Performance Evaluation of an Evaporatively-Cooled Split-System Air Conditioner

ET 08.08 Report

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Design & Engineering Services
Customer Service Business Unit
Southern California Edison

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Disclaimer

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# Abbreviations and Acronyms

<table>
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<th>Abbreviation</th>
<th>Description</th>
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<tr>
<td>A/C</td>
<td>Air Conditioner</td>
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<tr>
<td>AHU</td>
<td>Air Handler Unit</td>
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<tr>
<td>AHRI</td>
<td>Air Conditioning, Heating and Refrigeration Institute (formerly ARI)</td>
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<td>ARI</td>
<td>Air Conditioning and Refrigeration Institute</td>
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<tr>
<td>ASHRAE</td>
<td>American Society of Heating, Refrigeration, and Air Conditioning Engineers</td>
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<tr>
<td>BTU</td>
<td>British Thermal Unit</td>
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<tr>
<td>CFM</td>
<td>Cubic Feet per Minute, (\text{ft}^3/\text{min})</td>
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<tr>
<td>CPU</td>
<td>Central Processing Unit</td>
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<tr>
<td>CTAC</td>
<td>Customer Technology Application Center</td>
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<tr>
<td>CTZ</td>
<td>California Thermal Zone (also: Climate Zone)</td>
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<tr>
<td>DAT</td>
<td>Discharge Air Temperature, °F</td>
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<tr>
<td>DB</td>
<td>Dry-bulb temperature, °F</td>
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<tr>
<td>DX</td>
<td>Direct Expansion</td>
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<td>ECCU</td>
<td>Evaporatively-cooled Condensing Unit</td>
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<tr>
<td>EER</td>
<td>Energy Efficiency Ratio</td>
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<tr>
<td>EMS</td>
<td>Energy Management System</td>
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<tr>
<td>fpm</td>
<td>Feet per minute</td>
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<td>HDAC</td>
<td>Hot and Dry (region) Air Conditioner</td>
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<tr>
<td>hg</td>
<td>Inches of Water Gauge</td>
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<tr>
<td>HP</td>
<td>Horse Power</td>
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<tr>
<td>HVAC</td>
<td>Heating, Ventilation, and Air Conditioning</td>
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<tr>
<td>NIST</td>
<td>National Institute of Standards and Technology</td>
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<tr>
<td>PSI</td>
<td>Pounds per Square Inch (gauge)</td>
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<tr>
<td>Abbreviation</td>
<td>Description</td>
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<td>--------------</td>
<td>-----------------------------------------------</td>
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<tr>
<td>PSIA</td>
<td>Pounds per Square Inch (Absolute)</td>
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<tr>
<td>RH</td>
<td>Relative Humidity, %Rh</td>
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<tr>
<td>RMS</td>
<td>Root Mean Square</td>
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<tr>
<td>RTD</td>
<td>Resistive Thermal Device</td>
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<tr>
<td>RTTC</td>
<td>Refrigeration and Thermal Test Center</td>
</tr>
<tr>
<td>SCE</td>
<td>Southern California Edison</td>
</tr>
<tr>
<td>SCFM</td>
<td>Standard Cubic Feet per Minute</td>
</tr>
<tr>
<td>SH</td>
<td>Superheat</td>
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<tr>
<td>SHR</td>
<td>Sensible Heat Ratio</td>
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<tr>
<td>TC</td>
<td>Thermocouple</td>
</tr>
<tr>
<td>TXV</td>
<td>Thermostatic Expansion Valve</td>
</tr>
<tr>
<td>TTC</td>
<td>Technology Test Centers</td>
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<tr>
<td>UA</td>
<td>Heat Transfer Coefficient</td>
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<tr>
<td>VSD</td>
<td>Variable Speed Drive</td>
</tr>
<tr>
<td>WB</td>
<td>Wet-bulb temperature, °F</td>
</tr>
<tr>
<td>WCEC</td>
<td>Western Cooling Efficiency Center</td>
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EXECUTIVE SUMMARY

It has long been known that evaporatively-cooled condensers in air-conditioning systems provide increased efficiency over air cooled condenser technology. The increased efficiency is especially effective during peak demand periods that correspond with the hottest part of the day. Some of the unknowns are the performance of these units over a wide range of ambient conditions, the comparative performance to air cooled units, and information about water consumption due to purging and evaporation.

The primary goal of this project is to evaluate the performance of an evaporatively-cooled condensing unit as part of a residential split-system air conditioner in some of the climate zone conditions found in Southern California Edison’s (SCEs) service territory. Additional goals are to determine the performance degradation of the unit with increasingly harsh climate conditions, look at general water consumption of the unit in these different climate conditions, and to compare normalized performance data to existing information on air-cooled condenser type air conditioning units from previous lab research conducted at SCEs Technology Test Centers (TTC).

The evaporatively-cooled condensing unit of the residential split-system was installed in a controlled environment room of the TTC and every major component was instrumented for data collection. Likewise, the indoor unit of the split-system was installed in a separate controlled environment room and every major component instrumented for data collection.

Baseline tests were then conducted where a minimum of two hours of test data was collected under established indoor and outdoor conditions. These respective temperatures and humidities derive from the rating conditions required by the Air-Conditioning, Heating & Refrigeration Institute's Standard 210/240-2003 for Unitary Air-Conditioning and Air-Source Heat Pump Equipment.

The results of the Technology Test Center’s baseline data were then compared to the results of the baseline tests previously conducted by the manufacturer at identical test conditions. The closely matching results of this comparison established confidence in the results of these independently conducted tests.

After the results were confirmed to be in close agreement, further tests were conducted at various outdoor climate zone conditions. Each different climate zone test was conducted under steady-state operation with the appropriate constant temperature and humidity in each of the indoor and outdoor test chambers.

Data was collected for a minimum of two hours under several outdoor conditions representative of climate zones within SCE service territory. Measured parameters were later compared for performance variations across the different climate zone conditions.

The findings show that evaporatively-cooled condenser technology is able to produce the same cooling capacity at lower energy consumption than current air-cooled condenser technology. The efficiency of the evaporatively-cooled condenser decreased slightly (approximately 10%) at the extreme climate zone conditions of high dry bulb and low wet bulb but was largely unchanged across less extreme climate zone conditions. By contrast, the efficiency of previously tested air cooled condensing units decreased severely (a drop of 34%) while increasing energy consumption at similar high dry bulb conditions.
As expected, the evaporatively-cooled condenser technology showed increased water consumption at hotter and dryer climate zone conditions. While the amount of water consumed was not excessive, it should be weighed against the electrical energy savings achieved in climate zone conditions where water consumption is an issue.

The manufacturer of the unit was able to down-size the compressor by taking advantage of the heat transfer effectiveness of water and the evaporative cooling process to lower the refrigerant saturated condensing temperature thereby increase the refrigeration effect. This smaller compressor consumes less power for the same cooling capacity as an air-cooled condensing unit with a larger compressor. The ability to obtain the same cooling capacity with a smaller compressor results in an overall energy savings.

The performance of this evaporatively-cooled condensing unit shows the potential for significant energy savings over similar air-cooled condensing technology of the same capacity. The total energy savings could be substantial given significant market penetration. This technology merits investigation into the creation of a customer rebate program that will encourage market penetration.

Water consumption, due to evaporation and purge cycles, may be an issue to consider in regions of water scarcity or in hot and dry climate zones where water consumption will naturally be higher.

For purposes of determining energy savings, the amount of energy saved should account for the amount of water consumed by evaporation and purge cycles, especially in regions where water usage is of concern. Whether to use an energy cost versus water cost basis or other basis is beyond the scope of this project.
INTRODUCTION

Southern California Edison’s (SCEs) significant peak electrical demand is a result of the vast number of air conditioners operating during the hot summer days. The efficiency of conventional air-cooled air conditioning systems equipped with air cooled condensers decreases as the outdoor temperature rises (see Reference 1). Because electricity cannot be economically stored on a large scale, new power plants must be constructed to meet the increasing peak demands of the state. A wide-scale reduction in energy consumption of air conditioners on hot days would dramatically reduce peak demand.

It has long been known that evaporatively-cooled condensers in air conditioning systems provide increased efficiency over air-cooled condenser technology. The increased efficiency is especially effective during peak demand periods that correspond with the hottest summer days.

A commercially available air conditioning system was developed using an evaporative-cooled condensing unit (ECCU), which operates at much higher efficiency in hot weather than conventional units with air cooled condensers.

Some of the unknowns are the performance of these units over a wide range of ambient conditions, the comparative performance to air-cooled units, and information about water consumption due to purging and evaporation.

If the manufacturer's projections are validated, this technology could provide peak demand reduction benefits to SCE and potentially provide significant long-term air conditioning energy savings to customers in the hotter climate zones where air conditioning energy savings are needed most.
BACKGROUND

A large portion of SCE’s peak power demand during the summertime is due to operation of air conditioning systems. Broad implementation of a more efficient air conditioning technology can reduce customer electrical energy usage and result in a significant reduction in utility system peak demand requirements.

An evaporatively-cooled condensing unit was developed for a residential split-system air conditioner in which water is sprayed on the condenser coils during operation to cool the refrigerant in the coils. This provides a condensing temperature much lower than a conventional air-cooled condenser system, resulting in greater operating efficiency and reduced electrical demand.

The manufacturer claims efficiency improvements of up to 40% over air-cooled condenser air conditioning (A/C) units on the hottest days in areas with high outdoor temperatures and low humidity.

Water is consumed by evaporation in the cooling section of the condenser and by a periodic flushing of the sump to avoid excessive mineral build-up in the cooling water. The water usage can be a concern as there are many desert climate zones within SCE service territory.

The evaporative condenser air conditioner system is currently in limited production. Investigating manufacturer performance claims and documenting water use characteristics can help overcome market barriers and provide a basis for more widespread acceptance of this emerging technology.
ASSESSMENT OBJECTIVES

The primary objectives of this assessment are to quantify the cooling capacity, energy efficiency and the water consumption characteristics of the evaporative condenser A/C system in different SCE service territory climate zones.

In order to meet these primary objectives, several intermediate steps have to first be satisfied. The overall sequence is detailed as follows:

1. Install, instrument and put into operation the unit to be tested.
2. Perform baseline ASHRAE/AHRI tests to quantify baseline performance.
3. Compare baseline performance to manufacturer’s baseline results for agreement/confidence of the independent results.
4. Perform ASHRAE/AHRI tests at pre-selected SCE service territory climate zone conditions.
5. Compare cooling capacity performance and energy efficiency of the unit across the climate zones tested.
6. Quantify water consumption of the unit across the climate zones tested.
7. Compare cooling capacity performance and energy efficiency to water consumption across the climate zones tested.

In the interest of reducing peak summer electrical demand and bringing more energy efficient technology into use through market penetration, it is expected that this technology may be considered a candidate for rebates or other incentive programs. To facilitate this expectation, secondary assessment objectives were pursued to provide information about how this technology compares to existing air conditioning technology.

Secondary assessment objectives:

1. Compare the cooling capacity performance of the unit to previously tested air-cooled condenser A/C units for different climate zones.
2. Compare energy efficiency of the unit to previously tested air-cooled condenser A/C units for different climate zones.

