Pacific Gas and Electric Company

PY2005 Emerging Technologies Program

Application Assessment Report #0510

Laboratory Evaluation of the OASys™ Indirect/Direct Evaporative Cooling Unit

Issued: November 22, 2006

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EXECUTIVE SUMMARY

Testing was conducted to evaluate the performance of a mid-2006 production model OASys™ indirect/direct evaporative cooling unit designed for the residential market. The goal was to assess the performance of the OASys™ for consideration in PG&E's Mass Market program for evaporative coolers.

A test plan was developed based on ASHRAE test standards for evaporative coolers, which are primarily focused on the arrangement of the test apparatus, determining the supply airflow, and reporting on the results. A test condition matrix was established to evaluate system performance over a range of environmental conditions, which would capture the cooling design conditions for several locations in the PG&E service territory.

The advantage of an indirect/direct evaporative cooler such as the OASys™ is that it provides lower temperature air than a direct unit, and with less moisture. However, the increased flow resistance and the diversion of some of the intake airflow for indirect cooling results in significantly lower supplied airflow relative to typical direct evaporative coolers. Still, with the lower temperatures, less airflow is required to keep a space cool. The test unit also provided air that would keep a space within the ASHRAE comfort zone over a wider range of outdoor conditions than would a simple direct system.

Some key test results for this unit are summarized in *Table 1*. (For a more thorough description of the table contents and a comparison with other evaporative cooling systems, refer to *Table 5*, of which this is a subset.)

Table 1: Average Unit Performance

Fan Speed (RPM)	**High 1,122**	**Medium 992**	**Low 875**
Supply Airflow [1] (cfm)	**1,340**	**1,180**	**1,025**
Exhaust Airflow [1] (cfm)	**333**	**298**	**270**
Total Unit Power (W)	**581**	**430**	**384**
Effectiveness [2]	**107.5%**	**109.1%**	**110.3%**
CA T20 ECER (Btu/Wh)	**23.1**	**25.4**	**21.5**
Approximate Cost	**$2,850**		

[1] Measured outlet airflow referenced to the intake density, at zero inches of water supply resistance

[2] Average with 0.30" resistance on supply, 0.17" resistance on exhaust, and wet-bulb depression above 25°F (see Figure 21)

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INTRODUCTION

Background

The increasing use of compression-based air conditioning systems is the main contributor to PG&E's rising summer peak demand. This makes air conditioning systems a key focus area for research into methods to improve its efficiency or to find alternative cooling methods. One long-standing method for providing economical cooling is through the evaporation of water. Evaporative cooling technologies offer an alternative to conventional air conditioners in arid climates, and can provide some level of cooling for a fraction of the energy consumption. Because of this, PG&E promotes evaporative cooling technologies through rebates and information and education programs.

The main drawback to the acceptance of traditional "direct" evaporative coolers is that they exchange decreased temperature for increased humidity. An evaporatively cooled space can feel uncomfortable because the increased humidity can impair the body's ability to cool itself through perspiration. However, most direct systems supply relatively high airflow rates, which can make a space feel cooler than if the air was still.

A solution to the issue of increased humidity is to use evaporatively cooled air to cool another stream of dry air using an air-to-air heat exchanger. This "indirect" evaporative cooling takes advantage of the cooling done through evaporation, but without the increase in humidity in the conditioned space. A disadvantage is that because there are heat transfer inefficiencies involved with air-to-air heat exchangers, the supply air temperatures will be higher than with a direct system. They also typically have reduced supply airflow rates, and moving a second air stream creates an energy loss.

The OASys™, developed by the Davis Energy Group and manufactured by Speakman CRS, is a new combination indirect/direct evaporative cooling system being marketed to the residential and small commercial sector. In this system, the supply air is first evaporatively cooled indirectly through a heat exchanger, and then directly through a wetted pad. It is a compromise between direct and indirect-only systems, which attempts to reduce the negative impacts of either type of system alone.

Prior Research

PG&E's Technical and Land Services (TLS) 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. In 1998, tests were conducted on a prototype combined indirect/direct cooler to assist with its development. This "IDAC" system from Davis Energy Group was a direct predecessor to the current OASys™ product. This report builds upon more recent PG&E Emerging Technologies Program application assessments of different evaporative cooler technologies in 2003 and 2004, which produced three reports (References 8 through 10).

Objectives

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

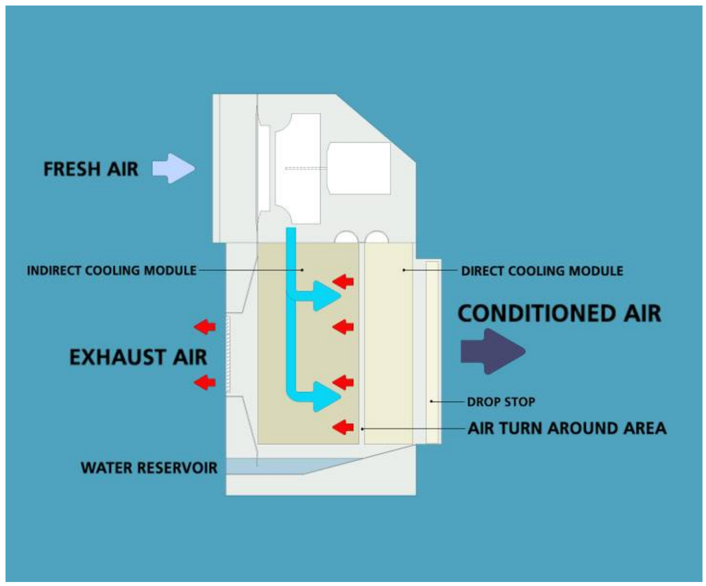
- airflow,
- evaporation (or wet-bulb) effectiveness,
- power demand,
- cooling capacity and efficiency,

as a function of the variables:

- intake air temperature and humidity,
- external resistance to flow (supply)
- exhaust airflow fraction
- fan speed, and
- line voltage.

System Description
Figure 1 shows two images of the OASys™ system (Model CRS1000), provided by Speakman CRS. As shown in the right-hand diagram, outside air is drawn in by a single centrifugal fan, which forces all of the air through the dry side of the indirect cooling module. As the air exits the first stage, it splits in two directions. The supply airflow passes forward through a direct evaporative cooler module on its way to the conditioned space. A smaller exhaust air stream reverses direction to pass through the wetted side of the indirect cooling module, and create the cooling medium for the intake air. The point at which this split occurs is the most significant change from the earlier IDAC system, which split the streams upstream of the indirect module as they exited the fan. This change enables a significant improvement in system performance, which will be discussed in more detail later. The level of water in the reservoir at the bottom of the unit is maintained by a float valve, and a single pump is used to circulate water from the common reservoir to the indirect and direct modules. The reservoir also has a flush pump, which is operated periodically to control the buildup of minerals as water is evaporated, and to drain the system when not in use. The water supply line also has a solenoid valve to shut off the supply when the system needs to be drained.

Figure 1: OASys™ System

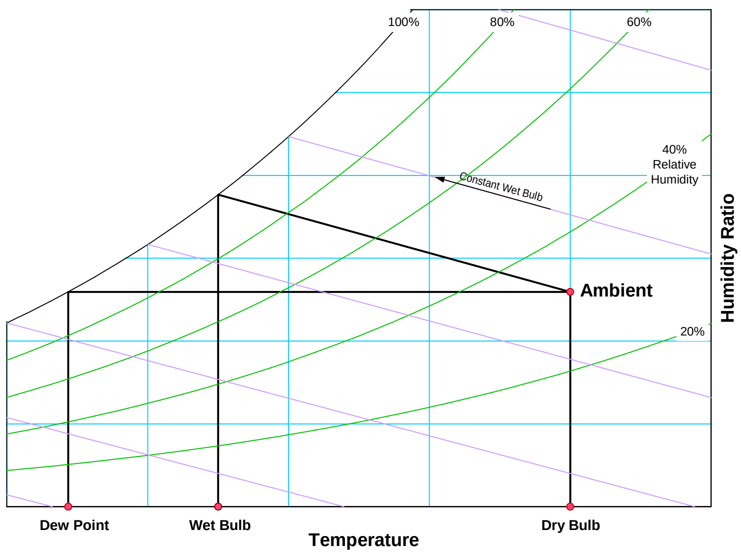


EXPERIMENTAL DESIGN AND PROCEDURE

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

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

Figure 2: Psychrometric Chart



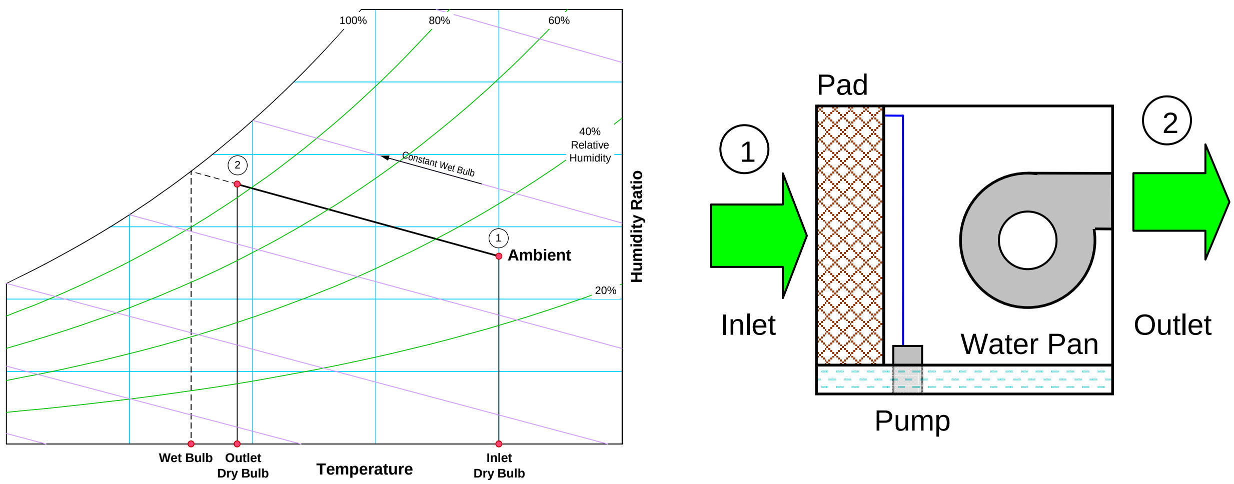
Direct Evaporative Cooling Process

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

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

where $T_{db_{in}}$ 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 effectiveness can also be described as the ratio of the actual sensible cooling done to the intake air to its wet-bulb depression. *Figure 3* shows the process for an 85%-effectiveness direct evaporative cooler on a psychrometric chart, along with the three temperatures used in the effectiveness calculation.

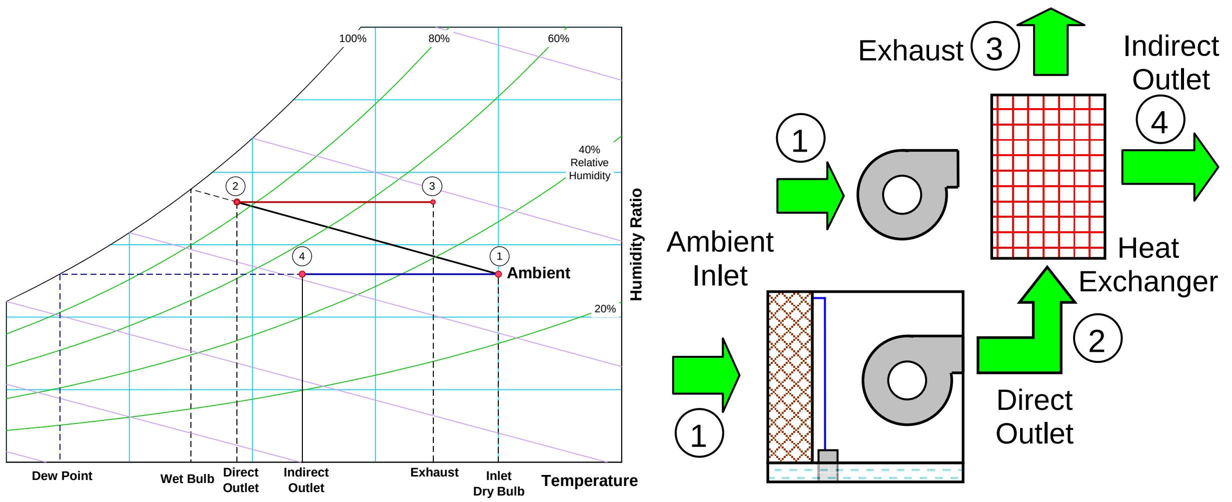
Figure 3: Simplified Direct Evaporative Cooler Process



Indirect Evaporative Cooling Process

Evaporatively cooled air can be used with an air-to-air heat exchanger to cool sensibly a second stream of air without changing its moisture content, thus creating an "indirect" evaporative cooler. Due to heat exchange inefficiencies, the temperature of the delivered air will be higher than that provided by the direct evaporative cooler. A simplified version of this process is shown in *Figure 4*. The resulting supply and exhaust temperatures will depend on the effectiveness of the heat exchanger and the relative magnitude of the two airflows. Although this example shows the intakes to both air paths from the same source (ambient air at point 1), the intake to either could be different. For example, indirectly cooling return air using outside air, or indirectly cooling outside air using exhaust air.

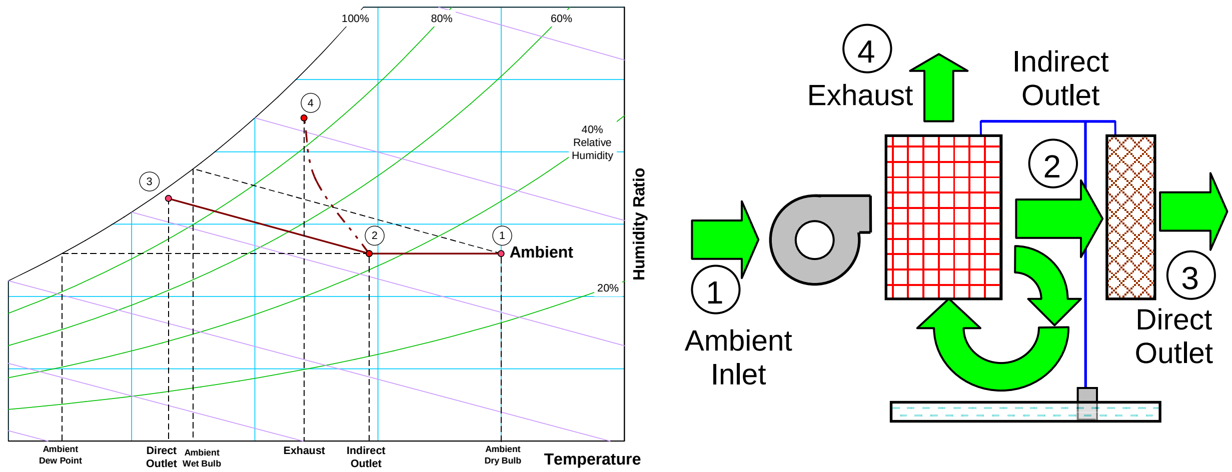
Figure 4: Simplified Indirect Evaporative Cooler Process



One advantage of the sensible cooling done through the heat exchanger is that it not only reduces the dry-bulb temperature of the air, but also its wet-bulb temperature. This means that the indirectly cooled air can be cooled further by using it as the inlet to another stage of direct or indirect evaporative cooling (Point 4 becomes Point 1 for the second stage).

The OASys™ system takes advantage of this by exhausting a portion of the sensibly cooled air to do the evaporative cooling, thus utilizing this lowered wet-bulb temperature. Rather than separate stages as in Figure 4, the OASys™ indirect cooling module uses a wetted section for the exhaust stream combining the evaporative cooler and heat exchanger into one. On the wetted side, heat from the dry side is absorbed through the walls to water flowing down the wall surface, which is then simultaneously cooled by evaporation to the exhaust air. The wetted section has a flocking applied to the wall surfaces to improve water distribution and surface contact for evaporation. *Figure 5* shows a diagram of the evaporative cooling process in the OASys™ system. In this example, the final supply temperature is actually below the ambient wet-bulb temperature, resulting in a wet-bulb effectiveness greater than 100%.

Figure 5: OASys™ Evaporative Cooler Process



Definition of Cooling Capacity

Ideally, the performance numbers obtained from testing can be used to compare performance against alternative cooling systems, including direct or indirect evaporative coolers and vapor compression air conditioners. However, the comparison between an evaporative system and a conventional vapor-compression air conditioner is not 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" in Btu/W-hr). Evaporative coolers are normally only rated in terms of airflow. The determination of a capacity for an evaporative cooler is open to some debate.

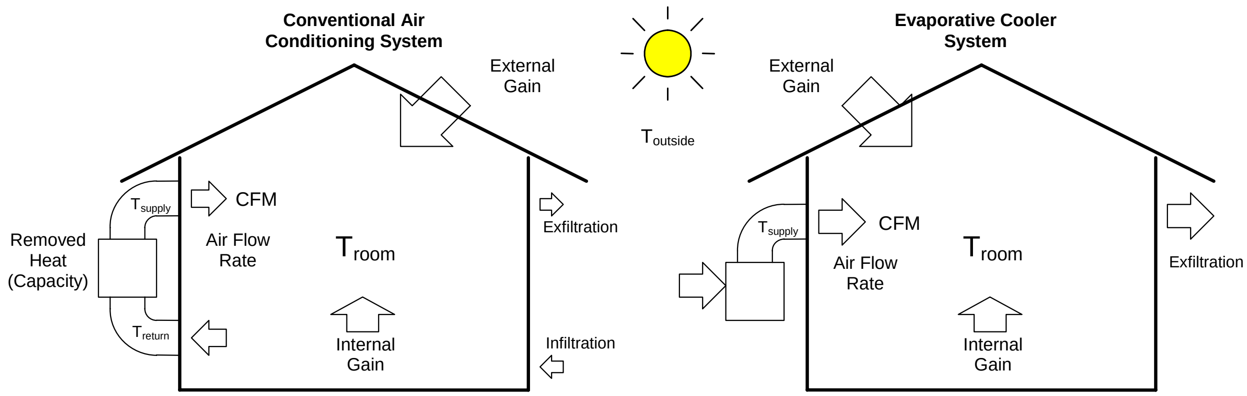
A conventional air conditioning system is designed to condition the air contained in a space, reducing the temperature (sensible heat) and moisture (latent heat) gained from various sources, while recirculating the same air repeatedly. The cooling capacity is measured at the evaporator coil as the product of the air mass flow rate across the coil and the enthalpy decrease between the return air from the conditioned space and the 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 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, the higher air velocities can make air feel cooler than air at rest. The high flow also means that evaporative coolers cannot normally be connected to a duct system sized for the velocities provided by a conventional air conditioner. Fans may need to be operated to assure adequate circulation throughout a building if the evaporative cooler supply is at only one location.

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

A graphical description of the difference between the two types of systems and the definition of cooling capacity is shown in *Figure 6*:

Figure 6: Definition of Cooling Capacity



$q_s = 1.08 \times CFM \times (T_r - T_s)$ (Sensible Cooling Load)

$q_t = 4.5 \times CFM \times (h_r - h_s)$ (Total Cooling Load)

(r = room or return s = supply)

The cooling capacity of an evaporative cooler as defined in this report is approximately:

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

where 1.08 is a units conversion factor combining standard air density and specific heat (0.075 lb/ft³ × 0.24 Btu/lb-°F × 60 min/hr), CFM is the flow rate of air through the unit in cubic feet per minute, Tdb_{supply} is the discharge dry-bulb temperature of the test unit, and Tdb_{room} is an assumed indoor space condition in °F. The selected room 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 (which only means that 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) may be determined by dividing it by the total input power in watts.

