Pacific Gas and Electric Company

PY2004 Emerging Technologies Program

Application Assessment Report #0401

Evaluation of a Thick Media Evaporative Cooler

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EXECUTIVE SUMMARY

Tests were conducted on an advanced whole-house evaporative cooler with a 12-inch thick cellulose pad. The results were then compared against the performance of three previous test units, particularly a nearly identical unit with an 8-inch thick pad. The test units were all new, off-the-shelf systems, and were not models specially configured for rating purposes. The goal was to determine if the thicker pad would provide enough of a performance gain to recommend its use by consumers, and whether to justify increased incentives for it.

The tests were done using identical conditions and methods as were used in the previous tests, documented in Pacific Gas and Electric Company PY2003 Emerging Technology Application Assessment Report #0307, "Evaluation of Advanced Evaporative Cooler Technologies" (Reference 8). This test unit has a two-speed primary fan motor like the others, and test data were collected at both speed settings.

In relation to the unit with the 8-inch pad (ECU2), this unit (ECU4) showed a significant gain in effectiveness (16%) for a moderate decrease in airflow (8%) at high speed. In the previous tests, the addition of an indirect evaporative precooler to the unit with the 8-inch media (ECU3) showed a larger increase in effectiveness (22%), but that came with a much greater decrease in airflow (27%), and ultimately a decrease in the system's capability for cooling a supply of outside air.

The effectiveness was nearly constant over the range of test conditions, and varied much less than that for the other three units. At high fan speed, the effectiveness range was 82 to 85% (averaging 84%), and at low speed, the range was 86 to 88% (averaging 87%).

In comparison with the unit with 8-inch media at an outdoor air condition of 100°Fdb and 70°Fwb, this unit showed an increase in capacity to cool a space to 80°F of 0.59 tons (74%) due to its cooler supply temperature. The unit with the indirect evaporative precooler showed a larger increase in indoor cooling capacity of 0.76 tons (96%). However, on an incremental cost basis (~$90 for the 12-inch media versus ~$1,100 for the indirect evaporative precooler) the capacity increase was at $150/ton of increase versus $1,450/ton for the precooler. The average test results for this unit along with a comparison with the previously tested evaporative cooling units (ECUs) is 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: Averaged Unit Performance

High speed at 0" w.g. outlet resistance

Test Unit	ECU1	ECU2	**ECU4**	ECU3	SEER 12
Mfr. Industry Standard Rating (cfm)	4,500	4,800	**4,800**		3-Ton A/C
Intake Airflow [1] (cfm)	3,790	3,320	**3,070**	2,440	1,200
Total Unit Power (W)	806	737	**726**	939	3,490
Effectiveness	42.0%	72.7%	**84.1%**	88.8%	N/A
Direct Section Pump Power (W)	27	36	**52**	36	
Indirect Section Pump Power (W)				49	
Indirect Section Fan Power (W)				277	
Indirect Section Airflow [1] (cfm)				610	
Approximate Cost	$510	$730	**$820**	$1,830	$2,800

Capacity measures with ~100°F_{db}/70°F_{wb} intake air (tons)

Test Unit	ECU1	ECU2	**ECU4**	ECU3	SEER 12
Sensible Room Capacity [2]	– [5]	0.62	**1.39**	1.56	
Sensible Intake Air Capacity [3]	4.5	6.2	**6.5**	5.7	
Total Intake Air Capacity [4]	-0.3	-0.5	**-0.4**	1.8	3.0
Change in moisture	+32%	+60%	**+64%**	+48%	-13%
Water consumption (gph)	10	13	**14**	14	-1
0.3" w.g. outlet resistance					
CEC Title-20 ECER (Btu/Wh)	– [5]	23.2	**25.7**	19.0	10.3 ARI "A" EER

[1] Supplied airflow referenced to the intake density.
[2] Sensible Room Capacity ≈ 1.08 × CFM × (80°F – T_{supply})/12,000
[3] Sensible Intake Air Capacity ≈ 1.08 × CFM × (T_{intake} – T_{supply})/12,000
[4] Total capacity is the sensible cooling less the latent heat added through evaporation.
[5] System did not achieve an outlet dry bulb temperature below 80°F at these outside conditions.

ECU1: Baseline "swamp cooler" with thin evaporative media
ECU2: Advanced direct evaporative cooler with 8"-thick cellulose pad
ECU3: ECU2 with add-on indirect evaporative precooler
ECU4: Advanced direct evaporative cooler with 12"-thick cellulose pad

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 now 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.

Traditional evaporative coolers use thin pads made out of aspen wood fiber or plastic. The problem with these systems is that the contact period between the air and water is short, resulting in low evaporation effectiveness. A significant improvement in cooling effectiveness is achieved in newer coolers that employ thick cellulose pads as the evaporative media, such that a single unit can be used to effectively cool an entire house. Beginning in 2002, PG&E has provided a rebate program for these Advanced Whole House Evaporative Coolers (AWHECs) as an energy-saving alternative to conventional air conditioners. (PG&E does not provide rebates for the traditional type of evaporative cooler.)

While it is well understood that the newer media does create a more effective cooler, there is insufficient information regarding how thick the media should be to achieve the desired cooling result.

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 evaporative air precoolers for the condenser air. The most recent tests were conducted at the end of 2003 on three sample evaporative cooling systems to evaluate their relative performance, and the results of those tests are documented in Pacific Gas and Electric Company PY2003 Emerging Technology Application Assessment Report #0307, "Evaluation of Advanced Evaporative Cooler Technologies" (Reference 8).

Objectives

The objective of this project was to assess the relative system performance of an evaporative cooling unit (ECU) with the thickest available pad, and whether this should have some effect on the available rebate. The performance parameters include:

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

as a function of the variables:

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

The test system (ECU4) is an advanced direct evaporative cooler, with its intake on only one side through a 12-inch thick cellulose pad. This test unit was obtained from the same manufacturer's product line as the test units used in the previous Emerging Technology Application Assessment documented in Report #0307. The previously evaluated systems included:

- ECU1: a traditional cooler with thin evaporative media (the unit has a side discharge rather than bottom, so it had pads on three sides rather than all four),
- ECU2: an advanced direct evaporative cooler, with its intake on only one side through an 8-inch thick cellulose pad,
- ECU3: ECU2 plus an add-on indirect evaporative precooler stage.

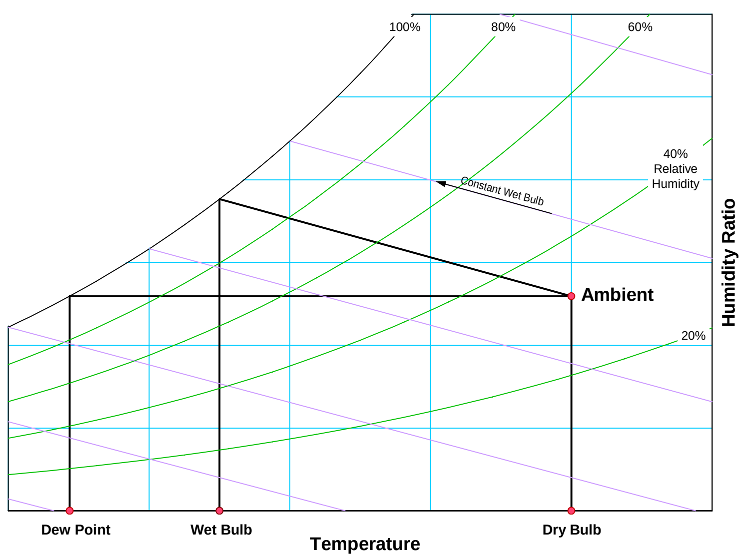
ECU4 is rated at the same nominal air delivery rate as ECU2, and uses the same size primary air fan. The only differences are that the "wet" section of ECU4 is longer to accommodate the thicker media, and it also uses a larger water circulation pump. To achieve more consistency in determining the effect of media thickness, the wet section of the new cooler was detached and mated with the fan section from ECU2.

EXPERIMENTAL DESIGN AND PROCEDURE

Performance Characteristics

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 air) against temperature. *Figure 1* 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* (temperature change only without a change in moisture content), the conditions of the air in the chart move along a horizontal line until it again reaches 100% relative humidity. The temperature at this point is called the "dew point" temperature.

Figure 1: Psychrometric Chart



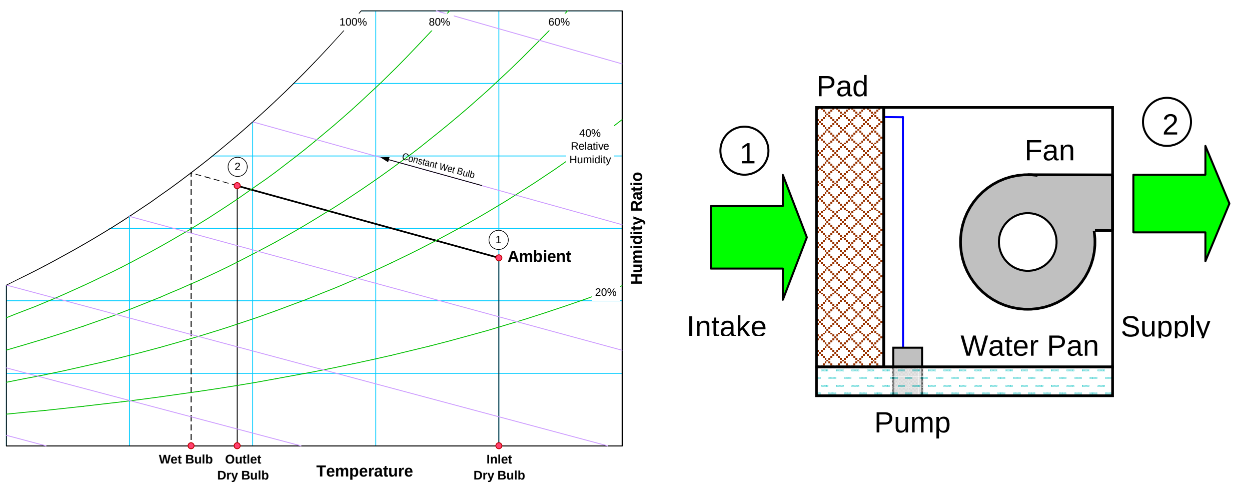
Direct evaporative coolers may be described as a constant wet-bulb temperature process (although there is some minor sensible heat gained from the fan). Their performance is related to how close the dry-bulb temperature of the outlet air approaches the intake wet-bulb temperature. The wet bulb "effectiveness" of an evaporative cooler is defined as follows:

:

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

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. *Figure 2* 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 2: Direct Evaporative Cooler Process



Ideally, the performance numbers obtained from the testing can be used to compare the performance against other cooling systems, not just other evaporative coolers. 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 capacity for an evaporative cooler is not defined in any American standard, and it is open to some interpretation.

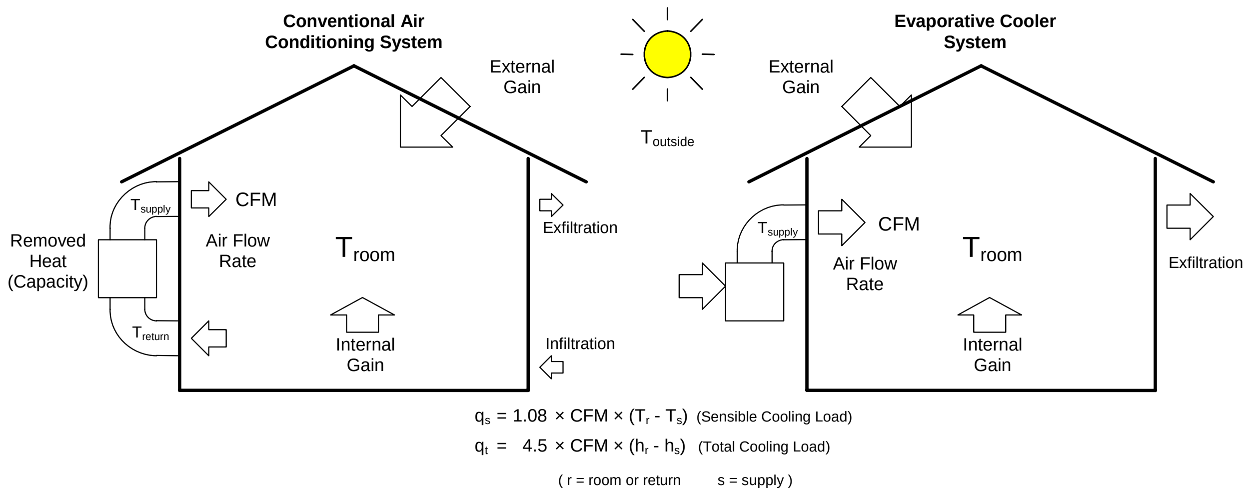
An air conditioning system is designed to cool the air contained in a space, removing the heat (sensible load) and moisture (latent load) gained from various sources, while recirculating the same air repeatedly. The cooling capacity is measured at the evaporator coil as the product of the air mass flow 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. Fortunately, 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. The furnace fan may still need to be operated to assure adequate circulation throughout a building if the evaporative cooler is not connected to ductwork.