In conclusion, this project provides the assessments that are necessary to confirm the electrical energy savings and peak demand reduction potential to SCE as required for reducing future utility system peak power demand and long-term energy consumption. It also provides validation of manufacturer’s performance claims to help reduce market barriers that will ultimately result in a more widespread use of this promising energy efficient technology.
**PRODUCT EVALUATED**

The product evaluated was a commercially available residential split-system air conditioner with an evaporatively-cooled condenser. The unit tested was rated at a capacity of three tons (36,000 Btu/hr) and used the R-410a blend of environmentally friendly refrigerant. The indoor portion of the A/C system was a standard 3-ton 'A' coil evaporator controlled by a thermal expansion valve. The indoor blower fan was integral to the standard gas pack heating unit - no heating components were tested. The supply side of the indoor unit was fitted with a sufficient length of flexible duct to create the static pressure across the evaporator required for testing. The key importance here is that the indoor unit was the same as used in the manufacturer’s tests where results were later used for comparison with Technology Testing Centers (TTC) tests.

**AIR COOLED VERSUS EVAPORATIVELY-COOLED CONDENSERS**

Conventional air-cooled condensing systems use outdoor air to cool and condense the refrigerant gas in the condensing unit of the air conditioning system. The efficiency of an air-cooled system is related to the outdoor dry bulb air temperature at the condensing unit, and the efficiency decreases as this outside air temperature increases.

In the draw-through type of evaporatively-cooled condenser system, water is sprayed down onto the refrigerant lines while the counter flowing condenser inlet air passes up through the water covered tubes of refrigerant. Condensing of the refrigerant occurs as a result of a two-stage process of sensible and then latent heat transfer. First, the heat of the refrigerant is conducted to the surface of the tubes containing it where a thin film of sprayed water resides. The water conducts heat away from the tube surface which increases the caloric heat content of the water film. As the continuous flow of water sprays down on the refrigerant lines, the upward flowing condenser inlet air absorbs moisture from the water spray and becomes cooler.

Then, in the second stage of the condensing process, the cooled inlet air continues to absorb moisture as it evaporates the water film on the refrigerant tube surfaces. It is this latent mass transfer of moisture into the air (evaporation) that results in further cooling of the refrigerant lines as the latent heat of vaporization of the water is given up to the air that carries away the absorbed mass of water. This process also describes the water consumption as more water continually replenishes that which is evaporated.

The refrigerant condensing temperature is thus limited by the ambient air water content or wet bulb temperature (for any given dry bulb temperature), that is normally 14°F to 25°F lower than the condensing temperature limitation of ambient dry bulb for air cooled condensers (Reference 5). The lower condensing temperature allows for more efficient compressor operation and increased capacity. This characteristic promises a performance advantage over air-cooled systems in climate zones with high outdoor temperatures.

Achieving the benefit of increased efficiency using the evaporative condenser technology requires the availability of sufficient water to meet the consumption requirements of the system. Water is consumed by evaporation in the cooling section of the condenser and by a
periodic purging of the sump to avoid excessive mineral build-up in the cooling water. The purge is scheduled to take place once per hour of run time and releases about two gallons of water each time. The manufacturer claims the water consumption is offset, to a small extent, by a reduction in water used during electricity generation due to the reduced electricity use of the higher efficiency evaporative condenser unit. However, there is also a slight offset in energy savings due to the pumping energy required to deliver water to the evaporative-condenser unit.

Performance parameters including capacity, power demand, and EER were evaluated for various climate zones within the SCE service territory. Water consumption associated with the operation of the system for the various climate zones was also quantified.
**TECHNICAL APPROACH/TEST METHODOLOGY**

**TEST FACILITY**

The TTC is a testing facility of approximately 7,600 square-feet located in SCE’s Customer Technology Application Center (CTAC) complex in Irwindale, California. The TTC is comprised of lighting, refrigeration, and air-conditioning test centers. Figure 1 is a schematic depicting the layout of the Refrigeration and Thermal Test Center (RTTC) portion of the TTC. Within the RTTC, there are several test chambers:

- Supermarket test chamber
- Refrigerated walk-in test chambers
- HVAC test chambers

The supermarket test area is comprised of a controlled environment room equipped with an independent dehumidification, humidification, heating, and cooling system. Three refrigeration racks and a variety of heat rejection equipment can serve various display case systems that are housed in a controlled environment room.

The refrigerated walk-in testing areas consist of a cooler, a freezer and a loading dock chamber. A multiplex compressor rack system equipped with sophisticated controls and advanced features, serves the fan coil systems of these three chambers.

![Figure 1. Entire Test Facility Layout](image)

The HVAC testing area is comprised of both an indoor and an outdoor controlled environment chamber. Both chambers are served by their own individual heating, cooling, dehumidification, and humidification systems. Two ultrasonic humidifiers, controlled by the
facility’s sophisticated CPU-based Energy Management System (EMS), Figure 2 (b), inject precise moisture quantities into both chambers. A centrifugal humidification system augments the capacity of the ultrasonic units. Each room is served by an Air Handler Unit (AHU), equipped with split Direct Expansion (DX) coils, Variable Speed Drive (VSD) controlled variable air volume fans, variable supply air temperature control, and an electric heating system controlled by pulse modulation circulates air through supply and return air plenums, shown in Figure 3. A multiplex rack system, shown in Figure 2 (a), consisting of two 15 HP scroll compressors, equipped with VSDs, shown in Figure 4, and variable suction control serves the two AHUs.
The unit to be tested was installed in the test rooms in accordance with the manufacturer’s installation instructions. Air velocity in the vicinity of the installed unit was monitored closely and maintained below 500 feet per minute (fpm). Figure 5 (a) shows the outdoor ambient controlled environment room with the evaporative condensing unit. The outdoor controlled environment room’s 10-foot high ceiling provided sufficient clearance (more than six feet) from condenser discharge. A distance of at least three feet was provided between the test room’s walls and the equipment side surfaces. Figure 5 (b) shows the indoor unit in the indoor ambient controlled environment room with the supply duct connected from the outlet of the airflow measurement station.
**TEST METHODOLOGY**

**TEST PROTOCOL**

The test protocol used in this performance evaluation was ASHRAE 37-2005 *Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment* (Reference 6). Instrumentation procedures, test setup and measurement methodologies were derived from this protocol that were then applied to the relevant calculations used.

The test method used in this evaluation was the 2003 ARI Standard 210/240 *Unitary Air-Conditioning and Air-Source Heat Pump Equipment* (Reference 3). Test procedures, indoor and outdoor baseline climate conditions and tolerances were derived from this test method and applied to the test procedure.

**TEST METHOD**

Determination of the capacity and performance characteristics of the unit under different climate conditions followed methods of testing specified by ASHRAE Standard 37 -2005 and AHRI 210/240-2003 (formerly ARI). All tests were performed at steady-state conditions for a period of at least two hours. As prescribed by the test standards, air-side and refrigerant-side measurements are used to determine performance, particularly cooling capacity and Energy Efficiency Ratio (EER).

Typically, the performance of a unit under test is to be validated by both air-side and refrigerant-side enthalpy analyses. These independent determinations provide confidence in the test results under ASHRAE guidelines.

It should be noted however, that ASHRAE does not require two simultaneous test methods, only that when two methods are used, the total cooling capacity shall be the evaporator side capacity of two simultaneously conducted methods of test which shall agree to within 6% to establish confidence in the results of the tests.

The AHRI rating standard requires the unit under test be operated at the baseline conditions for a minimum of one hour of steady state performance to establish stability in data measurements and assure consistency of performance results. During this one hour minimum, the indoor climate conditions are to be maintained at 80°F DB and 67°F WB.

Table 1 identifies the conditions specified by AHRI 210/240 that were used in the testing of this unit.

<table>
<thead>
<tr>
<th>TEST (for Cooling Capacity)</th>
<th>INDOOR UNIT Air Entering Evaporator DB (°F)/ WB (°F)</th>
<th>OUTDOOR UNIT Air Entering Air Cooled Condenser DB (°F)/ WB (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard Rating Conditions &quot;A&quot; Cooling Steady State</td>
<td>80°F/67°F</td>
<td>95°F/75°F</td>
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**TABLE 1. OPERATING CONDITIONS FOR STANDARD RATING USING AHRI 210/240**
In addition to the baseline test condition requirements of the performance rating methodology, SCE has interest in the performance variations of the test unit across different climate conditions within SCE service territory. Climate zones are defined by ambient dry bulb and wet bulb temperatures specified for each zone.

Table 2 shows design conditions for several different climate zones within SCE service territory in which the performance of this test unit may be of interest. Representative cities and a description of the climate conditions were taken from the ASHRAE reference: Climatic Data For Region X (see Reference 4 of appendix). These values represent the 0.5% mean temperature conditions as given by the ASHRAE Region X (10) weather data (Reference 4).

The capacity, power demand, EER, and water consumption of the unit were determined through the TTC tests for baseline conditions as well as for climate zones 6, 7\(^1\), 8, 9, 10, 13, 14, and 15. An additional climate zone specified as Hot and Dry Air Conditioner (HDAC) was included because it was previously used to identify an extreme hot, dry climate condition that is somewhat more severe than Climate Zone 15. See Table 2.

<table>
<thead>
<tr>
<th>Climate Zone</th>
<th>Temperature</th>
<th>Representative City</th>
<th>Description</th>
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<tbody>
<tr>
<td>TTC Baseline</td>
<td>95°F 75°F</td>
<td>ASHRAE 37-2005</td>
<td>ASHRAE Baseline</td>
</tr>
<tr>
<td>CTZ 6</td>
<td>84°F 67°F</td>
<td>Los Angeles</td>
<td>South Coast</td>
</tr>
<tr>
<td>CTZ 7(^1)</td>
<td>83°F 69°F</td>
<td>San Diego(^1)</td>
<td>South Coast</td>
</tr>
<tr>
<td>CTZ 8</td>
<td>89°F 69°F</td>
<td>El Toro</td>
<td>South Coast</td>
</tr>
<tr>
<td>CTZ 9</td>
<td>94°F 68°F</td>
<td>Pasadena</td>
<td>South Coast</td>
</tr>
<tr>
<td>CTZ 10</td>
<td>100°F 69°F</td>
<td>Riverside</td>
<td>South Coast</td>
</tr>
<tr>
<td>CTZ 13</td>
<td>101°F 71°F</td>
<td>Fresno</td>
<td>Central Valley</td>
</tr>
<tr>
<td>CTZ 14</td>
<td>108°F 69°F</td>
<td>China Lake</td>
<td>Desert</td>
</tr>
<tr>
<td>CTZ 15</td>
<td>111°F 73°F</td>
<td>El Centro</td>
<td>Desert</td>
</tr>
<tr>
<td>CTZ HDAC</td>
<td>115°F 74°F</td>
<td>(Similar to) Needles @ 114/72</td>
<td>Desert</td>
</tr>
</tbody>
</table>

\(^1\) Climate Zone 7 is not in SCE service territory.
Figure 6 below shows the psychrometric plot of all the design condition climate zones tested as well as the baseline and the HDAC conditions.

**FIGURE 6. PSYCHROMETRIC PLOT OF ALL DESIGN CONDITION CLIMATE ZONES TESTED**

**INSTRUMENTATION AND DATA ACQUISITION**

This section addresses instrumentation requirements for temperature, pressure, electrical equipment, refrigerant flow, and condensate collection. With the objective of minimizing random and systematic uncertainties, careful attention was paid to the design of the data acquisition system. With this in mind, the following steps were taken:

- Use of sensors with very high accuracy
- Minimization of random errors by use of multiple sensors
- Use of calibration standard instruments of very high accuracy.

