Pacific Gas and Electric Company

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Application Assessment Report #0402

Laboratory Evaluation of the Coolerado Cooler™ Indirect Evaporative Cooling Unit

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EXECUTIVE SUMMARY

Testing was conducted to evaluate the performance of an early-2005 model Coolerado Cooler™ indirect evaporative cooling unit designed for the residential and small commercial market. The goal was to assess the performance of the Coolerado Cooler™ for consideration in PG&E's rebate program for evaporative coolers.

A test plan was developed based on ASHRAE test standards for evaporative coolers, which are primarily focused on the arrangement of the test apparatus and determining the supply airflow. A test condition matrix was established through research into the cooling design conditions for various locations in the PG&E service territory. Additional testing was done to see the effect of varying supply voltage, and also fan speed with the addition of an optional speed controller.

The advantage of an indirect evaporative cooler is that it accomplishes low-cost evaporative cooling without adding any moisture to the air supplied to the conditioned space. However, the high flow resistance of the heat and mass exchange modules in this unit resulted in significantly lower supplied airflow and higher power consumption relative to typical direct evaporative coolers. Still, it produced conditions that would keep a space within the ASHRAE comfort zone over a wider range of outdoor conditions than would those other systems.

The wet-bulb effectiveness over the test conditions ranged from 81% to 91% (averaging 86%), although it is theoretically capable of achieving a wet-bulb effectiveness greater than 100%. The average test results for this unit are included in *Table 1*. (For a more thorough description of the table contents, refer to the description of *Table 4*, of which this is a subset.)

Table 1: Average Unit Performance

Supply Airflow[1] (cfm)	**1,500**
Exhaust Airflow[1] (cfm)	**1,320**
Total Unit Power (W)	**1,329**
Effectiveness	**86.0%**
CEC T20 ECER	**10.4**
Approximate Cost	**$2,900**

[1] Measured outlet airflow referenced to the intake density.

INTRODUCTION

Background

The use of central air conditioners in the residential sector has been increasing significantly in recent years, in retrofit applications as well as in new construction where air conditioning is often seen as a standard feature. This puts an increasing strain on California's generation, transmission, and distribution infrastructure to handle the demand for electricity. Evaporative cooling technologies offer an alternative to conventional air conditioners in hot, dry climates, and can provide some level of cooling for a fraction of the energy consumption. Because of this, PG&E promotes evaporative cooling technologies through rebates and information and education programs.

The biggest drawback to the acceptance of traditional evaporative coolers is that they exchange decreased temperature for increased humidity. An evaporatively cooled space can feel uncomfortable because the increased humidity can impair the body's ability to cool itself through perspiration. A solution is to use direct evaporatively cooled air to cool another stream of air using an air-to-air heat exchanger, which is then supplied to the space. This "indirect" evaporative cooling takes advantage of the inexpensive cooling done through evaporation without the increase in humidity in the conditioned space. Indirect, or combined indirect/direct or two-stage systems, are more complex and costly than simple direct evaporative coolers, but their advantages in interior comfort may make them more attractive to consumers. Rebates may need to be increased for these technologies to offset their higher costs if they prove to have a significant efficiency and comfort advantage over simple direct systems.

The Coolerado Cooler™, manufactured by Idalex, is a new indirect evaporative cooling system being marketed to the residential and small commercial sector. The system is built around a heat and mass exchange process developed by Dr. Valeriy Maisotsenko, which was developed for many cooling applications in dry climates, from space cooling to combustion turbine intake air cooling. While most evaporative cooling technologies are limited by the wet-bulb temperature of the entering air, this process is actually limited by the dew point temperature, which creates the potential for lower delivered air temperatures in addition to no moisture gain.

Prior Research

PG&E's Technical and Ecological Services (TES) has done extensive evaluations of various air conditioning technologies, including advanced evaporative cooling systems. The first tests on evaporative coolers were done in the summer of 1993, and included six sample systems available at the time. Additional testing was done in 1998 on a prototype combined indirect/direct cooler to assist with its development. Other tests done at TES have involved small commercial and residential air conditioning systems, including some using evaporatively cooled condensers. This project builds upon more recent PG&E Emerging Technologies Program Application Assessments of different evaporative cooler technologies in 2003 and 2004, which produced two reports (References 8 and 9).

Objectives

The objective of this project was to assess the performance of this evaporative cooling unit (ECU), as defined by:

- airflow,
- evaporation (or wet-bulb) effectiveness,
- power demand,
- cooling capacity and efficiency,

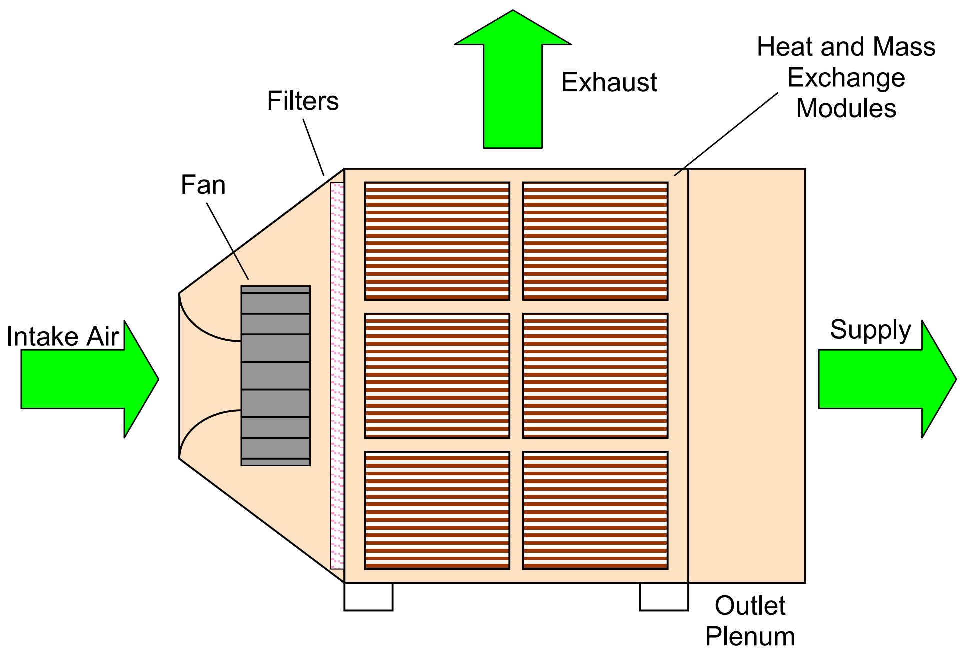
as a function of the variables:

- intake air temperature and humidity,
- external resistance to flow
- fan speed, and
- line voltage.

System Description
The test system is a recent application of the Coolerado's proprietary indirect evaporative cooling process to a unit applicable for residential or small commercial service. As such, this system is still subject to further development and may not represent their future retail product. It is still essentially custom built, so it does not yet have the economic advantages of mass production and is thus considerably more expensive than other evaporative systems. They are currently working on a contract with Delphi Corporation to mass produce their heat and mass transfer modules, which will help reduce the system cost.

Figure 1 shows a cross section of the Coolerado Cooler™ and the locations of key system components. Outside air is drawn in by a single backward-inclined, centrifugal fan, which forces air through filters and into the heat and mass exchange modules where it is split into two streams (supply and exhaust). There is no water pump in the system as the water flow is a once-through process with no recirculation. The water supplied to the modules is controlled by a solenoid valve on the water supply line, which is operated based on the flow of water draining from the system (too little and the valve opens, too much and the valve closes). Thus, the system has no pan of standing water that could become stagnant or grow algae.

Figure 1: Coolerado Cooler™ System

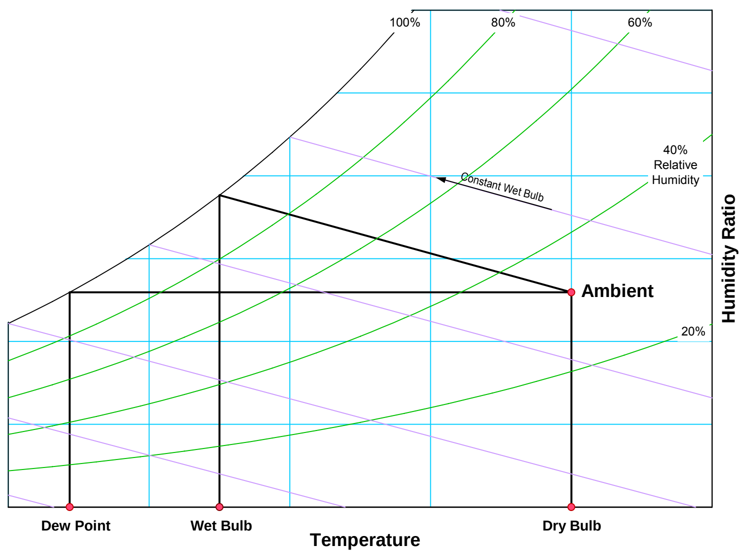


EXPERIMENTAL DESIGN AND PROCEDURE

Process Description and Performance Characteristics
Performance data are required in order to document the ability of evaporative systems to maintain comfort under various conditions. The data collected are intended to provide enough information to adequately model their performance, and thus to perform further analysis to determine the annual energy usage and peak demand for different climates. The results may be disseminated through Emerging Technologies program information transfer activities, and may be used to develop marketing materials for future rebate or incentive programs.

The performance of an evaporative cooler is best described using a psychrometric chart, which displays moisture content (humidity ratio in mass of water vapor per mass of dry air) against temperature. *Figure 2* shows a simplified psychrometric chart with some of the basic concepts and terms identified. When dry air is exposed to liquid water, some of the heat contained in the air will be absorbed through the evaporation of the water, causing a decrease in the air temperature. (Hot, dry air is converted to cool, humid air.) If continued long enough, air will become saturated with water vapor (100% relative humidity), and reach what is called its "wet-bulb" temperature. This term comes from the measurement method of wrapping the bulb of a thermometer in moistened fabric, and then blowing air across it. To avoid confusion, the actual air temperature is normally referred to as the "dry-bulb" temperature. This evaporative cooling process is shown in the chart as a diagonal line of decreasing temperature and increasing humidity ratio. If air is cooled "sensibly" (without a change in moisture content), the conditions of the air in the chart move along a horizontal line of constant humidity ratio until it again reaches 100% relative humidity. The temperature at this point is called the "dew point" temperature.

Figure 2: Psychrometric Chart



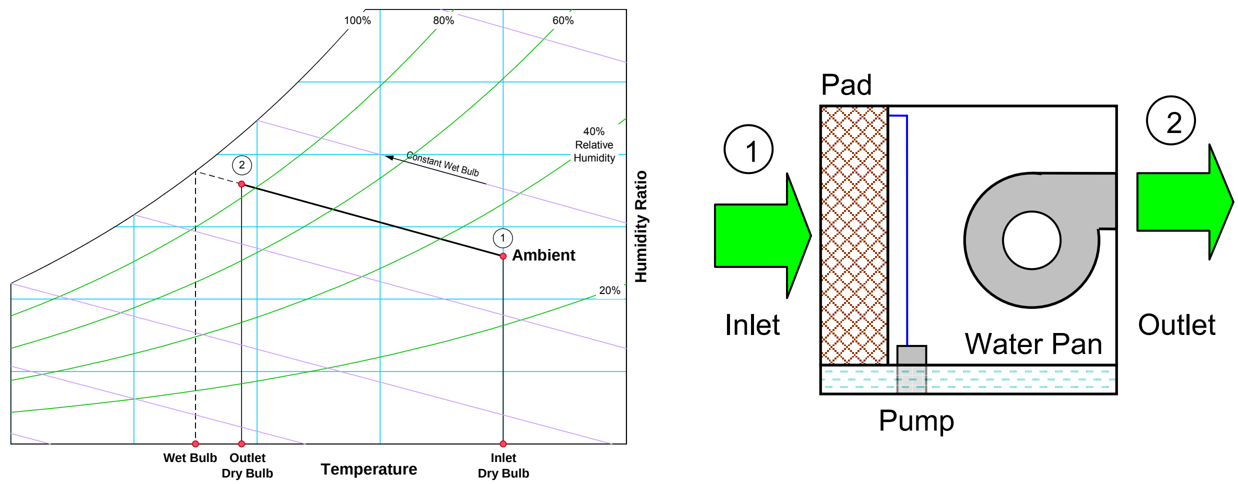
Direct Evaporative Cooling Process

In a direct evaporative cooler, the supply air is cooled by exposing it directly to liquid water. Hot, dry air is converted to cool, moist air, or "sensible" heat is converted to "latent" heat (water vapor). Direct evaporative coolers may be described as a constant wet-bulb temperature process, although there is some minor sensible heat gained from the fan. Their performance is related to how close the dry-bulb temperature of the supply air approaches the wet-bulb temperature of the intake air. The wet-bulb "effectiveness" of an evaporative cooler is defined as follows:

$$\text{Effectiveness}\,(\varepsilon) = \left(\frac{T_{db,in} - T_{db,out}}{T_{db,in} - T_{wb,in}}\right) \times 100\% \qquad \text{(Equation 1)}$$

where Tdb_{in} and Twb_{in} are the intake dry and wet-bulb temperatures, respectively, and Tdb_{out} is the dry-bulb temperature at the air outlet. The effectiveness can also be described as the ratio of the actual sensible cooling done to the air to its wet bulb depression. *Figure 3* shows the process for an 85%-effectiveness direct evaporative cooler on a psychrometric chart, along with the three temperatures used in the effectiveness calculation.

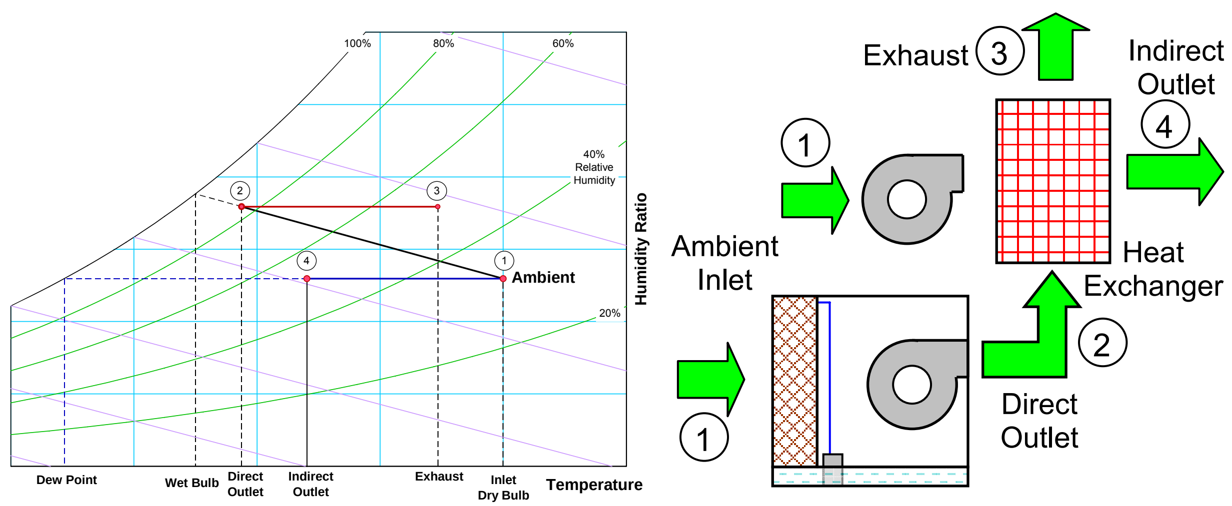
Figure 3: Direct Evaporative Cooler Process



Indirect Evaporative Cooling Process

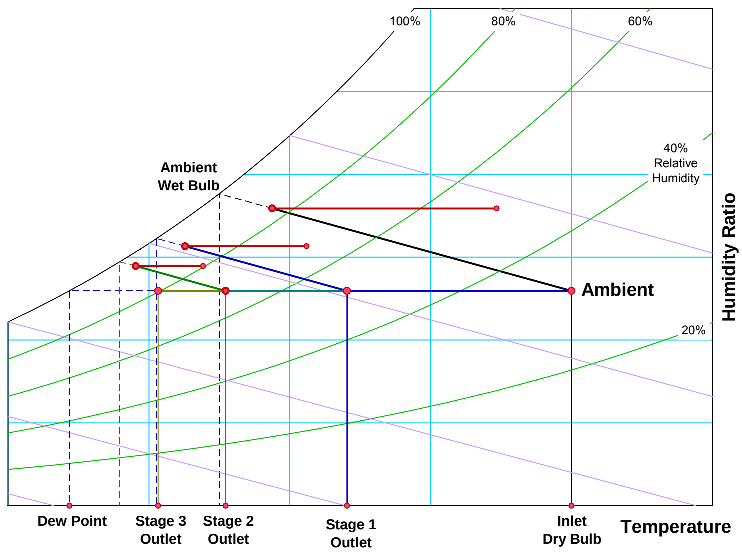
Evaporatively cooled air can be used with an air-to-air heat exchanger to sensibly cool a second stream of air without changing its moisture content, thus creating an "indirect" evaporative cooler. Due to heat exchange inefficiencies, the temperature of the delivered air will be higher than that provided by the direct evaporative cooler. A simplified version of this process is shown in *Figure 4*. The resulting supply and exhaust temperatures will depend on the effectiveness of the heat exchanger and the relative magnitude of the two airflows. While this example shows the source of the supply air being from the outside, this system can also be applied to indirectly cool return air from a space so long as the outlet from the direct stage is cooler than the return air.

