Evaluation of Advanced Evaporative Cooler Technologies

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EXECUTIVE SUMMARY

A series of tests were conducted on three types of evaporative coolers:

- A traditional-style "swamp cooler" with thin plastic pads,
- An advanced whole-house evaporative cooler with a thick (8") cellulose pad, and
- The advanced cooler supplemented with an add-on indirect evaporative precooler.

The test units were all new, off-the-shelf systems, and were not special models configured for rating purposes. The goal was to determine if the more advanced systems would provide enough of a performance gain to recommend their use by consumers, and whether to continue to support their purchase with incentives.

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. The two direct units were tested simultaneously while exposed to the same environmental conditions. Both systems had two-speed primary fan motors, and test data were collected at both speed settings.

The traditional-style unit may have been a bad sample of this type of cooler, and had low measurements of evaporation (or saturation) effectiveness (how close the outlet temperature gets to the entering wet-bulb temperature). At high speed, its effectiveness ranged from 32 to 47% (averaging 41%), while at low speed the range was 38 to 53% (averaging 49%). The lowest effectiveness values were recorded under conditions of low wet-bulb temperature, and may have been the result of the pads forming dry spots or water channels. The one advantage of this cooler was that because of the low flow resistance of the thin pads, it was able to provide the most airflow (3,800 cfm at high speed, and 2,500 cfm at low speed), even with a smaller fan motor (½ versus ¾-hp). Its average power demand was about 810 Watts at high speed, and 390 Watts at low speed.

The advanced unit had a nearly constant effectiveness over the range of test conditions, affected only by airflow. At high speed, the effectiveness range was 68 to 74% (averaging 73%), and at low speed, the range was 77 to 80% (averaging 78%). This was still lower than the anticipated 85 to 90% that was expected for this unit. The test unit was one with an 8"-thick cellulose pad, and the optional 12" pad may be needed to achieve the higher results. Because of the higher resistance of the pad, this unit's airflow was 12% less than the traditional-style unit at 3,300 cfm high speed (2,100 cfm low speed), despite a larger fan motor. This unit's average power demand was 740 Watts at high speed and 360 Watts at low speed.

For both of these units, the unit power factor was very low (averaging 0.77 and 0.62, respectively). This effect not only creates problems from a power delivery standpoint, but it also contributes to motor heating. The motor heat is eventually transferred to the air stream supplied to the conditioned space, thus reducing the cooling capability of the unit.

Adding the indirect evaporative precooler to the advanced unit increased its overall effectiveness while reducing the absolute humidity of the delivered air. The overall system effectiveness at high speed ranged from 85 to 91% (averaging 89%), and at low speed ranged from 92 to 98% (averaging 95%). However, this system also acted to reduce the supply airflow by 26% to only 2,400 cfm. The high cost of this add-on (at about 1½ times the cost of the direct evaporative cooler it was attached to) presents a poor benefit to cost ratio. This unit had the highest power consumption as the result of having an additional fan and pump. Its power demand was 940 Watts at high speed and 640 Watts at low speed.

The tests suggest that it would be worthwhile to continue to provide incentives towards the installation of the advanced whole-house evaporative coolers as an alternative to both traditional-style coolers and

conventional air conditioners. The add-on indirect evaporative cooler technology could use more development to improve its performance and reduce its cost, and if these advances are achieved, may be worth considering for future incentive programs. Although not included in the scope of this project, the obtained test results should be sufficient for use as inputs to computer models to calculate and compare their annualized energy use against that of a conventional air conditioner.

INTRODUCTION

Background

Central air conditioning can be the largest electrical load in a home during the summer months. It is also used at times when the demand for electricity is high in other sectors, contributing to a system-wide peak for which generation, transmission, and distribution systems must be designed to support. The high demand also drives up the real-time cost for power, which is not normally seen by the consumer or recovered by the utility. This effect puts central air conditioning systems under scrutiny for ways to make them more efficient, or to develop energy-saving alternatives, and thus to reduce their impact on the system peak.

One alternative technology that has been around for many years, but which has attracted new interest due to technological advances and the rising cost of energy, is utilizing the effect of evaporating water for comfort cooling. So-called "direct" evaporative coolers work by drawing air across a wetted pad, where the air temperature is reduced as the water on the pads is evaporated. The only energy consumed is for a fan to move the air and a pump to circulate water for evaporation, and their combined usage is considerably less than that for a conventional air conditioner. Another advantage is that it can improve indoor air quality by flushing an indoor space with 100% outside air, which is effectively "washed" of most pollen and dust.

Traditional evaporative coolers use thin pads made out of wood fiber or plastic, while newer coolers employ different evaporative media (thick cellulose pads) to increase the evaporation effectiveness. 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 (compression refrigeration).

The major drawback to the acceptance of direct evaporative coolers is the high humidity of the supply air. (The increased humidity produced by older evaporative coolers has earned them the term "swamp coolers".) Another advanced technology is an "indirect" evaporative system that still cools by evaporation of water, but does not add this moisture to the supplied air stream. Indirect, or combined indirect / direct 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 advantage over simple direct systems.

Prior Research

PG&E's Technical and Ecological Services (TES) has done some previous evaluation of other air conditioning systems, including other combined indirect / direct evaporative cooler technologies. The first tests 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 integral unit 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.

Objectives

The purpose of this project was to evaluate current evaporative cooling systems to determine the advantages of a typical advanced evaporative coolers in relation to a traditional cooler, and to evaluate the benefits of adding an indirect evaporative precooler. The information gained may be used to reevaluate the level of incentives provided for evaporative cooling systems.

The objective was to examine the relative system performance of three different evaporative cooling units (ECUs), as defined by:

- airflow,
- evaporation (or saturation) effectiveness,
- power demand,

- cooling capacity and efficiency,

as a function of the variables:

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

The test 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 (intake on only one side through a thick cellulose pad),
- ECU3: ECU2 plus an add-on indirect stage

Performance data are required in order to document the ability of these systems to maintain comfort under the various conditions. The data collected are intended to provide enough information to adequately model the performance of the different types of evaporative coolers, and thus to determine the annual energy usage and peak demand for different cities. The results may be made into technical information sheets for promoting evaporative coolers, and may be used to develop marketing materials for future rebate or incentive programs.

EXPERIMENTAL DESIGN AND PROCEDURE

Performance Characteristics

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 measured air temperature is normally referred to as the “dry-bulb” temperature.)

Direct evaporative coolers are described as a constant wet-bulb temperature process. Thus, their performance is related to how close the discharge air reaches the wet-bulb temperature, or its “effectiveness”. It is defined as follows:

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

where $T_{db_{in}}$ and $T_{wb_{in}}$ are the intake dry and wet-bulb temperatures, respectively, and $T_{db_{out}}$ is the dry-bulb temperature at the air outlet. The wet-bulb is actually only a limit for a single-stage direct or indirect evaporative cooler. When combined, a high performance indirect-direct evaporative cooler can achieve an effectiveness greater than 100% because the indirect stage reduces both the dry and wet-bulb temperatures. One could imagine an infinite number of indirect stages with decreasing returns at each stage, and the theoretical limiting temperature for this system would actually be the dew point temperature of the entering air.

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 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 number for an evaporative cooler is not defined in any standard, and it is open to some interpretation.

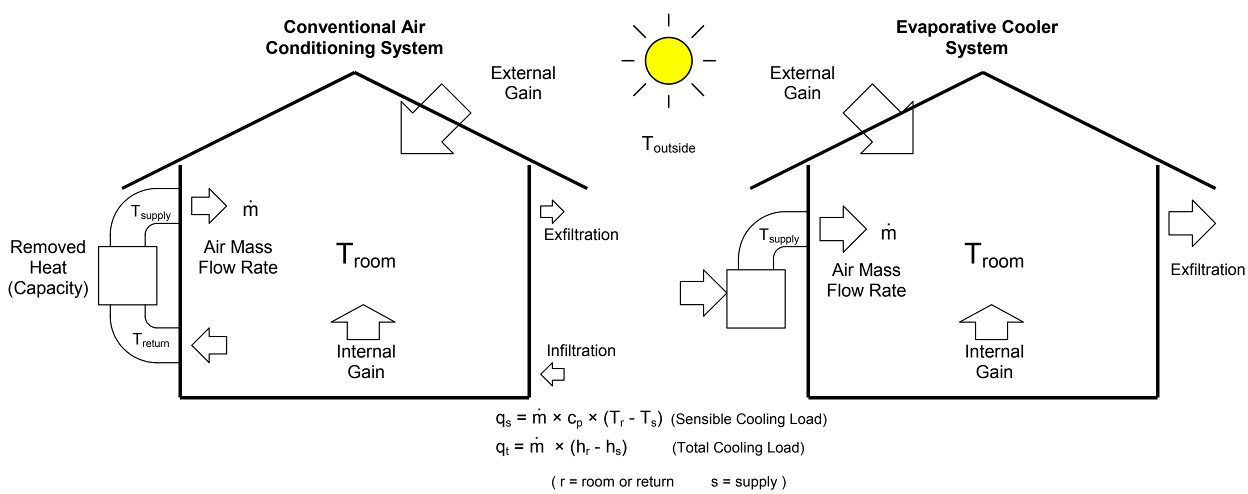
A residential air conditioning system is designed to condition the air contained in a space, removing the heat (sensible load) and moisture (latent load) gained from various sources, while recirculating the same air repeatedly. The cooling capacity is measured at the evaporator coil as the product of 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 from the space 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 its rest temperature. The high flow also means that evaporative coolers cannot be connected to a duct system sized for the velocities provided by a conventional air conditioner or furnace. The building cooling load may also be reduced if the exhaust air is vented out through the attic rather than through open windows, since that will lower the temperature in the attic and reduce the heat gain to the living space through the ceiling.

To determine a cooling capacity for an evaporative system, a design space condition needs to be assumed. The selected condition is an 80°F dry-bulb, which is the same as what is used in the test standards for return air for rating conventional air conditioning systems. It is 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. The sensible cooling load only uses the temperature difference between the room setpoint and the supplied air. Once a cooling capacity is determined, the energy efficiency ratio is then determined as the cooling load divided by the power consumption.

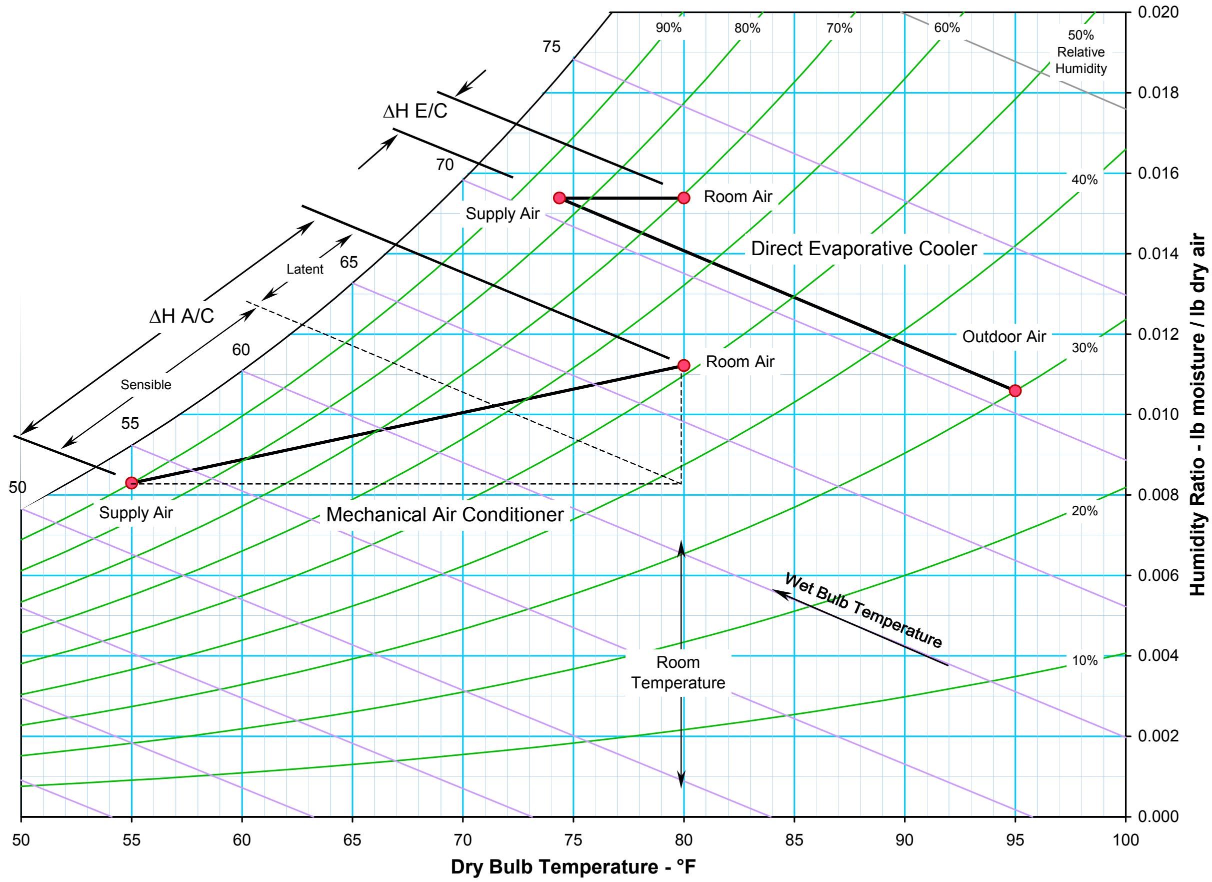
A graphical description of the difference between the two types of systems and the definition of cooling capacity is shown in *Figure 1*:

Figure 1: Definition of Cooling Capacity



The difference can also be represented on a psychrometric chart as shown in *Figure 2*. 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 55°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.

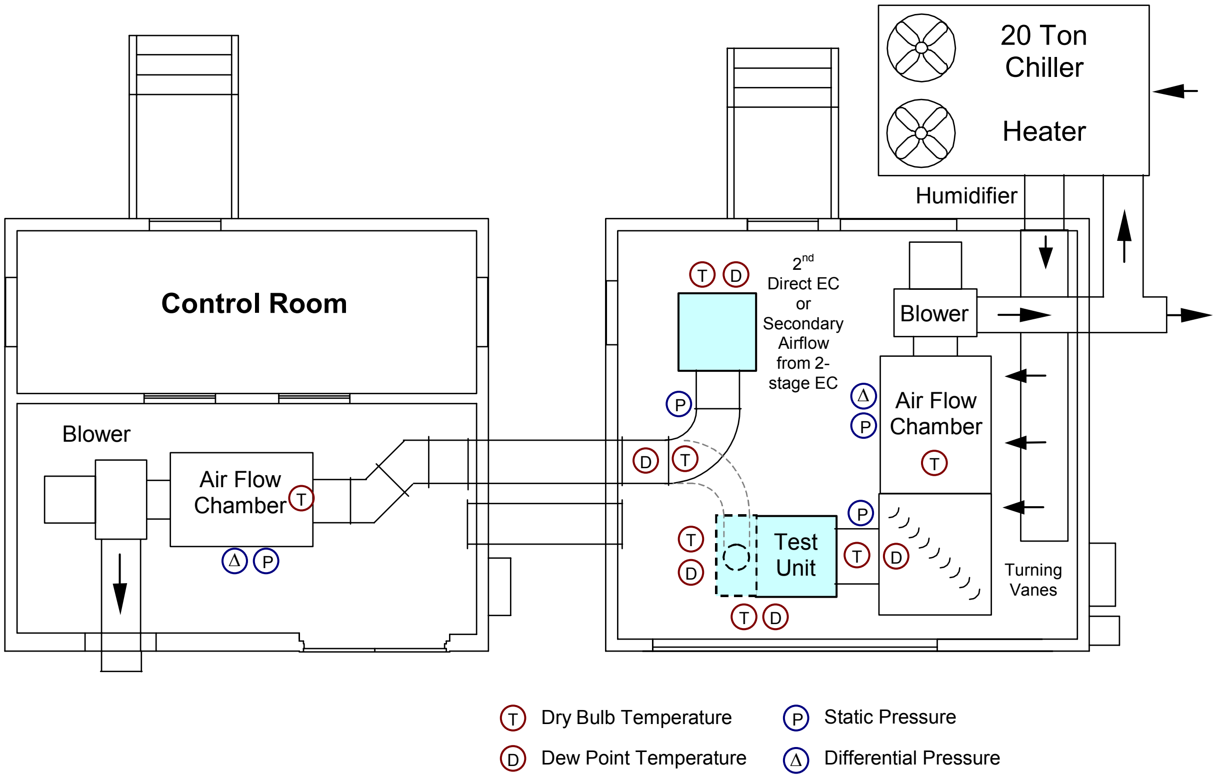
Figure 2: Comparison of Capacity between Air Conditioner and Evaporatvie Cooler
Psychrometric Chart showing the Enthalpy Change used to determine Capacity



Test Facility

The apparatus at the TES Thermal/Flow Test Facility in San Ramon has evolved over a number of years into a system providing good flexibility and measurement accuracy. *Figure 3* shows a layout of the test facility configured for the evaporative cooler testing. The test units were all placed in the larger environmental room, which was conditioned by a 20-ton air conditioner, a resistance heater, and a humidifier. Outside air dampers allowed for some recirculation of the test unit exhaust to control supplied air humidity.