The thermal load in a space served by an evaporative cooler should actually 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 (air leaks). 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. It is also often considered acceptable to ignore the latent load in determining the capacity for an evaporative cooler since any moisture gains in a space will be exhausted.

A graphical description of the difference between the two types of systems and the definition of cooling capacity is shown in *Figure 3*:

Figure 3: Definition of Cooling Capacity



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

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

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. (In the actual test results, measured values of density and specific heat are used rather than the nominal 0.075 lb/ft³ and 0.24 Btu/lb-°F.)

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, which substitutes in the equation for effectiveness solved for the supply air temperature, as follows:

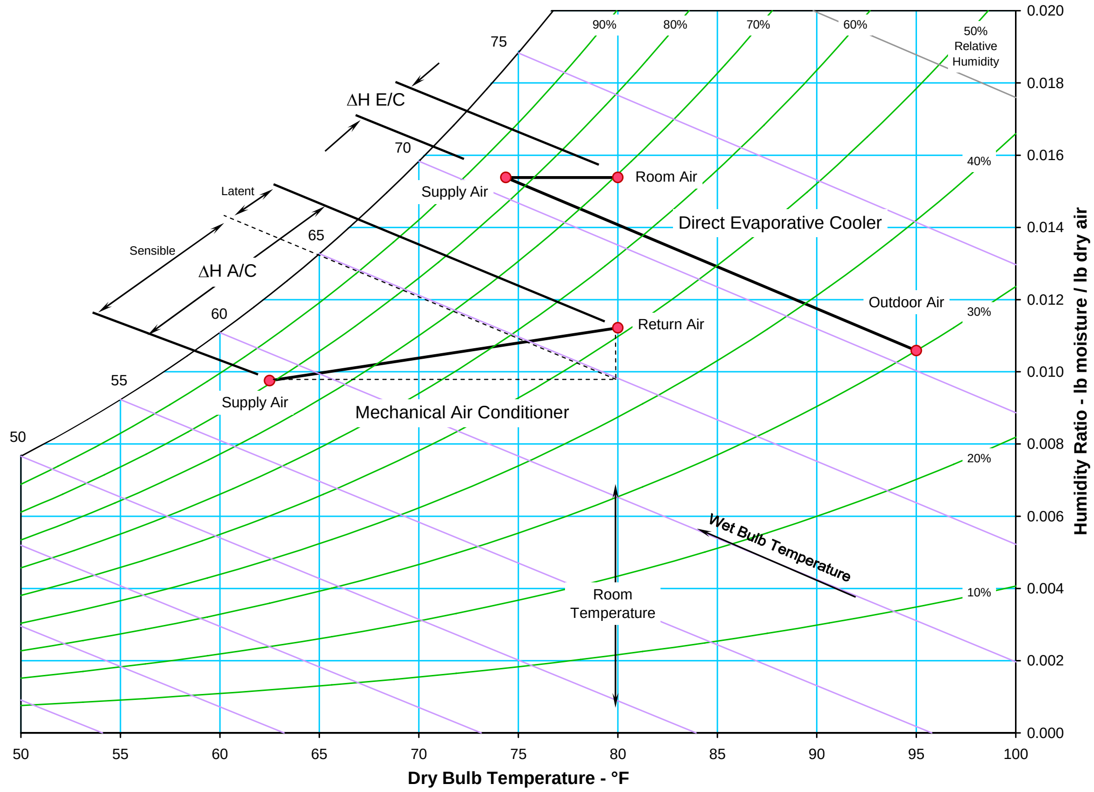
$$ECER = 1.08 \times CFM \times (Tdb_{room} - (T_{db,\,in} - \varepsilon \times (T_{db,\,in} - T_{wb,\,in}))) / W$$

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).

The difference in capacity between an evaporative cooler and an air conditioner can also be represented on a psychrometric chart as shown in *Figure 4*. The conventional air conditioner is shown as taking room air at 80°F dry-bulb and 67°F wet-bulb (ARI rating conditions) and discharging back into the room at 62.5°F. The example evaporative cooler takes in outside air at 95°F and 30% relative humidity, and is assumed to have an 85% effectiveness. The chart shows that for an evaporative system to handle the same cooling load, it must have a greater mass flow rate because the enthalpy difference (ΔH) is much smaller. It also shows that the resulting humidity in the space will be much greater with the evaporative cooler than for the air conditioner under these conditions.

Figure 4: Comparison of Capacity between Air Conditioner and Evaporatvie Cooler

Psychrometric Chart showing the Enthalpy Change used to determine Capacity

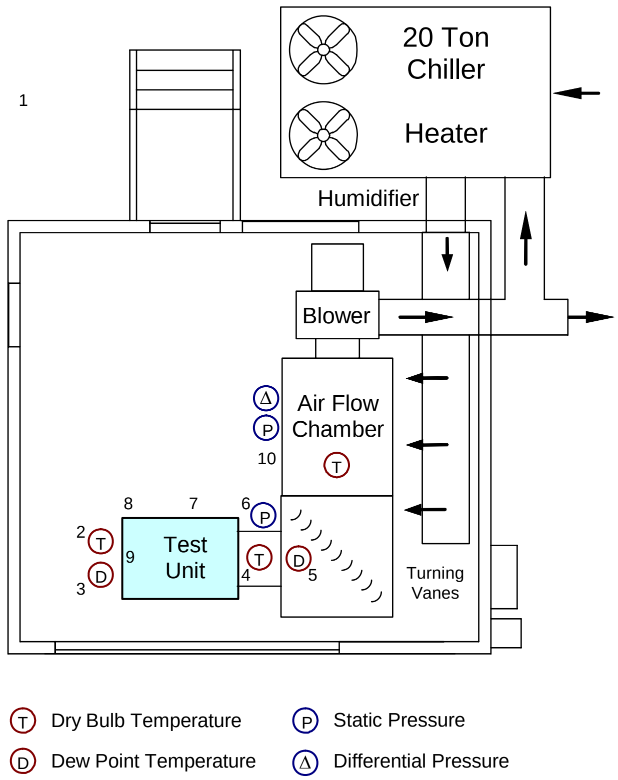


Test Facility

Figure 5 shows a layout of the test facility configured for the single evaporative cooler testing. The test unit was placed in an environmentally-controlled room, which was conditioned by a 20-ton heat pump / air conditioner, a variable-output resistance heater, and a humidifier. Outside air dampers allowed for some recirculation of the test unit exhaust to control supplied air humidity. The outlet of the test unit was connected to an airflow measurement station, consisting of a sealed chamber with several flow nozzles, designed in accordance with ASHRAE specifications (per References 3 and 4). The chamber has four 9-inch nozzles, and can measure flow rates between 1,300 and 12,400 cfm. A variable-speed blower on the outlet of the chamber is set to maintain the desired outlet static pressure and compensate for the added resistance of the measurement system and ductwork.

Figure 5: 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 direct evaporator cooler test standard (Reference 5). The following is a listing of the measurements taken and the instruments used for the direct evaporative cooler test:

1. Barometric pressure, using an electronic barometer.
2. Entering air dry-bulb temperature, using four resistance temperature devices (RTDs).

   The sensors were mounted about 2” in front of the intake at the center of equal face areas.
3. Entering air dew-point temperature, using a chilled mirror sensor.

   A sampling system mixed air from eight points around the intake area through non-hygroscopic copper tubing to the sensor.
4. Leaving air dry-bulb temperature, using four RTDs inserted through the duct wall.

   The tips of the inserted probes were positioned at the center of equal-area sections of the duct. The location of the sensors was far enough downstream from the cooler outlet to allow for adequate mixing and an even flow profile, yet not so far as to incur heat gain from the outside. The duct was also insulated.
5. Leaving air dew-point temperature, using a chilled mirror sensor and a sampling tube.
6. Outlet 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. Total power using a power meter; and pump power using a watt transducer.

8. Make-up water flow rate, using a low-range thermal flow meter with an analog output and a high-range positive displacement flow meter with a pulse output connected in series.

   The two flow meters had an overlapping range that allowed them to be directly compared. The thermal meter could not read above about 12.3 gallons per hour (GPH), and the positive displacement could not read below about 4 GPH.

9. Water temperature in the evaporative cooler sump near the pump intake, using two immersed RTDs.

10. Airflow rate, using a nozzle chamber and measurements of differential and inlet static pressure and inlet temperature.

All of the temperature instruments were calibrated 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 (110°F). The raw measurements were adjusted to match the reading from a secondary temperature standard RTD placed in the same environment. The transmitters for the differential and static pressure measurements were calibrated using a water manometer with a micrometer adjustment, accurate to below 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 fan were also provided by the NI system. The RTDs were all connected to a Fluke Helios data logger, which connected to one of the computer's serial ports. The two dew point sensors were also connected through serial ports. Total power measurements were made with a Yokogawa power meter, which communicated through a GPIB interface.

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. The program also received the readings from the two chilled mirror sensors 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.

Test Conditions

The ASHRAE test standard for direct 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 the following limits for the environmental conditions:

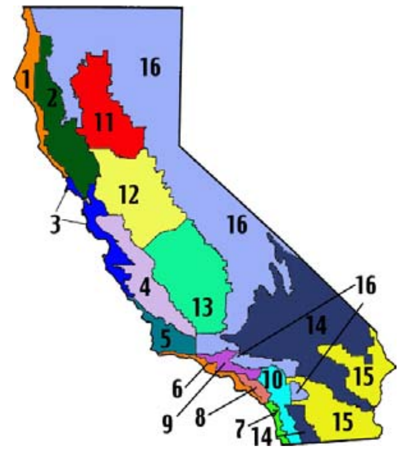
- A maximum dry-bulb temperature of 115°F
- A minimum wet-bulb temperature of 41°F
- A minimum wet-bulb depression (difference between dry and wet-bulb temperatures) of 25°F

A third test standard (Australian) 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)

Figure 6: Title 24 Climate Zones



This test unit was subjected to the same set of conditions as were used in the previous evaporative cooler tests to allow for an easy comparison. 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). 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 per year). 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 per year), 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 7*) 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:

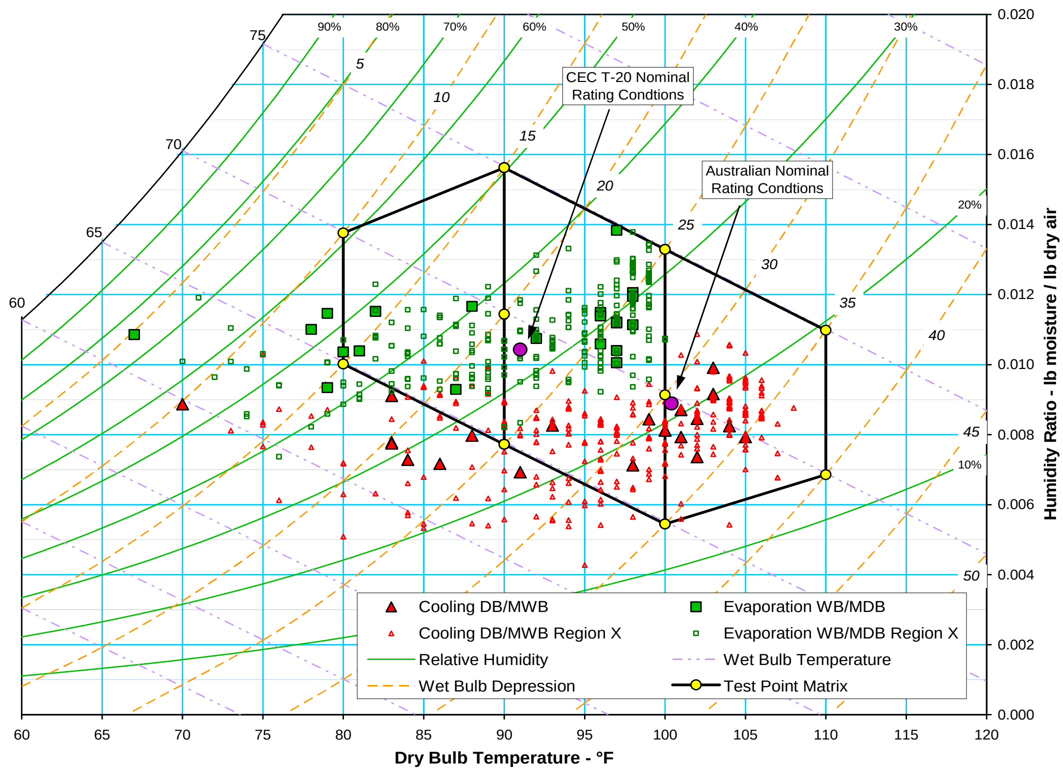
Table 3: Test Point Matrix

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

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 *Figure 7*. 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 7: 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 control was set to maintain a zero static pressure on the outlet (except during airflow sensitivity tests).
4. 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.
5. Since the test unit has a two-speed fan, the fan speed was changed after the initial set of data was recorded, and the test was continued at the second speed with the same environmental conditions.
6. The room conditioning system was adjusted to the next set of conditions, and steps 4 and 5 were repeated.