The test facility is equipped with a sophisticated computer-based data acquisition system. The National Instruments SCXI high-performance signal conditioning and instrumentation system for PC-based data acquisition and control was used to acquire and log test data. The data acquisition system was programmed to process and average 100 reads from 110 data channels every 20 seconds. This system was calibrated at the factory, and is traceable to the National Institute of Standards and Technology's (NIST) standards. The collected and stored data for each sensor was then checked for consistency and accuracy at the end of each test scenario. Consistently, the operating parameters were checked and deemed to be within acceptable limits before the next run began.
The Data Acquisition System consists of LabVIEW 8.6 software and SCXI hardware. The software includes a graphical data acquisition environment, and a data logging and supervisory control add-on, see Figure 7.

![Figure 7. Graphical Data Acquisition Environment](image)

The hardware includes three high-performance signal conditioning and switching platforms (SCXI-1000) with 4-slot chassis where each platform can house four modules, see Figure 8.

![Figure 8. Modules for SCXI High Performance Signal Conditioning and Switching Platform](image)

There are eight 32-channel SCXI-1100 analog input modules, and two 16-channel SCXI-1122 isolated sensor modules. In addition, a PCI-6052E 333kS/s, 16-bit, and 16-channel analog input card was used to convert analog data to digital, see Figure 9.

![Figure 9. PCI 6052E Analog Input Card](image)

The collected data points from the 20-second intervals were averaged into one-minute intervals and used for further screening of the test data. The advantage of using one-minute averages is that the data trends can still be displayed with an acceptable resolution while enabling the engineering model to generate relevant calculated hourly results such as cooling loads. The primary data points used for comparative analysis are based on refrigerant enthalpy results.

After the data was compiled into one-minute averages within the engineering model, tabular and graphical representations of various correlations and calculated parameters were produced. Several graphs were created to initially screen the calculated data. Various critical raw data was continuously screened for validation prior to importing the data into the TTC’s engineering model. After careful examination and upon validation of the initial screening plots, the informational plots were produced. This set of plots provided relationships...
between the calculated quantities. In cases where data flaws were detected, a series of diagnostic investigations were carried out, and through this process, corrections were made and tests were repeated.

Under steady-state test conditions, every 20 seconds, the data acquisition system sampled and averaged 100-points per channel of scanned data, which was then saved to a file at the end of each test. SCE engineers reviewed the initial data at the TTC to ensure that the control parameters were within range. In the event that any of the control parameters fell outside acceptable limits, the problem was flagged. In such cases, test runs were repeated until the problem was corrected.

**Temperature Measurements**

All temperature measurements followed ASHRAE Standard 41.1-1986.

**Thermocouples**

All thermocouples (TCs) were Type-T (copper-constantan) and had their junctions secured with soft solder and electrically isolated with heat shrink tubing. Upon application to the contact surface to be measured, the TCs were attached with thermally conductive paste for optimum heat transfer. Each TC was individually calibrated within 0.18°F accuracy, exceeding ASHRAE’s requirement of 0.2°F.

**Refrigerant Temperature Measurements**

Refrigerant vapor and liquid line temperatures were determined by attaching thermocouples on the suction, discharge, and liquid lines as described previously. For stability of readings, an eight inch length of the TC wire was wrapped around the location of pipe to be measured after the tip of the TC was properly affixed with thermal paste. Three separate layers of insulation and heat shield material were then wrapped around the TC element and pipe.

**Air Temperature & Humidity Measurements**

Air temperature measurements were taken within the 2‘x2‘ air duct at four locations throughout the air distribution ducts. For critical measurements, including air entering and leaving the evaporator, a temperature grid and a dew point sampling grid were constructed and mounted before and after each side of the ‘A’ shaped evaporator. The air inlet and outlet temperature grids each consisted of four TCs. The two air inlet dew point sampling grids were plumbed together and routed to a DewPrime® dew point temperature measurement instrument. Likewise, the two air outlet dew point sampling grids were plumbed and routed to a separate DewPrime instrument. All probe connections were insulated to minimize air leakage.

Similarly, for the air temperature at the inlet of the condenser unit, a grid of six TCs was installed on each of the three air inlet sides of the condenser. Each of these TC channels was recorded separately and their average was calculated as the average inlet air temperature to the condenser. A similar grid of 8 TCs was used at the outlet of the condenser to record the condenser outlet air temperature.
The inlet and outlet dew point temperatures of the condensing unit were each measured using a grid of three sampling points for inlet and outlet dew point temperature, respectively.

**Pressure Measurements**

The accuracy of all the pressure measuring instruments was ± 0.13% of full scale. The smallest scale division of the pressure sensors used for these tests never exceeded 2.5 times their specified accuracy. All duct static pressures were measured with manometers having an accuracy of ± 1% full scale. In each designated location, one side of the manometer was connected to four externally manifolded pressure taps in the supply duct. The other side of the manometer was connected to four externally manifolded pressure taps centered in each face of the return duct before inlet to the unit, see Figure 10.

To determine the external static pressure, ASHRAE 37 requires that pressure taps be centered in each discharge duct face at a distance of twice the square root of the cross sectional dimension from the test unit's outlet. Due to the space restrictions in the test facility, taps were installed at a distance shorter than those specified by the ASHRAE standard.

![Figure 10. Four Externally Manifolded Pressure Taps Before Inlet of the Test Unit](image)

**Electrical Measurements**

The total power supplied to the test unit was delivered through a STACO® voltage stabilization device that provides consistent stable voltage at 208 volts AC with an accuracy of ±1.2 volts.

To measure electrical component parameters, a power transducer was used to sample current and voltage for each electrical component of the system. The compressor, water circulation pump, condenser fan, water purge pump, indoor blower fan, entire indoor unit, and entire condensing unit were each measured separately. Each output of these transducers was recorded on a separate channel by the data acquisition system. Additionally, the total power, voltage, current, phase angle and frequency supplied to the unit were measured by an independent Nexus® power analysis unit. The output data from the Nexus was also recorded by the data acquisition system. The results of these two separate sets of data were compared to establish confidence in the readings by ensuring the sum of the individual
measurements was in close agreement with the totalizing unit. See Figure 11. All tests were performed at 208 volts and 60 hertz.

**Condensate and Purge Measurements**

A special piping assembly was constructed to transfer condensate from the evaporator pan to a separate container placed on a digital scale through a gravity fed system, see Figure 12. Water removed from the condenser cooling water sump during hourly purge cycles was also routed to the scale. Purge cycles were recognized in the data as step increases on the graph of water accumulated over time while the evaporator condensation rate was determined from the slope of the line of the rate of water accumulation over time. The collective weight of the condensate and the purge water accumulated was measured every 20 seconds by a 31,000 gram capacity digital scale with a standard deviation of 0.1 g, linearity of $\pm$ 0.2 g, and minimum deviation of 0.01%. The scale system automatically purged when necessary.
**Condenser Sump and Spray Water Measurements**

The average water temperature in the sump at the bottom of the condensing unit was measured at the inlet to the circulation pump using three TCs. The data recorded from each of these three channels was averaged together in the data analysis.

The temperature of the water spray onto the refrigerant condensing coils was measured using an array of eight TCs attached to the water distribution lines within the body of the condensing unit. The data recorded from each of these eight channels was also averaged together in the data analysis.

**Condenser Supply Water Measurements**

Cooling water input to the condenser was provided through a constant temperature and pressure supply line to the condenser water sump. Water was supplied to replace evaporation and to replace water removed during the hourly purge of the sump. The hourly purge is designed to prevent mineral buildup in the cooling water.

Water was supplied to the condenser sump by drawing, as needed, from a continuously circulating constant temperature water supply loop where the condenser inlet water was controlled to a constant 85°F by means of a heater and chiller arrangement, Figure 13 (a). Condenser inlet water temperature and pressure were also measured at the condenser inlet point of the circulation loop, Figure 13 (b). To measure total water consumption, a Coriolis mass flow meter was installed on the supply side of the constant temperature circulation loop as show in Figure 13 (c).

![Figure 13](image-url)
AIRFLOW MEASUREMENTS

To enable precise measurements of airflow rates for Heating, Ventilation, and Air Conditioning (HVAC) equipment, TTC staff designed and supervised construction of an ASHRAE airflow measurement station, as shown in Figure 14. The airflow station was built according to ASHRAE Standard 41.2-1987 and contains four calibrated flow nozzles with flow straighteners located both up- and down-stream of the nozzles. This measurement station was located in the indoor room and incorporated a supplemental blower fan with a variable frequency drive. Adjustments to this variable frequency drive fan allowed for a wide range of air velocity and static pressure conditions. Adjustments to the fan speed of the airflow station allowed matching the static pressure and cubit feet per minute (cfm) across the evaporator coil of the unit under test to the requirements of the ARI 210/240-2003 Standard.

FIGURE 14. ASHRAE AIR FLOW MEASUREMENT STATION
# Specifications of Sensors

The list of sensors used and their respective NIST traceable accuracies are shown in Table 3, below.

## Table 3. Specifications of Sensors Used

<table>
<thead>
<tr>
<th>Sensor Type</th>
<th>Make/Model</th>
<th>Accuracy [NIST Traceable]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature (TC)</td>
<td>Type T in Teflon Jacket</td>
<td>±0.1°C (±0.18°F)</td>
</tr>
<tr>
<td>Temperature (RTD)</td>
<td>Hy-Cal Engineering Model RTS-37-A-100</td>
<td>±0.06°C (±0.108°F)</td>
</tr>
<tr>
<td>Dew Point</td>
<td>EDGETECH Model 2000 Dew Prime DF Dew Point Hygrometer- S2 Sensor</td>
<td>±0.2°C (±0.36°F)</td>
</tr>
<tr>
<td>Relative Humidity</td>
<td>General Eastern Humiscan</td>
<td>±1.0%RH (for 0.5-90% RH range)</td>
</tr>
<tr>
<td>Pressure (static)</td>
<td>MAMAC Systems Model PR-274/275</td>
<td>±1.0%F.S.</td>
</tr>
<tr>
<td>Pressure (barometric)</td>
<td>PTB 100A Barometric Pressure Transducer</td>
<td>±0.3 hPa</td>
</tr>
<tr>
<td>Pressure (differential)</td>
<td>Dresser Industries Inc. ASHCROFT IxLdp</td>
<td>0.25% F.S.</td>
</tr>
<tr>
<td>Pressure</td>
<td>Setra Model 207, 100-500 psig pressure ranges</td>
<td>±0.13%F.S.</td>
</tr>
<tr>
<td>Power</td>
<td>Ohio Semitronics Model GW-5 E.I.L. AC Watt Transducer Electro Industries Nexus 1250</td>
<td>0.2% of reading 0.5% F.S. 0.06 of reading</td>
</tr>
<tr>
<td>Refrigerant Mass Flow</td>
<td>Endress &amp; Hauser Promass 80 M15 Coriolis Mass Flow Measuring System</td>
<td>±0.25% (gas) ±0.05% (liquid)</td>
</tr>
<tr>
<td>Condenser Cooling Water Mass Flow</td>
<td>Endress &amp; Hauser Promass 80 F08 Coriolis Mass Flow Measuring System</td>
<td>±0.25% (gas) ±0.05% (liquid)</td>
</tr>
<tr>
<td>Velometer</td>
<td>TSI Inc. Model #8455 Air Velocity Transducer</td>
<td>±0.5% F.S. ±2.0% of reading</td>
</tr>
<tr>
<td>Velometer</td>
<td>TSI Inc. Model #8475 Air Velocity Transducer</td>
<td>±1.0% F.S. ±3.0% of reading</td>
</tr>
<tr>
<td>Scale</td>
<td>HP-30K</td>
<td>±0.1 Gram</td>
</tr>
</tbody>
</table>
The minimum amount of data that is required by AHRI 210/240 to be recorded during the cooling capacity test, is shown in Table 4.