Figure 3: Test Facility and Measurement Locations



The outlet of each test unit was connected to an airflow measurement station, consisting of a sealed chamber with several flow nozzles, designed in accordance with Air Movement and Control Association (AMCA) 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 units was used for the primary airflow from the advanced evaporative cooler. 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 alternately measure the airflow from the traditional direct cooler and the secondary airflow through the add-on indirect cooler. 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.

Measurements and Instrumentation

The test set-up followed the guidelines described in the ASHRAE evaporator cooler test standards (References 5 and 6). An exception to the described measurement method is in regards to the temperature measurements. The standards describe using a sampling system that draws air across a matched pair of dry and wet-bulb thermometers in sequence. Aspirated wet-bulb sensors are prone to error due problems maintaining the proper wetness of the sensor bulb. Instead, the dry-bulb temperature was measured with multiple resistance temperature detectors (RTDs) inserted into the air stream, and the moisture content was measured with a chilled mirror dew point sensor connected to a multi-port sampling system.

The following is a listing of the measurements taken and the instruments used for the initial phase of direct evaporative cooler tests:

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

ECU1 used six sensors (2 for each of three entry faces), while ECU2 used four sensors across its single opening. The sensors were mounted about 2" in front of the inlets at the center of equal face areas.

3. Entering air dew point temperature, using a chilled mirror sensor.

   A sampling system drew and mixed air from several points around the inlet 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.

5. Leaving air wet-bulb 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 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 paddlewheel flow meter.

9. Water temperature in the evaporative cooler sump near the pump intake, using an immersed RTD.

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

Figure 4: ECU1 under Test

Figure 5: ECU2 under Test

When the switch was made to test the combined indirect / direct evaporative cooler (ECU3), the instruments used for ECU1 were transferred to the indirect device, and the secondary air stream was treated much like the primary air stream of a direct unit. The instrumentation for the intake of ECU2 was left in place to measure the intermediate temperature on the primary air stream between the indirect and direct stages.

All of the temperature instruments were calibrated against laboratory standards prior to the tests. The calibration included a low point using an ice bath (32°F), and a high point using a dry hot block calibrator (120°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 an inclined water manometer, accurate to better than 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 transducers, 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, which connected to one of the computer's serial ports. Total power measurements were made with a Yokogawa power meter, which communicated connected through a GPIB interface card. The four dew point sensors connected to a multiple serial port interface, which "listened" to each one for the readings that they sent out at approximately 1-second intervals.

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 4 Hz to provide a fast feedback control signal to the booster fans. The Fluke and Yokogawa were read at 10-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

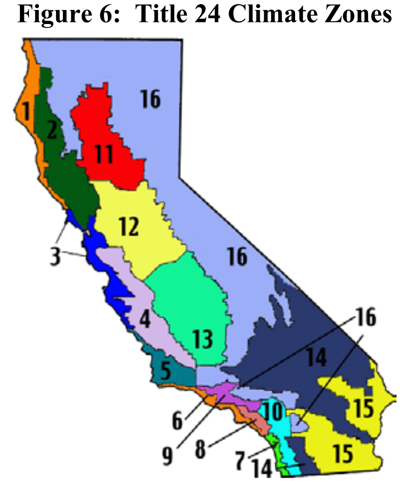
The ASHRAE test standards for evaporative coolers (references 5 and 6) primarily specify the arrangement of the apparatus, the measurements to be taken, and the accuracy of instruments. Neither gives specifics for the test conditions, other than some general guidelines, since evaporative cooling devices are mainly rated in terms of airflow. Reference 5 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 (also specified in reference 6).

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



For this series of tests, the plan was to test over a range of environmental conditions that adequately represents the conditions found during the cooling season at various locations in the PG&E service territory. The PG&E service territory covers nine of the sixteen distinct climate zones identified by the California Energy Commission for Title 24 analysis. 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)
- Maximum dew point temperature and coincident dry-bulb temperature (used for sizing dehumidification equipment)

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

Table 1: ASHRAE Design Condition Cities in PG&E Service Territory

City	Climate Zone	Elev.	Std P PSIA	Cooling DB/MWB				Evaporation WB/MDB				Dehumidification DP/MDB				
				DB	MWB	WBD	RH	WB	MDB	WBD	RH	DP	MDB	WB	WBD	RH
Alameda NAS	3	13	14.688	83	65	18	38%	67	79	12	54%	62	70	65	5	76%
Arcata / Eureka	1	217	14.581	70	60	10	56%	62	67	5	76%	60	64	61	3	87%
Bakersfield	13	492	14.436	104	70	34	18%	73	98	25	31%	64	84	70	14	51%
Blue Canyon	16	5,285	12.097	84	59	25	24%	62	80	18	39%	54	70	59	11	57%
Fairfield (Travis AFB)	12	62	14.662	98	67	31	18%	70	92	22	33%	62	76	67	9	62%
Fresno	13	328	14.522	103	71	32	20%	73	98	25	30%	64	85	71	14	49%
Lemoore (Reeves NAS)	13	236	14.570	103	72	31	22%	75	97	22	36%	67	89	73	16	48%
Marysville (Beale AFB)	11	112	14.636	101	70	31	21%	72	97	25	30%	63	85	70	15	48%
Merced (Castle AFB)	12	187	14.596	99	69	30	21%	72	96	24	31%	64	81	69	12	56%
Mount Shasta	16	3,543	12.909	91	62	29	20%	64	87	23	30%	56	74	62	12	53%
Mountain View (Moffat NAS)	4	39	14.675	88	65	23	28%	68	82	14	49%	62	74	66	8	66%
Paso Robles	4	837	14.257	102	68	34	16%	70	97	27	26%	61	76	66	10	60%
Red Bluff	11	354	14.508	105	70	35	16%	72	98	26	28%	65	82	70	12	56%
Sacramento (NE - McClellan AFB)	12	75	14.655	102	70	32	19%	72	97	25	30%	63	84	70	14	49%
Sacramento (NW - Metro AP)	12	23	14.683	100	69	31	20%	72	96	24	31%	62	82	69	13	51%
Sacramento (SE - Mather Field)	12	95	14.645	101	69	32	19%	71	97	26	28%	61	79	67	12	54%
Salinas	3	85	14.650	83	63	20	32%	66	78	12	53%	62	69	64	5	78%
San Francisco	3	16	14.687	83	63	20	32%	64	79	15	44%	59	67	62	5	75%
San Jose (Int'l AP)	4	56	14.666	93	67	26	25%	70	88	18	41%	63	77	68	9	62%
Santa Maria	5	240	14.569	86	63	23	27%	66	81	15	45%	61	70	64	6	73%
Stockton	12	26	14.681	100	69	31	20%	71	96	25	29%	62	78	67	11	58%

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 dehumidification numbers are not particularly important since the maximum dew point temperature tends to occur when dry-bulb temperatures and the need for cooling are relatively low.)

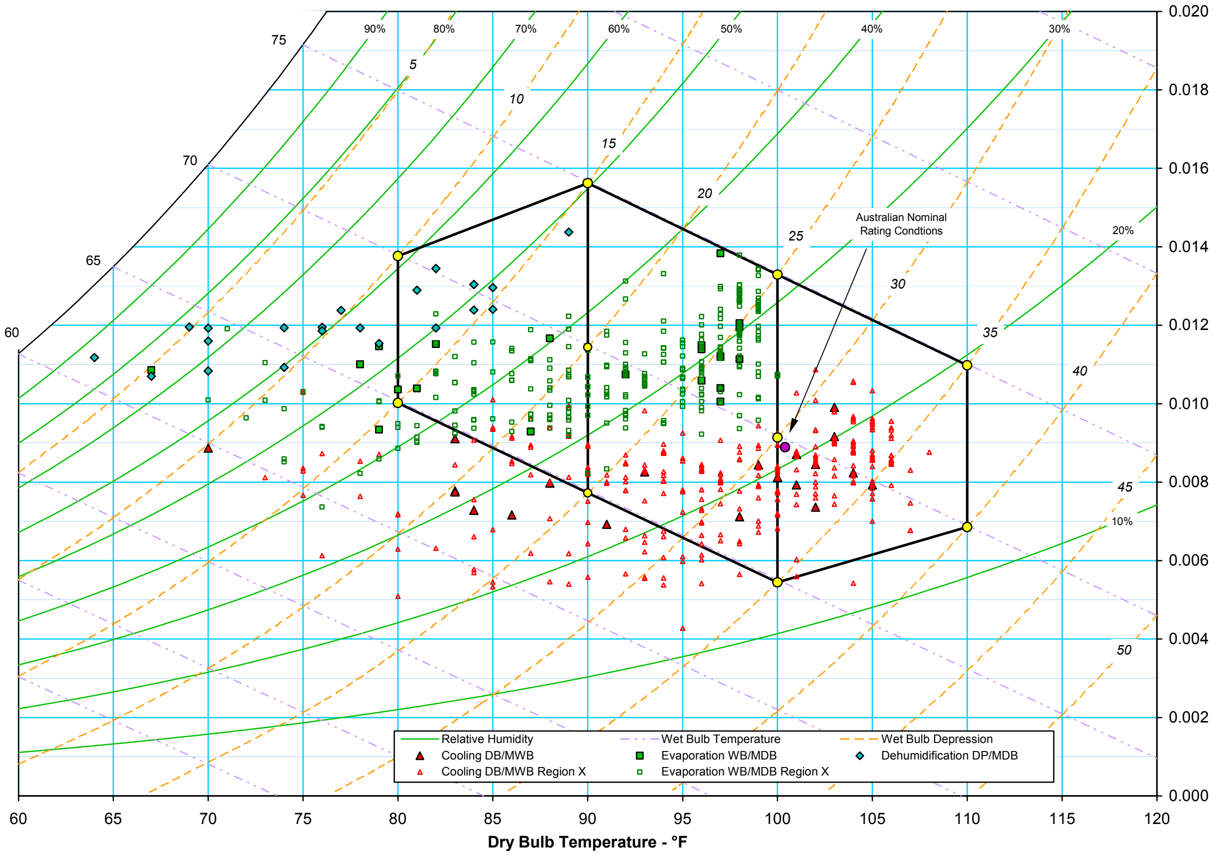
The selection of the number of test points needed to balance adequately representing the probable operating conditions, yet not be so great as to extend the testing period. Therefore, the following matrix of 10 test conditions was selected:

Table 2: Test Point Matrix

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

The test points and the climatic design conditions are shown together in the psychrometric chart. The few climatic conditions outside of the test point matrix include many of the dehumidification points, and many of the conditions for the coastal areas where residential cooling systems are normally unnecessary. It was also decided not to test at an ambient temperature below 80°F where straight ventilation cooling might be adequate.

Figure 7: Psychrometric Chart with Climate Deisgn Data and Test Points



Test Procedure

The test units were "broken in" by running them off and on for a number of 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. A small quantity of dish soap was also added to the pan water during this time to assist in breaking down the film.

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 control system, and the room conditioning system was started to control the room environment.
3. The test unit (or units) 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). (For ECU3, the secondary air outlet was always maintained at 0-inches of water column (WC). This represents an actual installation where the secondary stream draws from and discharges to outside air.)
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 both test units had two-speed fans, 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 relative performance of the test units is determined based on a number of parameters calculated from measured data averaged over stable recorded periods. The results from the tests are shown in several tables and figures. Most of the figures are located at the end of the report in the Appendix, as is a detailed summary of all the test measurements and results.

Table 3 lists several parameters for the three test units that are not particularly affected by the inlet air conditions, and the averaged results over the course of the tests. Also shown is the manufacturer's industry standard rating for airflow for the two direct systems. The parameters included are mainly concerned with airflow and power measurements. The primary fans on the units 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 ECU1 and ECU2 show that at the lower fan speed, the units provide about 2/3 the airflow for half the power relative to their high-speed setting.

Another item of particular interest in this table is the low power factor common to all of the units. 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.

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 / 67°F WB, condenser inlet: 95°F DB), and the results are from prior tests in this facility. The key factor is that the evaporative units consume about one quarter the power of the air conditioner.

Table 3: Average Results for Airflow and Power

	ECU1	ECU2	ECU3	SEER 12
Mfr. Industry Standard Rating (cfm)	4,500	4,800		3-Ton A/C
High Speed				ARI "A"
Intake Airflow (cfm)	3,790	3,320	2,440	1,200
Total Unit Power (W)	806	737	939	3,480
Total Unit Power Factor	0.77	0.62	0.60	0.96
Primary Fan Power* (W)	779	700	578	480
Primary Fan Efficiency (cfm / W)	4.87	4.77	4.27	2.47
Low Speed				
Intake Airflow (cfm)	2,540 (67%)	2,120 (64%)	1,480 (61%)	
Total Unit Power (W)	394 (49%)	360 (49%)	644 (69%)	
Total Unit Power Factor	0.71	0.58	0.62	
Primary Fan Power* (W)	366 (47%)	323 (46%)	282 (49%)	
Primary Fan Efficiency (cfm / W)	6.93	6.54	5.31	
Direct Section Pump Power (W)	27	36	36	
Indirect Section Pump Power (W)			49	
Indirect Section Fan Power (W)			277	
Indirect Section Airflow (cfm)			610	

* "Primary Fan Power" is included in the "Total Unit Power" listed immediately above

Table 4 lists the measured unit outlet 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 systems under different environmental conditions. There were a number of missed test points for ECU1 and ECU2 due to an inability to achieve the required conditions, or to supply sufficient conditioned air to the environmental chamber. Of particular interest are the results for ECU3 at low speed where the discharge temperature is consistently about 1°F greater than the entering air wet-bulb temperature.

Table 5 lists the resulting unit effectiveness in the same format as Table 4. The results indicate that all of the units show an improvement in effectiveness as the airflow 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. Both ECU2 and ECU3 show little sensitivity to the inlet air conditions, with an effectiveness range of less than 6 percentage points over all of test conditions at either speed. ECU1, on the other hand, showed a marked sensitivity to wet-bulb temperature, with a general increase in effectiveness at the higher wet-bulb temperatures. The reason for this is not known precisely, but it is suspected to be the result of channeling of the water through its relatively thin pad and the formation of dry spots under relatively dry inlet air conditions.

Table 4: Discharge Temperatures (°F)

ECU1 (High Speed)

Tdb	Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	*75*	*76*	
90	*81*	*81*	*84*
100	*88*	*87*	*88*
110		*N/A*	*93*

ECU2 (High Speed)

Tdb	Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	*69*	*73*	
90	*71*	*75*	*79*
100	*75*	*77*	*82*
110		*80*	*84*

ECU3 (High Speed)

Tdb	Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	*67*	*71*	
90	*68*	*73*	*76*
100	*68*	*72*	*78*
110		*74*	*79*

ECU1 (Low Speed)

Tdb	Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	*N/A*	*N/A*	
90	*79*	*80*	*82*
100	*87*	*84*	*87*
110		*N/A*	*92*

ECU2 (Low Speed)

Tdb	Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	*N/A*	*72*	
90	*71*	*75*	*79*
100	*73*	*76*	*80*
110		*79*	*83*

ECU3 (Low Speed)

Tdb	Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	*66*	*71*	
90	*66*	*71*	*76*
100	*66*	*71*	*76*
110		*71*	*76*

Table 5: Unit Effectiveness

ECU1 (High Speed)

Tdb	Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	*32%*	*41%*	
90	*36%*	*44%*	*44%*
100	*34%*	*43%*	*47%*
110		*N/A*	*47%*

ECU2 (High Speed)

Tdb	Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	*74%*	*68%*	
90	*73%*	*73%*	*73%*
100	*74%*	*73%*	*73%*
110		*73%*	*73%*

ECU3 (High Speed)

Tdb	Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	*88%*	*85%*	
90	*90%*	*88%*	*88%*
100	*90%*	*90%*	*88%*
110		*91%*	*90%*

ECU1 (Low Speed)

Tdb	Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	*N/A*	*N/A*	
90	*44%*	*50%*	*50%*
100	*38%*	*51%*	*53%*
110		*N/A*	*53%*

ECU2 (Low Speed)

Tdb	Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	*N/A*	*74%*	
90	*77%*	*78%*	*77%*
100	*78%*	*78%*	*80%*
110		*78%*	*78%*

ECU3 (Low Speed)

Tdb	Wet-bulb Temperature (°F)		
(°F)	65	70	75
80	*95%*	*92%*	
90	*96%*	*95%*	*94%*
100	*97%*	*96%*	*94%*
110		*97%*	*98%*

Figures 8 and 9 show examples of the process for each test unit on psychrometric charts, using actual test data for high primary fan speed and one particular set of inlet conditions (100°F DB, 70°F WB). The figures are meant to help describe the constant wet-bulb process through the direct stage of an evaporative cooler, and to graphically describe the effectiveness. The resulting outlet temperatures were 86°F for ECU1, 78°F for ECU2, and 74°F for ECU3, resulting in effectiveness values of 46%, 73%, and 90%, respectively. Figure 9 also includes the process for the secondary air through the indirect stage of ECU3. The results show that this process does not follow a line of constant wet-bulb, but is steeper due to the heat absorbed from the primary air stream. On the primary-air side, it shows cooling with no moisture addition through the indirect stage, and then a constant wet-bulb process through the direct cooler. The indirect stage reduced the dry-bulb temperature of the air by 12°F to 88°F, but the wet-bulb temperature of the inlet air was only reduced by 3°F to 67°F. The effectiveness of the direct stage acting on the new condition was 70%.