Figure 4: Indirect Evaporative Cooler Process



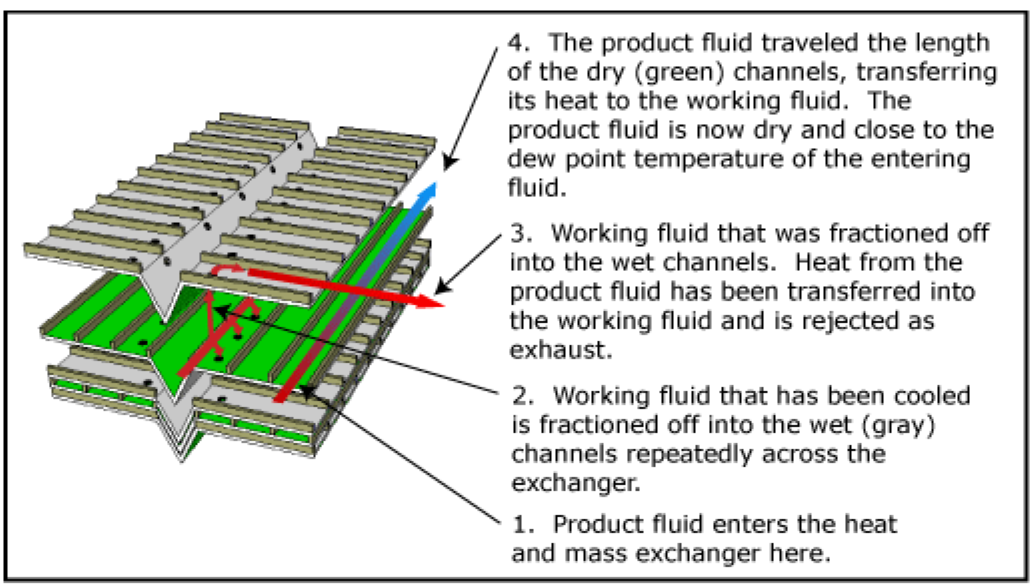
One upside of this process is that the sensible cooling done through the heat exchanger not only reduces the dry-bulb temperature of the air, but also its wet-bulb temperature. This means that the process can be repeated using the sensibly cooled air as the inlet to another stage of direct or indirect evaporative cooling (Point 4 becomes Point 1 for the second stage). *Figure 5* shows an example of three stages of indirect evaporative cooling, with diminishing temperature drops with each stage since wet-bulb depression decreases as the air is sensibly cooled. In this example, the final supply temperature is actually below the entering air wet-bulb temperature, resulting in a wet-bulb effectiveness greater than 100%. With each stage, the process approaches the ultimately limiting value of the dew point temperature; so, indirect evaporative coolers may be better compared based on a dew-point effectiveness.

Figure 5: Staged Indirect Evaporative Cooler Process



This example is a simplified version of the process that takes place in the Coolerado Cooler™. Instead of distinct stages, the heat and mass exchange modules used inside this system consist of hundreds of channels that split off portions of the supply air stream for evaporative cooling and heat exchange. *Figure 6* is an image from the Coolerado Cooler™ web site (www.coolerado.com) that shows a detailed diagram of the heat and mass exchanger. The other difference with this process and that described in *Figure 5* is that the exhaust air continues to evaporate water while absorbing heat from the air that is ultimately supplied to the space, and the resulting mixed exhaust is saturated.

Figure 6: Cut-away Diagram of a Heat and Mass Exchange Module



Source: www.coolerado.com

Definition of Cooling Capacity

Ideally, the performance numbers obtained from the testing can be used to compare the performance against alternative cooling systems, including direct or indirect evaporative coolers and vapor compression air conditioners. However, the comparison between an evaporative system and a conventional vapor-compression air conditioner is not very straightforward. Conventional air conditioners are rated in terms of their cooling capacity (Btu/hr or tons) and efficiency (capacity divided by power consumption, given as its energy efficiency ratio or "EER"). Evaporative coolers are normally only rated in terms of airflow. The determination of a capacity for an evaporative cooler is open to some debate.

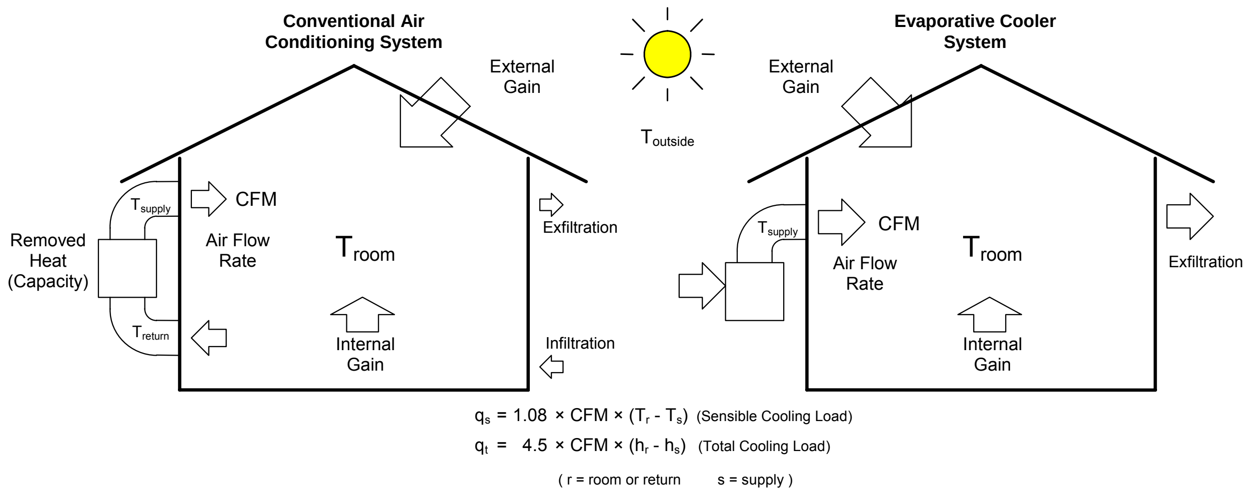
A conventional air conditioning system is designed to condition the air contained in a space, reducing the temperature (sensible heat) and moisture (latent heat) gained from various sources, while recirculating the same air repeatedly. The cooling capacity is measured at the evaporator coil as the product of the air mass flow rate across the coil and the enthalpy decrease between the return air from the conditioned space and the discharged supply air. (Enthalpy is a measure of the relative energy content of the air/water vapor mixture. A constant wet-bulb temperature process like a direct evaporative cooler is close to a constant enthalpy process.)

In contrast, an evaporative cooler is a once-through, displacement system. It pushes 100% outside air into a space, and the same amount must be exhausted back outside. Evaporative systems usually supply air at a higher temperature than a conventional air conditioner, so they need a much higher airflow rate to provide adequate cooling. Additionally, the higher air velocities can make air feel cooler than air at rest. The high flow also means that evaporative coolers cannot normally be connected to a duct system sized for the velocities provided by a conventional air conditioner or furnace. A fan may still need to be operated to assure adequate circulation throughout a building if the evaporative cooler supply is at only one location.

The thermal load in a space served by an evaporative cooler should be less than the thermal load in the same space if served by an air conditioner. Since an evaporative cooler keeps the space at a positive pressure, there is no thermal gain from infiltration. Also, if the exhaust air is vented out through the attic rather than through open windows, it will lower the temperature in the attic and reduce the heat gain to the living space through the ceiling. There is also no latent load in an evaporatively cooled space because any moisture generated within the space is exhausted and does not need to be condensed out of the air.

A graphical description of the difference between the two types of systems and the definition of cooling capacity is shown in *Figure 7*:

Figure 7: Definition of Cooling Capacity



For this and the previous PG&E Emerging Technology Application Assessment reports, the cooling capacity of an evaporative cooler is defined as:

$$\textit{Room Capacity (Btu/hr)} \approx 1.08 \times CFM \times (Tdb_{room} - Tdb_{supply}) \qquad \text{(Equation 2)}$$

where 1.08 is a units conversion factor combining standard air density and specific heat (0.075 lb/ft³ × 0.24 Btu/lb-°F × 60 min/hr), CFM is the flow rate of air through the unit in cubic feet per minute, Tdb_{supply} is the discharge dry-bulb temperature of the test unit, and Tdb_{room} is an assumed indoor space condition in °F. The selected temperature is 80°F, which was chosen since it is what is used for return air in the ARI test standards for rating conventional air conditioning systems (Reference 6). This definition means that if a system is unable to achieve a supply temperature less than 80°F, then its capacity will be negative, and the space will settle out at a higher temperature than 80°F. A test standard from Australia (Reference 7) lists a similar formula for capacity, but defines the interior space condition at 81.3°F (27.4°C). Once a cooling capacity is determined, an energy efficiency ratio (EER) is then determined by dividing it by the power consumption.

An evaporative cooler rating parameter has been recently developed by the California Energy Commission (CEC) for its Appliance Efficiency Regulations (Title-20). Their Evaporative Cooler Efficiency Ratio (ECER) uses a slightly modified version of the above equation for capacity (Equation 2), which substitutes in the equation for effectiveness (Equation 1) solved for the supply air temperature, as follows:

$$ECER = 1.08 \times CFM \times (Tdb_{room} - (T_{db,\,in} - \varepsilon \times (T_{db,\,in} - T_{wb,\,in}))) \,/\, W \qquad \text{(Equation 3)}$$

The effectiveness (ε), power (W), and airflow (CFM) are measured with an external static pressure of 0.3 inches of water, and the ECER is then calculated at standard rating temperatures of $T_{db,\,in}$ = 91°F, $T_{wb,\,in}$ = 69°F, and Tdb_{room} = 80°F (which is the same as what was chosen for this analysis before the CEC's method was published). This parameter only looks at the sensible cooling done by an evaporative cooler, and does not reflect the increased comfort provided by indirect systems through not adding moisture to the supply air. Thus, this parameter should only be used to compare like-systems (e.g. direct to direct).

An alternative measure of capacity is defined in ASHRAE Standard 143 (Reference 5), which uses the same basic equation, but uses the intake dry-bulb temperature in place of the assumed room temperature. This is because an indirect evaporative cooler could use return air as the intake to the indirect cooling section and outside air as the intake to the evaporative section, although the Coolerado Cooler™ uses outside air for both. Both of these measures of capacity will be included in the analysis, with this second measure described as the sensible cooling of intake air, or intake air capacity.

$$IA\ Capacity\ (Btu/hr) \approx 1.08 \times CFM \times (Tdb_{intake} - Tdb_{supply}) \qquad \text{(Equation 4)}$$

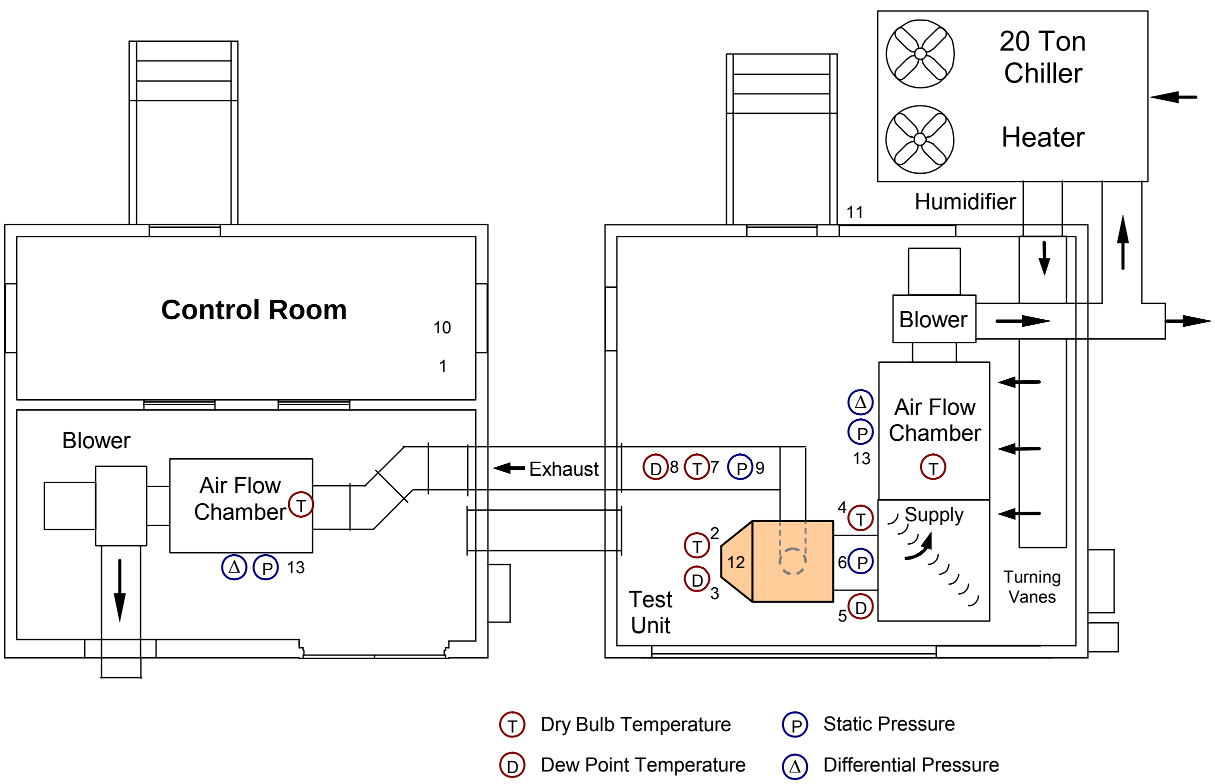
(The capacity parameters listed in Equations 2 and 4 are shown as approximations due to the nominal values of density and specific heat that produce the 1.08. For the reported results calculations, measured values of density and specific heat are used; except for the ECER values, which use Equation 3 directly.)