<table>
<thead>
<tr>
<th>ITEM</th>
<th>AIR ENTHALPY METHOD</th>
<th>REFRIGERANT ENTHALPY METHOD</th>
</tr>
</thead>
<tbody>
<tr>
<td>Barometric pressure (in. Hg)</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Power input to equipment (W or Wh)</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Applied voltage(s)</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Frequency (Hz)</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>External resistance to airflow (in. Hg)</td>
<td>X</td>
<td>-----</td>
</tr>
<tr>
<td>Fan speed, if adjustable (rpm)</td>
<td>X</td>
<td>-----</td>
</tr>
<tr>
<td>Dry-bulb temp of air entering equipment (°F)</td>
<td>X</td>
<td>-----</td>
</tr>
<tr>
<td>Wet-bulb temp of air entering equipment (°F)</td>
<td>X</td>
<td>-----</td>
</tr>
<tr>
<td>Dry-bulb temp of air leaving equipment (°F)</td>
<td>X</td>
<td>-----</td>
</tr>
<tr>
<td>Wet-bulb temp of air leaving equipment (°F)</td>
<td>X</td>
<td>-----</td>
</tr>
<tr>
<td>Condensing pressure or temp (psig/°F)</td>
<td>-----</td>
<td>X</td>
</tr>
<tr>
<td>Evaporator pressure or temp (psig/°F)</td>
<td>-----</td>
<td>X</td>
</tr>
<tr>
<td>Refrigerant-oil flow rate (ft³)</td>
<td>-----</td>
<td>X</td>
</tr>
<tr>
<td>Volume of refrigerant in refrigerant-oil mixture (ft³/ft³)</td>
<td>-----</td>
<td>X**</td>
</tr>
<tr>
<td>Refrigerant liquid temp, indoor side (°F)</td>
<td>-----</td>
<td>X</td>
</tr>
<tr>
<td>Item</td>
<td>Air Enthalpy Method</td>
<td>Refrigerant Enthalpy Method</td>
</tr>
<tr>
<td>-----------------------------------------------------------</td>
<td>---------------------</td>
<td>-----------------------------</td>
</tr>
<tr>
<td>Refrigerant liquid temp, outdoor side (°F) - Required only for line loss adjustment</td>
<td>-----</td>
<td>X</td>
</tr>
<tr>
<td>Refrigerant vapor temp, indoor side (°F)</td>
<td>-----</td>
<td>X</td>
</tr>
<tr>
<td>Refrigerant vapor temp, outdoor side (°F) - Required only for line loss adjustment</td>
<td>-----</td>
<td>X</td>
</tr>
<tr>
<td>Refrigerant vapor pressure, indoor side (psig)</td>
<td>-----</td>
<td>X</td>
</tr>
</tbody>
</table>

**Volume of refrigerant in refrigerant-oil mixture was not captured in this project**

The AHRI Standard requires the cooling capacity tests to yield the following results:

- Total cooling capacity, Btu/hr
- Sensible cooling capacity, Btu/hr
- Latent cooling capacity, Btu/hr
- Indoor side airflow rate, CFM standard air
- External resistance to indoor airflow, in-wg
- Total power input to equipment or all equipment components, watts
- Air temperatures:
  - Outdoor and indoor dry-bulb
  - Outdoor and indoor wet-bulb
- EER, Btu/hr/watts

Test results generated under this project, however, provided information beyond the AHRI requirements. The following lists additional results that were obtained.

- Coil sensible heat ratio
- Supply CFM/ton
- Coil superheat
- Condenser and total system sub-cooling
- Supply air temperature profile
- Refrigeration effect
- Condenser coil’s arithmetic mean temperature difference
- Evaporator and condenser coils heat transfer coefficient (UA)
- System power factor
- Work of compression
- Heat of compression
EVALUATIONS

APPROACH

The cooling capacity of the air-conditioning equipment was determined based on the ASHRAE 37 -2005 Standard. According to this standard, the cooling capacity can be determined based on indoor air-enthalpy and refrigerant enthalpy methods. The total cooling capacity of two simultaneously conducted methods of tests should agree within 6.0%. In addition to cooling capacity of the unit under consideration, the condenser performance, system EER, and water consumption were also evaluated.

The thermodynamic properties of air and airflow inside ducts were determined according to the ASHRAE Handbook of Fundamentals. Thermal Analysis Partners, LLC’s refrigerant property program, xProps® version 1.3, was used to determine refrigerant properties. The saturation temperatures were also determined using the xProps software.

COOLING CAPACITY

The gross cooling capacity is the rate of cooling or heat removal (in Btu/hr) that takes place at the evaporator coil of the unit. Cooling capacity was determined based on two methods; the air-enthalpy and refrigerant-enthalpy methods. In the air-enthalpy method, cooling capacity was determined based on properties of entering and leaving air, and the associated air mass flow rate. The cooling capacity using the refrigerant-enthalpy method was determined based on mass flow rate of refrigerant, and the refrigerant properties at the inlet and outlet of the evaporator coil.

According to AHRI 210/240, the cooling capacity rating should include the effects of blower fan heat, but not include supplementary heat.

AIR-ENTHALPY METHOD

The air-enthalpy method employs the measured psychrometric properties of air flowing across the evaporator coil of the unit. These measurements included dry-bulb and dew-point temperatures of the air up-stream and down-stream of the coil, volumetric airflow rates through the coil as well as static pressure drop across the coil. The enthalpy and mass flow rates of the air were then determined through calculation using these measured values. The resulting gross and net cooling capacities were then determined.

Prior to determining the cooling capacity of the test unit, the volumetric airflow in the duct was determined based on velocity pressure readings and air density. The equation for determining volumetric airflow from measured velocity pressure and obtained air density is shown in Equation 1.
EQUATION 1. VOLUMETRIC AIRFLOW RATE

\[
\text{cfm} = \left[ C \cdot \sqrt{\frac{2 \cdot P_v \cdot g_c}{\rho}} \right] \cdot A
\]

where

- \( \text{cfm} \) = volumetric airflow rate, \( \text{ft}^3/\text{min} \)
- \( C \) = unit conversion factor, (136.8)
- \( P_v \) = velocity pressure, in-wg
- \( g_c \) = gravitational constant, (32.174 \( \text{lb}_m\)-ft/lb\( r\)-s\(^2\))
- \( \rho \) = density of air, lb/ft\(^3\)
- \( A \) = duct cross sectional area, ft\(^2\)

The standard volumetric airflow rate was determined based on measured and standard air specific volume, which corresponds to about 60\(^\circ\)F at saturation and 69\(^\circ\)F dry air at 14.7 psia. The obtained volumetric airflow rate and the ratio of the measured and standard specific volume of air were used to obtain the standard volumetric airflow rate, Equation 2:

EQUATION 2. STANDARD VOLUMETRIC AIRFLOW RATE

\[
\text{SCFM} = \text{cfm} \cdot \left( \frac{\rho_{\text{std}}}{\rho_{\text{air}}} \right)
\]

where

- \( \text{SCFM} \) = standard volumetric airflow rate, \( \text{ft}^3/\text{min} \)
- \( \rho_{\text{std}} \) = specific volume of air at standard conditions, (13.33 \( \text{ft}^3/\text{lb} \))
- \( \rho_{\text{air}} \) = measured specific volume of air, \( \text{ft}^3/\text{lb} \)

After the standard volumetric airflow rate is determined, the air mass flow rate is determined by simply multiplying the volumetric airflow rate by the air density using Equation 3:

EQUATION 3. AIR MASS FLOW RATE

\[
\text{m}_{\text{air}} = \text{SCFM} \cdot \rho \cdot k
\]

where

- \( \text{m}_{\text{air}} \) = airflow rate inside the duct, lb/hr
- \( k \) = conversion factor, (60 min/hr)
- \( \text{SCFM} \) = standard volumetric airflow rate, \( \text{ft}^3/\text{min} \)
- \( \rho \) = density of air, lb/ft\(^3\)
The measured air properties, specifically dew point and dry-bulb temperatures, were used to determine air enthalpies at the inlet and outlet of the evaporator coil, based on the ASHRAE Handbook of Fundamentals. After the air enthalpies and airflow rate are determined, Equation 4 is used to obtain the gross cooling capacity.

**Equation 4. Gross Cooling Capacity - Air-Enthalpy**

\[
\dot{Q} = m_{\text{air}} \cdot (h_{\text{air - in}} - h_{\text{air - out}})
\]

where

- \( \dot{Q} \) = gross cooling capacity, Btu/hr
- \( h_{\text{air - in}} \) = entering air enthalpy, Btu/lb
- \( h_{\text{air - out}} \) = leaving air enthalpy, Btu/lb

It is sometimes useful to determine the cooling capacity in tons. Thus, the cooling capacity of the unitary air conditioning equipment was divided by 12,000, a conversion factor for Btu/hr to tons. See Equation 5.

**Equation 5. Converting Btu/hr to Tons**

\[
\dot{Q}_{(\text{tons})} = \frac{\dot{Q}}{12,000}
\]

where

- \( \dot{Q}_{(\text{tons})} \) = gross cooling capacity of air, tons

The net cooling capacity can be determined by simply subtracting the heat gain due to the evaporator fan from the gross cooling capacity, Equation 6. This methodology was used to exclude the heat input from the evaporator fan motor.

**Equation 6. Net Cooling Capacity**

\[
\dot{Q}_{\text{net}} = \dot{Q} - (kW_{\text{evap - fan}} \cdot k)
\]

where

- \( \dot{Q}_{\text{net}} \) = net cooling capacity of air, Btu/hr
- \( kW_{\text{evap - fan}} \) = evaporator fan power, kW
- \( k \) = conversion factor, (3,413 Btu/hr/kW)
Two methodologies were used to determine the latent indoor cooling capacity, psychrometric data and values taken directly from the condensate scale reading. The mass of collected condensate was determined psychrometrically by using airflow rate and absolute humidity of air at the inlet and outlet of the evaporator coil, Equation 7.

\[
\text{Equation 7. Mass of Collected Condensate - Psychrometric}
\]

\[
m_{wp} = m_{air} \times (\omega_{\text{air in}} - \omega_{\text{air out}})
\]

where

- \(m_{wp}\) = mass of collected condensate based on psychrometric data, lb/hr
- \(\omega_{\text{air in}}\) = absolute humidity of air at the evaporator inlet, lb\text{w}/lb\text{a}
- \(\omega_{\text{air out}}\) = absolute humidity of air at the evaporator outlet, lb\text{w}/lb\text{a}

After the condensate weight was determined, the latent cooling capacity was obtained by simply multiplying the condensate mass by the heat of vaporization of water. Equation 8 and Equation 9 show the latent cooling capacity calculations using psychrometric and scale reading methodologies, respectively.

\[
\text{Equation 8. Latent Cooling Capacity - Psychrometric}
\]

\[
Q_{lp} = m_{wp} \times h_{fg}
\]

where

- \(Q_{lp}\) = latent indoor cooling capacity using psychrometric data, Btu/hr
- \(h_{fg}\) = heat of vaporization of water, (1,060 Btu/lb)
- \(m_{wp}\) = mass of collected condensate based on psychrometric data, lb/hr

\[
\text{Equation 9. Latent Cooling Capacity - Condensate}
\]

\[
Q_{ls} = m_{ws} \times h_{fg}
\]

where

- \(Q_{ls}\) = latent indoor cooling capacity based on scale reading, Btu/hr
- \(m_{ws}\) = mass of collected condensate from scale reading, lb/hr
- \(h_{fg}\) = heat of vaporization of water, (1,060 Btu/lb)
After the gross cooling capacity and latent cooling capacity were determined, the sensible cooling capacity was obtained by using Equation 10.