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

Table 4 lists several parameters averaged over all of the tests for each unit that are not particularly affected by the intake air conditions. This is an expansion of the table included in the earlier report to compare with the results for ECU4. The results for ECU4 are inserted between those for ECU2 and ECU3 because that is where most of the performance factors fit relative to the other two. Also shown is the manufacturer's industry standard rating for airflow for the three direct units. The primary fans had both high and low speed settings, and the results are shown for each setting. Included with the low speed results is the relative magnitude of the airflow and power compared with the high-speed results. The results for the direct units show that at the lower fan speed, they provide about 2/3 the airflow for half the power relative to their high-speed setting.

The table also includes the comparative values for a conventional air conditioner of a size appropriate for a square footage in the sizing recommendations for these evaporative coolers. The example unit is a 3-ton SEER 12 unit operating under ARI "A" test conditions (evaporator inlet: 80°F_{db} and 67°F_{wb}, condenser inlet: 95°Fdb), and the results are from prior tests in this facility. The key finding is that the evaporative units consume less than one quarter the power of the air conditioner.

Also included as an addendum are measures of system cooling capacity at a particular operating condition, specifically 100°F_{db}/70°F_{wb} (which approximates the Australian rating conditions) with an outlet resistance of zero inches of water. Included is the sensible capacity referenced to a room temperature of 80°F defined earlier, plus another commonly used measure of capacity, which is the sensible cooling of the intake air:

$$\textit{Intake Air Capacity (Btu/hr)} \approx 1.08 \times CFM \times (Tdb_{intake} - Tdb_{supply})$$

Energy efficiency ratios (EER) for each are calculated by dividing by the total power draw. Finally, the calculated CEC Title-20 ECER is listed by itself.

Table 4: Averaged Results for Airflow and Power

0" w.g. outlet resistance

Test Unit	ECU1	ECU2	ECU4	ECU3	SEER 12
Mfr. Industry Standard Rating (cfm)	4,500	4,800	**4,800**		3-Ton A/C
High Speed					ARI "A"
Intake Airflow [1] (cfm)	3,790	3,320	**3,070**	2,440	1,200
Total Unit Power (W)	806	737	**726**	939	3,490
Total Unit Power Factor	0.77	0.62	**0.56**	0.60	0.96
Effectiveness	42.0%	72.7%	**84.1%**	88.8%	N/A
Low Speed					
Intake Airflow [1] (cfm)	2,540	2,120	**1,940**	1,480	
Total Unit Power (W)	394	360	**366**	644	
Total Unit Power Factor	0.71	0.58	**0.55**	0.62	
Effectiveness	50.0%	77.6%	**87.2%**	95.3%	
Direct Section Pump Power (W)	27	36	**52**	36	
Indirect Section Pump Power (W)				49	
Indirect Section Fan Power (W)				277	
Indirect Section Airflow [1] (cfm)				610	
Sensible capacity measures with ~100°F$_{db}$/70°F$_{wb}$ intake air and 0" w.g. outlet resistance					
High Speed					ARI "A"
Room Capacity (tons) [2]	– [4]	0.62	**1.39**	1.56	3.0
Room EER (Btu/Wh) [2]	– [4]	10.1	**22.9**	19.9	10.3
Intake Air Capacity (tons) [3]	4.5	6.2	**6.46**	5.7	
Intake Air EER (Btu/Wh) [3]	66.5	101.1	**106.5**	73.1	
Low Speed					
Room Capacity (tons) [2]	– [4]	0.70	**1.00**	1.14	
Room EER (Btu/Wh) [2]	– [4]	23.2	**32.6**	21.3	
Intake Air Capacity (tons) [3]	3.43	4.25	**4.23**	3.69	
Intake Air EER (Btu/Wh) [3]	104.5	141.7	**138.3**	68.8	
0.3" w.g. outlet resistance					
CEC Title-20 ECER (Btu/Wh)	– [4]	23.2	**25.7**	19.0	10.3

[1] Supplied airflow referenced to the intake density.
[2] Room Capacity ≈ 1.08 × CFM × (80°F – T_{supply})/12,000
[3] Intake Air Capacity ≈ 1.08 × CFM × (T_{intake} – T_{supply})/12,000
[4] System did not achieve an outlet dry bulb temperature below 80°F at these outside conditions.

ECU1: Baseline "swamp cooler" with thin evaporative media
ECU2: Advanced direct evaporative cooler with 8"-thick cellulose pad
ECU3: ECU2 with add-on indirect evaporative precooler
ECU4: Advanced direct evaporative cooler with 12"-thick cellulose pad

Table 5 lists the measured unit supply temperatures as a function of the inlet dry and wet-bulb temperatures and fan speed. When more than one test was done at a particular condition, the results were averaged. These results are intended to show the range of discharge temperatures that would be provided by the test unit under different environmental conditions. Similar tables for the other units were listed in the earlier report.

Table 6 lists the resulting unit effectiveness in the same format as *Table 5*. The results indicate that the unit shows an improvement in effectiveness when the fan speed is decreased. This is an expected result as the lower airflow rate increases the contact time with the wetted pads and thereby increases the evaporation, and is consistent with the results for the other units. The test unit shows less sensitivity to the inlet air conditions than the other three, with an effectiveness range of less than three percentage points over all of test conditions at either speed.

Table 5: Supply Temperatures (°F)

ECU4 (High Speed)

Inlet Tdb	Inlet Wet Bulb Temperature (°F)		
(°F)	65	70	75
80	68	71	
90	69	73	78
100	71	75	79
110		76	80

ECU4 (Low Speed)

Inlet Tdb	Inlet Wet Bulb Temperature (°F)		
(°F)	65	70	75
80	67	71	
90	69	72	77
100	70	74	78
110		75	80

Table 6: Effectiveness

ECU4 (High Speed)

Inlet Tdb	Inlet Wet Bulb Temperature (°F)		
(°F)	65	70	75
80	83%	82%	
90	84%	85%	84%
100	83%	85%	85%
110		85%	84%

ECU4 (Low Speed)

Inlet Tdb	Inlet Wet Bulb Temperature (°F)		
(°F)	65	70	75
80	86%	86%	
90	87%	87%	87%
100	86%	88%	88%
110		88%	87%

Figure 8 is a chart of several evaporative cooler performance parameters as a function of the supplied airflow rate (referenced to the intake density). This chart is presented here rather than in the Appendix because it is the recommended method for presenting the results from the ASHRAE test standard. Included on the chart are the measurements of effectiveness, power, and outlet static pressure that the ASHRAE standard requires. Also included are the corresponding measures of the CEC Title-20 ECER, although the only value that matters is when the outlet static pressure at 0.3" w.g., which is emphasized in the chart. The results in the chart show that as the outlet static pressure is increased, the airflow, power, and ECER decrease, while its effectiveness remains almost constant, although it does appear to reach a peak at 0.3" w.g. resistance.

Figure 8: Performance versus Airflow Rate

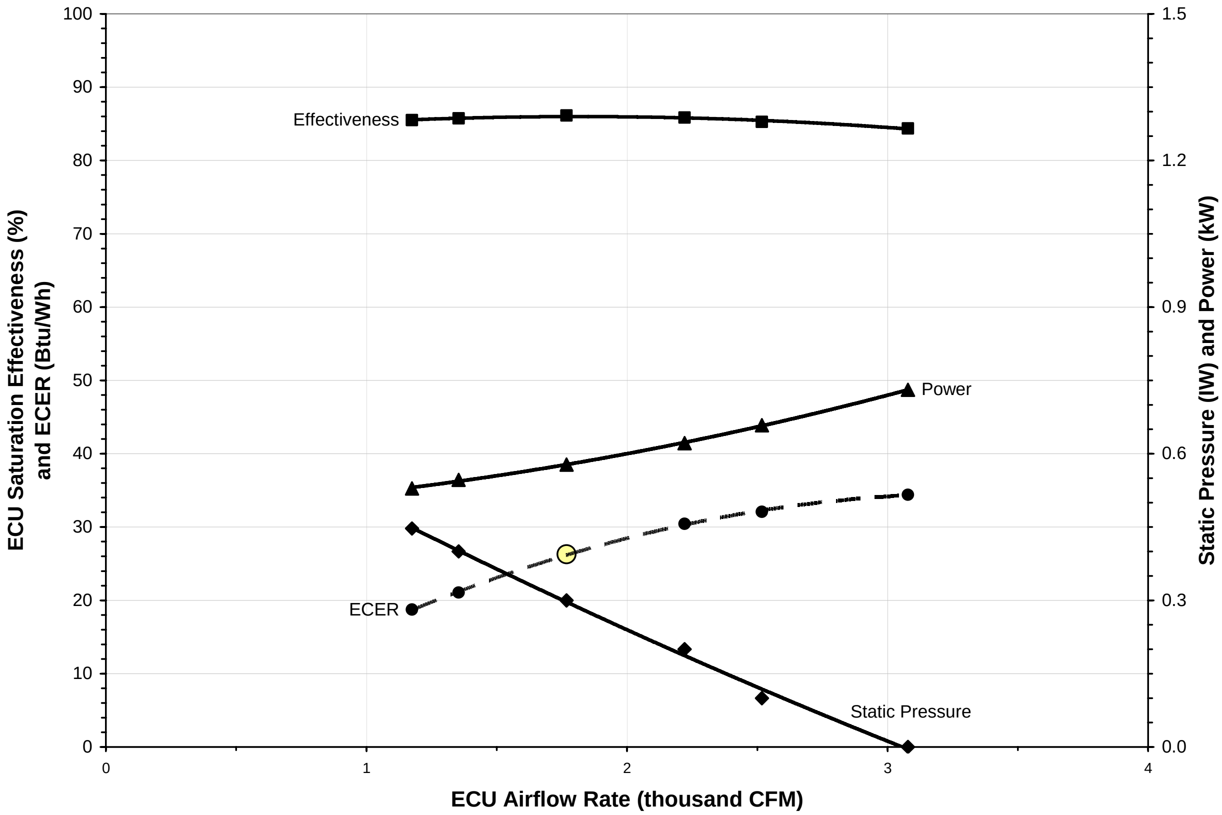


Figure 9 and *Figure 10* (in the Appendix) are charts of power consumption as a function of primary airflow rate for all of the test units, but with the results for ECU4 emphasized. (The airflow rate is the measured outlet airflow adjusted to the density of the intake air, as required by Reference 5.) The first figure shows the overall unit energy consumption, while the second shows only the power for the primary air fan. The data points in each chart are indicated as either being at the high or low speed setting of the fan, or the result of increasing the external resistance on the unit. ECU4 with its 12-inch media had more resistance to airflow than ECU2 with the 8-inch media, so the airflow was reduced. Total power consumption was slightly reduced at high speed and not at all at low speed due to the power savings from lower airflow being offset by the increase in circulation pump power (since ECU4 uses a larger pump motor). The second chart better shows the effect of increasing flow resistance on reducing the primary fan power since it removes the pump power. In this chart, the results for ECU2, ECU3, and ECU4 all appear to be aligned since they all used the same primary fan.

Figure 11 and *Figure 12* show basically the same information as the previous two charts, except that instead of power, the airflow divided by the power is graphed to give an indication of the fan efficiency (in cfm/W). Again, the results for the primary fan in ECU2, ECU3, and ECU4 appear to be aligned along the curves of increasing resistance to flow. It does not appear to make any significant difference if the resistance is upstream of the fan (increased pad friction) or downstream (increased duct friction or back pressure).

Figure 13 plots the overall system power factor as a function of airflow rate (a chart that was not included in the previous report). The results show that for all of the units, power factor tends to decrease with decreasing flow, and that ECU4 had the poorest power factor of all of the test units. Since the primary fans were identical for all but ECU1, the reason the power factor for ECU4 was the lowest was probably related to the water circulation pump. Motor power factor typically decreases sharply with decreasing shaft power rating, but the effect of the low pump power factor on the total was more apparent for ECU4 because this pump was larger than those in the previously tested evaporative coolers and represented a greater fraction of the total power consumption (7.2% for ECU4 versus 4.9% for ECU2). The low power factor is a trait common to many fractional horsepower induction motors, and is something that could be improved. The reactive power does no real work in the motor, but can contribute to significant heat generation. Since the motors are located in the air stream, the heat produced by the motor will be delivered to the conditioned space, degrading the performance of the cooler.