Test Facility

Figure 8 shows a layout of the test facility configured for the Coolerado Cooler™ testing. The test unit is placed in an environmentally-controlled room, which is conditioned by a 20-ton heat pump / air conditioner, a variable-output resistance heater, and a humidifier. Outside air dampers allow for some recirculation and mixing with outside air to control supplied air temperature and humidity. Both the supply air and exhaust streams of the test unit were connected to separate airflow measurement stations, each consisting of a sealed chamber with several flow nozzles, designed in accordance with ASHRAE specifications (per References 3 and 4). The chambers consist of a square tunnel, with flow conditioning screens at the entrance and exit and a partition in the middle having four flow nozzles. The chamber in the room with the test unit was used for the supply airflow. It has four 9" nozzles, and can measure flow rates between 1,300 and 12,400 cfm. The chamber located in the other building was used to measure the exhaust airflow. That chamber has 8", 6", and two 4" nozzles, and is capable of measuring flow rates between 260 and 5,000 cfm. A variable-speed blower on the outlet of each chamber is set to maintain the desired outlet static pressure and compensate for the added resistance of the measurement system and ductwork.

Figure 8: Test Facility and Measurement Locations

(The numbers correspond to the locations of the instruments described in the next section)



Measurements and Instrumentation
The test set-up followed the guidelines described in the ASHRAE indirect evaporator cooler test standard (Reference 5). The following is a listing of the measurements taken and the instruments used for the testing:

1. Barometric pressure, using an electronic barometer.
2. Entering air dry-bulb temperature, using four resistance temperature devices (RTDs).
3. Entering air dew-point temperature, using a chilled mirror sensor.
4. Supply air dry-bulb temperature, using four RTDs inserted through the duct wall.
5. Supply air dew-point temperature, using a chilled mirror sensor and a sampling tube.
6. Supply static pressure, using a low-range static pressure transmitter.

   Four taps were made in the outlet duct at a distance downstream equal to the average of the duct height and width, and at the middle of each duct face. The taps were connected together with a ring of tubing and tees, with an additional tee leading to the transmitter.
7. Exhaust air dry-bulb temperature, using four RTDs inserted through the duct wall.
8. Exhaust air dew-point temperature, using a chilled mirror sensor and a sampling tube.
9. Exhaust static pressure, using a low-range static pressure transmitter.
10. Total power, using a true-RMS power meter.
11. Make-up water flow rate, using a positive displacement flow meter with a pulse output.
12. Fan speed, using an optical tachometer.
13. Airflow rates, using a nozzle chamber and measurements of differential and inlet static pressure and inlet temperature.

All of the temperature instruments were calibrated simultaneously against a laboratory standard prior to the tests. The calibration included a low point using an ice bath (32°F), and a high point using a hot water bath (~120°F). The raw measurements were adjusted to match the reading from a secondary temperature standard RTD placed in the same bath. The transmitters for the differential and static pressure measurements were calibrated using a water manometer with a micrometer adjustment, accurate to 0.01 inch of water.

Data Acquisition System
The instruments were connected through several data acquisition devices to a central personal computer. The pressure transmitters, power transducer, and water flow meters were all connected to a high-speed data acquisition system from National Instruments (NI). The NI system used a PCI-bus data acquisition card to transfer the measurements to the computer. Digital and analog feedback control signals for the room conditioning systems and airflow chamber booster fans were also provided by the NI system. The RTDs were all connected to a Fluke Helios data logger, and total power measurements were made with a Yokogawa power meter. The data logger, power meter, three dew point temperature sensors, and tachometer all communicated with the computer through serial ports.

The computer ran a program written in National Instruments' LabVIEW graphical programming language. This program was required to read all the measurement devices, display the readings and calculated values on screen, and save the data to disk for later analysis, as well as control the conditions in the test rooms according to operator instructions. The scan rate for NI system was set at 20 Hz to provide a fast feedback control signal to the booster fans. The data logger and power meter were set to scan and report at 10-second intervals, which was also the rate at which the data were saved to disk. The program also received the readings from the three chilled mirror sensors and tachometer as they were sent at 1-

second intervals. The data that are displayed and saved to disk include the single measurements from the slow scan, plus the averages of all the high speed scan measurements taken in the same interval.

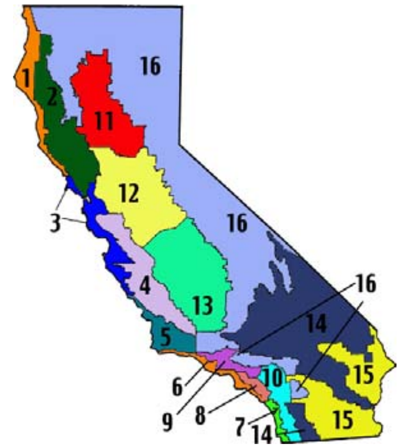


Figure 9: Title 24 Climate Zones

Test Conditions
The ASHRAE test standard for indirect evaporative coolers (Reference 5) primarily specifies the arrangement of the apparatus, the measurements to be taken, and the accuracy of instruments. It does not give specifics for the test conditions, other than some general guidelines, since evaporative cooling devices are mainly rated in terms of airflow. It does specify a minimum wet-bulb depression (difference between dry and wet-bulb temperatures) of 25°F.

An Australian test standard was reviewed that did provide some specifics for nominal test conditions. Reference 7 lists the following conditions:

- Inlet dry-bulb temperature: 38°C (100.4°F)
- Inlet wet-bulb temperature: 21°C (69.8°F)
- Room dry-bulb temperature: 27.4°C (81.3°F)
  (used in calculation of cooling capacity)

These numbers are close to those selected for the CEC Title-20 ECER rating, except for the inlet dry-bulb temperature, which is about 10°F higher.

The Coolerado Cooler™ was subjected to the same set of conditions as were developed in the previous evaporative cooler tests. These conditions were selected based on the desire to evaluate the performance of the test units over a range of environmental conditions that adequately represent the conditions found during the cooling season at various locations in PG&E's service territory. This territory covers nine of the sixteen distinct climate zones identified by the California Energy Commission for Title 24 analysis. The ASHRAE Handbook of Fundamentals (Reference 1) gives tables of cooling design condition for a large number of cities, including 19 within the PG&E service territory, representing all but one of the 9 climate zones (Zone 2 – Napa, Santa Rosa, Ukiah; see *Figure 9*). The tables list a number of useful climate design conditions, and of particular interest are the listings for conditions that are exceeded less than 0.4% of a year on average (about 35 hours). These design conditions include:

- Maximum dry-bulb temperature and coincident wet-bulb temperature (used in determining the cooling load on a building).
- Maximum wet-bulb temperature and coincident dry-bulb temperature (used for sizing cooling towers and other evaporative equipment)

An excerpt from this table showing the cities in the PG&E service territory is shown in *Table 2*:

Table 2: ASHRAE Design Conditions for Cities in PG&E Service Territory

City	Climate Zone	Elev.	Std P PSIA	Cooling DB/MWB				Evaporation WB/MDB			
				DB	MWB	WBD	RH	WB	MDB	WBD	RH
Alameda NAS	3	13	14.688	83	65	18	38%	67	79	12	54%
Arcata / Eureka	1	217	14.581	70	60	10	56%	62	67	5	76%
Bakersfield	13	492	14.436	104	70	34	18%	73	98	25	31%
Blue Canyon	16	5,285	12.097	84	59	25	24%	62	80	18	39%
Fairfield (Travis AFB)	12	62	14.662	98	67	31	18%	70	92	22	33%
Fresno	13	328	14.522	103	71	32	20%	73	98	25	30%
Lemoore (Reeves NAS)	13	236	14.570	103	72	31	22%	75	97	22	36%
Marysville (Beale AFB)	11	112	14.636	101	70	31	21%	72	97	25	30%
Merced (Castle AFB)	12	187	14.596	99	69	30	21%	72	96	24	31%
Mount Shasta	16	3,543	12.909	91	62	29	20%	64	87	23	30%
Mountain View (Moffat NAS)	4	39	14.675	88	65	23	28%	68	82	14	49%
Paso Robles	4	837	14.257	102	68	34	16%	70	97	27	26%
Red Bluff	11	354	14.508	105	70	35	16%	72	98	26	28%
Sacramento (NE - McClellan AFB)	12	75	14.655	102	70	32	19%	72	97	25	30%
Sacramento (NW - Metro AP)	12	23	14.683	100	69	31	20%	72	96	24	31%
Sacramento (SE - Mather Field)	12	95	14.645	101	69	32	19%	71	97	26	28%
Salinas	3	85	14.650	83	63	20	32%	66	78	12	53%
San Francisco	3	16	14.687	83	63	20	32%	64	79	15	44%
San Jose (Int'l AP)	4	56	14.666	93	67	26	25%	70	88	18	41%
Santa Maria	5	240	14.569	86	63	23	27%	66	81	15	45%
Stockton	12	26	14.681	100	69	31	20%	71	96	25	29%

ASHRAE also publishes a regional set of climatic data from which values for other cities can be obtained (Reference 2). From this source, about 300 more sites were obtained that are in or adjacent to the PG&E service territory. This source lists the design cooling dry-bulb and coincident wet-bulb, but unfortunately only the design wet-bulb without the coincident dry-bulb. Thus, an approximation was made for the appropriate dry-bulb temperatures based on the values in Reference 1. Reference 2 also lists those temperatures that are exceeded on average less than 0.1% of a year (about 9 hours), rather than the 0.4% values given in Reference 1, so the values tend to be about 1-2°F higher for the same locations.

The numbers from both sources were then plotted on a psychrometric chart (*Figure 10*) in order to determine a matrix of test points that would bracket the majority of these design conditions. The selection of the number of test points needed to balance having enough to adequately represent the probable operating conditions, yet not be so great as to extend the testing period. *Table 3* lists the selected matrix of ten test conditions:

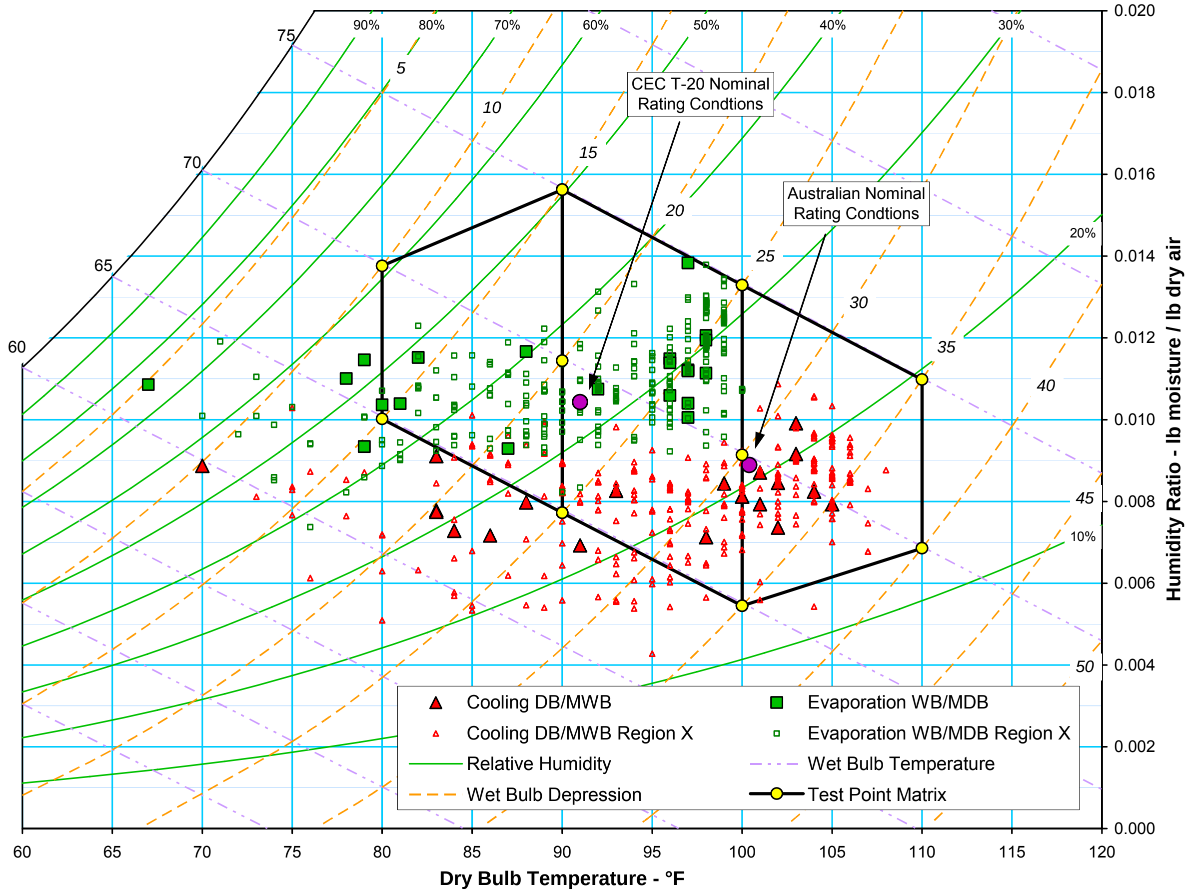
Table 3: Test Point Matrix

(Highlighted cells have less than the 25°F wet-bulb depression required by ASHRAE test standards)

Dry-bulb	Wet-bulb Temperature		
Temp. °F	65°F	70°F	75°F
80	×	×	
90	×	×	×
100	×	×	×
110		×	×

The test point at 100°Fdb and 70°Fwb is very close to the rating point used in the Australian test procedure (100.4°Fdb and 69.8°Fwb), while that at 90°Fdb and 70°Fwb is close to the CEC Title-20 rating point (91°Fdb and 69°Fwb). Both conditions are also indicated in the figure. It was also decided not to test at ambient temperatures below 80°F where straight ventilation cooling might be adequate. Four of the points have less than the minimum 25°F wet-bulb depression specified in the ASHRAE test standard (highlighted in *Table 3*) and were only included to provide performance details at high relative humidity.

Figure 10: Psychrometric Chart with Climate Deisgn Data and Test Points



Test Procedure

The test unit was "broken in" by running it off and on for about eight hours. This allowed for the breakdown of water repelling films or oils on the evaporative media to ensure that it is thoroughly wetted during testing.

The tests proceeded as follows:

1. The data acquisition system was started, and all instruments were ensured to be reading correctly.
2. The control points for room temperature and humidity were set into the computer, and the room conditioning system was started to control the room environment.
3. The test unit was turned on, and airflow station booster fan controls were set to maintain the desired static pressure at the outlet, and zero at the exhaust.
4. Other than the test done to evaluate sensitivity to line voltage, the voltage was adjusted as necessary with a Variac to maintain 110 VAC.
5. Once the desired environmental conditions were achieved and stable for at least 15 minutes, a data log file was opened on the computer and the instrument readings were recorded for another 30 minutes. Any operational problems observed were documented.
6. The room conditioning system was then adjusted to the next set of conditions.

The recorded test data were averaged over the stable test period, and the averaged values were used to calculate the performance characteristics. The results from all of the tests were tabulated, and analyzed graphically by plotting the results as a function of the control parameters.