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

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

The effectiveness (ε), power (W), and airflow (CFM) are measured with an external static pressure of 0.3 inches of water; and in accordance with the ASHRAE test standards, with a minimum entering wet-bulb depression of 25°F. The ECER is then calculated at standard rating temperatures of $Tdb_{,\ in}$ = 91°F, $T_{wb,\ in}$ = 69°F, and Tdb_{room} = 80°F. This parameter only looks at the sensible cooling done by an evaporative cooler, and does not reflect the increased comfort provided by indirect systems through not adding moisture to the supply air. Thus, this parameter should only be used to compare like-systems (e.g. direct to direct).

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

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

When the intake air is the same as the outside air (as with this system), this equation reduces to:

$$OA\ Capacity\ (Btu/hr) \approx 1.08 \times CFM \times \varepsilon \times WBD \qquad \text{(Equation 5)}$$

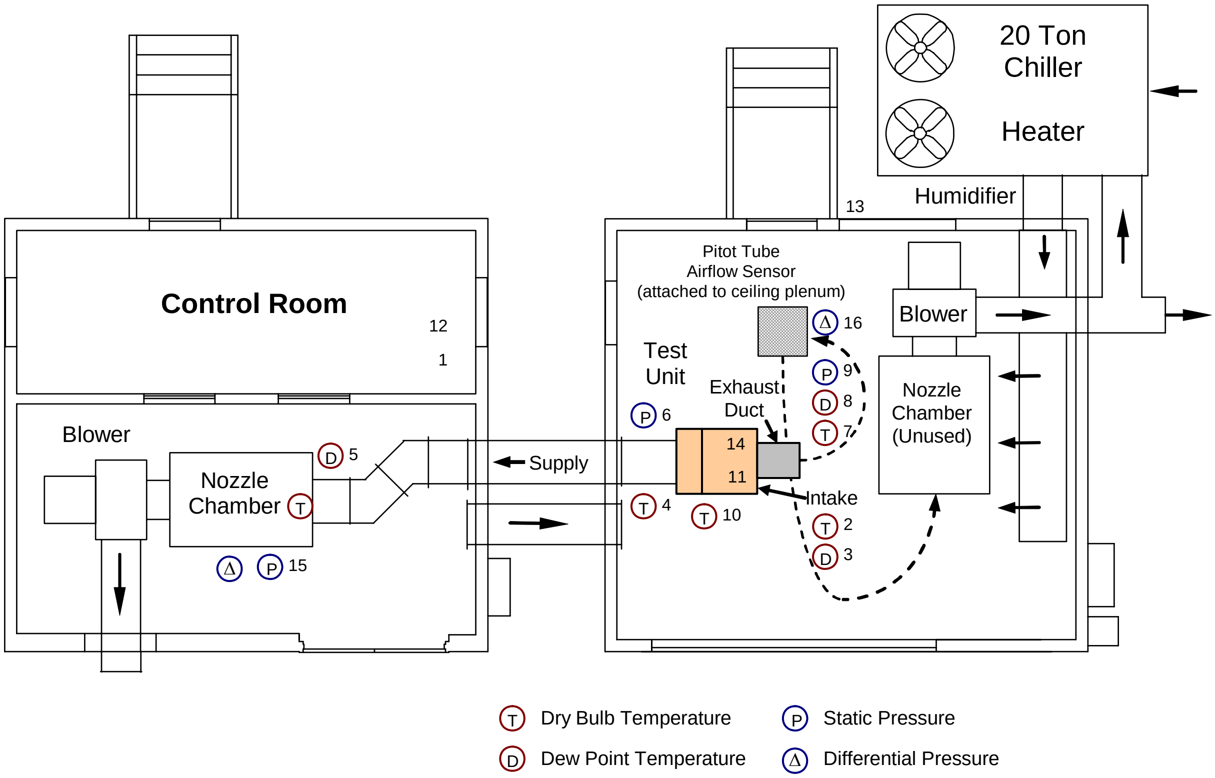
where WBD is the outside air wet-bulb depression (difference between the outside dry- and wet-bulb temperatures).

(The capacity parameters listed in Equations 2, 4 and 5 are shown as approximations due to the nominal values of density and specific heat that produce the 1.08. For the reported results calculations, the calculations of capacity are made by determining the air mass flow rate and enthalpy from the measurements; except for the ECER values, which use Equation 3 directly.)

Test Facility

Figure 7 shows a layout of the test facility configured for the OASys™ testing. The test unit was placed in a controlled environment room to maintain the conditions for testing. Both the supply air and exhaust streams of the test unit were connected to their own airflow measurement systems. The supply airflow measurement system was located in an adjacent building, and consisted of a sealed chamber with several flow nozzles designed in accordance with ASHRAE specifications (per References 2 and 3). The exhaust air flow was too low to be measured accurately by a second nozzle chamber in the same room as the test unit, so a sensor was used that consists of an averaging Pitot tube array with a flow conditioner and a ultra-low range differential pressure sensor. This device was previously calibrated against the nozzle chamber used for the supply airflow. A short section of rigid duct was attached to the exhaust outlet to provide a structure for temperature and pressure measurements. Flexible ducting was used to connect between this duct and the airflow sensor, which was attached to the plenum that feeds into the second nozzle chamber. Variable-speed blowers on the outlet of each chamber were set to maintain the desired outlet static pressures and compensate for the added resistance of the measurement systems and ductwork.

Figure 7: Test Facility and Measurement Locations
(The numbers correspond to the descriptions of the instruments in the next section)



Measurements and Instrumentation

The test set-up followed the guidelines described in the ASHRAE evaporator cooler test standards (References 4 and 5). The following is a listing of the measurements taken and the instruments used for the testing:

1. Barometric pressure, using an electronic barometer.
2. Intake air dry-bulb temperature, using four resistance temperature devices (RTDs).
3. Intake air dew-point temperature, using a chilled mirror sensor.
4. Supply air dry-bulb temperature, using six RTDs (located about 8” downstream from unit outlet)
5. Supply air dew-point temperature, using a chilled mirror sensor.
6. Supply static pressure, using a low-range static pressure transmitter.
   Four taps were made in the supply duct at the middle of each duct face. The taps were connected together with a ring of tubing and tees, with an additional tee leading to the transmitter.
7. Exhaust air dry-bulb temperature, using six RTDs inserted through the duct wall.
8. Exhaust air dew-point temperature, using a chilled mirror sensor and a sampling tube.
9. Exhaust static pressure, using a low-range static pressure transmitter.
10. Inter-stage air dry-bulb temperature, using six RTDs inserted through the test unit enclosure.
11. Water basin temperature, using two RTDs.
12. Total power, using a true-RMS power meter.

13. Make-up water flow rate, using two flow sensors with overlapping ranges (positive displacement flow meter with a pulse output, and a paddlewheel sensor with a voltage output).
14. Fan speed, using an optical tachometer.
15. Supply airflow rate, using a nozzle chamber and measurements of differential and inlet static pressure and inlet temperature.
16. Exhaust airflow rate, using an averaging Pitot tube sensor and an ultra-low range differential pressure sensor

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

Data Acquisition System

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

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

Test Conditions

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

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

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

These numbers are close to those selected for the CEC Title-20 ECER rating (91°F$_{db}$ / 69°F$_{wb}$), except for the inlet dry-bulb temperature.

The OASys™ was subjected to most of the same conditions that were developed for the previous evaporative cooler tests. These conditions were selected based on the desire to evaluate the performance of the test units over a range of environmental conditions that would adequately represent the climate during the cooling season at various locations in PG&E's service territory. 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. 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 (about 35 hours) on average. 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/CWB				Evaporation WB/CDB			
				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%

The numbers from this table were then plotted on a psychrometric chart (*Figure 8*) in order to determine a matrix of test points that would then capture the majority of those design conditions where the need for cooling is greatest. The selection of the 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 eight test conditions:

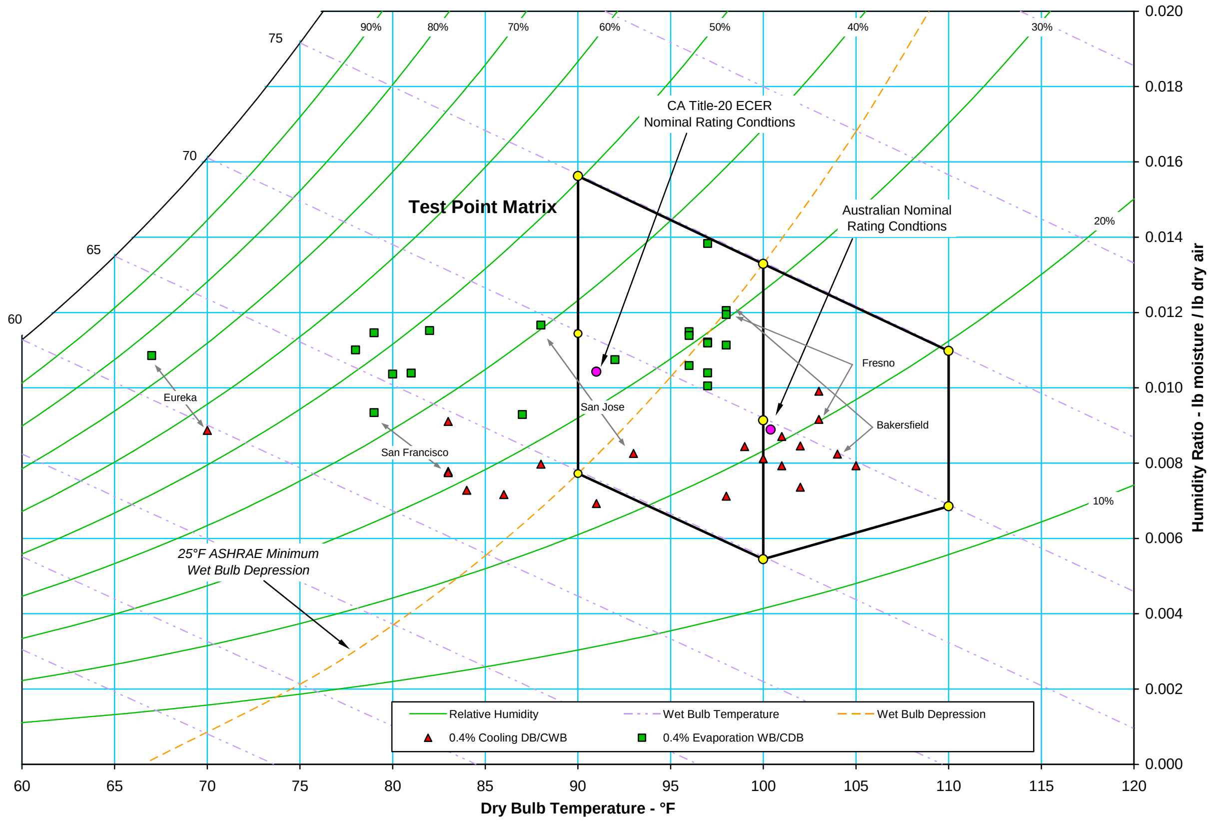
Table 3: Test Point Matrix

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

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

The test point at 90°Fdb and 70°Fwb is close enough to the CEC Title-20 rating point (91°Fdb and 69°Fwb) that these rating conditions were used instead. Both the Title-20 and Australian rating conditions are indicated in the figure. It was also decided not to test at ambient temperatures below 90°F where ventilation cooling alone might be adequate. Two of the points have less than the minimum 25°F wet-bulb depression specified in the ASHRAE test standard (highlighted in *Table 3*), including that selected for the Title-20 ECER.

Figure 8: Psychrometric Chart with Climate Deisgn Data and Test Points



In addition to the intake air conditions, the other controllable parameters were the blower speed (three settings), the supply external resistance (or backpressure), and the fraction of the total intake flow directed to the exhaust, as controlled by a damper. The number of settings for each of these parameters acts as a multiplier in the number of conditions under which the system could be run. Some compromises were necessary to limit the number of tests.

Test Procedure

The tests proceeded as follows:

1. The data acquisition system was started, and all instruments were ensured to be reading correctly.

2. The control points for room temperature and humidity were set into the computer, and the room conditioning system was started to control the room environment.
3. The test unit was turned on, and airflow station booster fan controls were set to maintain the desired static pressure at the outlet, and zero at the exhaust.
4. Other than the test done to evaluate sensitivity to line voltage, the voltage was adjusted as necessary with a Variac to maintain 115 VAC.
5. Once the desired environmental conditions were achieved and stable for at least 15 minutes, a data log file was opened on the computer and the instrument readings were recorded for another 10 to 30 minutes. Any operational problems observed were documented.
6. Adjustments for different fan speeds and different outlet static pressures were made while at the same environmental conditions, and test data collected from a stable operating period.
7. The room conditioning system was then adjusted to the next set of conditions. A flush of the water basin was normally initiated at this time, since it would normally take a long time for the new conditions to stabilize. (The basin was also drained completely at the end of a test day.)

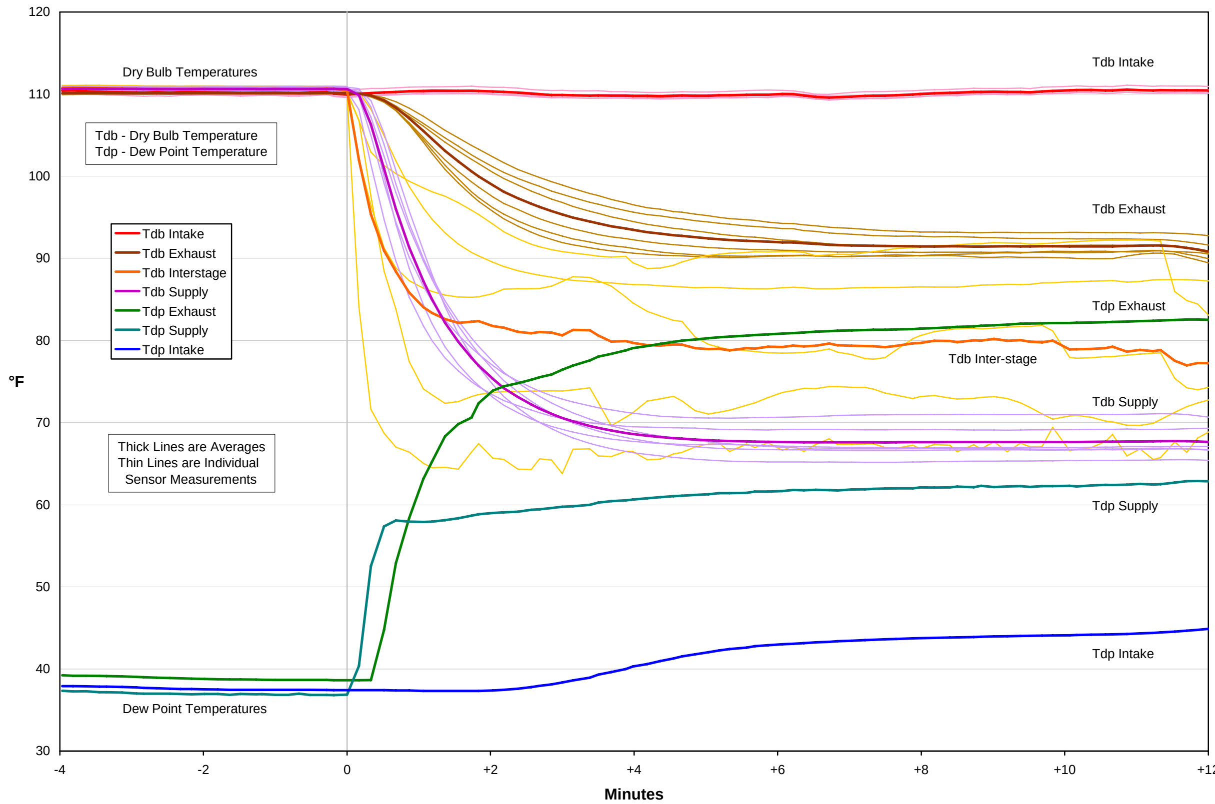
The recorded test data were averaged over the stable test period, and the averaged values were used to calculate the performance characteristics. The results from all of the tests were tabulated, and analyzed graphically by plotting the results as a function of the control parameters.

RESULTS

The testing of the sample unit went smoothly, although there was one minor system failure. During one test at high temperature, it was observed that the flow of water to the unit had stopped. This was traced to a problem with the system control board that prevented operation of the solenoid valve on the water supply line. The valve was reactivated using an external power supply, and the tests were continued.

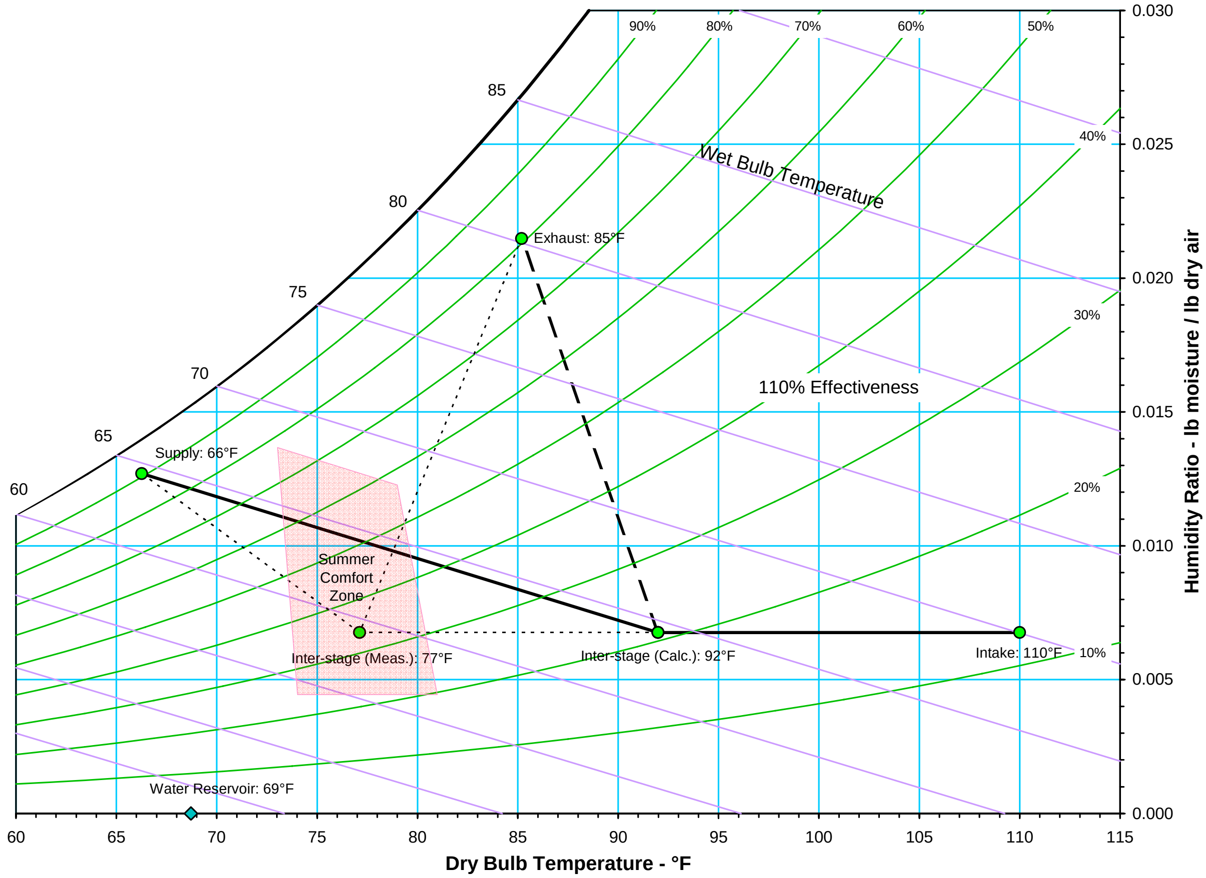
Another problem (but not with the test unit) was with the attempted measurement of the inter-stage temperature. The space between the indirect stage and the direct stage is small, and the temperature sensors that were inserted into this space were subject to wetting, resulting in lower than expected readings. In addition, the flow path and heat exchange process through the indirect stage is not uniform, leading to a wide distribution in the outlet temperatures. These effects are shown in Figure 9, which shows the changes in measured system temperatures following the startup of the water circulation pump. This is after a long period of operation with just the fan running at medium speed and allowing everything to dry out and all of the temperatures to equalize. After the pump start, the individual dry-bulb temperature measurements (other than the intake) begin to spread out from their location average, but none more so than the inter-stage measurement. Sensor wetting is evidenced by some of the inter-stage temperatures reading about the same as the supply air downstream of the direct stage. Also worth noting from this chart is that the supply temperature becomes stable after only about six minutes, showing a quick response time to system changes.