Figure 14 is a chart of overall system effectiveness as a function of airflow rate for all of the test units, which was also not included in the previous report. Low speed and high speed results for each unit are enclosed in ovals. This chart shows how ECU4 had the least variation in system effectiveness despite wide differences in the intake air conditions. It also shows how the performance of ECU4 falls between those for ECU2 and ECU3: airflow is reduced slightly from ECU2, and the effectiveness is nearly as good as ECU3, at least at high speed.

Figure 15 and *Figure 16* show the data contained in *Table 6* in graphical form. All of the individual test points are plotted, without averaging the data for the same test conditions. 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). Like *Figure 14*, these charts show how consistent the effectiveness was for ECU4 in relation to the others.

Figure 17 and *Figure 18* examine the relative cooling capacity and overall system energy efficiency for ECU4 alone. These factors combine the effects of airflow rate and supply temperature to demonstrate the ability to cool off a space. Similar charts for the other units were included in the previous report. As discussed previously, the capacity is defined as the ability of the unit to maintain a space at 80°F. Thus, if the supply temperature is above this value, the capacity of the unit is negative. (What this really means is the space will reach an equilibrium temperature somewhere above 80°F.) The capacity is listed in tons (12,000 Btu/hr) and the energy efficiency ratio (or EER, which is capacity divided by the total unit power) is listed in Btu/Wh. The results are graphed as a function of entering dry-bulb temperature, and grouped by entering wet-bulb temperature and fan speed. The results show that this unit will be able to provide some cooling effect under all but the highest combined temperature and humidity condition.

Figure 19 shows an alternative way of considering the cooling capacity: the ability of the test unit to sensibly cool the outside air supply. In this case, the capacity (again given in tons) is calculated by taking the temperature difference between the intake and supply air dry-bulb temperatures, and multiplying by the air mass flow rate and specific heat. The calculated values for some of these points are also included in *Table 4* under "Sensible Cooling of OA". The results were then graphed as a function of the entering air wet-bulb depression. With this view, ECU4 shows the best performance of all of the test units at high speed with a combination of adequate airflow and high effectiveness, but it is virtually identical to ECU2 at low speed. This may be a better measure to evaluate the relative performance of an evaporative cooler than the other definition because it takes the "fixed" interior space condition out of the equation. By this chart, the outside air cooling capability of ECU4 is about 5% better than ECU2. This should allow ECU4 to run less to provide the same cooling to a space as ECU2.

Figure 20 examines the sensitivity of ECU4 to increasing the backpressure, or the external resistance to flow. Similar charts for the other units were provided in the previous report. Included in the chart are the measurements of airflow rate, total unit power, effectiveness, and outside air capacity (the same factor as graphed in *Figure 17*), for both high and low speed settings of the primary fan. The values are graphed as the relative magnitude compared with the measured parameters with no backpressure (0" of water column), the values of which are given in the chart legend. The intake air condition was maintained at 100°F_{db} and 70°F_{wb}. The results show that there is a decrease in airflow, capacity, and power with increasing resistance, but that the effectiveness was only slightly increased.

The last two charts (*Figure 21* and *Figure 22*) examine the relative water consumption rates of all the test units. The first chart displays the flow rate recorded by the series flow meters installed in the makeup water line, and shows the total consumption for both evaporation and bleed off. A bleed off line was required to reduce the buildup of dissolved solids as water is evaporated. The line was attached to the discharge of the water circulation pump, and its flow was controlled with a clamp. After an initial set of tests, the bleed flow was observed to be lower than recommended at only about 1 GPH, and was subsequently increased to about 5 GPH. This is the cause for the jump in results on this chart. The results are graphed as a function of the wet-bulb depression, and the charts show a general increase in water consumption as the wet-bulb depression rises. Reducing the fan speed reduced the evaporation water consumption, but not the bleed flow.

The second figure looks at the water evaporation rate only, as determined from measurements on the air side of the process. It is determined by taking the moisture (humidity ratio) rise from intake to supply, and multiplying by the air mass flow rate. To achieve its high effectiveness, this unit had the highest water consumption rates of all the test units at high speed, but had an identical consumption rate to ECU2 at their low speeds.

The results from this graphical analysis are summarized in *Table 7*. The table lists the slopes of the graphed trend lines to indicate the correlation between water consumption and the entering wet-bulb depression.

Table 7: Summary of Water Consumption Rates

Test Unit	GPH	GPH per °F of Wet-bulb Depression				Approx. Total GPH Usage at 100°Fdb/70°Fwb
	Make-up Water Flowmeter Measurements			Air-side Measurements		
	Bleed Flow	High Speed	Low Speed	High Speed	Low Speed	High Speed
ECU1	3.9	0.209	0.162	0.167	0.135	10
ECU2	6.0	0.292	0.209	0.293	0.198	13
ECU3	–	–	–	0.293	0.213	14
ECU4	**0.95** **5.30**	**0.302** **0.301**	**0.194** **0.202**	**0.302**	**0.198**	**10** **14**

CONCLUSIONS

This study investigated the effect of media thickness on the performance of advanced evaporative coolers. Tests were conducted on an advanced evaporative cooler with a 12-inch thick pad at various conditions of outside air temperature, humidity, unit fan speed, and external resistance, and the results were compared with those previously obtained for three other evaporative coolers. Some of the key findings are summarized below.

1. The increase in evaporative media thickness exchanged some airflow for increased effectiveness and a lower supply air temperature. Relative to an identical unit with 8-inch thick evaporative media at high fan speed (ECU2), the test unit with 12-inch thick media (ECU4) showed:
    - An increase in effectiveness of 15.7% (from 73% to 84%)
    - A decrease in airflow of 7.5% (from 3,320 cfm to 3,070 cfm)
    - A decrease in power consumption of 1.5% (from 737W to 726 W).
    - An increase in water consumption for evaporation as a function of wet bulb depression of 3.1%
    - An increase in outdoor air sensible cooling capability as a function of wet bulb depression of 4.6%

At low fan speed:

- Effectiveness increased by 12.4% (from 78% to 87%)
- Airflow decreased by 8.5% (from 2,120 cfm to 1,940 cfm)
- Power consumption increased by 1.7% (from 360W to 366 W).
- Water consumption for evaporation and outdoor air sensible cooling capability were unchanged.

2. With a large jump in effectiveness for a small decrease in airflow, the 12-inch media unit had a slightly higher capability for cooling a space, and showed the highest cooling capability of the four test units in terms of the sensible heat removed from the outside air supply. While ECU3 with the indirect evaporative precooler did have a greater effectiveness, it sacrificed too much airflow.

3. Comparing the cooling capacity of ECU4 and ECU2 at one outside air condition (100°Fdb and 70°Fwb):

   In terms of the capacity for sensibly cooling a space to 80°F:

   - The capacity of ECU2 is 0.62 tons at a power consumption of 737W; thus in one hour it can do 0.62 ton-hours of cooling to keep a space at 80°F, while using 0.737 kWh.
   - The capacity of ECU4 is 1.39 tons at a power consumption of 728W. To produce the same amount of cooling (0.62 ton-hours), it would only need to run for 0.45 hours and use 0.325 kWh.
   - Thus, for the same level of capacity, ECU4 saves 0.412 kWh/hour, or about 4.1¢/hour at 10¢/kWh.

   In terms of the outside air sensible cooling capability:

   - The capability of ECU2 is 6.2 tons at a power consumption of 737W; thus in one hour it does 6.2 ton-hours of outside air cooling and uses 0.737 kWh.
   - The capability of ECU4 is 6.5 tons at a power consumption of 728W. To produce the same amount of cooling (6.2 ton-hours), it would only need to run for 0.96 hours and use 0.700 kWh.
   - Thus, for the same level of outside air cooling, ECU4 saves 0.037 kWh/hour, or about 0.4¢/hour at 10¢/kWh.

   This difference in how the cooling capacity of an evaporative cooler is defined presents the difficulty in comparing how particular systems will operate to keep a space cool. At higher temperatures, any cooler will likely be running full time, and the interior space temperature will float depending on the gain on the space and the outdoor air cooling capability. More intensive computer modeling or monitoring of installed systems may be necessary to determine operating patterns and total cooling season savings.

4. Increasing the thickness of the evaporative media from 8-inches to 12-inches does appear to have an economic advantage in terms of cooling capacity and EER. However, since the cost difference between the two units was only $87, or an increase of 12% over the cost of the unit with 8-inch media (list prices of $822 versus $735), the performance increase is not sufficient to justify increasing the rebate from the current $300 for either of these two systems.

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Figure 9: Total Power versus Airflow

