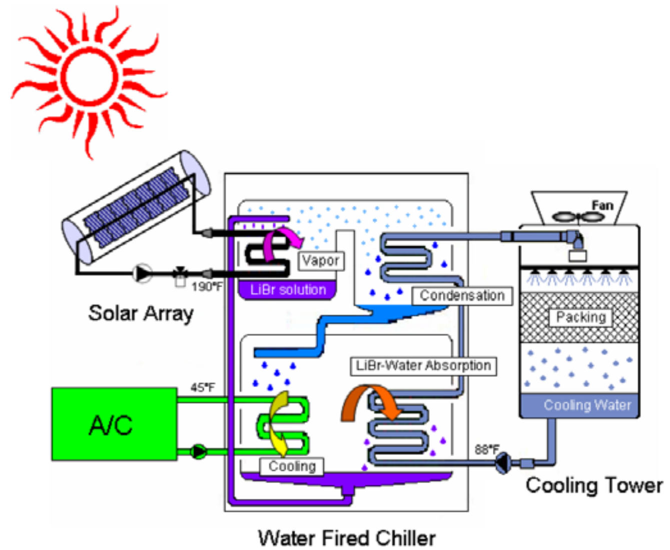

FIGURE 3: SOLAR COOLING SYSTEM SCHEMATIC

Table 1 provides a summary of the major components including power measurements of all electric equipment. It also provides individual component and an overall estimated installed cost for the entire system including labor. Actual cost was somewhat higher than reported due to SRP's requirement of extra measures of safety and instrumentation.

TABLE 1: EQUIPMENT SUMMARY

Description	Manufacturer	Auxiliary Watts	Cost
Solar Array - 416 tubes, 780sf	Sunda Seido1	-	$40,000
Solar Field Pump	Ebara w/VFD	400	$3,500
Hot Water Storage Tank - 4,640 Liter	Hanson	-	$15,000
Hot Water Pump	Ebara	260	$1,500
Cooling Tower Fan	Marley 492A	250	$8,000
Cooling Tower Pump	Pool Pump	1500	$300
Absorption Chiller 10-Ton	Yazaki WFC-SC10	150	$20,000
Chilled Water Pump	Grundfos	750	$1,000
Piping, Valves, Control System			$55,000
Total Auxiliary Watts		**3,310**	
Subtotal Equipment Cost			**$144,300**
Installation and Markup @ 30%			$187,590
Total			**$187,590**

INSTRUMENTATION

The system was instrumented with a Campbell Scientific CR1000 data logger and weather station, matched Kele thermisters, calibrated Sponsler turbine flow meters, and Continental Control Systems WattNode AC power and energy meters. Data recorded included wind speed, ambient temperature, solar insolation, supply and return water temperatures, water flows from each major component and true power measurement from each individual electric component. Data was collected on a one minute interval during the test period and then integrated into hourly performance. An energy balance was performed across the absorption chiller to verify that the instrumentation was properly calibrated, as shown in Figure 4.

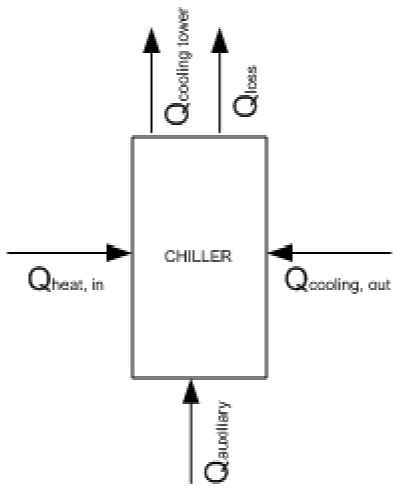

FIGURE 4: CHILLER ENERGY BALANCE

The auxiliary electrical input, expressed as $Q_{auxiliary}$, was measured using energy meters. The energy balance was calculated using the measured energy flow rate for each of the streams around the chiller. Thermal energy flow rate, expressed as Q, was calculated by measuring the fluid flow rate and temperature difference for each stream.

$$Q = mc_p\Delta T \quad (1)$$

Where m is the mass flow rate, c_p is the heat capacity of the heat transfer fluid, and ΔT is the temperature difference between the supply and return.

$$Q_{heat,in} + Q_{cooling,out} + Q_{aux} = Q_{cooling\ tower} + Q_{loss} \quad (2)$$

A stacked line graph showing the measured chiller balance for a typical day is shown in Figure 5. The goal is to verify the accuracy of the data logging equipment by confirming that the sum of the input energy is equal to the heat rejected to the cooling tower; the three solid lines should total the dashed line (note: auxiliary almost overlays the chiller line).