Figure 9: Temperature Measurements Following a Pump Start



The problem with the inter-stage temperature measurement may also be seen in a psychrometric chart of the actual test data, as shown in Figure 10 (which shows the steady-state performance test following the data shown in Figure 9). For this test, the measured average inter-stage temperature is about 15°F less than the value calculated by finding the intersection between the intake humidity ratio and the supply wet-bulb temperature. Also shown in this chart is the summer comfort zone as defined in the ASHRAE Fundamentals Handbook (Reference 1), which defines the conditions that should be maintained in a space to keep most occupants in a working environment comfortable. The air supplied to the space should be at a condition to the left and slightly below the summer comfort zone to allow for sensible and latent heat gains within the space. A direct evaporative cooler under these outside conditions would provide supply air that is too humid to maintain adequate comfort.

Figure 10: Process Description at one Test Condition



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

Fixed Inlet Conditions, Variable Outlet Resistance (and Voltage)
For the first set of tests, the intake conditions were maintained constant while the supply outlet resistance and fan speed were varied. The intake condition selected was that used for the Title-20 ECER: 91°Fdb / 69°Fwb, even though the wet-bulb depression was only 22°F rather than the ASHRAE required 25°F. One other variable was the position of the damper on the exhaust outlet, which is used to control the amount of airflow through the wet side of the indirect stage. To observe the full range of operation, tests were conducted with this damper in its fully open and fully closed positions. In both cases, a zero static pressure relative to the intake was maintained in the duct downstream of the damper. In the fully closed position, the damper does not completely shut off the exhaust airflow, but only creates a maximum restriction. Several charts in the Appendix are derived from this first set of data.

Figure 12 shows the results of a special test to see the effect of varying line voltage, since demand for cooling typically coincides with periods of high electrical demand when there may also be voltage sags. These tests were conducted during the period when the exhaust damper was in its closed position, and were conducted with a supply outlet resistance of 0.3 inches of water. The results are shown relative to the points taken at a line voltage of 115V, and ranged from 95 to 120V. The results show that there is a decrease in speed, power, and airflow as the voltage is decreased, and that the effectiveness rises slightly (most likely in response to the lower airflow). Also shown is a calculation of the Title-20 ECER, which combines the effects of airflow, effectiveness and power. The resulting curve shows that under these conditions, the ECER reaches a maximum at about 110V.

In *Figure 13*, the total intake airflow is graphed as a function of the supply air outlet resistance in inches of water. The outlet resistance was varied by changing the speed of the booster fan on the nozzle chamber to maintain the required value. This chart demonstrates the simple relationship that as the overall system resistance increases, either by increasing the resistance at the supply outlet or by closing the exhaust damper, the total intake airflow decreases. The decrease in airflow is nearly linear with the increasing supply outlet resistance over this range, and the trends are relatively parallel for the three fan speeds. The airflow at medium fan speed is about 12% less than at high speed, and 23% less at low speed.

This chart is actually graphed opposite of how the results are supposed to be presented according to the ASHRAE test standards, which prescribe graphing the performance measures (power, effectiveness, external resistance) as a function of airflow rate. However, this means the airflow supplied to the space, whereas this first chart shows the total airflow into the unit, including the exhaust. This graphing method was preferred for this parameter since the outlet resistance is the controlled variable. The remaining charts in this section are all drawn in accordance with the ASHRAE standards as functions of the supply airflow referenced to the intake air density. One change from the standards is that while they specify that all of the main parameters be plotted on the same chart for a particular fan speed, it was decided to put each parameter on its own chart for clarity and to show the effect of speed and damper position.

Figure 14 is almost a rotation of the previous chart, where in this case the supply outlet resistance is graphed as a function of the supply airflow. This chart shows that the supply airflow has a much greater sensitivity to the damper position and supply resistance than the total airflow. This is because as the exhaust damper is opened or as the supply resistance is increased, more of the intake airflow is diverted through the exhaust path, while the total system resistance is only increased slightly. More air through the exhaust will likely improve the thermal performance of the indirect stage at the expense of less air to cool the space.

Figure 15 examines the diversion of air to the exhaust stream in more detail. In this chart, the parameter being shown is the fraction of the total intake airflow that is directed to the exhaust in relation to the supply airflow, again as a function of the fan speed and damper position. As reported by the system designers, the recommended optimum for the exhaust air is between 20 and 30% of the total. As shown in the chart, this can be achieved with many combinations of fan speed and damper position, but it will depend on the actual supply outlet pressure. To achieve this fraction in the field would require some tuning of the system to its installation characteristics, and may require a different damper setting for each fan speed.

Figure 16 is a graph of total power consumption, which shows a sensitivity to fan speed only, with very little variation due to supply airflow and damper position. It is worth comparing this chart with that of the intake airflow. In particular, the increase in the total airflow is about the same going from low to medium speed as going from medium to high speed. However, the power consumption increase between medium and high speed is significantly greater than between low and medium speed. This is related to the fan affinity laws, which say that airflow increases linearly with speed, but that power increases with the cube of the speed.

Figure 17 shows the trends of different measures of wet-bulb effectiveness. The graph includes the overall system effectiveness, plus the calculated effectiveness for the indirect and direct stages separately. The values for the stages are based on the calculated average inter-stage condition, found at the intersection of the supply wet-bulb temperature and the intake dew point temperature. Thus, the indirect stage effectiveness is the dry-bulb temperature difference between the intake and the inter-stage divided by the intake wet-bulb depression, and the direct stage effectiveness is the dry-bulb temperature difference between the inter-stage and the supply air divided by the inter-stage air wet-bulb depression.

Since these two measures use different denominators, they are not additive to the overall system effectiveness (which like the indirect stage uses the intake wet-bulb depression).

Reducing the supply airflow by increasing the external resistance has two effects. First, more air is diverted through the exhaust, improving the effectiveness of the indirect section. Second, when the velocity decreases through the direct stage, there is more contact time with the water, resulting in another increase in effectiveness. Thus, it is not surprising that the effectiveness is highest (and thus the outlet temperature is lowest) at low airflow rates. One deceptive feature of graphing the effectiveness as a function of the supply airflow is that it appears that the three fan speeds have overlapping ranges of airflows where each speed will have the highest effectiveness. The reason for this is that for a particular supply airflow, the higher the fan speed, the larger the amount of air that must be diverted to the exhaust.

Figure 18 is a graph of the California Title-20 ECER, which combines the effects of airflow, power, and effectiveness. This is not a parameter that is specified by the ASHRAE test standards, but has significant interest in regards to the relative efficiency of the unit. This value is calculated in accordance with Equation 3 with fixed temperature values, even though all of these tests were actually conducted at the prescribed temperatures. The key data points of interest are those that occur when the outlet resistance is at 0.3 inches of water, and these points are emphasized and a curve is drawn between those at different fan settings. The normal range of the ECER should be somewhere between these two curves depending on the damper position. The shape of the curves implies that the ECER reaches its highest value at the medium fan speed. It is lower at low speed due to low airflow, and at high speed because of high power consumption.

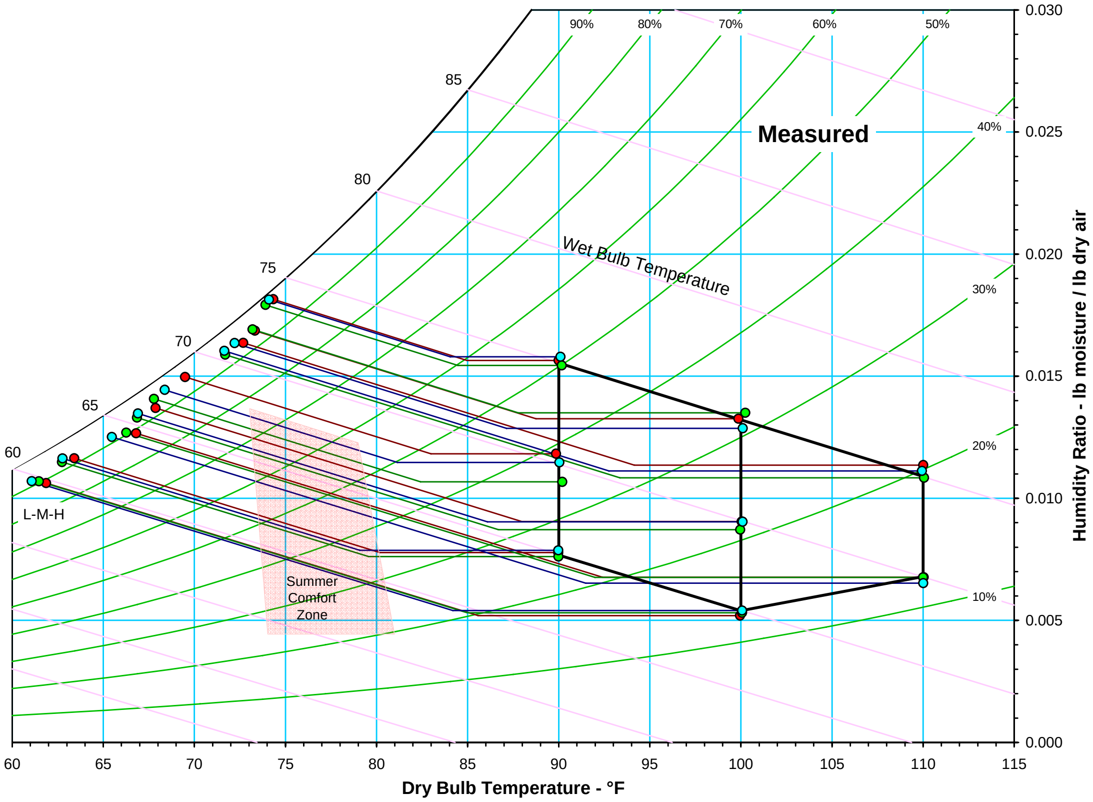
The final chart of this group, *Figure 19*, shows the measured water consumption rate. In general, this chart shows increasing water consumption with increasing airflow, both in terms of supply airflow and exhaust airflow (as the result of opening the exhaust damper). This does not show the effect of different intake conditions, which can have a more significant impact.

Fixed Outlet Resistance, Variable Intake Conditions

For the next set of tests, the supply outlet resistance was kept constant at 0.3 inches of water (to provide data for the ECER calculation) while the intake conditions were varied in accordance with the selected test matrix (Table 3). For the exhaust flow, it was decided to collect data at some intermediate point between the damper open and the damper closed positions. Rather than resetting the damper position, it was left in the fully open position and the airflow chamber booster fan was shut off, thus creating a resistance due to the outlet ducting that could actually be measured. Rather than a fixed value, the exhaust outlet pressure could vary with the airflow, just as it would with the damper in one fixed position. As it turned out, this did produce exhaust airflow fractions that were about half way between the open and closed damper settings in the previous tests. Measured exhaust pressures ranged from 0.15 to 0.19 inches of water, or about half of that at the supply outlet.

Figure 11 shows the resulting supply air conditions from all of the tests in this group on a psychrometric chart. The first chart shows the actual measured test data, which demonstrates that the desired test conditions could not always be maintained precisely. To better observe the effects of the variables, the results were adjusted to the desired common settings using the measured effectiveness and degree of humidification. The results from this second graph show a minor increase in effectiveness (or lowering of the supply air temperature) as the fan speed is decreased. Of greater interest is that as the outside air temperature increases for a fixed wet-bulb temperature, the supply air temperature decreases and capacity improves. This is the opposite of what happens with a conventional air conditioner, which looses capacity and efficiency with increasing ambient temperature. The summer comfort zone conditions would not be met with this system if it were already humid outside, but only in terms of the indoor humidity. The supply air temperatures were all at or below the comfort zone conditions.

Figure 11: Performance at All Variable Intake Conditions



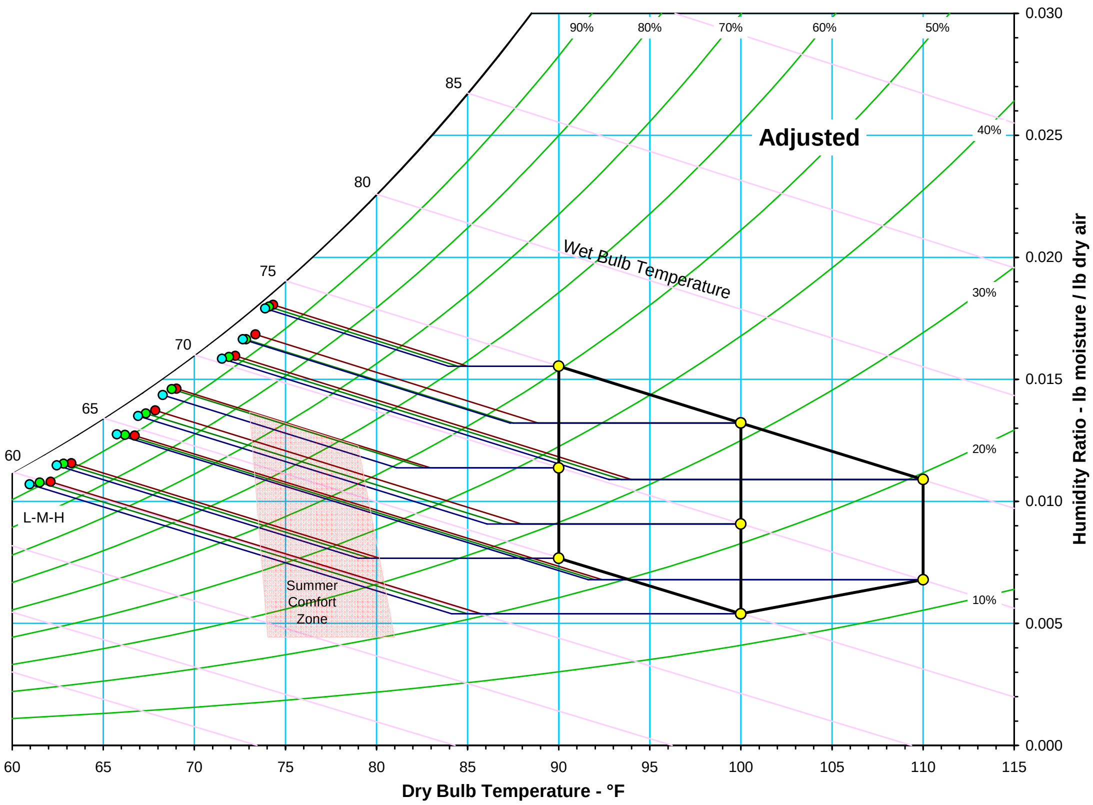


Table 4 contains three different measures from these tests as a function of the inlet dry and wet-bulb temperatures: the resulting supply and exhaust temperatures, and wet-bulb effectiveness. The data shown are for the medium fan speed, representing the middle of the three points shown for the supply conditions in Figure 11. The points that do not have the 25°F wet-bulb depression required by ASHRAE are shaded. The results indicate that the unit shows an improvement in effectiveness at higher ambient temperatures. The key point being that its performance improves as temperatures get higher, as opposed to the performance of a conventional air conditioner, which gets worse. The effectiveness shows very little sensitivity to the inlet air conditions with a wet-bulb depression above 25°F, but the table does show why a minimum wet-bulb depression is specified in the ASHRAE test standards.

Table 4: Performance Measures at Medium Fan Speed and 0.3" of Water Supply Pressure

Supply Temperatures (°F)

Intake Tdb	Intake Wet-bulb Temperature (°F)		
(°F)	65	70	75
90	63	68	74
100	61	67	73
110		66	72

Exhaust Temperatures (°F)

Intake Tdb	Intake Wet-bulb Temperature (°F)		
(°F)	65	70	75
90	75	78	81
100	80	81	84
110		85	87

Wet-Bulb Effectiveness

Intake Tdb	Intake Wet-bulb Temperature (°F)		
(°F)	65	70	75
90	109%	106%	106%
100	110%	109%	109%
110		110%	109%

Most of the other charts from this set of the data are drawn as a function of the intake wet-bulb depression to demonstrate the effect of the changing intake conditions. *Figure 20* (in the Appendix) is a complement to Figure 19 in that they both show the water consumption of this unit, but as a function of different variables. In this chart, the water consumption is graphed as a function of the intake wet-bulb depression for the three fan speeds, and shows a nearly linear increase in water consumption as the air becomes dryer. This chart shows two different measures of the water consumption. The first is the measurement from the flow meter on the supply line, and the second is the calculated water evaporation rate as determined from the measures of airflow and humidity. Since the system did not have a constant bleed for maintaining water quality, and the flush system was not allowed to operate during a test, these measures are essentially equivalent. The slopes of the lines fitted to the results are indicated for comparison with the results reported in earlier reports.

Figure 21 shows the wet-bulb effectiveness as a function of the intake wet-bulb depression, grouped by fan speed and intake dry-bulb temperature. This chart shows that there is an apparent increase in effectiveness as the wet-bulb depression rises, but a small decrease for the same depression at a higher dry-bulb temperature. This may be due to changes in airflow due to its decreased intake density, increased evaporation rates, or thermal expansion of air passages. Of particular concern from this chart is that the ASHRAE test standards only specify a minimum intake wet-bulb depression of 25°F, but the effectiveness does not remain constant above this level. However, there does appear to be a sharper downturn in effectiveness below this value. The averages for all of the effectiveness measures at or above the ASHRAE minimum for each fan speed are also indicated in this chart, and are what may be used to calculate the Title-20 ECER.

Figure 22 and *Figure 23* are plots of the two measures of system sensible capacity (Equations 2 and 4) and their corresponding energy efficiency ratios (EERs), respectively. The capacity is listed in tons (12,000 Btu/hr) and the EER (which is capacity divided by the total unit power) is listed in Btu/Wh. Both measures of capacity show an increase with rising intake wet-bulb depression. As shown in Equation 5, the outside air capacity is a direct function of the wet-bulb depression, with some variation due to minor changes in airflow and effectiveness. Curve fits of the outside air capacity and EER over the full range of wet-bulb depression are included for each fan speed for comparison with previous results. It is interesting to see that while the highest fan speed provided the highest capacity by either measure, the medium fan speed produced the highest efficiency because of the large jump in power at high speed.

