Laboratory Testing of Performance Enhancements for Rooftop Packaged Air Conditioners

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LEGAL NOTICE

ABBREVIATIONS AND ACRONYMS

ACCA	Air Conditioning Contractors of America
AHRI	Air-Conditioning, Heating & Refrigeration Institute
ANSI	American National Standards Institute
ASHRAE	American Society of Heating, Refrigeration and Air-conditioning Engineers
ATS	PG&E Applied Technology Services
Btu	British Thermal Unit – a unit of energy required to raise 1 pound of water by 1°F
CEC	California Energy Commission
CFM	Cubic feet per minute – a unit of air flow
COP	Coefficient of Performance – a unit of efficiency (unit less)
CPUC	California Public Utilities Commission
DB	Dry bulb temperature (as in Tdb)
DEC	Direct Evaporative Cooling (or Cooler)
DP	Dew point temperature (as in Tdp)
DX	Direct eXpansion, as a descriptor for vapor compression air conditioning
EE	Evaporative Effectiveness, as a percent of wet-bulb depression
EER	Energy Efficiency Ratio – a unit of efficiency in Btu/Wh
EA	Exhaust Air from condenser (EAT = Exhaust Air Temperature)
GPH	Gallons per hour
HVAC	Heating, Ventilation and Air Conditioning
HR or W	Humidity Ratio or Absolute Humidity (mass fraction of water vapor to dry air)
IEC	Indirect Evaporative Cooling (or Cooler)
IOU	Investor Owned Utility
IW	Inches of water column - a unit of pressure
MA	Mixed Air – a blend of OA and RA (MAT = Mixed Air Temperature)

OA	Outside Air (OAT = Outside Air Temperature)
OEM	Original Equipment Manufacturer
PG&E	Pacific Gas and Electric Company
QM	Quality Maintenance
RA	Return Air from space (RAT = Return Air Temperature)
RH	Relative Humidity
RTD	Resistance Temperature Detector or Resistance Thermometer
RTU	Rooftop Unit (packaged air conditioner)
SA	Supply Air to space (SAT = Supply Air Temperature)
Ton	A unit of cooling capacity equal to 12,000 Btu/hr
WB	Wet Bulb temperature (as in Twb)
WBD	Wet Bulb Depression (the difference between the dry and wet bulb temperature of an air sample
WCEC	Western Cooling Efficiency Center of UC Davis (http://wcec.ucdavis.edu)

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EXECUTIVE SUMMARY

The research conducted for this project directly supports the California Energy Efficiency Strategic Plan goals to accelerate marketplace penetration of climate appropriate air conditioning technologies. The work was executed by PG&E Applied Technology Services, with leadership, vision, project management, and funding provided by the PG&E Emerging Technologies program. This report records results of a detailed laboratory evaluation of several enhancements that can be made to an operational rooftop unitary air conditioner (RTU).

The project began as an evaluation of a package of products that attach to a RTU that was submitted as part of the Western Cooling Challenge, a program that encourages manufacturers to develop and commercialize climate appropriate unitary air conditioning equipment. The Western Cooling Challenge works to characterize and compare the performance of these technologies in order to better inform utility program planning, customer investments in energy efficiency, and industry planning related to low energy mechanical design strategies optimized for western climates.

At the time the evaluation project was begun, there was a parallel effort being conducted by other consultants under CPUC Work Order 32 to evaluate the potential for energy improvement on RTUs. Since this project also involved performance testing of an RTU, the scope was expanded to include tests that would supplement the CPUC effort, specifically in regards to the performance of economizers.

California IOUs have also recently begun an air conditioner service program referred to as Quality Maintenance, but there is a lack of backing evidence to support the energy savings claims. A series of tests were conducted in order to quantify the savings, which involved an outside contractor certified to conduct the QM service.

Finally, since the WCC submitted system included as one of its components a condenser air evaporative pre-cooler, three additional examples of this technology were also evaluated.

The testing of economizers, evaporative condenser air pre-coolers and ventilation air pre-coolers is not covered under any existing testing standard. Therefore, appropriate methods to evaluate these products needed to be developed, and could be considered contributions towards future standards development.

PROJECT GOAL

The primary goal of this project was to measure and quantify the relative performance gains that can be achieved by a typical rooftop air conditioner from various enhancement technologies. This involved baseline testing of the air conditioner under a range of operating conditions, and then conducting similar tests with the enhancements applied. The project scope was limited to the collection of performance data, which can be later used to develop models for estimating the potential energy savings in different climates.

The testing is not intended to rank similar technologies against each other, but rather to provide perspective on the range of performance changes that can be achieved. Thus, an effort has been made to avoid naming manufacturers.

PROJECT DESCRIPTION

The project report is divided into five phases of laboratory testing:

1. Benchmark testing of an old RTU for comparison with the manufacturer's specifications
2. Execution of a Quality Maintenance service with follow-up performance testing and comparison with the benchmark to determine savings
3. Evaluation of the performance of economizers, which included:
   a. Tests for the airflow through the outside air damper as a function of damper position and pressure difference for two damper assemblies
   b. Demonstration of the functional performance of two recently-introduced digital economizer controllers
4. Performance testing of the RTU with four samples of evaporative condenser air pre-coolers, including one that combined a coil for ventilation air pre-cooling
5. Performance testing of the RTU with separate systems for indirect evaporative cooling of ventilation air and direct evaporative cooling of condenser intake air, packaged as a submission to the Western Cooling Challenge[1] and subject to its testing protocol

As the testing progressed and improvements were made to the RTU, new performance benchmarks were established to evaluate the savings resulting from the later enhancements.

PROJECT FINDINGS/RESULTS

There were numerous findings from the various test phases, and some of the key points from each phase are listed below.

1. Benchmark testing
   a. At the AHRI Standard 340/360 (Reference 1) rating conditions (95°Fdb outdoor and 80°Fdb/67°Fwb indoor), the subject 23-year-old RTU had a measured capacity that was 16% less than the manufacturer's rating, a power consumption that was higher by 5 to 8% (depending on the reference used), and a resulting energy efficiency ratio (EER) that was low by 20 to 22%.
   b. The measured supply airflow was low by 12 to 23% from its rating, which is affected by the presence of an economizer as well as a less-than-optimum ducting arrangement applied to the test unit. The manufacturer's literature does mention the increased system resistance created by an economizer when specifying the indoor fan size. Standard 340/360 does not include either OAS or an installed economizer when its rated value is determined.
   c. Modeling the performance of RTUs needs to account for the inefficiencies produced by low airflow and high duct resistance, as well as the drier climate in California creating less latent load and thus less total capacity. Average system age should also be considered.

[1] http://wcec.ucdavis.edu/programs/western-cooling-challenge/

2. Quality Maintenance (QM)
    a. The actions performed as the result of the QM service recommendations included coil cleaning, and filter and fan belt replacement. An economizer damper and control assembly replacement was also done.
    b. At the AHRI Standard rating conditions, the actions resulted in a 4% improvement in capacity, a 2% reduction in power, and a resulting 7% improvement in EER. However, the supply airflow was reduced by an additional 10% due to the higher resistance of the new economizer return air damper.
    c. An encouraging result of the QM service was that the capacity improvement increased with rising outside temperature when it is most needed. At an outside temperature of 115°F, the capacity improvement was 9% and the EER improvement was 11%. (The power reduction was still around 2%.)
3. Economizers
    a. Four methods for determining the outside air fraction of the supply air were tried without a clear favorite over the full range of test conditions. This measurement retains significant uncertainty.
    b. Tests on closed outside air dampers of the two after-market economizer damper assemblies tested indicated leakage rates of 10 to 20% of the RTU supply airflow if return duct static pressure loss is allowed. The leakage rate is dependent on the pressure difference across the damper, and increased return duct static pressure creates a larger pressure difference between ambient and return plenum.
    c. Leakage is not necessarily a bad thing since some ventilation air is usually a requirement for occupied spaces, but its uncontrolled nature is problematic. The ventilation requirement of ASHRAE 62.1[2] may actually be met with a closed outside air damper with these samples, so that setting the recommended minimum open position may result in over-ventilation. During heat storms, this results in less net cooling capacity and increased peak kW demand. Demand controlled ventilation based on CO_2 concentration in the conditioned space would allow a minimum set point of "closed", and is starting to be required by building energy efficiency codes.
    d. The recent Title-24 change disallowing linkage-controlled dampers may need to be reviewed, as the tested linkage damper had better leakage performance than the tested geared damper. However, these tests do not account for long-term reliability and which system is more likely to bind.
    e. The new generation of digital controllers provides more options for the control of the damper, as well as providing the means to turn off compressors when are not needed. They are required by the California Energy Commission Building Energy Efficiency Standards Title 24, Part 6[3]. They also provide a friendlier user interface and some fault detection capability. With the added options and set points, proper training of service technicians is required to ensure that they are set up properly for each installation.

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[2] https://www.ashrae.org/resources--publications/bookstore/standards-62-1--62-2 - Ventilation for Indoor Air Quality.

[3] http://www.energy.ca.gov/title24/equipment_cert/fdd/FDD_Certification_Guidance.pdf

f. To gain understanding into how the controllers operate, a test procedure was developed involving a steady ramp down and back up in outside temperature while observing the RTU performance in response to control signals from the economizer and the on-board controls of the RTU. This procedure could be enhanced and possibly refined into a working test standard. Since the outdoor dry bulb temperature plot over time looks like a "V", the test is referred to by that name.

4. Evaporative condenser air pre-coolers

   a. The performance improvement from pre-cooling increases at higher outside temperatures when it is needed the most. The effect is a function of wet-bulb depression, or the difference between the dry and wet-bulb temperatures. In hot/dry climates found in the Western United States, wet-bulb depressions often exceed 30°F.

   b. The maximum observed system demand reduction was 14% and the maximum increase in system capacity was 29%. Both of these values were from the same pre-cooler at an outside dry-bulb temperature of 115°F and a wet-bulb temperature of 75°F (40°F wet-bulb depression). Operations at a wet-bulb depression below about 15°F did not produce significant performance benefits from any of the test units. (Less than 5% improvement in efficiency in most cases.)

   c. All of the test pre-coolers affected the condenser airflow to some extent; up to an 11% reduction. This has a negative effect on performance offsetting some of the positive gains from the evaporative cooling. Up to a 4% reduction in RTU efficiency was observed from operation with a dry pre-cooler.

   d. The average saturation effectiveness of the four tested systems ranged from a low of 28% up to a high of 80%. (A saturation effectiveness of 100% would result in the condenser intake air being at the wet bulb temperature.)

   e. Based on the improvement in RTU efficiency, the better systems use less than 10 gallons of water for every kWh saved. This result can be used to determine the economic viability of the process. In addition, adaptive controls can be used to optimize water consumption, energy efficiency, and peak kW demand.

5. Indirect Evaporative Pre-Cooling of Ventilation Air

   a. In this phase of testing, an indirect evaporative cooler (IEC) was installed at the OAS inlet of the RTU. An IEC can often provide space cooling without the attached RTU operating, depending on the ambient conditions. However, RTU system leaks and high internal resistance may not deliver the full output airflow from the IEC to the conditioned space. Thus, the IEC may be better suited as a stand-alone product operating in parallel with its own open supply path to the space, and thermostat control set to a temperature below that of the remaining RTUs so that it is operated first.

   b. The saturation effectiveness of the IEC averaged 113%, meaning that it usually supplied air at a temperature below the entering wet-bulb temperature, which will commonly be below 70°F. (In Title-24, highest design wet-bulb temperature in California is 76°F.)

   c. The exhaust from the IEC was always cooler than the entering outside air, and could be used for condenser air pre-cooling, although it is not as cool as from a dedicated direct system.

d. With the conditioning of the ventilation air, the amount of fresh air provided to the occupants can be increased without a significant penalty to the cooling capacity. It can also expand the range of "free cooling" when compressor operation can be curtailed.

e. The testing involved steady-state periods with different combinations of component operations and environmental conditions under manual control. In actual operation, the system will need to have a sophisticated control system to allow operation in its most efficient configuration for the current climate, which may be developed based on these test results.

f. Qualification of the combined system components for the Western Cooling Challenge will be done by the Western Cooling Efficiency Center based on the collected laboratory test data and the results of their field testing, and provided in a separate report.

PROJECT RECOMMENDATIONS

The results from these tests provide justification for the continuing or implementing incentive programs. The results following the Quality Maintenance service provide needed performance improvement data to support the continuation of the program as an energy efficiency or market transformation measure. The results from the pre-cooler testing are already being used to develop models of system performance to be applied to the various California climate zones and eventually used to develop work papers for appropriate incentives. Economizers are already required by Title-24 in RTUs above a specific capacity, but the study results can be used to support retrofits of existing units and replacement of controllers.

As with most experimentation, the results from these tests raise more questions that require further study. One of the more crucial areas as demonstrated in the last phase is how to best control a variety of system components to provide the required level of cooling for the least cost in energy, and in some cases water. There is interest in the HVAC community towards developing an annualized or load-based performance metric for RTUs that takes into account the intake of outside air, whether required or when it is actually beneficial in reducing the refrigeration requirement. It is hoped that the results from these tests can be used either through the collected data or the experimental test methods as a step towards development of this metric.

INTRODUCTION

Over half of the commercial floor space in California is conditioned by packaged rooftop air conditioning units (RTUs), with the majority of these systems in the 5 to 10-ton cooling capacity range. With their general simplicity and good reliability if properly maintained, these systems can be in service for more than their expected life of 12-15 years. In the meantime, advances in technology are pushing the efficiency of new products, increasing their attractiveness for early replacement. However, replacement still remains a capital intensive prospect and the economics of the energy efficiency improvement may not be sufficient to justify replacement of an operational system, despite utility incentives. There then remains an entrenched inventory of aging RTUs having the potential for decreasing capacity and efficiency over time, and thus higher electric energy use and power demand and the resulting higher cost of operation, which presents an opportunity for products or services that can maintain or improve system performance until such time that replacement becomes economic or necessary.

The California Investor Owned Utilities (IOUs) recently implemented a Commercial Quality Maintenance program (http://www.commercialhvacqm.com) wherein incentives are paid for regular service that follows a procedural checklist for RTUs based on ANSI/ASHRAE/ACCA Standard 180 (Reference 4). The program also includes contractor training in following the Standard, and in the use of approved Quality Maintenance tools. Quantifying the performance improvement achieved through proper maintenance is difficult in a field installation, and thus translating the results of system service into utility savings that can support such a program remains difficult. Individual system savings are also subject to the condition of the RTU and its past maintenance practices, and results will vary widely. With this in mind, there is still a desire to see if there are measurable performance gains from conducting the Quality Maintenance service, and this provided the initiative to explore this in a laboratory environment.

One of the most common enhancements to a RTU is the addition of an economizer: an assembly of dampers, actuator, and controls designed to select the lowest energy air source for cooling between normal building return air and outside air. When there is a call for cooling and the outside air has a lower temperature than the return air (or enthalpy if in a humid climate), the economizer opens a damper to the outside and closes a damper on the return air. An integrated economizer can also interrupt compressor operation as needed to meet supply air dry bulb temperature requirements using just ventilation air, thereby extending the operation range of the economizer. Economizers are so common that checking their function is included in the Quality Maintenance service, particularly since they have been identified from field studies as having a high rate of failure.

Because an economizer provides access to outside air, it is also often tasked with providing the ventilation air to a space as required by code. This means that even when the outside air has more energy than the return air, some outside air is still brought in through a minimum open position. Setting this minimum open position is problematical in that the outside air volume is a function of several internal system pressures, and the result is often that too much outside air is being delivered to the space. In hot weather, this creates increased load on the space; or more apparently, a decrease in the apparent cooling capacity of the RTU.

In recent years, a new generation of stand-alone digital economizer controllers has appeared with the added benefits of simplifying setup, fault indication, and integration with the operation of the compressors and fans with the aim of greater system efficiency. The hope is that these new systems will support correct installation, commissioning, and fault detection, thus helping to reduce the high failure rate that economizers experience.

The vast majority of RTUs have air cooled condensers for simplicity and ease of maintenance. The drawback to air cooled systems is that as the outside temperatures rise, the capacity and efficiency of the RTU decreases at a time when the capacity is needed the most and electric demands are high. Most of California's climate zones that experience high temperature also have relatively low humidity, so one method available to improve cooling efficiency is through the use of evaporative cooling. The majority of large HVAC and refrigeration systems (such as hydronic chillers) use evaporative condensers or cooling towers because of their efficiency and capacity benefits. For air-cooled systems, performance may be enhanced by applying direct evaporative cooling (DEC) to the air supplied to the condenser. The trade-offs for the performance improvements are a significant increase in the maintenance required with the evaporative system, and also concern over water consumption, especially for drought-prone California.

While the air used for condensing can be conditioned by direct application of evaporative cooling, the ventilation air usually cannot be treated in this manner because it can create an environment that may be too humid for comfort. However, systems that apply indirect evaporative cooling can be used for this purpose. In an indirect system, there are two air paths: one that is cooled by direct evaporation and a second that is cooled by heat exchange with the first. This second stream is then supplied to the space without an increase in moisture content from the outside air. Some of the more advanced products can actually achieve wet-bulb effectiveness in excess of 100%, because their limiting temperature is actually the entering air dew point.

This project began as the evaluation of a package of an indirect evaporative cooler for the ventilation air drawn in through the economizer combined with an evaporative condenser air pre-cooler to a rooftop unit as a submission to the Western Cooling Challenge. In order to properly evaluate the effect of the system, the baseline performance of the RTU needed to be determined and repairs made as necessary, as well as determine the effect of outside air intake that is not available from manufacturer's data. Thus, the project expanded to be a comprehensive examination of the effects from each of these sub-systems and performance adjustments.

BACKGROUND

Some of the investigation covered in this report echoes a previous study into rooftop unit performance from 1999 (Reference 6), which also included the effect from the addition of an evaporative pre-cooler. The PG&E Applied Technology Services HVAC laboratory has also evaluated several different evaporative cooling products over the years, and many of the test reports are available at the Emerging Technologies Coordinating Council website (http://www.etcc-ca.com). More recently, an evaluation of another product submission to the Western Cooling Challenge was conducted just prior to this current effort (Reference 8), and the report on it is also available at the same location.

The UC Davis Western Cooling Challenge is a program that encourages manufacturers to develop and commercialize climate appropriate unitary air conditioning equipment. The program was initiated at the behest of California utilities and various major energy end users who recognize the need for substantial HVAC energy savings. The program defines a climate appropriate test methodology for laboratory evaluation of air conditioning equipment.

Western Cooling Challenge certified equipment must demonstrate a 40% savings at peak load conditions compared to standard efficiency equipment. All of the technologies evaluated thus far for the Western Cooling Challenge employ some form of evaporative cooling system because of the higher heat rejection potential over simple air cooling. The program and evaluation criteria were designed around comparison to conventional rooftop packaged air conditioners in commercial applications that operate with some minimum ventilation rate. The Western Cooling Challenge test criteria do not apply specifically to add-on components like the submitted package, but the system evaluated here was tested under similar environmental conditions, and savings were projected using the same assumptions about system operating constraints that have been used for test of other Western Cooling Challenge equipment.

Much of the additional testing was done to add to validate the effort being conducted by others under CPUC Work Order 32: HVAC Impact Evaluation, Measurement and

Verification[4]. This particularly applies to the impact of outside air being drawn in through the economizer, and whether it is excessive.

The evaluation of evaporative condenser air pre-cooling systems is also being used to support the development of a uniform testing method. ASHRAE Standard Project Committee 212 is working towards a standardized method of test to rate the performance of pre-coolers in terms of evaporative effectiveness and water utilization. The testing work conducted herein helped to evaluate the feasibility of the proposed testing method.

TECHNICAL APPROACH/TEST METHODOLOGY

TEST PLAN

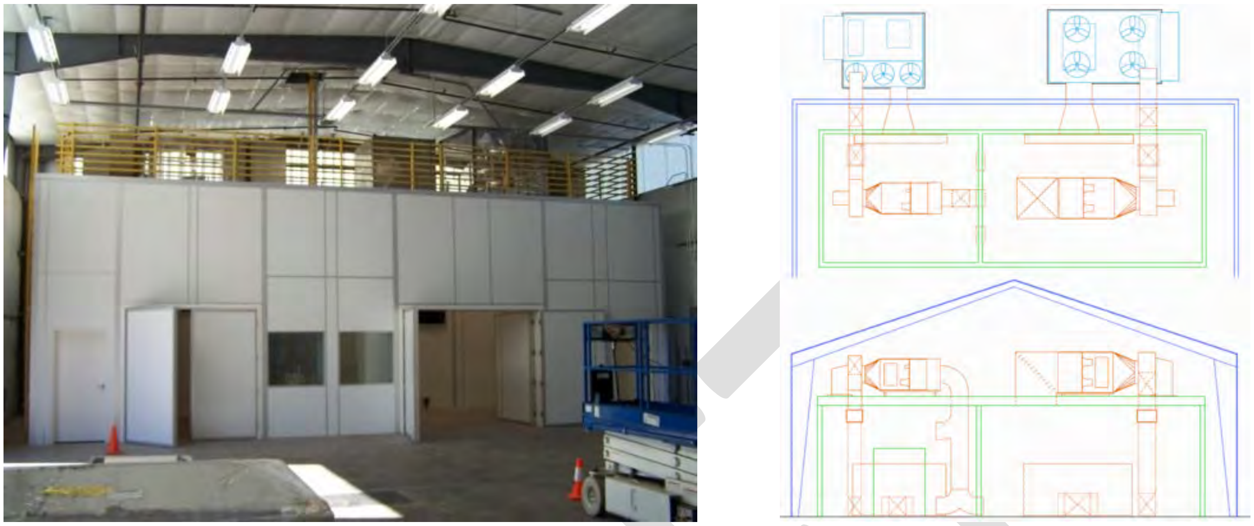
The testing plan was primarily to examine the relative change in performance as a result of applying various measures or changes in system operation. It was not intended to exactly replicate the same test setup as is used by certified laboratories to obtain a system's performance rating. However, the apparatus and testing procedures followed established standards as closely as the project goals allowed. The particular standards followed include ASHRAE Standard 37 (Reference 2) for the testing apparatus and instrument specifications, and AHRI Standard 340/360 (Reference 1) for setting standard rating conditions.

LABORATORY FACILITY

All testing was performed in the HVAC testing apparatus in the Advanced Technology Performance Lab (ATPL) at PG&E's San Ramon Technology Center. The apparatus consists of two side-by-side environmental chambers designed following ASHRAE Standard 37. The two chambers have independent conditioning systems for maintaining temperature and humidity, and each has its own airflow measurement apparatus or "code tester". The airflow measurement apparatus follow ASHRAE standard design, and consist of a sealed box with a partition having several flow nozzles that can be opened or sealed in combination to provide the required range of differential pressure for the current airflow. Variable-speed blowers on the outlets of each station can be set to maintain the desired outlet static pressures or airflow rates and compensate for the added resistance of the flow measurement system and ductwork. The smaller of the two chambers is conditioned to maintain the required return air conditions to the RTU and its code tester is used to measure the supply airflow from the RTU. Whereas the larger chamber is used to maintain the required outside air conditions to the RTU condenser and its code tester is used to measure the condenser exhaust airflow. The test RTU and all add-on components were located completely inside the large chamber.