**EQUATION 10. SENSIBLE COOLING CAPACITY**

\[ Q_s = Q - Q_{ls} \]

where

\[ Q_s \] = sensible indoor cooling capacity, Btu/hr

The standard volumetric airflow rate per gross cooling capacity of the air-conditioning equipment was obtained using Equation 11.

**EQUATION 11. VOLUMETRIC FLOW RATE PER GROSS COOLING CAPACITY**

\[ \frac{SCFM}{Ton} = \frac{SCFM}{Q_{(tons)}} \]

where

\[ SCFM_{Ton} \] = Standard volumetric airflow rate per ton of cooling capacity, ft3/min/tons

The EER of the unit depends on the total power usage, as well as the net cooling capacity of the unit. The total power usage includes compressor, condenser fan, and evaporator fan. The EER of the unit can be determined by simply dividing the net cooling capacity by the measured total input power to the unit, Equation 12.

**EQUATION 12. EER – ENERGY EFFICIENCY RATIO**

\[ EER = \frac{Q_{net}}{W_{total}} \]

where

\[ EER \] = energy efficiency ratio of the unit, Btu/hr/watts

\[ W_{total} \] = measured total input power to the unit, watts

The sensible heat ratio (SHR) was determined using Equation 13. It is used to compare the amount of sensible cooling being done on the air to the gross cooling capacity, which includes the amount of moisture removal.
### Equation 13. Sensible Heat Ratio

\[
\text{SHR} = \frac{\dot{Q}_s}{Q}
\]

where

\(\text{SHR}\) = sensible heat ratio, unit-less

---

**Refrigerant-Enthalpy Method**

Pressures and temperatures were measured across all components of the refrigeration system to verify the enthalpies at all points in the refrigeration cycle. These measurements were taken in the refrigerant lines approximately 10 inches from the relevant components such as compressor, condenser coil, and evaporator. A Coriolis mass flow meter was installed in the liquid line at the evaporator unit, Figure 15. This assembly was positioned upstream of the refrigerant metering device. Pressure drop across the flow meter was closely monitored so that liquid refrigerant did not flash and undergo a temperature drop larger than 3°F.

![Coriolis Refrigerant Mass Flow Meter](image)

**Figure 15. Coriolis Refrigerant Mass Flow Meter**

Additionally, temperature and pressure sensors as well as a sight glass were installed immediately downstream of the mass flow meter. This provided confirmation that the total sub-cooling of liquid refrigerant had not exceeded 3°F (exiting the flow meter) and that no vapor bubbles passed through the flow meter.

The refrigerant enthalpy method is generally an easier and more reliable method for determining cooling capacity than the air-enthalpy method as there are fewer properties to be measured and fewer components in the cooling capacity equation. Pressures, temperatures, and the mass flow of refrigerant can be measured directly and the behavior of refrigerant is generally more consistent and stable over time compared to properties of air. As an example, opening the door to the indoor test chamber for a short period during a test would affect the air-side properties much more than the refrigerant-side properties.
The refrigerant enthalpy method requires refrigerant temperature and pressure measurements entering and leaving the evaporator coil as well as the mass flow rate of the refrigerant through the system. A change in enthalpy across the evaporator is determined from the refrigerant pressure and temperature measurements at the inlet to the evaporator (before the expansion device) minus the corresponding enthalpy due to the pressure and temperature of the refrigerant exiting the evaporator. This difference is referred to as the refrigeration effect (Equation 14 below) and is the quantity of heat that each unit of mass of refrigerant absorbs to cool the indoor space. It simply represents the capacity of the evaporator per pound of refrigerant. The xProps program was used to determine refrigerant vapor and liquid refrigerant enthalpies from the temperature and pressure readings.

**Equation 14. Refrigeration Effect**

\[ RE = h_{\text{refrig - out}} - h_{\text{refrig - in}} \]

where

- \( RE \) = refrigeration effect, Btu/lb
- \( h_{\text{refrig - out}} \) = superheated refrigerant enthalpy at the evaporator outlet, Btu/lb
- \( h_{\text{refrig - in}} \) = subcooled liquid refrigerant enthalpy at the expansion device inlet, Btu/lb

The gross cooling capacity of the unit was determined simply by multiplying the mass flow rate of refrigerant by the refrigeration effect, Equation 15.

**Equation 15. Gross Cooling Capacity - Refrigerant-Enthalpy**

\[ Q = m_{\text{refrig}} \times RE \]

where

- \( Q \) = gross cooling capacity of refrigerant, Btu/hr
- \( m_{\text{refrig}} \) = mass flow rate of refrigerant, lb/hr
- \( RE \) = refrigeration effect, Btu/lb

As was the case with the air-enthalpy gross cooling capacity, Equation 5 is used to convert the refrigerant-enthalpy gross cooling capacity to tons of cooling. The net cooling capacity can be determined by subtracting the heat gain in Btu/hr due to the evaporator fan motor from the gross cooling capacity, as shown in Equation 6.

Equation 9 was used to determine the latent cooling capacity. With the latent and net cooling capacities calculated the sensible cooling capacity can be calculated (Equation 10).

The standard volumetric airflow rate per gross cooling capacity of refrigerant was obtained using Equation 11.
The same equations were used to calculate the EER and SHR using the refrigerant-enthalpy data as with the air-enthalpy. These are Equation 12 and Equation 13, respectively.

The compressor efficiency was determined based on the gross cooling capacity and the compressor input power, Equation 16. The compressor efficiency represents the gross cooling capacity per power input to the compressor.

\[
\text{COMPRESSOR EFFICIENCY}
\]

\[
\text{Eff} = \frac{\dot{Q}}{W_{\text{comp}}}
\]

where

- \( \text{Eff} \) = compressor efficiency in terms of cooling capacity per input power, Btu/hr/watts
- \( W_{\text{comp}} \) = measured total input power to the compressor, watts

Additional supplemental performance evaluation information pertaining to evaporator coil characteristics, condenser coil characteristics and related parameters is presented in Appendix A.
RESULTS

CAPACITY AND PERFORMANCE OF THE UNIT

The first test at the TTC of the evaporative condensing A/C unit system was conducted to verify the manufacturer’s stated performance of the air conditioner and to develop a baseline for comparing the performance of the unit in different SCE climate zones. The TTC tests showed nearly identical capacity to the manufacturer’s test data but at a 13% greater power demand resulting in a 12% reduction in EER compared to the manufacturer’s test data, shown in Figure 16.

![Figure 16. Performance Comparison of Manufacturer’s Test Data to TTC’s Test Data at AHRI 210/240 Baseline Conditions]

One of the reasons for the difference in power is that the manufacturer’s tests did not actually measure indoor blower fan wattage but used the ASHRAE indoor blower fan default value of 438 watts for their calculations. The SCE tests however, used the actual measured values of indoor blower fan wattage in determining total unit power. These measured values were consistently higher than the ASHRAE default value but the difference does not entirely account for the difference in power between the two independent tests.

The methodology of each party’s tests was based on the AHRI 210/240 baseline conditions of 95°F dry bulb and 75°F wet bulb using the refrigerant enthalpy method for determining performance capacity of the unit. This is the more reliable and preferred method of performance verification as there are fewer measured data values and thus fewer possibilities for introducing error to the results compared with the air enthalpy method. Additionally, attaining closely matched results using the same methodology in analyzing the independently conducted baseline tests provides increased confidence in the results.
The TTC baseline and manufacturer’s baseline tests were determined to be in agreement based on the refrigerant enthalpy test methodology used and the relatively close values attained by each testing group. Therefore, for the purpose of comparison, the subsequent tests across different climate zones were conducted.

As mentioned previously in the Test Methodology section, ASHRAE requires that when two simultaneous test methods are employed that the evaporator side cooling capacity of the two simultaneous methods match within 6%. However, the air-enthalpy and refrigerant enthalpy methods for the TTC tests did not match to within 6.0% as required when two simultaneous methods are used. The reason for the greater than 6.0% difference between air- and refrigerant enthalpy method results was suspected to be related to errors in the measurement instruments or incorrect calibrations of the air-side instruments.

Investigation of these presumptions later revealed a problem with the cold-junction calibrations of the TCs in the old data acquisition chassis compared to those of the new chassis added shortly before the beginning of this project. However, the close agreement of the independently conducted baseline tests at identical test conditions provides confidence in the TTC varying climate zone test results based on the analysis using only the refrigerant-enthalpy method.

The results of the different climate zone tests are shown in Figure 17 where each climate zone is specified by a dry bulb (DB) temperature and a wet bulb (WB) temperature to define a state point for that CTZ condition. The Capacity, Power, and EER were fairly uniform over Climate Zones 6, 7, 8, 9, and 10, with small declines (up to a maximum of 10%) in EER for Climate Zones 13, 14, 15, and the HDAC condition. Capacity was maintained within 2% of the CTZ 6 value for Climate Zones 6, 7, 8, 9, and 10, with a maximum reduction of 5.7% for the HDAC condition.

![Figure 17. Test Results of Evaporative Condenser A/C Unit in Various Outdoor Climate Zones](image-url)
These results show that the test unit experienced a relatively small degradation in performance in the hotter climate zones. For the HDAC zone, the hottest condition evaluated, the unit is able to deliver 94.3% of capacity and 90% of the EER it achieved in Climate Zone 6, one of the cooler zones evaluated.

A summary table of the performance parameters across the different climate zones tested is listed in Appendix B at the end of this report. To put these results in perspective, they were compared to similar tests performed at the TTC in 2004 on air conditioners with conventional air-cooled condensers. This comparison is developed in a subsequent section of this report.

**WATER CONSUMPTION**

Total water consumption of the evaporatively-cooled condensing unit was due to the amount of water consumed through evaporation and the amount of water purged during the purge cycles. Each of these is explained in more detail below.

**PURGE**

Generally, supply water to an evaporative cooler inherently contains dissolved calcium, lime and other minerals. During the evaporation process, only the water component evaporates leaving an increased concentration of minerals in the sump water as the supply continuously replenishes the water that is evaporated. The purge activity is a method of eliminating the water that becomes concentrated with minerals that might otherwise build up as scale and subsequently reduce the effectiveness of heat transfer surfaces.