Table 5 provides a summary of some key performance measures for comparison with the results from previous evaporative cooler tests. The results from two of the test units are included in the table, and these are an advanced direct evaporative cooler with 8-inch rigid media, and the same unit but with an add-on indirect evaporative cooling section. This second unit provides a direct comparison to another two-stage system, but which has a different arrangement of components: the secondary/exhaust air is drawn through by an additional fan, and its source is from the intake rather than the outlet of the indirect stage. Since the performance of the comparison systems in the second section of the table are reported at an intake condition of 100°F_{db} and 70°F_{wb}, the numbers for this unit are also. However, only a couple tests were run at this intake condition, and these were not at the zero resistance used for comparison. This required some extrapolation of the performance measures the trends from both sets of tests, and the results are listed as estimates. (As discussed previously, the performance improves with higher dry-bulb temperatures at a constant wet-bulb, so these numbers are better than the measured performance at 91°F_{db} and 69°F_{wb}.) Also included in the table is a comparison with a 3-ton 12-SEER split-system air conditioner for comparison with some selected parameters.

Most of the table shows measures with no external resistance on the supply. As was shown in Figure 17, the effectiveness of the OASys™ unit improves with increasing resistance as more air is diverted to the exhaust; so with no resistance, its effectiveness is at a minimum. Even so, it has the highest effectiveness of all of the systems that have been tested in this facility, and is the only one to achieve an effectiveness over 100%.

Table 5: Averaged Results for Airflow and Power

0" w.g. outlet resistance

Test Unit	Dir	In/Dir	**OASys**		SEER 12
Exhaust Damper Position			**Open**	**Closed**	3-Ton A/C
High Speed					ARI "A"
Supply Airflow [1] (cfm)	3,320	2,440	**1,330**	**1,457**	1,200
Exhaust Airflow [1] (cfm)		610	**360**	**189**	
Total Unit Power (W)	737	939	**584**	**581**	3,490
Effectiveness	72.7%	88.8%	**104.2%**	**98.7%**	N/A
Medium Speed					
Supply Airflow [1] (cfm)			**1,190**	**1,293**	
Exhaust Airflow [1] (cfm)		610	**310**	**175**	
Total Unit Power (W)			**436**	**436**	
Effectiveness			**104.1%**	**100.5%**	
Low Speed					
Supply Airflow [1] (cfm)	2,120	1,480	**1,040**	**1,111**	
Exhaust Airflow [1] (cfm)		610	**260**	**165**	
Total Unit Power (W)	360	644	**385**	**386**	
Effectiveness	77.6%	95.3%	**104.1%**	**100.8%**	

Sensible capacity measures with ~100°F_{db}/70°F_{wb} intake air
and 0" w.g. outlet resistance

Test Unit	Dir	In/Dir	OASys		SEER 12
High Speed			**(Est.)**		ARI "A"
Room Capacity (tons) [2]	0.62	1.56	**1.28**	**1.19**	3.0
Room EER (Btu/Wh) [2]	10.1	19.9	**26.4**	**24.9**	10.3
Outside Air Capacity (tons) [3]	6.21	5.72	**3.52**	**3.70**	
Outside Air EER (Btu/Wh) [3]	101.1	73.1	**72.5**	**77.3**	
Water Consumption (GPH) [4]	13	14	**5**	**5**	-1
Medium Speed					
Room Capacity (tons) [2]			**1.14**	**1.13**	
Room EER (Btu/Wh) [2]			**31.6**	**31.4**	
Outside Air Capacity (tons) [3]			**3.16**	**3.35**	
Outside Air EER (Btu/Wh) [3]			**88.0**	**93.1**	
Low Speed					
Room Capacity (tons) [2]	0.70	1.14	**1.02**	**0.89**	
Room EER (Btu/Wh)[2]	23.2	21.3	**32.4**	**28.1**	
Intake Air Capacity (tons) [3]	4.25	3.69	**2.84**	**2.82**	
Intake Air EER (Btu/Wh) [3]	141.7	68.8	**90.1**	**88.7**	

0.3" w.g. outlet resistance

CA Title-20 ECER (Btu/Wh) High	23.2	19.0	**23.1**	10.3
Medium			**25.4**	
Low			**21.5**	

[1] Supplied airflow referenced to the intake density.
[2] Room Capacity ≈ $1.08 \times CFM \times (80°F - T_{supply})/12{,}000$
[3] Outside Air Capacity ≈ $1.08 \times CFM \times (T_{intake} - T_{supply})/12{,}000$
[4] The comparison units had continuous bleed systems
Dir: Advanced direct evaporative cooler with 8"-thick cellulose pad
In/Dir: "Dir" with add-on indirect evaporative precooler

CONCLUSIONS

This study investigated the performance of a commercially available example of the OASys™ two-stage evaporative cooler. The primary advantage of the OASys™ in relation to direct evaporative coolers is that it can achieve lower supply temperatures while simultaneously adding less moisture to the space. The disadvantages are that the system is more complex and more expensive than direct units are, and the supply airflow is low. However, this unit still uses considerably less power than a conventional air conditioner or most other evaporative coolers, and will likely keep a space more comfortable than other evaporative coolers for more of the cooling season. As this evaluation was only a series of short-term tests, they give no indication of its long-term reliability or maintenance requirements in actual use.

Some of the key findings are summarized below.

1. With three fan speeds and various combinations of exhaust damper position, supply outlet resistance, and intake dry- and wet-bulb temperatures, there were more tests done on this unit than for any of the evaporative cooler systems evaluated in previous application assessments. However, even with the larger number, it was not a goal to determine which combination of settings provided the best operating performance, and the test results did not point towards one.
2. The wet-bulb effectiveness varied from 98% to 112% over the range of test conditions. It was only under 100% for a very few tests when there was no resistance on the supply outlet and the fan was on

high speed. Increasing the outlet resistance increases the effectiveness significantly by diverting more air through the secondary side of the indirect module.

3. Increasing effectiveness for a particular set of intake conditions means:
   - Lower supply air temperatures, which means that:
   - Less airflow will be needed to cool a space, which in turn requires:
   - Less fan power.
4. Supply air temperatures ranged between 61 and 74°F over the range of test conditions. This means that it always provided some cooling effect compared to a room reference temperature of 80°F. This is particularly impressive since the intake dry-bulb temperature was as high as 110°F, which showed a temperature reduction to the outside air of 37 to 44°F
5. The supply airflow at high speed and zero resistance was about 1,340 cfm with the exhaust damper open and 1,450 cfm with it closed. This is about 50-60% less than the airflow provided by some direct evaporative coolers, but is compensated for somewhat by having a lower temperature. This low airflow does have another advantage in that the system could be connected to existing ductwork in a residence if it is sized to handle about a 3½-ton or smaller central air conditioner, and it also creates lower noise levels.
6. The power consumption of this system averaged:
   - 580W at high speed,
   - 430W at medium speed, and
   - 380W at low speed.

   At high speed, this is about 20% less than a comparison direct evaporative cooler, and 83% less than a 3-ton air conditioner.
7. The new California Title 20 evaporative cooler efficiency ratio (ECER) does not reflect increased comfort from the reduced moisture addition to the supply air, and thus treats this and other systems with indirect components unfairly when compared to direct systems. However, with its high effectiveness and efficient fan operation, this unit provided an ECER comparable to that of a direct system (23.1 Btu/Wh at high speed).
8. The cost of this unit was considerably more than other evaporative cooling systems. However, this system is still at an early stage of production, and system costs should go down as production rates are increased. While comparable in initial cost to a conventional air conditioner, the energy savings reduce its life-cycle cost.
9. This system has application wherever evaporative coolers are currently used or that require large amounts of fresh, outside air (e.g. residences, commercial kitchens, gymnasiums).

Recommendations for Follow-on Activities

To evaluate the OASys™ and other evaporative cooling technologies to air conditioning technologies a combination of laboratory, computer modeling and field monitoring data must be collected and analyzed. These laboratory results may be used to develop a computer model to evaluate how much of a cooling season this system would be able to keep a house comfortable in different climates, and for what cost in energy and water use. This model should be calibrated using field data to estimate more accurately the annual cost to operate.

As this system had three different speed settings, it demonstrated an interesting trend in the new Title-20 ECER calculation as a function of fan speed and the resulting airflow. The ECER is a function of wet-bulb effectiveness, airflow rate, and power consumption (Equation 3), and as the test results show, the effectiveness and power consumption are both functions of airflow. As ECER is related directly to the airflow rate, its value is zero at zero airflow. In a true variable speed fan situation, as the airflow is increased from zero, the ECER will also increase up to some peak, at which point the near-cubic function of power versus airflow takes over and causes the ECER to fall off at higher airflow rates. This is why the ECER number for this system was highest at the medium fan speed, which turned out to be the fan speed that came closest to the peak for this system. As the ECER rating number becomes more prevalent, manufacturers may begin to design the fan speed of their systems around its maximum ECER to make them stand out, which may not be the best solution for the consumer. ECER should always be reported with supply airflow.

Perhaps a refinement of the ECER is required for variable or multiple speed systems is needed to account for the times that the system will operate at lower fan speeds than the maximum, which will likely be most of the time. Additional system modeling or testing could be done with the goal of determining a seasonal ECER (integrated total cooling provided by the total power consumed), and a set of measurements that could be done to predict this result to create a better system rating. While a better indicator, this still does not include the reduced humidification done by systems having indirect components, like the OASys™.

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

REFERENCES

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. ANSI/ASHRAE Standard 37-1988, “Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment”, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, NE, Atlanta, GA 30329, 1988.
3. ANSI/ASHRAE 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.
4. ANSI/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.
5. ANSI/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.
6. ARI Standard 210/240, “Standard for Unitary Air-conditioning and Air-source Heat Pump Equipment”, Air-conditioning and Refrigeration Institute, 4301 N. Fairfax Drive, Arlington, VA 22203, 2003.
7. Australian Standard AS 2913-2000, “Evaporative Air Conditioning Equipment”, 2000.
8. Davis, R. Pacific Gas and Electric Company PY2003 Emerging Technology Application Assessment Report #0307, “Evaluation of Advanced Evaporative Cooler Technologies”, PG&E/TES Report 491-04.07, February 2004.
9. Davis, R. Pacific Gas and Electric Company PY2004 Emerging Technology Application Assessment Report #0401, “Evaluation of a Thick Media Evaporative Cooler”, PG&E/TES Report 491-04.31, November 2004.
10. Davis, R. Pacific Gas and Electric Company PY2005 Emerging Technology Application Assessment Report #0402, “Laboratory Evaluation of the Coolerado Cooler™ Indirect Evaporative Cooling Unit”, PG&E/TES Report 491-05.6, March 2006.

APPENDIX

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Figure 12: Normalized Performance Sensitivity to Line Voltage