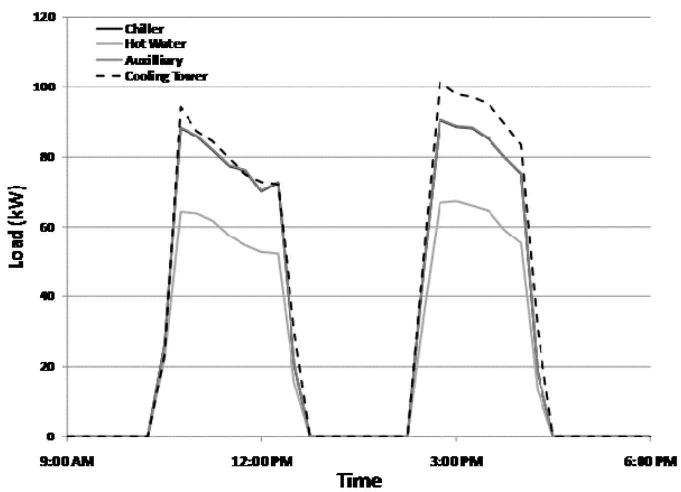

FIGURE 5: MEASURED CHILLER ENERGY BALANCE

OPERATIONAL AND DESIGN ISSUES

The solar cooling system was purchased by SRP as a turn-key project from a vendor. The original project plan was to commission the system in the spring of 2007. Once the system was fully installed and functioning SRP would transfer ownership to DEMA. It quickly became apparent after installation of the system that there were a number of design deficiencies that needed to be addressed. This section details these issues and the design changes that occurred.

The project ran into an environmental compliance issue early in 2007. The cooling tower blow down was routed into a storm drain. SRP designed and installed a pumping station and sewer tap to send cooling tower blow down into the sewer.

Once the cooling tower water issue was resolved in August, SRP attempted to restart the solar cooling system. Unfortunately the chiller would trip off daily as result of high heat medium inlet water temperature. A three-way diverting valve controlled by the building's Energy Management System (EMS) was installed in the heat medium circuit between the hot water storage tank and the chiller.

In September it became apparent that the amount of heat available to drive the chiller was insufficient. The vendor originally designed the solar array to produce 62 kW (210,000 Btu) per hour. Due to frictional head loss in the vacuum tube manifolds and pump selection, the array was only achieving 22 kW (75,000 Btu/hr). SRP redesigned the piping layout to shorten the manifolds and plumbed them in a reverse-return configuration. At this time analog flow meters were installed on the array and the array pump was re-sized.

In December of 2007 the chiller service representative attempted to re-commission the chiller but was unable to do so because the cooling tower water loop was too cold. The chiller's cooling tower control logic was not adequate to maintain the cooling water loop temperature. SRP designed and procured a three-way bypass valve on the cooling tower and a variable frequency drive (VFD) on the cooling tower fan, both controlled by the EMS for cooling water temperature control.

Early in the spring of 2008, the chiller started tripping from high inlet water temperature again. SRP realized that the three way valve on the heat medium supply to the chiller needed to be faster acting and configured as a mixing valve to prevent the overheated water from entering the chiller.

By August of 2008 with improved controls, the solar thermal cooler was running about 4.5 hours a day. Accurately measuring and logging the energy streams of the solar cooler was problematic. As the data acquisition system (DAS) equipment was improved to increase accuracy it became apparent that the sum of the energy around the chiller was not adding up to zero as expected. The DAS still needed work. The chiller manufacturer suggested installing balancing valves on each of the water circuits to ensure proper flow rates. The chiller commissioning contractor discovered a thermo siphon effect in the heat medium circuit keeping the chiller's generator hot even when the chiller was turned off. A test of the specific gravity of the $LiBr$-H_2O solution verified that the chiller had become crystallized.

The solar array pump was designed to shut off when the storage tank reached a high temperature limit of 93˚C (200˚F). In the spring and fall when building demand for cooling was low the array often produced steam due to stagnation. Rather than turn off the array pump, SRP installed a three way valve to a heat rejection coil for high temperature rejection.