Figure 8: Process Diagram for ECU1 & ECU2

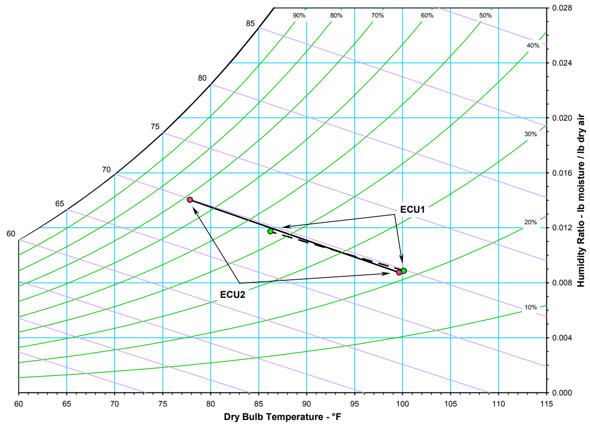
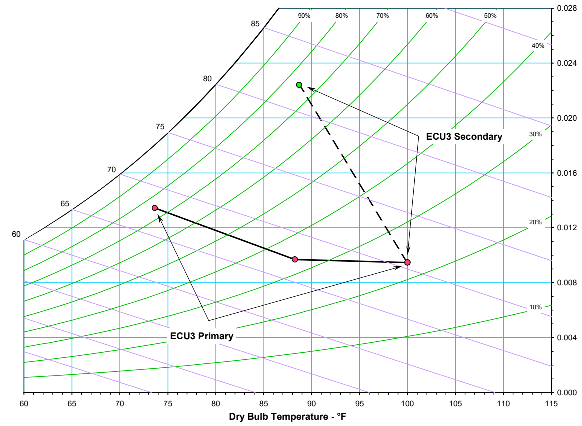


Figure 9: Process Diagram for ECU3



Figures 10 and 11 (in the Appendix) are charts of power consumption as a function of primary airflow rate for all of the test units. (The airflow rate is the measured outlet airflow adjusted to the density of the inlet 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. As expected, the two-stage ECU3 uses the most power (due to an additional fan and pump) and supplied the least airflow (due to the added resistance of the indirect stage). The fact that the resistance of the indirect stage is the cause for the reduced airflow is obvious from the second chart, which shows that the unrestricted test points for ECU3 coincide with the data for increased resistance on ECU2.

Another point of interest is that the fan motor in ECU1 was ½-hp, while that in ECU2 and ECU3 was a ¾-hp motor. The larger horsepower motor is used in ECU2 to overcome the added resistance of the thicker pad. The lower horsepower motor in ECU1 used more power because it pushed more airflow through a much smaller resistance, and power consumption tends to rise with the cube of the airflow rate. One final comment: the data for the secondary fan on ECU3 was added to the second chart, and shows that this axial fan used about the same amount of power as the centrifugal primary fan on low speed, but moved only about a third of the airflow.

Figures 12 and 13 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). In the first chart, which uses the total system power, it is interesting to see that for ECU1 and ECU2, the airflow efficiency increases at low speed, but it decreases for ECU3. This is because only the speed of the primary fan is changed; the secondary fan continues to draw the same airflow and power. The second figure again shows that the primary air fan is just responding to the added resistance of the indirect stage. (Aside: The default air handler blower efficiency used in ARI Standard 210 for calculating SEER is 2.74 cfm/W, or slightly higher than that for the ECU3 secondary fan.)

Figures 14 and 15 show the data contained in Table 5 in graphical form. (Actually, 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). These charts show how stable the effectiveness was for the advanced units (ECU2 and ECU3), and how the results for the traditional-style ECU1 are relatively more scattered.

The next group of six charts (*Figures 16 through 21*) examine the relative cooling capacity and overall system energy efficiency for each of the three test units. As discussed previously, the capacity is defined as the ability of the unit to maintain a space at 80°F. Thus, if the outlet temperature is above this value, the capacity of the unit is negative. (What this really means is the space will reach an equilibrium temperature 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. These results combine the effects of air velocity and outlet temperature.

The charts for ECU1 show that despite its greater airflow rate, it is unable to deliver supply air at a temperature below 80°F except when the inlet air temperature is below about 88°F. Its greater airflow is reflected in the relatively steep slope of the data trends as a function of temperature. ECU2 shows better results, but does lose its cooling ability at the combined high temperature and humidity test conditions. Because of its greater overall effectiveness, ECU3 was able to provide some cooling capacity under all test conditions, and actually showed very little sensitivity to the entering dry-bulb temperature.

Figure 22 shows an alternative way of considering the cooling capacity: the ability of the test unit to sensibly cool the outside air. In this case, the capacity (again given in tons) is calculated by taking the temperature difference between the entering and supply air temperatures, and multiplying by the air mass flow rate and specific heat. The results were then graphed as a function of the entering air wet-bulb depression. With this arrangement, the advanced evaporative cooler (ECU2) comes out looking the best, with a combination of good airflow and effectiveness. The results for ECU1 are the lowest because of its poor effectiveness, despite providing the greatest airflow. ECU3 shows a lesser penalty for low airflow, despite having the best effectiveness.

The next set of three charts (*Figures 23 through 25*) examine each unit's sensitivity to increasing the backpressure on the unit (or the external resistance to flow). Included in the charts are the measurements of airflow rate, total unit power, and effectiveness. The values are graphed as the relative magnitude compared to the measured parameters with no backpressure (0" of water column), the values of which are given in the chart legend. The test conditions for ECU1 and ECU2 were at 90°F DB / 65°F WB, while ECU3 was tested at 100°F DB / 70°F WB. All of the units show similar decreases in airflow and power, and increased effectiveness as the backpressure is increased.

The last two charts (*Figures 26 and 27*) examine the relative water consumption rates. The first chart displays the flow rates recorded by low-range paddle wheel flow meters installed in the makeup water lines. Unfortunately, many flow conditions were below the lower sensitivity limit of these meters, so no useful information was recorded. In particular, no useful flow data were recorded at all for ECU3 because the individual flow rates to the two sections were always under the sensitivity limit. 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. The point at which the trend lines intersect with a wet-bulb depression of zero indicates the constant bleed flow from the pump discharge to a drain, which is meant to reduce the concentration of dissolved solids in the pan water. For ECU1, the bleed was held at about 4 gallons per hour (gph), and for ECU2, it was about 6 gph. Reducing the fan speed reduced the evaporation water consumption, but not the bleed flow.

The second figure looks at the total water consumption from measurements on the air side of the process. It is determined by taking the moisture (humidity ratio) rise from inlet to outlet, and multiplying by the air mass flow rate. Since this only measures the amount of water evaporated into the air, it does not include the excess water used in the bleed. One surprising result from this examination is that ECU2 and ECU3 evaporated water at the same rate when on high speed, even though ECU3 evaporated part of it to a secondary air stream. The overall water usage would still be slightly greater for ECU3 because the two separate water systems would need two distinct bleeds. ECU3 consumed more water than ECU2 at low speed because the fan speed of the secondary air system was not reduced, so its water consumption rate remained the same.

The results from this graphical analysis are summarized in Table 6. The table lists the slopes of the graphed trend lines to indicate the correlation between water consumption and the entering wet-bulb depression. The table shows that the measured water consumption rate for ECU2 corresponded almost exactly to the calculated air-side evaporation rate, although ECU1 showed higher consumption on the water-side than the air-side.

Table 6: Summary of Water Consumption Rates

Test Unit	GPH	GPH per °F of Wet-bulb Depression			
	Makeup Water Flowmeter Measurements			Air-side Measurements	
	Bleed Flow	High Speed	Low Speed	High Speed	Low Speed
ECU1	3.89	0.209	0.162	0.167	0.135
ECU2	5.98	0.292	0.209	0.293	0.198
ECU3	N/A	N/A	N/A	0.293	0.213

DISCUSSION

The test units were all obtained through distributors and not directly from the factory, so as not to bias the results with a special unit. The downside to this is that there is no assurance of the quality of the units. In particular, the old-style evaporative cooler (ECU1) performed much worse than was expected. The unit came with pads made out of a plastic material instead of the traditional "aspen wood" natural fiber pads, and it is possible that the water-repelling nature of plastic may need additional time to break down. The test units were all run for an extended period prior to testing as an attempt to ensure that the pads were thoroughly wetted during the tests, and during this time, a small quantity of dish soap was added to accelerate the removal of the water repelling films. This still may have not been long enough, and could help to explain the greater level of scatter in the results for this unit. The problem could also be the result of poor distribution of the water to the pads, or inadequate water flow.

ECU1 had another problem before testing, but this was most likely the result of a setup error. When the bleed line was installed, it was inadvertently placed where it contacted the belt wheel on the fan. Eventually, the moving fan cut through the tube, and the resulting spray inside the unit may have caused a short in the blower motor. The problem was evidenced by the motor switching from high to low speed on its own, and not being able to hold at high speed. The solution was to obtain a replacement motor, which unfortunately took six weeks to get from the factory, but at least was simple to change.

Evaporative coolers fall into a niche market as a low-cost option providing a limited cooling effect, somewhere between ventilation fans and conventional air conditioners. Under hot, dry conditions, the systems work very well and may provide adequate comfort to the occupants. As humidity rises, their cooling capability decreases, but their power demand remains relatively constant. In contrast, as outside temperatures rise, the cooling capability of a conventional air conditioner will also decrease somewhat, but its power demand will increase significantly. Evaporative coolers also cannot be easily adapted to existing ductwork designed for the lower airflows for heating or air conditioning, and need to have their own distribution system. An evaporative cooler could be used in conjunction with a conventional system as an alternative cooling method when the outside conditions allow it to provide adequate comfort. This would effectively reduce the hours of operation for a conventional system (much like a whole house fan), but may not alleviate the impact on the residential peak demand when the air conditioner needs to be turned on.

The addition of an indirect evaporative precooler may not be economically feasible at this time. While it did improve the conditions of the supplied air in terms of lower temperatures and comparatively low humidity, it did restrict the delivered airflow. Its biggest drawback is cost, which was significantly more than the direct evaporative cooler to which it was attached. (The prices paid for the test units were $522 for ECU1, $735 for ECU2, and $1,100 for the add-on precooler.) The higher cost of this product is probably due to the complexity of constructing the indirect evaporative heat exchanger, and also the result of low production rates since it is difficult to market very many at these prices.

CONCLUSIONS

This study tested three samples of currently available evaporative cooling systems. The test units included a traditional-style cooler having thin (~1") plastic pads, an advanced whole-house evaporative cooler that using a thick (8") cellulose pad, and an add-on indirect evaporative precooler attached to the advanced unit. The tests on each system were done at various conditions of outside air temperature, humidity, unit fan speed, and external resistance. Many findings are discussed in previous sections of the report and are summarized below.

1. The effectiveness of all of the test units was less than what was expected prior to testing. This was especially true of the traditional-style system, which had effectiveness numbers about half of what the advanced system reached. This may also represent evidence to suggest that these older systems may be performing worse in actual use than was suspected.
2. The low effectiveness results may be the result of not allowing enough time to break in the evaporative pads. This suggests that there may be some period of peak performance for evaporative cooler pads between the time of installation and the time of replacement when they have become contaminated with dissolved solids.
3. Because of having less flow resistance and a smaller motor, the traditional-style unit did have the best fan efficiency (in terms of cfm/W) and provided the most airflow. The resulting higher velocities may allow its higher temperature air to feel cooler than it actually is.
4. The advanced direct evaporative cooler provided the most cooling capability in terms of the sensible heat removed from the supplied outside air. The traditional unit provided more airflow, but at a higher temperature; and the combined indirect/direct cooler provided cooler temperatures, but with less airflow.
5. The addition of the indirect precooler to the advanced system increased the cooling effect at high speed by up to 6°F (representing an increase in unit effectiveness by 17 percentage points), but reduced the delivered airflow by 26%. At low speed, the cooling effect was raised by up to 8°F (representing an increase in effectiveness by 19 percentage points), but with a 30% reduction in airflow.
6. The reduced airflow and high cost of the indirect system may make it unattractive for most consumers, except those willing to pay the extra for slightly better comfort provided through less humid supply air.

The test units were off-the-shelf purchases, and may not be representatives the best or even the average quality of units produced of these models. As such, the results of these tests are somewhat inconclusive in determining the relative merits of different types of evaporative cooler units. The results do support the concept of the advanced systems providing better performance than the traditional systems, and are therefore worthy of financial incentives to offset their higher cost. However, there is still the concern that the tested traditional system was a poor example, and thus the differences may not be as great as the results suggest. Still, since one new unit was found to be operating this poorly, there may be new or existing systems that are worse.

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1. ASHRAE, *"Handbook of Fundamentals, Chapter 27: Climatic Design Information"*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, NE, Atlanta, GA 30329, 2001.
2. ASHRAE "*Climatic Data for Region X: Arizona, California, Hawaii, Nevada"*, Fifth Edition, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, NE, Atlanta, GA 30329, May 1982.
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4. ASHRAE/ANSI Standard 41.2-1992: "*Standard Methods for Laboratory Airflow Measurements*", American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, NE, Atlanta, GA 30329, 1992.
5. ASHRAE Standard 133-2001, *"Method of Testing Direct Evaporative Air Coolers"*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, NE, Atlanta, GA 30329, 2001.
6. ASHRAE Standard 143-2000, *"Method of Test for Rating Indirect Evaporative Coolers*", American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, NE, Atlanta, GA 30329, 2000.
7. Australian Standard AS 2913-2000, "*Evaporative Air Conditioning Equipment*", 2000.