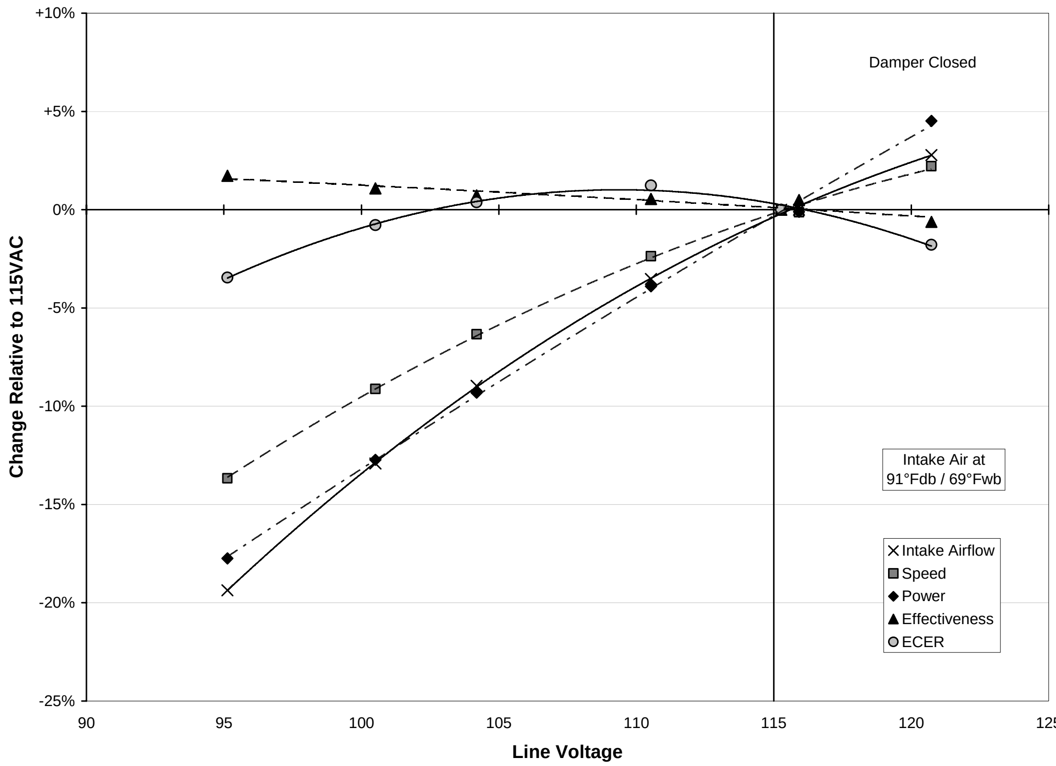

Figure 13: Intake Airflow Sensitivity to Supply Backpressure

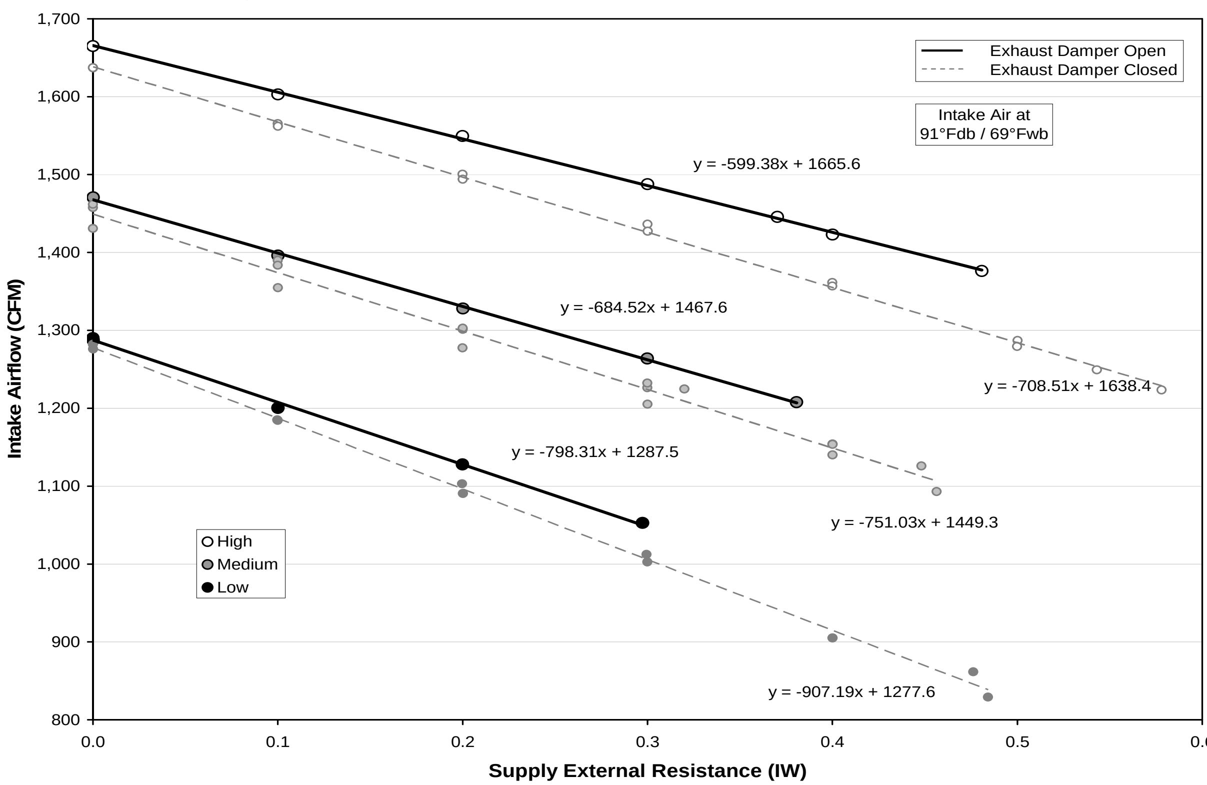

Figure 14: Supply Backpressure versus Supply Airflow

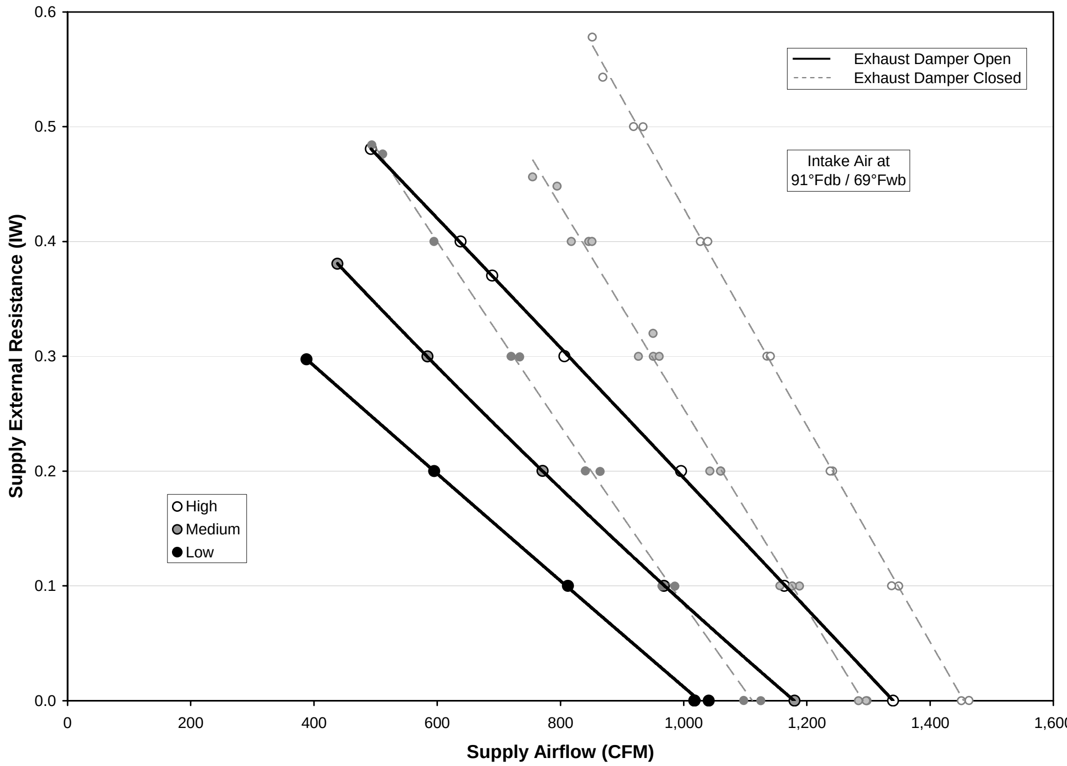

Figure 15: Exhaust Airflow Fraction versus Supply Airflow

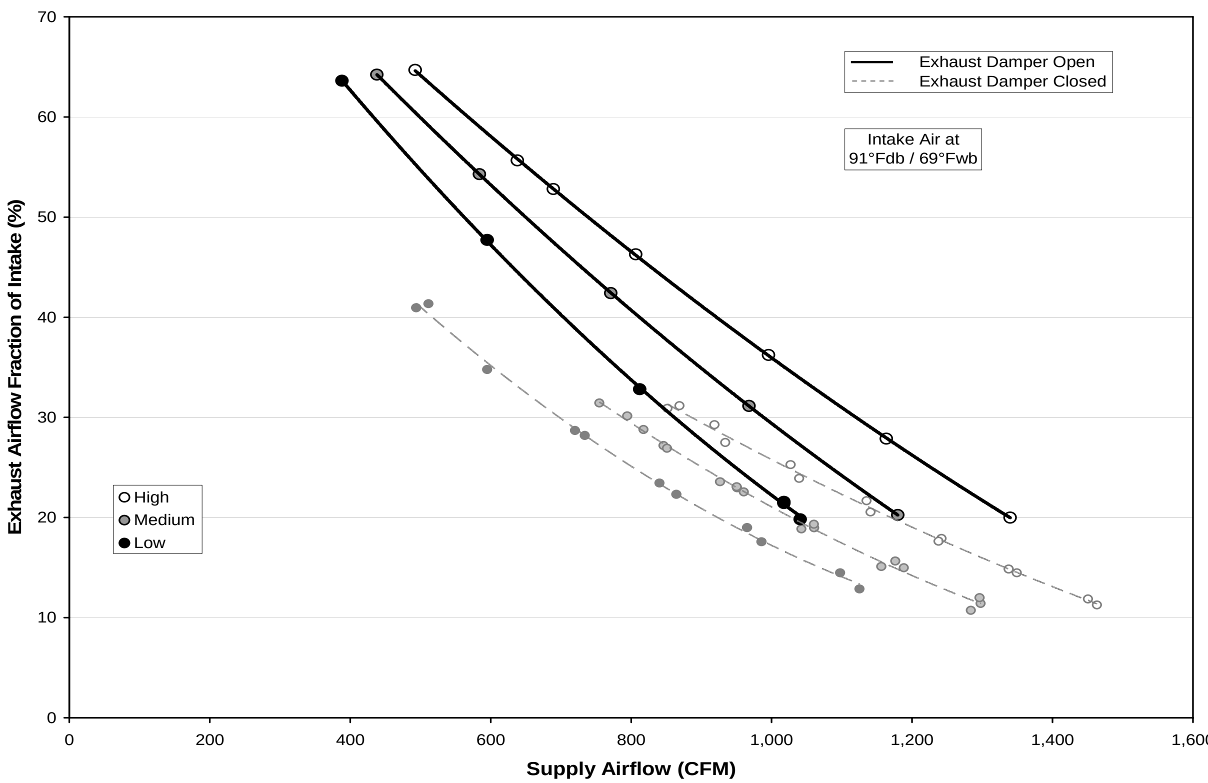

Figure 16: Power versus Supply Airflow

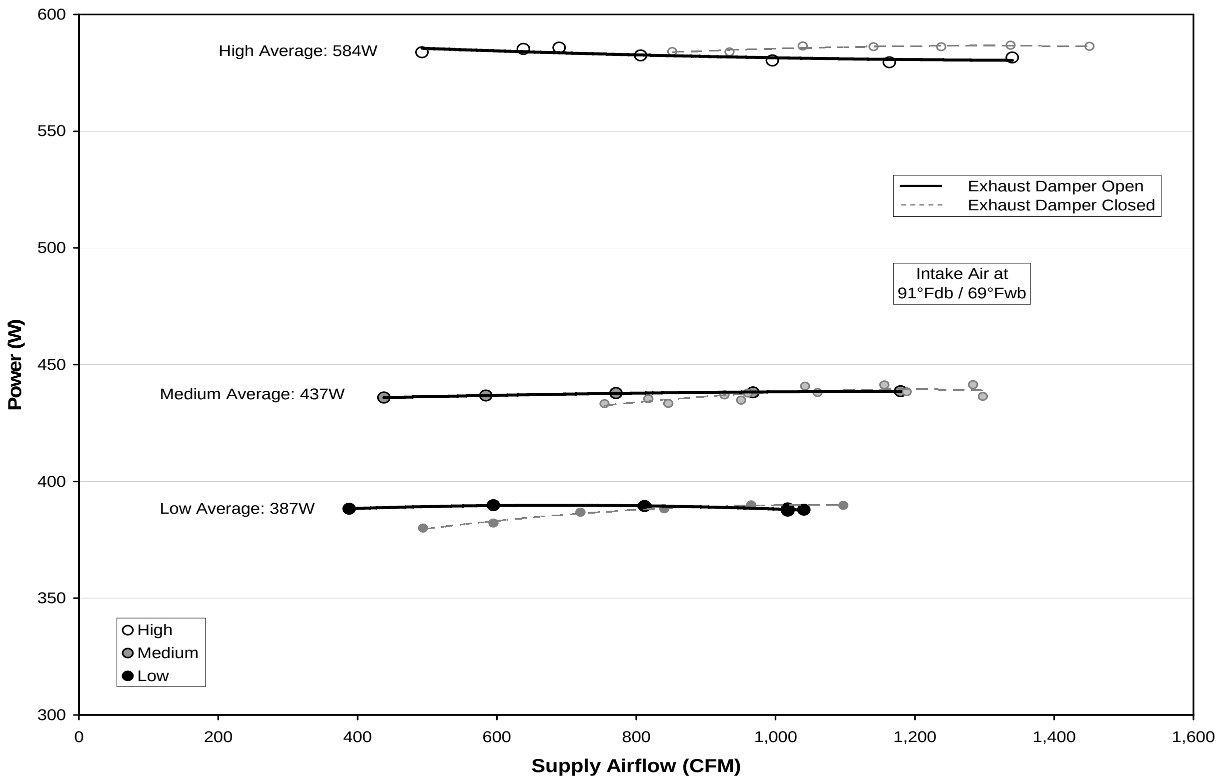

Figure 17: Effectiveness versus Supply Airflow

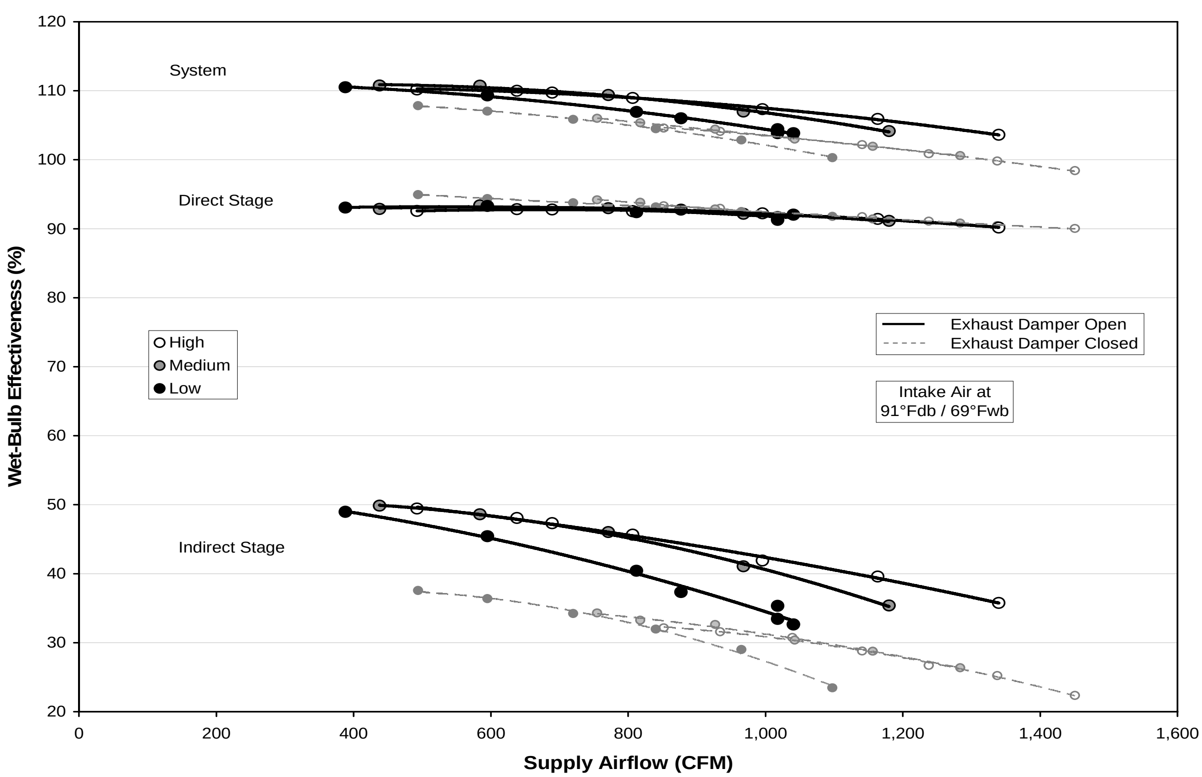

Figure 18: CA T20 ECER versus Supply Airflow

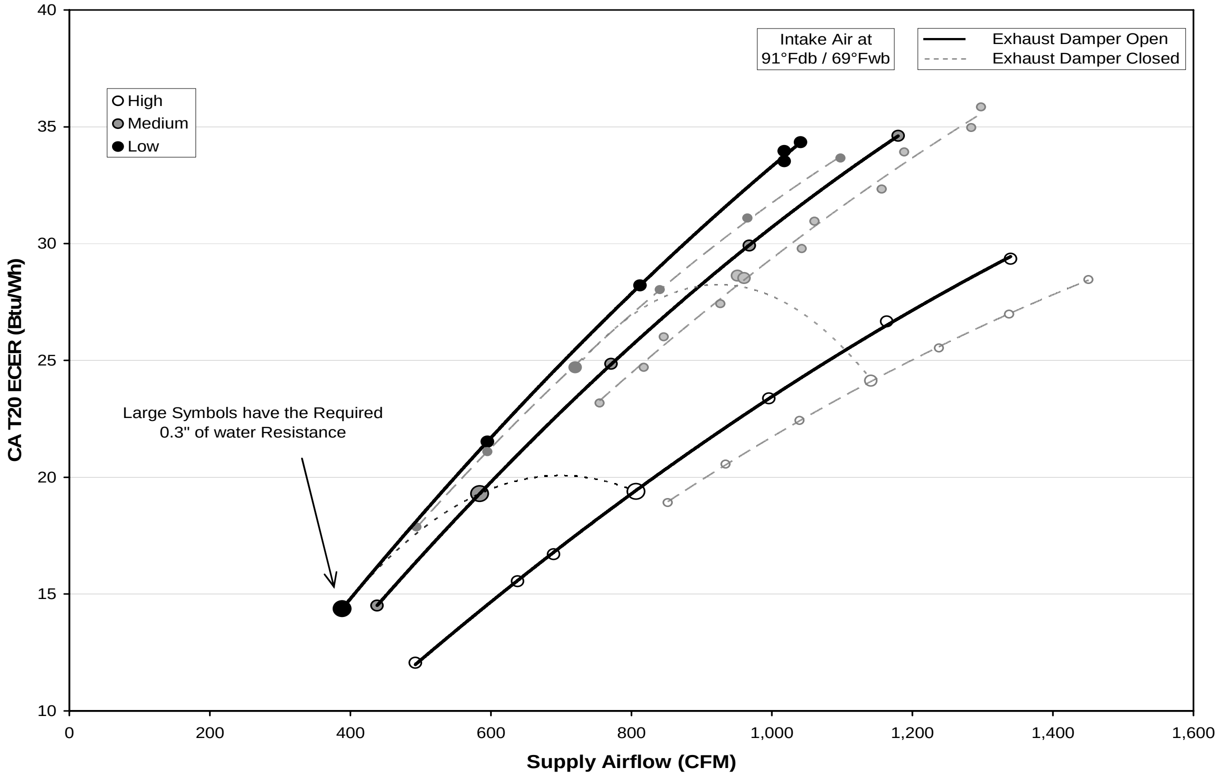

Figure 19: Water Consumption Rate versus Supply Airflow

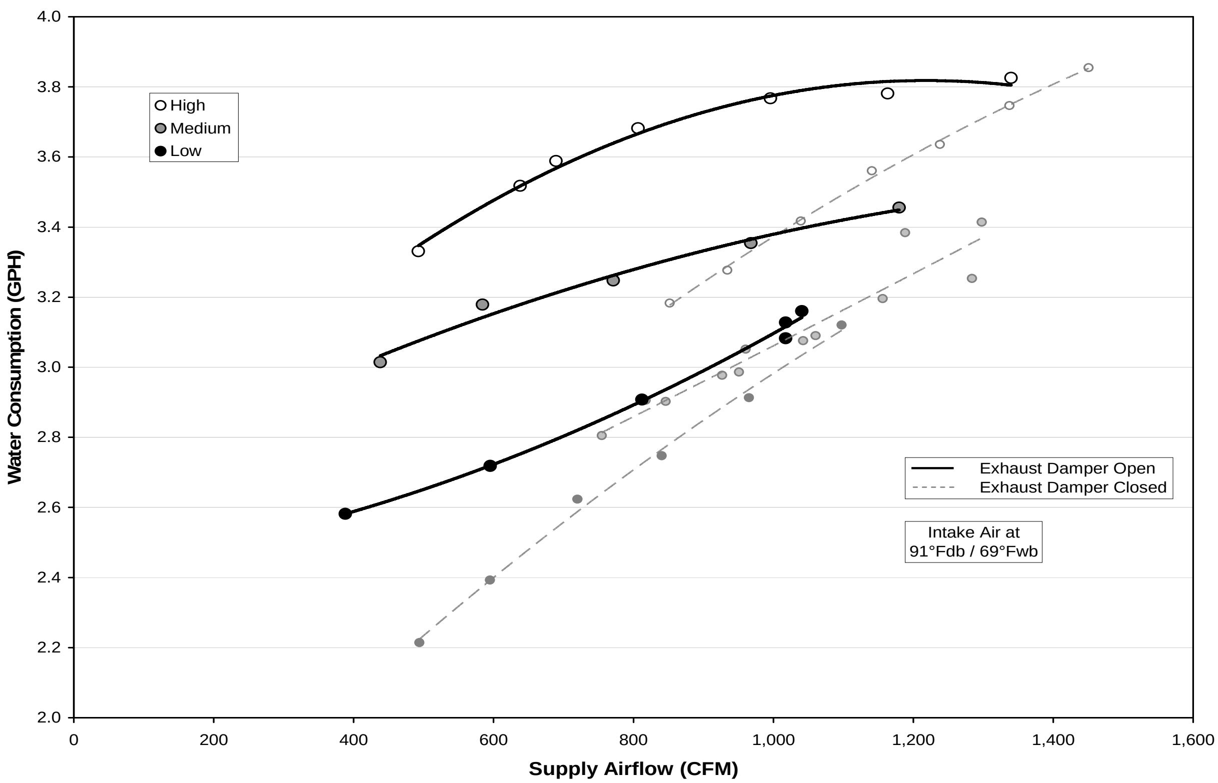

Figure 20: Water Consumption Rate versus Intake Wet-Bulb Depression

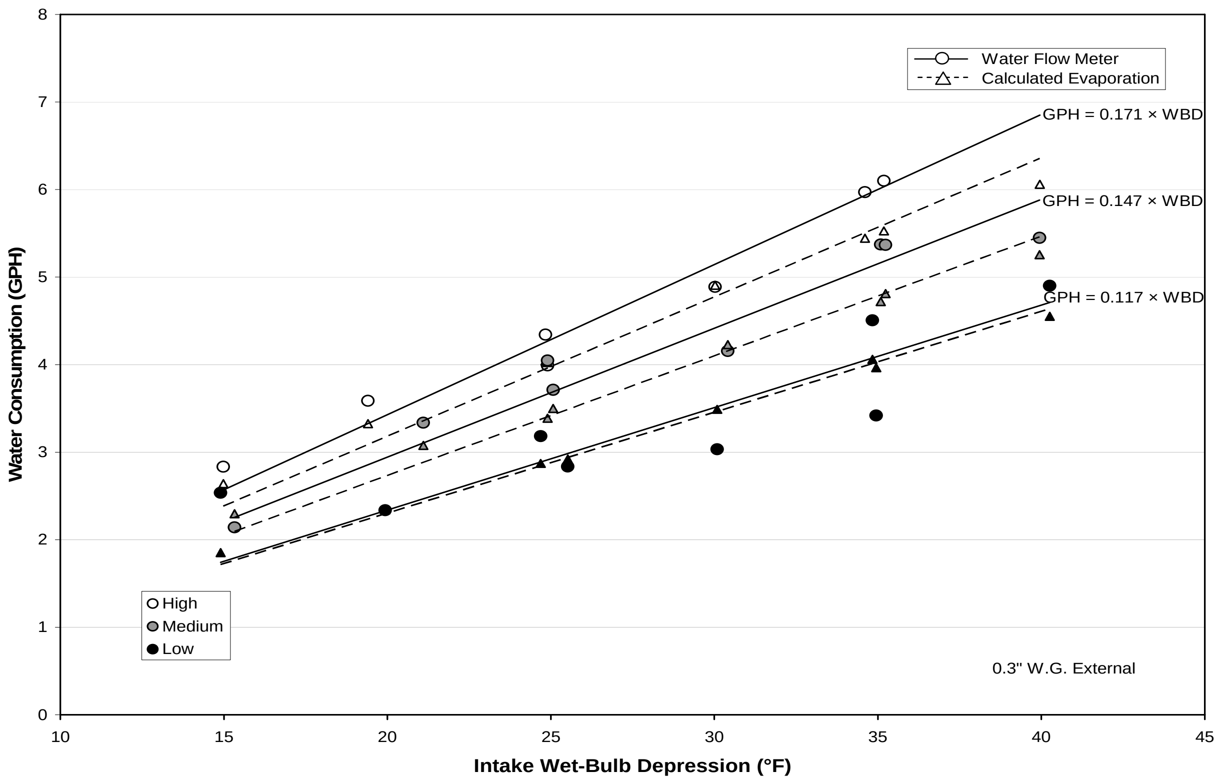

Figure 21: Overall Effectiveness versus Wet-Bulb Depression

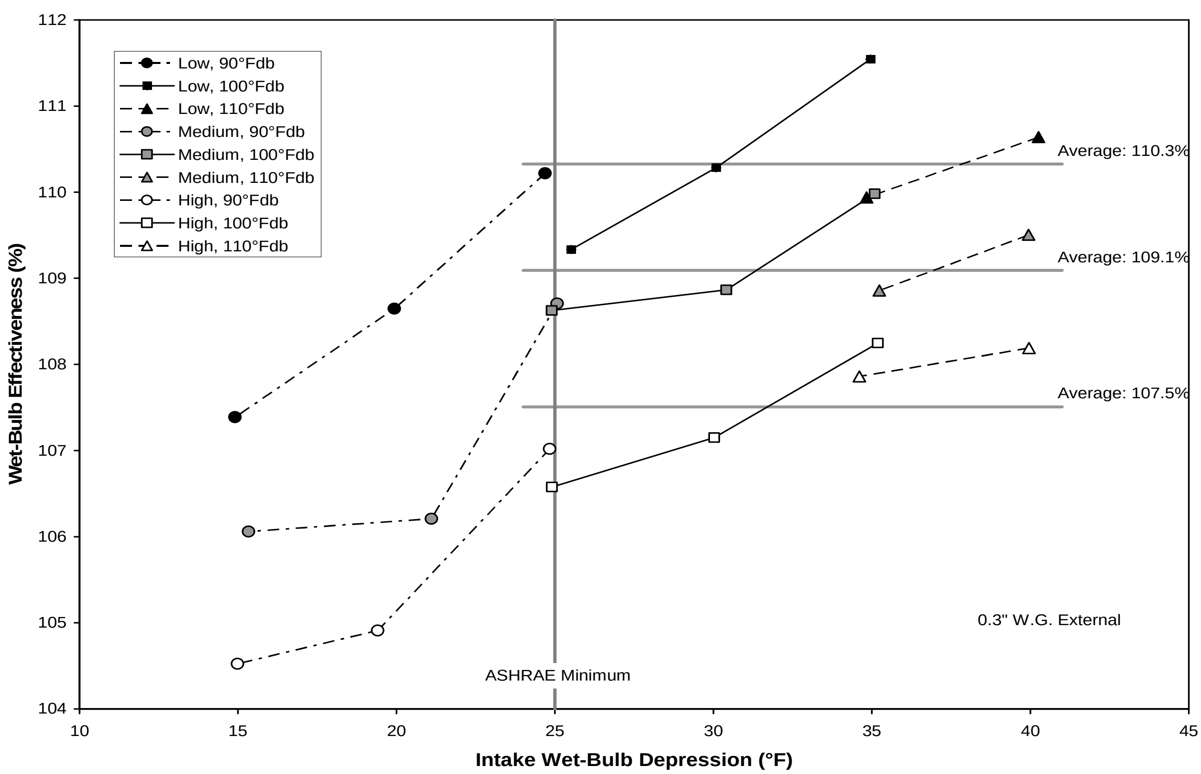

Figure 22: Sensible Capacity versus Wet-Bulb Depression

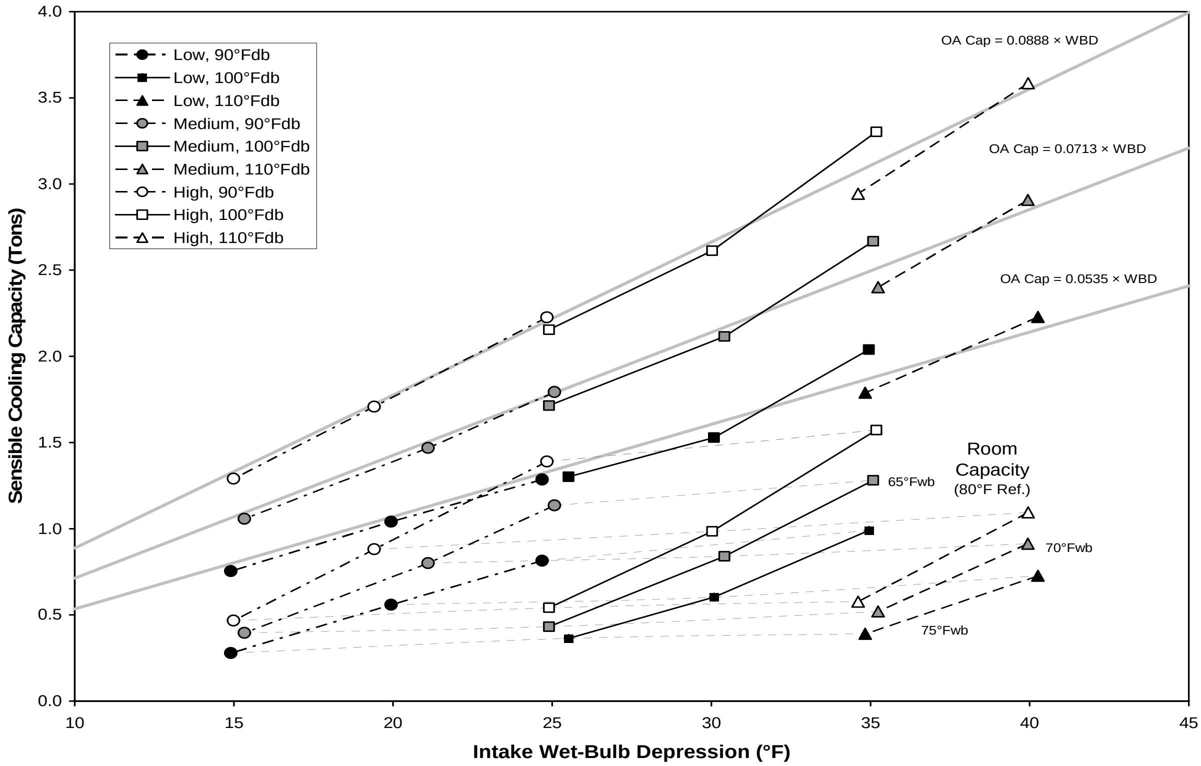

Figure 23: Sensible EER versus Wet-Bulb Depression

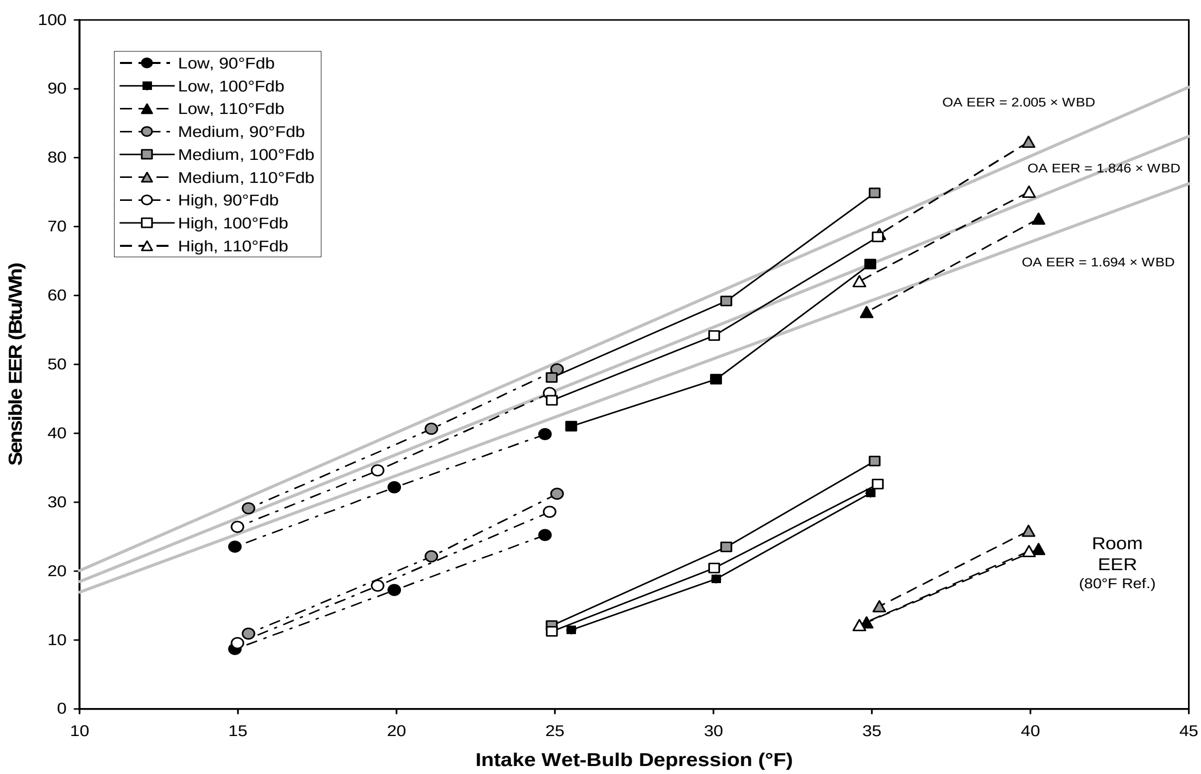

Table 6: OASys™ Test Data

Test Summary Information	Sensitivity to External Resistance - Exhaust Damper **Closed**, Intake at 91°Fdb / 69°Fwb																	
General	Low Fan Speed						Medium Fan Speed											
Date (2006)	26-Sep	26-Sep	26-Sep	26-Sep	26-Sep	26-Sep	21-Sep	27-Sep	21-Sep	27-Sep	21-Sep	27-Sep	21-Sep	21-Sep	27-Sep	21-Sep	27-Sep	27-Sep
Start Time	3:29p	3:52p	4:16p	4:36p	4:51p	5:10p	1:50p	11:32a	12:49p	11:50a	11:47a	12:04p	11:05a	2:36p	12:20p	2:18p	1:37p	2:00p
Duration (minutes)	10	10	10	10	10	10	10	10	30	10	10	10	30	10	30	10	10	10
Barometric Pressure (in. of Hg)	29.50	29.50	29.50	29.50	29.50	29.50	29.31	29.62	29.37	29.61	29.40	29.61	29.37	29.33	29.60	29.34	29.58	29.57
Inlet Air Properties																		
Dry Bulb Temperature (°F)	91.0	90.9	91.0	91.0	91.0	91.0	91.0	91.0	91.1	91.1	90.9	91.0	91.0	91.0	91.0	91.2	91.0	91.0
Dew Point Temperature (°F)	57.7	57.7	57.4	56.6	56.9	57.0	57.9	57.7	57.8	57.9	58.6	58.1	58.2	58.0	58.3	57.9	58.2	58.1
Wet Bulb Temperature (°F)	68.9	68.9	68.8	68.4	68.5	68.6	69.0	69.0	69.0	69.1	69.3	69.2	69.2	69.1	69.3	69.1	69.2	69.1
Wet Bulb Depression (°F)	22.1	22.0	22.2	22.6	22.5	22.4	22.0	22.1	22.1	22.0	21.6	21.8	21.8	21.9	21.8	22.1	21.8	21.8
Relative Humidity (%)	32.7	32.8	32.4	31.4	31.8	31.9	32.9	32.6	32.8	32.9	33.9	33.2	33.3	33.1	33.4	32.8	33.3	33.2
Interstage Air Properties																		
Measured Dry Bulb Temperature (°F)	75.6	75.5	75.1	74.0	72.9	72.2	74.4	73.2	72.4	72.2	71.7	72.4	71.1	74.4	71.7	73.4	71.3	71.8
Calculated Dry Bulb Temperature (°F) [1]	85.9	84.6	83.9	83.3	82.8	82.6	86.0	85.2	85.4	84.7	84.5	84.4	84.0	86.0	83.9	86.0	83.8	83.5
Supply Air Properties																		
Dry Bulb Temperature (°F)	68.9	68.3	67.8	67.1	67.0	66.8	68.9	68.8	68.4	68.6	68.5	68.5	68.1	68.0	68.3	67.7	68.0	67.8
Dew Point Temperature (°F)	66.1	65.8	65.5	65.0	65.0	65.1	66.3	65.8	66.2	65.9	66.3	66.0	66.0	66.0	66.0	65.9	66.0	65.9
Wet Bulb Temperature (°F)	67.4	67.0	66.6	66.0	66.0	66.0	67.5	67.2	67.2	67.1	67.4	67.1	67.1	67.0	67.1	66.9	67.0	66.9
Relative Humidity (%)	92.7	93.5	94.2	94.7	95.3	95.8	93.3	92.0	94.5	92.7	94.7	93.3	95.1	95.1	94.1	95.7	94.9	95.2
External Resistance (IW)	**0.00**	**0.10**	**0.20**	**0.30**	**0.40**	**0.48**	**0.00**	**0.00**	**0.10**	**0.10**	**0.20**	**0.20**	**0.30**	**0.30**	**0.30**	**0.40**	**0.40**	**0.46**
Exhaust Air Properties																		
Dry Bulb Temperature (°F)	83.8	81.9	80.5	79.2	78.3	77.5	84.3	84.2	83.2	82.9	81.3	81.8	80.5	80.7	80.9	80.0	80.3	79.7
Dew Point Temperature (°F)	77.3	76.3	75.5	74.5	73.7	73.0	77.2	77.3	76.9	76.9	76.0	76.4	75.2	75.5	75.8	74.6	75.0	74.6
Wet Bulb Temperature (°F)	78.9	77.7	76.8	75.7	74.9	74.2	79.0	79.0	78.5	78.4	77.3	77.7	76.6	76.9	77.1	76.0	76.4	75.9
Relative Humidity (%)	80.8	83.2	84.8	85.4	85.7	86.0	79.3	80.0	81.6	82.2	83.7	83.6	84.1	84.1	84.4	83.8	84.0	84.4
Water Properties																		
Basin Temperature (°F)	71.9	71.2	70.2	69.0	68.4	67.8	72.5	72.1	71.9	71.3	72.3	71.0	71.7	70.4	70.2	69.7	69.8	69.5
Makeup Water Flow (gph) - measured	3.6	3.0	3.2	2.9	2.5	2.1	3.4	3.3	3.3	3.2	3.4	3.4	2.7	3.5	2.9	3.3	2.7	3.0
Total Evaporation Rate (gph) - calculated	3.1	2.9	2.8	2.6	2.4	2.2	3.4	3.3	3.4	3.2	3.1	3.1	3.0	3.1	3.0	2.9	2.9	2.8
Fraction to Supply Air (%)	70%	63%	57%	52%	46%	41%	75%	76%	69%	68%	64%	63%	59%	59%	57%	55%	52%	49%
Power Consumption																		
Voltage (V)	115	115	115	115	115	115	115	115	115	115	115	115	115	116	114	115	115	115
Current (A)	4.0	4.0	4.0	3.9	3.9	3.9	4.4	4.4	4.4	4.4	4.4	4.4	4.3	4.3	4.4	4.3	4.3	4.3
Power (W)	390	390	388	387	382	380	436	441	438	441	438	441	435	438	437	433	435	433
Power Factor	0.85	0.85	0.85	0.85	0.85	0.85	0.87	0.87	0.87	0.87	0.87	0.87	0.87	0.87	0.87	0.87	0.87	0.87
Fan Speed (RPM)	877	877	877	882	887	897	1,000	996	1,000	994	1,000	994	998	1,002	992	1,001	996	998
Performance																		
Dry Bulb Temperature Drop (°F)	22.2	22.7	23.2	23.9	24.0	24.2	22.2	22.2	22.7	22.4	22.4	22.5	23.0	23.0	22.7	23.5	23.0	23.1
Dew Point Temperature Rise (°F)	8.4	8.1	8.1	8.4	8.1	8.0	8.4	8.2	8.4	8.0	7.8	7.8	7.8	8.0	7.7	8.0	7.8	7.7
Wet-Bulb Effectiveness (%)	100.3	102.9	104.5	105.9	107.0	107.9	100.7	100.6	102.7	101.9	103.8	103.0	105.1	104.7	104.4	106.1	105.4	106.0