The interval of the ECCU purge cycle may be manually adjusted inside the unit to accommodate different water qualities depending on the quality of the feed water for any particular installation. The purge interval for all tests conducted in this evaluation was set to one hour. When the water quality contains a high concentration of dissolved minerals, it may be desirable to purge the water more frequently to minimize scale buildup on internal heat transfer surfaces and pump components.

The amount of water consumed by the purge activity is independent of climate zone test conditions. The measured purge amount is a function of the purge frequency interval (one hour), the pump on-time duration and the purge pump flow rate. The amount of water purged at each purge interval is relatively constant at approximately 1.8 gallons per one hour flush interval (see Figure 18 below).

Climate zone 7 test conditions had the lowest purge water amount at 1.6 gal/hr while the highest purge water amounts were at CTZ 10 and CTZ 13 each with 2.1 gal/hr. This small variation from lowest to highest of approximately 0.5 gal/hr is likely related to variations in purge pump operating parameters such as flow rate or time to establish flow, such as in priming the pump.

**EVAPORATION**

The remainder and larger component of the water consumption is due to evaporation during the cooling process and is directly related to the climate zone conditions as explained below.

Water consumption of the ECCU due to evaporation tends to increase with hotter and dryer climate zones. The increase is approximately 27% from the lowest usage of 5
gal/hr for the mild condition of climate zone 7 to the highest usage of almost 7 gal/hr for the extreme conditions of climate zones 14, 15, and the HDAC condition (see Figure 18).

Total water usage for evaporation and the hourly purge of water in the sump over all climate zones was in the 6.5 to 8.5 gal/hr range during continuous operation. This is equivalent to about 2.1 to 2.8 gal/hr per ton of air conditioning capacity for this 3-ton unit.

![Figure 18. Water Consumed by Purge and Evaporation at Various Climate Zone Conditions](image)

The amount of water consumed due to evaporation in the process of cooling the refrigerant is related to the climate zone conditions in which the unit operates. Specifically, the hotter and dryer the ECCU inlet air, the more water is consumed by the evaporative cooling process. Any given climate zone condition can be specified by a dry bulb (DB) temperature and a wet bulb (WB) temperature to define a state point for that CTZ condition. The water content of such a state point is referenced by the humidity ratio ($\omega$).

Figure 19 shows a psychrometric chart plot of the ECCU inlet and outlet air state points under the relatively mild conditions of climate zone 7. Though this particular climate condition is not in SCE service territory, it is useful to compare with more extreme climate zone conditions. The vertical graduations are dry bulb temperatures, the horizontal graduations are humidity ratios, and the diagonal lines represent the enthalpy or total heat content of the air at any given state point. The enthalpy at any given state point is described in the ASHRAE Fundamentals Handbook by the following relationship:
As the condenser inlet air passes through the ECCU, it absorbs moisture in the evaporation process until the air becomes saturated with water vapor. This is shown in Figure 19 as the Condenser Outlet state point on the saturation curve of the psychrometric chart plot.

The difference or change in humidity ratio (Δω) of the condenser outlet to condenser inlet air state points represents the amount of water absorbed by the air as it passes through the condensing unit (again, Figure 19). This is the water consumption due to evaporation. It is interesting to note that for this climate zone, the Δω is nearly proportional to the change in enthalpy (Δh) or heat content of the air after passing through the ECCU. The increase in the air enthalpy comes from the heat of the refrigerant as it is transferred to the water film surface through conduction and then to the air through the latent process of evaporation.

Figure 19 shows a small drop (~1.5°F) in dry bulb temperature along the x-axis and an increase in humidity ratio along the right y-axis as the state point of the air moves from the Condenser Inlet condition to the Condenser Outlet condition.
condenser inlet air has a much greater ability to absorb moisture compared to the milder CTZ 7 condenser inlet air conditions. In this extreme climate condition, the x-axis of Figure 20 also shows a large drop in the dry bulb temperature of the condenser outlet air. This indicates that some of the water consumed during evaporation in this extreme climate condition is inadvertently used to cool the condenser outlet air in addition to that used in cooling the refrigerant. This is inherent to the use of the evaporation process in a hot and dry climate zone.

The greater $\Delta \omega$ of the extreme climate zone conditions results in a higher water consumption at the higher dry bulb temperature CTZs. It is interesting to note that the overall change in the enthalpy ($\Delta h$) of the air is approximately the same as that for the mild climate zone condition, CTZ 7 ($\Delta h_{CTZ\ 7} = 12.3$ versus $\Delta h_{CTZ\ HDAC} = 11.9$). This is due to evaporation of the water contributing to the large drop in air dry bulb temperature at the discharge of the ECCU during operation in hot and dry climate zone conditions.

Figure 21, below, shows the increasing total water consumption rate as a function of the increasing change in humidity ratio ($\Delta \omega$) for the outdoor climate zone conditions. Specifically, this change in humidity ratio ($\Delta \omega$) is the difference between what the humidity ratio is at saturation and the actual humidity ratio for the specific dry bulb of each climate zone condition. Thus, the dryer the entering condenser air, the greater is the $\Delta \omega$ or the ability of that air to absorb moisture, therefore the higher the water consumption due to evaporation. The data shows that CTZ 7 has the lowest evaporation rate that corresponds to the smallest change in humidity ratio ($\Delta \omega$) or ability to absorb moisture. Likewise, the CTZ HDAC conditions have the highest ability to absorb moisture ($\Delta \omega$) and the greatest water consumption due to evaporation.
Figure 21 charts the total water consumption rate. The rate due to evaporation only is the difference between the total water consumption and the purge water consumption, or approximately 2 gal/hr/ton less at each data point.

\[ y = 37.618x + 6.3742 \]

\[ R^2 = 0.9509 \]

As relative humidity is a more general reference to the moisture content of the air for any given dry bulb temperature, we would expect to see a correlation similar to that of the change in humidity ratio (\(\Delta \omega\)). Figure 22 shows reference to the more common metric of relative humidity. Though the correlation is not as precise as to the change in humidity ratio (\(\Delta \omega\)), it may be easier to use as a reference. This graph shows total water consumed in gallons per hour and references Table 4 of dry bulb and wet bulb conditions for the different climate zones tested.

Water consumption in hot and dry areas can become an issue, but the test data provides an accurate basis for estimating water consumption needs in various climate zones corresponding to future installation projections for the units. The trend line of Figure 22 indicates an increase of approximately 0.5 gal/hr total water consumption per ton of net cooling capacity for each 10% decrease in relative humidity.
**FIGURE 22. TOTAL WATER CONSUMPTION AS A FUNCTION OF RELATIVE HUMIDITY**

**PERFORMANCE AND WATER CONSUMPTION**

Figure 23 shows a comparison of net cooling capacity, EER and water consumption across the different climate zone conditions tested. Notice the general trend of increased water consumption as the hot and dry climate conditions become more extreme. Also, the increasing hot and dry conditions reflect a slight but general trend in decreased cooling capacity performance.

\[ y = -0.0468x + 8.9062 \quad R^2 = 0.9208 \]
Overall, it can be concluded that the evaporatively-cooled condenser system offers the potential of more consistent performance in hot climate zones compared to conventional air-cooled condenser systems that experience substantial performance and efficiency degradation when outdoor temperatures are high.

**Performance Comparison To Air-Cooled Air Conditioning Systems**

In 2004, tests were conducted at the TTC to evaluate the performance of six conventional air cooled air conditioning systems at high ambient temperatures (Reference 1). The 5-ton rooftop package units evaluated included standard and high-efficiency models from three different manufacturers. The performance of each unit was evaluated by conducting controlled environment tests of varying outdoor dry bulb while maintaining constant indoor conditions (temperature and humidity). The outdoor dry bulb temperature was incrementally increased in several stages from 85°F to 130°F to capture the performance parameters at each stage. The performance parameters evaluated included capacity, power demand, and EER.

For the purpose of a general comparison to the ECCU over the 85°F to 115°F dry bulb temperature range, the results of all six air-cooled units (standard and high-efficiency) are averaged together and presented in Figure 24 below. These results were taken from Reference 1 and re-formatted for presentation here. Over the tested temperature range of 85°F to 115°F, the average performance data of the six air cooled units showed a 13% decrease in net cooling capacity with a simultaneous 26% increase in power (kW) consumption with increasing dry bulb temperature. The resulting Energy Efficiency Ratio (EER) decreased by 34%.
It should be noted that the tests on the air-cooled units were conducted over an outdoor temperature range of 85°F to 130°F under no specified wet bulb temperature as wet bulb does not affect the performance of an air-cooled condenser unit. The results of these air cooled units were determined as a function of outdoor dry bulb temperature only. The tests on the evaporatively-cooled unit however, were performed based on a series of climate zone conditions defined by specified dry bulb and wet bulb temperatures as the evaporative cooling process and thus the evaporative condenser performance depend on the wet bulb temperature. The results of the tests on the ECCU were determined as a function of both dry bulb and wet bulb.

It should also be noted that the air-cooled units referenced in Figure 24 were 5-ton roof-top package units while the ECCU tested was a 3-ton residential split-system. All general performance comparisons, therefore, were normalized for a ‘per-ton of net cooling capacity’ comparison between the two unit types.

**NET COOLING CAPACITY**

In lieu of a direct comparison between the two unit types, a general comparison can be made by examining the performance at several outdoor dry bulb temperatures. Figure 25 shows, for example, CTZ 6 can be represented by a dry bulb temperature of 84°F, and the HDAC zone by 115°F. Comparing these points with the air-cooled condenser test results for an 85°F to 115°F range, it can be seen that the capacity of the ECCU is reduced by 5.7% while the capacity for the average of the air-cooled units is reduced by 13% over this range (see Figure 25).
It is interesting to note the air cooled units have a high net cooling capacity per ton at low dry bulb temperature but a low net cooling capacity per ton at the higher dry bulb temperatures. This is because the condensing temperature of air-cooled units is directly dependent on the outdoor ambient dry bulb temperature. As the outdoor temperature gets cooler, the condenser is able to more efficiently reject heat to the outside air and the cooling capacity of the unit increases. The ECCU condensing temperature, however, is driven by the moisture content of the condenser inlet air as previously discussed and so does not show as significant a change in cooling capacity with lower dry bulb temperatures.

At the AHRI 210/240 Standard Rating Condition of 95°F for outdoor air-cooled units, the net cooling capacity is slightly higher than 12,000 Btu/hr/ton. This is an indication that the units are designed to be slightly oversized or over rated at low dry bulb temperatures in order for the performance to be acceptable at higher dry bulb temperatures. This over sizing is, apparently, necessary because the performance decreases significantly with increasing dry bulb temperatures.

It is also interesting to note that the ECCU is slightly under sized at the same AHRI Standard Rating Condition of 95°F. In fact, the net cooling capacity per ton is mostly below 12,000 Btu/hr/ton over the same temperature range. The ECCU however, does not show as significant a loss of net cooling capacity as the air-cooled unit over the same temperature range. This is indicated by the slope of their respective trend lines.
The average net cooling capacity of the air-cooled units decreased 13% or 54.4 Btu/hr/ton per °F increase over this temperature range. Compare this to the less severe decrease in net cooling capacity of only 5.7% or 22 Btu/hr/ton per degree °F increase for the ECCU over the same dry bulb temperature range. This is an absolute drop in net cooling capacity of 1,632 Btu/hr/ton for the air-cooled units and only 690 Btu/hr/ton for the ECCU.