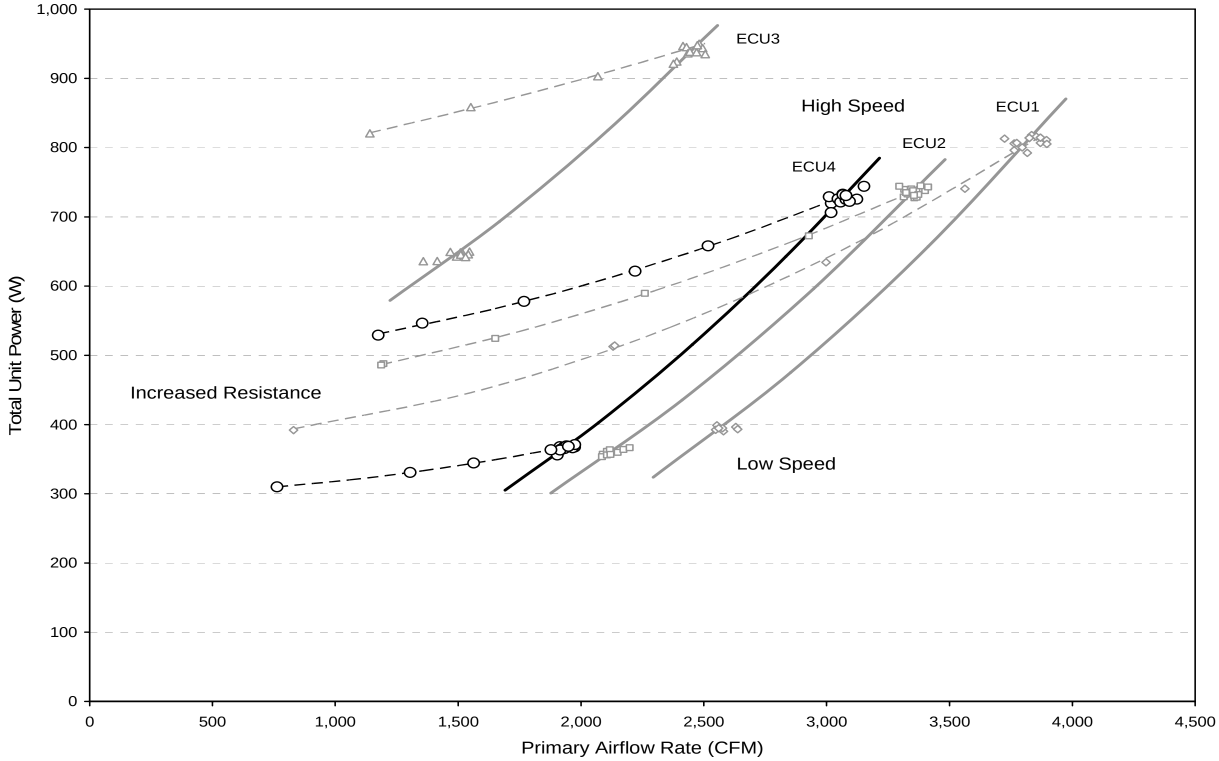

Figure 10: Primary Fan Power versus Airflow

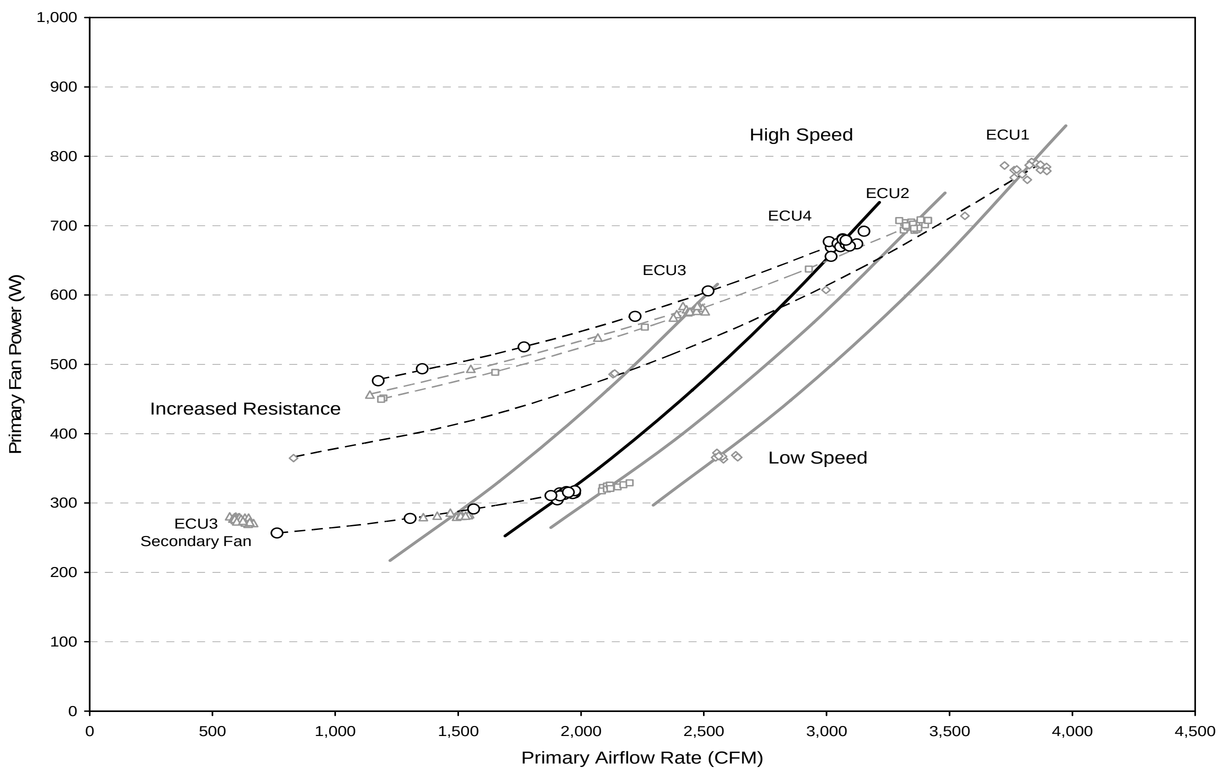

Figure 11: Total CFM / W versus Airflow

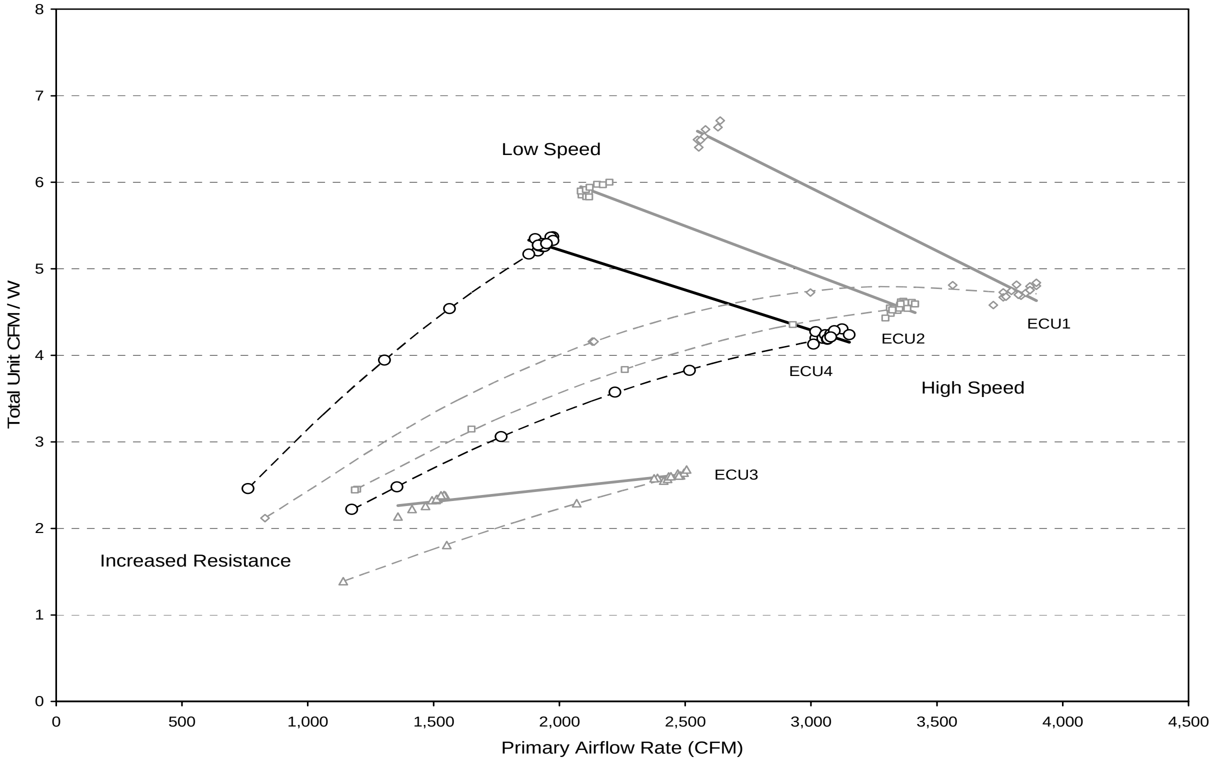

Figure 12: Primary Fan CFM / W versus Airflow

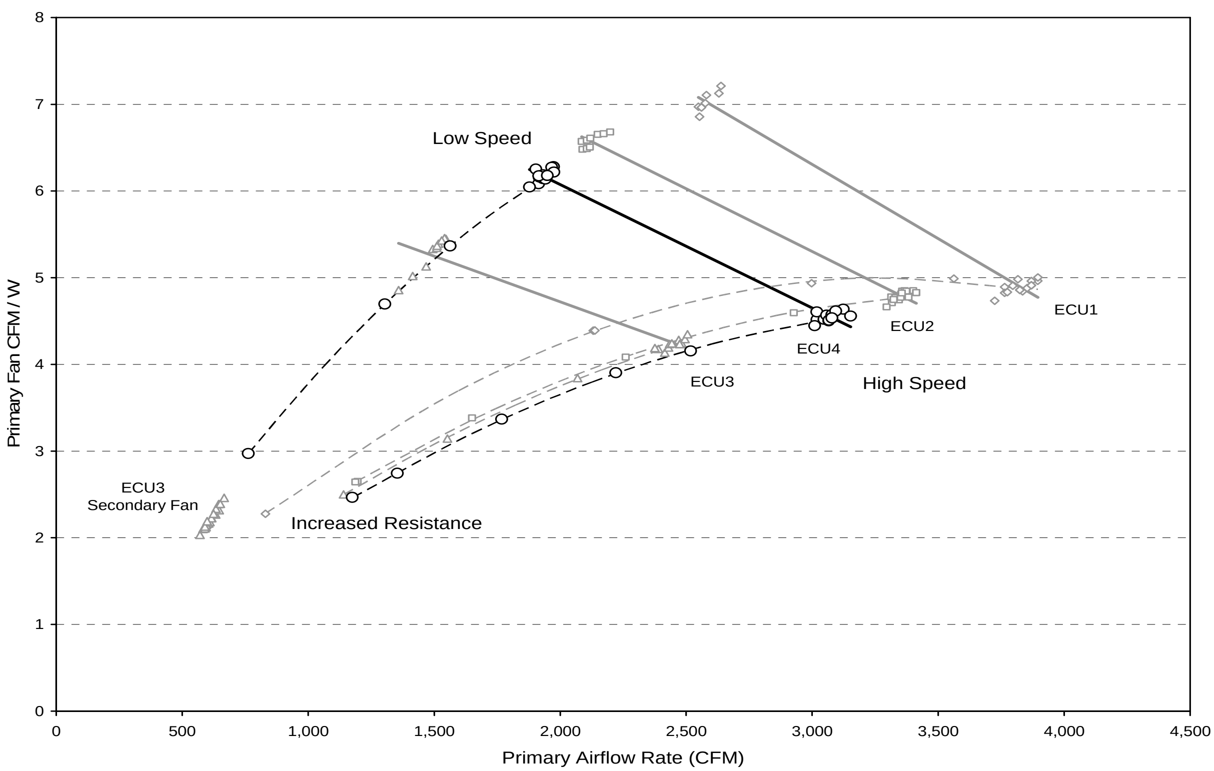

Figure 13: Power Factor versus Airflow

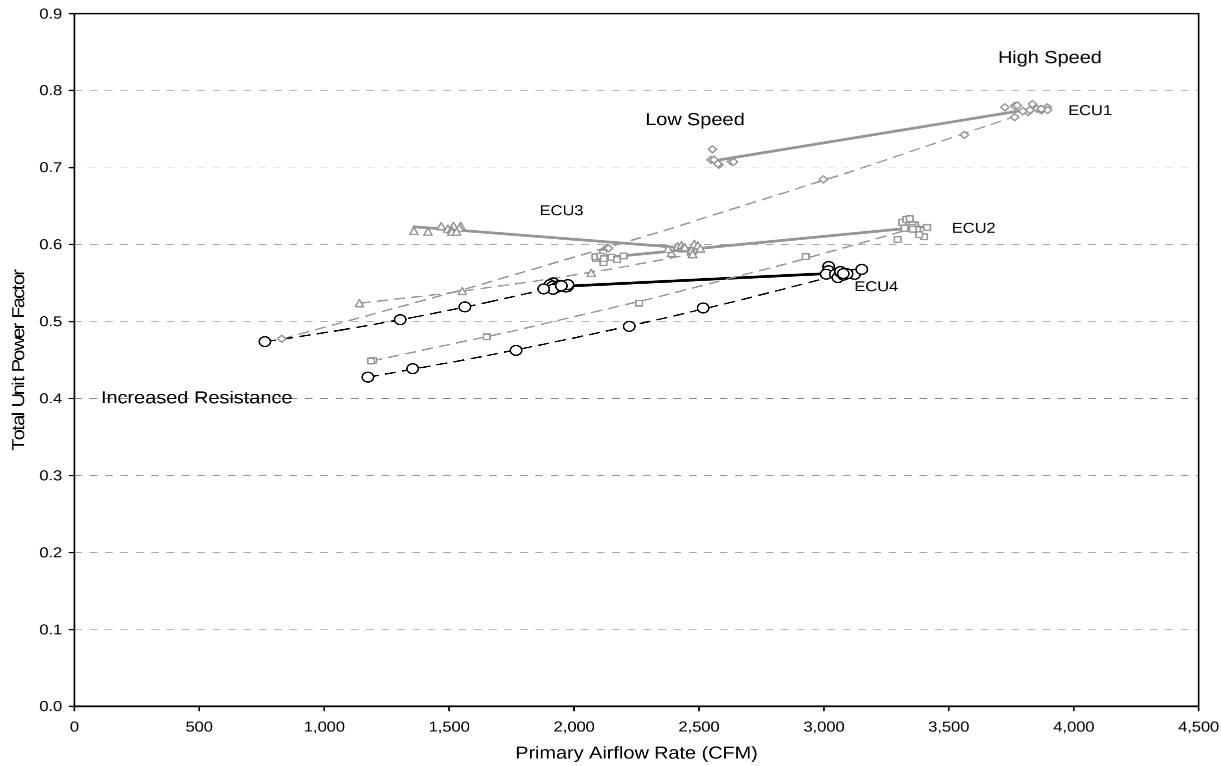

Figure 14: Effectiveness versus Airflow