Supply Airflow Rate (CFM)	1,098	965	840	720	595	494	1,298	1,284	1,188	1,156	1,061	1,043	951	960	927	846	818	755
Exhaust Airflow Rate (CFM)	185	225	256	288	315	339	166	153	208	205	247	241	282	278	284	314	328	344
Intake Airflow Rate (CFM)	1,276	1,184	1,091	1,002	905	829	1,458	1,431	1,390	1,355	1,301	1,278	1,226	1,232	1,205	1,154	1,140	1,093
Fan Intake CFM / W	3.28	3.04	2.81	2.59	2.37	2.18	3.34	3.24	3.17	3.07	2.97	2.90	2.82	2.81	2.76	2.66	2.62	2.52
Room Capacity (tons; 80°F reference)	1.05	0.97	0.88	0.80	0.67	0.56	1.23	1.24	1.18	1.13	1.05	1.03	0.97	0.98	0.94	0.89	0.84	0.79
Room EER (Btu/Wh, 80°F reference)	32.3	29.9	27.2	24.8	20.9	17.6	33.9	33.6	32.3	30.8	28.7	28.1	26.8	26.9	25.7	24.6	23.2	21.9
CA T-20 ECER (Btu/Wh) [2]	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
Sensible Cooling of Outside Air (tons)	2.09	1.88	1.68	1.48	1.23	1.03	2.46	2.46	2.31	2.23	2.03	2.02	1.87	1.88	1.82	1.69	1.62	1.50
Outside Air EER (Btu/Wh)	64.4	57.8	51.8	45.9	38.6	32.4	67.6	66.9	63.1	60.8	55.7	55.0	51.5	51.6	49.9	46.9	44.6	41.6

[1] Calculated interstage temperature is the intersection of the intake dew point and the supply wet bulb.

[2] ECER can only be derived when the supply resistance is 0.3" and the wet-bulb depression is >=25°F

Table 6: OASys™ Test Data (Continued)

Test Summary Information	Sensitivity to External Resistance						Voltage Sensitivity - Damper **Closed**, Intake at 91°Fdb / 69°Fwb						
General	High Fan Speed						Medium Fan Speed						
Date (2006)	27-Sep	27-Sep	27-Sep	27-Sep	27-Sep	27-Sep	21-Sep	21-Sep	21-Sep	21-Sep	21-Sep	21-Sep	21-Sep
Start Time	2:39p	2:54p	3:09p	3:24p	3:40p	3:47p	3:10p	2:50p	4:45p	3:34p	4:00p	4:22p	4:33p
Duration (minutes)	10	10	10	10	5	5	10	15	10	10	10	10	10
Barometric Pressure (in. of Hg)	29.56	29.56	29.56	29.56	29.55	29.55	29.33	29.33	29.25	29.32	29.31	29.29	29.26
Inlet Air Properties													
Dry Bulb Temperature (°F)	91.0	91.0	91.0	91.0	90.9	91.0	90.9	91.0	91.1	91.0	91.0	91.0	91.0
Dew Point Temperature (°F)	57.9	57.9	58.0	57.6	57.6	57.7	56.7	57.9	57.7	58.0	58.3	58.0	57.3
Wet Bulb Temperature (°F)	69.1	69.1	69.1	68.9	68.9	69.0	68.4	69.0	68.9	69.0	69.2	69.0	68.7
Wet Bulb Depression (°F)	22.0	22.0	21.9	22.1	22.1	22.1	22.6	22.0	22.2	22.0	21.8	22.0	22.3
Relative Humidity (%)	33.0	32.9	33.1	32.6	32.6	32.7	31.6	33.0	32.6	33.0	33.4	33.0	32.3
Interstage Air Properties													
Measured Dry Bulb Temperature (°F)	72.7	73.7	72.8	73.6	73.7	74.2	71.8	73.8	70.8	75.3	72.9	74.1	73.5
Calculated Dry Bulb Temperature (°F) [1]	86.1	85.5	85.2	84.6	84.2	84.1	84.3	84.0	84.0	83.8	83.8	83.4	83.1
Supply Air Properties													
Dry Bulb Temperature (°F)	69.4	69.1	68.9	68.4	68.1	68.1	67.3	67.8	67.6	67.7	67.8	67.6	67.1
Dew Point Temperature (°F)	66.1	66.0	66.0	65.7	65.6	65.7	65.3	65.9	65.8	65.9	66.0	65.7	65.3
Wet Bulb Temperature (°F)	67.6	67.4	67.3	66.9	66.8	66.9	66.3	66.9	66.7	66.8	67.0	66.7	66.3
Relative Humidity (%)	91.2	91.7	92.3	92.9	93.4	94.0	95.2	95.4	95.7	95.7	95.8	95.6	96.0
External Resistance (IW)	**0.00**	**0.10**	**0.20**	**0.30**	**0.40**	**0.50**	0.30	0.30	0.30	0.30	0.30	0.30	0.30
Exhaust Air Properties													
Dry Bulb Temperature (°F)	84.6	83.4	82.4	81.5	80.8	80.1	80.2	80.5	80.1	80.5	80.3	80.0	79.7
Dew Point Temperature (°F)	77.8	77.4	76.8	76.1	75.5	75.0	74.6	75.2	75.0	75.2	75.2	75.1	74.8
Wet Bulb Temperature (°F)	79.5	78.9	78.2	77.5	76.9	76.3	76.1	76.6	76.3	76.6	76.5	76.4	76.0
Relative Humidity (%)	80.3	82.0	83.2	83.8	83.9	84.4	83.3	83.8	84.3	83.8	84.5	85.2	85.0
Water Properties													
Basin Temperature (°F)	72.1	71.6	71.7	70.7	70.1	69.7	70.1	70.2	69.5	70.4	70.0	69.8	69.5
Makeup Water Flow (gph) - measured	3.4	3.8	3.1	3.9	2.9	2.6	3.3	2.7	3.7	3.1	2.9	2.7	2.6
Total Evaporation Rate (gph) - calculated	3.9	3.8	3.6	3.6	3.4	3.3	3.2	3.0	3.0	2.9	2.7	2.6	2.5
Fraction to Supply Air (%)	73%	68%	64%	61%	57%	53%	61%	59%	60%	58%	57%	55%	54%
Power Consumption													
Voltage (V)	115	115	115	115	115	115	**121**	**115**	**116**	**111**	**104**	**101**	**95**
Current (A)	6.4	6.4	6.4	6.4	6.4	6.4	4.4	4.3	4.3	4.3	4.3	4.3	4.3
Power (W)	586	587	586	586	587	584	456	436	436	419	396	381	359
Power Factor	0.80	0.80	0.80	0.80	0.80	0.80	0.87	0.87	0.87	0.88	0.88	0.88	0.88
Fan Speed (RPM)	1,121	1,121	1,121	1,121	1,121	1,122	1,023	1,001	1,000	977	938	910	864
Performance													
Dry Bulb Temperature Drop (°F)	21.6	21.9	22.1	22.6	22.8	23.0	23.6	23.2	23.5	23.3	23.2	23.4	23.9
Dew Point Temperature Rise (°F)	8.2	8.1	8.0	8.1	8.0	8.0	8.6	8.0	8.1	7.9	7.7	7.8	8.0
Wet-Bulb Effectiveness (%)	98.4	99.8	100.9	102.2	103.3	104.1	104.7	105.4	105.9	106.0	106.2	106.5	107.2
Supply Airflow Rate (CFM)	1,451	1,338	1,238	1,141	1,039	934	981	944	954	909	848	801	726
Exhaust Airflow Rate (CFM)	194	232	263	293	324	352	283	285	275	277	271	269	265
Intake Airflow Rate (CFM)	1,637	1,562	1,494	1,427	1,357	1,279	1,257	1,223	1,223	1,180	1,114	1,065	986
Fan Intake CFM / W	2.79	2.66	2.55	2.44	2.31	2.19	2.76	2.80	2.81	2.82	2.82	2.80	2.75
Room Capacity (tons; 80°F reference)	1.32	1.26	1.18	1.14	1.06	0.96	1.06	0.98	1.01	0.95	0.88	0.85	0.80
Room EER (Btu/Wh, 80°F reference)	27.1	25.7	24.2	23.3	21.7	19.7	28.0	27.0	27.8	27.3	26.7	26.7	26.7
CA T-20 ECER (Btu/Wh) [2]	-	-	-	-	-	-	-	-	-	-	-	-	-
Sensible Cooling of Outside Air (tons)	2.70	2.52	2.35	2.22	2.04	1.84	1.98	1.87	1.91	1.81	1.68	1.60	1.48
Outside Air EER (Btu/Wh)	55.2	51.6	48.2	45.4	41.7	37.9	52.1	51.4	52.6	51.7	50.8	50.5	49.5