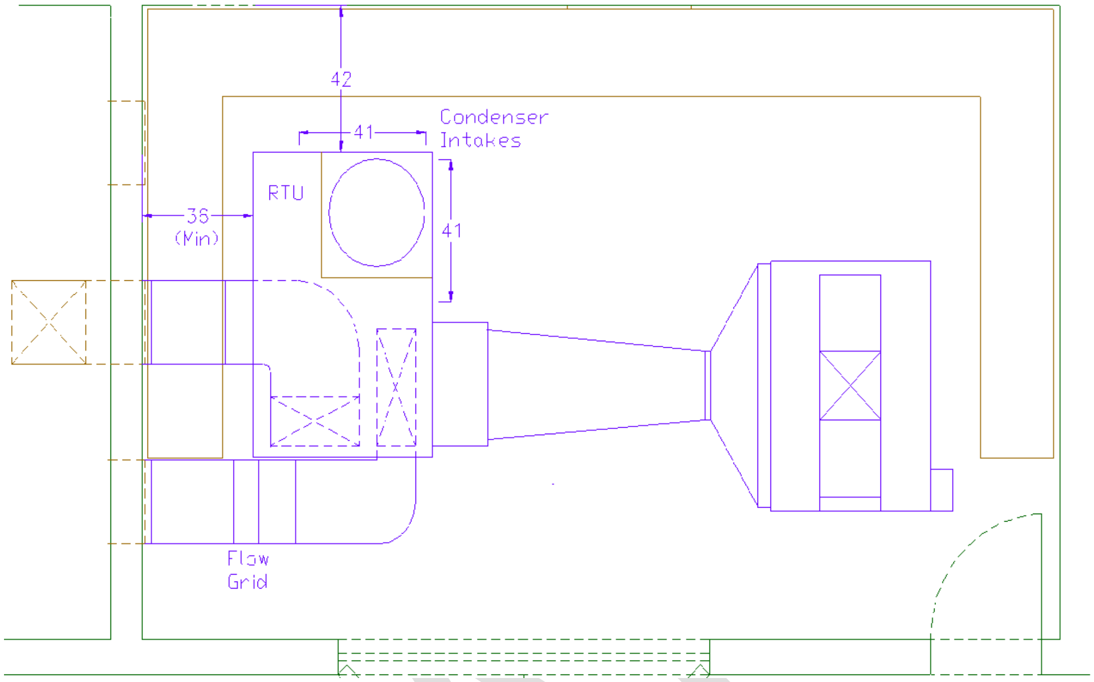
[4] http://www.performancealliance.org/Portals/4/Documents/WO32-EMV-Presentation-11-7-13-Part-1.pdf

FIGURE 1: HVAC TEST LAB
(INDOOR ROOM ON LEFT, OUTDOOR ROOM ON RIGHT)



Some deviations from the ASHRAE standard were necessary due to access and space considerations to allow for the installation of the add-on components; the most significant of which was the requirement for straight duct attached to the return and supply openings of the RTU. The test unit was designed for vertical connections, so it needed to be raised onto a stand to bring the ducting in from below. Because of the plan to eventually install an indirect evaporative cooler to the outside air intake of the economizer, the test unit also needed to be oriented such that the outside air intake was facing in the long dimension of the room. On the return side, ASHRAE Standard 37 requires straight duct of the same dimensions as the rectangular opening with a length equal to 1.7 times the square root of the product of the two dimensions of the rectangular opening for a vertical return, with the pressure taps taken at 0.5 times the same factor. For this unit, the return duct would need to be 35 inches long. On the supply side, the length multiplier is 2.5 with a pressure tap location multiplier of 2. Again for this unit the supply duct would need to be about 50 inches long. To accommodate this, the test RTU would have needed to be raised to nearly 6 feet above the floor to allow for elbows to connect the ducting to the adjacent indoor room. To keep the system accessible, the unit was raised only enough to attach elbows directly beneath the unit, which created an undesirable set of two non-coplanar elbows on both the return and supply. Pressure taps for the return and supply were made using a tubing ring connecting penetrations of all four faces of the duct directly at the connection to the RTU. This is not ideal, but may actually be closer to a real-world field installation than the requirements of the Standard. The performance analysis is not being used for the purposes of providing a rating, but rather is being done to look for the relative change in performance between different modifications. Thus, this configuration was acceptable for the purpose of this test program.

FIGURE 2: LAB LAYOUT WITH TEST RTU AND IEC



Instrumentation Plan

Table A-1 in the Appendix documents the instrumentation and its accuracy for the measurements taken in the laboratory facility. The majority of the temperature and pressure measurements were taken external to the unit, typically in the attached ductwork. The exception are the direct refrigerant measurements, for which pressure transmitters were attached to the existing Schrader valves and surface temperature sensors were applied to the refrigerant tubing, all inside the RTU case. The temperature and pressure transmitters were calibrated against laboratory standards through the data acquisition system prior to testing. For the temperatures, the calibration included a low point using an ice bath (32°F) and a high point using a hot block calibrator (100°F). The raw measurements were adjusted to match the reading from a secondary temperature standard placed in the same environment. The four dew-point sensors had received a factory calibration in December 2012.

All of the instruments were connected to signal conditioning modules based on the National Instruments C-series architecture, connected to six Compact-RIO chasses. The modules included different units for RTDs, thermocouples, voltage, current, and pulse counting, plus both analog and digital output modules to control the room conditioning systems and the test RTU. Two of the Compact-RIO chasses were connected by serial cables to a weather station and a digital scale used for weighing condensate. The default chassis internal scan rate for reading the module inputs is 10 Hz, although the weather station and scale updated once every second.

The six Compact-RIO chasses communicate over an Ethernet network to a central host computer, which ran a custom data acquisition and control program developed with National

Instruments LabVIEW™ graphical programming language. The program acquired readings from the chasses at a rate of 2 Hz, applied calibration scaling and maintained a running average for each measurement, and logged the averages to a file every 10 to 15 seconds. The scaled values and other calculated values were also displayed on screen in both text and graphical form, and used to generate feedback control signals to the space conditioning systems.

The logged data was saved in an ASCII text format that is easily imported into Microsoft Excel for analysis. A macro is run on the raw data file to apply formatting, calculate statistics, and create trend charts. The result is then analyzed to isolate a period of stable operation. For most of the tests, the target period duration was 30 minutes as specified by Standards, although shorter duration periods were accepted when thermal stability was not critical (e.g. damper mapping), or on rare occasions when some operating anomaly reduced the acceptable data set. Once this period is identified, the statistics (average, standard deviation, range) are isolated to just this period and then copied over to another spreadsheet with one row per test. Operating performance metrics are then calculated from these values, and the results are checked for the test tolerances specified in ASHRAE Standard 37.

RESULTS

PHASE 1: BASELINE UNIT PERFORMANCE

The RTU used for this testing program served for 22 years as the space conditioning system for the ATS cafeteria, as shown in ***Figure 3***.

FIGURE 3: ORIGINAL LOCATION OF TEST RTU ON ATS BUILDING

The cafeteria space was an addition to the main building that received its own separate conditioning systems when it was constructed: one for the eating area and one for the kitchen. The cafeteria was being used increasingly as a large meeting room, creating more load than the RTU was designed to handle as a lower population density cafeteria. The decision was eventually made to tie the cafeteria area into the main building conditioning system, and which point the subject RTU was removed from the building and recovered for the testing program.

The test unit is a Carrier Model DJD009 with a nominal capacity of 8½-tons (100,000 Btu/hr). The unit has a fixed speed 1½-hp indoor blower (constant volume), a single ¾-hp condenser fan, and two compressors operating with R-22 refrigerant. Two Product Data references for this unit were found online (Forms 48DJ-4PD and 48DJ-5PD), both from 1991 but with slightly different performance information. The former lists the standard airflow as 3400 CFM at total power of 11.2 kW and a rated EER of 8.9 Btu/Wh; while the latter lists a standard airflow of 3000 CFM (although it says 3400 CFM later in the Physical Data table), total power of 10.9 kW and a rated EER of 9.15 Btu/Wh. It is uncertain which of these the test unit more closely resembles.

The references contained Performance Data tables listing total and sensible cooling (gross coil cooling, which excludes heating from the indoor blower) and compressor power as a function of condenser intake air dry bulb temperature, evaporator return air wet bulb temperature and evaporator air flow rate. These tables are excerpted below:

TABLE 1: MANUFACTURER'S REFERENCE PERFORMANCE DATA TABLES

Form 48DJ-4PD

48DJ009 (8½ Tons)

Temp (F) Air Ent Cond (Edb)		Evap Air - CFM 2550			Evap Air - CFM 3400		
		Evap Air - Ewb (F) 72	67	62	72	67	62
85	TC	116.5	104.8	93.3	123.2	111.7	100.0
	SHC	58.0	71.1	83.1	64.5	81.8	96.7
	kW	8.52	8.21	7.89	8.71	8.41	8.10
95	TC	110.7	98.8	87.2	117.0	**105.3**	94.0
	SHC	55.9	68.8	80.1	62.2	**79.7**	93.0
	kW	9.14	8.80	8.45	9.33	**9.00**	8.68
105	TC	104.7	92.6	80.9	110.5	98.8	88.7
	SHC	53.7	66.3	77.0	60.0	77.1	88.6
	kW	9.76	9.40	9.02	9.95	9.59	9.27
115	TC	98.3	85.9	73.7	103.7	92.2	83.1
	SHC	51.3	63.6	73.1	57.4	74.6	83.1
	kW	10.34	9.98	9.60	10.54	10.17	9.91

Form 48DJ-5PD

48DJ009 (8½ Tons)

Temp (F) Air Ent Cond (Edb)		Evap Air - CFM 2550			Evap Air - CFM 3400		
		Evap Air - Ewb (F) 72	67	62	72	67	62
85	TC	118.5	106.6	94.1	125.3	113.3	100.4
	SHC	58.0	70.9	82.8	64.3	81.7	96.6
	kW	8.22	7.95	7.68	8.37	8.12	7.87
95	TC	112.7	100.4	87.8	119.2	**107.1**	94.9
	SHC	56.0	68.8	79.8	62.2	**79.3**	93.1
	kW	8.83	8.54	8.25	9.00	**8.73**	8.43
105	TC	106.6	94.4	81.3	112.4	100.7	89.0
	SHC	53.7	66.2	76.6	59.6	76.9	89.0
	kW	9.43	9.13	8.82	9.59	9.31	9.03
115	TC	100.5	87.4	73.8	105.5	94.2	83.4
	SHC	51.5	63.6	72.6	57.1	74.3	83.4
	kW	10.04	9.73	9.42	10.19	9.89	9.67

Average Manufacturer's Reference

48DJ009 (8½ Tons)

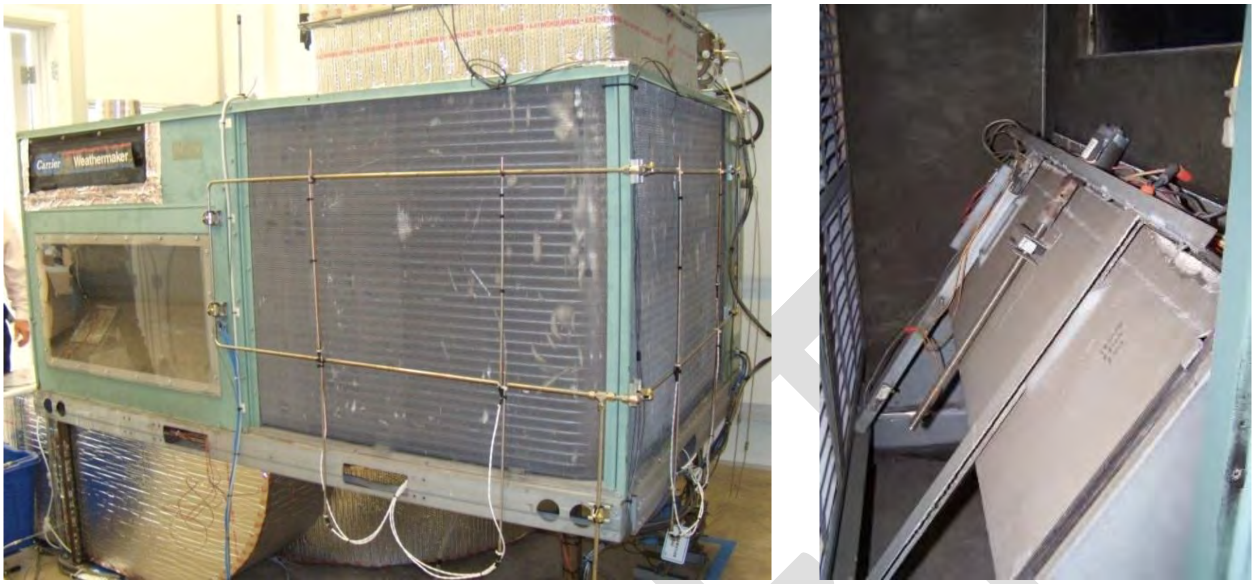
Temp (F) Air Ent Cond (Edb)		Evap Air - CFM 2550			Evap Air - CFM 3400		
		Evap Air - Ewb (F) 72	67	62	72	67	62
85	TC	117.5	105.7	93.7	124.3	112.5	100.2
	SHC	58.0	71.0	83.0	64.4	81.8	96.7
	kW	8.37	8.08	7.79	8.54	8.27	7.99
95	TC	111.7	99.6	87.5	118.1	**106.2**	94.5
	SHC	56.0	68.8	80.0	62.2	**79.5**	93.1
	kW	8.99	8.67	8.35	9.17	**8.87**	8.56
105	TC	105.7	93.5	81.1	111.5	99.8	88.9
	SHC	53.7	66.3	76.8	59.8	77.0	88.8
	kW	9.60	9.27	8.92	9.77	9.45	9.15
115	TC	99.4	86.7	73.8	104.6	93.2	83.3
	SHC	51.4	63.6	72.9	57.3	74.5	83.3
	kW	10.19	9.86	9.51	10.37	10.03	9.79

TC Total Cooling (1,000 Btu/hr)
SHC Sensible Cooling (1,000 Btu/hr)
kW Compressor Power Only

(The manufacturer's tables also contain data for an airflow rate of 4250 CFM, but this was left out because it would not be achievable with this unit as it would have required a different blower motor.) Based on the total power reported by each reference at the rating condition, the combined fan power between the indoor blower and the condenser fan is 2.2 kW.

The test unit was equipped with and OEM economizer that consists of a sliding plate on a jack screw that moved between outside air and return air. The damper had been permanently fixed in the 100% return air position with a small panel open for a minimum outside air intake. Several wires were broken and the economizer function was found to be unusable. To lessen the impact of outside air leakage during the baseline testing, the outside air opening was sealed closed with an acrylic panel, but the economizer was left in place.

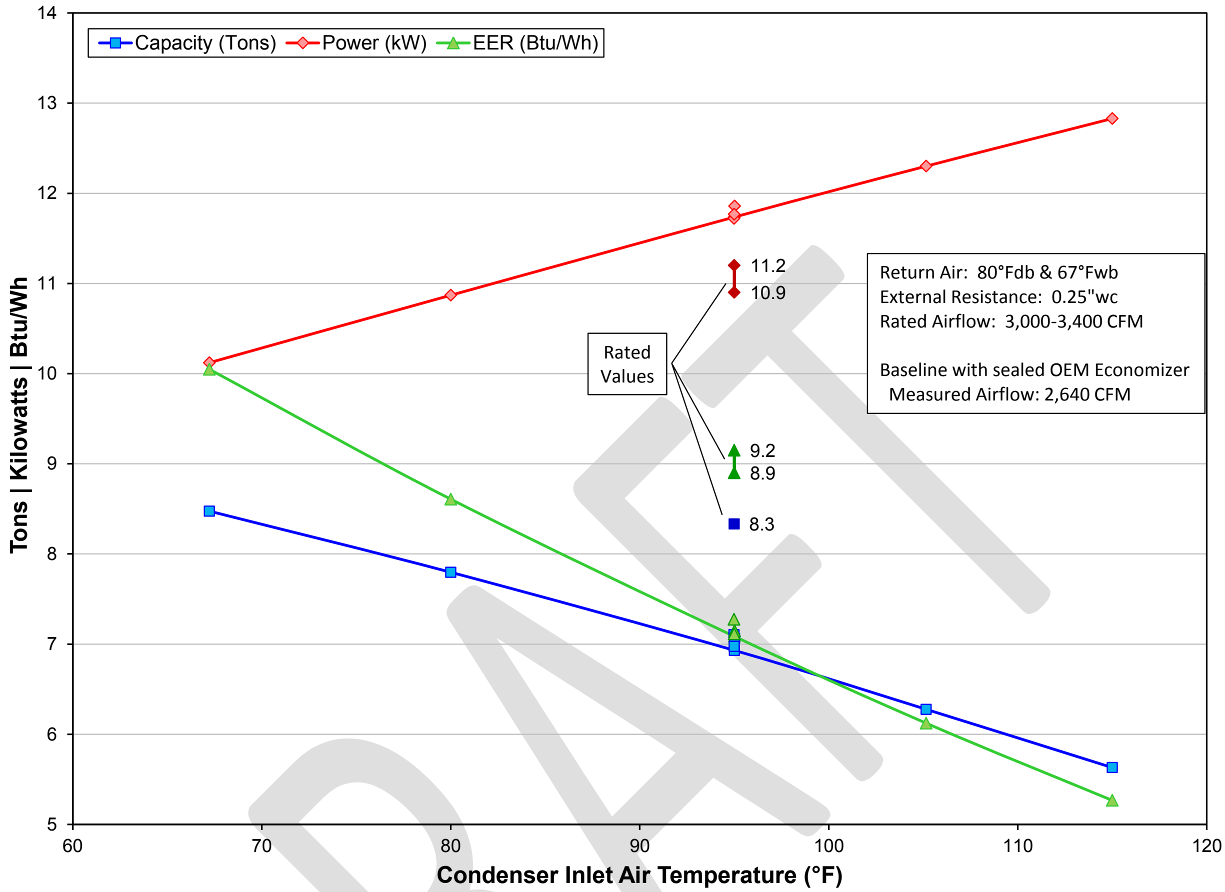
FIGURE 4: TEST UNIT INSTALLED IN LAB, AND OEM ECONOMIZER



Prior to the start of performance testing, there was one mishap with the unit installation. Both of the two refrigerant circuits were connected to pressure transmitters to continuously measure the compressor suction and the condenser outlet pressures. The connections to transmitters were made using tees that connected to the existing Schrader valve fill ports at these locations. The tees had the transmitter attached to one branch and a duplicate Schrader valve on the other so that refrigerant charge adjustments could be made without disconnecting the transmitters. When a technician hand-checked the tightness of the fitting on the suction of the primary compressor refrigerant circuit, the valve stem snapped off from the refrigerant line, releasing the refrigerant charge. A service technician was brought in to repair the break, which consisted of sealing up the hole left by the stem and providing a new fill port, pressure testing the line with nitrogen, pulling a vacuum on the circuit while allowing time for entrained water to evaporate, and then recharging the circuit to the factory weight of new R-22 refrigerant. (The second circuit did not need repair; at least not at this time.) So while the original plan was to capture baseline data on the RTU as it would have been immediately after being relocated from the roof, this could no longer be achieved. The baseline then became this repaired state with a new charge in the primary circuit.

There were two parts to the baseline performance testing. The first was testing with the external static pressure as specified by the AHRI Standard (0.25 IW for the rated capacity of this unit) and with the standard return air conditions of 80°Fdb/67°Fwb, and then mapping the system performance over a range of condenser inlet temperatures, from a low of 67°F to a high of 115°F. The result from this part is the performance map shown in ***Figure 5***.

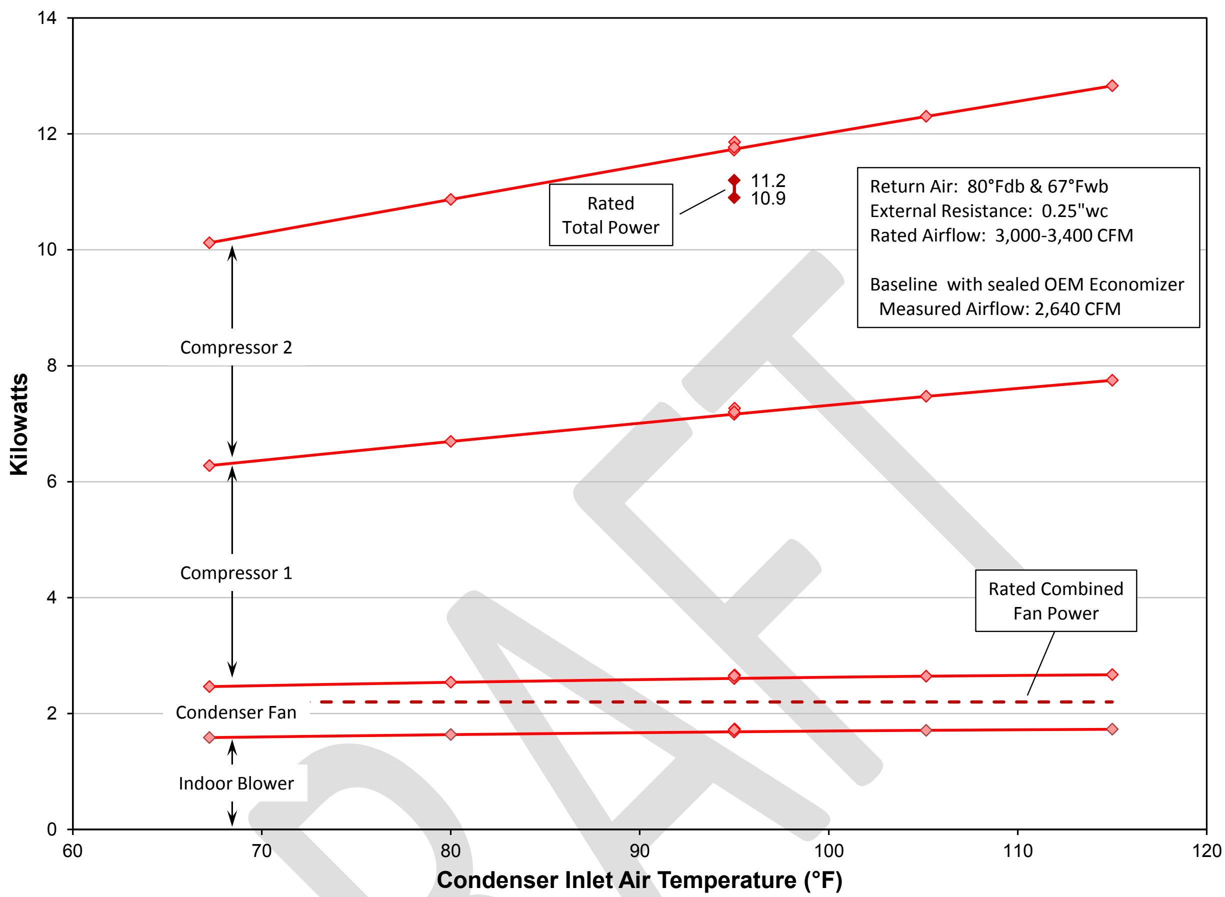
FIGURE 5: INITIAL RTU BASELINE TEST PERFORMANCE MAP



The AHRI rated performance from the two manufacturer's reference documents have been added for comparison, which are all at the standard rating condition of 95°F condenser inlet air. The results show that the measured power consumption is high by 5 to 8%, the cooling capacity is low by 16%, and the resulting energy efficiency ratio (EER) is low by 20 to 22%. Of particular interest, the measured airflow through the unit is 12 to 23% low (depending on the reference used), which is likely a contributing factor to the low capacity. One of the reasons for the low airflow is the presence of the economizer, as the manufacturer's ratings are for a system without one. Even with the return damper wide open and the outside air intake sealed, there is still added flow resistance that would not exist if the economizer was not there.