Figure 25 indicates that at approximately 101°F the net cooling capacity per ton of the ECCU becomes greater than that of the air-cooled units. Also, neither the air-cooled units nor the ECCU unit was able to produce their rated cooling capacity at the extreme temperature of 115°F; though the net cooling capacity per ton of the ECCU was greater than that of the air cooled units at this temperature.

**POWER**

Perhaps one of the strongest advantages to the evaporative condenser system is the relatively flat or steady power consumption across a wide range of outdoor dry bulb temperatures (see Figure 26). This is where the performance of air-cooled condenser systems is greatly compromised. As outlined previously, when temperatures get hotter the cooling capacity of air cooled units decreases while they consume even more power to try to satisfy the need for cooling. This is an inherent problem with air cooled condenser technology as the compressor has to work harder to reject the heat of the refrigerant to the hotter outside air. The result is evident as Figure 26 shows the increase in power consumption with increasing outdoor temperatures for the air-cooled condenser units versus the relatively unchanged power consumption for the ECCU.

Additionally, at the low dry bulb temperatures where the air-cooled units perform best, the power demand per ton of cooling capacity for the ECCU was 0.3 kW/ton more efficient. This is indicated by the upward offset in the power demand curve for the average of the air-cooled units.

Another interesting point to note is that for the same cooling capacity per ton at approximately 101°F dry bulb (refer to Figure 25), the trend line of the power consumed per ton at 101°F (Figure 26) indicates the air-cooled unit consumes significantly more power than the ECCU for the same output cooling capacity. This implies for this dry bulb, that even when the net cooling capacities per ton are the same, the air-cooled unit requires 0.379 kW/ton more power to produce that cooling capacity than does the ECCU.
The change in power demand shown for the air-cooled units as temperature increased was 27% or 10 watts/ton per °F over the range of temperatures tested. The change in power demand for the ECCU unit over the same temperature range was nearly zero at less than 0.1% or less than 1 Watt/ton per °F. This is an approximation based on the trend line of the experimental data. This result closely matches the expected trend for power demand because the saturated condensing temperature and thus the condenser discharge pressure are dependent on the moisture content of the condenser inlet air. This means for a given wet bulb, as the dry bulb temperature increases the amount of water evaporated within the unit increases because more water can be evaporated from the air and the thin water film covering the refrigerant tubes.

As confirmation of this, Figure 27 shows the ECCU power consumption as a function of increasing ambient wet bulb temperature changes only slightly as the wet bulb temperature increases.
Similarly, Figure 28 shows that the ECCU discharge pressure that caused the change in power consumption also follows the same trend of a slight increase with increasing wet bulb temperature.
Figure 29 shows that the ECCU saturated refrigerant condensing temperature that caused the change in discharge pressure is also seen to follow the same slightly increasing trend with increasing wet bulb temperature. This average saturated condensing temperature of 84°F is determined by the average 84°F wet bulb temperature of the water on the refrigerant tubes.

\[
y = 0.6776x + 81.789 \\
R^2 = 0.83
\]

The trend of Figure 29 back to Figure 27 indicates that as the inlet condenser air is more humid, there is less capacity of that air to perform the mass transfer process of evaporating the water on the condenser refrigerant tubes so the compressor has to work harder (more power) to reject the heat of the refrigerant in the condensing unit. However, the ECCU does not severely increase power consumption with increasing wet bulb temperatures nor with increasing dry bulb temperatures as does air-cooled condenser technology.

**Energy Efficiency Ratio (EER)**

The EER is defined as the ratio of net cooling capacity over the power required to produce that cooling capacity. This ratio is defined at the AHRI 210/240 rating temperature of 95°F DB for air cooled condensers and 95°F DB at 75°F WB for evaporatively-cooled condensers. Although the EER is defined at a single point for reporting purposes, Figure 30 shows that it changes as outdoor temperatures change.
Similar to the decrease in net cooling capacity shown in Figure 25, the EER of the evaporatively-cooled condenser also decreased (by approximately 10%) as a function of increasing temperature conditions (Figure 30). However, the decrease in EER is more severe (approximately 34%) as a function of increasing temperature conditions for the air cooled condensing units. This is a direct result of the decrease in net cooling capacity that is coincident with the increased power consumption as the dry bulb temperatures increased.

Overall, the performance of the evaporatively-cooled condensing unit suffers less in terms of net cooling capacity and energy consumption than does an air-cooled condensing unit. However, this performance benefit involves consumption of water, which can become an issue in arid areas with limited water supply.
CONCLUSIONS

The tests demonstrated that the evaporatively-cooled system experiences substantially less performance degradation under hot climate conditions than conventional air-cooled air conditioner systems. The energy savings over air-cooled condenser technology increase as the outdoor temperature increases where conventional air-cooled systems become less efficient. Additionally, the performance advantage of the evaporative condenser system is greatest in climate zones with high outdoor temperatures and low humidity.

PERFORMANCE

The humidity ratio of air passing through the evaporatively-cooled condenser drives the saturated condensing temperature of the refrigerant toward the wet bulb temperature of the air that is always lower than the dry bulb temperature. Condensing temperature is thus limited by ambient wet bulb temperature that is typically 14°F - 25°F below the ambient dry bulb temperature.

The TTC test results confirm the performance of evaporatively-cooled condenser technology to be significantly better than that of air-cooled condenser technology in hot and dry climate zones. In mild climate conditions of less than 101°F DB the air-cooled units demonstrated higher cooling capacity. However, the test results also showed the ECCU had up to 3.6% higher net cooling capacity per ton than the air-cooled units in the extreme hot and dry climate zone conditions. At temperatures above 101°F DB the performance of air-cooled condenser technology A/C degrades dramatically. The measured test data also showed up to 5 points greater EER and 0.5 kW/ton less power consumption for the ECCU than for the air-cooled units across the climate/temperature conditions tested.

Under ARI-Test A conditions of 95°F DB and 75°F WB, the ECCU produced a net cooling capacity of 11,825 Btu/hr/ton. The unit consumed 0.9 kW/ton of power and yielded an EER of 13.5 Btu/hr/watt.

In general, at higher DB temperatures the evaporatively-cooled condenser technology allows a higher net cooling capacity per ton than that of the air-cooled condenser technology without compromising the comfort conditions of supply air temperature and humidity. Additionally, the ECCU technology does not severely increase power consumption with increasing outdoor temperatures as does the air-cooled condenser technology.

WATER CONSUMPTION

Taking advantage of the evaporative cooling process using water is beneficial to energy efficiency. The lowered saturated refrigerant condensing temperature due to wet bulb temperature results in a significant benefit of reduced energy consumption. Water consumption versus water availability, however, may be an issue in areas where evaporatively-cooled condensing units perform optimally.

Under ARI-Test A conditions of 95°F DB and 75°F WB, the ECCU purged 0.59 gal/hr/ton of water each hour and evaporated 1.86 gal/hr/ton for a total water consumption of 2.45 gal/hr/ton.

The total ECCU water consumption increased with increasing changes in the humidity ratio (Δω) of the condenser inlet air or about 0.5 gal/hr increase for every 10% decrease in relative humidity.
In regard to water consumption, it should be noted that electrical power generation plants consume fresh water if their steam condensers operate with water from rivers or lakes. In this case, on a grid-level scale some of the water consumption may be offset to some extent by a reduction in water used in electricity generation proportional to the reduction in electrical energy consumption by the more efficient ECCU units. However, it takes energy to pump water to the site of an ECCU and there are thus energy losses in the water distribution system that may offset some of these gains.

**WATER QUALITY AND MAINTENANCE**

Study of the maintenance requirements of the ECCU system was outside the scope of this performance evaluation. It should be noted however, that the ECCU system includes additional components over an air-cooled system that are required to pump and handle the water and these may introduce the potential for higher maintenance costs. In contrast, however, it should be noted that in hot climate zones where the ECCU system is most advantageous, the maintenance needs of conventional air-cooled air conditioners will likely be higher than average because the system is operating much of the time under extreme conditions.

Another concern is the long-term performance of the ECCU technology that may be affected by buildup of hard water deposits on the refrigerant lines. This in turn decreases the heat transfer effectiveness from the refrigerant lines to the surface water film and influences the ability to cool the refrigerant. This buildup of hard water deposits ultimately results in decreased performance of the unit. The degree of decreased performance is a function of the water quality (hardness), period of exposure, and evaporation rate or climate zone. While it is certain that water quality is a long-term issue with this technology, the long-term effects of water quality are beyond the scope of this evaluation.

The long-term effects of water quality on the performance of the ECCU will next be evaluated by the Western Cooling Efficiency Center (WCEC) in Northern California. The intent here is to take the unit to failure mode while evaluating the long-term operational effects of hard water deposits on the refrigerant lines and other internal components of the condensing unit.
RECOMMENDATIONS

To further assess the potential for commercial viability and possible value of an incentive program for evaporatively-cooled condenser technology on air conditioning systems, some additional evaluations should be undertaken.

1. Conduct field testing to verify that laboratory test results can be achieved under actual operating conditions in the field.

2. Investigate impact of water usage on water supply, environmental impact, and cost issues in hot dry climate zones where the system is most effective. The water usage may be partially offset by a reduction in power plant water usage proportional to the reduction in electrical energy consumption by the unit.

3. Investigate potential for reducing the system water consumption. Options may include potential recycling of condensate water for cooling, reduction in sump purge volumes or cycles, and use of purge water for irrigation.

4. Investigate potential maintenance, reliability, corrosion, scaling, and other water related operational issues.

5. Perform cost/benefit assessment of installation of systems in hotter SCE climate zones. Include current and future cost projections, maintenance, water consumption, lifetime, operating costs, and other related issues.
REFERENCES


APPENDICES

APPENDIX A

SUPPLEMENTAL PERFORMANCE EVALUATIONS

EVAPORATOR COIL CHARACTERISTIC PERFORMANCE

Due to temperature stratifications on the exit side of the evaporator coil, the air temperature at this point was calculated using the measured temperature after the evaporator fan and subtracting the change in temperature caused by the evaporator fan motor. The temperature differential across the evaporator coil was determined based on the measured air temperature at the inlet of the evaporator coil, and the computed air temperature at the outlet of the evaporator coil, Equation 18.

EQUATION 18. TEMPERATURE DIFFERENTIAL ACROSS EVAPORATOR

\[ \Delta T_{\text{evap}} = T_{\text{evap-in}} - T_{\text{evap-out}} = T_{\text{evap-in}} \left(1 - \frac{kW_{\text{evap-fan}} \cdot k}{0.24 \cdot m_{\text{air}}}\right) \]

where
\( \Delta T_{\text{evap}} \) = temperature differential across the evaporator coil, °F
\( T_{\text{evap-in}} \) = air temperature at the inlet of the evaporator coil, °F
\( T_{\text{evap-out}} \) = air temperature at the outlet of the evaporator coil, °F
\( T_{\text{evap-fan-out}} \) = air temperature at the outlet of the evaporator fan, °F
\( k \) = conversion factor, (3,413 Btu/hr/kW)

Another indication of coil performance is the evaporator temperature difference (TD). It is defined as the difference in temperature between the temperature of the air leaving the evaporator and the saturation temperature of the refrigerant corresponding to the pressure at the compressor inlet, Equation 19. The saturation temperature at the inlet of the compressor is determined using the xProps program.