RESULTS

The testing of the sample unit went relatively smoothly, although there was one problem that resulted in some preliminary data sets being discarded. As mentioned in the description of the system, the solenoid valve controlling the water supplied to the unit is activated based on the flow of water through the drain line. This flow rate is actually only a level measurement in an inclined ½-inch pipe, using a pair of top and bottom electrodes. When enough water is flowing through the drain, a contact is made between the two electrodes through the water, and the supply valve is signaled to close. When the flow is low, contact is broken and the valve opens. The problem that occurred was that a small piece of foam packing material had found its way into the drain line and become lodged between the electrodes. The debris acted like a sponge holding on to water even when there was no flow in the tube. Thus, it kept the supply valve closed until the debris dried out enough to break electrical contact. This caused the unit to be frequently starved for water, and showed up as unstable behavior. The performance degradation was observed, which caused a search for the cause, and led to the discovery of the debris and its removal.

The results from the tests are shown in several tables and figures, of which most are located at the end of the report in the Appendix. Also in the Appendix is a detailed summary of all the averaged test measurements and calculated results.

Table 4 lists several parameters averaged over all of the tests conducted at variable environmental conditions, but at constant fan speed, line voltage, and with no external resistance. The exception are the values listed for capacity and EER, which are averaged from only those tests done at one specific outdoor condition. This was chosen to be the point at 100°Fdb / 70°Fwb, which approximates the Australian rating condition, but without their requirement of 80 Pa (0.32 inches of water) external resistance. This table is similar to tables included in the earlier reports, and is provided to compare the results for the Coolerado Cooler™ with other systems. As obtained, the Coolerado Cooler™ was a single speed system; however, part way into the testing, the system was equipped with a variable speed fan controller, which will be offered as an option on future systems. The results shown in the "low speed" section of the table for the Coolerado Cooler™ are for a single test done at about the same percentage of airflow reduction as for the other units at low speed.

Table 4: Average Results for Airflow and Power

Parameter	High Speed	Low Speed[1]
Supply Airflow[2] (cfm)	**1,500**	**1,020**
Exhaust Airflow[2] (cfm)	**1,320**	**950**
Total Unit Power (W)	**1,329**	**984**
Effectiveness	**86.0%**	**88.2%**
Room Capacity (tons)[3]	**0.95**	**0.89**
Room EER (Btu/Wh)[3]	**9.1**	**10.8**
IA Capacity (tons)[4]	**3.46**	**2.61**
IA EER (Btu/Wh)[4]	**32.9**	**31.8**
T20 ECER (Btu/Wh)	**10.4**	

1 "Low speed" is a single selected point for the variable speed fan.
2 Measured outlet airflow referenced to the intake density.
3 *Room Capacity* $\approx 1.08 \times CFM \times (Tdb_{room} - Tdb_{supply})/12{,}000$ *(Equation 2);* intake air at 100°Fdb / 70°Fwb, referenced to a room temperature of 80°F
4 *IA Capacity* $\approx 1.08 \times CFM \times (Tdb_{intake} - Tdb_{supply})/12{,}000$ *(Equation 4),* intake air at 100°Fdb / 70°Fwb

Table 5 through *Table 7* contain three different parameters as a function of the inlet dry and wet-bulb temperatures: the resulting supply and exhaust temperatures, and wet-bulb effectiveness. The points that do not have the 25°F wet-bulb depression required by ASHRAE are shaded. The results indicate that the unit shows an improvement in effectiveness at higher ambient temperatures. This may be a result of expansion of the air passages allowing for easier flow, or from increased evaporation rates. The key point being that its performance improves as temperatures get higher, as opposed to the performance of a conventional air conditioner, which gets worse. The test unit shows some sensitivity to the inlet air conditions, with an effectiveness range of ten percentage points over all of test conditions.

Table 5: Supply Temperatures (°F)

Intake Tdb	Intake Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	68	72	
90	69	73	78
100	70	72	78
110		74	78

Table 6: Exhaust Temperatures (°F)

Intake Tdb	Intake Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	71	74	
90	74	77	80
100	77	78	83
110		82	85

Table 7: Wet-Bulb Effectiveness

Intake Tdb	Intake Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	81%	81%	
90	83%	85%	83%
100	89%	90%	86%
110		91%	90%

Figure 11 shows an example of the process for the Coolerado Cooler™ on a psychrometric chart, using actual test data for one particular set of inlet conditions (100°Fdb/70°Fwb, which is highlighted in the previous tables). (The control scheme for a test was to achieve the wet-bulb temperature within ±1°F; thus, the averaged inlet conditions did not always fall exactly at the desired conditions. The average inlet air wet-bulb temperature for this test was 69.2°F) This figure is meant to help describe the constant dew-point process through the indirect evaporative cooler, and to graphically describe the effectiveness. The resulting supply temperature was 72.4°F, resulting in an effectiveness value of 90%.

The ASHRAE summer and winter comfort zones are highlighted in the chart to show the conditions that should be maintained in a space to keep most occupants in a working environment comfortable. The air supplied to the space should be at a condition to the left and slightly below the summer comfort zone to allow for sensible and latent heat gains within the space. A direct evaporative cooler would provide supply air that is too humid under these outside conditions to maintain adequate comfort.

Also indicated in the chart is the condition of the exhaust stream where the evaporative cooling has taken place. This and Table 6 show that the temperature of the exhaust is also considerably cooler than the entering air. Since the exhaust airflow is almost as large as the supply rate, this cool exhaust could be put to use. An ideal application of the Coolerado Cooler™ is in conjunction with a rooftop packaged unit to condition the required minimum outside air. Not only will the pre-cooled supply stream decrease the load on the packaged unit, but the evaporatively cooled exhaust stream could be directed at its air-cooled condenser to improve its efficiency. With the increasing emphasis on indoor air quality, providing adequate ventilation air has become an important issue, and this device would help to reduce the overall energy impact.

Figure 11: Process Description at one Test Condition

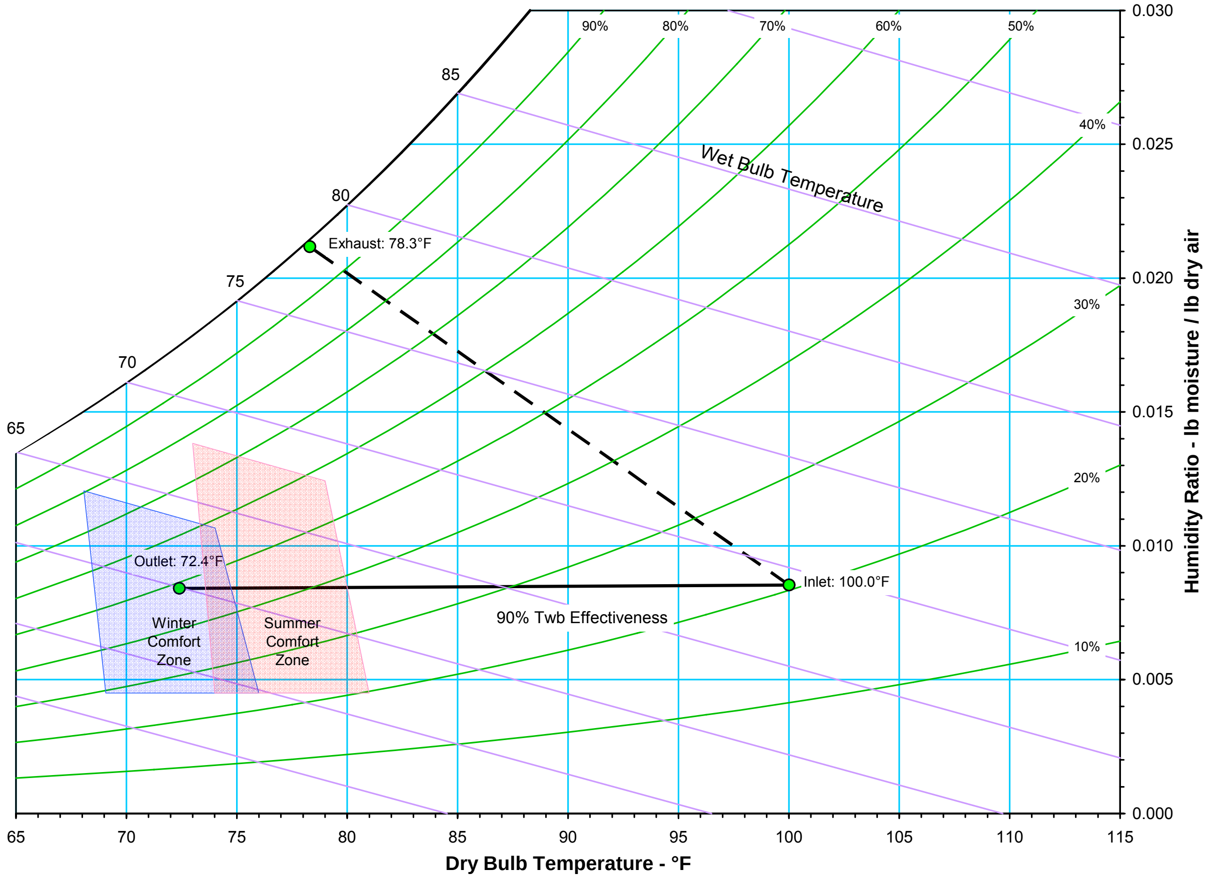
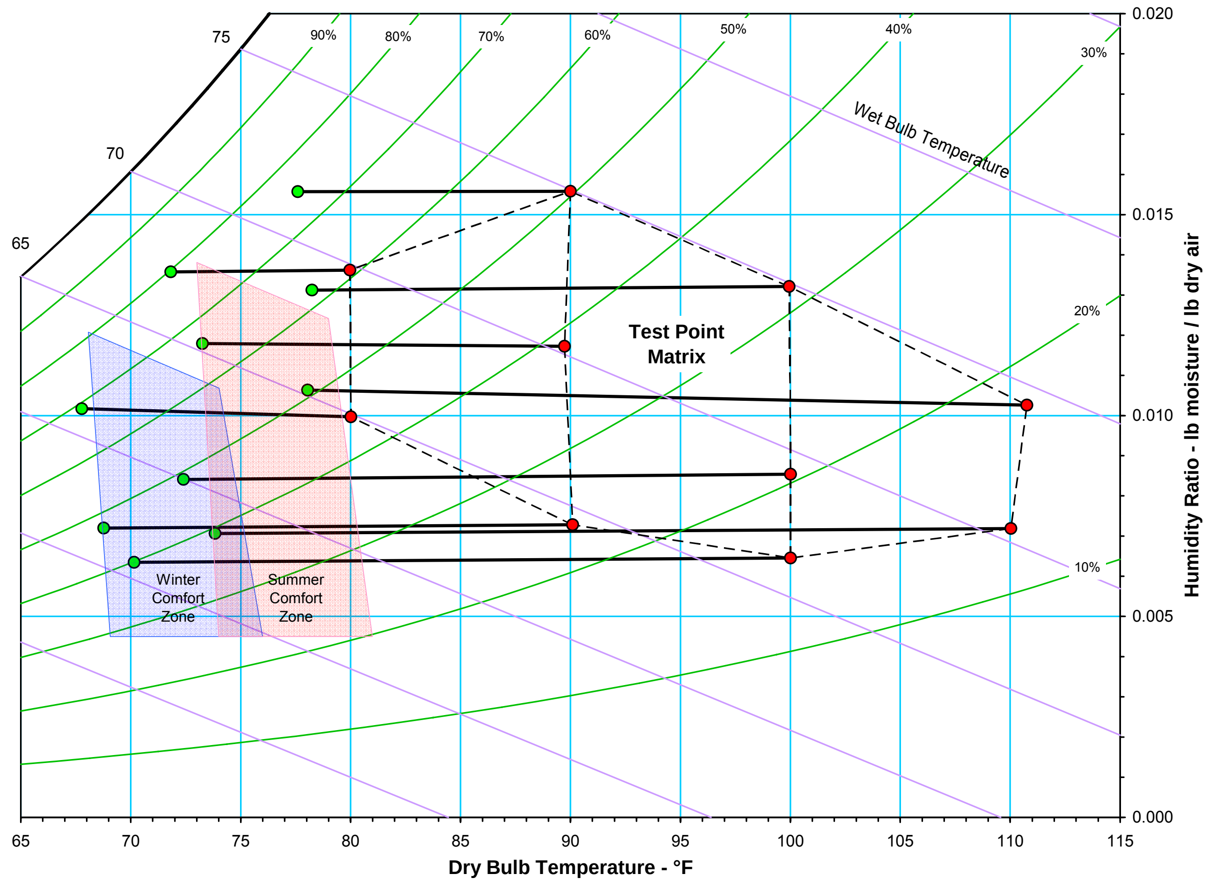


Figure 12 shows the results from the tests at all of the conditions in the test point matrix, as a graphic representation of *Table 5*. As shown in this chart, not all of the conditions in the test point matrix could be achieved precisely, particularly in terms of humidity. The ambient air was also never dry enough during the testing period to achieve the desired condition of 100°Fdb / 65°Fwb. The results show that only the three test conditions that had an intake dew point temperature above 64°F did not produce supply conditions that would meet the comfort zone requirements, although all of the tests resulted in supply conditions less than the 80°F used as the basis for calculating cooling load. Since this system does not change the absolute humidity or dew point temperature of the entering air, some comfort may be lost if it is very humid outside.

Figure 12: Performance at All Test Conditions



The following section discusses several charts of the test results that are located in the Appendix. Most of these are similar to the charts included in the previous PG&E Emerging Technology Application assessment reports for comparison.

Figure 13 and *Figure 14* examine the sensitivity of the Coolerado Cooler™ to increasing the backpressure (or the external resistance to flow) on the supply air stream. The inlet air condition was maintained at 100°Fdb / 70°Fwb for all of these tests. In the first figure, the results are plotted as a function of the supply airflow rate, which is the recommended method of displaying results for evaporative coolers from the ASHRAE test standards. This chart includes the measured results for wet-bulb effectiveness, power, and backpressure, plus the calculated CEC Title-20 ECER values. Of the latter, the value of particular interest is where the backpressure was equal to 0.3 inches of water, and this point has been emphasized. The observation from this chart is that power and efficiency are not significantly sensitive to backpressure, and there is a slight improvement in effectiveness as the backpressure is increased. The low sensitivity of for power may be because the unit already has considerable resistance to flow, and increasing the backpressure on the outlet has only a small impact. In the second figure, the values are graphed normalized to the measured parameters with no backpressure (zero inches of water), the values of which are given in the chart legend. Included in this chart are the measurements of supply, exhaust, and total intake airflow rates, total power, and effectiveness. The results show that as the resistance is increased, airflow is diverted from the supply stream to the exhaust, with a relatively small decrease in the overall intake airflow. With the small decrease in total airflow combined with a small rise in the static pressure at the fan outlet, the power consumption is virtually unchanged. Here again is shown the slight increase in the system effectiveness as the higher exhaust flow removes more heat from the supply stream.

Figure 15 shows the results of a test to see the effect of varying line voltage, since demand for cooling typically coincides with periods of high electrical demand when there may also be voltage sags. The results are shown relative to the points taken at a line voltage of 110V, and ranged from 95 to 130V. The results show that there is a decrease in speed, power, and airflow as the voltage is decreased, but that the effectiveness remains relatively stable since the relative magnitudes of the supply and exhaust airflows remain the same. Thus, its cooling capacity will decrease slightly during a voltage sag. Also shown in the chart are measurements of motor power factor, which indicates that it actually reaches a maximum at about 115V, and decreases slightly to either side.