The total power consumption for the test unit from this chart can also be broken down into its individual contributors: the two compressors and two fans. This has been done in ***Figure 6***, based on the ratio of the individual current measurements from each component to their sum as applied to the total system power. The suggested manufacturer's rating for the combined fan power of 2.2 kW is included in the figure. The comparison indicates that high fan power may be the main contributing factor to the higher overall power draw of the system, as the difference between the measured total power and the rated total power is about the same as the difference between the rated combined fan power and its measurement.

Figure 6 Breakdown of System Total Power Consumption by Contributor



The second part to the baseline testing was to replicate the performance data tables from the manufacturer's reference as much as possible. This expands the other test into two additional return air wet bulb temperatures and two fixed supply airflow rates. While the first set of tests held the external resistance constant, for these tests the pressure was allowed to float while the booster fan on the airflow measurement apparatus was set to meet the required airflow rate. For the 2550 CFM set, the external pressure was slightly higher than the rating condition at an average of 0.30 IW. However, to achieve the 3400 CFM set, the booster fan had to pull a vacuum on the supply side, thus creating a situation that the unit could not achieve on its own. This additional external fan power is not included in the system performance metrics, and may make the results look better than they should be. Acting in opposition to this, drawing a vacuum on the RTU creates an increased opportunity to draw in outside air through case leaks, resulting in added load when the outside air is hotter than the return air.

There was a system fault that occurred during four of these tests that was not noticed until the data was processed because it did not show an effect on the cooling capacity of the unit. Rather, the fault evidenced itself through about a 32% higher than normal compressor power demand. This fault cleared itself later in the same day of testing, and did not reoccur until much later into the testing program (during the evaporative pre-cooler testing). At that future date, the fault was traced using the current measurements to the primary circuit compressor, and from there to one leg of the compressor contactor. The contactor was closing in with a high resistance on one leg, creating a high current imbalance on the remaining two legs, and a high power indication. Cleaning the contacts with emery paper cleared up the issue. This fault demonstrated the need for service technicians to check the current on all three phases to the unit to confirm that they are relatively well balanced.

The results from this phase of the baseline testing are shown in ***Table 2*** and ***Figure 7***. Some modification of the actual measured results was necessary to properly match the manufacturer's data since capacity is determined from measurements external to the RTU and total unit power was being measured. Both the total (TC) and sensible (SHC) cooling capacity values have been adjusted to deduct the temperature rise caused by the blower, and the power consumption of just the compressors was separated from the total unit power measurement using current measurement ratios. In the table, the average manufacturer's reference performance data is repeated alongside the results obtained from the testing, with an additional table showing the relative difference from the manufacturer's average. The compressor power data from the four tests that had the dirty contactor fault are included but crossed out to show that the results are invalid.

TABLE 2 BASELINE TEST RESULTS RELATIVE TO MANUFACTURER'S REFERENCE

Average Manufacturer's Reference
48DJ009 (8½ Tons)

Temp (F) Air Ent Cond (Edb)		2550			3400		
		72	**67**	**62**	**72**	**67**	**62**
85	**TC**	117.5	105.7	93.7	124.3	112.5	100.2
	SHC	58.0	71.0	83.0	64.4	81.8	96.7
	kW	8.37	8.08	7.79	8.54	8.27	7.99
95	**TC**	111.7	99.6	87.5	118.1	**106.2**	94.5
	SHC	56.0	68.8	80.0	62.2	**79.5**	93.1
	kW	8.99	8.67	8.35	9.17	**8.87**	8.56
105	**TC**	105.7	93.5	81.1	111.5	99.8	88.9
	SHC	53.7	66.3	76.8	59.8	77.0	88.8
	kW	9.60	9.27	8.92	9.77	9.45	9.15
115	**TC**	99.4	86.7	73.8	104.6	93.2	83.3
	SHC	51.4	63.6	72.9	57.3	74.5	83.3
	kW	10.19	9.86	9.51	10.37	10.03	9.79

Evap Air - CFM; Evap Air - Ewb (F)

Test Data - Baseline

Temp (F) Air Ent Cond (Edb)		2550			3400		
		72	**67**	**62**	**72**	**67**	**62**
85	**TC**	110.1	99.0	87.4	140.1	115.2	103.8
	SHC	52.2	66.0	76.6	60.1	76.0	90.5
	kW	~~10.53~~	7.87	7.52	~~13.03~~	11.52	11.14
95	**TC**	102.3	90.8	80.8	130.2	102.2	97.8
	SHC	48.7	61.4	72.4	56.8	71.4	85.2
	kW	~~11.53~~	8.34	7.92	~~14.24~~	12.12	11.68
105	**TC**	97.1	82.6	72.4	137.6	105.6	82.9
	SHC	46.7	57.4	68.5	53.8	66.0	79.2
	kW	9.20	8.82	8.31	13.13	12.80	12.23
115	**TC**	86.3	77.3	65.6	128.6	114.8	84.5
	SHC	42.8	54.1	63.9	50.6	63.1	74.8
	kW	9.77	9.28	8.85	13.92	13.23	12.77

Evap Air - CFM; Evap Air - Ewb (F)

Difference from Reference

Temp (F) Air Ent Cond (Edb)		2550			3400		
		72	**67**	**62**	**72**	**67**	**62**
85	**TC**	-6%	-6%	-7%	+13%	+2%	+4%
	SHC	-10%	-7%	-8%	-7%	-7%	-6%
	kW	~~+26%~~	-3%	-3%	~~+53%~~	+39%	+40%
95	**TC**	-8%	-9%	-8%	+10%	-4%	+4%
	SHC	-13%	-11%	-9%	-9%	-10%	-8%
	kW	~~+28%~~	-4%	-5%	~~+55%~~	+37%	+36%
105	**TC**	-8%	-12%	-11%	+23%	+6%	-7%
	SHC	-13%	-13%	-11%	-10%	-14%	-11%
	kW	-4%	-5%	-7%	+34%	+35%	+34%
115	**TC**	-13%	-11%	-11%	+23%	+23%	+1%
	SHC	-17%	-15%	-12%	-12%	-15%	-10%
	kW	-4%	-6%	-7%	+34%	+32%	+30%

Evap Air - CFM; Evap Air - Ewb (F)

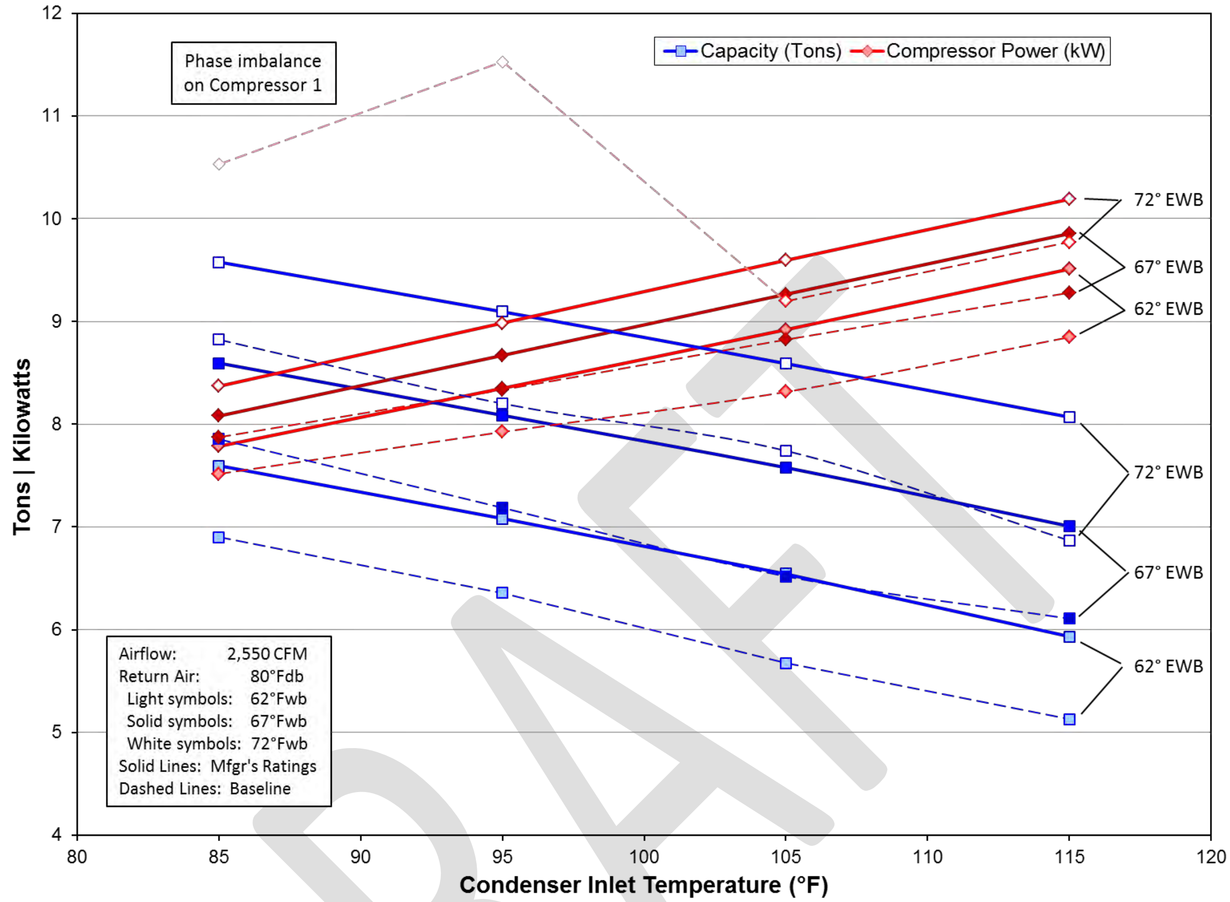
TC Total Cooling (1,000 Btu/hr)
SHC Sensible Cooling (1,000 Btu/hr)
kW Compressor Power Only

Power measurements crossed-out for tests that had poor contact on one of the phases of compressor #1

The values in the table for total net cooling and compressor power are shown in ***Figure 7*** for just the 2550 CFM airflow condition.

The conclusion from the baseline testing is that this 22+ year old RTU is performing with lower than rated airflow and capacity and higher than rated power over a wide range of conditions. The age of the unit is only one contributor to this difference, with the others being the compromises made in the testing apparatus (ducting), the presence of an economizer, and the general leakage of the unit. Given the age of the unit, the fact that it is still operating with reasonably good performance is impressive.

FIGURE 7: COMPARISON OF TEST DATA TO MANUFACTURER'S REFERENCE AT 2550 CFM



PHASE 2: QUALITY MAINTENANCE SERVICE

Once the complete set of baseline tests was performed, the system was scheduled for an in-situ service from a Quality Maintenance-certified contractor. The contractor used Field Diagnostics' HVAC Service Assistant™ on a tablet computer[5] to step through the service procedures outlined in ANSI/ASHRAE/ACCA Standard 180, and diagnose issues with the system using inputs from field instrumentation measuring refrigerant pressures and temperatures and air temperatures. The software then identifies potential problems and where to direct service efforts.

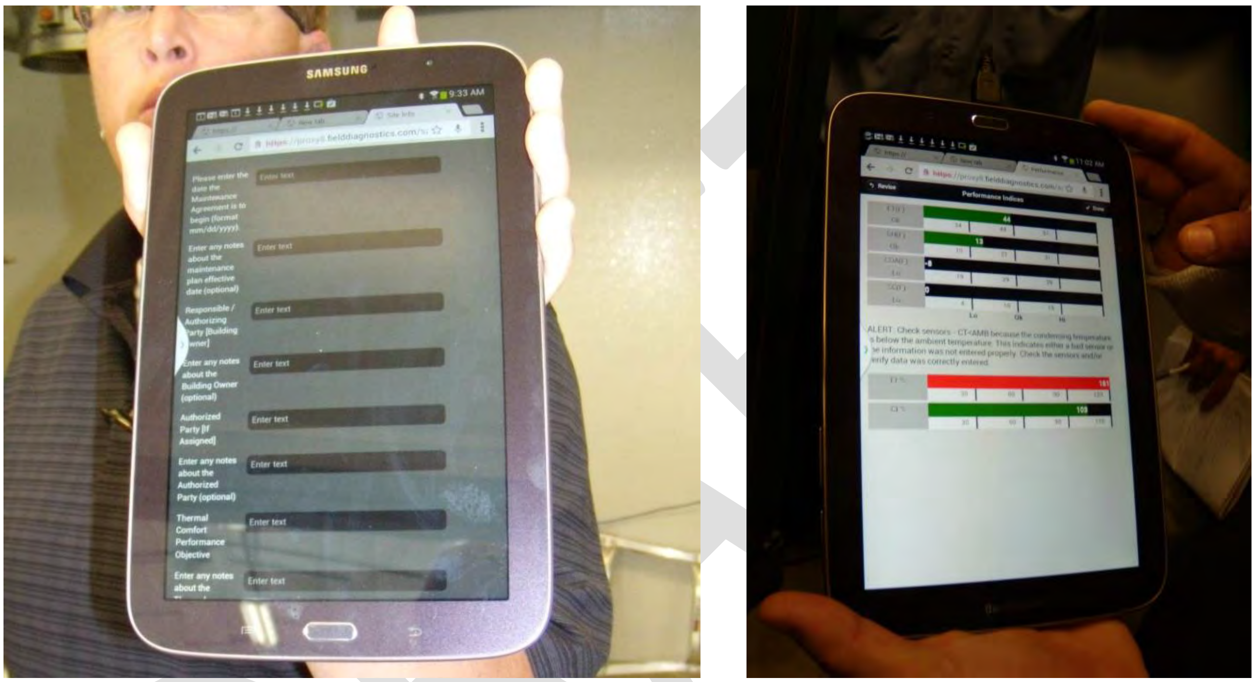
The outdoor room was maintained at a comfortable 80°F for the service rather than the rating condition of 95°F. This temperature was also selected because it minimized the effect from any air leakage into the unit as the return air was also held at 80°Fdb (and 67°Fwb). The initial diagnosis from the Service Assistant™ was that the condensing over ambient (COA) reading and the evaporator saturation temperature (ET) were both in their acceptable range, but superheat (SH) and subcooling (SC) were both off-scale on the low side. The device also gave the message:

[5] https://www.fielddiagnostics.com/

> "ALERT: Check sensors – CT<AMB because the condensing temperature is below the ambient temperature. This indicates either a bad sensor or the information was not entered properly. Check sensors and/or verify data was correctly entered."

This message relates that some entered measurement (either ambient temperature or the condenser pressure) was incorrect as the condensing temperature cannot be less than the ambient air temperature that it is rejecting heat to.

FIGURE 8: FIELD DIAGNOSTICS' HVAC SERVICE ASSISTANT™ DATA ENTRY AND DIAGNOSTIC SCREENS



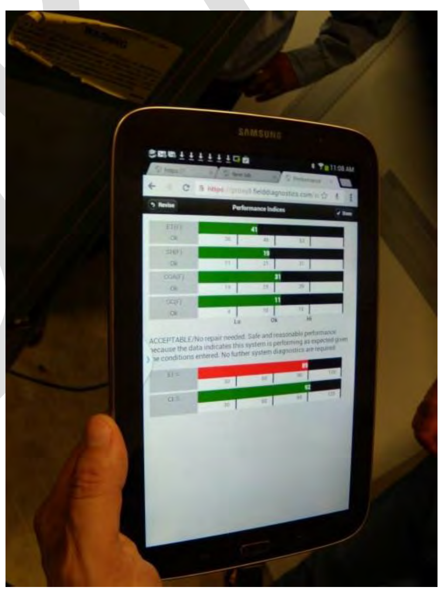
The system diagnostic suggested possibilities for system faults, including low charge or heat transfer problems with either the evaporator or condenser. One factor that may have contributed to the diagnostic of poor condenser heat transfer is that a side panel of the RTU needed to be removed to access the Schrader valve ports and tubing for the refrigerant pressure and temperature measurements, and this creates a short-circuit for air flow to bypass the condenser. Some attempt was made to minimize the opening while the measurements were made, but it still was not the same as if the panel were still there. This provides justification for having pressure access points external to the unit. In response to theft and inhalation ("huffing") of refrigerant, units on the market do not have service valves that are accessible from the outside. To address the requirements of charge testing, valves need to be located behind a removable panel in a compartment that is separated from the condenser section.

Changing the refrigerant charge is the most invasive of the three diagnoses and the most expensive, while airflow issues should be dealt with first in any HVAC diagnostic step. The service technician then performed a cleaning of the evaporator and condenser coils using a backpack spray rig with water and cleaning solution. The spray was directed in the same direction as the airflow because that direction is the most convenient to reach. (Upstream side of the evaporator coils after moving the filters, and the outside of the condenser coil.) The coils were not particularly dirty, and the spray was continued until the runoff ran clear. There are some that are critical of this approach and even go so far as to recommend removal of the coils to clean them. This may be needed in extreme cases but it remains to be proven that spraying counter to airflow direction is required.

FIGURE 9 CLEANING EVAPORATOR AND CONDENSER COILS

The inputs required by the HVAC Service Assistant™ were then measured again, and the result this time was overall acceptable; other than the estimated relative system efficiency metric at 89%, which is just below the acceptable value of 90%. Lab testing verified that performance was about this much below the rated value.

FIGURE 10 DIAGNOSTIC RUN AFTER COIL CLEANING



The message indicated on the display is:

> "ACCEPTABLE/No repair needed. Safe and reasonable performance because the data indicates this system is performing as expected from the conditions entered. No further system diagnostics are required."

Refrigerant charge and filter cleanliness were both judged to be acceptable from this result, and the Quality Maintenance part of the service was concluded. (A detailed review of this analysis tool is available in Reference 7, using data from previous tests on residential split systems).

As an extra step in the service, the economizer assembly was replaced by the technician with an after-market unit having linked dampers that move in parallel. The digital controller installed with it is one that met Title 24 requirements. The technician ensured that all of the wiring to the controller and the RTU were connected properly and that the economizer damper function worked as intended. The performance of this economizer is analyzed in Phase 3, and all of the Post-QM data collection would be done with the economizer in its 100% return air position and with the outside air opening enclosed in plastic sheeting.

Following the recommendation of the service technician, the original smooth indoor blower fan belt was replaced with a new cogged belt to reduce slippage, once it could be obtained (since the service technician did not have a suitable replacement). It was also decided to go ahead with replacing the air filters even though they were not particularly dirty. After these two follow-up steps, most of the baseline performance tests were repeated, with the exception of the tests at 3400 CFM as these still required extra pull from the booster fan to achieve.

FIGURE 11 ORIGINAL SMOOTH AND REPLACEMENT COGGED FAN BELTS

The results from the subsequent tests with an external resistance of 0.25 IW applied to the unit are overlaid on the baseline figure for comparison (***Figure 12***). The results show that there was a small but measurable improvement in performance as a result of the Quality Maintenance service. There was both a reduction in power and an increase in capacity over the entire range of outside temperatures, both contributing to increased system efficiency. Of particular interest is that the increase in cooling capacity was larger at higher outside temperatures when having extra capacity is most important. The only downside was that the supply airflow was further reduced by 9% from the baseline, most likely as the result of the added resistance of the new economizer dampers. The blower speed remained the same between the two tests at about 775 RPM.

FIGURE 12: PRE- AND POST-QUALITY MAINTENANCE SERVICE RESULTS AT AHRI STANDARD CONDITIONS

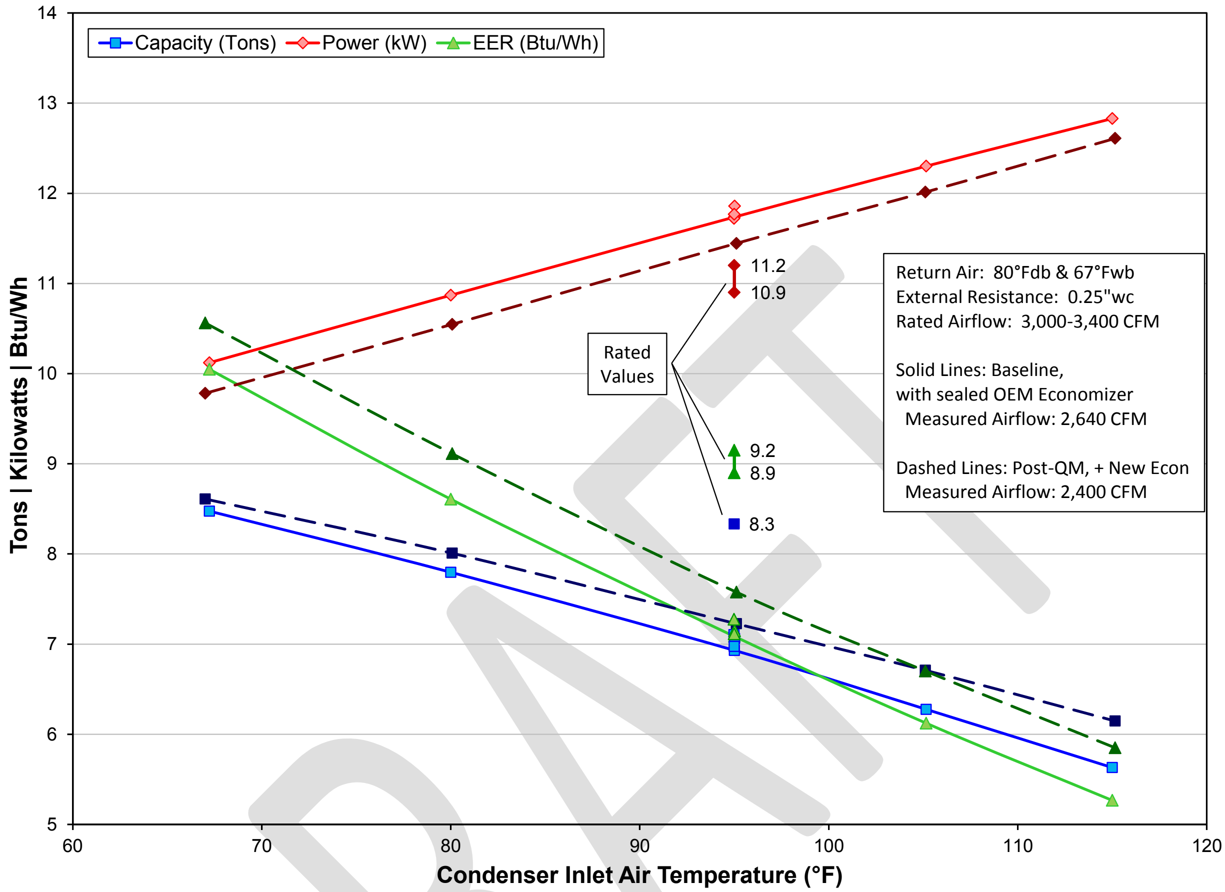


Table 3 and ***Figure 13*** are the continuation of the previous excerpt from the reference Performance Data tables showing the relative change from before and after the Quality Maintenance service. The largest increase in total cooling capacity was 11%, although the largest decrease in compressor power was only 2%. The small decrease in compressor power means that the larger decrease seen in total power in the previous figure is likely due to lower indoor blower motor power. This is linked to the reduction in airflow, because throttling a fan by restricting airflow on the intake (as the new economizer damper is doing) will result in reduced fan power.