[1] Calculated interstage temperature is the intersection of the intake dew point and the supply wet bulb.

[2] ECER can only be derived when the supply resistance is 0.3" and the wet-bulb depression is >=25°F

Table 6: OASys™ Test Data (Continued)

Test Summary Information	Sensitivity to External Resistance - Exhaust Damper **Open**, Intake at 91°Fdb / 69°Fwb																
General	Low Fan Speed					Medium Fan Speed					High Fan Speed						
Date (2006)	16-Oct	16-Oct	16-Oct	16-Oct	16-Oct	13-Oct	13-Oct	13-Oct	13-Oct	13-Oct	13-Oct	13-Oct	13-Oct	13-Oct	13-Oct	13-Oct	13-Oct
Start Time	1:48p	2:40p	2:57p	3:15p	3:48p	4:19p	3:53p	3:31p	3:12p	2:53p	10:21a	11:00a	11:44a	12:46p	1:24p	1:48p	2:23p
Duration (minutes)	15	10	10	10	15	10	10	10	10	15	30	30	15	30	15	15	15
Barometric Pressure (in. of Hg)	29.33	29.32	29.31	29.31	29.31	29.29	29.29	29.29	29.29	29.29	29.34	29.32	29.30	29.28	29.29	29.30	29.29
Inlet Air Properties																	
Dry Bulb Temperature (°F)	91.0	91.0	90.9	90.9	91.0	90.9	91.0	91.0	91.0	91.0	91.0	91.0	91.0	91.0	91.0	91.0	91.0
Dew Point Temperature (°F)	56.4	57.7	57.8	58.4	57.8	58.0	58.1	57.8	57.0	57.4	57.7	58.1	58.0	57.8	57.7	57.9	58.7
Wet Bulb Temperature (°F)	68.3	68.9	68.9	69.2	69.0	69.0	69.1	69.0	68.6	68.7	68.9	69.1	69.0	68.9	68.9	69.0	69.4
Wet Bulb Depression (°F)	22.7	22.1	22.0	21.7	22.1	21.9	21.9	22.1	22.4	22.2	22.1	21.9	21.9	22.1	22.1	22.0	21.6
Relative Humidity (%)	31.3	32.7	33.0	33.6	32.8	33.2	33.2	32.9	32.0	32.4	32.7	33.2	33.1	32.8	32.7	32.9	34.0
Interstage Air Properties																	
Measured Dry Bulb Temperature (°F)	73.6	73.3	70.1	69.3	68.9	72.2	69.7	69.4	68.2	68.4	73.3	72.1	69.6	68.3	68.6	67.6	67.7
Calculated Dry Bulb Temperature (°F) [1]	82.9	83.6	82.0	81.1	80.2	83.2	82.0	80.9	80.1	79.9	83.1	82.3	81.8	80.9	80.5	80.4	80.3
Supply Air Properties																	
Dry Bulb Temperature (°F)	67.3	68.1	67.4	67.2	66.6	68.1	67.5	66.9	66.2	66.4	68.1	67.8	67.4	67.0	66.7	66.8	67.2
Dew Point Temperature (°F)	64.5	65.4	65.1	65.2	64.6	65.4	65.2	64.8	64.1	64.3	65.2	65.3	65.1	64.8	64.6	64.7	65.1
Wet Bulb Temperature (°F)	65.8	66.6	66.2	66.2	65.6	66.7	66.3	65.8	65.2	65.3	66.5	66.4	66.2	65.8	65.7	65.7	66.2
Relative Humidity (%)	92.6	93.1	94.0	95.1	94.9	92.9	94.0	94.7	95.0	94.8	92.0	93.3	94.0	94.4	94.6	94.8	94.8
External Resistance (IW)	**0.00**	**0.00**	**0.10**	**0.20**	**0.30**	**0.00**	**0.10**	**0.20**	**0.30**	**0.38**	**0.00**	**0.10**	**0.20**	**0.30**	**0.37**	**0.40**	**0.48**
Exhaust Air Properties																	
Dry Bulb Temperature (°F)	80.6	80.9	78.5	76.4	73.9	81.0	78.6	76.3	74.0	73.1	80.8	79.1	77.3	75.0	74.1	73.9	73.3
Dew Point Temperature (°F)	75.0	75.5	73.6	71.9	70.4	76.2	74.0	72.5	71.0	70.3	75.4	74.3	73.3	72.1	71.4	71.2	70.8
Wet Bulb Temperature (°F)	76.5	76.9	74.9	73.2	71.4	77.4	75.2	73.5	71.8	71.1	76.8	75.6	74.4	72.9	72.2	72.0	71.5
Relative Humidity (%)	83.1	83.5	84.9	86.1	88.6	85.5	86.0	87.8	90.1	90.9	83.6	85.5	87.5	90.9	91.2	91.4	92.1
Water Properties																	
Basin Temperature (°F)	70.1	70.8	69.3	68.4	66.8	70.3	68.9	67.7	66.6	66.3	71.2	70.2	69.4	68.0	67.3	67.1	67.1
Makeup Water Flow (gph) - measured	3.7	3.2	3.6	3.9	3.3	3.8	3.6	3.6	3.8	2.8	3.9	3.9	4.2	4.1	4.1	4.1	4.1
Total Evaporation Rate (gph) - calculated	3.1	3.1	2.9	2.7	2.6	3.5	3.4	3.3	3.2	3.0	3.8	3.8	3.8	3.7	3.6	3.5	3.3
Fraction to Supply Air (%)	59%	60%	47%	35%	24%	59%	48%	38%	30%	23%	61%	52%	44%	35%	31%	29%	22%
Power Consumption																	
Voltage (V)	115	115	115	115	116	115	115	115	115	115	114	114	114	114	115	115	115
Current (A)	4.0	4.0	4.0	4.0	4.0	4.4	4.4	4.4	4.4	4.3	6.4	6.4	6.4	6.4	6.4	6.4	6.4
Power (W)	387	388	389	390	388	439	438	438	437	436	582	579	580	582	586	585	584
Power Factor	0.85	0.85	0.85	0.85	0.85	0.87	0.87	0.87	0.87	0.87	0.80	0.80	0.80	0.80	0.80	0.80	0.80
Fan Speed (RPM)	873	875	875	877	880	999	998	997	996	998	1,122	1,121	1,121	1,122	1,122	1,122	1,122
Performance																	
Dry Bulb Temperature Drop (°F)	23.7	23.0	23.5	23.7	24.4	22.8	23.5	24.1	24.8	24.6	22.9	23.2	23.5	24.1	24.3	24.2	23.8
Dew Point Temperature Rise (°F)	8.1	7.7	7.3	6.8	6.8	7.4	7.1	7.0	7.1	6.8	7.5	7.2	7.1	7.0	6.9	6.8	6.4
Wet-Bulb Effectiveness (%)	104.4	103.9	106.9	109.3	110.5	104.2	107.0	109.4	110.7	110.8	103.6	105.9	107.3	108.9	109.7	110.0	110.2
Supply Airflow Rate (CFM)	1,018	1,018	812	595	388	1,180	968	771	584	438	1,340	1,164	996	807	689	638	492
Exhaust Airflow Rate (CFM)	278	276	394	538	670	298	435	563	686	776	333	447	561	689	763	792	890
Intake Airflow Rate (CFM)	1,290	1,288	1,200	1,128	1,053	1,471	1,396	1,328	1,264	1,208	1,665	1,603	1,549	1,488	1,446	1,423	1,376
Fan Intake CFM / W	3.33	3.31	3.08	2.89	2.71	3.35	3.19	3.03	2.89	2.77	2.86	2.77	2.67	2.55	2.47	2.43	2.36
Room Capacity (tons; 80°F reference)	1.11	1.04	0.87	0.65	0.44	1.20	1.03	0.86	0.69	0.51	1.36	1.21	1.07	0.90	0.78	0.72	0.54
Room EER (Btu/Wh, 80°F reference)	34.3	32.0	26.9	20.0	13.7	32.7	28.2	23.6	19.0	14.0	28.1	25.1	22.1	18.5	16.0	14.8	11.0
CA T-20 ECER (Btu/Wh) [2]	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
Sensible Cooling of Outside Air (tons)	2.06	2.00	1.63	1.21	0.81	2.29	1.94	1.59	1.24	0.92	2.62	2.30	2.00	1.65	1.43	1.32	1.00
Outside Air EER (Btu/Wh)	63.8	61.6	50.2	37.1	25.0	62.8	53.0	43.5	34.0	25.3	54.1	47.7	41.3	34.0	29.2	27.0	20.5

[1] Calculated interstage temperature is the intersection of the intake dew point and the supply wet bulb.

[2] ECER can only be derived when the supply resistance is 0.3" and the wet-bulb depression is >=25°F

Table 6: OASys™ Test Data (Continued)

Test Summary Information	Sensitivity to External Resistance - Exhaust Damper **Closed**, Intake at 100°Fdb / 70°Fwb																	
General	Low Fan Speed					Medium Fan Speed						High Fan Speed						
Date (2006)	22-Sep	22-Sep	22-Sep	22-Sep	22-Sep	22-Sep	22-Sep	22-Sep	22-Sep	22-Sep	22-Sep	22-Sep	22-Sep	22-Sep	22-Sep	22-Sep	22-Sep	22-Sep
Start Time	5:01p	4:50p	4:37p	4:25p	3:50p	10:27a	9:58a	9:38a	8:07a	8:51a	9:15a	11:23a	11:48a	1:27p	1:03p	1:55p	2:12p	2:35p
Duration (minutes)	5	5	5	5	30	15	15	15	30	15	15	15	15	15	15	15	10	15
Barometric Pressure (in. of Hg)	29.24	29.24	29.24	29.24	29.25	29.29	29.29	29.29	29.27	29.28	29.29	29.29	29.29	29.28	29.29	29.28	29.25	29.25
Inlet Air Properties																		
Dry Bulb Temperature (°F)	100.4	100.5	100.4	100.4	100.4	100.4	100.3	100.3	100.6	100.8	101.0	100.4	100.4	100.4	100.4	100.4	100.4	100.4
Dew Point Temperature (°F)	56.1	47.7	53.6	53.2	53.1	53.4	54.2	51.5	53.6	53.5	53.1	53.9	53.9	53.7	52.8	51.6	57.6	53.4
Wet Bulb Temperature (°F)	70.9	67.5	69.8	69.6	69.6	69.7	70.0	68.9	69.8	69.9	69.8	70.0	69.9	69.9	69.5	69.0	71.6	69.7
Wet Bulb Depression (°F)	29.6	33.0	30.6	30.8	30.8	30.7	30.3	31.4	30.8	31.0	31.2	30.5	30.5	30.6	31.0	31.4	28.8	30.7
Relative Humidity (%)	23.1	16.9	21.2	20.8	20.8	21.0	21.7	19.7	21.0	20.8	20.4	21.3	21.3	21.2	20.5	19.6	24.5	20.9
Interstage Air Properties																		
Measured Dry Bulb Temperature (°F)	74.6	71.3	72.1	72.0	70.5	76.0	75.5	73.6	72.8	73.2	72.4	78.4	78.3	77.6	76.3	76.1	76.0	73.8
Calculated Dry Bulb Temperature (°F) [1]	92.9	92.9	91.1	90.0	88.9	93.3	92.1	90.3	90.8	90.3	90.3	93.6	93.0	92.5	91.7	91.4	90.7	90.3
Supply Air Properties																		
Dry Bulb Temperature (°F)	70.5	67.1	68.5	67.7	67.1	69.6	69.5	68.1	68.4	68.1	67.8	70.3	69.9	69.4	68.6	67.9	70.1	68.0
Dew Point Temperature (°F)	67.3	63.6	65.8	65.4	65.0	66.1	66.1	64.8	65.6	65.5	65.3	66.2	66.1	66.0	65.5	64.8	67.6	65.4
Wet Bulb Temperature (°F)	68.7	65.1	67.0	66.5	66.1	67.6	67.5	66.2	66.9	66.7	66.5	67.9	67.7	67.5	66.8	66.2	68.8	66.6
Relative Humidity (%)	91.4	90.1	92.6	93.7	94.9	90.2	90.8	91.0	92.8	92.9	93.5	88.8	89.6	90.8	91.5	91.7	93.7	93.3
External Resistance (IW)	**0.00**	**0.10**	**0.20**	**0.30**	**0.48**	**0.00**	**0.10**	**0.20**	**0.32**	**0.40**	**0.45**	**0.00**	**0.10**	**0.20**	**0.30**	**0.40**	**0.50**	**0.54**
Exhaust Air Properties																		
Dry Bulb Temperature (°F)	91.2	88.1	86.4	84.9	82.6	90.7	88.8	86.7	84.6	84.2	83.8	91.1	89.5	87.7	86.2	85.2	85.5	84.3
Dew Point Temperature (°F)	82.0	79.3	79.1	77.5	75.4	81.7	81.1	79.5	78.4	77.4	77.0	82.2	81.4	80.1	78.9	77.8	78.7	76.9
Wet Bulb Temperature (°F)	84.0	81.4	80.9	79.3	77.2	83.7	82.8	81.1	79.9	79.1	78.7	84.1	83.2	81.9	80.6	79.6	80.3	78.8
Relative Humidity (%)	74.7	75.4	79.0	78.4	78.8	75.0	78.1	79.2	81.7	80.2	80.1	75.2	77.3	78.3	79.1	78.6	80.3	78.7
Water Properties																		
Basin Temperature (°F)	73.9	70.4	70.5	69.1	67.8	74.1	73.2	71.3	70.8	70.2	69.5	74.5	73.4	73.1	71.9	70.8	72.2	70.2
Makeup Water Flow (gph) - measured	4.1	4.4	3.6	3.6	3.3	5.0	4.9	4.4	4.1	4.1	4.3	5.3	5.1	5.2	4.8	4.9	3.6	4.7
Total Evaporation Rate (gph) - calculated	4.0	4.4	3.8	3.5	3.1	4.8	4.5	4.6	4.2	3.9	4.0	5.3	5.2	5.1	5.0	4.9	4.2	4.4
Fraction to Supply Air (%)	71%	65%	58%	52%	41%	73%	66%	62%	58%	54%	51%	74%	68%	64%	60%	56%	51%	50%
Power Consumption																		
Voltage (V)	115	115	115	115	115	115	115	114	115	114	114	114	114	114	114	115	115	115
Current (A)	3.9	3.9	3.9	3.9	3.8	4.3	4.3	4.3	4.3	4.3	4.3	6.3	6.3	6.3	6.3	6.4	6.3	6.3
Power (W)	382	381	383	380	374	431	431	430	432	427	425	575	576	576	577	578	576	574
Power Factor	0.85	0.85	0.85	0.85	0.85	0.87	0.87	0.87	0.87	0.87	0.87	0.80	0.80	0.80	0.80	0.79	0.79	0.79
Fan Speed (RPM)	878	875	881	884	900	1,001	999	998	1,005	1,002	1,003	1,122	1,122	1,121	1,122	1,122	1,123	1,123
Performance																		
Dry Bulb Temperature Drop (°F)	29.9	33.4	31.9	32.7	33.3	30.8	30.8	32.2	32.3	32.7	33.2	30.2	30.6	31.0	31.8	32.5	30.3	32.4
Dew Point Temperature Rise (°F)	11.2	15.9	12.1	12.1	11.9	12.7	11.9	13.3	12.1	12.0	12.2	12.4	12.2	12.3	12.6	13.3	10.0	12.1
Wet-Bulb Effectiveness (%)	101.3	101.2	104.2	106.1	108.0	100.3	101.8	102.6	104.8	105.5	106.3	99.0	100.2	101.5	102.8	103.4	105.2	105.5
Supply Airflow Rate (CFM)	1,125	986	865	734	512	1,296	1,176	1,060	951	851	795	1,463	1,349	1,242	1,135	1,027	919	869
Exhaust Airflow Rate (CFM)	165	208	246	285	356	175	216	252	283	310	339	184	227	269	311	344	377	389
Intake Airflow Rate (CFM)	1,282	1,185	1,103	1,012	862	1,462	1,384	1,303	1,225	1,154	1,126	1,637	1,565	1,501	1,436	1,361	1,287	1,249
Fan Intake CFM / W	3.36	3.11	2.88	2.66	2.30	3.39	3.21	3.03	2.83	2.70	2.65	2.85	2.72	2.61	2.49	2.35	2.24	2.18
Room Capacity (tons; 80°F reference)	0.90	1.07	0.83	0.75	0.55	1.13	1.04	1.06	0.93	0.85	0.81	1.19	1.15	1.10	1.09	1.04	0.76	0.87
Room EER (Btu/Wh, 80°F reference)	28.1	33.6	26.2	23.8	17.7	31.4	28.9	29.6	25.8	23.8	22.9	24.9	23.9	23.0	22.6	21.7	15.9	18.3
CA T-20 ECER (Btu/Wh) [2]	-	-	-	25.7	-	-	-	-	-	-	-	-	-	-	24.7	-	-	-
Sensible Cooling of Outside Air (tons)	2.82	2.76	2.31	2.01	1.43	3.35	3.04	2.86	2.57	2.33	2.21	3.71	3.46	3.23	3.03	2.80	2.33	2.36
Outside Air EER (Btu/Wh)	88.7	86.9	72.5	63.4	45.7	93.1	84.7	79.9	71.3	65.6	62.4	77.3	72.1	67.3	63.0	58.1	48.7	49.3

[1] Calculated interstage temperature is the intersection of the intake dew point and the supply wet bulb.