Figure 10: Total Unit Power

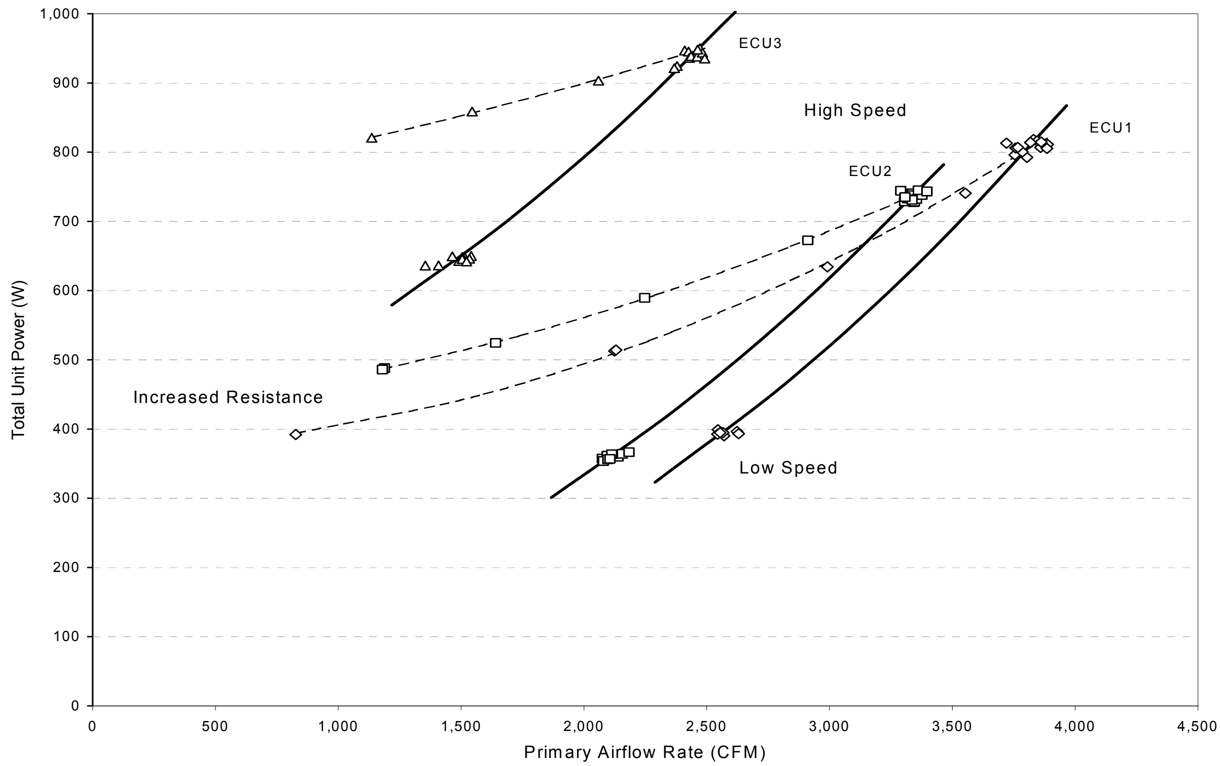


Figure 11: Primary Fan Power

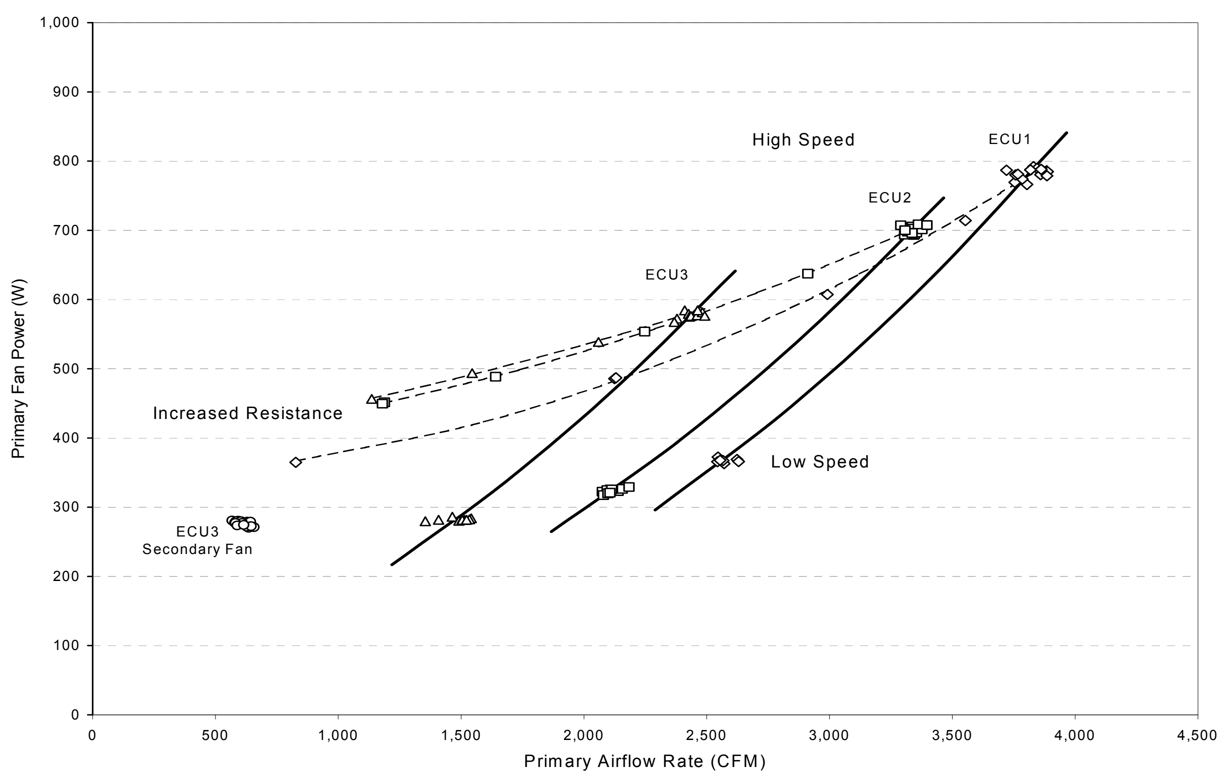


Figure 12: Total Unit cfm / W

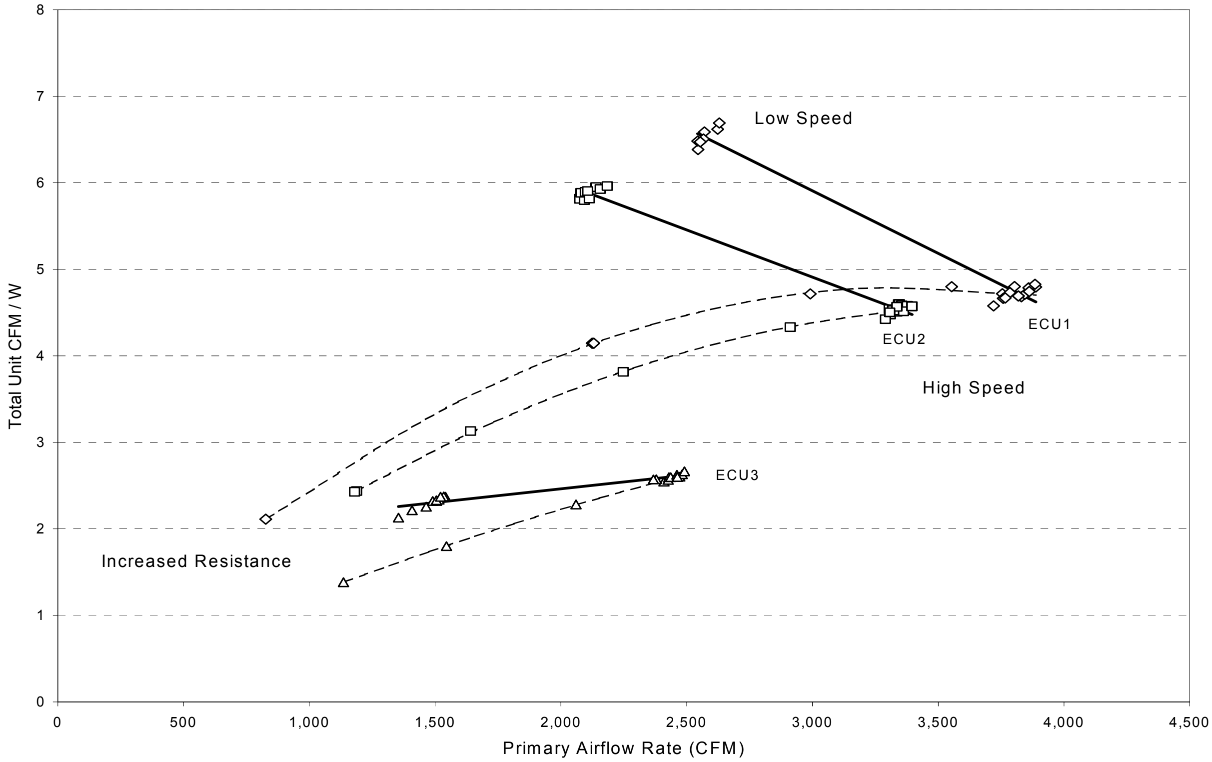


Figure 13: Primary Fan cfm / W

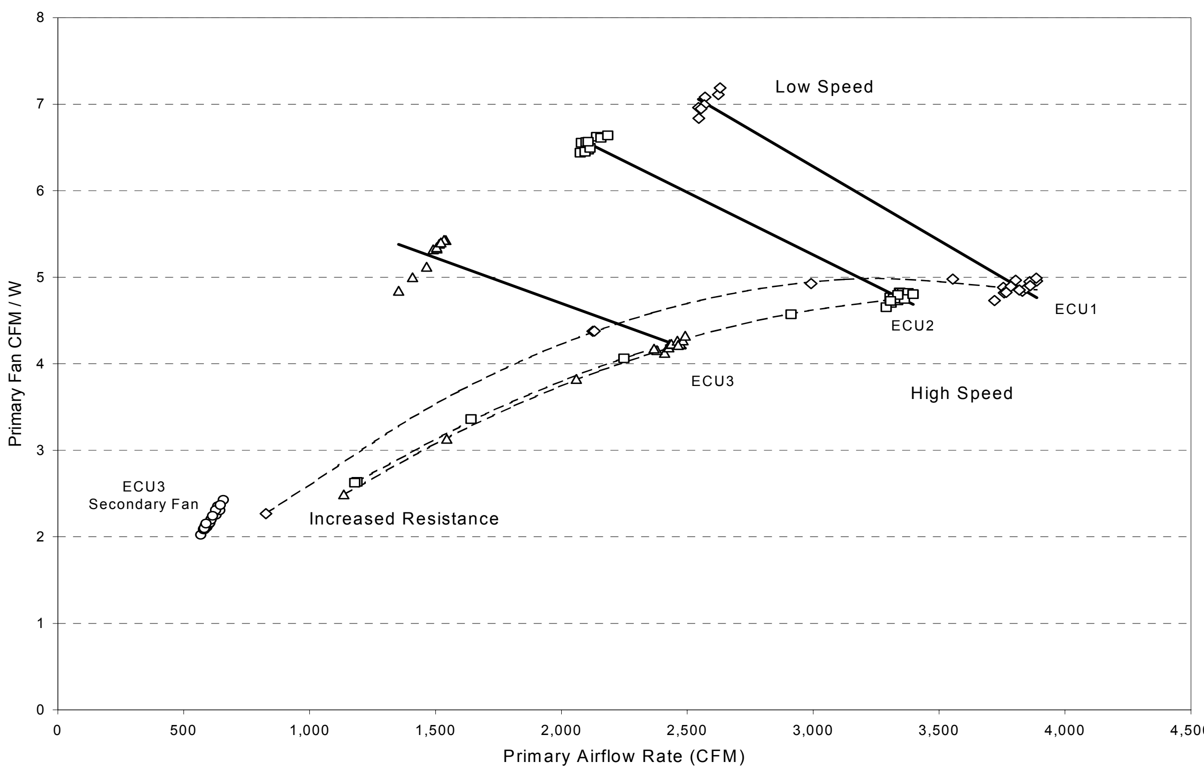


Figure 14: Unit Effectiveness versus Intake Dry-bulb Temperature

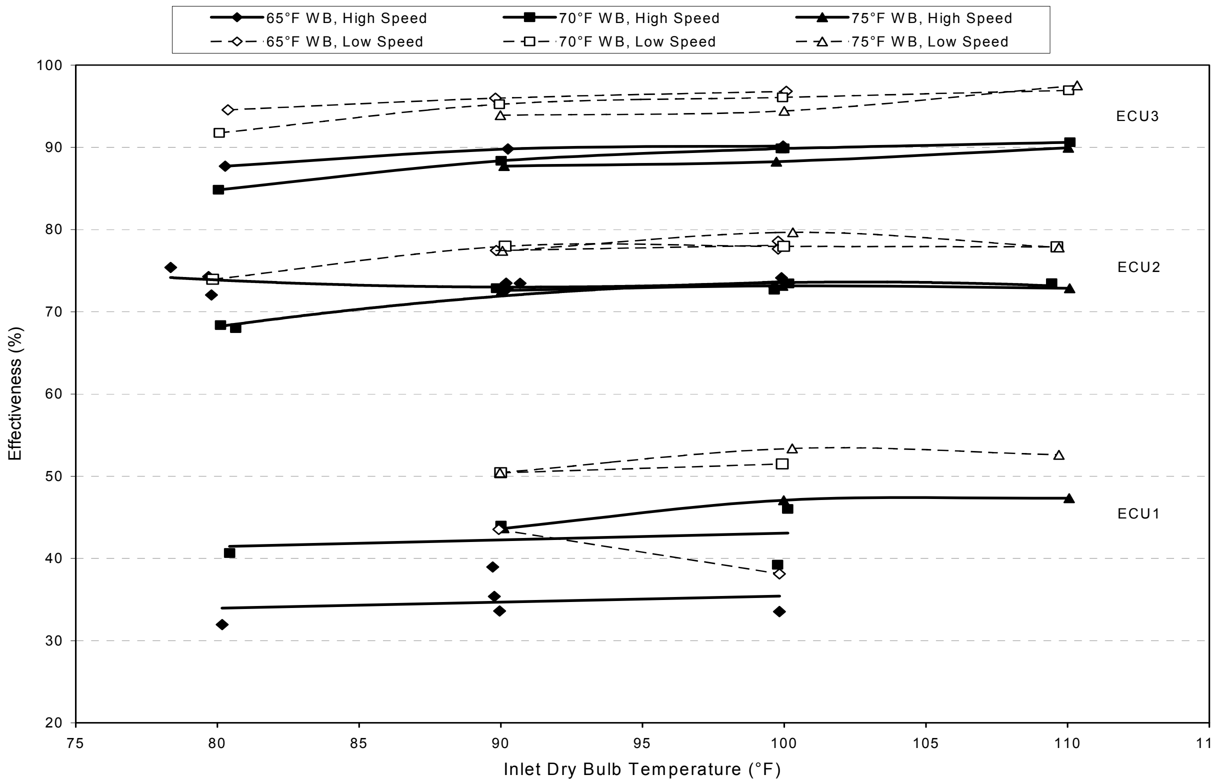


Figure 15: Unit Effectiveness versus Intake Wet-bulb Depression

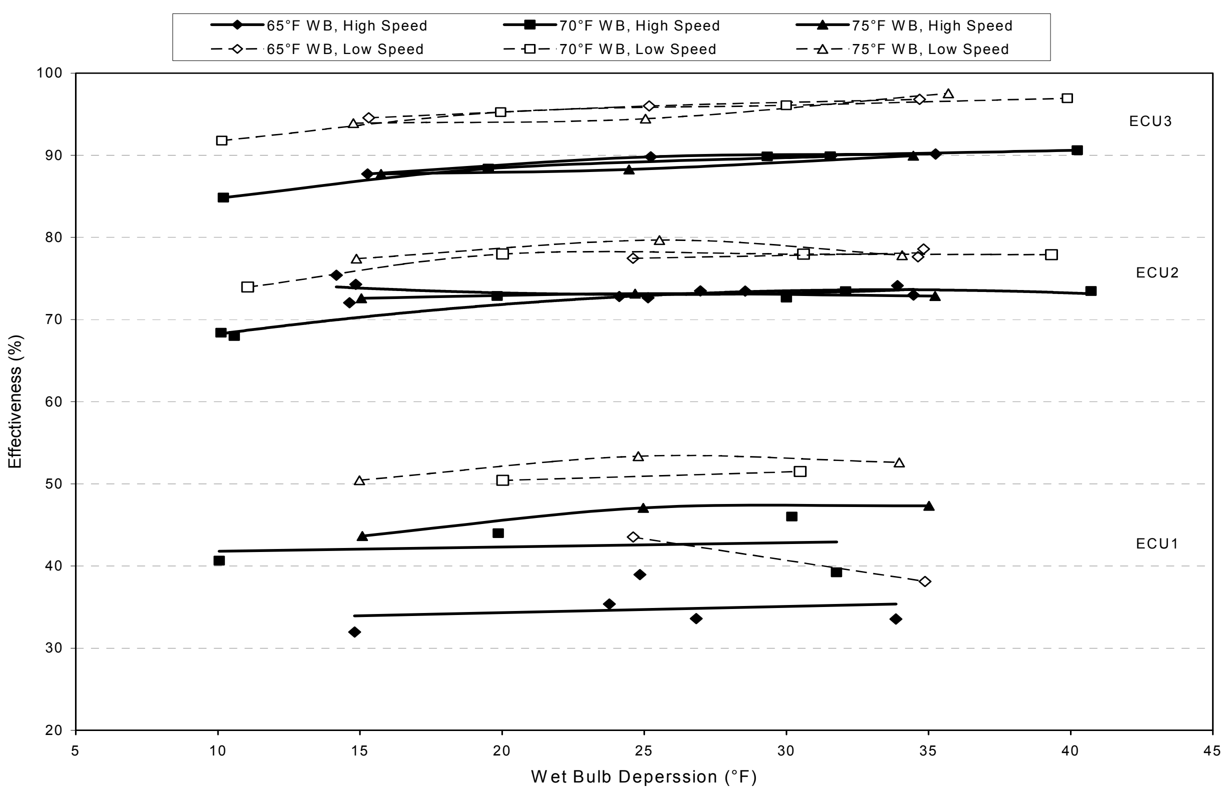


Figure 16: ECU1 Capacity
(80°F Room Temperature Basis)

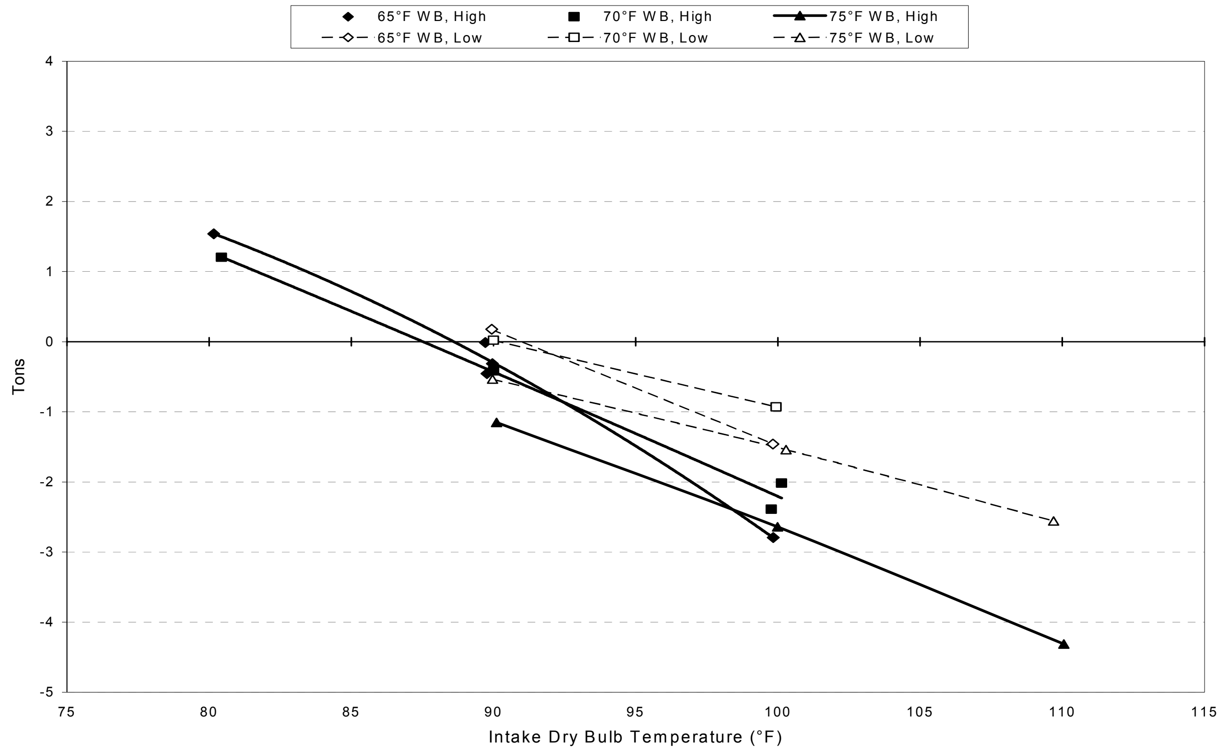


Figure 17: ECU1 Energy Efficiency Ratio
(80°F Room Temperature Basis)