TABLE 3: POST-QUALITY MAINTENANCE PERFORMANCE DATA RELATIVE TO BASELINE

Test Data - Baseline

Temp (F) Air Ent Cond (Edb)		Evap Air - CFM 2550, Evap Air - Ewb (F)		
		72	67	62
85	TC	110.1	99.0	87.4
	SHC	52.2	66.0	76.6
	kW	~~10.53~~	7.87	7.52
95	TC	102.3	90.8	80.8
	SHC	48.7	61.4	72.4
	kW	~~11.53~~	8.34	7.92
105	TC	97.1	82.6	72.4
	SHC	46.7	57.4	68.5
	kW	9.20	8.82	8.31
115	TC	86.3	77.3	65.6
	SHC	42.8	54.1	63.9
	kW	9.77	9.28	8.85

Test Data - Post-QM

Temp (F) Air Ent Cond (Edb)		Evap Air - CFM 2550, Evap Air - Ewb (F)		
		72	67	62
85	TC	113.6	99.0	88.2
	SHC	51.9	65.0	77.7
	kW	7.98	7.71	7.38
95	TC	107.3	93.2	81.7
	SHC	48.9	61.9	74.2
	kW	8.57	8.25	7.85
105	TC	100.6	86.1	75.0
	SHC	46.0	58.4	69.8
	kW	9.08	8.74	8.30
115	TC	94.8	81.3	69.4
	SHC	43.0	55.3	66.1
	kW	9.71	9.21	8.79

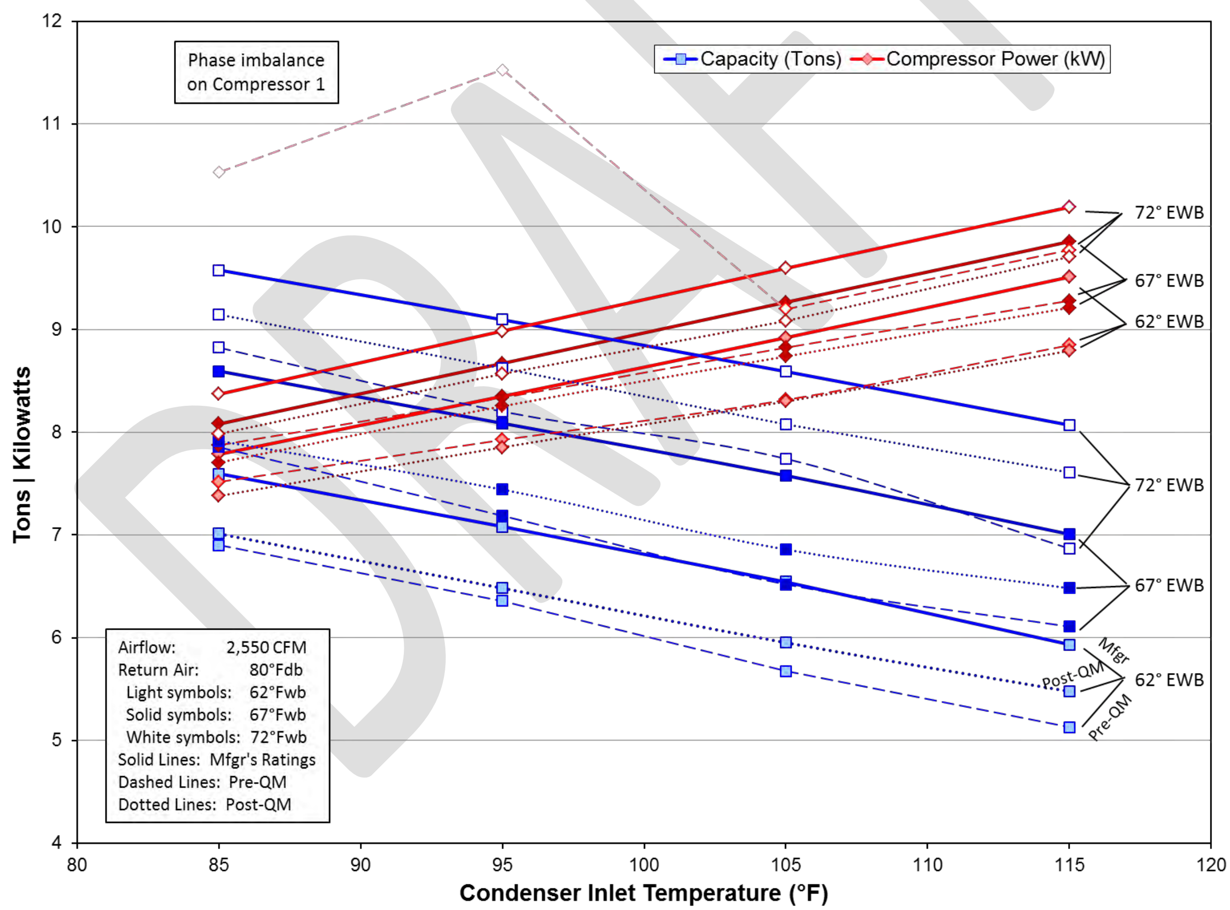
Difference from Baseline

Temp (F) Air Ent Cond (Edb)		Evap Air - CFM 2550, Evap Air - Ewb (F)		
		72	67	62
85	TC	+3%	+0%	+1%
	SHC	-0%	-1%	+1%
	kW	~~-24%~~	-2%	-2%
95	TC	+5%	+3%	+1%
	SHC	+0%	+1%	+2%
	kW	~~-26%~~	-1%	-1%
105	TC	+4%	+4%	+4%
	SHC	-1%	+2%	+2%
	kW	-1%	-1%	-0%
115	TC	+10%	+5%	+6%
	SHC	+0%	+2%	+3%
	kW	-1%	-1%	-1%

TC Total Cooling (1,000 Btu/hr)
SHC Sensible Cooling (1,000 Btu/hr)
kW Compressor Power Only

Power measurements crossed-out for tests that had poor contact on one of the phases of compressor #1

FIGURE 13: COMPARISON OF PRE- AND POST-QM TEST DATA TO MANUFACTURER'S REFERENCE AT 2550 CFM



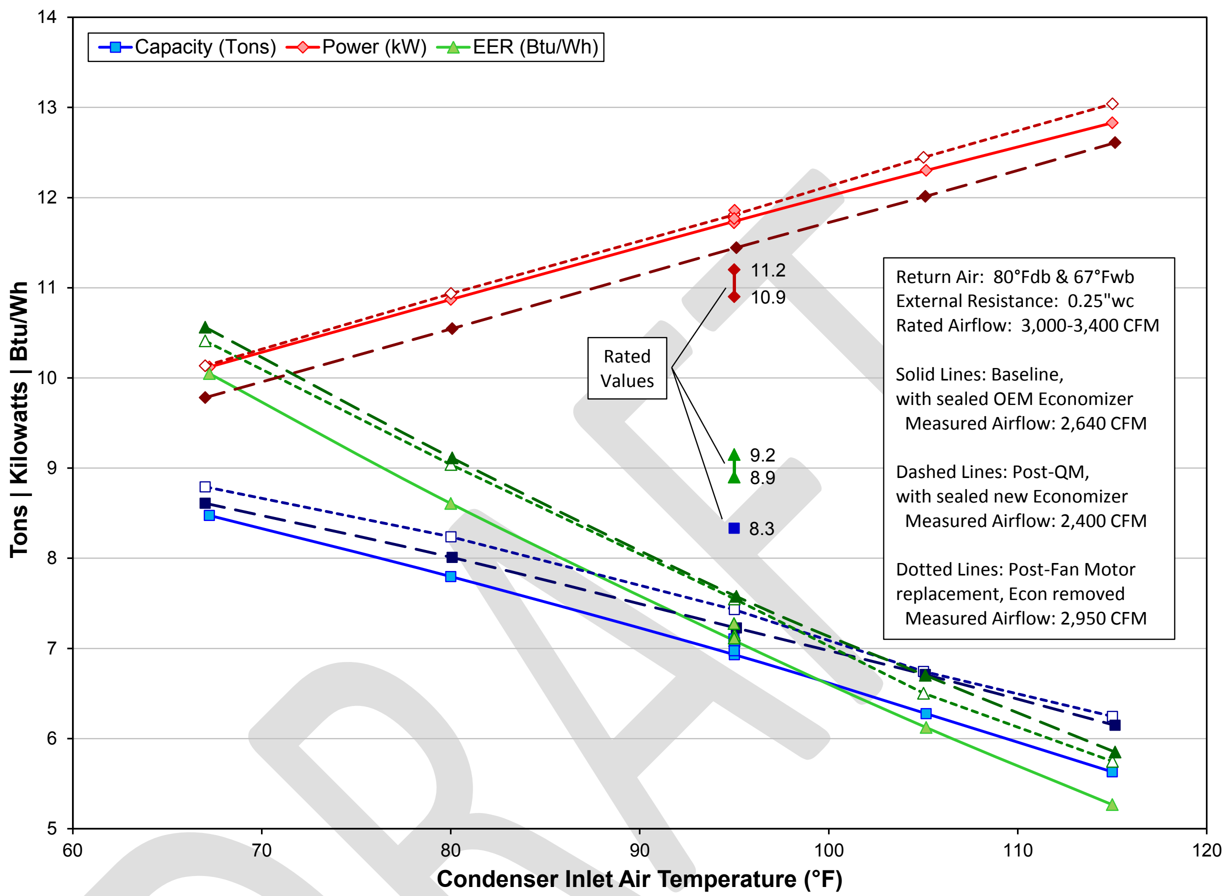
An additional system modification was made to the unit by a team that had been working on CPUC Work Order 32 at another facility. They came to this test unit primarily to compare their field instruments against the laboratory instruments (particularly airflow), but also to diagnose the low airflow performance that had been identified for this test unit. Their system modifications included replacing the indoor blower motor with a new motor of the same model, and making adjustments to the diameter of the pulley at the motor to increase the fan speed. Because the rated system airflow is without an installed economizer, the after-market damper assembly was removed and the opening covered with a sheet of plywood.

FIGURE 14: INVESTIGATION OF LOW SUPPLY AIRFLOW (ECONOMIZER DAMPER REMOVED)

The fan speed was increased slightly from 775 RPM to 800 RPM through the pulley adjustment, and the result was an improvement in supply airflow to nearly the lower of the two rated values at 2950 CFM. The set of tests at the standard external resistance were repeated once again with the new configuration. The increase in airflow came with the cost of increased fan power to bring the total above the original baseline result, but this was compensated for by an increase in capacity, particularly at lower temperatures. The resulting system efficiency (EER) was about the same as from the previous post-Quality Maintenance tests.

The changes in performance were small but measurable under laboratory conditions. Under field conditions using field instrumentation, measurement uncertainty can be larger than the measured savings. Following the guidance of Standard 180, quality maintenance provides the means to achieve the goal of "maintained" status on all systems. However, the results of RTU maintenance energy efficiency programs are mixed and there is not yet a data set that can definitively establish the distribution of quality maintenance impacts, much less the distribution of RTU faults and in-situ performance.

FIGURE 15: PERFORMANCE MAP COMPARING BASELINE, QUALITY MAINTENANCE AND BLOWER ADJUSTMENT

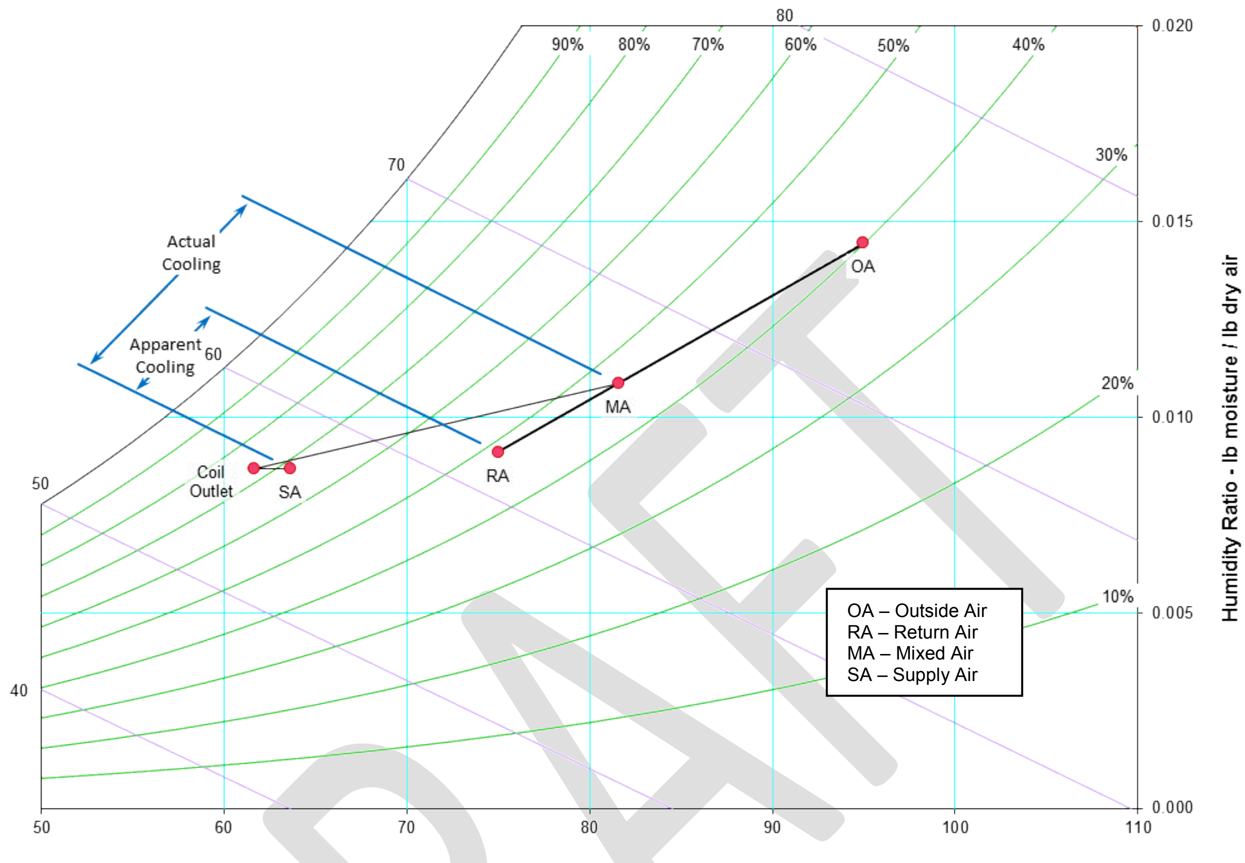


PHASE 3A: ECONOMIZER DAMPER AIRFLOW

The previous testing of the RTU identified that the addition of an economizer will reduce the rated system airflow even with a fully open return damper, potentially by a significant amount. Restricted airflow normally results in a shift in the sensible heat ratio (SHR) of the system towards more latent cooling (moisture removal) and less sensible cooling (temperature reduction). With California's relatively dry climate, a high SHR is desirable, so ensuring good airflow is a primary concern.

Of equal and possibly greater importance is how much outside air is allowed to enter the RTU, particularly when its condition creates added system load. As shown in the following psychrometric chart (Figure 16), if the outside air is hot and humid and a large amount of it is allowed to enter the mixed air plenum of the RTU, then the apparent cooling capacity of the RTU can be significantly reduced even though the actual cooling may not have changed much. In this example, the apparent cooling is about half of the actual cooling. The difference is the cooling required by the additional ventilation air.

FIGURE 16: PSYCHROMETRIC CHART DEMONSTRATING THE EFFECT OF OUTSIDE AIR LEAKAGE ON COOLING CAPACITY



The measurement of outside air leakage rates for economizers is not a simple process, nor one that has a standard measurement method. ANSI/AMCA Standard 500-D-12 describes laboratory methods of testing dampers for rating, but this standard is intended for single duct-mounted dampers and describes measuring leakage rate as a function differential pressure across the damper. Economizers consist of a pair of dampers – outside and return air – which are normally linked together: as one opens, the other closes. For the tests in this program, the desire was to measure the leakage rate of an economizer damper in-situ, with the potential for adapting the methodology to field use. To further complicate the situation, pressure changes caused by damper operation will impact the leakage of the RTU case.

With an economizer damper assembly installed in an RTU, the factors that can be controlled in the lab include the position of dampers and the air pressures at the supply and return connections to the RTU. What is not easily controlled is the pressure in the mixed air plenum, which is a function of all three. The test plan was to measure the outside air fraction of the supply airflow and the mixed air plenum pressure as a function of these three parameters. Controlling the supply static pressure and the damper position were already easily achievable, but controlling the return air pressure required a modification of the test apparatus. This modification consisted of adding a propeller fan to a return duct extension followed by a damper that can be adjusted to reduce the pressure added by the fan to the desired value. (Completely closing this damper would create a stagnated fan, and the motor would overheat and eventually shut off.)

The measurement of the outside air fraction can be done by several methods, four of which were tried with varying results, and often not agreeing with each other. The four methods are as follows:

Method 1: Difference between measured supply and return airflow rates

In addition to the fan and damper on the return duct, a flow station was added to the duct consisting of a flow straightener and an averaging pitot tube array. Between the damper and the flow station was added about ten feet of straight duct to reduce the turbulence from the fan and damper. This is not quite the recommended amount of straight duct, but space limitations prevented the addition of more.

One disadvantage of this type of flow element is that the measured pressure is a function of the square of the air velocity, as shown in the following equation:

$$\text{Flow Velocity (ft/min)} = 1097 \times \sqrt{\frac{\text{Velocity Pressure (inches of water)}}{\text{Air Density (lb/ft}^3\text{)}}} \qquad \text{(Equation 1)}$$

Thus, if the airflow rate is reduced by half, the velocity pressure is reduced by three quarters. Low flow rates then become increasingly difficult to measure accurately. The flow element was also sized to fit the full return duct area of 4 square feet, so the maximum velocity pressure at 3000 CFM is approximately 0.035 IW. This required the use of a precision low-range pressure transmitter (full span of 0.1 IW).

Another disadvantage is that the measurement does not account for other potential leakage sites between it and the supply flow measurement, including through the RTU exterior panels, interior leakage from the condenser side through penetrations for wire or tubing, or connecting duct leaks. ***Figure 17*** examines the difference between the flow measured at the return duct and that measured by the supply air flow measurement apparatus, along with the static pressure measured in the mixed air plenum. This test was done with an economizer damper in place but wide open to the return flow and with the outside air intake sealed with plastic sheeting and foil tape. The damper downstream of the return air fan was kept wide open, and the flow was varied by adjusting the speed of the booster fan on the supply air flow measurement apparatus.

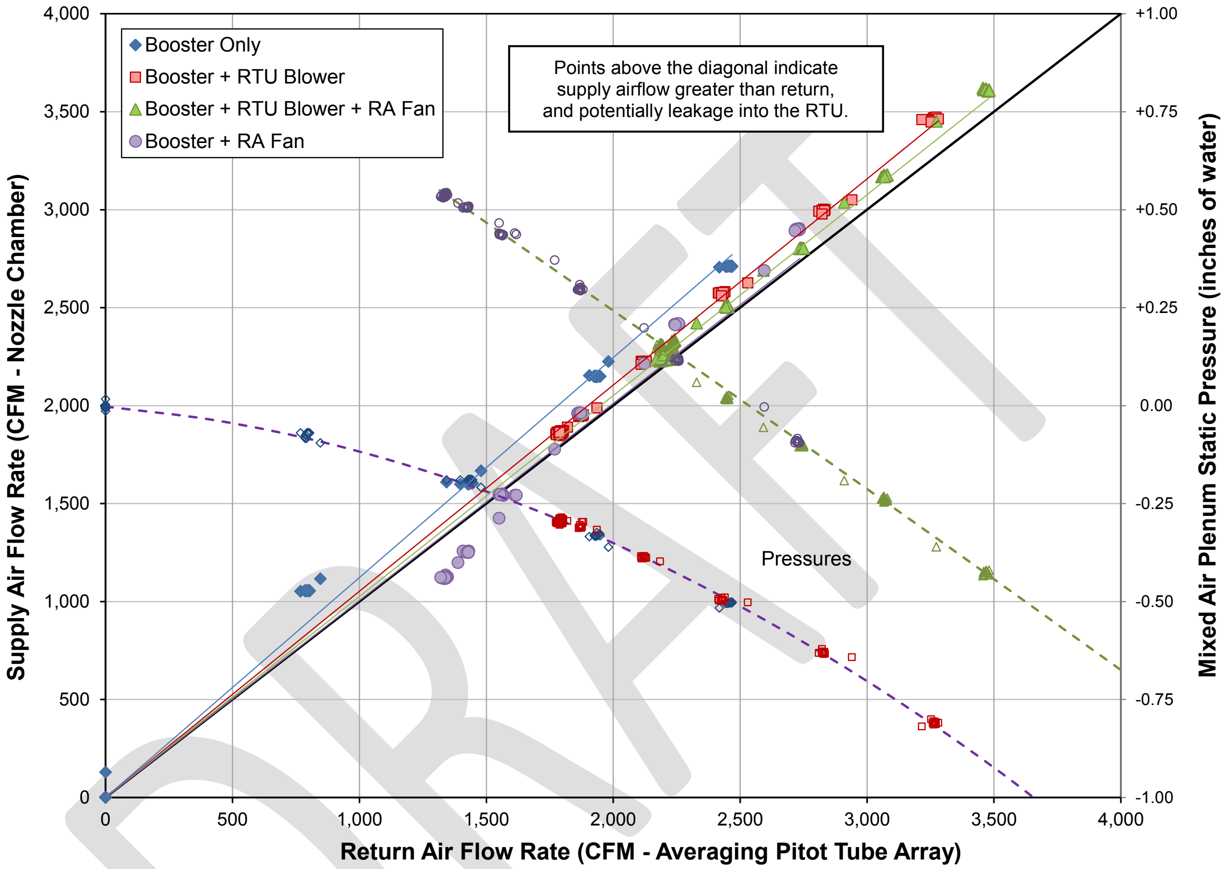
The first test case was with only the supply booster fan operating, and thus pulling negative pressure along the entire length of the flow path. At the maximum airflow rate, the supply flow appears to be nearly 10% higher than the return air flow, suggesting that additional outside air is being drawn in to the system somewhere in between. For the next set, the RTU's indoor blower was turned on, thus creating positive pressure at the supply outlet and negative pressure at the return inlet. The pressure indicated at the mixed air plenum for both of these cases followed the same trend, demonstrating the frictional pressure loss through the entire return duct (including fan, damper, flow grid, elbows and economizer return damper). The supply air flow was now much closer to the return air flow, suggesting that leaks *out* after the blower are compensating for the leaks *in* before the blower.

For the next two sets of data with the same combination of supply booster and RTU blowers, the return air fan was turned on, sometimes resulting in positive pressure in the mixed air plenum. These tests further reduced the difference between the two flow measurements as the negative pressure through the system is reduced or completely eliminated. At the maximum measured mixed air pressure (positive relative to the outside), the return air flow element actually showed higher flow than the supply, suggesting that air is leaking out of the system.