By July of 2009 the chiller was still only operating four hours per day. Some of the pumps needed to be replaced to regain the optimal flow rates required by the chiller after the plumbing reconfigurations and wear, SRP was still working with the chiller company to resolve the crystallization issue, and there was more performance to be gained by tuning the EMS. Unfortunately at the end of August the solar array pump failed and SRP elected to discontinue further testing.

THEORETICAL BASIS

The solar collector consists of an array of evacuated tubes. The usable energy collected from the array, Q_{array} is determined by the area of the collector absorber plate, A_{array}, the solar irradiance, I_{rad}, and the array efficiency, η_{array}.

$$Q_{array} = A_{array} I_{rad} \eta_{array} \tag{3}$$

For this project, available heat to the chiller, $Q_{heat,abs}$ is the energy from the array minus distribution losses, Q_{losses} which include pipe and tank losses.

$$Q_{heat,abs} = Q_{array} - Q_{losses} \tag{4}$$

The coefficient of performance (COP) for the absorption chiller, COP_{abs} is determined by the cooling output, $Q_{cooling,out}$, divided by the heat input to the chiller.

$$COP_{abs} = \frac{Q_{cooling,out}}{Q_{heat,abs}} = \frac{Q_{cooling,out}}{I_{rad} A_{array} \eta_{array} - Q_{losses}} \tag{5}$$

Therefore, the overall system efficiency can be determined as follows. The auxiliary loads, Q_{aux} are considered here to be an electrical input and consist of pumps and fans; in addition the denominator includes the energy available from the solar irradiance.

$$COP_{overall} = \frac{Q_{cooling,out}}{I_{rad} A_{array} + Q_{aux}} = \frac{[I_{rad} A_{array} \eta_{array} - Q_{losses}] COP_{abs}}{I_{rad} A_{array} + Q_{aux}} \tag{6}$$

In order to provide a meaningful comparison with a solar electric system, it is assumed that the photovoltaics are used to power conventional air conditioning equipment. The following evaluation shows the conversion of solar electric to solar cooling using an air conditioner assuming a COP of 3 at Air-Conditioning and Refrigeration Institute (ARI) conditions (EER 10).

$$Q_{Elec} = A_{PV,array} I_{rad} \eta_{PV,array} \tag{7}$$

Energy available to the air conditioning unit, $Q_{Elec,usable}$ is the energy from the photovoltaic cells minus losses, $Q_{Elec,losses}$ which include inverter and distribution losses, which are on the order of 20-25%.

$$Q_{Elec,usable} = Q_{Elec} - Q_{Elec,losses} \tag{8}$$

The COP for the photovoltaic powered air conditioning system alone (exclusive of losses from the PV) is determined by the cooling output divided by the usable energy delivered from the photovoltaics.

$$COP_{Elec} = \frac{Q_{cooling,out}}{Q_{Elec,usable}} = \frac{Q_{cooling,out}}{I_{rad} A_{PV,array} \eta_{PV,array} - Q_{Elec,losses}} \tag{9}$$

Therefore, the overall system COP can be determined as follows.

$$COP_{PV,overall} = \frac{[I_{rad} A_{PV,array} \eta_{PV,array} - Q_{PV,losses}] COP_{Elec}}{I_{rad} A_{PV,array}} \tag{10}$$

IN-SITU PERFORMANCE

The actual cooling load output profile of the system on August 24th 2009 is depicted in Figure 6. Due to insufficient heat being generated by the solar array, the chiller shuts down at mid-day until the tank is sufficiently recharged to begin providing heat to the chiller again. Note the chiller output is limited to approximately 21 kW (6 tons) due to the crystallization issues discussed previously.

The dashed line represents the “best case” operation of the system had the solar collectors produced sufficient heat and if the chiller was able to operate at its full rated capacity. This “best-case” profile is used for the annual economic analysis that follows.