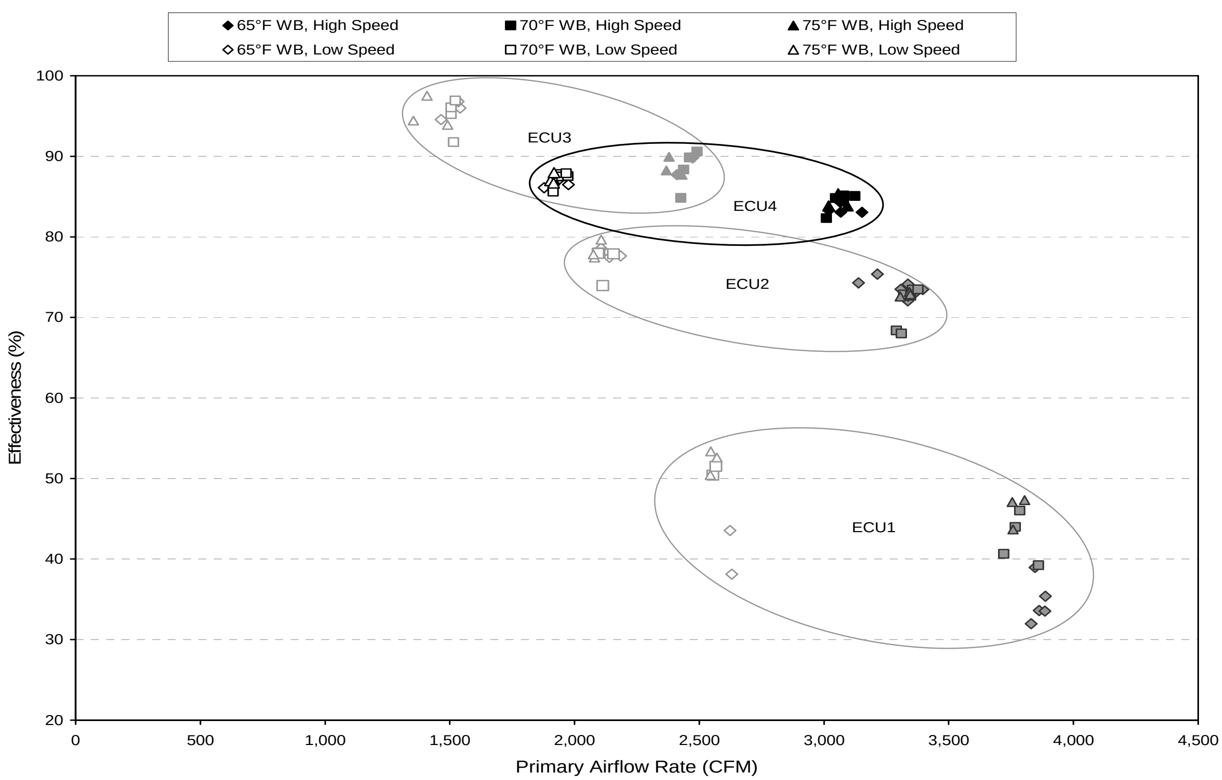

Figure 15: Effectiveness versus Intake Dry-bulb Temperature

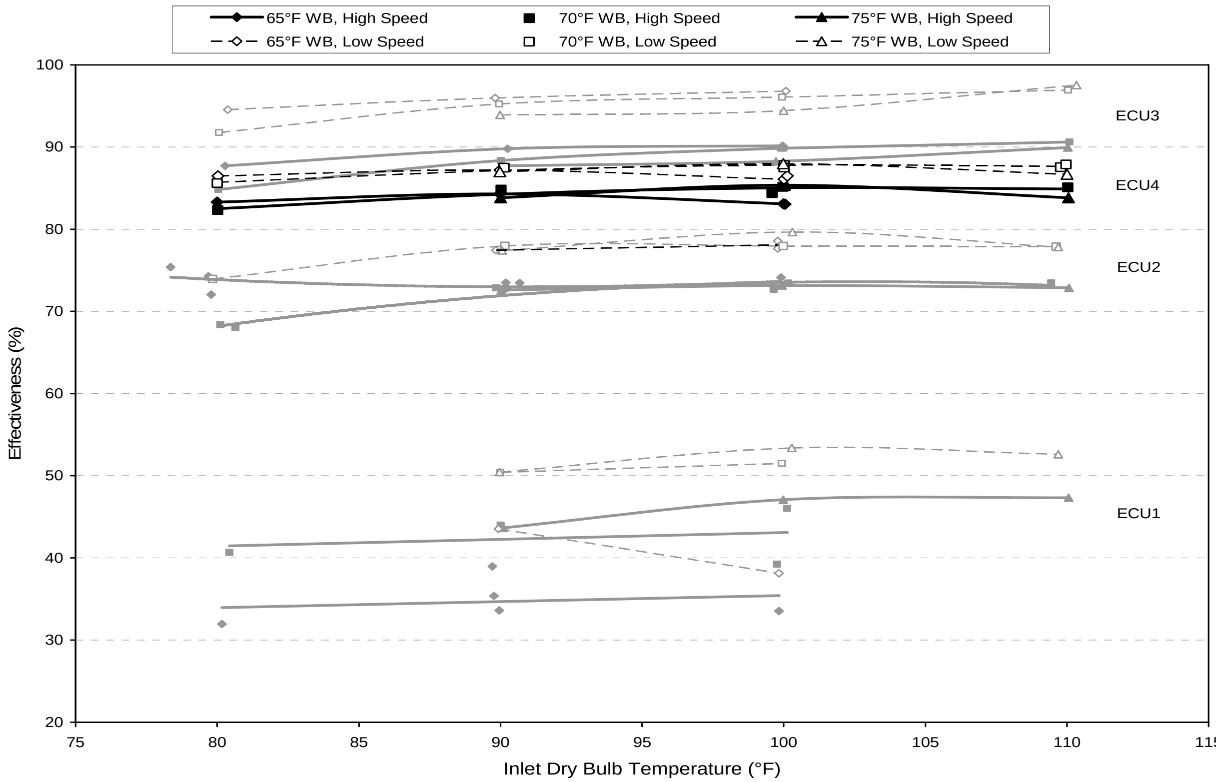

Figure 16: Effectiveness versus Intake Wet-bulb Depression

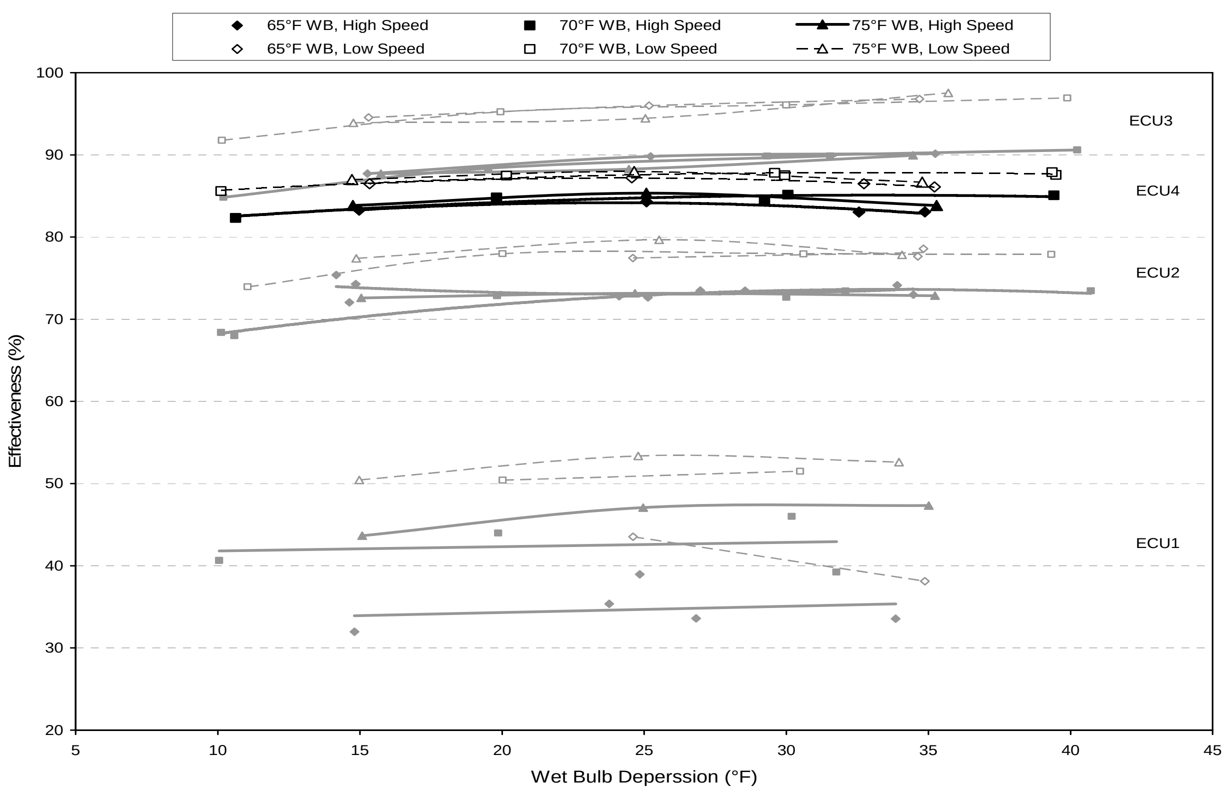

Figure 17: ECU4 Capacity
(80°F Room Temperature Basis)

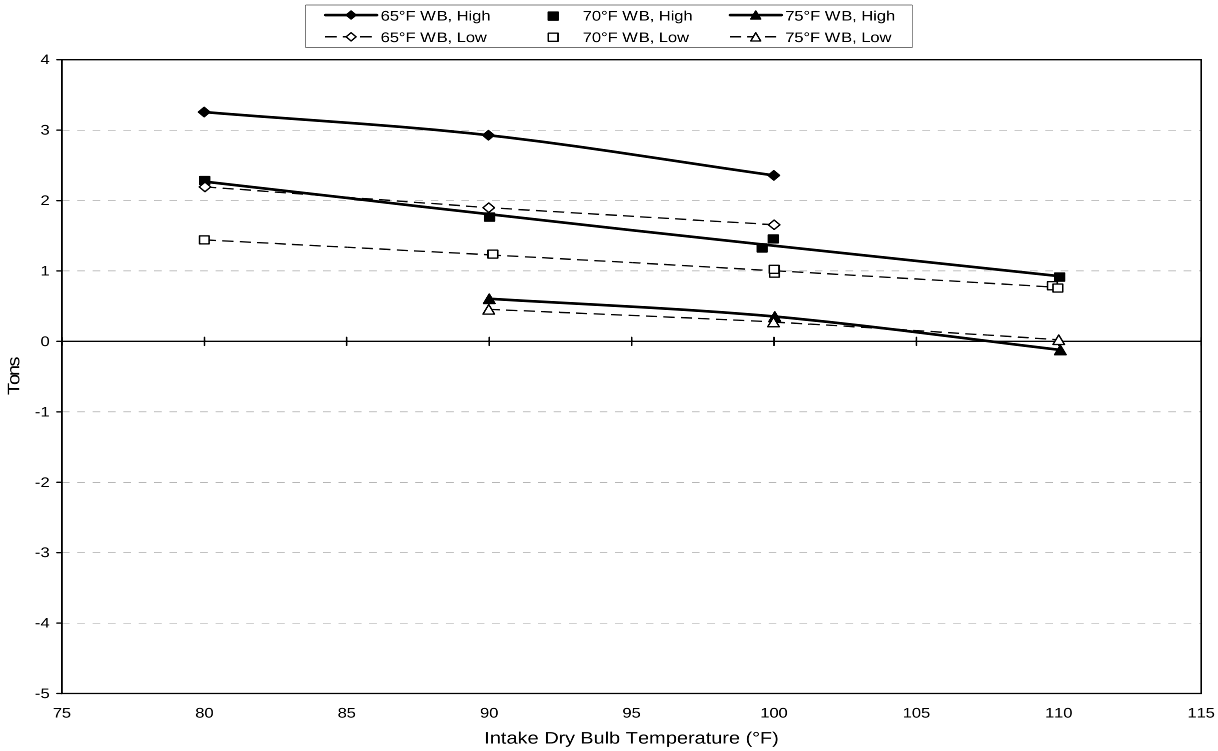

Figure 18: ECU4 Energy Efficiency Ratio
(80°F Room Temperature Basis)