If there were no system leaks and the airflows were perfectly balanced, then the outside air fraction can be calculated as one minus the ratio of the mass flow rate measured at the return to the mass flow rate measured on the supply, as in the following equation:

$$\%\,Outside\ Air\ (RA\ Flow) = \left[1 - \frac{MFR_{RA}}{MFR_{SA}}\right] \times 100\% \qquad \text{(MFR = Mass Flow Rate)} \qquad \text{(Equation 2)}$$

FIGURE 17: COMPARISON OF RETURN AIR FLOW ELEMENT TO SUPPLY AIR FLOW



The measurement of any air flow rate is difficult to achieve with accuracy in the field, and is typically done with a flow grid similar to a pitot tube array, hand-held air velocity sensors, or a flow hood. In the field, a direct measurement of outside air to the economizer would likely be easier than the return air flow, but supply air flow would still need to be measured to calculate the outside air fraction.

Method 2: Measurement of mixed air temperature

The next method is to measure the temperature of the air in the mixed air plenum upstream of the evaporator coil. This method requires that there is sufficient mixing of the air in the plenum, which is unlikely given the short distances involved with this test unit. It also requires a fairly large temperature difference between the return air and outside air to achieve an accurate measurement.

Knowing that air mixing in the plenum is not very good, the average mixed air temperature measurement was improved through averaging measurements from multiple locations. For these tests, an array of twelve thermocouples was installed. To reduce the effect of radiation heat transfer to the coil (if cooling), the thermocouples were arranged on the front of the air filters. Ideally, the thermocouple measurements should be weighted by the air velocity measured at the same location as the thermocouple, such that measurements taken in dead zones are not included. In addition to the thermocouples, a 48-inch long flexible RTD that averaged temperature along its entire length was bent into a U-shape and also attached to the filter rack.

With measures of return (T_{RA}), outside (T_{OA}) and mixed air temperatures (T_{MA}), the outside air fraction can be calculated as a simple ratio of the temperature differences:

$$\%\ Outside\ Air\ (MA\ Temp) = \frac{T_{MA}-T_{RA}}{T_{OA}-T_{RA}} \times 100\% \qquad \text{(Equation 3)}$$

One advantage of this method is that it can be done with the system actively cooling, assuming the mixed air temperature sensor(s) can be adequately shielded from the cold coil.

Method 3: Measurement of supply air temperature

Because of the difficulty in accurately measuring an average air temperature in the mixed air plenum, an alternative is to measure the supply air temperature downstream of the blower. The advantage to this method is that the blower becomes an air mixer to create a more uniform temperature measurement in the supply duct. The disadvantage is that this method can only be used if the RTU is not actively cooling, and the evaporator coil is dry and has reached thermal equilibrium with the air flowing through it.

The other disadvantage is that this active blower heats the air by between 1 and 2°F. Again, if there is a substantial temperature difference between the return and outside, then this temperature rise has less of an impact. It can also be compensated for by subtracting it out. A more accurate estimate of the temperature gain can be calculated by dividing the power consumption of the blower (converted to Btus) by the airflow rate multiplied by the factor 1.08 Btu/hr-°F per CFM for standard air, as in the following equation.

$$\%\ Outside\ Air\ (SA\ Temp) = \frac{(T_{SA}-\Delta T)-T_{RA}}{T_{OA}-T_{RA}} \times 100\% \quad \left[\Delta T = \frac{Blower\ kW \times 3{,}412}{1.08 \times Supply\ CFM}\right] \qquad \text{(Equation 4)}$$

Method 4: Measurement of supply air humidity

If the return air and outside air conditions of temperature and humidity are plotted on a psychrometric chart and a line drawn between them forms a diagonal, then the outside air fraction can also be found by finding where the supply air humidity ratio intersects with this line. While Methods 2 and 3 used differential temperatures (the horizontal axis of the psychrometric chart), this method uses the differences in the humidity ratio (the vertical axis of the psychrometric chart; represented with "W"). This method thus requires a substantial difference in the humidity ratio between the return and outside, and also that the coil be off, completely dry, and warm enough to not cause condensation. The temperature rise caused by the blower has no effect on the humidity ratio, and the blower still provides mixing. This method also requires accurate sensors for determining the humidity ratio.

$$\%\ Outside\ Air\ (SA\ Humid) = \frac{W_{SA}-W_{RA}}{W_{OA}-W_{RA}} \times 100\% \qquad \text{(Equation 5)}$$

This method could be thought of as using water vapor as a tracer gas. The tracer gas method is another way to measure the outside air fraction, and works by injecting a gas that doesn't exist in large quantities in either the outside or return air stream into one of them at a controlled and measured flow rate, and then measuring the concentration of the gas in the supply using an instrument that is sensitive to small quantities of the gas. Without such an instrument, this method was not attempted. It would also be difficult to apply in an enclosed environment where the RTU supply air is recirculated back to the return along with the added tracer gas.

The test methodology used for the leakage tests was to set the return air to a combination of low temperature and low humidity (usually 70°Fdb/55°Fwb, but sometimes 75°Fdb/62°Fwb), set the outside air to a combination of high temperature and high humidity (usually the AHRI Standard condition of 95°Fdb/75°Fwb, but in later tests 100°Fdb/80°Fwb for a larger spread), for a temperature difference of between 20 and 30°F and a humidity ratio difference between 0.004 and 0.009 pounds of water vapor per pound of dry air. Testing then followed a matrix of varying:

- damper position (0, 15, 30, 45, 60, 75, and 100% open to outside air),
- supply static pressure (0.25, 0.50 and 0.75 IW), and
- return static pressure (constant zero relative to outside and uncompensated return duct friction loss – i.e. fan off and damper open).

For the tests where the friction loss through the return duct was left at whatever the duct friction created, the supply static pressure was adjusted to account for the changes in supply flow rate with the change in damper position. The specified supply static pressure was only applied to the initial test with the outside air damper closed. At this condition, a supply "duct friction factor" was calculated as the square of the airflow rate divided by the supply static pressure divided by the supply air density, as follows.

Duct Friction Factor = $\frac{CFM^2}{(\Delta P/\rho)}$

As the outside damper opens and the return damper closes, the return duct resistance lessens and the system preferentially uses outside air, resulting in an increase in supply flow rate. This was compensated for by adjusting the supply static pressure upwards to maintain the same duct factor, thus simulating the friction loss behavior of an actual duct to rising airflow.

Two after-market damper assemblies were run through this test scenario. Since the test RTU is a very old model, the damper assemblies were also of an older design, and do not follow the leakage specification set in the current version of the Title-24 building code[6]. The first was the damper installed during the Quality Maintenance service consisting of two blades in both the return and outside air position that operate in parallel and are connected with linkages. The actuator that drives the dampers is attached to the lower of the two outside blades. As the return damper opens, the flow is directed towards the filters and coil, while as the outside air damper opens, it directs the flow towards the return air and the bottom of the filters.

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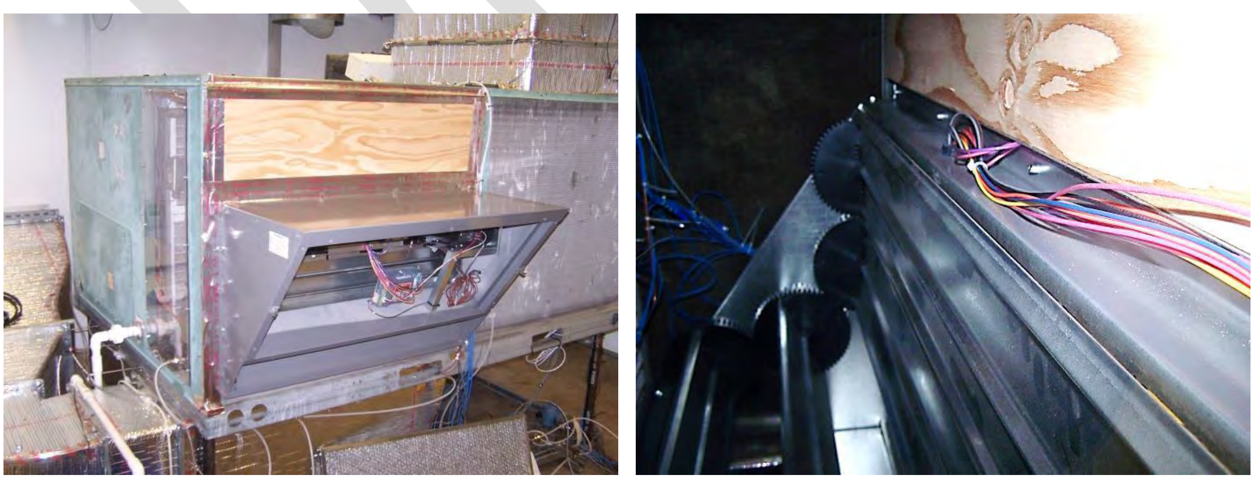
6 http://www.energy.ca.gov/title24/2013standards/nonresidential_manual.html, section 140.4(e)4: "Economizer outside air and return dampers shall be certified in accordance with AMCA Standard 500 to have a maximum leakage rate of 10 cfm/sf at 1.0 in. w.g."

FIGURE 18: LINKED PARALLEL BLADE DAMPER

The second damper assembly also had two blades in each position, but with a geared connection and blades that moved in an opposed rotation. The damper actuator connects to a drive gear rather than directly to a blade. The pair of outside blades opens outwards from the middle, which tends to direct the outside air to the top and bottom of the mixing chamber. The opening that is produced in the middle is also blocked somewhat by a support beam for the actuator that stretches across the entire opening. The return damper blades open up from the middle, tending to funnel the airflow to the center.

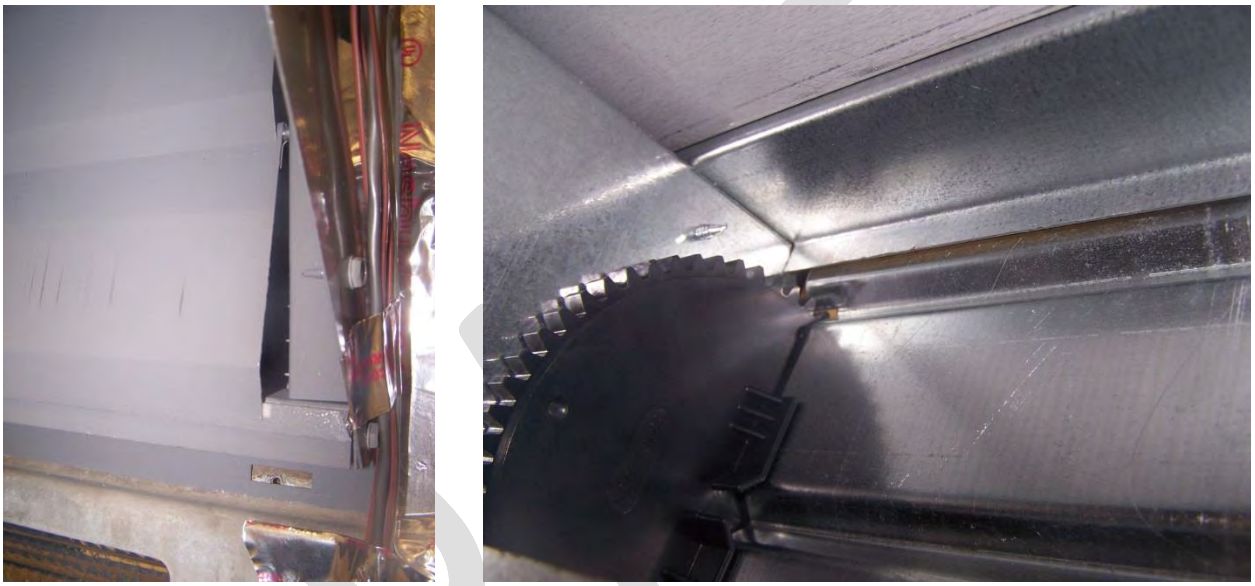
The parallel blade damper assembly included rubber seals on the blade edges to ensure a tighter closure, but these were not included with the opposed blade damper, possibly as an oversight. When the outside damper was in its fully closed position, there was still a visible gap along the top and bottom edges, although there was no gap where the two blades meet.

FIGURE 19: GEARED OPPOSED BLADE DAMPER



Both of the damper assemblies had several points of potential leakage of outside air. One of the most obvious is around the edges of the barometric relief damper at the bottom of both damper assemblies. The relief damper is needed because as the outside damper opens and brings outside air into the conditioned space, there needs to be a path for the displaced room air to escape to avoid over-pressurizing the space. This damper is merely a hinged flap of metal over an opening (two in the case of the second assembly). When the outside damper is closed and the indoor blower is operating, there usually exists negative pressure on the other side of this damper due to return duct friction, and outside air can seep in through the gaps on the side of the relief damper. Blade side edges and gaps for gears are other sites for unintended leakage when the outside damper is closed.

FIGURE 20: LEAKAGE POINTS IN GEARED OPPOSED BLADE DAMPER ASSEMBLY (LEFT: BAROMETRIC DAMPER EDGE, RIGHT: GAPS AT GEAR AND BLADE EDGE)



The results from the damper leakage tests are listed in ***Table 4*** and ***Table 5*** and plotted into six figures in the Appendix (***Figure A-1*** through ***Figure A-6***). The tables are grouped by the outside air fraction analysis method and by the damper assembly. (Of the four methods, Method 3 is believed to be the most accurate.) Each table is arrayed vertically by actuator position (percent open to the outside air, which means the percent open to the return air is 100% minus this number) and horizontally by the supply air pressure and return air pressure. For the return air pressure, the test case where the return duct pressure was allowed to float with the airflow friction loss is identified with the heading "Var" to signify that the return pressure was variable.

TABLE 4: LINKED PARALLEL DAMPER MEASURED OUTSIDE AIR FRACTIONS (% OF SUPPLY AIR CFM)

Method 1: Return Air Flow

	Supply Pressure (IW)	0.25		0.50		0.75	
	Return Pressure (IW)	0	Var	0	Var	0	Var
Actuator Position	0%	7	15	2	15	0	16
	15%	11	24	7	24	0	26
	30%	24	44	19	47	12	51
	45%	33	57	34	62	28	68
	60%	50	69	48	74	51	81
	75%	69	82	70	89	75	90
	100%	90	90	100	100	100	100

Method 2: Mixed Air Temperature

	Supply Pressure (IW)	0.25		0.50		0.75	
	Return Pressure (IW)	0	Var	0	Var	0	Var
Actuator Position	0%	6	10	6	11	6	12
	15%	16	23	16	24	16	25
	30%	44	51	43	53	44	57
	45%	60	66	60	69	60	74
	60%	73	73	73	76	74	82
	75%	84	84	84	89	85	90
	100%	93	93	98	98	99	99

Method 3: Supply Air Temperature

Supply Pressure (IW)		0.25		0.50		0.75	
Return Pressure (IW)		0	Var	0	Var	0	Var
Actuator Position	0%	8	12	8	12	8	13
	15%	15	22	14	22	14	23
	30%	32	41	32	42	33	46
	45%	49	55	48	57	49	64
	60%	66	66	65	70	66	78
	75%	80	80	80	87	80	87
	100%	92	92	97	97	97	97

Method 4: Supply Air Humidity

Supply Pressure (IW)		0.25		0.50		0.75	
Return Pressure (IW)		0	Var	0	Var	0	Var
Actuator Position	0%	8	14	7	14	8	15
	15%	16	23	12	22	13	24
	30%	29	40	30	41	30	45
	45%	45	52	44	54	45	60
	60%	59	61	59	65	60	72
	75%	70	72	69	78	72	80
	100%	79	79	87	87	89	89

TABLE 5: GEARED OPPOSED DAMPER MEASURED OUTSIDE AIR FRACTIONS (% OF SUPPLY AIR CFM)

Method 1: Return Air Flow

Supply Pressure (IW)		0.25		0.50		0.75	
Return Pressure (IW)		0	Var	0	Var	0	Var
Actuator Position	0%	2	20	0	22	0	10
	15%	5	29	2	31	0	21
	30%	10	40	8	45	0	31
	45%	18	49	15	53	16	40
	60%	37	58	35	65	33	50
	75%	55	68	56	75	52	65
	100%	75	75	78	78	75	77

Method 2: Mixed Air Temperature

Supply Pressure (IW)		0.25		0.50		0.75	
Return Pressure (IW)		0	Var	0	Var	0	Var
Actuator Position	0%	4	16	8	17	8	14
	15%	10	23	11	23	14	24
	30%	17	33	20	36	26	34
	45%	29	44	33	49	39	44
	60%	51	58	53	65	65	58
	75%	74	73	75	76	77	77
	100%	91	91	91	91	97	96

Method 3: Supply Air Temperature

Supply Pressure (IW)		0.25		0.50		0.75	
Return Pressure (IW)		0	Var	0	Var	0	Var
Actuator Position	0%	7	20	9	21	10	18
	15%	12	30	13	30	16	29
	30%	19	41	23	43	28	39
	45%	32	50	35	53	42	47
	60%	51	60	54	64	62	57
	75%	70	72	72	75	75	75
	100%	91	91	89	89	95	94

Method 4: Supply Air Humidity

Supply Pressure (IW)		0.25		0.50		0.75	
Return Pressure (IW)		0	Var	0	Var	0	Var
Actuator Position	0%	8	26	10	20	9	20
	15%	13	37	14	28	16	29
	30%	21	48	24	38	29	39
	45%	32	57	37	48	43	47
	60%	53	68	56	58	63	57
	75%	75	82	78	69	78	77
	100%	88	88	100	100	98	96

Of primary interest in these results is the amount of leakage air when the outside damper is closed. For the test cases where the return duct friction loss was compensated for by holding the return pressure to zero relative to the outside room using the return fan and damper, the leakage rate was usually below 10% by all of the analysis methods. However, when the duct friction was allowed (and thus creating a lower pressure in the mixed air plenum), the leakage rate increased to between 10 and 20%. Since this the in-situ scenario, this leakage has a significant impact on operating efficiency and capacity. Technicians who implement a minimum OA setting of 15% damper open are over-ventilating under almost all conditions.

Figure 21 and ***Figure 22*** show the measured outside air fractions by the four methods for a closed damper as a function of the pressure drop across the damper, or rather the negative static pressure in the mixed air plenum. This pressure tap was through the panel where the RTU would have connected to a horizontal duct, and in a corner out of the major airflow streams. It may still have some influence from turbulence, adding to the variability of the outside air fraction measurements. The trends observed from the results show a much higher leakage rate for the opposed blade damper (second figure) than the parallel blade damper.

FIGURE 21: CLOSED DAMPER LEAKAGE RATES AS A FUNCTION OF PRESSURE DIFFERENTIAL – PARALLEL BLADE

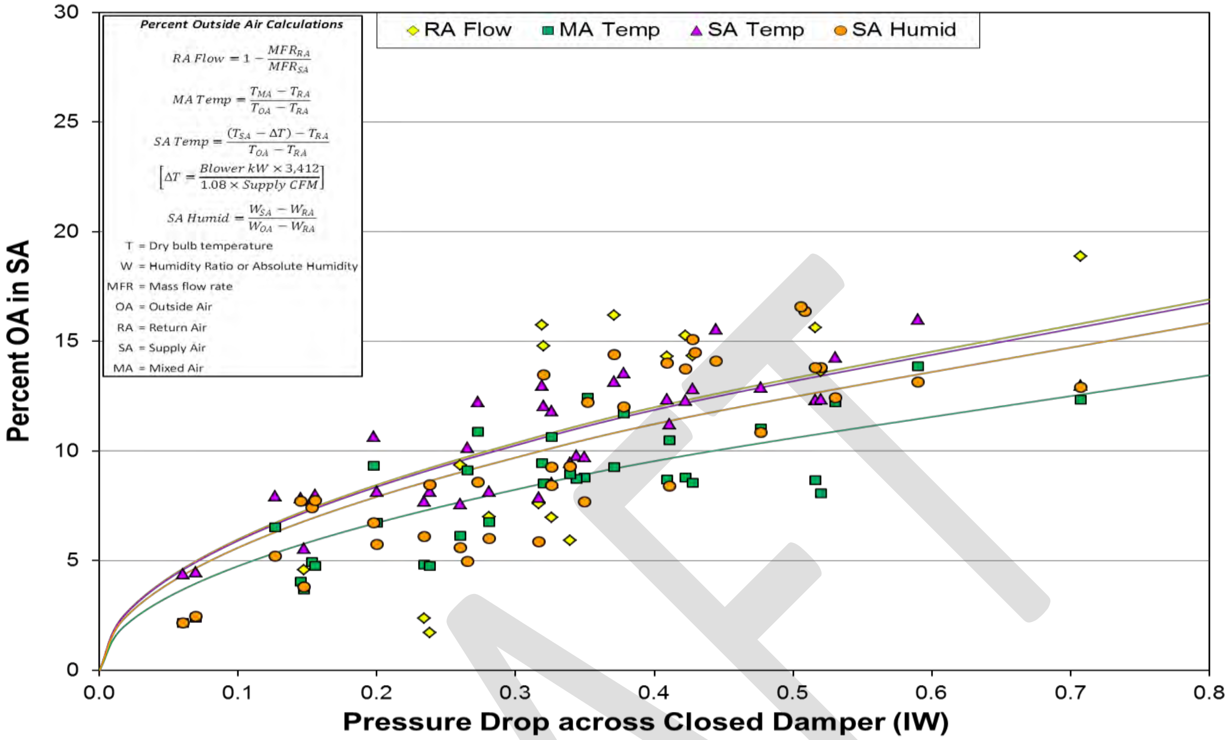
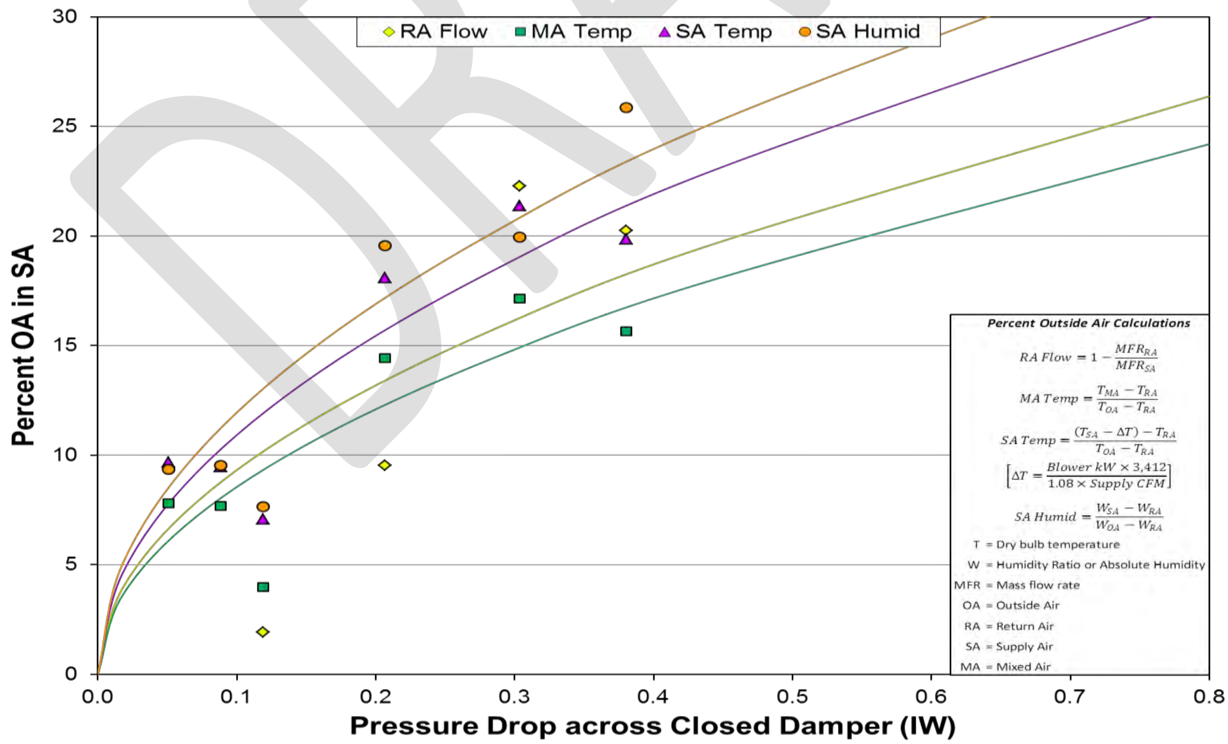


FIGURE 22: CLOSED DAMPER LEAKAGE RATES AS A FUNCTION OF PRESSURE DIFFERENTIAL – OPPOSED BLADE



The 2013 version of the California Title-24 building code (section 140.4(e)4) specifies a maximum closed damper leakage rate of 10 CFM per square foot of intake area at 1 IW differential pressure. (The rate will decrease as a function of the square root of the differential pressure; thus at 0.5 IW, the maximum leakage rate would be about 7 CFM/ft².) The damper openings for the two damper assemblies are about the same, with the parallel blade economizer outside air damper having an open area of 3.11 ft² and the opposed blade damper at 3.03 ft². With these areas, the maximum leakage rate would be close to 30 CFM at 1 IW (or 21 CFM at 0.5 IW). With a supply airflow rate for these tests in the range of 1,500 to 3,000 CFM, the outside air fraction limitation would be less than 2%; which is significantly less than what was measured. However, not all of the air leakage can be attributed to the damper, as there are other avenues by which outside air can leak in to the RTU, such as the relief damper and around ill-fitting panels.