Figure 16 shows the results from tests done with the optional fan speed controller attached, and the fan set to several different speed settings while the external resistance was held at zero static. The graph includes plots of power, power factor, supply and exhaust airflow, and effectiveness as a function of fan speed relative to their values at full speed. The graph shows a decrease in power and airflow with decreasing speed, but an increase in system effectiveness. The power factor of the motor also falls off significantly using the speed controller, which is likely due to how the electronics control the motor speed. The trends of airflow rates must be taken with some skepticism, as most of the reduced speed conditions resulted in a pressure drop at the measurement nozzle less than what is required by the airflow measurement standards for an accurate measurement. This is particularly true for the supply airflow measurement nozzle, which has a lower accuracy limit of about 1,300 cfm, which is only slightly less than the full speed airflow. The flow measurement calculation can still be made, but the uncertainty is growing exponentially. At the 400 RPM fan speed, the airflow is likely not zero; it was just too low to produce a measurable pressure drop with the applied nozzle arrangements.

Figure 17 is a chart of power consumption as a function of total intake airflow rate. The measurements from each of the previous sensitivity tests along with the results from the changing intake condition tests are shown with different symbols. The results show that the power consumption is slightly sensitive to temperature and voltage variations, but not to backpressure. The slight temperature sensitivity may be due to fan motor losses, or a result of the change in air density on the fan. The reduction in fan speed produced an almost linear reduction in fan power relative to airflow rate, which was somewhat unusual since the general fan laws imply a cubic relation of power to flow. This again may be due to an inaccurate measure of airflow at the lower speeds.

Figure 18 plots the fan motor power factor as a function of airflow rate. The results indicate that the power factor is not greatly affected by environmental conditions or flow resistance, but shows the slight sensitivity to line voltage and the more significant sensitivity to the fan speed controller. The normal power factor for this motor at about 0.98 is high relative to most other evaporative cooler fan motors, which tend to be in the 0.6 to 0.7 range. This may be due to a number of factors, including that it is a larger horsepower than the others, it is operated at a different load point with the higher flow resistance, and because it is a better quality motor. A low power factor is a trait common to many fractional horsepower, single-phase induction motors, and these results show that it can be improved. The reactive power does no real work in the motor, but can contribute to significant heat generation. Since the motors are typically located in the air stream in most evaporative coolers, the heat produced by the motor will be delivered to the conditioned space, which can degrade the performance of coolers with low power factor motors.

Figure 19 and *Figure 20* both show the data contained in *Table 7* in graphical form. The first figure shows overall unit effectiveness as a function of the entering dry-bulb temperature, while the second plots it as a function of the entering wet-bulb depression (difference between the dry and wet-bulb temperatures). These charts show that there is an apparent increase in effectiveness as the dry-bulb temperature or wet-bulb depression rises. This may be due to changes in airflow due to its decreased intake density, increased evaporation rates, or thermal expansion of the heat and mass exchange modules allowing for freer airflow.

Figure 21 and *Figure 22* examine the cooling capacity and energy efficiency ratio, which combine the effects of airflow rate and supply temperature to demonstrate its ability to cool off a space. As with the previous charts of effectiveness, the results are graphed as a function of entering dry-bulb temperature (Figure 21) and entering wet-bulb depression (Figure 22), and grouped by entering wet-bulb temperature. As defined previously, the evaporative cooler capacity is defined as the ability of the unit to maintain a space at 80°F, disregarding humidity. The capacity is listed in tons (12,000 Btu/hr) and the energy efficiency ratio (which is capacity divided by the total unit power) is listed in Btu/Wh. The results show that this unit will be able to provide some cooling effect under all of the tested conditions (the supply temperature was always below 80°F); but that its cooling effect looks low because of its low airflow.

Figure 23 shows the alternative method of calculating capacity and efficiency for an indirect evaporative cooler, which is its ability to cool off the entering air. A similar chart was included in the previous test reports on evaporative coolers, but only looked at their sensible cooling effect (temperature reduction). Since this system does sensible cooling with no increase in moisture, the cooling effect can be considered total cooling. The total intake air cooling capacity of a direct evaporative cooler will be negative because of the exchange of sensible heat for latent heat, and the addition of fan energy. The results are graphed as functions of the inlet wet-bulb depression, and show a nearly linear relationship. The product literature on the Coolerado Cooler™ claims a capacity of 5.4 to 6.0 tons and an EER of 40+. While these test results do not quite confirm the capacity numbers, it does confirm an EER over 40 under high wet-bulb depression conditions using this definition of capacity.

The last chart (*Figure 24*) examines the relative water consumption rates of all the test units. This figure looks at only the water evaporation rate (as determined from measurements on the air side of the process) as a function of the intake wet-bulb depression. The evaporation rate is determined by taking the moisture (humidity ratio) rise from inlet to outlet, and multiplying by the air mass flow rate. For this unit, the evaporation is all taking place in the exhaust stream. The total actual water consumption of the Coolerado Cooler™ was difficult to measure over the short test periods due to the irregular operation of the solenoid valve. The valve could be open or shut through the duration of an entire test period. While the total flow was recorded during each test, the results are erratic due to the intermittent valve operation, and are therefore inconclusive. The average water consumption rate at an outside condition of 100°Fdb/70°Fwb is estimated to be about 12 gallons per hour, with ⅔ of that going to evaporation and ⅓ going out the drain.

CONCLUSIONS

This study investigated the performance of a first generation example of the Coolerado Cooler™. The primary advantage of the Coolerado Cooler™ in relation to the other evaporative cooling units is that it cools without adding any moisture to the conditioned space. This advantage comes at a price, though; and not just in terms of initial cost. The heat and mass exchange modules have a very high resistance to flow, resulting in decreased delivered airflow and higher power consumption than other evaporative cooler systems, as evaluated under previous PG&E Emerging Technologies projects. However, this unit still uses considerably less power than a conventional air conditioner, and will likely keep a space more comfortable than other evaporative coolers for more of the cooling season. As this evaluation was only a series of short-term tests, they give no indication of its long-term reliability or maintenance requirements in actual use.

Some of the key findings are summarized below.

1. The wet-bulb effectiveness of this unit varied from 81% to 91% over the range of test conditions. This was somewhat less than anticipated since this system has the capability to achieve an effectiveness in excess of 100%. However, it is comparable to the more advanced systems previously tested, and is better than what a basic single-stage indirect evaporative cooler could achieve (direct evaporative cooler attached to an air-to-air heat exchanger).

2. The effectiveness improved at higher intake dry-bulb temperatures and wet-bulb depressions, and for all of the applied test conditions, the unit was able to produce supply temperatures less than 80°F (even at an intake temperature of 110°F).
3. This system does pure sensible cooling of the intake air, with no moisture addition or removal. Under conditions of low outside humidity, this unit will produce indoor conditions that are closer to the ASHRAE comfort zone than single stage direct or two-stage indirect/direct evaporative coolers.
4. The airflow at full speed was about 1,500 cfm, which is 40-60% less than the airflow provided by direct evaporative coolers. This reduces its apparent cooling capability relative to the other systems if their moisture addition is ignored. However, a dry air supply flow does not have to be as large as a humid supply to feel as cool. This low airflow does have another advantage in that the system could be connected to existing ductwork in a residence if it is sized to handle up to a 3½ to 4 ton central air conditioner, and it also creates lower noise levels.
5. The power consumption of this system was 40-80% greater than that of the other evaluated evaporative coolers at about 1,330 watts. However, its power factor was considerably better, such that in terms of apparent power (Volt-Amperes), this unit used from 30% more to 14% less VA than the others. Since a utility company has to produce or obtain the VA even though only the watts are billed, the higher power consumption of this unit may not be as much of a disadvantage from the utility standpoint. Also, discussions with the manufacturer indicated that future products will use a more advanced fan motor, which is expected to cut the power consumption by nearly half.
6. The new California Energy Commission Title 20 evaporative cooler efficiency ratio (ECER) does not reflect increased comfort from the lack of moisture addition to the supply air, and thus treats this and other indirect systems unfairly when compared to direct systems. A caution should be added to Title 20 to only use the rating to compare similar systems, and that lower numbers for indirect systems relative to direct systems do not indicate poorer performance.
7. The cost of this unit was considerably more than other evaporative cooling systems. However, this system is at an early stage of development, and system costs should go down as production rates are increased. While comparable in initial cost to a conventional air conditioner, the energy savings reduce its life-cycle cost.
8. This system has application wherever evaporative coolers are currently used or that require large amounts of fresh, outside air (e.g. residences, commercial kitchens, gymnasiums). It can also be used in conjunction with conventional air conditioning systems to treat the required ventilation air, something that should not be done with a direct evaporative cooler because the conventional system would then have to remove the added moisture. The relatively cool exhaust stream could also be directed at the air-cooled condenser coils to improve the air conditioner efficiency.

Recommendations for Follow-on Activities

It was brought to our attention by Rick Gillan of Idalex and Dave Bisbee of the Sacramento Municipal Utility District (SMUD), during the draft report review process that it is difficult to compare this system to either a direct evaporative cooler or a conventional air conditioner. This is because an indirect evaporative cooler does not increase indoor moisture and it introduces 100% outside air into the house. Based on their input this report has been limited to presenting only the laboratory performance of the Coolerado Cooler.

To effectively evaluate the Coolerado Cooler and other evaporative cooling technologies to air conditioning technologies a combination of laboratory, computer modeling and field monitoring data must be collected and analyzed. The laboratory results should be used to develop a computer modeled house in which the characteristics of different systems could be applied. This model should be calibrated using field data to show how much of the cooling season the interior space can be kept within ASHRAE comfort conditions, and to more accurately estimate the annual cost to operate the different systems.

Based on its system effectiveness, the Coolerado Cooler should be qualified for the highest rebate tier. The attractiveness of this system for providing a more comfortable environment will need to be high enough to counter the higher relative cost of the system, even after a rebate.

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APPENDIX

Figure 13: Performance Sensitivity to Outlet Backpressure

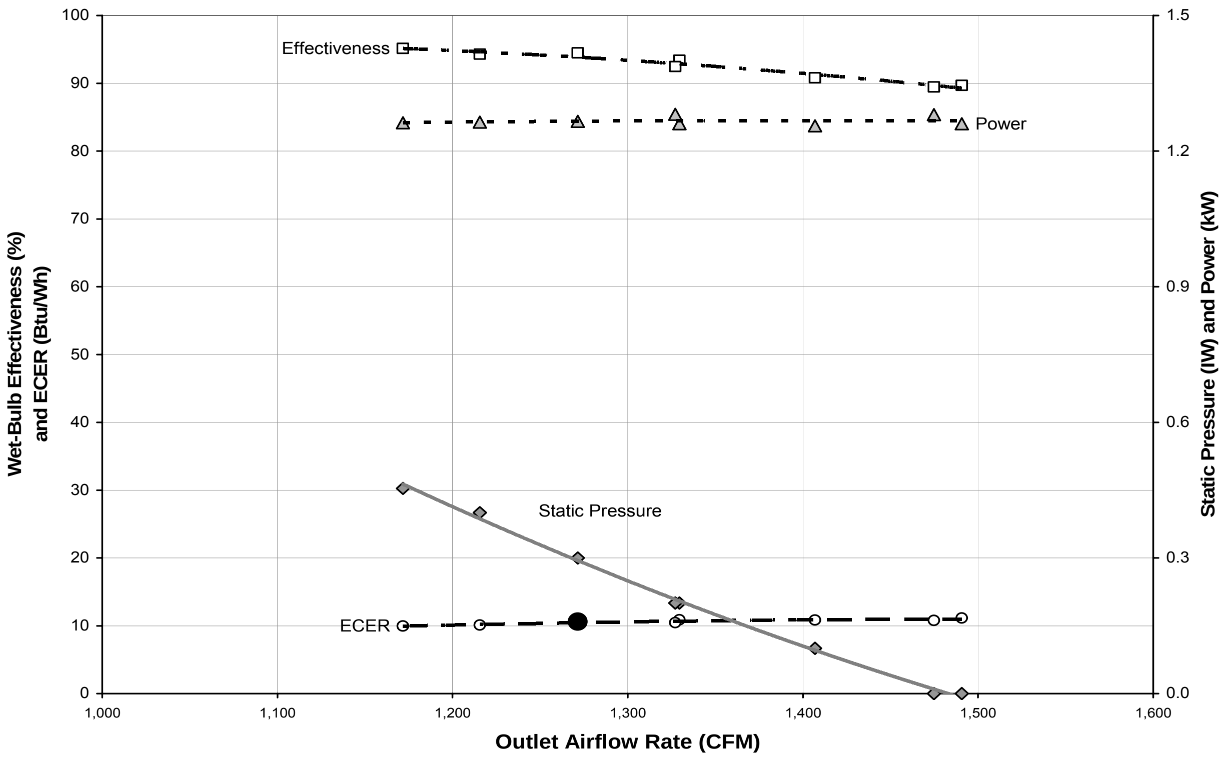

Figure 14: Normalized Performance Sensitivity to Outlet Backpressure
(Relative to no external resistance)

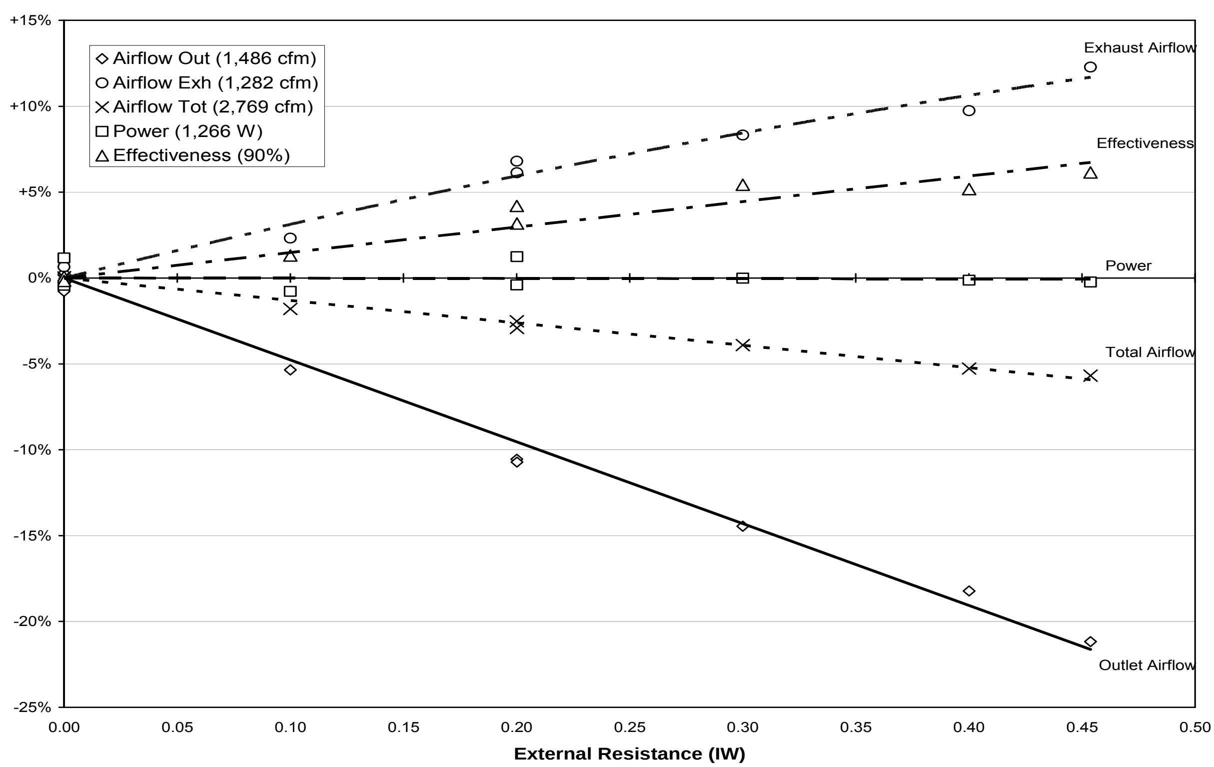

Figure 15: Normalized Performance Sensitivity to Line Voltage
(Relative to 110 VAC)