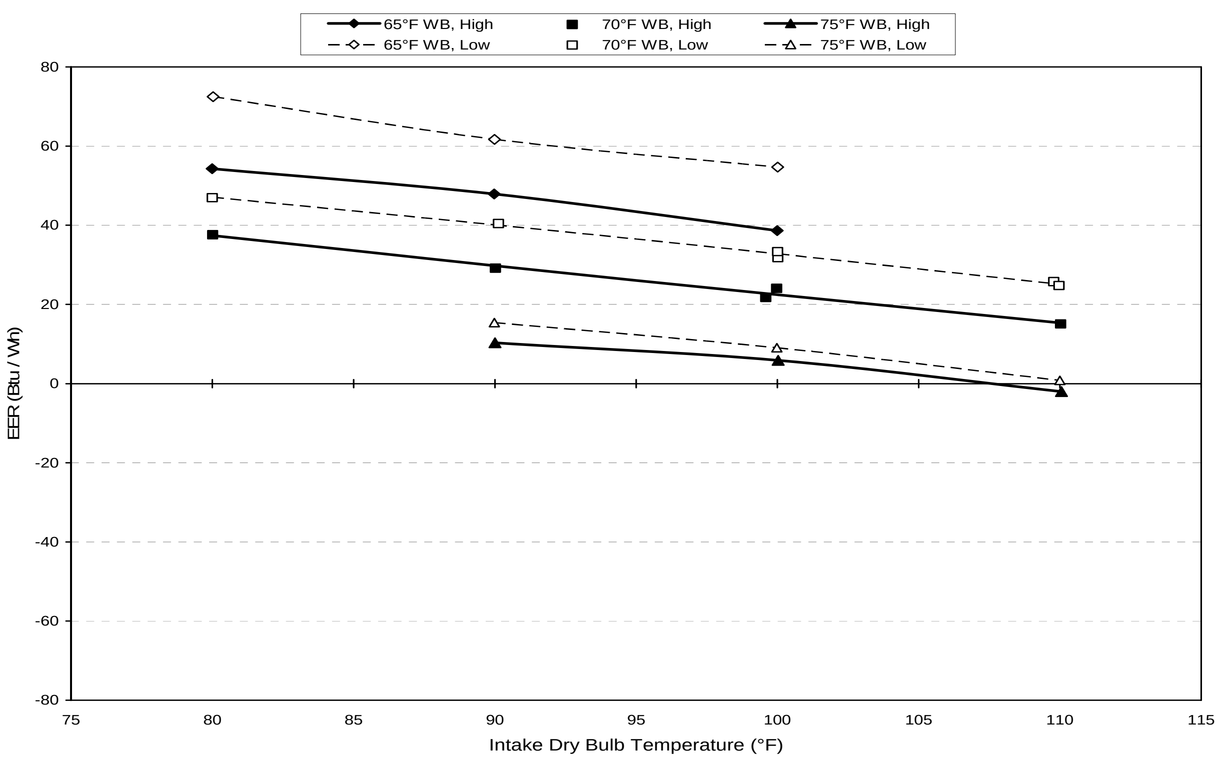

Figure 19: Sensible Cooling of Intake Air

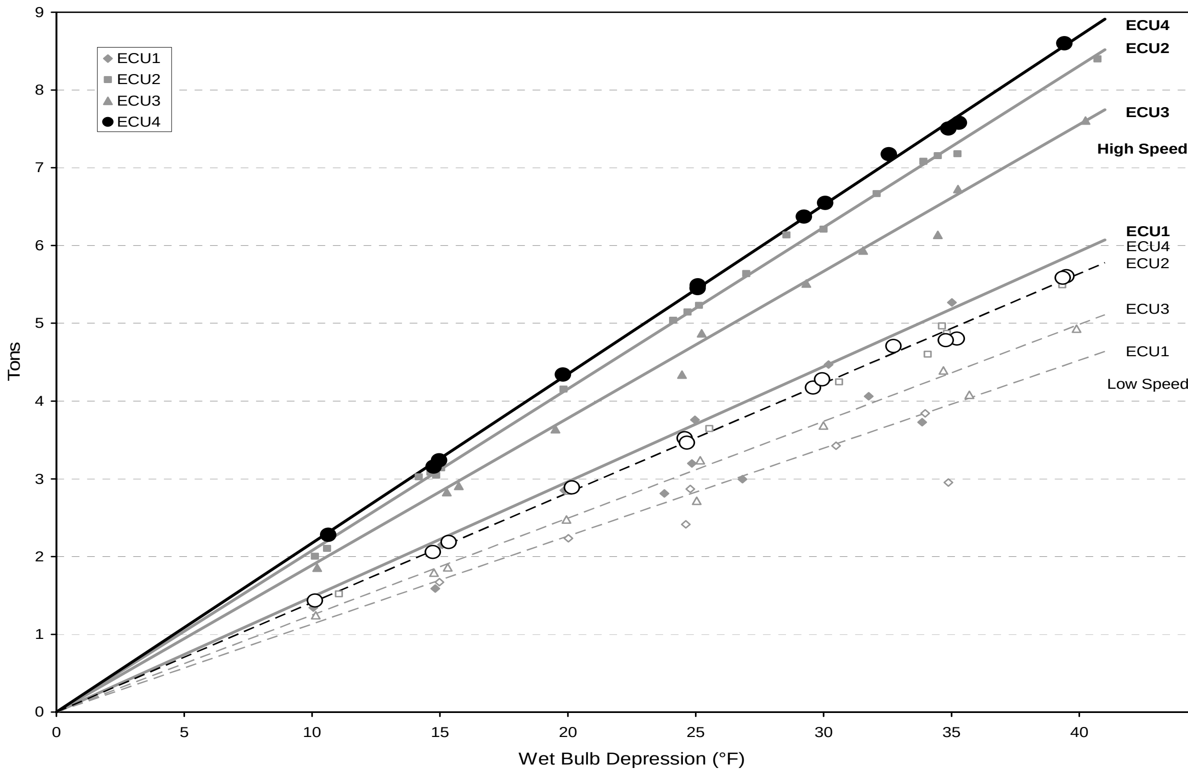

Figure 20: ECU4 Performance Sensitivity to Backpressure

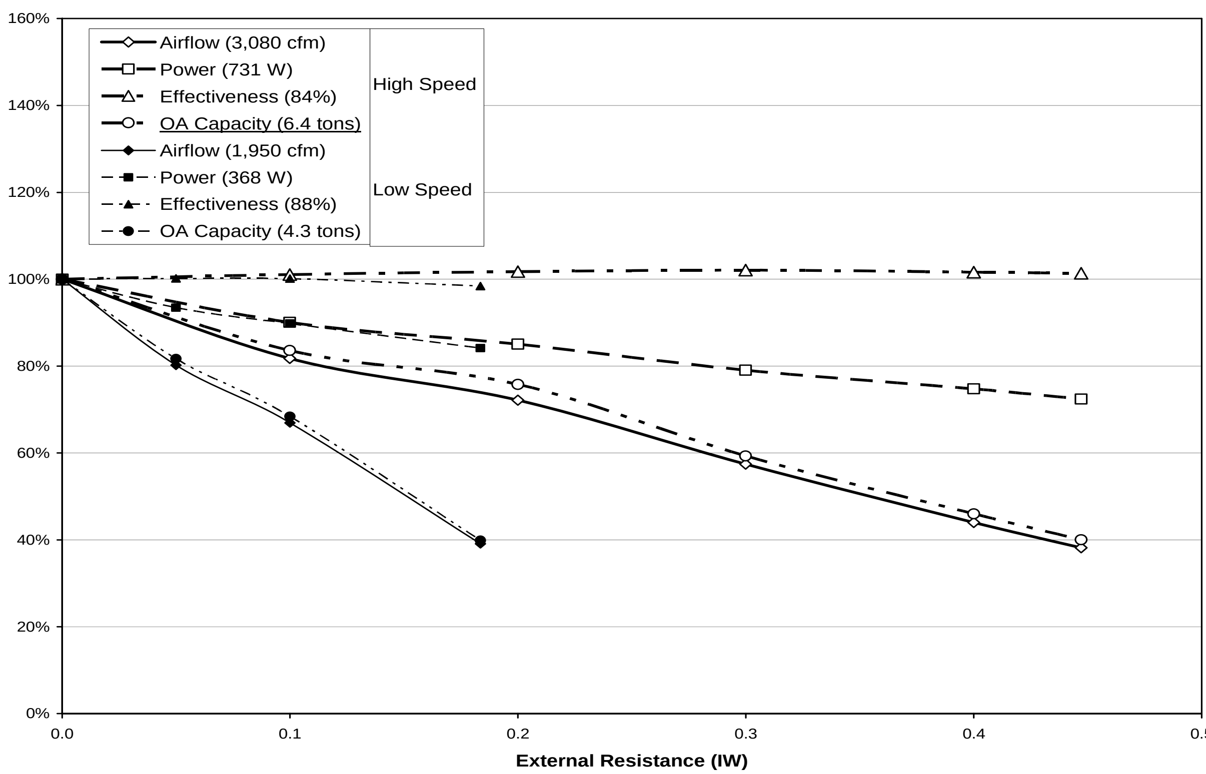

Figure 21: Measured Make-up Water Flow

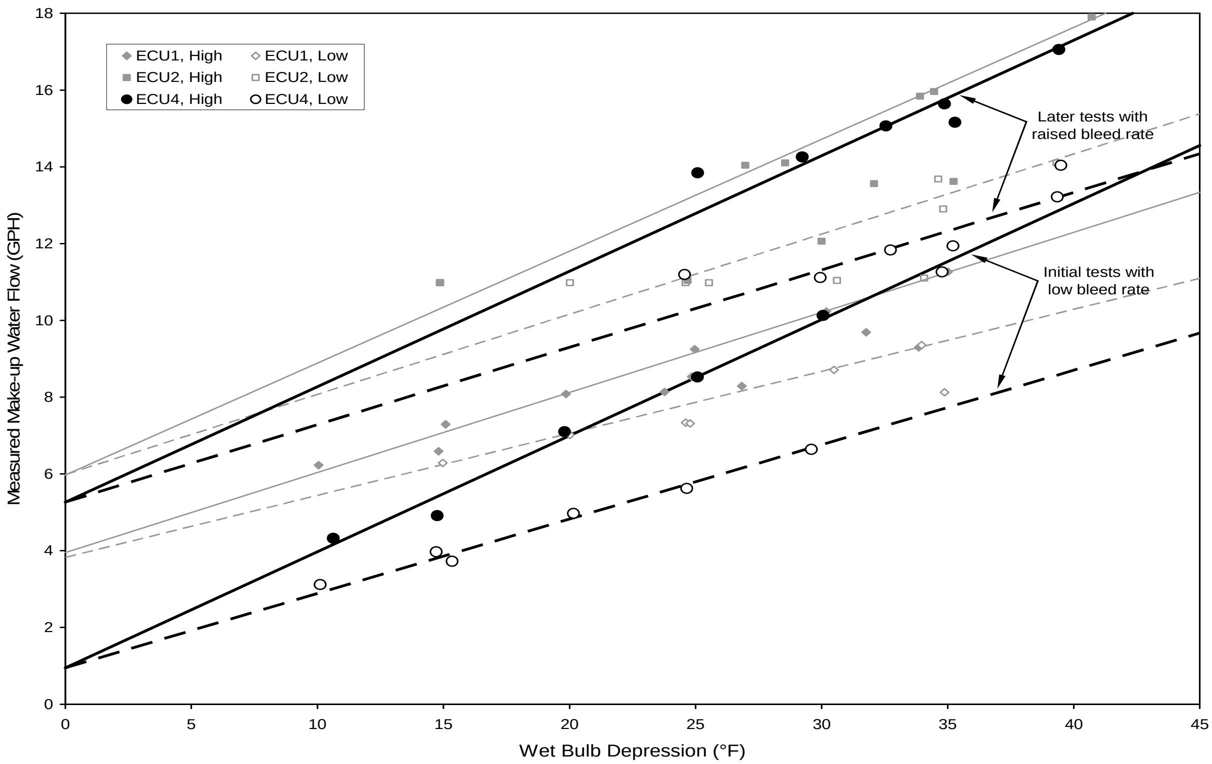

Figure 22: Calculated Evaporation Rate

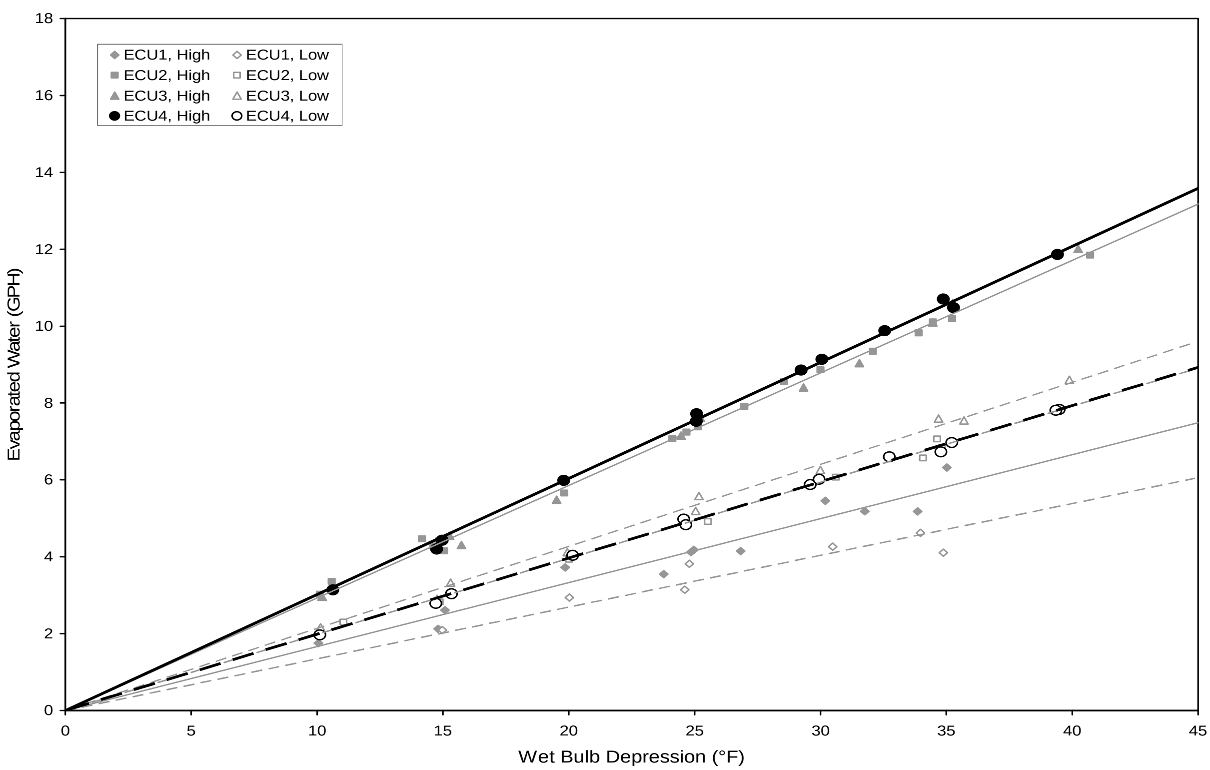

Table 8: ECU4 Test Data

Test Summary Information	High Speed											Variable Resistance (High Speed)					
General																	
Date (all year 2004)	Oct 26	Oct 27	Oct 27	Oct 27	Oct 26	Oct 29	Oct 27	Oct 28	Oct 27	Oct 28	Oct 29	Nov 1	Jan 0	Oct 28	Oct 28	Oct 28	Oct 28
Start Time	12:55p	8:10a	4:12p	9:38a	2:56p	7:45a	2:25p	11:34a	11:56a	8:33a	9:49a	11:11a	12:00a	11:34a	11:51a	12:36p	1:01p
Duration (minutes)	31	20	30	30	30	32	30	15	30	30	30	30	0	15	43	23	36
Barometric Pressure (in. of Hg)	29.14	29.25	29.29	29.28	29.10	29.70	29.28	29.54	29.30	29.52	29.72	29.80	0.00	29.54	29.55	29.54	29.54
Nominal Test Conditions																	
Inlet Dry Bulb Temperature (°F)	80	80	90	90	90	100	100	100	100	110	110	100	0	100	100	100	100
Inlet Wet Bulb Temperature (°F)	65	70	65	70	75	65	70	70	75	70	75	65	0	70	70	70	70
Fan Speed	High	High	High	High	High	High	High	High	High	High	High	Low	0	High	High	High	High
Inlet Air Properties																	
Dry Bulb Temperature (°F)	80.0	80.0	90.0	90.0	90.0	100.1	100.0	99.6	100.0	110.0	110.1	100.0	0.0	99.6	98.9	99.8	100.1
Dew Point Temperature (°F)	56.9	64.6	49.6	60.9	69.6	47.7	54.2	55.3	64.5	48.3	58.5	39.5	0.0	55.3	53.5	53.4	55.2
Relative Humidity (%) - calculated	45.2	59.3	25.1	37.8	51.3	17.2	21.9	23.0	31.6	13.1	19.0	12.6	0.0	23.0	22.0	21.4	22.6
Wet Bulb Temperature (°F) - calc.	65.0	69.4	64.9	70.2	75.3	*67.5*	69.9	70.4	75.0	70.6	74.8	64.8	0.0	70.4	69.4	69.6	70.5
Wet Bulb Depression (°F)	15.0	10.6	25.1	19.8	14.8	32.6	30.1	29.2	25.1	39.4	35.3	35.2	0.0	29.2	29.5	30.2	29.6
Outlet Air Properties									External Resistance (IW) -->			**0.00**	**0.00**	**0.00**	**0.10**	**0.20**	**0.30**
Dry Bulb Temperature (°F)	67.5	71.3	68.8	73.2	77.6	73.0	74.4	74.9	78.6	76.5	80.5	69.7	0.0	74.9	73.8	73.9	74.6
Dew Point Temperature (°F)	63.8	68.6	63.3	69.0	74.3	65.0	68.3	68.6	73.7	68.3	72.7	63.3	0.0	68.6	67.8	68.2	69.3
Relative Humidity (%) - calculated	87.9	91.3	82.5	86.7	89.4	76.1	81.2	80.7	84.8	75.7	77.4	80.2	0.0	80.7	81.8	82.3	83.7
Wet Bulb Temperature (°F) - calc.	65.0	69.4	65.1	70.3	75.2	67.6	70.1	70.5	75.0	70.7	74.9	65.4	0.0	70.5	69.6	69.9	70.9
Pan Water Temperature (°F)	65.3	69.5	65.5	70.5	75.2	67.7	70.5	70.8	75.3	71.2	75.1	65.8	0.0	70.8	69.9	70.0	70.9
Performance																	
Dry Bulb ΔT (°F)	-12.5	-8.8	-21.1	-16.8	-12.4	-27.0	-25.6	-24.7	-21.4	-33.5	-29.6	-30.3	0.0	-24.7	-25.2	-25.9	-25.5
Wet Bulb ΔT (°F)	0.0	0.0	0.2	0.1	-0.1	0.1	0.2	0.1	0.1	0.1	0.1	0.7	0.0	0.1	0.2	0.3	0.4
Effectiveness (%)	83.3	82.3	84.3	84.8	83.8	83.1	85.2	84.4	85.4	85.1	83.8	86.1	0.0	84.4	85.3	85.9	86.1
Intake Airflow Rate (CFM)	3,020	3,010	3,070	3,050	3,020	3,150	3,080	3,080	3,060	3,120	3,090	1,880	0	3,080	2,520	2,220	1,770
Sensible Capacity (tons)	3.26	2.28	2.93	1.76	0.61	1.88	1.45	1.32	0.35	0.91	-0.12	1.66	0.00	1.32	1.34	1.15	0.81
Evaporation Rate (gph)	4.4	3.1	7.7	6.0	4.2	9.9	9.1	8.9	7.5	11.9	10.5	7.0	0.0	8.9	7.5	6.8	5.4
Makeup Water Usage (gph)	7.8	4.3	13.8	7.1	4.9	15.1	10.1	14.3	8.5	17.1	15.2	11.9	0.0	14.3	13.0	12.3	10.8
Power Consumption																	
Fan (W)	669	677	681	674	655	692	673	679	669	674	670	311	0	679	606	569	525
Pump (W)	51	52	52	52	51	52	52	52	52	52	52	53	0	52	52	52	53
Total (W)	720	729	733	726	706	744	725	731	721	725	722	363	0	731	658	621	578
Unit Power Factor	0.57	0.56	0.56	0.56	0.57	0.57	0.56	0.56	0.56	0.56	0.56	0.54	0.00	0.56	0.52	0.49	0.46