In ***Figure 23*** and ***Figure 24***, the data shown in *Figure 21* and *Figure 22* have been converted into the units used by Title-24, by multiplying the outside air fraction by the measured supply air flow rate, dividing by the damper face area and by the square root of the mixed air plenum pressure to normalize the result to 1 IW, as follows.

$$\text{CFM/ft}^2 \text{ @ 1 IW} = \frac{Supply\ CFM \times \%OA}{Damper\ Area \times \sqrt{\Delta P}}$$

The result should be relatively constant, but the actual values vary widely due to the afore-mentioned measurement errors and also because of the pressure division, especially at low values. *Figure 23* for the parallel blade damper using the supply air temperature method (Method 3) shows the most consistent result with an average around 130 CFM/ft² @ 1 IW, or 13-times the Title-24 limit. Again, these aftermarket dampers followed an old design for this old RTU and newer designs should have better performance, but this may be indicative of the current state of many field installations.

Figure 25 shows the effect on several temperature measurements from sealing off the outside air intake with plastic, including the barometric relief damper. This is for the closed opposed blade damper assembly, and with return duct friction allowed thus creating a lower pressure in the mixed air plenum. This example shows the removal of the plastic rather than its application because it was quicker. Prior to removal, the average mixed air temperature measured by either the array of twelve thermocouples (MAT-TC avg) or by the long averaging RTD (MAT-RTD) were both consistent with the return air temperature (RAT). The supply air temperature (SAT) was higher because of the blower heating. After removal, the mixed air temperature rose and the individual thermocouple readings spread out as different regions received differing amounts of leakage air from the outside. This exercise just emphasizes the inherent inaccuracy in the measurement of mixed air temperature.

FIGURE 23: CLOSED DAMPER LEAKAGE RATES IN TITLE-24 UNITS – PARALLEL BLADE

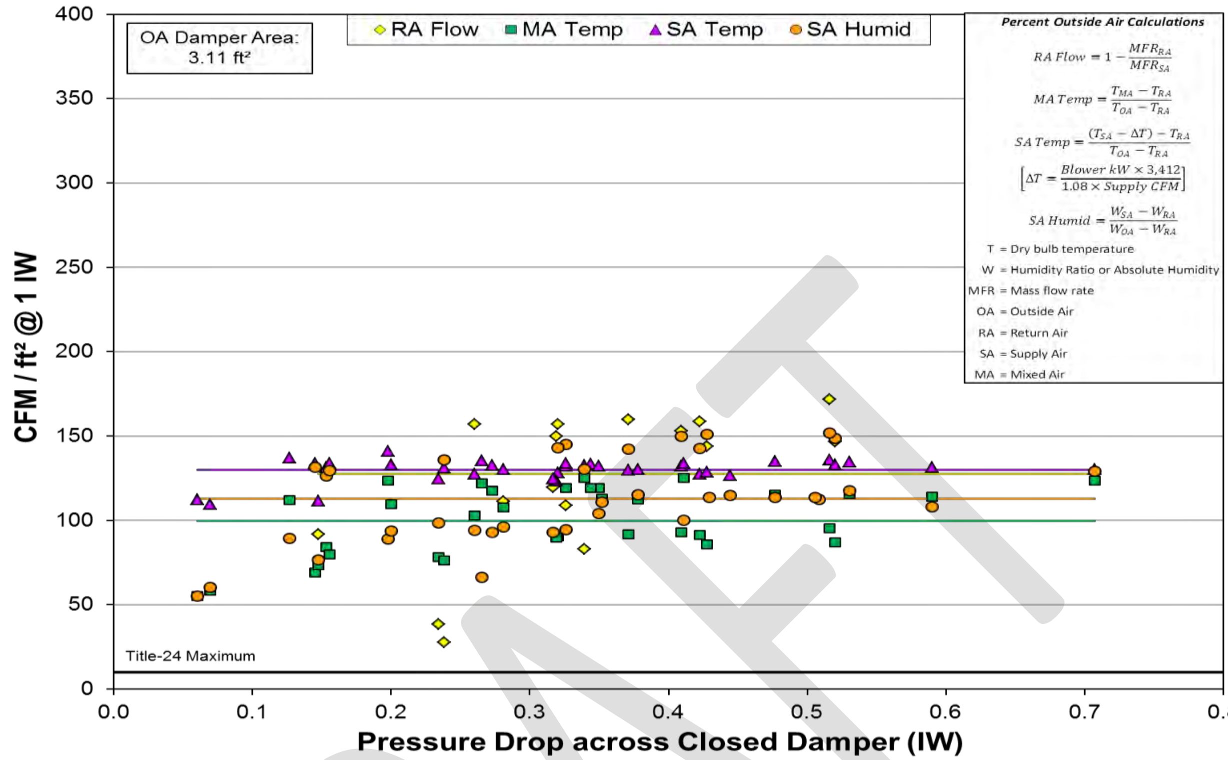


FIGURE 24: CLOSED DAMPER LEAKAGE RATES IN TITLE-24 UNITS – OPPOSED BLADE

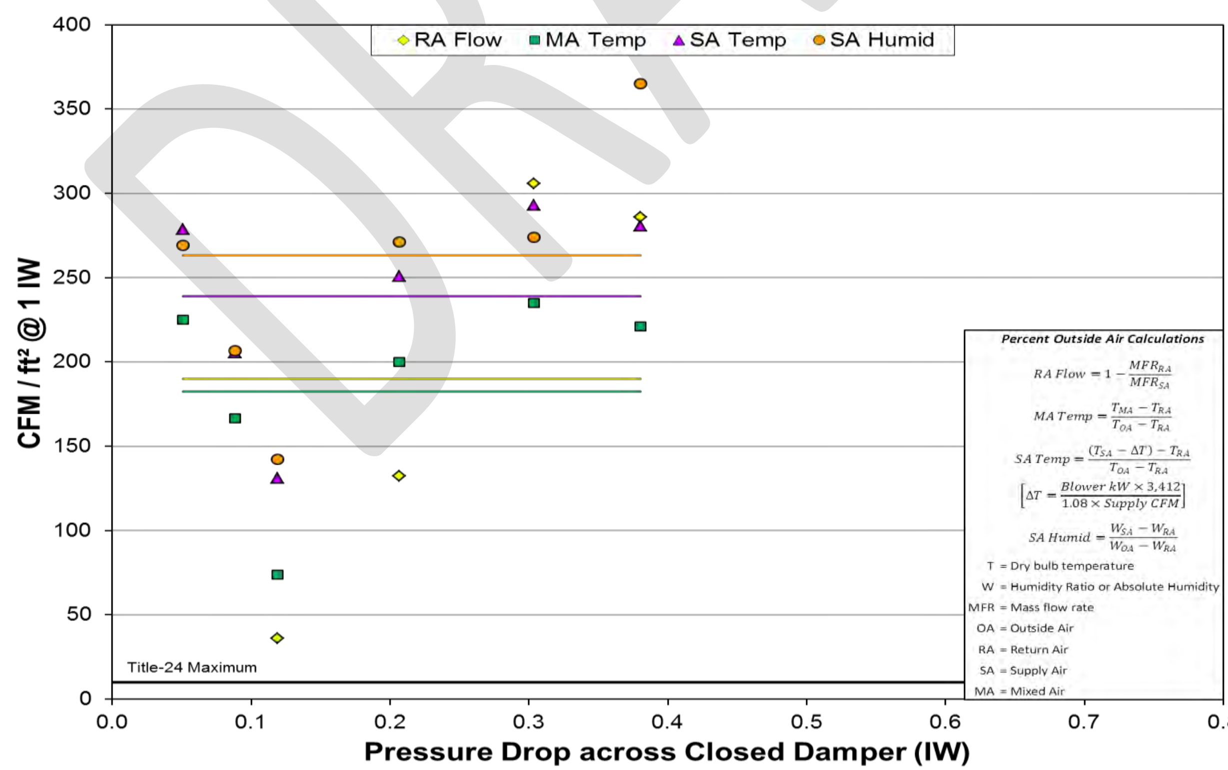
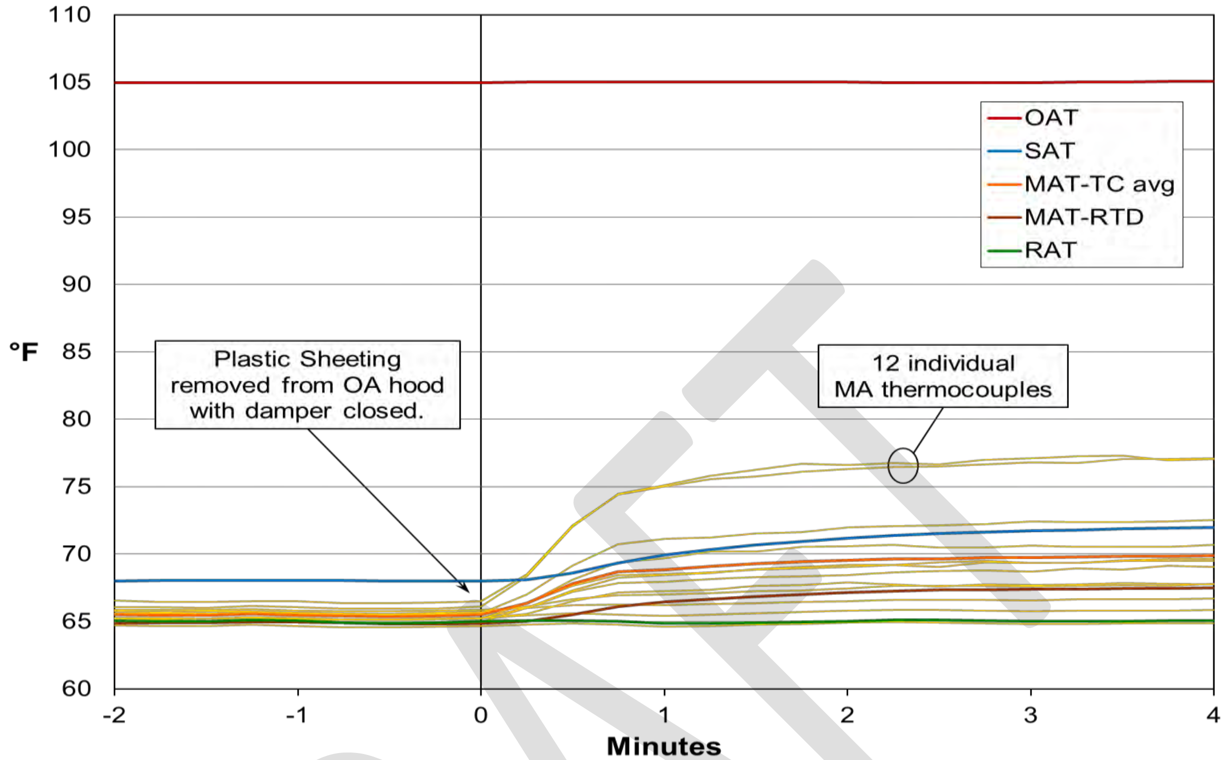


FIGURE 25: EFFECT OF SEALING OA INTAKE ON GEARED OPPOSED BLADE DAMPER ASSEMBLY WHILE CLOSED



The figures of the test results in the Appendix include what the negative pressure was at the return when the outside air damper was fully closed, which varied from about negative 0.3 IW at the supply pressure of 0.25 IW, down to about negative 0.2 IW at the supply pressure of 0.75 IW. Thus, the higher supply static resulted in a lower pressure difference between the outside and the mixed air plenum, and less leakage into the unit from other sites.

Also of interest is the case where the outside damper is fully open and the outside air fraction of the supply air should be 100%. The reason that many of the calculation methods do not show this is mainly the result of over-pressurizing the indoor room (relative to the outdoor room), which pushes some air back through the return duct. Most of this excess is exhausted through the barometric relief damper back into the outdoor room, where it can potentially be recirculated back in through the open outside air damper above it, but some air also manages to leak past the closed return damper. The open outside air damper still creates a small pressure drop, which imparts a negative pressure in the mixed air plenum relative to the return that can draw air through the gaps around the return damper. (For the tests with the outside damper fully open, there was usually only one test done with the return duct fan off - the "Var" column - since the measured static pressure at the unit return was zero even when uncompensated. Where there is a difference, it is because one test was done at the specified supply pressure and the other was done with the supply pressure adjusted for the rise in supply airflow.)

The variation between the individual mixed air thermocouple measurements is examined visually in several tables following the figures in the Appendix (***Table A-2*** through ***Table A-7***). The tables represent the face of the filter rack in the RTU with return air entering from below and with the outside air from the same direction as the view. Shown in a grid are the twelve thermocouple measurements in their specific positions, which have been colored according to their value relative to the applied return and outside temperatures (blue for return and red for outside), along with the absolute range between the maximum and minimum reading. The tables also show the average of the twelve thermocouple measurements, the reading from the long averaging RTD placed alongside the thermocouples (which should read close to the thermocouple average), the supply, return and mixed air plenum static pressures, and the supply flow rate. Finally, the tables include the values of the outside air damper position setting, the return, and outside temperatures, and the outside air fraction calculated using the thermocouple average.

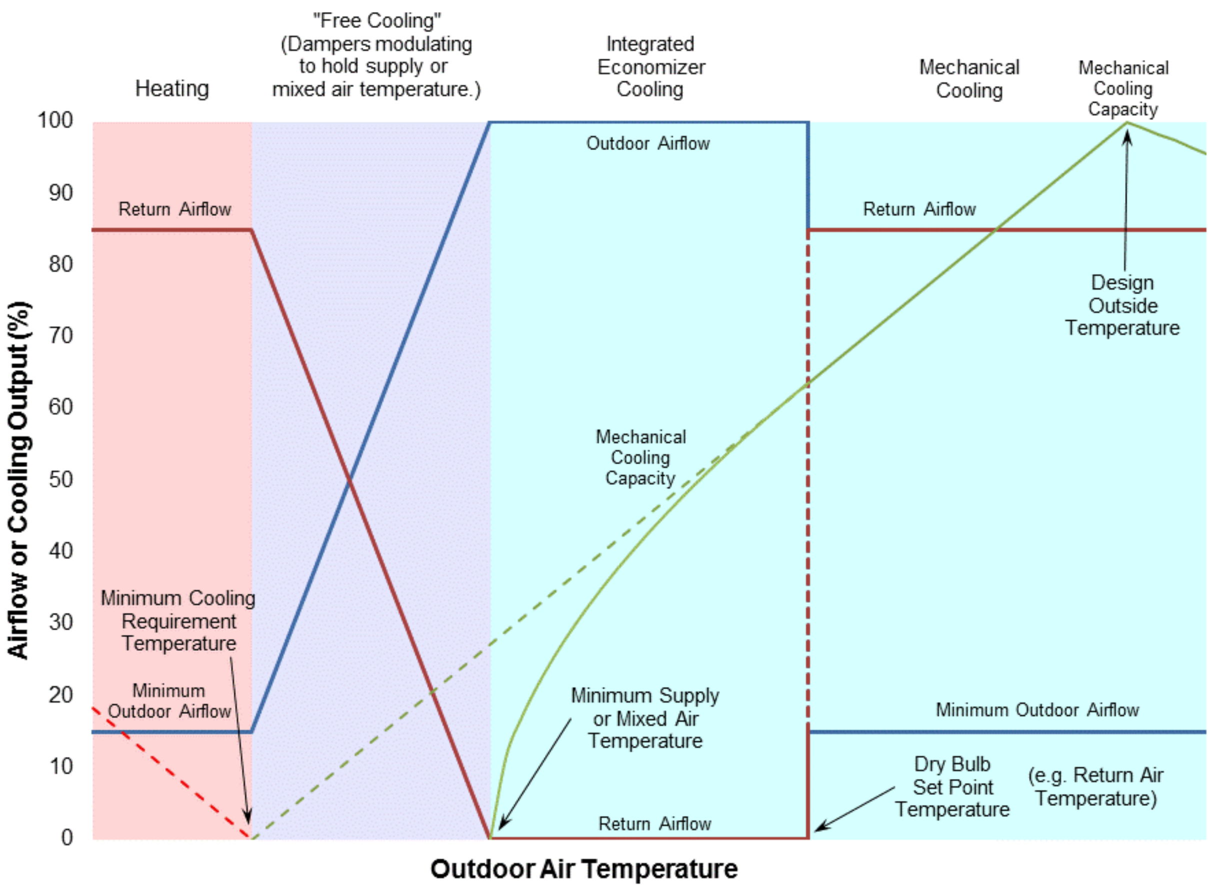
These tables show that for the parallel blade damper, the outside air begins to influence the thermocouples towards the top of the filter panel first with the bottom remaining close to the return air, but the influence moves down to the middle as the damper opens wider. Towards the end of the sequence (75% open), the top of the filters are actually cooler than the bottom, which may be an indication of air stagnation. The opposed blade damper follows a similar pattern, except that the bottom of the filter bank stays cooler longer, perhaps because with this damper the airflow is less directional. For both damper assemblies, the range of the measurements is greatest at the intermediate points of the damper opening when there is roughly the same quantity of return and outside air. The main point of this is to emphasize that obtaining an accurate measure of the mixed air temperature is very difficult, particularly if a single measurement point is used.

PHASE 3B: ECONOMIZER CONTROLLER PERFORMANCE

The performance of the damper assembly in allowing or preventing outside air from entering involves only one part of an economizer. The other part is the "brains" that tells the dampers when to open or close. Many of the field problems can be traced to this controller module and the difficulty with properly setting it up.

Figure 26 describes the general function that an economizer is intended to perform. When an air conditioner system is designed, its rated capacity is selected based on continuous operation at design specifications of room temperature and outside climate, taking into account various space loads including ventilation. Starting at the far right of the figure, when the outside temperature rises above the building cooling system design outside temperature, the output of the air conditioner begins to drop below 100% of its rated capacity as demonstrated by the earlier performance mapping, and it cannot maintain the design room temperature. Below the design outside condition, the unit has sufficient capacity to maintain the room temperature and will modulate its compressor operation (usually by cycling) as controlled by the room thermostat. In these regions, the outside air damper is set to a point that will allow for the minimum level of ventilation air required by code for the conditioned space. When the outside air temperature (or in some cases the combination of temperature and humidity represented by enthalpy) drops below that of the return air or some other defined set point, the outside air damper moves to a fully open position and the return air damper closes to take advantage of the lower energy air source from the outside. The compressor operation continues to modulate, but with decreasing on-time because of the decreasing temperature (or enthalpy) of the mixed air reaching the coil. Eventually, the outside air temperature reaches the supply (or mixed) air temperature lower limit, and the compressor operation can completely stop. Between this temperature and the temperature at which the conditioned zone switches from requiring cooling to requiring heating is the zone of "free cooling", where the cooling load is being satisfied with only ventilation air. In this zone, the damper will modulate to blend the room return air with outside air to meet the supply or mixed air minimum temperature. Once the conditioned space transitions to heating mode, the damper returns to the position of supplying the minimum amount of outside air for ventilation. In the figure, the area between the actual mechanical cooling capacity (represented by the solid green curve) and the capacity that would be required if the unit had no economizer and always used return air (represented by the dashed green line) is the area where energy savings can be achieved from using outside air.

FIGURE 26: GENERAL ECONOMIZER FUNCTION

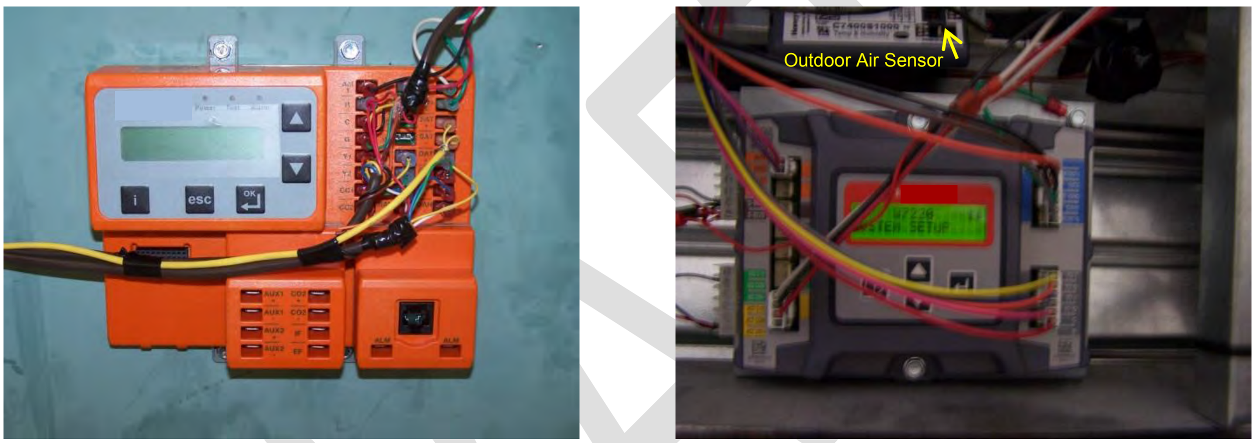


This testing program included two recently introduced integrated economizer controllers that meet the requirements of the Title 24 Building Energy Efficiency Standards, and have been designed with added features for ease of use, such as an LCD display and menu driven settings. They also have ability to use and report from multiple sensors and identify faults in programming or operation in the display or by warning lights or even remote indicators. The term "integrated" refers to the controller as also being able to interrupt operation of the compressors in addition to operating the damper assembly. The systems are normally wired in between the thermostat and the control circuitry of the RTU. For the lab tests, no thermostat was used, and the signals for blower operation (G) and the two stages of cooling (Y1 and Y2) were controlled by manual relays. For evaluation of the controllers, these signal wires were connected to input terminals on the controllers with the "G" wire giving an indication of the space being occupied as well as operating the blower. The controllers then provided controlled outputs for the compressors as well as the damper actuator.