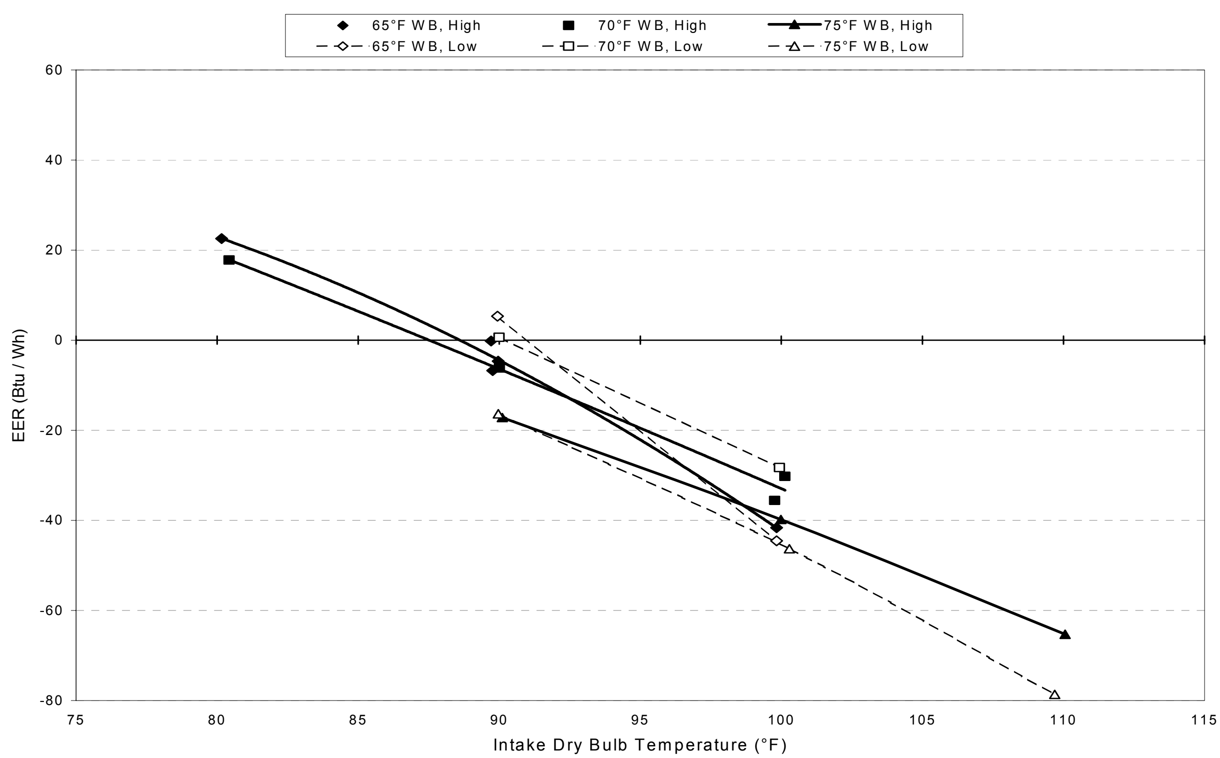


Figure 18: ECU2 Capacity
(80°F Room Temperature Basis)

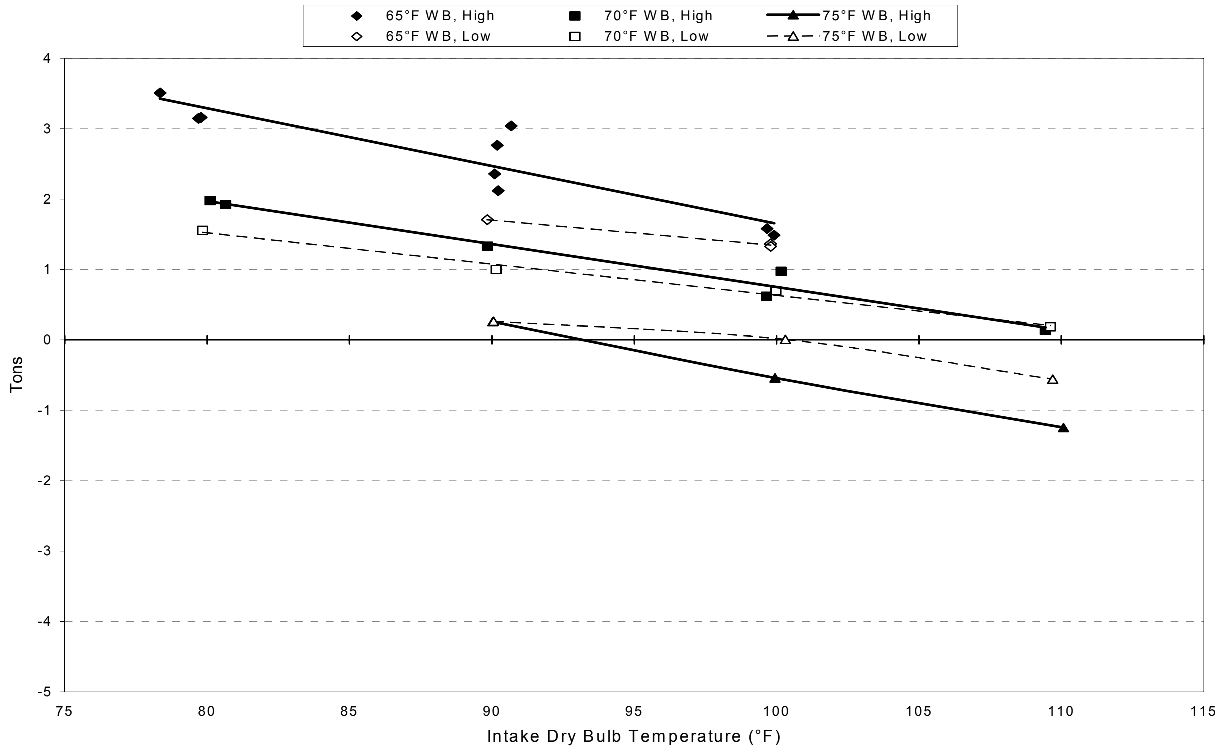


Figure 19: ECU2 Energy Efficiency Ratio
(80°F Room Temperature Basis)

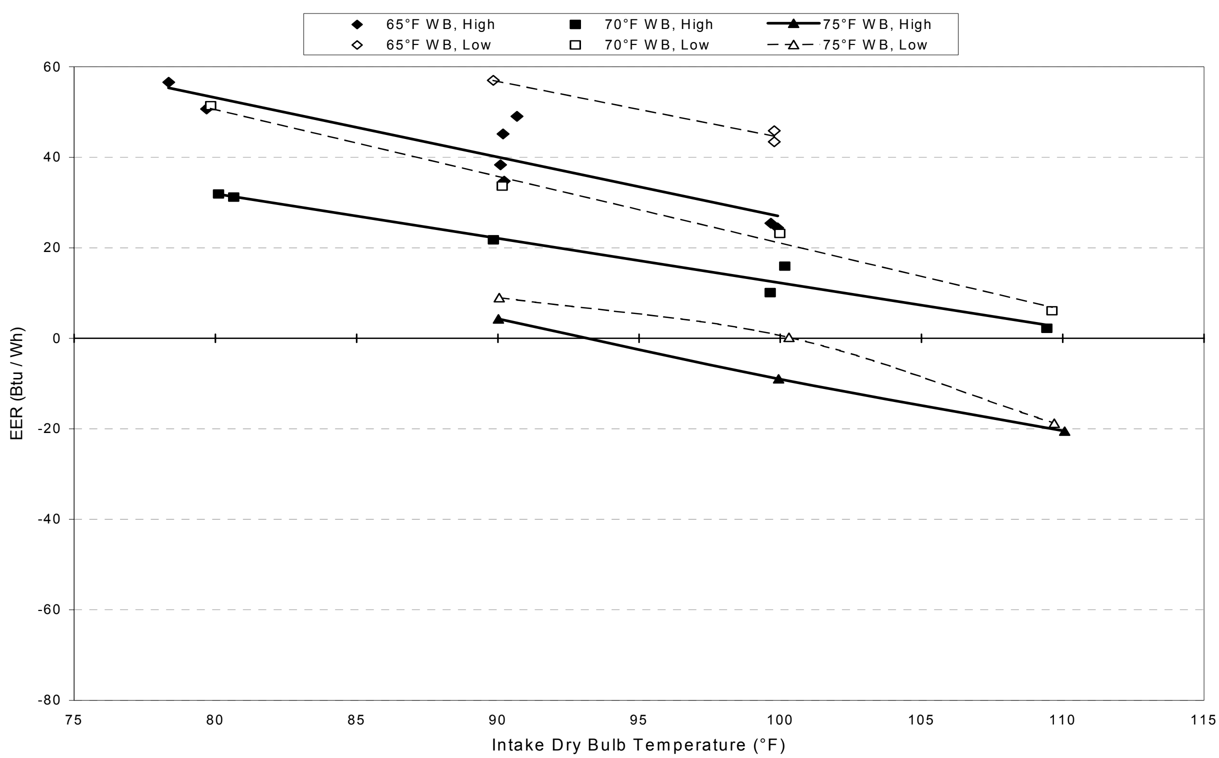


Figure 20: ECU3 Capacity
(80°F Room Temperature Basis)

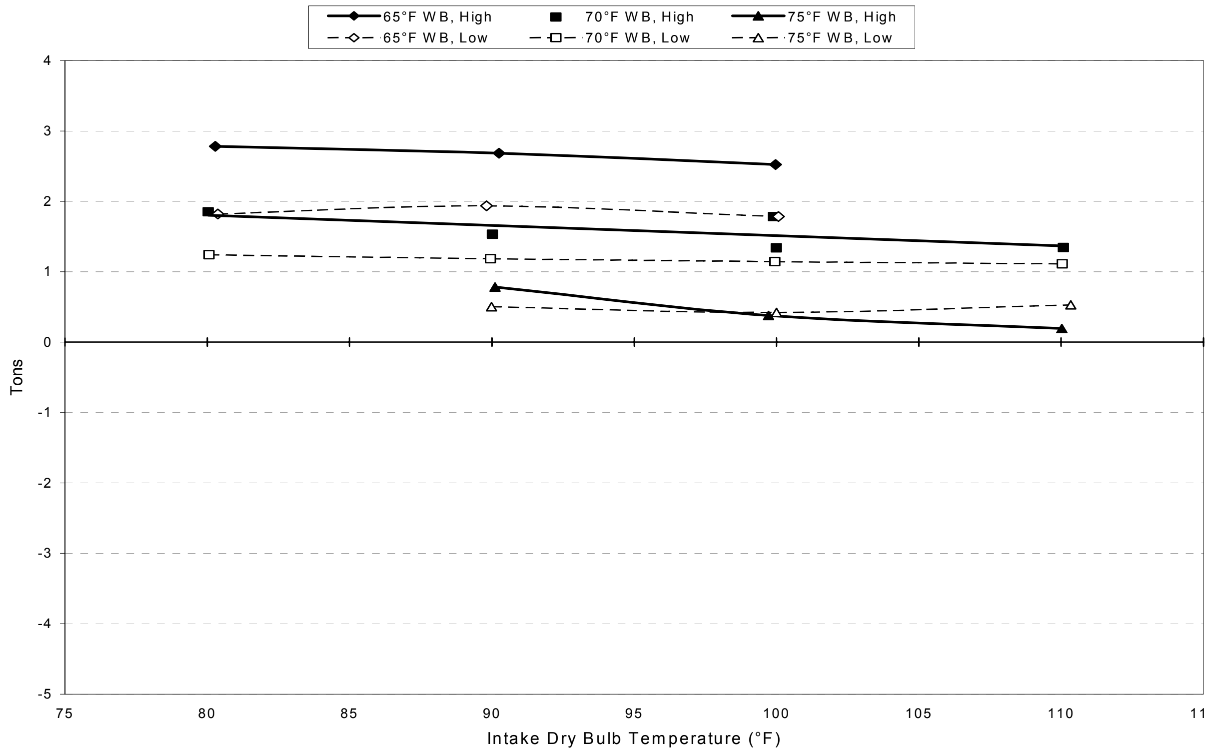


Figure 21: ECU3 Energy Efficiency Ratio
(80°F Room Temperature Basis)