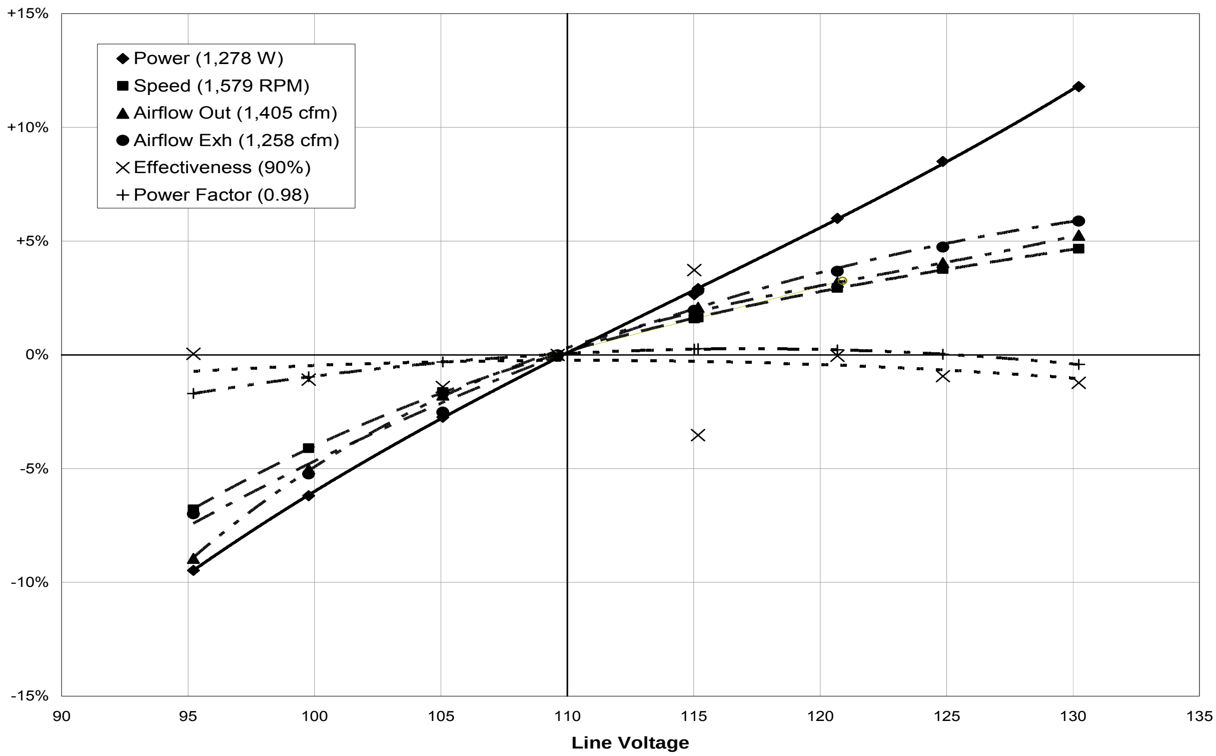

Figure 16: Normalized Performance Sensitivity to Fan Speed
(Relative to full speed at ~1,600 RPM)