Fan CFM / W	4.52	4.45	4.50	4.52	4.61	4.56	4.57	4.54	4.57	4.64	4.61	6.05	0.00	4.54	4.16	3.90	3.37
Unit CFM / W	4.20	4.13	4.18	4.20	4.27	4.24	4.25	4.21	4.24	4.31	4.28	5.17	0.00	4.21	3.83	3.57	3.06
Energy Efficiency Ratio (Btu / Wh)	54.31	37.59	47.93	29.15	10.31	30.26	24.05	21.75	5.87	15.07	-2.02	54.67	0.00	21.75	24.35	22.20	16.81

Table 8: ECU4 Test Data (Continued)

Test Summary Information	Low Speed												Variable Resistance (Low Speed)			
General																
Date (all year 2004)	Oct 26	Oct 27	Oct 27	Oct 27	Oct 26	Oct 29	Oct 27	Oct 28	Oct 27	Oct 28	Oct 28	Oct 29	Oct 28	Jan 0	Oct 28	Oct 28
Start Time	1:48p	8:43a	5:06p	10:52a	4:50p	8:34a	3:23p	2:37p	1:05p	9:11a	10:03a	11:01a	2:06p	12:00a	2:37p	4:18p
Duration (minutes)	30	30	24	24	30	30	30	30	30	12	27	30	19	0	30	20
Barometric Pressure (in. of Hg)	29.13	29.26	29.29	29.29	29.10	29.71	29.28	29.53	29.29	29.52	29.53	29.72	29.53	0.00	29.53	29.55
Nominal Test Conditions																
Inlet Dry Bulb Temperature (°F)	80	80	90	90	90	100	100	100	100	110	110	110	100	0	100	100
Inlet Wet Bulb Temperature (°F)	65	70	65	70	75	65	70	70	75	70	70	75	70	0	70	70
Fan Speed	Low	Low	Low	Low	Low	Low	Low	Low	Low	Low	Low	Low	High	0	Low	Low
Inlet Air Properties																
Dry Bulb Temperature (°F)	80.0	80.0	90.0	90.1	90.0	100.1	100.0	100.0	100.0	109.8	110.0	110.0	99.9	0.0	100.0	100.0
Dew Point Temperature (°F)	56.2	65.4	50.9	60.3	69.6	47.4	55.3	54.3	65.2	47.6	48.3	59.5	53.4	0.0	54.3	53.2
Relative Humidity (%) - calculated	44.1	61.1	26.3	36.9	51.4	16.9	22.8	22.0	32.4	12.8	13.1	19.7	21.3	0.0	22.0	21.1
Wet Bulb Temperature (°F) - calc.	64.7	69.9	65.4	70.0	75.3	*67.4*	70.4	70.1	75.3	70.3	70.6	75.2	69.6	0.0	70.1	69.6
Wet Bulb Depression (°F)	15.3	10.1	24.6	20.2	14.7	32.7	29.6	30.0	24.7	39.5	39.4	34.8	30.3	0.0	30.0	30.4
Outlet Air Properties										External Resistance (IW) -->			**0.45**	**0.00**	**0.00**	**0.05**
Dry Bulb Temperature (°F)	66.8	71.4	68.6	72.5	77.2	71.8	74.0	73.8	78.3	75.2	75.4	79.9	74.0	0.0	73.8	73.3
Dew Point Temperature (°F)	63.8	69.3	64.3	69.1	74.5	65.7	69.3	68.8	74.4	68.5	69.0	73.8	68.8	0.0	68.8	68.5
Relative Humidity (%) - calculated	90.2	93.2	86.2	89.0	91.5	81.1	85.1	84.3	87.8	79.8	80.5	81.9	83.8	0.0	84.3	84.9
Wet Bulb Temperature (°F) - calc.	64.8	69.9	65.7	70.1	75.2	67.6	70.7	70.3	75.4	70.5	70.9	75.4	70.4	0.0	70.3	70.0
Pan Water Temperature (°F)	65.0	69.9	65.9	70.2	75.3	67.8	71.0	70.5	75.6	70.8	71.1	75.6	70.1	0.0	70.5	70.0
Performance																
Dry Bulb ΔT (°F)	-13.3	-8.7	-21.4	-17.6	-12.8	-28.3	-26.0	-26.2	-21.7	-34.6	-34.6	-30.2	-25.9	0.0	-26.2	-26.7
Wet Bulb ΔT (°F)	0.1	0.0	0.3	0.1	-0.1	0.3	0.2	0.2	0.1	0.2	0.2	0.2	0.7	0.0	0.2	0.4
Effectiveness (%)	86.5	85.6	87.2	87.5	87.0	86.5	87.8	87.5	88.0	87.6	87.9	86.7	85.5	0.0	87.5	87.7
Intake Airflow Rate (CFM)	1,920	1,910	1,940	1,930	1,900	1,980	1,930	1,950	1,920	1,970	1,970	1,920	1,170	0	1,950	1,560
Sensible Capacity (tons)	2.20	1.44	1.90	1.24	0.46	1.38	0.97	1.02	0.27	0.79	0.76	0.02	0.60	0.00	1.02	0.89
Evaporation Rate (gph)	3.0	2.0	5.0	4.0	2.8	6.6	5.9	6.0	4.8	7.8	7.8	6.7	3.8	0.0	6.0	5.0
Makeup Water Usage (gph)	3.7	3.1	11.2	5.0	4.0	11.8	6.6	11.1	5.6	14.0	13.2	11.3	9.0	0.0	11.1	10.1
Power Consumption																
Fan (W)	311	315	316	314	305	318	312	315	311	314	313	310	476	0	315	291
Pump (W)	52	53	53	53	52	53	53	53	53	53	53	53	53	0	53	53
Total (W)	363	368	369	368	356	371	365	368	364	368	367	363	529	0	368	344
Unit Power Factor	0.55	0.55	0.55	0.55	0.55	0.55	0.54	0.55	0.54	0.55	0.55	0.54	0.43	0.00	0.55	0.52
Fan CFM / W	6.16	6.08	6.14	6.14	6.25	6.22	6.19	6.18	6.16	6.28	6.27	6.18	2.47	0.00	6.18	5.37
Unit CFM / W	5.28	5.20	5.26	5.26	5.35	5.33	5.29	5.29	5.26	5.37	5.37	5.27	2.22	0.00	5.29	4.54
Energy Efficiency Ratio (Btu / Wh)	72.49	46.95	61.69	40.44	15.38	44.57	31.84	33.32	9.04	25.76	24.77	0.81	13.50	0.00	33.32	30.86