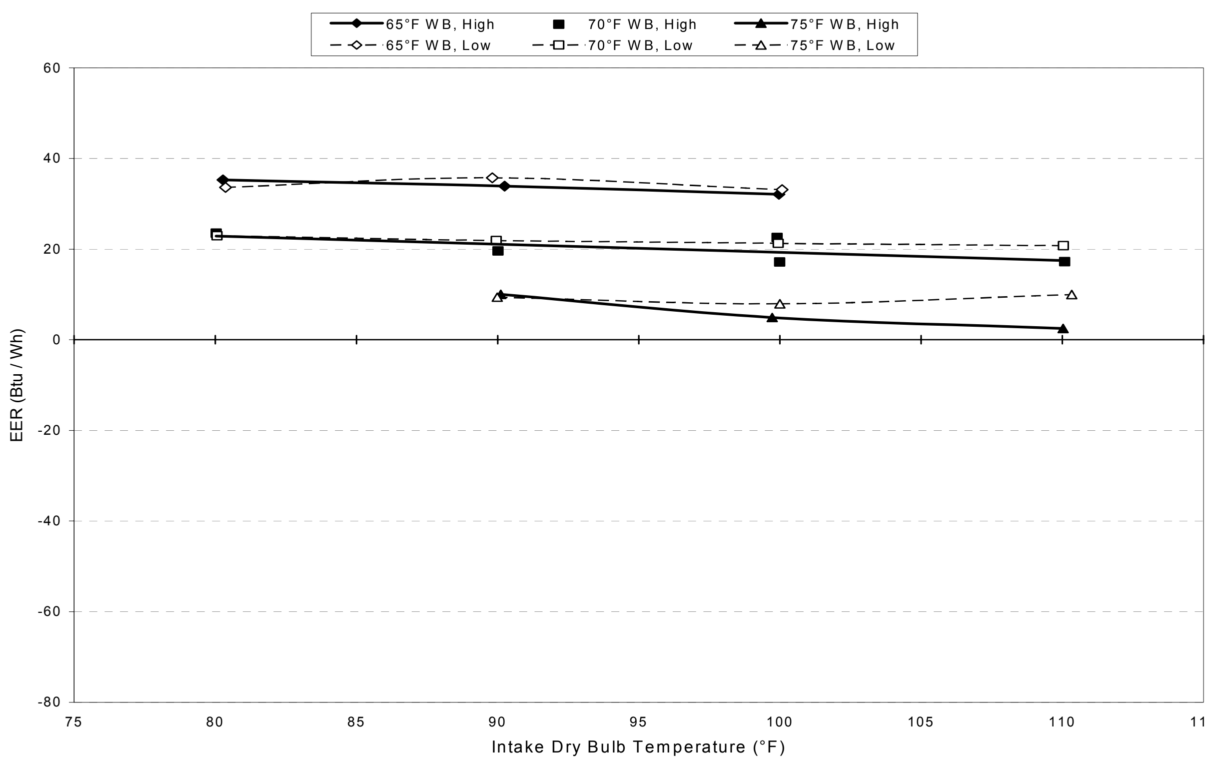


Figure 22: Sensible Cooling of Intake Air

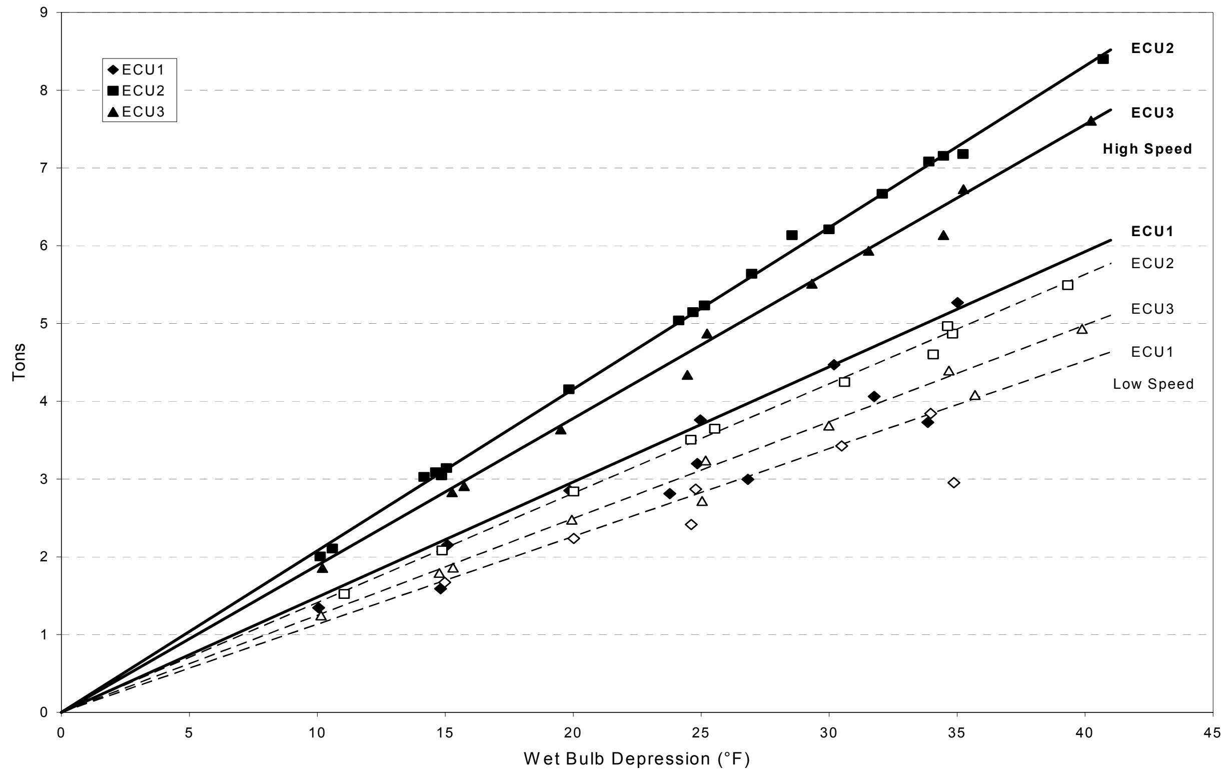


Figure 23: ECU1 Performance Sensitivity to Backpressure

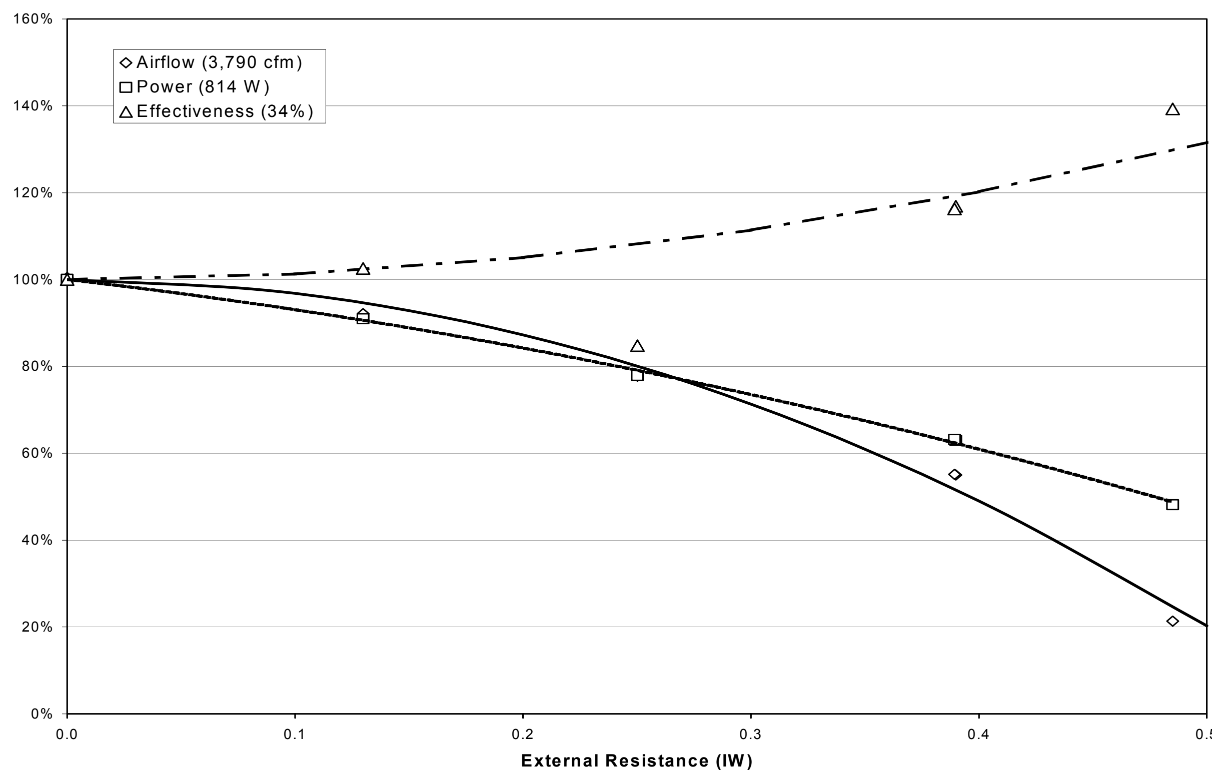


Figure 24: ECU2 Performance Sensitivity to Backpressure

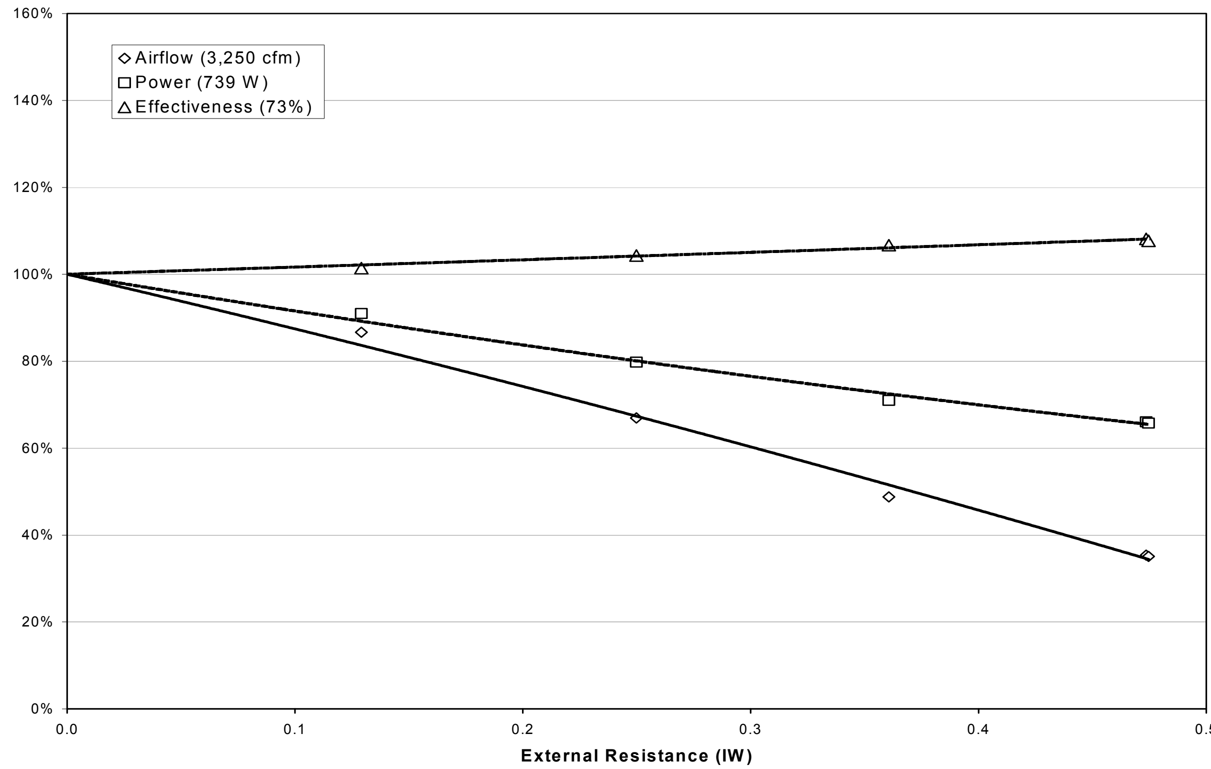


Figure 25: ECU3 Performance Sensitivity to Backpressure

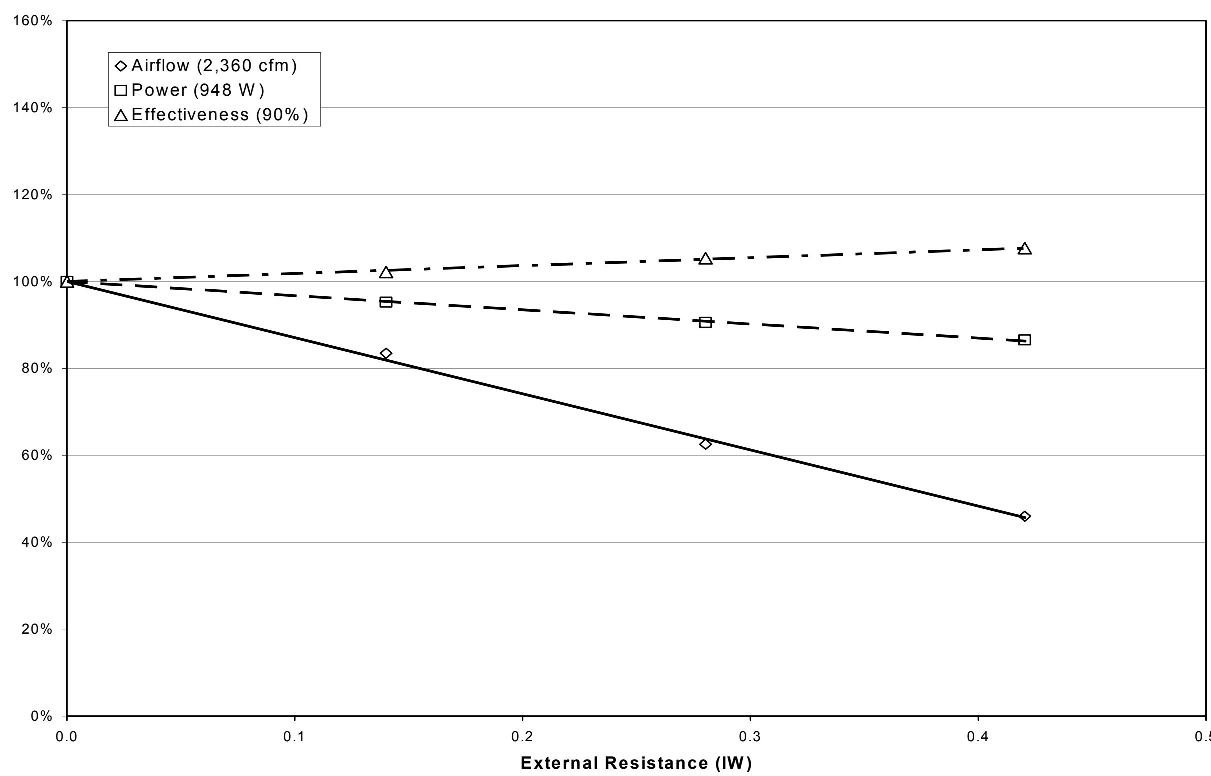


Figure 26: Measured Make-up Water Flow

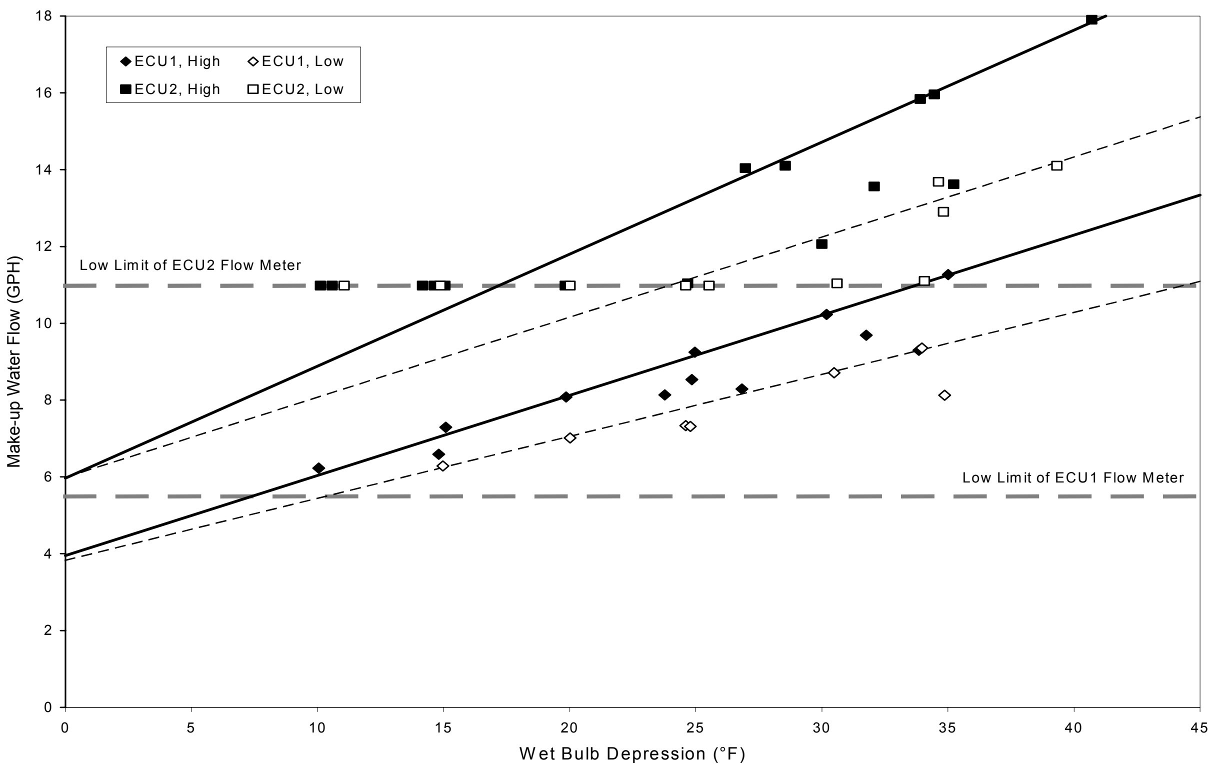
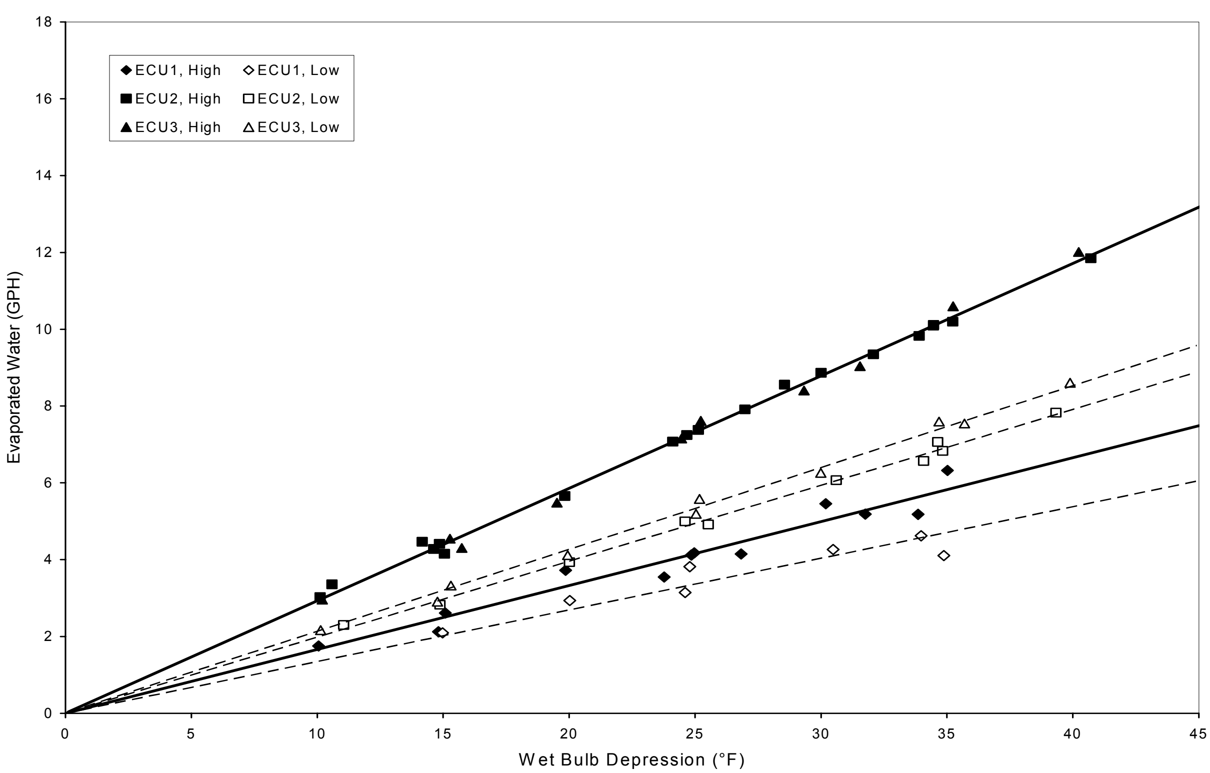