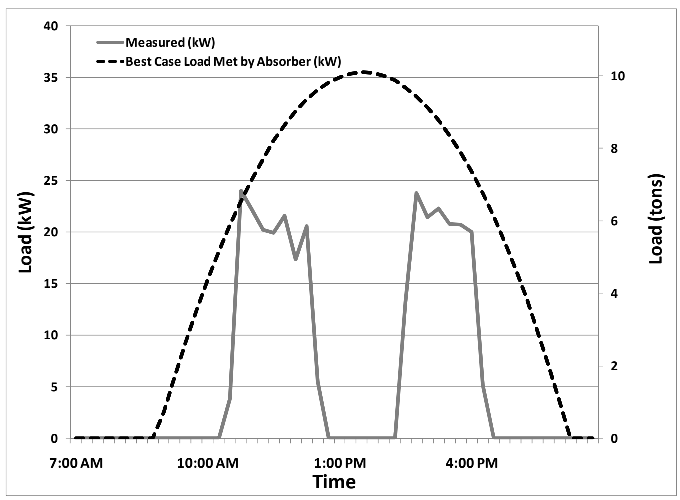

FIGURE 6: MEASURED AND BEST CASE COOLING LOAD PROFILES, 08/24/09

The solar array efficiency (heat output from the collector divided by incident solar radiation) was measured to be approximately 41% during the test period. The COP of the chiller was measured to be 0.40 (due to the crystallization issues) as compared to the rated 0.70 by the manufacturer. The resultant overall system COP, cooling output divided by incident solar radiation is approximately 0.15.

TABLE 2: SYSTEM PERFORMANCE

	Array Efficiency	Cooling COP	Overall System COP
Actual Solar Absorber	41.5%	0.4	0.15
Best Case Solar Absorber	41.5%	0.7	0.28
PV to Unitary Equipment	15.0%	3.0	0.38

If the chiller operated at its rated performance, COP of 0.70, this would have increased the overall system COP to 0.28. For comparison purposes, the overall efficiency of a photovoltaic powered air conditioning system, cooling output divided by incident solar radiation, is on the order of 0.38.

ANALYSIS METHODOLOGY

It is clear from our testing that performance is significantly lower than an optimally designed and operating solar cooling system. To properly assess the potential of a solar cooling system in Arizona, the following economic analysis assumes a "best case" operation. The "best case" assumes sufficient heat from the solar array, proper flow, system controls, and an absorption chiller able to achieve its full rated 35 kW (10-ton) capacity.

In addition, a minimum 35 kW (10 ton) base cooling load was assumed to exist 7 days a week for the entire year to maximize the utilization of the system. The dashed line superimposed over the actual system performance in Figure 6 reflects this operation. In total, an optimized system is capable of producing approximately 250 kWh (70 ton-hours) per day in the peak summer months. This value will be reduced somewhat in the winter months due to reduced solar insolation. The Maui Solar Energy Software Corporation's Solar Design Studio software program was used to come up with monthly adjustments to the profile in terms of monthly heat output for the array. These values ranged from 61.5% to 100%. Figure 6 depicts the load profile after adjusting for the reduced output providing an estimated total annual cooling load of 59,400 kWh (16,900 ton-hrs).

The economics of the solar cooling system was compared to conventional air-cooled equipment with a SEER rating of 14 and a conventional air conditioning system powered by solar electric (photovoltaic). Figure 7 provides a nominal comparison of the COP of the conventional packaged equipment at 35˚C (95˚F) and the solar cooling system. Note the solar cooling COP is for the electric use only, not thermal. A brief discussion of each alternative follows.

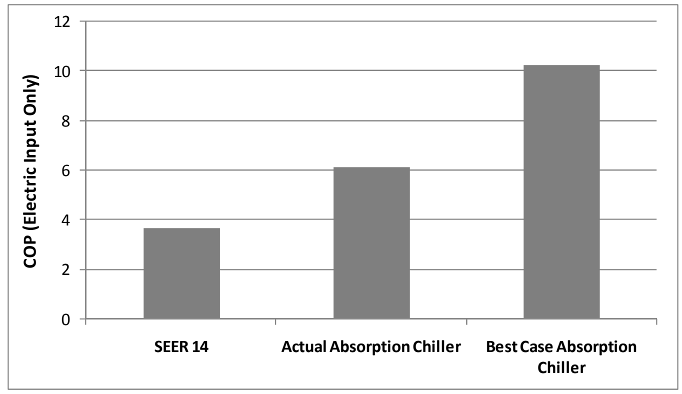

FIGURE 7: HVAC EQUIPMENT COMPARISON OF ELECTRIC USAGE

Unitary Air Conditioning is the simplest and lowest first cost system. It is the most common system for small commercial and residential building sectors. Energy use consists of a constant volume indoor fan, and compressor/outdoor condenser fans that cycle with load. The performance of these systems is dependent on the ambient air temperature with the ARI rated condition of 35˚C (95˚F). At temperatures above 35˚C (95˚F) the system capacity and performance is reduced. This is important since ambient daytime temperatures in Arizona can be well over 38˚C (100˚F) during the summer months. Figure 8 illustrates the efficiency of the SEER 14 unit as a function of outdoor temperature.