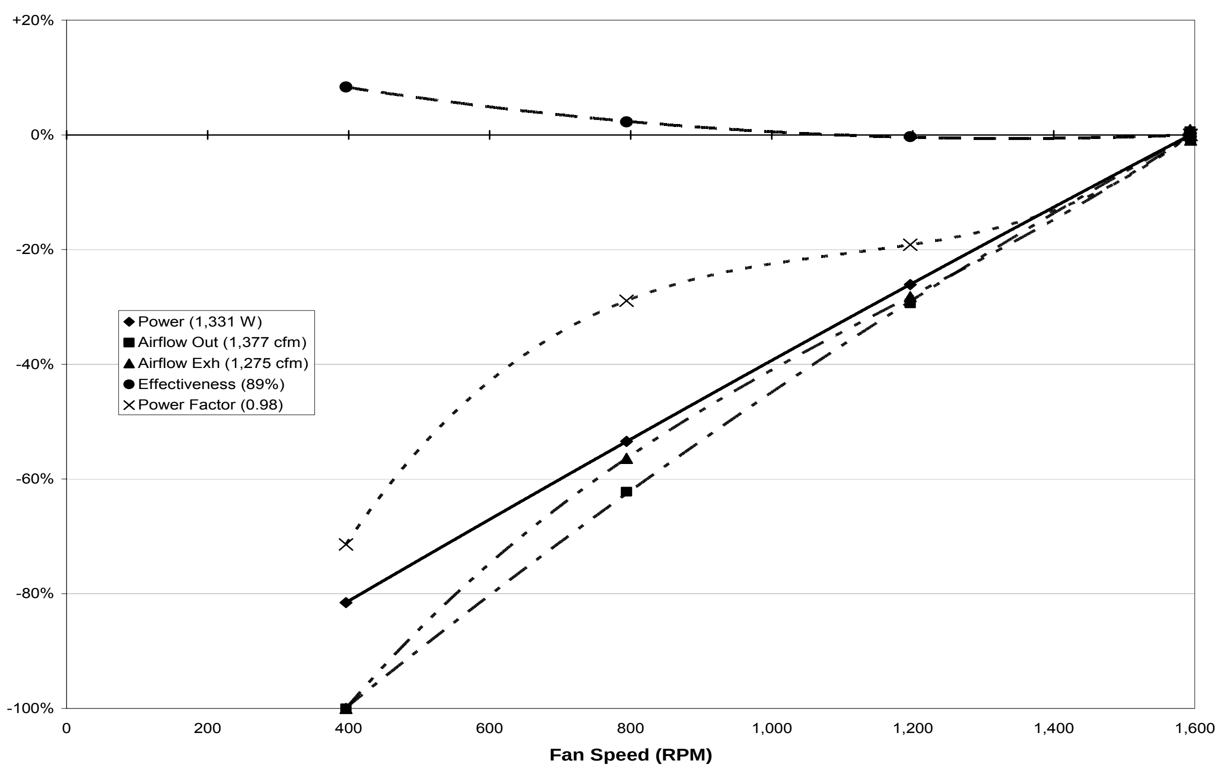

Figure 17: Total Power versus Airflow

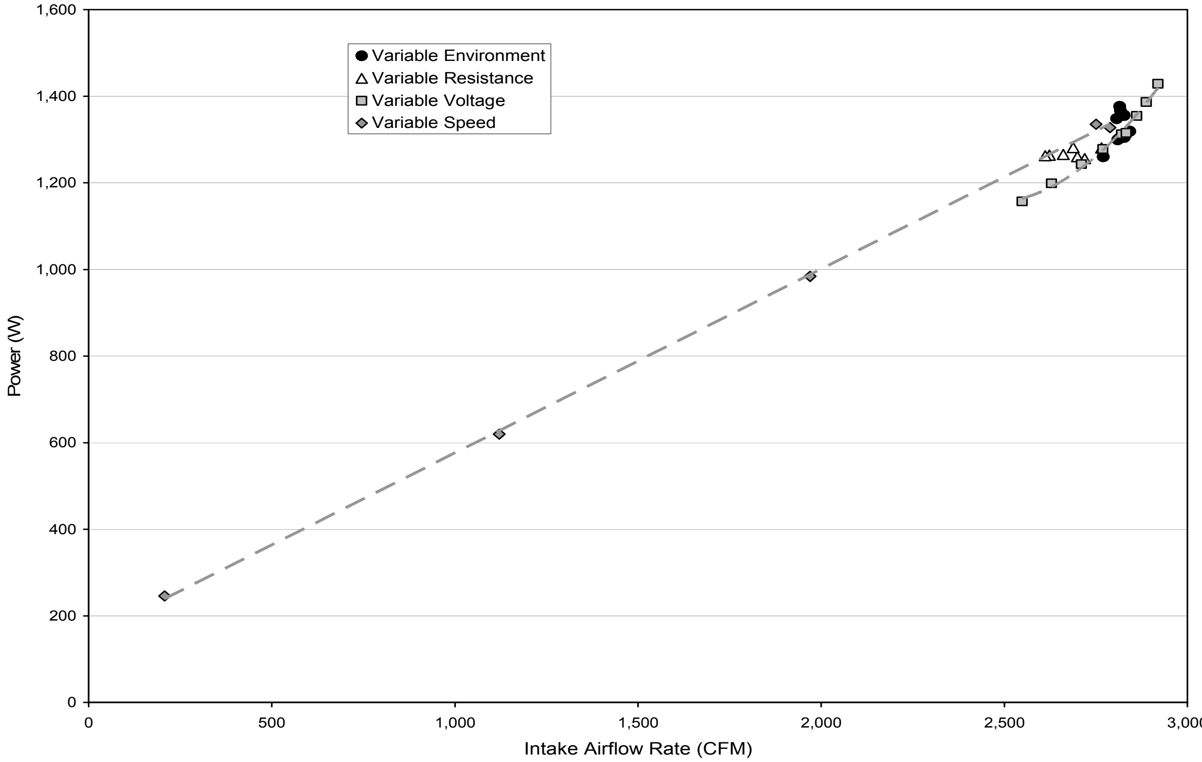

Figure 18: Power Factor versus Airflow

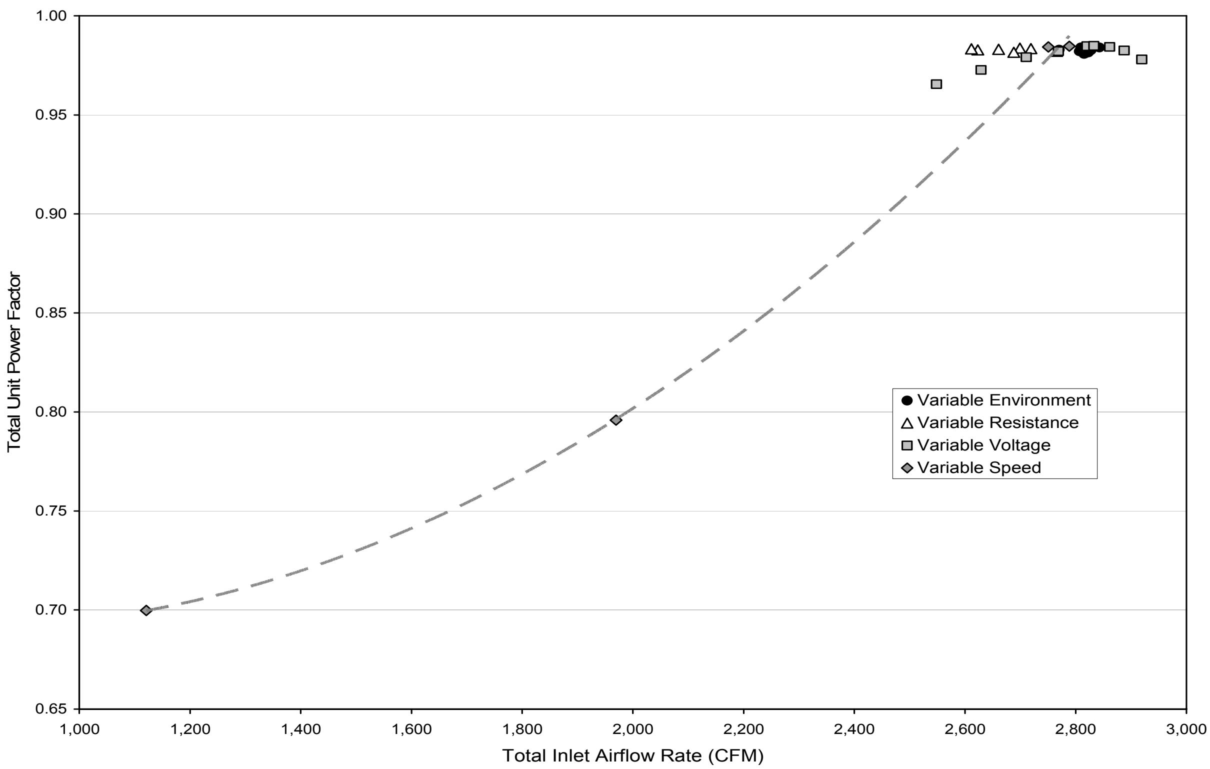

Figure 19: Effectiveness versus Dry-Bulb Temperature

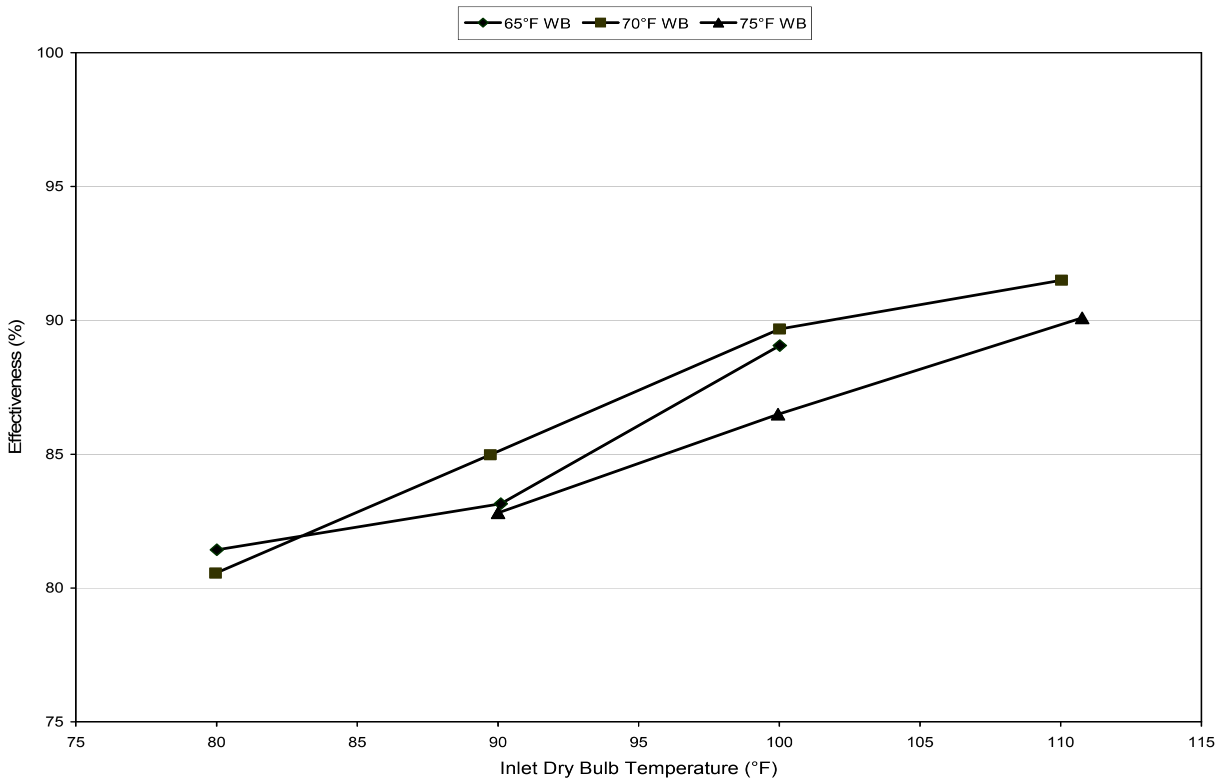

Figure 20: Effectiveness versus Wet-Bulb Depression

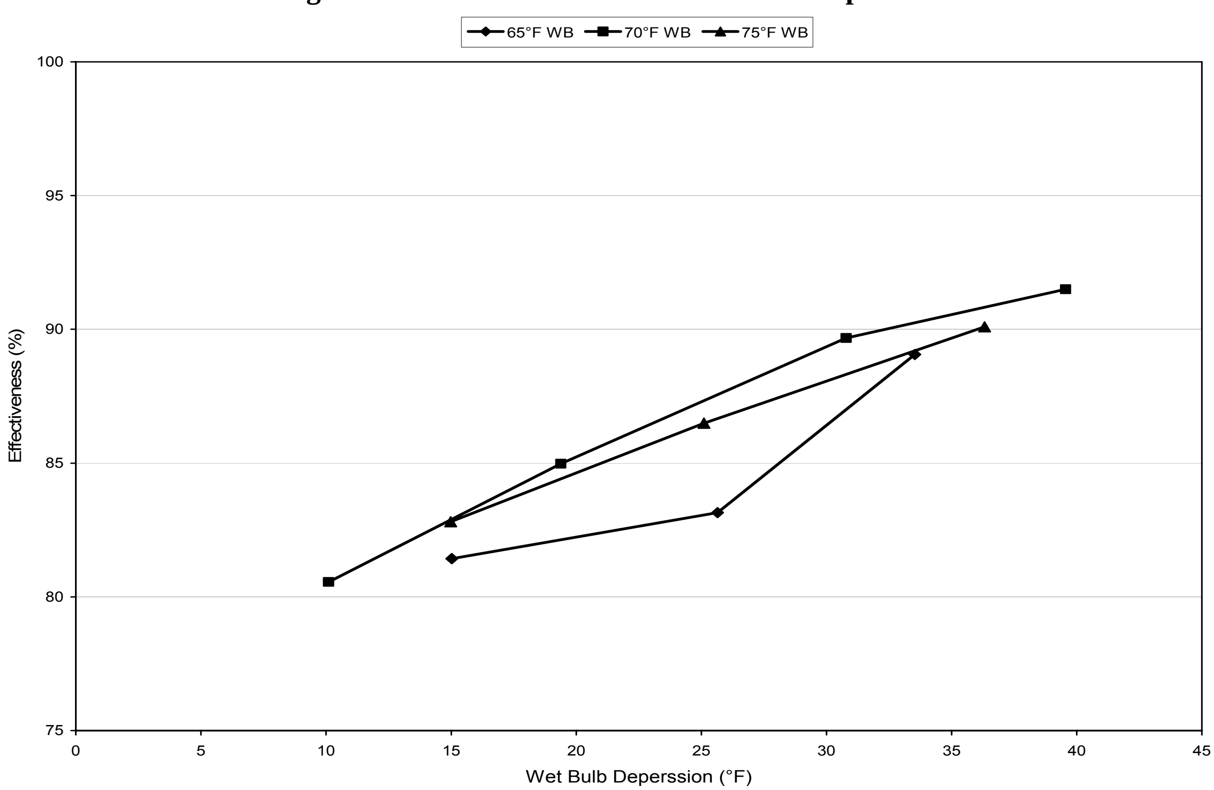

Figure 21: Room Cooling Capacity and EER versus Dry-Bulb Temperature
(80°F Room Temperature Basis)

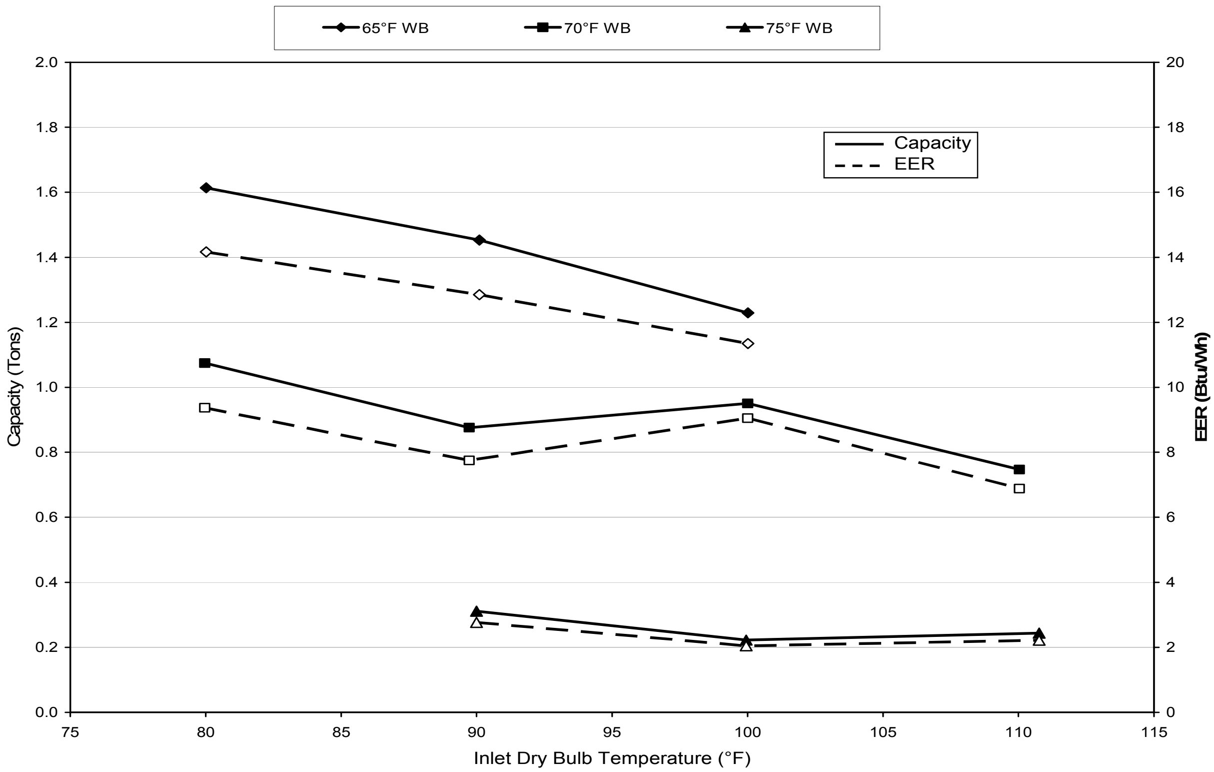

Figure 22: Room Capacity and EER versus Wet-Bulb Depression
(80°F Room Temperature Basis)

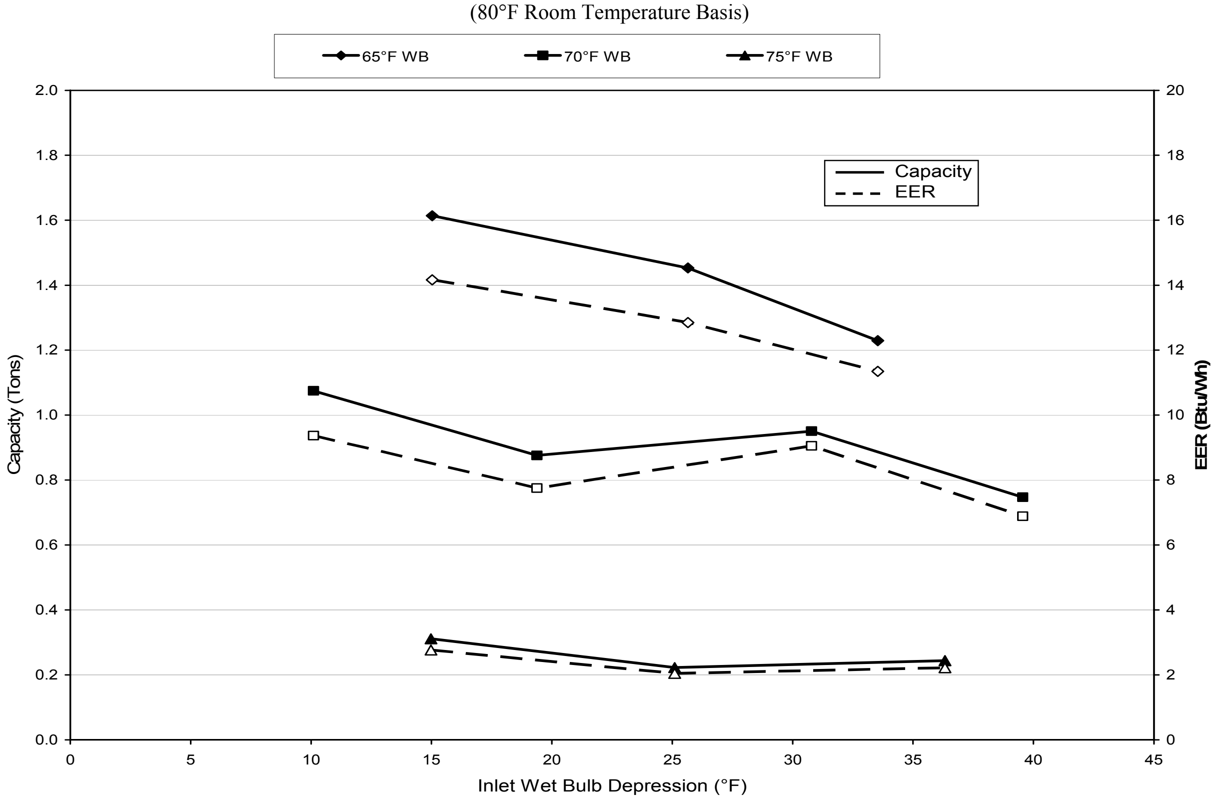

Figure 23: Total Cooling of Inlet Air
(with no outlet backpressure)

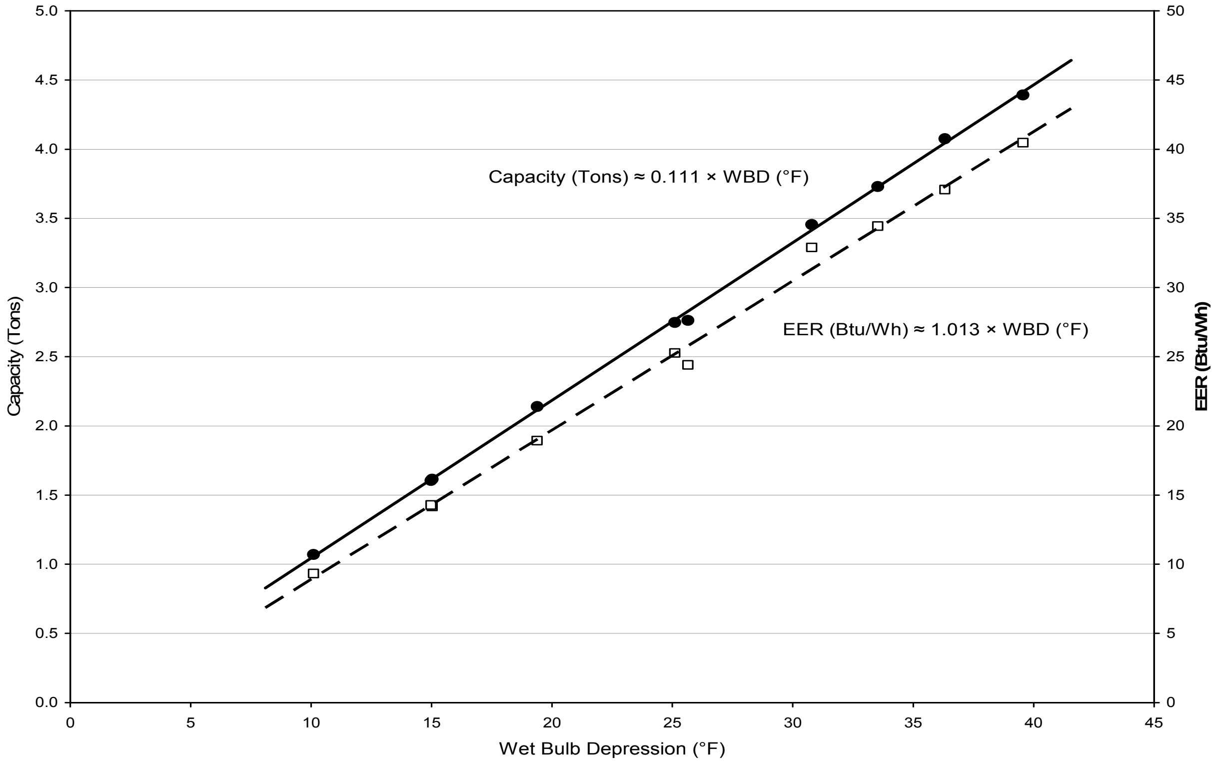

Figure 24: Calculated Evaporation Rate
(with no outlet backpressure)