Figure 27: Calculated Evaporation Rate



ECU1 Test Data

Test Summary Information	High Speed												
General													
Date (all year 2003)	Nov 20	Nov 18	Nov 20	Nov 24	Nov 24	Nov 20	Nov 18	Nov 18	Nov 24	Nov 17	Nov 20	Nov 17	Nov 17
Start Time	7:52a	8:28a	1:07p	10:08a	2:15p	2:41p	1:01p	9:58a	3:16p	3:52p	3:47p	12:03p	12:53p
Duration (minutes)	30	29	30	30	30	30	30	30	30	30	30	30	30
Barometric Pressure (in. of Hg)	29.62	29.86	29.58	29.70	29.61	29.56	29.79	29.87	29.61	29.79	29.56	29.79	29.78
Nominal Test Conditions													
Inlet Dry Bulb Temperature (°F)	80	80	90	90	90	90	90	90	100	100	100	100	110
Inlet Wet Bulb Temperature (°F)	65	70	65	65	65	65	70	75	65	70	70	75	75
Inlet Air Properties													
Dry Bulb Temperature (°F)	80.2	80.4	89.8	90.1	90.0	89.7	90.0	90.1	99.8	100.1	99.8	100.0	110.1
Dew Point Temperature (°F)	57.3	65.9	52.2	41.3	44.8	49.6	60.5	69.0	43.7	53.8	49.5	64.5	59.1
Relative Humidity (%) - calculated	45.6	61.3	27.8	18.3	21.0	25.3	37.3	50.1	14.8	21.4	18.5	31.6	19.4
Wet Bulb Temperature (°F) - calc.	65.4	70.4	66.0	62.0	63.1	64.9	70.2	75.0	66.0	69.9	68.0	75.0	75.0
Wet Bulb Depression (°F)	14.8	10.1	23.8	28.1	26.8	24.9	19.9	15.1	33.9	30.2	31.8	25.0	35.0
Outlet Air Properties													
Dry Bulb Temperature (°F)	75.4	76.3	81.4	83.3	80.9	80.0	81.3	83.5	88.5	86.2	87.3	88.2	93.5
Dew Point Temperature (°F)	60.0	67.7	57.4	47.6	52.4	56.1	64.9	71.5	53.3	61.5	57.6	68.9	66.7
Relative Humidity (%) - calculated	58.8	74.7	44.0	28.8	37.1	43.9	57.7	67.2	30.2	43.5	36.7	52.9	41.6
Wet Bulb Temperature (°F) - calc.	65.3	70.3	65.8	61.9	63.2	64.7	70.0	74.8	66.1	69.5	67.8	74.4	74.5
Pan Water Temperature (°F)	65.8	69.9	67.5	63.4	64.3	66.3	70.6	74.8	67.9	70.8	69.5	74.9	75.4
Performance													
Dry Bulb ΔT (°F)	-4.7	-4.1	-8.4	-6.8	-9.0	-9.7	-8.7	-6.6	-11.4	-13.9	-12.5	-11.8	-16.6
Wet Bulb ΔT (°F)	0.0	-0.1	-0.2	0.0	0.1	-0.2	-0.1	-0.2	0.1	-0.4	-0.2	-0.6	-0.5
Effectiveness (%)	32.0	40.7	35.4	24.1	33.6	39.0	44.0	43.7	33.6	46.0	39.2	47.1	47.3
Intake Airflow Rate (CFM)	3,830	3,720	3,890	3,820	3,860	3,850	3,770	3,760	3,890	3,790	3,860	3,760	3,800
Sensible Capacity (tons)	1.54	1.21	-0.46	-1.10	-0.32	-0.01	-0.42	-1.15	-2.80	-2.02	-2.39	-2.64	-4.31
Makeup Water Usage (gph)	6.6	6.2	8.1	7.1	8.3	8.5	8.1	7.3	9.3	10.2	9.7	9.3	11.3
Power Consumption													
Fan (W)	792	787	785	787	788	789	781	780	779	774	780	769	766
Pump (W)	26	26	26	27	26	27	26	26	26	26	27	27	26
Total (W)	818	813	811	814	814	816	807	806	805	800	807	796	792
Unit Power Factor	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.77	0.77	0.77	0.77
Fan CFM / W	4.84	4.73	4.96	4.85	4.90	4.87	4.82	4.82	4.99	4.89	4.95	4.88	4.97
Unit CFM / W	4.68	4.58	4.80	4.69	4.74	4.71	4.67	4.66	4.83	4.73	4.79	4.72	4.80
Energy Efficiency Ratio (Btu/Wh)	51.26	31.18	34.72	49.07	45.15	38.34	21.77	4.27	24.42	10.09	15.96	0.22	-20.53

ECU1 Test Data (Continued)

Test Summary Information	Low Speed							Variable Resistance					
General													
Date (all year 2003)	Nov 20	Nov 18	Nov 18	Nov 24	Nov 17	Nov 17	Nov 17	Nov 24	Nov 24	Nov 24	Nov 24	Nov 24	Nov 24
Start Time	1:56p	12:19p	11:00a	4:03p	2:59p	10:31a	2:00p	2:15p	1:30p	10:51a	1:10p	12:36p	11:20a
Duration (minutes)	30	30	30	30	30	30	30	30	20	20	11	30	20
Barometric Pressure (in. of Hg)	29.57	29.80	29.84	29.60	29.78	29.82	29.78	29.61	29.62	29.69	29.62	29.62	29.68
Nominal Test Conditions													
Inlet Dry Bulb Temperature (°F)	90	90	90	100	100	100	110	90	90	90	90	90	90
Inlet Wet Bulb Temperature (°F)	65	70	75	65	70	75	75	65	65	65	65	65	65
Inlet Air Properties													
Dry Bulb Temperature (°F)	89.9	90.0	90.0	99.8	99.9	100.3	109.7	90.0	90.0	89.6	90.4	90.2	90.0
Dew Point Temperature (°F)	50.5	60.2	69.0	40.5	52.8	65.2	60.8	44.8	34.1	36.6	32.4	32.1	32.6
Relative Humidity (%) - calculated	26.0	36.9	50.3	13.1	20.8	32.1	20.8	21.0	13.8	15.4	12.7	12.6	13.0
Wet Bulb Temperature (°F) - calc.	65.3	70.0	75.0	65.0	69.5	75.5	75.7	63.1	59.7	60.3	59.4	59.3	59.4
Wet Bulb Depression (°F)	24.6	20.0	15.0	34.9	30.5	24.8	34.0	26.8	30.3	29.3	31.0	31.0	30.6
Outlet Air Properties					External Resistance (IW) -->			**0.00**	**0.13**	**0.25**	**0.39**	**0.39**	**0.49**
Dry Bulb Temperature (°F)	79.2	79.9	82.4	86.5	84.2	87.1	91.8	80.9	79.6	81.3	78.3	78.2	75.6
Dew Point Temperature (°F)	57.6	65.3	71.9	52.5	61.7	70.9	68.6	52.4	45.7	45.7	47.1	46.8	49.8
Relative Humidity (%) - calculated	47.5	61.1	70.6	31.1	46.8	58.9	46.7	37.1	30.2	28.7	33.3	33.1	40.3
Wet Bulb Temperature (°F) - calc.	65.2	69.9	74.8	65.1	69.1	75.4	75.2	63.2	59.8	60.4	59.9	59.7	60.2
Pan Water Temperature (°F)	67.3	70.5	74.9	67.2	70.8	75.4	76.2	64.3	62.2	62.8	62.0	61.8	62.6
Performance													
Dry Bulb ΔT (°F)	-10.7	-10.1	-7.6	-13.3	-15.7	-13.2	-17.9	-9.0	-10.4	-8.4	-12.2	-12.1	-14.3
Wet Bulb ΔT (°F)	-0.1	-0.1	-0.2	0.1	-0.4	-0.1	-0.6	0.1	0.1	0.1	0.5	0.5	0.8
Effectiveness (%)	43.5	50.4	50.4	38.1	51.5	53.4	52.6	33.6	34.5	28.5	39.3	39.1	46.8
Intake Airflow Rate (CFM)	2,620	2,550	2,540	2,630	2,570	2,550	2,570	3,860	3,550	2,990	2,120	2,130	830
Sensible Capacity (tons)	0.18	0.02	-0.54	-1.46	-0.93	-1.54	-2.56	-0.32	0.13	-0.33	0.32	0.34	0.31
Makeup Water Usage (gph)	7.3	7.0	6.3	8.1	8.7	7.3	9.4	8.3	8.4	7.0	7.2	7.1	5.5
Power Consumption													
Fan (W)	369	368	366	366	367	372	363	788	714	607	486	487	365
Pump (W)	27	27	27	27	27	26	27	26	27	27	27	27	27
Total (W)	396	395	393	393	395	399	390	814	741	634	513	514	392
Unit Power Factor	0.71	0.71	0.71	0.71	0.71	0.72	0.70	0.78	0.74	0.69	0.60	0.60	0.48
Fan CFM / W	7.11	6.95	6.96	7.19	6.99	6.84	7.08	4.90	4.98	4.93	4.38	4.38	2.27
Unit CFM / W	6.62	6.47	6.48	6.69	6.51	6.38	6.59	4.74	4.80	4.72	4.15	4.15	2.11
Energy Efficiency Ratio (Btu/Wh)	57.01	33.63	8.99	45.88	23.17	-8.95	-18.74	45.15	57.49	50.47	45.72	36.54	36.42

ECU2 Test Data

Test Summary Information	High Speed															
General																
Date (all year 2003)	Nov 20	Nov 18	Nov 21	Nov 20	Nov 20	Nov 24	Nov 24	Nov 18	Nov 18	Nov 21	Nov 24	Nov 17	Nov 20	Nov 17	Nov 21	Nov 17
Start Time	7:52a	8:28a	9:52a	1:07p	2:41p	10:08a	2:15p	1:01p	9:58a	3:16p	3:16p	3:52p	3:47p	10:31a	1:03p	12:53p
Duration (minutes)	30	29	30	30	30	30	30	30	30	30	30	30	30	30	30	30
Barometric Pressure (in. of Hg)	29.62	29.86	29.70	29.58	29.56	29.70	29.61	29.79	29.87	29.64	29.61	29.79	29.56	29.82	29.64	29.78
Nominal Test Conditions																
Inlet Dry Bulb Temperature (°F)	80	80	80	90	90	90	90	90	90	100	100	100	100	100	110	110
Inlet Wet Bulb Temperature (°F)	65	70	70	65	65	65	65	70	75	65	65	70	70	75	70	75
Inlet Air Properties																
Dry Bulb Temperature (°F)	79.8	80.7	80.1	90.2	90.1	90.7	90.2	89.9	90.0	99.7	99.9	99.7	100.2	99.9	109.4	110.1
Dew Point Temperature (°F)	57.1	65.3	65.4	52.1	49.5	41.2	44.8	60.4	69.0	41.4	43.7	53.4	49.3	64.9	42.9	58.6
Relative Humidity (%) - calculated	45.9	59.5	61.0	27.3	24.9	17.9	20.8	37.3	50.2	13.7	14.8	21.5	18.2	32.1	10.8	19.1
Wet Bulb Temperature (°F) - calc.	65.2	70.1	70.0	66.1	65.0	62.1	63.2	70.0	75.0	65.2	66.0	69.7	68.1	75.3	68.7	74.8
Wet Bulb Depression (°F)	14.6	10.6	10.1	24.1	25.1	28.6	27.0	19.8	15.1	34.5	33.9	30.0	32.1	24.7	40.7	35.2
Outlet Air Properties																
Dry Bulb Temperature (°F)	69.3	73.5	73.2	72.7	71.8	69.7	70.4	75.4	79.1	74.5	74.8	77.8	76.6	81.9	79.5	84.4
Dew Point Temperature (°F)	63.2	69.1	68.9	63.1	61.8	58.1	59.6	67.7	73.3	60.9	61.8	66.4	64.4	73.0	64.2	71.6
Relative Humidity (%) - calculated	81.1	86.3	86.5	72.0	70.6	66.7	68.8	76.9	82.4	62.5	64.1	68.0	66.0	74.5	59.5	65.5
Wet Bulb Temperature (°F) - calc.	65.2	70.4	70.2	66.3	65.2	62.3	63.4	70.0	74.9	65.5	66.2	70.0	68.3	75.4	69.1	75.1
Pan Water Temperature (°F)	65.2	70.7	70.1	66.4	65.4	62.3	63.4	70.1	75.0	65.6	66.3	70.0	68.5	75.4	69.2	75.2
Performance																
Dry Bulb ΔT (°F)	-10.5	-7.2	-6.9	-17.6	-18.3	-21.0	-19.8	-14.5	-10.9	-25.2	-25.1	-21.8	-23.6	-18.1	-29.9	-25.7
Wet Bulb ΔT (°F)	0.1	0.4	0.2	0.2	0.2	0.2	0.2	0.0	-0.1	0.3	0.2	0.3	0.2	0.1	0.3	0.3
Effectiveness (%)	72.1	68.0	68.4	72.8	72.7	73.5	73.5	72.9	72.6	73.0	74.1	72.7	73.5	73.2	73.5	72.9
Intake Airflow Rate (CFM)	3,340	3,310	3,290	3,340	3,340	3,400	3,310	3,320	3,310	3,360	3,340	3,350	3,350	3,340	3,380	3,350
Sensible Capacity (tons)	3.16	1.92	1.98	2.12	2.36	3.04	2.77	1.33	0.26	1.58	1.49	0.62	0.97	-0.54	0.13	-1.25
Makeup Water Usage (gph)				11.3	11.4	14.1	14.0			16.0	15.8	12.1	13.6		18.1	13.6
Power Consumption																
Fan (W)	705	704	707	697	702	707	700	698	694	708	696	702	696	693	701	694
Pump (W)	35	35	37	35	36	36	35	35	35	37	35	35	36	34	37	34
Total (W)	740	740	744	733	738	743	735	733	729	745	731	737	732	727	738	728
Unit Power Factor	0.63	0.63	0.61	0.63	0.62	0.62	0.62	0.63	0.63	0.61	0.62	0.63	0.62	0.62	0.61	0.62
Fan CFM / W	4.73	4.70	4.65	4.79	4.75	4.80	4.73	4.76	4.77	4.75	4.79	4.77	4.82	4.82	4.82	4.82
Unit CFM / W	4.51	4.48	4.42	4.56	4.52	4.57	4.50	4.53	4.54	4.51	4.57	4.54	4.58	4.60	4.58	4.60
Energy Efficiency Ratio (Btu / Wh)	51.26	31.18	31.89	34.72	38.34	49.07	45.15	21.77	4.27	25.44	24.42	10.09	15.96	-8.95	2.18	-20.53

ECU2 Test Data (Continued)

Test Summary Information	Low Speed										Variable Resistance						
General																	
Date (all year 2003)	Nov 21	Nov 20	Nov 18	Nov 18	Nov 21	Nov 24	Nov 17	Nov 17	Nov 21	Nov 17	Nov 24	Nov 24	Nov 24	Nov 24	Nov 24	Nov 24	Nov 24
Start Time	10:31a	1:56p	12:19p	11:00a	4:30p	4:03p	2:59p	12:03p	1:56p	2:00p	10:08a	2:15p	1:30p	10:51a	1:10p	11:20a	12:36p
Duration (minutes)	30	30	30	30	30	30	30	30	30	30	30	30	20	20	11	20	30
Barometric Pressure (in. of Hg)	29.70	29.57	29.80	29.84	29.64	29.60	29.78	29.79	29.63	29.78	29.70	29.61	29.62	29.69	29.62	29.68	29.62
Nominal Test Conditions																	
Inlet Dry Bulb Temperature (°F)	80	90	90	90	100	100	100	100	110	110	90	90	90	90	90	90	90
Inlet Wet Bulb Temperature (°F)	70	65	70	75	65	65	70	75	70	75	65	65	65	65	65	65	65
Inlet Air Properties																	
Dry Bulb Temperature (°F)	79.8	89.8	90.2	90.1	99.8	99.8	100.0	100.3	109.6	109.7	90.7	90.2	89.8	90.1	90.3	90.3	90.1
Dew Point Temperature (°F)	63.6	50.4	60.4	69.3	41.2	40.6	52.6	63.8	47.6	60.6	41.2	44.8	33.7	36.3	31.9	32.2	31.6
Relative Humidity (%) - calculated	57.6	25.9	37.0	50.7	13.5	13.2	20.6	30.6	12.9	20.6	17.9	20.8	13.6	15.0	12.5	12.7	12.4
Wet Bulb Temperature (°F) - calc.	68.8	65.2	70.1	75.2	65.2	65.0	69.4	74.8	70.3	75.6	62.1	63.2	59.6	60.4	59.3	59.3	59.1
Wet Bulb Depression (°F)	11.1	24.6	20.0	14.9	34.6	34.8	30.6	25.5	39.3	34.1	28.6	27.0	30.3	29.7	31.1	30.9	31.0
Outlet Air Properties								External Resistance (IW) -->			**73.05**	**77.35**	**66.43**	**70.23**	**68.98**	**70.59**	**70.29**
Dry Bulb Temperature (°F)	71.7	70.8	74.6	78.5	72.9	72.4	76.1	80.0	79.0	83.2	69.7	70.4	67.3	67.3	66.0	65.7	65.6
Dew Point Temperature (°F)	67.9	62.9	68.3	73.9	61.8	61.6	66.9	72.7	67.1	73.3	58.1	59.6	55.5	57.1	56.5	57.2	57.0
Relative Humidity (%) - calculated	87.9	76.1	80.9	85.7	68.2	68.7	73.1	78.5	67.0	72.1	66.7	68.8	65.8	69.6	71.4	74.0	73.8
Wet Bulb Temperature (°F) - calc.	69.1	65.5	70.2	75.2	65.6	65.2	69.7	74.7	70.7	75.9	62.3	63.4	60.0	60.9	60.1	60.4	60.2
Pan Water Temperature (°F)	69.0	65.7	70.3	75.3	65.7	65.3	69.6	74.3	70.8	75.8	62.3	63.4	59.9	60.7	59.7	59.8	59.7
Performance																	
Dry Bulb ΔT (°F)	-8.2	-19.1	-15.6	-11.5	-26.9	-27.4	-23.9	-20.3	-30.6	-26.5	-21.0	-19.8	-22.6	-22.8	-24.4	-24.6	-24.5
Wet Bulb ΔT (°F)	0.3	0.3	0.1	0.0	0.4	0.3	0.3	-0.1	0.4	0.3	0.2	0.2	0.4	0.5	0.8	1.0	1.1
Effectiveness (%)	73.9	77.5	78.0	77.4	77.6	78.6	78.0	79.7	77.9	77.8	73.5	73.5	74.5	76.7	78.4	79.5	79.2
Intake Airflow Rate (CFM)	2,110	2,140	2,100	2,080	2,180	2,110	2,090	2,110	2,160	2,080	3,400	3,310	2,910	2,250	1,640	1,190	1,180
Sensible Capacity (tons)	1.56	1.71	1.00	0.27	1.33	1.37	0.70	0.01	0.18	-0.56	3.04	2.77	3.22	2.48	2.00	1.48	1.48
Makeup Water Usage (gph)					13.7	12.9			14.1	11.1	14.1	14.0	14.2	11.5			
Power Consumption																	
Fan (W)	326	323	320	317	329	321	325	325	326	322	707	700	637	553	488	451	449
Pump (W)	38	37	36	36	37	36	37	34	38	35	36	35	35	36	36	37	36
Total (W)	363	360	356	354	366	357	361	360	364	357	743	735	672	589	524	488	486
Unit Power Factor	0.58	0.58	0.59	0.58	0.59	0.58	0.58	0.59	0.58	0.58	0.62	0.62	0.58	0.52	0.48	0.45	0.45
Fan CFM / W	6.49	6.62	6.56	6.55	6.64	6.57	6.45	6.47	6.61	6.44	4.80	4.73	4.57	4.06	3.36	2.63	2.63
Unit CFM / W	5.82	5.95	5.89	5.88	5.96	5.90	5.80	5.85	5.93	5.81	4.57	4.50	4.33	3.81	3.13	2.44	2.43
Energy Efficiency Ratio (Btu / Wh)	51.40	57.01	33.63	8.99	43.43	45.88	23.17	0.22	6.08	-18.74	49.07	45.15	57.49	50.47	45.72	36.42	36.54