EQUATION 19. EVAPORATOR TEMPERATURE DIFFERENCE

\[ \text{TD}_{\text{evap}} = T_{\text{evap-out}} - \text{SST} \]

where
\( \text{TD}_{\text{evap}} \) = evaporator TD, °F
\( \text{SST} \) = saturated suction temperature based on compressor inlet pressure, °F
Additionally, the evaporator coil effectiveness was determined. As shown in Equation 20, the effectiveness is defined as the ratio of actual to maximum possible heat transfer rate.

\[
\varepsilon = \frac{\Delta T_{evap}}{T_{evap\, in} - SST}
\]

where
\[
\varepsilon = \text{evaporator coil effectiveness, unit-less}
\]

**EQUATION 20. EVAPORATOR COIL EFFECTIVENESS**

**CONDENSER COIL CHARACTERISTIC PERFORMANCE**

The temperature differential across the condenser coil was determined based on measured air temperatures at the inlet and outlet of the condenser coil, Equation 21.

\[
\Delta T_{cond} = T_{cond\, in} - T_{cond\, out}
\]

where
\[
\Delta T_{cond} = \text{temperature differential across the condenser coil, } ^\circ\text{F}
\]
\[
T_{cond\, in} = \text{air temperature at the inlet of the condenser coil, } ^\circ\text{F}
\]
\[
T_{cond\, out} = \text{air temperature at the outlet of the condenser coil, } ^\circ\text{F}
\]

The condenser TD is defined as the difference in temperature between the temperature of the air entering the condenser and the saturation temperature of the refrigerant corresponding to the pressure at the compressor outlet, Equation 22. The saturation temperature at the outlet of the compressor was determined using the xProps program.

\[
TD_{cond} = SCT - T_{cond\, in}
\]

where
\[
TD_{cond} = \text{condenser TD, } ^\circ\text{F}
\]
\[
SCT = \text{saturated condensing temperature based on compressor outlet pressure, } ^\circ\text{F}
\]

The heat exchange effectiveness of the condenser is also dependent on its LMTD and its UA. However, utilizing the arithmetic mean temperature difference can provide an approximation to the actual LMTD. The arithmetic mean temperature difference is determined using the air temperatures at the inlet and outlet of the condenser and the saturated condensing temperature based on compressor outlet pressure according to Equation 23.
Prior to determining the UA of the condenser coil, the heat of rejection at the condenser needed to be obtained. According to Equation 24, the heat of rejection is the sum of gross cooling capacity and the heat of compression, that can be directly determined from the input power to the compressor.

\[ Q_{rej} = m_{refrig} (h_{refrig-in} - h_{refrig-out}) \]

where

- \( Q_{rej} \) = heat of rejection at the condenser, Btu/hr
- \( m_{refrig} \) = mass flow rate of refrigerant, lb/hr
- \( h_{refrig-in} \) = subcooled liquid refrigerant enthalpy at the condenser inlet, Btu/lb
- \( h_{refrig-out} \) = superheated refrigerant enthalpy at the condenser outlet, Btu/lb

After the heat of rejection and arithmetic mean temperature difference were determined, Equation 25 was used to obtain the effective overall heat transfer coefficient of the condenser coil (UA).

\[ UA_{cond} = \frac{Q_{rej}}{\Delta T_{mean}} \]

where

- \( UA_{cond} \) = effective overall heat transfer coefficient of the condenser coil, Btu/hr-°F
**SYSTEM CHARACTERISTIC PERFORMANCE**

One of the system parameters is the evaporator coil superheat. The evaporator coil superheat was determined based on vapor refrigerant temperature at the evaporator coil outlet and the saturation temperature of the refrigerant corresponding to the pressure at the compressor inlet according to Equation 26.

**EQUATION 26. EVAPORATOR COIL SUPERHEAT**

\[ T_{SH-evap} = T_v - SST \]

where

- \( T_{SH-evap} \) = evaporator coil superheat, °F
- \( T_v \) = vapor refrigerant temperature at the evaporator coil outlet, °F

Equation 27 was used to determine the condenser subcooling. Condenser subcooling was obtained by subtracting the liquid refrigerant temperature at the condenser coil outlet from the saturated condensing temperature based on compressor outlet pressure.

**EQUATION 27. CONDENSER COIL SUBCOOLING**

\[ T_{SC-cond} = SCT - T_L \]

where

- \( T_{SC-cond} \) = condenser coil subcooling, °F
- \( SCT \) = saturated condensing temperature based on compressor outlet pressure, °F
- \( T_L \) = liquid refrigerant temperature at the condenser coil outlet, °F

The total system subcooling, that takes place after the liquid refrigerant leaves the condenser coil until it reaches the refrigerant metering device of the unit, was determined using Equation 28, i.e., subtracting the liquid refrigerant temperature at the metering device inlet from the saturation temperature of refrigerant corresponding to the pressure after the mass flow meter (metering device inlet). The liquid refrigerant temperature at the metering device inlet excludes the subcooling effect of the mass flow meter.

**EQUATION 28. TOTAL SYSTEM SUBCOOLING**

\[ T_{SC-total} = ST_{mfm-out} - T_{MD-in} \]

where

- \( T_{SC-total} \) = total system subcooling, °F
- \( ST_{mfm-out} \) = saturation temperature of refrigerant based on pressure after mass flow meter, °F
- \( T_{MD-in} \) = refrigerant temperature at the metering device inlet, less subcooling effect, °F
**HEAT AND WORK OF COMPRESSION**

The heat of compression, which is a function of refrigerant suction and discharge enthalpies, is obtained using Equation 29.

**EQUATION 29. HEAT OF COMPRESSION**

\[ \Delta h = h_{\text{discharge}} - h_{\text{suction}} \]

where

\( \Delta h \) = Heat of compression, Btu/lb

\( h_{\text{discharge}} \) = Enthalpy of superheat vapor discharged from compressor, Btu/lb

\( h_{\text{suction}} \) = Enthalpy of low pressure vapor at compressor inlet, Btu/lb

After determining the heat of compression, the work of compression was obtained by multiplying the heat of compression by the mass flow rate of refrigerant, Equation 30.

**EQUATION 30. WORK OF COMPRESSION**

\[ W_{\text{comp}} = m_{\text{refrig}} \cdot \Delta h \]

where

\( W_{\text{comp}} \) = Work of compression, Btu/hr

\( m_{\text{refrig}} \) = mass flow rate of refrigerant, lb/hr

\( \Delta h \) = Heat of compression, Btu/lb
## APPENDIX B

### SUMMARY TABLE OF ECCU PERFORMANCE PARAMETERS

#### TABLE 5. SUMMARY TABLE OF ECCU PERFORMANCE PARAMETERS ACROSS THE DIFFERENT CLIMATE ZONE CONDITIONS TESTED

<table>
<thead>
<tr>
<th>Parameter</th>
<th>CTZ Baseline</th>
<th>CTZ 6</th>
<th>CTZ 7</th>
<th>CTZ 8</th>
<th>CTZ 9</th>
<th>CTZ 10</th>
<th>CTZ 13</th>
<th>CTZ 14</th>
<th>CTZ 15</th>
<th>CTZ HDAC</th>
</tr>
</thead>
<tbody>
<tr>
<td>CTZ Dry Bulb Temp. (°F)</td>
<td>95</td>
<td>84</td>
<td>83</td>
<td>89</td>
<td>94</td>
<td>100</td>
<td>101</td>
<td>108</td>
<td>111</td>
<td>115</td>
</tr>
<tr>
<td>CTZ Wet Bulb Temp. (°F)</td>
<td>75</td>
<td>67</td>
<td>69</td>
<td>69</td>
<td>68</td>
<td>69</td>
<td>71</td>
<td>69</td>
<td>73</td>
<td>74</td>
</tr>
<tr>
<td>Net Cooling Capacity (Btu/hr)</td>
<td>35,474</td>
<td>36,343</td>
<td>35,840</td>
<td>35,753</td>
<td>35,736</td>
<td>36,188</td>
<td>35,258</td>
<td>34,580</td>
<td>34,766</td>
<td>34,274</td>
</tr>
<tr>
<td>Net Cooling Capacity per ton (Btu/hr/ton)</td>
<td>11,825</td>
<td>12,114</td>
<td>11,947</td>
<td>11,918</td>
<td>11,912</td>
<td>12,063</td>
<td>11,753</td>
<td>11,527</td>
<td>11,598</td>
<td>11,425</td>
</tr>
<tr>
<td>Power (kW)</td>
<td>2.62</td>
<td>2.50</td>
<td>2.56</td>
<td>2.55</td>
<td>2.53</td>
<td>2.55</td>
<td>2.57</td>
<td>2.54</td>
<td>2.61</td>
<td>2.61</td>
</tr>
<tr>
<td>Power per ton (kW/ton)</td>
<td>0.87273</td>
<td>0.83254</td>
<td>0.85245</td>
<td>0.84997</td>
<td>0.84425</td>
<td>0.85099</td>
<td>0.85813</td>
<td>0.84581</td>
<td>0.86993</td>
<td>0.87135</td>
</tr>
<tr>
<td>EER (Btu/hr/watt)</td>
<td>13.5</td>
<td>14.6</td>
<td>14.0</td>
<td>14.0</td>
<td>14.1</td>
<td>14.2</td>
<td>13.7</td>
<td>13.6</td>
<td>13.3</td>
<td>13.1</td>
</tr>
<tr>
<td>Water Evaporated (gal/hr)</td>
<td>5.57</td>
<td>5.14</td>
<td>4.93</td>
<td>5.25</td>
<td>5.67</td>
<td>5.72</td>
<td>5.63</td>
<td>6.45</td>
<td>6.49</td>
<td>6.74</td>
</tr>
<tr>
<td>Water Purged (gal/hr)</td>
<td>1.71</td>
<td>1.86</td>
<td>1.58</td>
<td>1.88</td>
<td>1.48</td>
<td>2.12</td>
<td>2.03</td>
<td>1.82</td>
<td>1.83</td>
<td>1.79</td>
</tr>
<tr>
<td>Total Water Consumption (gal/hr)</td>
<td>7.35</td>
<td>7.01</td>
<td>6.52</td>
<td>7.14</td>
<td>7.46</td>
<td>7.83</td>
<td>7.75</td>
<td>8.27</td>
<td>8.32</td>
<td>8.53</td>
</tr>
<tr>
<td>Outdoor Humidity Ratio (ω)</td>
<td>0.01414</td>
<td>0.0103</td>
<td>0.01203</td>
<td>0.01064</td>
<td>0.00874</td>
<td>0.00812</td>
<td>0.00943</td>
<td>0.00629</td>
<td>0.00875</td>
<td>0.00867</td>
</tr>
<tr>
<td>Change in Humidity Ratio Across Condenser (Δω)</td>
<td>0.02261</td>
<td>0.01526</td>
<td>0.01269</td>
<td>0.01955</td>
<td>0.02683</td>
<td>0.03509</td>
<td>0.0352</td>
<td>0.04953</td>
<td>0.05264</td>
<td>0.061</td>
</tr>
<tr>
<td>Outdoor Relative Humidity (%Rh)</td>
<td>39.81</td>
<td>41.24</td>
<td>49.59</td>
<td>36.3</td>
<td>25.58</td>
<td>19.79</td>
<td>22.28</td>
<td>12.13</td>
<td>15.42</td>
<td>13.62</td>
</tr>
</tbody>
</table>