[2] ECER can only be derived when the supply resistance is 0.3" and the wet-bulb depression is >=25°F

Table 6: OASys™ Test Data (Continued)

Test Summary Information	**Estimated** Performance Sensitivity to External Resistance - Exhaust Damper **Open**, Intake at 100°Fdb / 70°Fwb														
General	Low Fan Speed				Medium Fan Speed					High Fan Speed					
Date (2006)															
Start Time															
Duration (minutes)															
Barometric Pressure (in. of Hg)	29.65	29.65	29.65	29.65	29.65	29.65	29.65	29.65	29.65	29.65	29.64	29.65	29.65	29.65	29.65
Inlet Air Properties															
Dry Bulb Temperature (°F)	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
Dew Point Temperature (°F)	54.1	54.1	54.1	54.1	54.1	54.1	54.1	54.1	54.1	54.1	54.1	54.1	54.1	54.1	54.1
Wet Bulb Temperature (°F)	70.0	70.0	70.0	70.0	70.0	69.6	70.0	70.0	70.0	70.0	70.1	70.0	70.0	70.0	70.0
Wet Bulb Depression (°F)	30.0	30.0	30.0	30.0	30.0	30.4	30.0	30.0	30.0	30.0	29.9	30.0	30.0	30.0	30.0
Relative Humidity (%)	21.8	21.8	21.8	21.8	21.8	21.8	21.8	21.8	21.8	21.8	21.8	21.8	21.8	21.8	21.8
Interstage Air Properties															
Measured Dry Bulb Temperature (°F)															
Calculated Dry Bulb Temperature (°F) [1]	90.4	88.1	86.7	85.6	89.9	88.1	86.9	86.1	85.6	88.8	88.0	86.9	86.0	85.3	84.7
Supply Air Properties															
Dry Bulb Temperature (°F)	68.8	67.9	67.2	66.8	68.8	67.5	67.3	67.0	66.8	68.6	68.1	67.5	67.1	66.7	66.5
Dew Point Temperature (°F)	66.3	65.6	65.3	65.0	66.0	65.2	65.3	65.1	64.9	65.6	65.6	65.2	65.0	64.8	64.6
Wet Bulb Temperature (°F)	67.1	66.4	65.9	65.6	66.9	65.9	66.0	65.7	65.6	66.6	66.4	66.0	65.7	65.5	65.3
Relative Humidity (%)	91.7	92.6	93.6	93.8	90.9	92.2	93.2	93.6	93.7	90.4	91.6	92.5	93.2	93.6	93.8
External Resistance (IW)	**0.00**	**0.10**	**0.20**	**0.30**	**0.00**	**0.10**	**0.20**	**0.30**	**0.40**	**0.00**	**0.10**	**0.20**	**0.30**	**0.40**	**0.50**
Exhaust Air Properties															
Dry Bulb Temperature (°F)															
Dew Point Temperature (°F)															
Wet Bulb Temperature (°F)															
Relative Humidity (%)															
Water Properties															
Basin Temperature (°F)															
Makeup Water Flow (gph) - measured															
Total Evaporation Rate (gph) - calculated	5.1	4.3	5.5	5.0	5.0	4.9	4.8	4.6	4.4	5.5	5.5	5.4	5.3	5.1	4.8
Fraction to Supply Air (%)															
Power Consumption															
Voltage (V)															
Current (A)															
Power (W)	378	379	378	374	431	431	431	431	430	582	581	582	583	583	584
Power Factor															
Fan Speed (RPM)															
Performance															
Dry Bulb Temperature Drop (°F)	31.2	32.1	32.8	33.2	31.2	32.5	32.7	33.0	33.2	31.4	31.9	32.5	32.9	33.3	33.5
Dew Point Temperature Rise (°F)	12.2	11.5	11.2	10.8	11.9	11.1	11.2	11.0	10.8	11.5	11.5	11.1	10.9	10.7	10.5
Wet-Bulb Effectiveness (%)	104.1	107.1	109.4	110.7	104.0	106.9	108.9	110.0	110.6	104.7	106.6	108.4	109.8	110.9	111.7
Supply Airflow Rate (CFM)	1,069	848	624	400	1,193	979	781	591	411	1,317	1,143	974	801	628	454
Exhaust Airflow Rate (CFM)	223	323	426	538	323	472	612	744	866	383	517	653	788	921	1,054
Intake Airflow Rate (CFM)	1,292	1,171	1,050	938	1,516	1,442	1,394	1,335	1,278	1,700	1,649	1,627	1,589	1,549	1,508
Fan Intake CFM / W	3.42	3.09	2.78	2.51	3.52	3.35	3.23	3.10	2.97	2.92	2.84	2.80	2.73	2.66	2.58
Room Capacity (tons; 80°F reference)	1.02	0.88	0.68	0.45	1.14	1.04	0.84	0.66	0.46	1.28	1.15	1.04	0.88	0.71	0.52
Room EER (Btu/Wh, 80°F reference)	32.4	27.7	21.6	14.4	31.6	28.9	23.4	18.3	12.9	26.4	23.8	21.4	18.2	14.6	10.8
CA T-20 ECER (Btu/Wh) [2]	-	-	-	15.4	-	-	-	19.6	-	-	-	-	19.5	-	-
Sensible Cooling of Outside Air (tons)	2.84	2.32	1.74	1.13	3.16	2.70	2.17	1.66	1.16	3.52	3.10	2.69	2.25	1.78	1.29
Outside Air EER (Btu/Wh)	90.1	73.3	55.2	36.2	88.0	75.2	60.4	46.2	32.4	72.5	64.0	55.5	46.2	36.6	26.6

[1] Calculated interstage temperature is the intersection of the intake dew point and the supply wet bulb.

[2] ECER can only be derived when the supply resistance is 0.3" and the wet-bulb depression is >=25°F

Table 6: OASys™ Test Data (Continued)

Test Summary Information	Sensitivity to Intake Conditions - Supply Backpressure at 0.3 IW															
General	Low Fan Speed								Medium Fan Speed							
Date (2006)	19-Oct	23-Oct	23-Oct	19-Oct	19-Oct	24-Oct	19-Oct	24-Oct	19-Oct	23-Oct	23-Oct	19-Oct	19-Oct	24-Oct	19-Oct	24-Oct
Start Time	3:17p	11:00a	4:21p	11:10a	2:30p	10:23a	10:38a	3:00p	3:53p	1:03p	3:39p	11:31a	2:07p	10:54a	10:21a	2:04p
Duration (minutes)	15	15	10	15	15	15	10	15	15	10	10	15	15	10	10	15
Barometric Pressure (in. of Hg)	29.56	29.49	29.39	29.66	29.57	29.48	29.67	29.37	29.55	29.45	29.42	29.65	29.58	29.48	29.67	29.38
Inlet Air Properties																
Dry Bulb Temperature (°F)	**90.0**	**90.0**	**90.1**	**100.1**	**100.1**	**100.1**	**110.0**	**109.9**	**90.0**	**90.2**	**90.2**	**100.1**	**100.0**	**100.3**	**110.0**	**110.1**
Dew Point Temperature (°F)	50.4	60.5	69.4	40.7	54.1	63.7	45.6	59.6	49.5	58.5	68.8	40.2	53.1	65.1	46.5	58.9
Wet Bulb Temperature (°F)	**65.3**	**70.1**	**75.2**	**65.1**	**70.0**	**74.6**	**69.8**	**75.1**	**64.9**	**69.1**	**74.8**	**65.0**	**69.6**	**75.4**	**70.1**	**74.8**
Wet Bulb Depression (°F)	24.7	19.9	14.9	35.0	30.1	25.5	40.3	34.8	25.1	21.1	15.3	35.1	30.4	24.9	40.0	35.2
Relative Humidity (%)	25.9	37.3	50.8	13.1	21.7	30.7	11.8	19.8	25.0	34.6	49.6	12.9	21.1	32.0	12.2	19.2
Interstage Air Properties																
Measured Dry Bulb Temperature (°F)	68.4	72.0	75.9	70.0	73.1	75.6	76.5	76.8	68.6	70.5	75.4	69.3	74.0	77.2	77.1	77.2
Calculated Dry Bulb Temperature (°F) [1]	79.1	81.1	84.0	84.2	86.1	87.1	91.5	92.8	79.6	82.4	84.5	85.0	86.7	87.8	92.0	93.4
Supply Air Properties																
Dry Bulb Temperature (°F)	62.8	68.4	74.1	61.1	66.9	72.2	65.5	71.7	62.7	67.8	73.9	61.5	66.9	73.2	66.3	71.7
Dew Point Temperature (°F)	60.6	66.4	72.7	58.4	64.6	72.3	62.6	69.2	60.2	65.7	72.4	58.4	64.2	72.9	63.0	69.0
Wet Bulb Temperature (°F)	61.6	67.4	73.6	59.6	65.7	71.0	63.9	70.4	61.4	66.7	73.3	59.8	65.5	72.0	64.5	70.2
Relative Humidity (%)	94.0	95.4	97.7	92.2	94.0	94.5	92.2	94.1	92.9	94.7	97.3	90.9	93.0	94.4	90.9	93.0
External Resistance (IW)	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30
Exhaust Air Properties (Resistance, IW)	0.16	0.16	0.15	0.16	0.16	0.15	0.16	0.16	0.17	0.16	0.16	0.18	0.17	0.16	0.17	0.16
Dry Bulb Temperature (°F)	74.1	77.2	80.2	78.5	79.9	82.6	83.5	85.9	75.4	77.6	80.9	79.6	80.7	84.0	85.2	87.3
Dew Point Temperature (°F)	69.5	73.2	77.7	72.4	74.7	78.4	77.1	80.4	70.3	73.5	77.9	73.5	75.4	79.8	78.6	81.4
Wet Bulb Temperature (°F)	70.8	74.3	78.3	74.1	76.1	79.4	78.7	81.7	71.8	74.6	78.7	75.2	76.8	80.8	80.2	82.8
Relative Humidity (%)	85.5	87.5	91.9	81.8	84.2	87.4	81.2	83.9	84.1	87.3	90.8	81.7	83.9	87.4	80.7	82.8
Water Properties																
Basin Temperature (°F)	64.0	69.4	74.6	63.4	68.0	73.3	67.6	73.0	64.2	69.3	74.6	64.0	68.4	74.6	68.7	73.4
Makeup Water Flow (gph) - measured	3.2	2.3	2.5	3.4	3.0	2.8	4.9	4.5	3.7	3.3	2.1	5.4	4.2	4.1	5.5	5.4
Total Evaporation Rate (gph) - calculated	2.9	2.3	1.9	4.0	3.5	2.9	4.6	4.1	3.5	3.1	2.3	4.7	4.2	3.4	5.3	4.8
Fraction to Supply Air (%)	36%	36%	34%	41%	34%	32%	39%	33%	43%	42%	41%	46%	41%	37%	44%	38%
Power Consumption																
Voltage (V)	116	115	116	115	116	116	115	116	116	115	116	115	115	115	115	115
Current (A)	3.9	4.0	3.9	3.9	3.9	3.9	3.9	3.8	4.3	4.3	4.3	4.3	4.3	4.3	4.2	4.2
Power (W)	387	388	384	379	383	381	376	373	437	434	436	427	429	428	424	418
Power Factor	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.87	0.87	0.87	0.87	0.87	0.87	0.87	0.87
Fan Speed (RPM)	840	849	848	879	852	858	886	885	994	994	999	994	999	998	1,000	1,006
Performance																
Dry Bulb Temperature Drop (°F)	27.2	21.7	16.0	39.0	33.2	27.9	44.5	38.3	27.3	22.4	16.3	38.6	33.1	27.1	43.7	38.4
Dew Point Temperature Rise (°F)	10.2	5.9	3.3	17.7	10.5	8.6	17.1	9.7	10.7	7.2	3.6	18.1	11.1	7.8	16.5	10.1
Wet-Bulb Effectiveness (%)	110.2	108.7	107.4	111.5	110.3	109.3	110.6	109.9	108.7	106.2	106.1	110.0	108.9	108.6	109.5	108.9
Supply Airflow Rate (CFM)	548	559	549	616	544	552	599	564	764	763	758	814	754	750	795	756
Exhaust Airflow Rate (CFM)	464	465	470	394	474	469	416	477	464	475	479	401	482	484	414	477
Intake Airflow Rate (CFM)	1,006	1,019	1,016	1,002	1,011	1,015	1,005	1,032	1,221	1,232	1,233	1,206	1,227	1,228	1,199	1,223
Fan Intake CFM / W	2.60	2.63	2.64	2.64	2.64	2.67	2.68	2.77	2.80	2.84	2.83	2.82	2.86	2.87	2.83	2.93
Room Capacity (tons; 80°F reference)	0.81	0.56	0.28	0.99	0.60	0.36	0.73	0.39	1.14	0.80	0.40	1.28	0.84	0.43	0.91	0.52
Room EER (Btu/Wh, 80°F reference)	25.2	17.3	8.7	31.3	18.9	11.5	23.2	12.6	31.2	22.1	10.9	35.9	23.5	12.1	25.8	14.9
CA T-20 ECER (Btu/Wh) [2]	-	-	-	23.7	20.3	20.4	23.0	21.6	24.4	-	-	27.1	24.6	24.4	26.5	25.3
Sensible Cooling of Outside Air (tons)	1.29	1.04	0.75	2.04	1.53	1.30	2.23	1.79	1.79	1.47	1.06	2.67	2.12	1.72	2.91	2.40
Outside Air EER (Btu/Wh)	39.9	32.1	23.5	64.6	47.8	41.0	71.2	57.6	49.3	40.6	29.1	74.9	59.2	48.1	82.3	68.9

[1] Calculated interstage temperature is the intersection of the intake dew point and the supply wet bulb.

[2] ECER can only be derived when the supply resistance is 0.3" and the wet-bulb depression is >=25°F

Table 6: OASys™ Test Data (Continued)

Test Summary Information	Sensitivity to Intake Conditions - Supply Backpressure at 0.3 IW							
General	High Fan Speed							
Date (2006)	19-Oct	23-Oct	23-Oct	19-Oct	19-Oct	24-Oct	19-Oct	24-Oct
Start Time	4:24p	2:01p	3:16p	11:54a	1:39p	11:46a	10:03a	1:34p
Duration (minutes)	15	10	15	15	15	10	10	15
Barometric Pressure (in. of Hg)	29.54	29.42	29.40	29.65	29.60	29.48	29.68	29.39
Inlet Air Properties								
Dry Bulb Temperature (°F)	**90.0**	**89.9**	**90.0**	**100.0**	**100.0**	**99.9**	**110.0**	**110.0**
Dew Point Temperature (°F)	50.1	61.3	69.2	39.7	54.2	64.6	46.5	60.2
Wet Bulb Temperature (°F)	**65.2**	**70.4**	**75.0**	**64.8**	**70.0**	**75.0**	**70.1**	**75.4**
Wet Bulb Depression (°F)	24.8	19.4	15.0	35.2	30.0	24.9	40.0	34.6
Relative Humidity (%)	25.5	38.6	50.5	12.7	21.8	31.8	12.2	20.2
Interstage Air Properties								
Measured Dry Bulb Temperature (°F)	71.5	72.0	74.7	67.2	75.3	77.0	74.3	78.1
Calculated Dry Bulb Temperature (°F) [1]	80.2	83.0	85.0	85.6	88.0	88.8	92.4	94.2
Supply Air Properties								
Dry Bulb Temperature (°F)	63.4	69.5	74.3	61.9	67.9	73.3	66.8	72.7
Dew Point Temperature (°F)	60.6	67.4	72.7	58.2	65.1	74.6	63.0	69.8
Wet Bulb Temperature (°F)	61.9	68.4	73.7	59.8	66.3	72.0	64.6	71.1
Relative Humidity (%)	91.9	94.8	97.0	89.0	92.5	93.7	89.1	92.7
External Resistance (IW)	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30
Exhaust Air Properties (Resistance, IW)	0.19	0.17	0.17	0.19	0.18	0.17	0.19	0.18
Dry Bulb Temperature (°F)	76.4	78.8	81.4	80.7	82.0	84.8	86.3	88.9
Dew Point Temperature (°F)	71.4	75.2	78.5	74.5	76.6	80.4	79.7	82.9
Wet Bulb Temperature (°F)	72.8	76.1	79.2	76.1	78.0	81.4	81.2	84.2
Relative Humidity (%)	84.3	88.5	90.9	81.5	83.9	87.0	80.8	82.5
Water Properties								
Basin Temperature (°F)	64.8	71.0	75.3	64.8	69.3	75.0	69.5	74.6
Makeup Water Flow (gph) - measured	4.3	3.6	2.8	6.1	4.9	4.0	0.0	6.0
Total Evaporation Rate (gph) - calculated	4.0	3.3	2.6	5.5	4.9	4.0	6.1	5.4
Fraction to Supply Air (%)	47%	46%	45%	50%	45%	42%	47%	42%
Power Consumption								
Voltage (V)	115	116	115	115	115	115	115	115
Current (A)	6.4	6.5	6.4	6.4	6.4	6.4	6.3	6.3
Power (W)	583	592	586	579	579	578	573	569
Power Factor	0.80	0.79	0.79	0.79	0.79	0.79	0.79	0.79
Fan Speed (RPM)	1,120	1,123	1,122	1,121	1,122	1,122	1,122	1,123
Performance								
Dry Bulb Temperature Drop (°F)	26.6	20.4	15.7	38.1	32.2	26.5	43.2	37.3
Dew Point Temperature Rise (°F)	10.5	6.0	3.6	18.5	10.9	10.1	16.5	9.6
Wet-Bulb Effectiveness (%)	107.0	104.9	104.5	108.3	107.2	106.6	108.2	107.9
Supply Airflow Rate (CFM)	973	977	960	1,021	957	960	992	952
Exhaust Airflow Rate (CFM)	472	484	491	414	494	488	426	481
Intake Airflow Rate (CFM)	1,437	1,454	1,446	1,423	1,442	1,440	1,406	1,422
Fan Intake CFM / W	2.46	2.46	2.47	2.46	2.49	2.49	2.45	2.50
Room Capacity (tons; 80°F reference)	1.39	0.88	0.47	1.57	0.99	0.54	1.09	0.58
Room EER (Btu/Wh, 80°F reference)	28.6	17.9	9.6	32.6	20.4	11.3	22.9	12.2
CA T-20 ECER (Btu/Wh) [2]	22.6	-	-	24.4	22.5	22.4	23.9	23.0
Sensible Cooling of Outside Air (tons)	2.23	1.71	1.29	3.30	2.61	2.15	3.58	2.94
Outside Air EER (Btu/Wh)	45.8	34.6	26.4	68.5	54.2	44.8	75.1	62.1

[1] Calculated interstage temperature is the intersection of the intake dew point and the supply wet bulb.

[2] ECER can only be derived when the supply resistance is 0.3" and the wet-bulb depression is >=25°F