ECU3 Test Data

Test Summary Information	High Speed										
General											
Date (all year 2003)	Dec 11	Dec 12	Dec 15	Dec 12	Dec 12	Dec 15	Dec 12	Dec 12	Dec 11	Dec 15	Dec 11
Start Time	8:31a	9:20a	9:36a	3:02p	11:05a	12:56p	4:05p	5:25p	2:28p	1:46p	12:50p
Duration (minutes)	30	30	30	30	30	30	30	20	30	30	30
Barometric Pressure (in. of Hg)	29.64	29.81	29.94	29.76	29.80	29.90	29.76	29.77	29.68	29.89	29.66
Nominal Test Conditions											
Inlet Dry Bulb Temperature (°F)	80	80	90	90	90	100	100	100	100	110	110
Inlet Wet Bulb Temperature (°F)	65	70	65	70	75	65	70	70	75	70	75
Inlet Air Properties											
Dry Bulb Temperature (°F)	80.3	80.0	90.3	90.0	90.1	100.0	100.0	99.9	99.7	110.1	110.0
Dew Point Temperature (°F)	56.5	65.2	49.3	61.2	68.0	39.1	55.6	50.1	65.1	45.5	60.3
Relative Humidity (%) - calculated	44.2	60.6	24.6	38.3	48.3	12.4	23.0	18.8	32.5	11.8	20.3
Wet Bulb Temperature (°F) - calc.	65.0	69.8	65.0	70.5	74.4	64.7	70.7	68.4	75.3	69.9	75.6
Wet Bulb Depression (°F)	15.3	10.2	25.2	19.5	15.7	35.3	29.3	31.5	24.5	40.2	34.5
Indirect Secondary Outlet Air											
Dry Bulb Temperature (°F)	75.9	77.3	81.3	83.1	84.6	86.9	88.7	88.0	90.6	93.7	95.9
Dew Point Temperature (°F)	69.6	73.5	73.9	77.1	79.6	76.4	79.9	78.6	82.0	82.4	85.0
Relative Humidity (%) - calculated	80.9	88.0	78.1	82.0	85.0	71.2	75.4	73.9	76.1	69.8	71.0
Wet Bulb Temperature (°F) - calc.	71.5	74.6	75.9	78.6	80.8	79.1	82.0	80.8	83.9	84.9	87.3
Pan Water Temperature (°F)	70.8	73.8	73.8	77.6	79.9	76.9	80.7	79.5	83.5	82.2	86.3
Indirect Primary Outlet Air Properties											
Dry Bulb Temperature (°F)	75.8	76.0	80.9	82.3	83.6	86.7	88.3	87.5	91.1	94.1	98.0
Dew Point Temperature (°F)	56.3	65.3	49.3	61.8	68.5	39.0	56.2	51.1	65.1	45.4	60.1
Relative Humidity (%) - calculated	50.8	69.6	33.3	50.1	60.6	18.6	33.8	28.7	42.4	18.8	28.7
Wet Bulb Temperature (°F) - calc.	63.4	68.7	61.9	68.5	72.9	60.1	67.4	64.8	73.0	64.8	72.2
Wet Bulb Depression (°F)	12.4	7.3	19.0	13.7	10.6	26.6	20.8	22.7	18.2	29.3	25.8
Direct Outlet Air Properties											
Dry Bulb Temperature (°F)	66.9	71.4	67.6	72.8	76.3	68.2	73.6	71.6	78.1	73.6	79.0
Dew Point Temperature (°F)	61.7	67.9	59.7	66.9	71.7	57.1	65.2	62.2	71.3	62.1	69.7
Relative Humidity (%) - calculated	83.4	88.8	75.7	81.9	85.6	67.8	74.9	72.4	79.5	67.1	73.2
Wet Bulb Temperature (°F) - calc.	63.5	69.0	62.5	68.7	73.0	61.2	67.9	65.4	73.2	66.0	72.4
Pan Water Temperature (°F)	63.1	68.7	62.0	68.5	72.7	61.1	67.7	65.0	73.2	65.8	72.3
Performance											
Indirect / Secondary Dry Bulb ΔT (°F)	-4.4	-2.7	-8.9	-6.9	-5.5	-13.1	-11.3	-12.0	-9.2	-16.4	-14.1
Indirect / Secondary Wet Bulb ΔT (°F)	6.5	4.7	10.9	8.1	6.4	14.4	11.3	12.5	8.7	15.0	11.7
Indirect / Primary Dry Bulb ΔT (°F)	-4.5	-4.1	-9.4	-7.8	-6.6	-13.3	-11.7	-12.4	-8.6	-16.0	-12.0
Indirect / Primary Dew Point ΔT (°F)	-0.2	0.1	0.1	0.6	0.5	-0.1	0.6	1.0	0.0	-0.2	-0.2
Indirect Stage Effectiveness (%)	29.3	40.1	37.2	39.8	41.7	37.7	40.0	39.4	35.1	39.7	34.8
Direct Stage Dry Bulb ΔT (°F)	-8.9	-4.6	-13.3	-9.5	-7.2	-18.5	-14.6	-15.9	-13.0	-20.5	-19.0
Direct Stage Wet Bulb ΔT (°F)	0.1	0.3	0.7	0.2	0.1	1.2	0.5	0.5	0.3	1.2	0.2
Direct Stage Effectiveness (%)	72.0	62.8	69.8	69.1	68.1	69.5	70.1	70.3	71.5	69.8	73.5
Overall Unit Effectiveness (%)	87.7	84.8	89.8	88.4	87.7	90.1	89.9	89.9	88.3	90.6	89.9
Indirect Secondary Airflow Rate (CFM)	630	570	580	580	580	580	590	590	630	590	640
Primary Intake Airflow Rate (CFM)	2,410	2,430	2,470	2,440	2,430	2,480	2,460	2,460	2,370	2,490	2,380
Sensible Capacity (tons)	2.78	1.85	2.68	1.53	0.78	2.52	1.34	1.78	0.38	1.34	0.19
Power Consumption											
Indirect Stage Fan (Secondary) (W)	279	281	280	277	277	277	275	277	272	274	270
Indirect Stage Pump (W)	49	49	49	49	49	49	49	49	48	48	49
Indirect Stage Total (W)	328	330	329	326	326	326	324	327	320	322	319
Indirect Stage Power Factor	0.76	0.76	0.76	0.76	0.76	0.76	0.76	0.75	0.76	0.76	0.75
Indirect Stage Fan CFM / W	2.25	2.02	2.09	2.09	2.09	2.10	2.13	2.12	2.31	2.15	2.35
Direct Stage Fan (W)	584	579	585	576	575	582	577	584	567	576	572
Direct Stage Pump (W)	34	36	36	36	35	36	37	37	34	37	33
Direct Stage Total (W)	618	616	621	612	610	618	614	621	601	613	606
Direct Stage Power Factor	0.54	0.54	0.54	0.54	0.53	0.54	0.53	0.53	0.53	0.54	0.53
Direct Stage Fan CFM / W	4.13	4.19	4.23	4.23	4.23	4.27	4.27	4.22	4.17	4.32	4.16
Total Unit Power (W)	947	945	950	939	936	944	938	948	921	935	925
Total Unit CFM / W	2.55	2.57	2.60	2.60	2.60	2.63	2.62	2.60	2.57	2.67	2.57
Energy Efficiency Ratio (Btu/Wh)	35.27	23.51	33.90	19.58	10.02	32.07	17.14	22.57	4.91	17.23	2.51

ECU3 Test Data (Continued)

Test Summary Information	Low Speed										Variable Resistance			
General														
Date (all year 2003)	Dec 11	Dec 12	Dec 15	Dec 12	Dec 12	Dec 15	Dec 12	Dec 11	Dec 15	Dec 11	Dec 12	Dec 12	Dec 12	Dec 12
Start Time	10:19a	8:38a	11:01a	2:27p	12:25p	12:15p	4:47p	3:21p	2:33p	11:03a	5:25p	5:50p	6:15p	6:33p
Duration (minutes)	30	30	30	30	30	30	30	30	30	30	20	20	15	20
Barometric Pressure (in. of Hg)	29.67	29.80	29.92	29.76	29.78	29.91	29.77	29.68	29.89	29.67	29.77	29.77	29.78	29.78
Nominal Test Conditions														
Inlet Dry Bulb Temperature (°F)	80	80	90	90	90	100	100	100	110	110	100	100	100	100
Inlet Wet Bulb Temperature (°F)	65	70	65	70	75	65	70	75	70	75	70	70	70	70
Inlet Air Properties														
Dry Bulb Temperature (°F)	80.4	80.1	89.8	90.0	90.0	100.1	99.9	100.0	110.0	110.3	99.9	100.1	100.0	100.1
Dew Point Temperature (°F)	56.6	65.3	48.7	60.3	69.4	41.3	53.9	64.4	46.5	58.1	50.1	49.9	49.8	49.6
Relative Humidity (%) - calculated	44.1	60.8	24.4	37.1	50.9	13.4	21.7	31.4	12.2	18.6	18.8	18.6	18.6	18.4
Wet Bulb Temperature (°F) - calc.	65.1	69.9	64.6	70.0	75.2	65.4	70.0	75.0	70.2	74.6	68.4	68.3	68.3	68.2
Wet Bulb Depression (°F)	15.3	10.1	25.2	20.0	14.8	34.7	30.0	25.0	39.9	35.7	31.5	31.7	31.7	31.8
Indirect Secondary Outlet Air														
Dry Bulb Temperature (°F)	74.8	76.5	79.2	81.4	83.7	84.7	86.5	88.8	91.3	93.1	88.0	87.2	86.0	84.9
Dew Point Temperature (°F)	68.5	72.5	71.1	75.0	78.7	73.8	76.9	80.1	79.2	82.0	78.6	77.4	75.9	74.5
Relative Humidity (%) - calculated	80.9	87.5	76.2	80.9	85.0	70.0	73.4	75.4	67.9	70.4	73.9	72.8	72.0	71.3
Wet Bulb Temperature (°F) - calc.	70.4	73.6	73.4	76.7	79.9	76.7	79.3	82.1	82.1	84.5	80.8	79.8	78.5	77.2
Pan Water Temperature (°F)	70.1	73.2	72.5	76.3	79.5	75.7	78.9	82.1	80.8	83.8	79.5	78.8	78.0	77.1
Indirect Primary Outlet Air Properties														
Dry Bulb Temperature (°F)	74.3	75.0	77.7	79.8	82.0	83.0	84.5	88.6	89.4	94.7	87.5	86.3	84.1	81.4
Dew Point Temperature (°F)	56.3	65.5	48.6	61.0	70.0	41.1	54.7	64.4	46.3	57.2	51.1	50.9	51.0	50.7
Relative Humidity (%) - calculated	53.4	72.2	36.0	52.7	67.0	22.7	36.1	44.7	22.6	28.7	28.7	29.7	31.9	34.4
Wet Bulb Temperature (°F) - calc.	62.9	68.5	60.4	67.3	73.4	59.4	65.5	71.9	63.6	69.8	64.8	64.4	63.7	62.6
Wet Bulb Depression (°F)	11.4	6.5	17.3	12.5	8.6	23.6	19.0	16.8	25.8	24.8	22.7	21.9	20.4	18.8
Direct Outlet Air Properties										External Resistance (IW) -->	**84.44**	**84.16**	**83.45**	**82.14**
Dry Bulb Temperature (°F)	65.9	70.8	65.7	71.0	76.1	66.5	71.1	76.4	71.4	75.5	71.6	70.9	70.0	69.3
Dew Point Temperature (°F)	61.6	68.0	59.1	66.1	72.6	57.8	64.1	70.6	62.1	67.5	62.2	62.1	61.9	61.4
Relative Humidity (%) - calculated	86.0	90.8	79.4	84.6	88.8	73.5	78.4	82.4	72.6	76.3	72.4	73.7	75.4	76.2
Wet Bulb Temperature (°F) - calc.	63.1	68.8	61.5	67.6	73.6	61.0	66.4	72.3	65.3	70.0	65.4	65.1	64.6	64.1
Pan Water Temperature (°F)	62.7	68.5	61.2	67.3	73.3	60.8	66.0	72.1	65.2	69.5	65.0	64.7	64.1	63.2
Performance														
Indirect / Secondary Dry Bulb ΔT (°F)	-5.6	-3.6	-10.6	-8.6	-6.3	-15.4	-13.5	-11.2	-18.7	-17.3	-12.0	-12.9	-14.0	-15.2
Indirect / Secondary Wet Bulb ΔT (°F)	5.3	3.7	8.8	6.7	4.7	11.3	9.4	7.2	11.9	9.9	12.5	11.5	10.2	9.0
Indirect / Primary Dry Bulb ΔT (°F)	-6.1	-5.1	-12.1	-10.2	-8.0	-17.1	-15.5	-11.4	-20.7	-15.7	-12.4	-13.8	-15.9	-18.6
Indirect / Primary Dew Point ΔT (°F)	-0.3	0.1	0.0	0.7	0.6	-0.2	0.8	0.1	-0.2	-0.9	1.0	1.0	1.2	1.1
Indirect Stage Effectiveness (%)	39.8	49.9	48.0	51.1	54.1	49.4	51.7	45.4	51.8	43.9	39.4	43.5	50.3	58.5
Direct Stage Dry Bulb ΔT (°F)	-8.4	-4.3	-12.1	-8.8	-5.9	-16.5	-13.3	-12.3	-18.0	-19.1	-15.9	-15.4	-14.1	-12.2
Direct Stage Wet Bulb ΔT (°F)	0.2	0.3	1.1	0.4	0.1	1.6	0.9	0.4	1.7	0.1	0.5	0.7	1.0	1.5
Direct Stage Effectiveness (%)	73.6	65.1	69.8	70.4	68.6	69.8	70.3	73.3	69.7	77.1	70.3	70.0	69.0	64.7
Overall Unit Effectiveness (%)	94.6	91.8	96.0	95.2	93.9	96.8	96.1	94.4	96.9	97.5	89.9	91.9	94.7	96.8
Indirect Secondary Airflow Rate (CFM)	640	590	600	600	600	610	610	650	610	660	590	600	610	620
Primary Intake Airflow Rate (CFM)	1,460	1,510	1,540	1,510	1,490	1,530	1,510	1,350	1,520	1,410	2,460	2,060	1,550	1,140
Sensible Capacity (tons)	1.82	1.24	1.94	1.18	0.50	1.78	1.14	0.42	1.11	0.53	1.78	1.60	1.33	1.05
Power Consumption														
Indirect Stage Fan (Secondary) (W)	279	281	280	280	278	277	277	273	274	271	277	278	278	278
Indirect Stage Pump (W)	49	49	49	50	49	49	49	49	49	49	49	49	50	49
Indirect Stage Total (W)	328	330	329	329	327	326	326	322	324	320	327	327	328	327
Indirect Stage Power Factor	0.76	0.76	0.76	0.75	0.76	0.76	0.75	0.75	0.75	0.75	0.75	0.75	0.75	0.75
Indirect Stage Fan CFM / W	2.31	2.11	2.16	2.15	2.15	2.20	2.20	2.37	2.24	2.43	2.12	2.16	2.20	2.23
Direct Stage Fan (W)	286	281	284	282	280	283	282	280	282	282	584	539	494	456
Direct Stage Pump (W)	35	37	37	37	36	37	37	35	37	34	37	37	38	37
Direct Stage Total (W)	321	318	321	320	316	320	319	314	318	316	621	576	531	494
Direct Stage Power Factor	0.53	0.53	0.53	0.52	0.52	0.53	0.52	0.52	0.52	0.52	0.53	0.49	0.46	0.44
Direct Stage Fan CFM / W	5.12	5.39	5.43	5.33	5.32	5.43	5.35	4.84	5.40	5.00	4.22	3.82	3.13	2.49
Total Unit Power (W)	649	648	650	649	643	646	645	636	642	636	948	903	859	821
Total Unit CFM / W	2.26	2.34	2.37	2.32	2.32	2.38	2.33	2.13	2.37	2.21	2.60	2.28	1.80	1.38
Energy Efficiency Ratio (Btu/Wh)	33.64	22.95	35.74	21.91	9.40	33.11	21.26	7.95	20.75	9.97	22.57	21.26	18.51	15.29