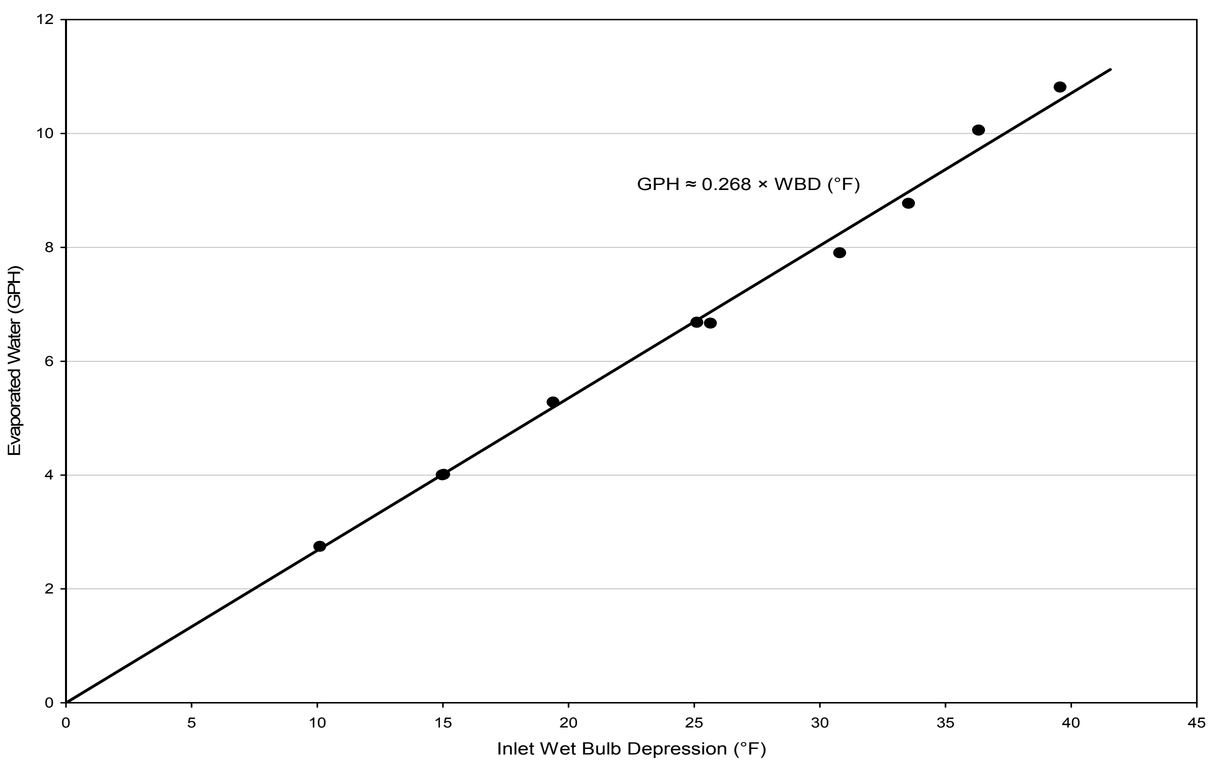

Table 8: Coolerado Cooler™ Test Data

Test Summary Information	Full Speed										Variable Speed				
General															
Date (all year 2005)	Feb 23	Feb 23	Feb 25	Feb 23	Feb 23	Feb 22	Feb 25	Feb 23	Feb 22	Feb 23	Mar 1	Mar 1	Mar 1	Mar 1	Mar 1
Start Time	8:13a	4:27p	12:45p	3:04p	10:18a	9:17a	3:44p	2:12p	3:10p	11:09a	2:51p	12:38p	1:22p	2:13p	2:51p
Duration (minutes)	30	28	30	30	30	33	30	30	30	10	30	30	30	30	30
Barometric Pressure (in. of Hg)	29.50	29.51	29.52	29.50	29.52	29.41	29.48	29.51	29.36	29.52	29.50	29.54	29.52	29.51	29.50
Nominal Test Conditions															
Inlet Dry Bulb Temperature (°F)	80	90	100	80	90	100	110	90	100	110	100	100	100	100	100
Inlet Wet Bulb Temperature (°F)	65	65	65	70	70	70	70	75	75	75	65	65	65	65	65
Fan Speed	1,590	1,600	1,610	1,590	1,600	1,590	1,610	1,590	1,610	1,610	400	1,590	1,200	790	400
Inlet Air Properties															
Dry Bulb Temperature (°F)	80.0	90.1	100.0	80.0	89.7	100.0	110.0	90.0	100.0	110.8	100.0	100.0	100.0	100.0	100.0
Dew Point Temperature (°F)	56.7	48.3	45.1	65.3	61.2	52.4	47.9	69.1	64.3	57.5	38.9	44.6	42.7	40.2	38.9
Relative Humidity (%) - calculated	44.8	23.8	15.6	61.0	38.5	20.5	12.9	50.5	31.5	17.9	12.3	15.3	14.2	12.9	12.3
Wet Bulb Temperature (°F) - calculated	65.0	64.5	66.5	69.9	70.4	69.2	70.5	75.0	74.9	74.5	64.5	66.3	65.7	64.9	64.5
Wet Bulb Depression (°F)	15.0	25.7	33.5	10.1	19.4	30.8	39.6	15.0	25.1	36.3	35.5	33.7	34.3	35.1	35.5
Exhaust Air Properties															
Dry Bulb Temperature (°F)	70.9	73.6	77.5	74.1	77.2	78.3	81.7	80.2	83.0	85.2	78.5	76.8	76.6	76.7	78.5
Dew Point Temperature (°F)	69.9	72.3	76.2	72.8	76.0	77.9	81.6	78.8	81.1	84.1	75.5	76.1	75.4	75.7	75.5
Relative Humidity (%) - calculated	96.5	95.8	95.9	95.9	96.2	98.7	99.7	95.6	93.9	96.4	90.5	97.8	96.1	96.7	90.5
Wet Bulb Temperature (°F) - calculated	70.2	72.7	76.5	73.2	76.3	78.0	81.5	79.2	81.5	84.3	76.3	76.3	75.7	76.0	76.3
Outlet Air Properties															
Dry Bulb Temperature (°F)	67.8	68.8	70.2	71.8	73.3	72.4	73.8	77.6	78.2	78.1	66.0	70.1	69.7	68.2	66.0
Dew Point Temperature (°F)	57.2	48.0	44.7	65.2	61.3	52.0	47.5	69.1	64.2	58.5	39.1	44.3	42.3	39.8	39.1
Relative Humidity (%) - calculated	69.0	47.4	39.9	79.8	66.2	48.7	39.2	75.2	62.0	51.0	37.2	39.4	37.1	35.4	37.2
Wet Bulb Temperature (°F) - calculated	61.1	56.7	55.8	67.3	65.4	60.0	58.4	71.6	68.6	65.3	51.8	55.6	54.6	53.1	51.8
Performance								Fan Speed (RPM) -->			**396**	**1,594**	**1,196**	**794**	**396**
Exhaust Dry Bulb □T (°F)	-9.1	-16.5	-22.6	-5.9	-12.6	-21.7	-28.4	-9.8	-16.9	-25.6	-21.5	-23.2	-23.4	-23.3	-21.5
Exhaust Wet Bulb □T (°F)	5.2	8.2	10.0	3.3	6.0	8.8	11.1	4.1	6.7	9.9	11.8	10.0	10.1	11.1	11.8
Outlet Dry Bulb □T (°F)	-12.2	-21.3	-29.9	-8.1	-16.5	-27.6	-36.2	-12.4	-21.7	-32.7	-34.0	-29.9	-30.3	-31.8	-34.0
Outlet Dew Point □T (°F)	-0.6	0.3	0.4	0.1	-0.2	0.4	0.5	0.0	0.2	-1.0	-0.1	0.3	0.4	0.4	-0.1
Wet-Bulb Effectiveness (%)	81.4	83.2	89.1	80.6	85.0	89.7	91.5	82.8	86.5	90.1	95.9	88.8	88.2	90.5	95.9
Supply Airflow Rate (CFM)	1,510	1,510	1,480	1,500	1,510	1,490	1,470	1,510	1,510	1,510	0	1,460	*1,020*	*550*	*0*
Exhaust Airflow Rate (CFM)	1,310	1,310	1,330	1,310	1,320	1,280	1,350	1,300	1,320	1,340	210	1,330	950	*570*	*210*
Room Capacity (tons; 80°F reference)	1.61	1.45	1.23	1.07	0.88	0.95	0.75	0.31	0.22	0.24	0.00	1.22	0.89	0.54	0.00
Room EER (Btu/Wh, 80°F reference)	14.2	12.8	11.3	9.4	7.7	9.1	6.9	2.8	2.0	2.2	0.0	11.0	10.8	10.5	0.0
Sensible Cooling of Intake Air (tons)	1.62	2.76	3.73	1.07	2.14	3.46	4.39	1.61	2.75	4.08	0.00	3.68	2.61	1.46	0.00
Intake Air EER (Btu/Wh)	14.2	24.4	34.4	9.3	18.9	32.9	40.5	14.3	25.3	37.1	0.0	33.2	31.8	28.4	0.0
Makeup Water Usage (gph)	11.9	13.7	11.7	4.8	6.9	19.0	1.3	7.1	1.4	11.6	13.4	8.6	18.9	10.9	13.4
Water Evaporation Rate (gph)	4.0	6.7	8.8	2.8	5.3	7.9	10.8	4.0	6.7	10.1	1.5	8.9	6.3	4.0	1.5
Power Consumption															
Total Unit Power (W)	1,367	1,357	1,299	1,376	1,356	1,260	1,302	1,348	1,305	1,319	246	1,327	984	620	246
Unit Power Factor	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.99	0.98	0.28	0.99	0.80	0.70	0.28
Total Unit CFM / W	2.06	2.08	2.16	2.05	2.09	2.20	2.16	2.08	2.17	2.16	0.84	2.10	2.00	1.81	0.84
Energy Efficiency Ratio (Btu/Wh)	14.17	12.85	11.35	9.37	7.75	9.05	6.88	2.77	2.05	2.22	0.00	11.00	10.83	10.51	0.00

Italicised airflow rates are below the recommended accuracy limit for the airflow chamber

Table 8: Coolerado Cooler™ Test Data (Continued)

Test Summary Information	Variable Resistance								Variable Voltage								
General																	
Date (all year 2005)	Feb 22	Feb 22	Feb 22	Feb 22	Feb 22	Feb 22	Feb 22	Feb 22	Feb 22	Feb 22	Feb 22	Feb 22	Feb 22	Feb 22	Feb 22	Feb 22	Feb 22
Start Time	9:17a	11:35a	10:02a	10:20a	11:21a	10:35a	10:50a	11:06a	12:58p	12:47p	12:34p	12:24p	1:07p	1:45p	1:17p	1:28p	1:37p
Duration (minutes)	33	12	15	12	12	12	12	12	8	9	9	9	8	8	9	8	7
Barometric Pressure (in. of Hg)	29.41	29.42	29.41	29.42	29.42	29.42	29.42	29.42	29.39	29.39	29.40	29.40	29.39	29.37	29.38	29.38	29.38
Nominal Test Conditions																	
Inlet Dry Bulb Temperature (°F)	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
Inlet Wet Bulb Temperature (°F)	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70
Fan Speed	1,590	1,580	1,580	1,590	1,580	1,590	1,590	1,590	1,470	1,510	1,550	1,580	1,600	1,610	1,630	1,640	1,650
Inlet Air Properties																	
Dry Bulb Temperature (°F)	100.0	100.0	100.0	99.9	100.0	100.2	100.7	99.7	100.0	100.0	100.1	99.5	100.0	100.1	100.0	100.0	100.0
Dew Point Temperature (°F)	52.4	52.5	52.8	53.2	52.6	54.0	54.5	52.5	49.7	50.2	51.6	51.5	55.7	53.4	54.0	53.9	54.7
Relative Humidity (%) - calculated	20.5	20.6	20.8	21.1	20.6	21.6	21.6	20.7	18.5	18.9	19.8	20.2	23.1	21.2	21.7	21.6	22.2
Wet Bulb Temperature (°F) - calculated	69.2	69.3	69.4	69.5	69.3	69.9	70.3	69.2	68.1	68.3	68.9	68.7	70.6	69.6	69.9	69.8	70.2
Wet Bulb Depression (°F)	30.8	30.8	30.6	30.4	30.8	30.2	30.4	30.6	31.9	31.7	31.2	30.8	29.4	30.4	30.1	30.2	29.9
Exhaust Air Properties																	
Dry Bulb Temperature (°F)	78.3	78.6	78.2	78.3	77.8	77.7	77.9	77.2	78.4	78.1	78.4	78.3	79.1	79.1	79.2	79.2	79.2
Dew Point Temperature (°F)	77.9	78.2	77.8	77.7	77.3	77.0	77.2	76.5	77.9	77.7	78.0	77.9	78.4	77.8	78.4	78.3	78.2
Relative Humidity (%) - calculated	98.7	98.7	98.6	98.0	98.3	97.7	98.0	97.7	98.4	98.7	98.8	98.5	97.7	95.9	97.4	97.1	96.6
Wet Bulb Temperature (°F) - calculated	78.0	78.3	77.9	77.8	77.4	77.2	77.4	76.7	78.0	77.8	78.1	78.0	78.6	78.2	78.6	78.5	78.4
Outlet Air Properties	**0.00**	**0.00**	**0.10**	**0.20**	**0.20**	**0.30**	**0.40**	**0.45**	<-- External Resistance								
Dry Bulb Temperature (°F)	72.4	72.5	72.2	71.5	71.6	71.6	72.0	70.7	71.4	72.0	72.5	71.9	72.7	73.8	73.0	73.3	73.6
Dew Point Temperature (°F)	52.0	52.4	52.4	52.6	52.3	53.6	54.1	52.3	49.6	50.2	51.5	51.0	54.3	52.7	53.3	53.3	53.5
Relative Humidity (%) - calculated	48.7	49.2	49.7	51.3	50.6	53.0	53.3	52.2	46.0	46.2	47.5	47.6	52.3	47.7	49.9	49.5	49.4
Wet Bulb Temperature (°F) - calculated	60.0	60.2	60.1	60.0	59.9	60.5	60.9	59.5	58.4	58.9	59.8	59.3	61.2	60.8	60.8	60.9	61.2
Performance																	
Exhaust Dry Bulb □T (°F)	-21.7	-21.4	-21.8	-21.6	-22.2	-22.5	-22.9	-22.5	-21.6	-21.9	-21.7	-21.1	-20.9	-21.0	-20.8	-20.8	-20.8
Exhaust Wet Bulb □T (°F)	8.8	9.1	8.5	8.3	8.2	7.2	7.1	7.6	10.0	9.5	9.3	9.3	8.0	8.5	8.7	8.7	8.3
Outlet Dry Bulb □T (°F)	-27.6	-27.5	-27.8	-28.4	-28.4	-28.6	-28.7	-29.1	-28.6	-28.1	-27.5	-27.5	-27.3	-26.3	-27.0	-26.8	-26.4
Outlet Dew Point □T (°F)	0.4	0.2	0.4	0.6	0.2	0.4	0.3	0.2	0.1	0.1	0.1	0.6	1.4	0.7	0.7	0.6	1.1
Wet-Bulb Effectiveness (%)	89.7	89.5	90.8	93.4	92.5	94.5	94.3	95.1	89.6	88.6	88.3	89.5	92.9	86.4	89.5	88.7	88.4
Supply Airflow Rate (CFM)	1,490	1,470	*1,410*	*1,330*	*1,330*	*1,270*	*1,220*	*1,170*	*1,340*	*1,400*	1,450	1,470	1,500	1,500	1,520	1,530	1,550
Exhaust Airflow Rate (CFM)	1,280	1,290	1,310	1,370	1,360	1,390	1,410	1,440	1,210	1,230	1,260	1,300	1,320	1,330	1,340	1,360	1,370
Room Capacity (tons; 80°F reference)	0.95	0.93	0.92	0.95	0.94	0.90	0.81	0.92	0.97	0.95	0.91	1.00	0.92	0.78	0.89	0.87	0.83
Room EER (Btu/Wh, 80°F reference)	9.1	8.7	8.8	9.0	8.8	8.5	7.7	8.7	10.0	9.5	8.8	9.4	8.4	7.1	7.9	7.5	7.0
Sensible Cooling of Intake Air (tons)	3.46	3.41	3.28	3.17	3.17	3.05	2.92	2.86	3.22	3.30	3.34	3.41	3.43	3.31	3.43	3.44	3.43
Intake Air EER (Btu/Wh)	32.9	31.9	31.4	30.2	29.7	28.9	27.8	27.2	33.4	33.0	32.3	32.0	31.4	30.2	30.4	29.7	28.8
Makeup Water Usage (gph)	19.0	5.7	11.4	3.4	17.2	18.9	16.2	1.3	1.3	8.6	8.9	11.9	3.7	1.3	1.3	1.3	1.3
Water Evaporation Rate (gph)	7.9	8.1	8.0	8.2	8.1	7.8	7.9	8.2	8.0	7.9	8.0	8.2	7.7	8.0	8.2	8.2	8.1
Power Consumption							Supply Voltage -->		**95.2**	**99.8**	**105.1**	**109.6**	**115.0**	**115.2**	**120.7**	**124.9**	**130.2**
Total Unit Power (W)	1,260	1,280	1,256	1,260	1,281	1,266	1,264	1,263	1,157	1,199	1,243	1,278	1,312	1,315	1,355	1,387	1,429
Unit Power Factor	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.97	0.97	0.98	0.98	0.99	0.99	0.98	0.98	0.98
Total Unit CFM / W	2.20	2.16	2.17	2.14	2.10	2.10	2.08	2.07	2.20	2.19	2.18	2.17	2.15	2.15	2.11	2.08	2.04
Energy Efficiency Ratio (Btu/Wh)	9.05	8.70	8.80	9.01	8.76	8.50	7.70	8.74	10.03	9.45	8.76	9.38	8.37	7.14	7.87	7.50	6.96

Italicised airflow rates are below the recommended accuracy limit for the airflow chamber