There is no established testing standard for economizers, so a test methodology was created to demonstrate how the controller responds to changes in the outside air conditions.

The procedure was to begin with the outdoor room temperature at a stable, high point where the outside damper would be in its minimum open position (the right side of *Figure 26*), and then ramp the temperature down at a slow but constant rate (1°F every 5 to 10 minutes) to a low temperature point at which the economizer should be in the free cooling and modulating mode. After a period of time at this temperature, the ramp is reversed back to the high temperature condition. Because of the appearance of the trend of outside temperature when graphed over time, this method has been nicknamed the "V" test. Throughout the sequence, the simulated thermostat signals of occupied (G) and calls for both stages of cooling (Y1 and Y2) were held constantly active, although in a real application this would likely not happen. The room condition control program was modified to allow for automatic application of the temperature trends, and due to their long duration, the tests were often run overnight unattended.

FIGURE 27: DIGITAL ECONOMIZER CONTROLLERS



ECONOMIZER #1

The first unit evaluated was paired with the parallel blade damper assembly. The high change over set point is chosen automatically by an entry of the Zip Code for the unit's location, and is then used to look up what is required by code for that location, relieving the technician of having to look it up. It can be overridden through the setup menus. For the test unit, the Zip Code of the lab was used (94583), which corresponds to California Climate Zone 12.

The unit takes sensor measurements at three locations on the RTU: outside air, return air and supply air. The supply air sensor measures temperature only, but the outside and return air sensors can be either temperature only sensors or temperature and humidity sensors for calculating enthalpy. With just the temperature sensors attached, the unit can operate in either fixed outside or differential temperature change-over mode. With the enthalpy sensors attached, the unit can be programed to operate in either differential or fixed outside enthalpy mode, or in differential or fixed outside temperature mode and ignore the humidity measurements. The readings from all of the sensors (including temperature, humidity, and enthalpy from the combination sensors) are available for display on the front panel.

The test unit included the two combination temperature and humidity sensors for the return and outside air, and a single temperature sensor for the supply air. The entire assembly was installed and the sensors placed by the technicians who performed the Quality

Maintenance service. The outside air sensor was placed near the damper actuator but not directly in the outside air stream, the return air sensor was dropped into the return duct underneath the unit, and the supply air temperature sensor was placed in the indoor blower cabinet because there it would sense the coldest temperature before picking up a degree or two from the blower.

The first two "V" test trends (shown in ***Figure 28***) were conducted with the system programmed to operate in differential temperature mode, with slightly different operating conditions. The first test was with the return or indoor room temperature (RAT) held at 75°F and the outside air temperature (OAT) was ramped at a rate of 1°F every 5 minutes from 80°F down to 50°F and then back up to 85°F. In the second test, the same ramp rate was used, but it began at 90°F, ramped down again to 50°F and then back up to 90°F. The return air was also held 5°F higher at 80°F dry bulb, and its wet bulb temperature was maintained at 67°F (AHRI Conditions).

Because of the lower return temperature and beginning outside temperature in the first test, the supply air temperature (SAT) dropped sooner into the zone where a "compressor protection and energy savings" strategy is triggered by the economizer.[7] In this strategy, the second stage compressor is cycled off for 3 minutes and back on for 3 minutes; so even though the thermostat signal into the economizer is calling for both stages of cooling, the economizer has determined that the supply temperature is cool enough without needing to run both compressors constantly. This results in some energy savings before the dampers are even moved. There was less of this cycling of the stage 2 compressor in the second test because with the higher return air temperature, the supply air temperature did not reach the threshold as soon, and stayed higher until the cooling capacity gain from the decreasing outside temperature could bring it down.

Eventually, the outside temperature drops below the differential temperature threshold set by the Zip Code to switch to economizing mode by opening the outside air damper and also shutting off the second stage compressor completely. In the first test, this occurred at an outside temperature of 66.0°F (9.0°F differential from the return air) and in the second test it occurred at an outside temperature of 67.6°F (12.4°F differential). The gain in cooling capacity resulting from the lower outside air temperature to the condenser combined with the lower temperature air entering the cooling coil means that the supply air temperature requirement could be met without needing to run the second compressor, even though the simulated thermostat signal was still calling for it.

As the outside temperature continued to drop, the supply air temperature reached another threshold where the economizer decided it was not necessary to run the first stage compressor either and could provide cooling with just ventilation air. It also began modulating the damper to keep the supply air temperature at a minimum value. With just the blower consuming power, the effective system efficiency (EER) using a capacity based on the difference between the supply and return or room air enthalpy was on the order of 50 Btu/Wh, versus its rating of around 9.

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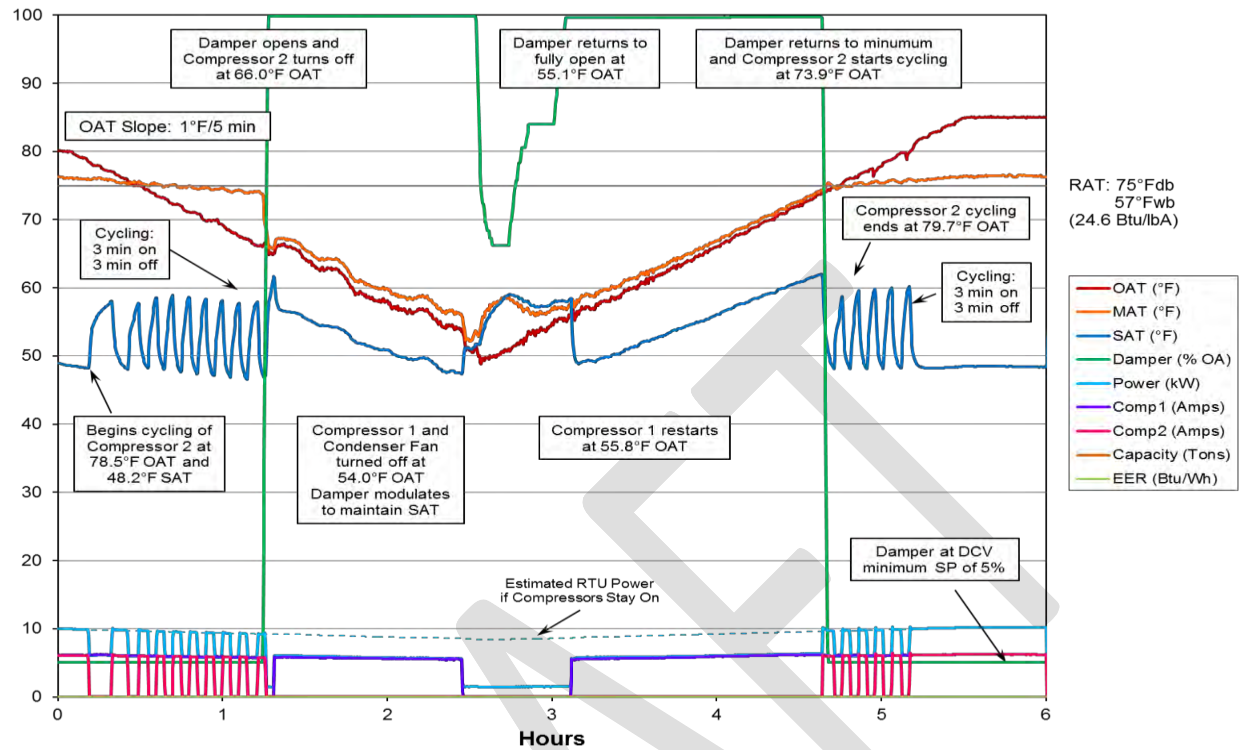
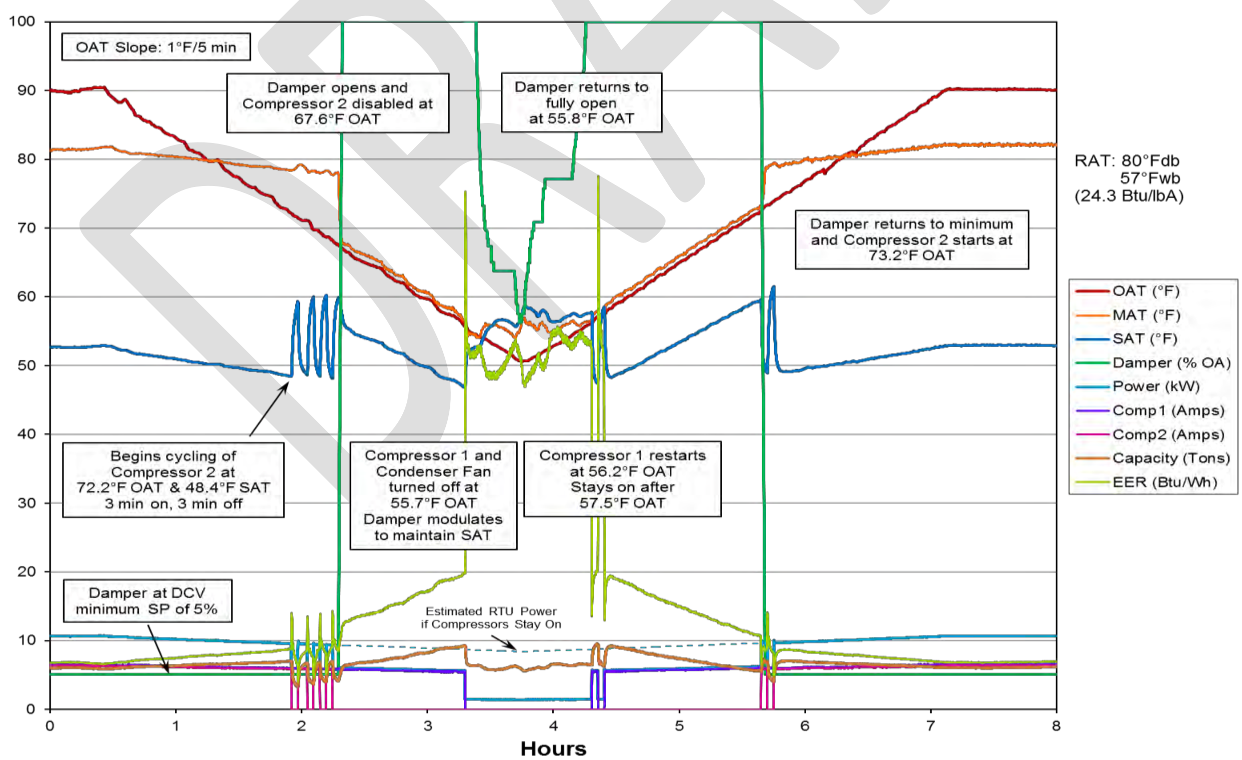
[7] The operation manual for the unit says that the threshold for this is a supply air temperature of 47°F, and the reason why the charts show the effects from this operation starting at slightly over 48°F is because the chart data is the measured supply air temperature from the RTU while the economizer sensor is in the blower cabinet, so the difference is the air heating by the blower.

The ramp back up in temperature was almost a mirror image of the ramp down, with the exception in the second test of one cycle of the primary compressor after coming out of the free cooling zone as the supply air temperature dropped too low during the initial run. There were again fewer cycles of the secondary compressor in the second test. Out of the six hours of the first test, the RTU was affected by the economizer (between the first and last cycle of the secondary compressor) for nearly 5 hours and consumed 29.4 kWh in that period; while if it had operated in continuous mode without any outside air but accounting for the change in power with outside temperature (as indicated by the dashed line in the figure), it would have consumed 45.7 kWh; a savings of 16.3 kWh or 36%. For the second test, the RTU had 3 hours and 50 minutes of the eight hour test period in which it was affected by the economizer and consumed 18.7 kWh while in this mode, but would have consumed 34.6 kWh over the same period without access to the outside air, for a savings of 15.9 kWh or 46%. This does not account for the likelihood that the room thermostat would have been satisfied and stopped calling for compressor operation.

FIGURE 28: ECONOMIZER #1 "V" TEST – DIFFERENTIAL TEMPERATURE MODE

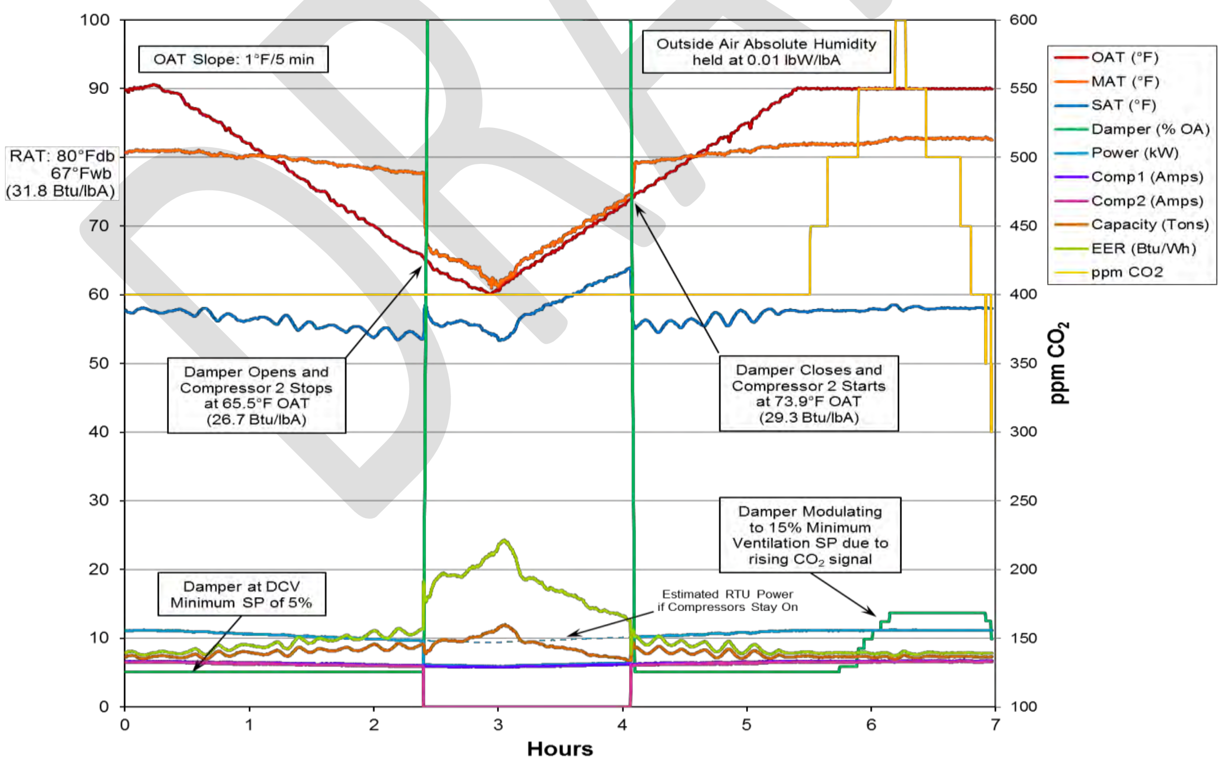




One additional test was run with this economizer, but this time switched to differential enthalpy control mode. In the previous tests, the absolute humidity (or humidity ratio) in the outdoor room was allowed to float with the ambient humidity; while in this test, the absolute humidity was held at a constant 0.010 pounds of water vapor per pound of dry air and the air temperature was again ramped down and up. The outside temperature was only lowered to 60°F rather than the 50°F achieved in the previous tests, and the supply air temperature never dropped to the minimum threshold for compressor cycling. When the economizer did switch the dampers over to outside air, it again shut off the second stage compressor and kept it off until the time when the outside temperature rose to the point at which the damper was returned to its minimum position. In this test scenario, that period lasted for 1 hour and 40 minutes, during which time the RTU consumed 10.2 kWh when it would have used 16.2 kWh if it had not shut off the second compressor; a savings of 6 kWh.

For the latter part of this test, a quick check of the demand controlled ventilation (DCV) function of the economizer was conducted. The economizer uses a second minimum damper position set point to allow the damper to be opened beyond the usual minimum position when a remote carbon dioxide (CO_2) sensor signals that the space air is stale and additional ventilation air is needed. For this trial, the second position was set to 15% open versus the 5% base minimum, and the DCV CO_2 concentration set point was put at 500 ppm (parts per million). The CO_2 sensor was simulated with a controlled voltage signal proportional to the concentration, and raised in steps from the baseline of 400 ppm to 600 ppm. The economizer reacted to the signal by slowly opening the damper to the second position, after some time lag. The effect from this is also seen in a small rise in the mixed air temperature (MAT) as more of the warmer outside air is brought in.

FIGURE 29: ECONOMIZER #1 "V" TEST – DIFFERENTIAL ENTHALPY MODE



ECONOMIZER #2

The second economizer profiled was paired with the opposed blade damper assembly. (Actually, these two components arrived as a complete packaged system from the aftermarket economizer manufacturer.) The original configuration of the economizer included just two temperature-only sensors: one for the outside air and one for the mixed air. The outside air sensor came attached to the support beam that also supports the actuator, and is in the airflow path through the outside air damper (see the right photo in *Figure 27*). The instructions were not specific about where to place the mixed air sensor, so it was located approximately in the middle of the filter rack amongst the test thermocouple array.

For the first attempted "V" test, the system was left with its factory default settings. This includes a damper changeover set point of 63°F, a minimum mixed air set point of 53°F for modulating the dampers, and a low temperature compressor lock-out set point of 32°F. The unit also defaults to a 2-hour timeout on the second stage compressor. That compressor will again shut off when the unit opens the outside air damper, but the economizer will bring it back on after the delay period if the thermostat is still calling for it (signal on Y2).

The test followed much the same scenario as before, with the return or indoor room air held to a constant AHRI Standard 80°Fdb/67°Fwb, and with the outdoor room air beginning at a high temperature of 90°F, ramping down at a constant rate to a low of 50°F at which point it stayed for a time before ramping back up to 90°F. The ramp rate for this test was halved from that used in the previous tests to 1°F every 10 minutes instead of every 5 minutes to increase the test duration. Because of this long duration, the tests on this economizer were usually run overnight without supervision, which had some consequences.

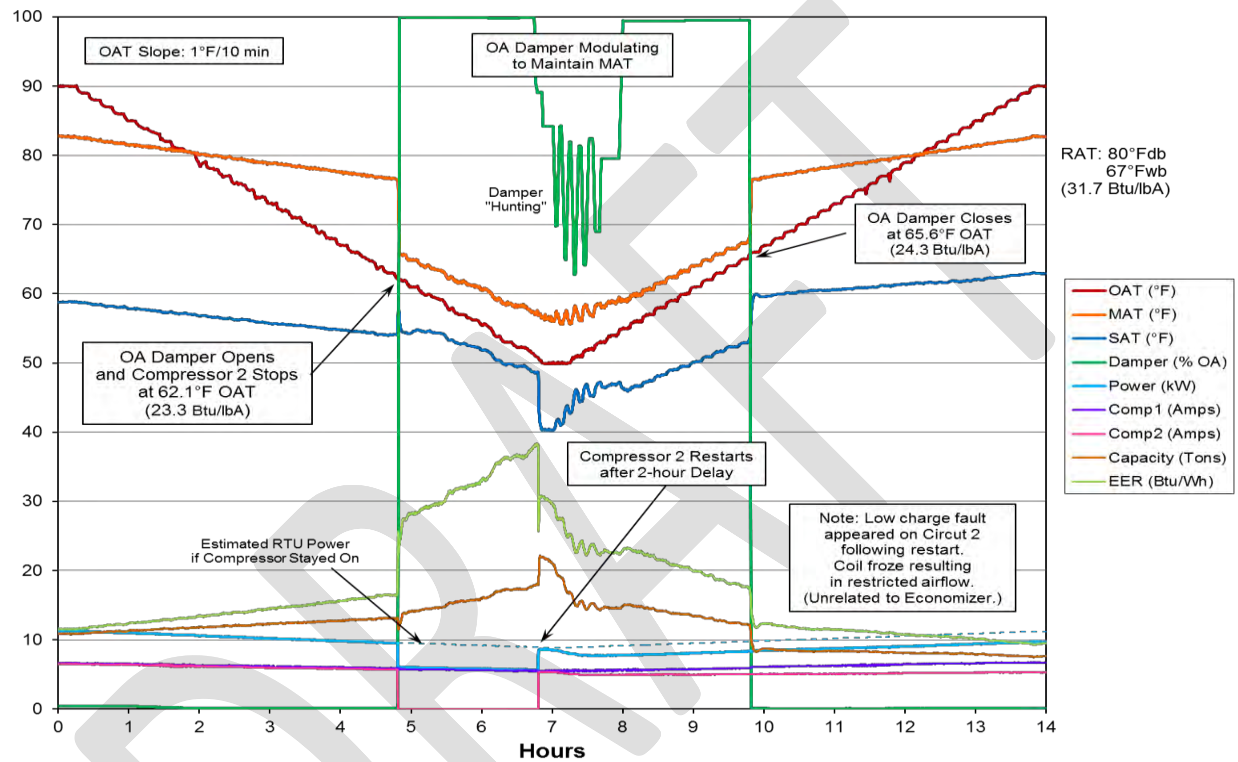
The default settings also used an output value of 2.8 volts to the damper actuator for its minimum position. If the actuator had a normal range of 2 to 10 volts representing zero to 100% open, then the default would represent 10% open. However, for this assembly, the actuator did not start to move the dampers until it received a signal close to 3 volts, which meant the minimum damper position was actually closed. As demonstrated by the earlier damper leakage tests, this is not necessarily a bad thing for this damper, which had more leakage than the other.

The results from this test are shown in ***Figure 30***. Even with a closed outside air damper, the mixed air temperature trend shows the result of leakage and the mixing between the fixed return air temperature and the ramping outside air temperature, with a ramp of its own. The unit started economizing when the outside air reached 62°F by opening the outside air damper and shutting down the second stage compressor. Since the simulated thermostat control signals were always on, the economizer brought the second stage compressor back on after the default 2-hour delay, despite producing a very low supply temperature which this configuration of the unit does not measure. With the second stage compressor off for only two hours, the energy savings from the system was 6.8 kWh or 37% of what the RTU would have used if it had continued to run.

Coincidentally at about the same time the second compressor was brought back, the outside temperature reached a point at which the mixed air sensor signaled the economizer of the need to modulate the damper to maintain the mixed air temperature set point. This had an interesting result with this damper assembly and the odd flow patterns it produces due to the support beam that blocks some airflow through the passage between the damper pair as they opened out. The placement of the mixed air sensor may be in a position where there is a sudden change in which of the two air streams hits it the most as the damper moves.

As the outside air damper modulated closed and the return damper opened, it appears as if the sensor caught too much of the return air and reversed direction. The mixed air eventually became too cold again and the actuator again reversed to bring in more return air. Since the system could not find a damper position that would produce the required mixed air temperature set point, it resulted in a period of damper oscillation or "hunting". Again, this is largely the result of sensor positioning and the effect on airflow from the damper style that would not be noticed without doing the "V" test.