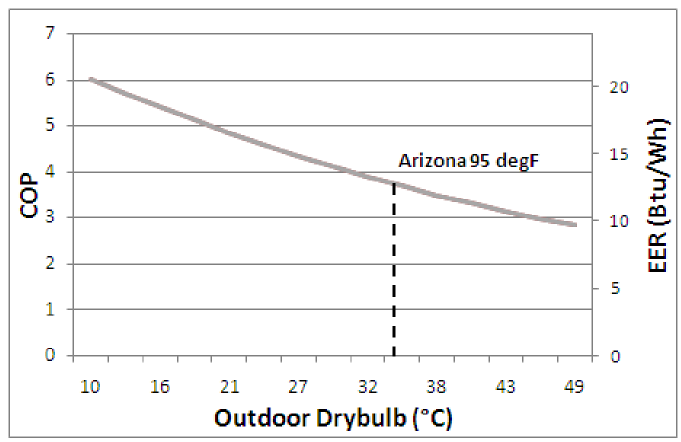

FIGURE 8: UNITARY EFFICIENCY VERSES AMBIENT TEMPERATURE

For the solar cooling system, the energy provided by the sun is "free", but additional energy input is required in the form of purchased electricity. There are a total of four pumps, an absorption chiller solution pump and a cooling tower fan required to transfer, store and convert the sun's heat to useful cooling. The total power, total electrical load, for the installed system, when fully loaded, is 3.3 kW or 10.7 COP (0.33 kW/ton). It is worth noting that if the chiller is running at less than full capacity as in the case for our installation, the power (kW) consumption of the solar cooling system does not change. At 21 kW (6 tons) of cooling output the system performance is 6.4 COP (0.55 kW/ton); approximately the same as an electric water-cooled chiller.

Water consumption is another key component to consider in assessing the economics of these systems. For the air-cooled unitary equipment there is no local water use associated with the cooling. For the solar cooling system, water is used by the cooling tower via evaporation to reject heat, as well as blow down to ensure the dissolved solids in the tower are maintained at a manageable level.

Figure 9 illustrates the relative water use for each system type. Note that with the solar cooling system, the total heat load rejected to the cooling tower encompasses both solar heat input and the building load.

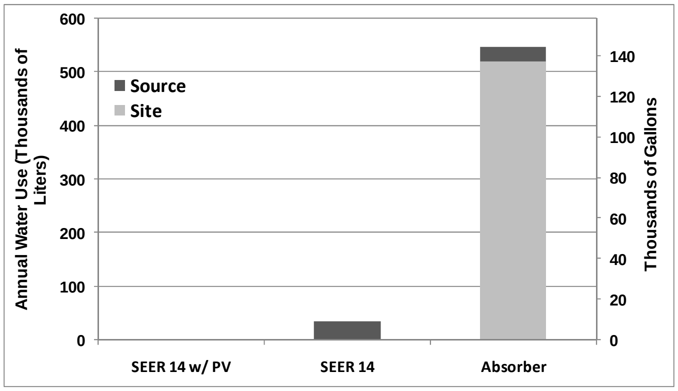

FIGURE 9: ESTIMATED ANNUAL WATER USE

The generation of electricity requires water use at the power plant. While there is no direct utility cost associated with this water consumption (it is encompassed as part of the electric tariff), from an environmental viewpoint all systems including the air-cooled unitary equipment use water. To this end Figure 9 also includes "source" water use for each system. To estimate this water use, 2.27 L/Wh (0.6 gal/Wh) was used to determine the source water usage for coal fired electricity production.

Solar Cooling versus Photovoltaic

As a final comparison it is meaningful to compare the economics of the solar cooling system to a photovoltaic system as a renewable technology. A 9.5 kW photovoltaic system is assumed to meet the load of a SEER 14 air conditioning system. A performance comparison is found in Table 2 and an energy and cost comparison is provided in Table 3.

RESULTS

A detailed economic analysis was done for each of the alternatives. The economic analysis encompasses first cost, electric and water utility costs.

For the cooling systems, it was assumed that a cooling load exists regardless of the source of input energy; thus no first cost was assumed for either case utilizing conventional air conditioning methods. First cost for the solar cooling system is greater than $18,000 per ton (see Table 1).

An installed cost of $6.50 per watt-DC was used for the photovoltaic system alternative. Based on the annual cooling load, a 9.5 kW system is sized. Note that for the solar cooling and photovoltaic systems, a direct analysis is performed with no tax benefits being applied.

Electric utility cost savings were based on a typical small commercial rate. The average cost for electricity was 0.10 per kWh year round. Water and sewer charges are based on local charges and are assumed to be $2.50 per 2,800 liters (100 cubic feet).

To simplify the analysis water treatment for the cooling tower was assumed to be included as part of the existing contract on the site. As such it has been assumed that no additional maintenance costs will be added for the solar cooling system. Based on SRP's experience, this is likely very optimistic.

Table 1 provides a comparison of the solar cooling system to both a unitary SEER 14 air conditioning and a SEER 14 air conditioning unit powered by photovoltaic. The absorption chiller saves approximately 3,420 kWh as compared to the conventional SEER 14 unit. A SEER 14 unit powered by photovoltaics theoretically requires no input energy to operate. When the cost of water is added the absorption chiller is actually more expensive to operate.

CONCLUSIONS

As a result of the extensive field testing and analysis, a number of conclusions can be drawn from this study. The most salient of these are summarized below.

Solar cooling systems of this size are not currently economically viable due to the high first cost compared to conventional air conditioning. Even if the system had zero electric usage, the amount of potential cooling savings would not justify the initial cost.

Solar cooling is not free. Auxiliary power for pumps and fans represent a significant electric load. The energy savings of a system of this size is not substantial, even at full load performance. If the chiller is operated under part load conditions, these auxiliaries represent the same electrical input but provide a lower cooling output, reducing the annual energy savings relative to conventional air conditioning.

Absorption cooling systems are relatively complex and can be damaged by improper operation. Absorption chillers require consistent operating temperature for the both heat source and cooling tower condenser water to ensure reliable operation. They also require consistent loads for ideal operation. Single stage absorption chillers require a significant amount of water for heat rejection. While these water costs are typically less than electric utility costs necessary to operate the chiller, they are significant.

From a renewable energy perspective, photovoltaics can achieve the same annual savings for significantly less cost. Photovoltaic systems are also much simpler and do not require on-going maintenance and the water treatment required by the absorption chiller.

As demonstrated by this study, a significant annual operating cost savings would be required to offset the capital cost of the solar cooling equipment. Water saving cooling water options and lower first cost of equipment would be required to encourage widespread adoption.

TABLE 3: THEORETICAL SOLAR COOLING VS. CONVENTIONAL AC how about add "annual" to savings v conventional ac

Cooling Alternative	Efficiency	First Cost	Annual Cooling		Purchased Annual Electric		Annual Water		Total
		($)	kWh	ton-hrs	kWh	$	kLiter	$	$
Conventional AC	SEER 14	$ -	59,400	16,900	15,130	$ 1,050	-	$ -	$1,050
Solar Electric (PV) w/ Conventional AC	SEER 14	$ 68,700	59,400	16,900	-	$ -	-	$ -	$ -
Solar Thermal w/ Absorber	3.3 kW (Aux)	$ 187,600	59,400	16,900	11,710	$ 800	521	$ 540	$1,340
Savings vs. Conventional AC									
Solar Electric (PV) w/ Conventional AC	SEER 14	$ (68,700)	-	-	15,130	$ 1,050	-	$ -	$1,050
Solar Thermal w/ Absorber	SEER 14	$ (187,600)	-	-	3,420	$ 250	(521)	$(540)	$ (290)

REFERENCES

[1] National Renewable Energy Laboratory, 4/2/2010. <http://www.nrel.gov/rredc/pvwatts/changing_parameters.html>

[2] Mittal V., Kasana K. S., Thakur N. S. (2006). "Modelling and simulation of a solar absorption cooling system for India", *Journal of Energy in Southern Africa*, **17**, 3, pp.65-70.

[3] Mittal V., Kasana K. S., Thakur N. S. (2005). "Performance evaluation of solar absorption cooling system of Bahal (Haryana)", *J. Indian Inst. Sci.*, Sep.-Oct. 2005, **85**, pp.295-305.

[4] Zidianaki G., Tsoutsos Th., Zografakis N. (2007). "Simulation of a solar absorption cooling system", 2nd PALENC Conference and 18th AIVC Conference on Building Low Energy Cooling and Advanced Ventilation Technologies in the 21st Century, Crete island, Greece, 2, pp. 1187-1194.

[5] Henning H. M. (2004). *Solar-Assisted Air-Conditioning in Buildings*. SpringerWienNewYork, Freiburg, Germany.