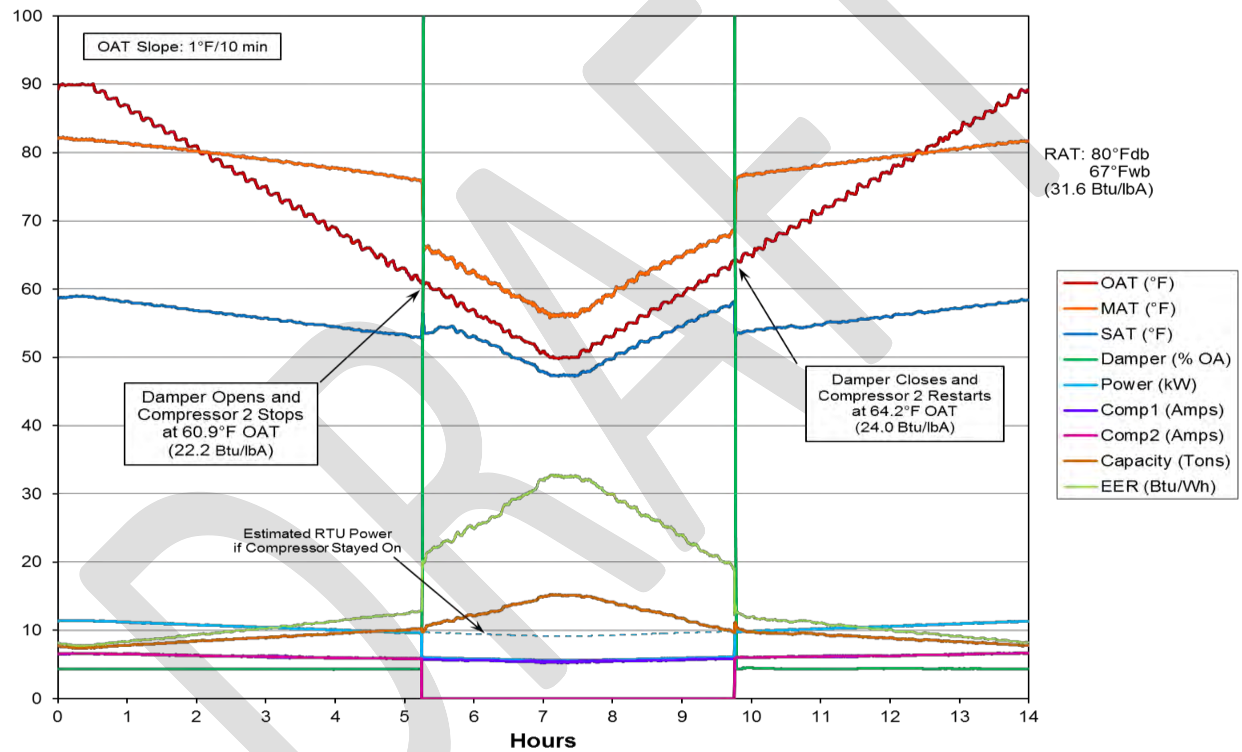
FIGURE 30: ECONOMIZER #2 "V" TEST – FACTORY DEFAULT SETTINGS ORIGINAL OAT & MAT SENSORS



Unrelated to the economizer operation, the test RTU revealed a system fault shortly after the second stage compressor restarted. As the result of what was eventually traced to be an intermittently leaking Schrader valve, the second stage circuit had lost a significant portion of its refrigerant charge. After the compressor started back up again, the fault mainfested itself initially through indications of very low suction pressures and temperatures. With the relatively high humidity return air, ice soon formed on the evaporator coil, which impacted airflow. The issue also demonstrated that the low suction pressure cut-out switch for the compressor was also faulty, as it allowed the compressor to continue operating. This was the refrigerant circuit that did not have to repaired at the beginning of the testing. In the lab, the fault was obvious; but if the unit was still in the field it could be a long time, if ever, before the loss of charge was discovered, particularly since the first stage compressor would still be providing cool air. Without a functioning low pressure cut-off protection it is possible that the compressor would be destroyed.

After the RTU was repaired and the second circuit recharged to its factory weight, another "V" test was conducted with some minor adjustments to the set points. The stage two compressor delay was set to "Off" such that it would not be brought back on while the outside air damper was open. Also, the minimum damper position was set such that the damper was at about 5% of its operating range towards open to outside. The results from this test are shown in ***Figure 31***. Even though the mixed air temperature sensor had not been moved, the damper hunting seen in the previous test did not occur. This is mainly the result of the mixed air sensor never reaching a temperature low enough to trigger the damper modulation. With the second stage compressor now off for a longer period (4½-hours), the savings over keeping it running were 16.4 kWh, or 39%. This trend also shows the reasoning behind the change over set point of 63°F as the cooling capacity[8] of the RTU barely changed from the transition between cooling return air with two compressors to cooling outside air with one compressor. This activity is similar to what a good commissioning technician would do to coordinate the RTU and economizer for best operation.

FIGURE 31: ECONOMIZER #2 "V" TEST – ADJUSTED SETTINGS ORIGINAL OAT & MAT SENSORS



For the next phase of the economizer evaluation, a new set of sensors was attached to the controller. This included replacing the outside air temperature sensor with a combination temperature and humidity sensor, and adding new combination sensors to the return and supply ducts. (The unit does not appear to do any control based on the supply air sensor other than enabling the display of its temperature reading. Its humidity sensor is unused and not available for display.) With sensors capable of determining air enthalpy, the

[8] Throughout this section and as shown in the charts, the measure of capacity (and the EER that is calculated from it) is based on the difference between the reference return or room air enthalpy and the supply air enthalpy multiplied by the supply air flow rate, even when the system is actually conditioning outside air or a mixture of return and outside. It is the "apparent" total capacity.

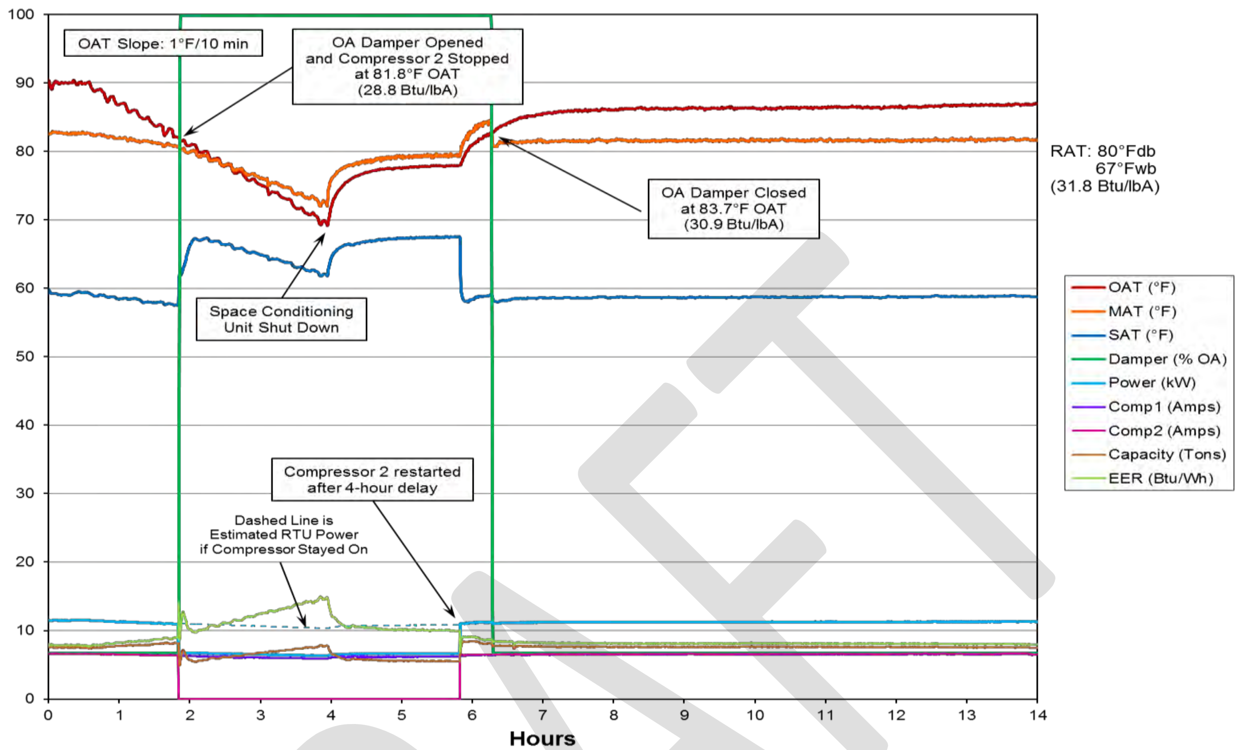
economizer automatically switches its operating mode to differential enthalpy control. This mode also uses an upper limit change-over curve defined at a maximum outside air temperature of 86°F and a maximum outside air enthalpy of 32.2 Btu/lb of dry air if the enthalpy difference between outside and return air does not changeover the economizer first.

A few attempts were made to observe the effects from this economizer with the new sensors, but with limited success due to problems with the space conditioning system for the outdoor room. This prevented reproduction of the nice "V" trends of outside temperature achieved in the previous tests. In the first attempt (shown in ***Figure 32***), the ramp down began well starting at 90°F, and the economizer switched over to outside air at an outside temperature of about 82°F. While this was a higher temperature than the return air, it had a lower humidity and consequently its enthalpy was lower by 3 Btu/lbm of dry air, which was enough of a difference to trigger the change. With this high temperature change over, the cooling capacity of the RTU saw a fairly significant drop when the second stage compressor was disabled.

When the outdoor air temperature ramp achieved 70°F, the outdoor room space conditioning system shut off completely for reasons unknown. (Again, this test was scheduled to run overnight and unattended.) This allowed the room temperature to rise until it reached equilibrium between the heat rejection from the RTU and the heat losses from the room. Even though the space conditioning system was off, its outside air damper remained open, and the booster fan on the airflow measurement apparatus achieved some air exchange with the outside.

As an attempt at a compromise between the default 2-hour delay on the second stage compressor restart and keeping it off for the duration of the economizing mode, the maximum delay time of 4-hours had been programmed into the economizer. After this 4-hour period expired, the second compressor restarted and contributed more to raising the now unconditioned outdoor room temperature. Eventually, the temperature reached the point where the dampers switched back to using return air. This occurred at an enthalpy difference between the outside and return air of about 1 Btu/lbm. So although the outside air temperature "V" trend could not be achieved, the system performed as expected given the available conditions. For the 4-hour period that the second stage compressor was off, the system avoided using 16.5 kWh of energy.

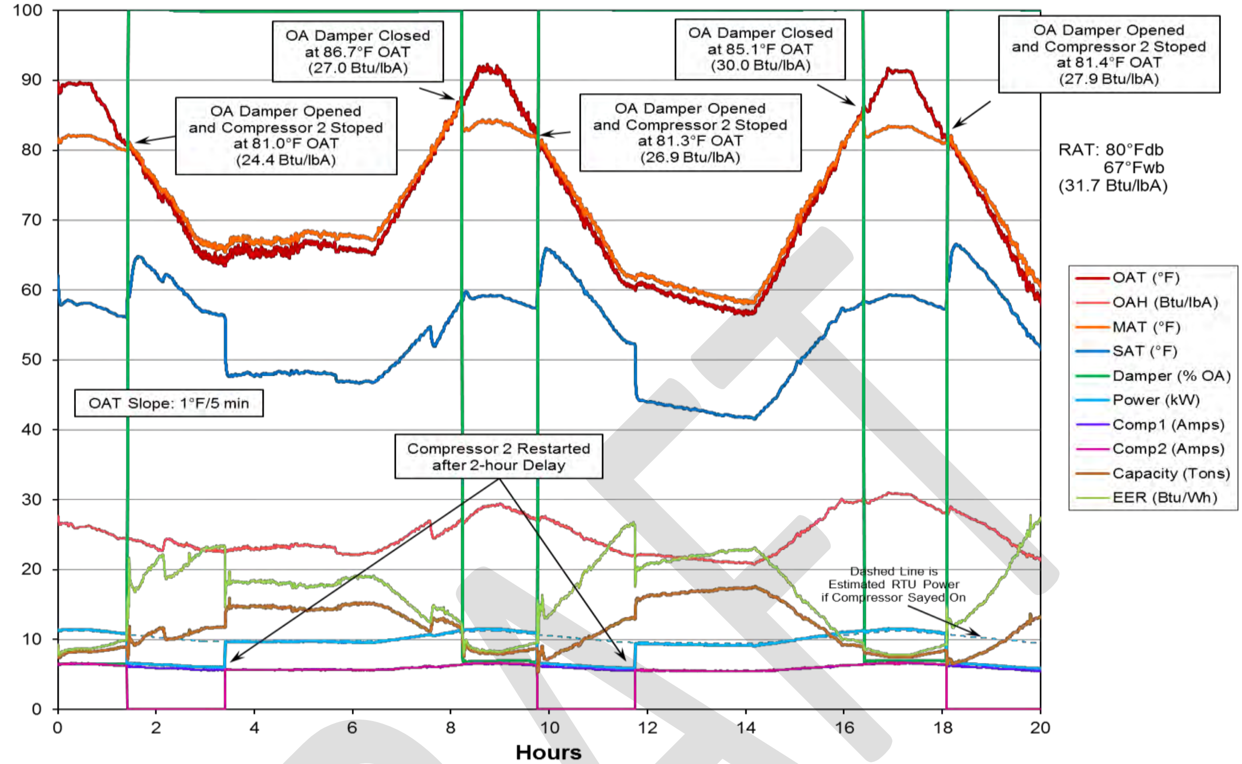
FIGURE 32: ECONOMIZER #2 "V" TEST, PARTIAL –DIFFERENTIAL ENTHALPY MODE NEW SENSORS



For the final test on this system, the ramp rate was increased back to 1°F every 5 minutes to decrease the cycle duration, but also set to repeat so that two or more complete cycles could be achieved in an overnight run to see if the response from the economizer is consistent. Once again, problems with the space conditioning system prevented full achievement of this goal. In this case, the space conditioning system did not shut off, but it failed to bring on all of its stages of cooling, which ultimately limited the lowest temperature that it could achieve. Thus, instead of achieving the desired low temperature of 50°F, the room temperature flattened out at whatever could be achieved with the space conditioning system running full out with its available compressors. In the first cycle, this minimum temperature was about 64°F, while in the second cycle (which occurred later in the night when it was cooler outside and the space conditioning system's capacity improved) the minimum temperature achieved was 57°F. The economizer controller's second stage delay was also reset to the default 2-hours for this test.

The outdoor room humidity was uncontrolled and floated with the ambient humidity. An addition to the collection of trends shown in ***Figure 33*** is the trend of outdoor room air enthalpy, which does not quite reflect the change in temperature due to humidity variations. Despite the difference seen in enthalpy, the transition into economizing mode for the three occurrences all happened at an outside temperature of about 81°F (with a range of less than ½°F). The transitions into and out of economizing mode all happened at enthalpy values less than the return air, with the leaving values higher than the entering values, reflecting an internal deadband.

Figure 33: Economizer #2 "W" Test – Differential Enthalpy Mode New Sensors



Throughout the evaluation of this product, it never shut down the primary compressor circuit, nor did it cycle either of the compressors like the other controller did. Based on the manual, the only time it would disable both compressors was when the outside temperature dropped below the low temperature compressor lock-out set point, which had been left at its default of 32°F, and the unit was never subjected to this temperature during the evaluation. There was nothing noticed in the operating instructions that indicates that the optional supply air temperature sensor would provide feedback on overcooling. Instead, this system relies on the room thermostat to say that the load has been satisfied to shut off or cycle the compressors, or on the compressor protection systems that would also prevent operation if the outside temperature is too low.

For either of these economizers, their actual operation is far more complicated than the demonstrations profiled here. Rather than the constant call for both stages of cooling that they were subjected to throughout these evaluations, the economizer will actually receive constantly changing indications from a thermostat responding to real changes in space temperature. These profiles also do not give the full picture of potential energy savings because they do not take into account the effect of the increase in cooling capacity at lower outside temperatures on the space load, as this will result in the unit operating less often. This enforces the need for developing load-based models of how systems will operate to capture a better representation of the potential energy savings. It is hoped that by understanding how the economizer controllers respond will help in the development of these models.

A final note is a concern with the placement of the return air sensor when the economizer is used in differential mode. When the system is using 100% outside air, the return duct is essentially stagnant, unless the conditioned space is tight enough that the only outlet for the excess ventilation air is back through the return duct and out the barometric relief damper. If the air in the return duct is stagnant, it may drift to a higher or lower temperature than the room air depending on the environment around the duct. With the economizer in a differential mode, this would affect the point at which the economizer transitions back to its minimum outside air position, and may pose an energy penalty. Thus, it may be prudent to locate this sensor closer to or in the conditioned space.

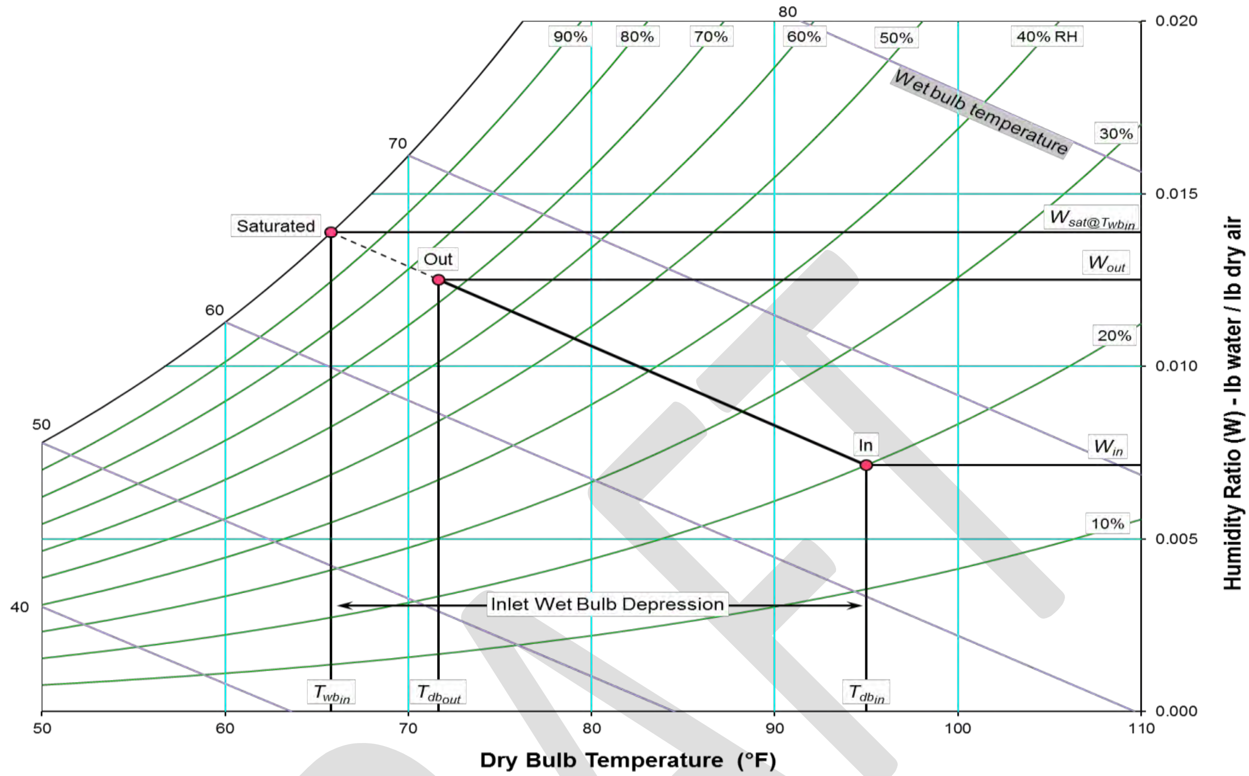
PHASE 4: EVAPORATIVE CONDENSER AIR PRE-COOLERS

As demonstrated through the RTU performance mapping, as the outside temperature rises, compressor power increases and cooling capacity decreases, which combine into a rapidly decreasing system efficiency. Higher temperatures also create higher space conditioning loads, so the conditioning system performance is diminished when it is most needed, and it may no longer be capable of maintaining a comfortable space temperature.

One method of effectively lowering air temperature is through direct evaporative cooling, where sensible heat (temperature) is adiabatically converted to latent heat (water vapor). Hot, dry air is put in contact with liquid water, which evaporates and absorbs heat from the air with the result of cool, humid air. If the process is adiabatic and the contact is long enough, then air will eventually become completely saturated (100% relative humidity) at what is termed the wet bulb temperature. The process is shown graphically on a psychrometric chart in ***Figure 34***.

By themselves, evaporative coolers have been used for direct space conditioning in dry climates where the cooling effect outweighs the feeling of the higher humidity. However, evaporative coolers can also be applied to the intake of an air-cooled condenser as a way to make it operate at a lower temperature, with a resulting increase in capacity and efficiency. Air-cooled condensers are unaffected by air humidity (other than from small changes in density), and since condenser air is not delivered to the space, the humidity increase does not affect the indoor humidity. When applied to an air-cooled condenser, the system is referred to as an evaporative "pre-cooler" since it cools the air before it is used to absorb heat from the condenser.

FIGURE 34: PSYCHROMETRIC CHART OF DIRECT EVAPORATIVE COOLING PROCESS



The main performance metric of an evaporative cooler is called the saturation or evaporation effectiveness (EE), and relates how close the outlet temperature gets to the wet bulb temperature at saturation, as shown in the following equation:

$$EE = \frac{T_{db_{in}} - T_{db_{out}}}{T_{db_{in}} - T_{wb_{in}}} \qquad \text{Equation 6}$$

The effectiveness is dependent on the contact time between the water and air, which is a function of several factors such as the relative velocity of the two streams and their distribution. As shown in the psychrometric chart, so long as the process is adiabatic and follows the trend of constant wet bulb temperature, the same value of evaporative effectiveness can be found using the change in humidity ratio or absolute humidity, represented by W.

$$EE = \frac{W_{out} - W_{in}}{W_{sat@T_{wb_{in}}} - W_{in}} \qquad \text{Equation 7}$$

This form of the equation becomes important when evaluating the effectiveness of an evaporative pre-cooler. Measuring the average outlet temperature between the cooler and the condenser coil can be difficult due to inadequate mixing and uneven wetting of the evaporative media resulting in wide spatial temperature variations, or droplet carry-over or radiant heat from the coil effects on the temperature sensors. The air that passes through the condenser is only heated by the coil and fan, and therefore its humidity ratio is unchanged. Thus, the humidity ratio at the RTU condenser air exhaust, downstream of a fan that provides some mixing action, should be the same as the average humidity ratio out of the evaporative pre-cooler. One factor that can affect this is air leakage into the RTU cabinet that bypasses the condenser coil and dilutes the condenser air, thus decreasing the humidity ratio and the apparent effectiveness.

The higher the condenser airflow and the higher the evaporative effectiveness, the greater the water consumption rate will be. The amount of water evaporated into the air is a function of the air flow rate and the rise in the humidity ratio, as follows:

Water Evaporation Rate (GPH) = 60 × CFM / *V* × (W_{out} − W_{in}) / (8.33 lb/gal H_2O) Equation 8

where *V* is the specific volume of the air in cubic feet per pound of dry air, determined from the conditions in the same location as the airflow measurement. Substituting in *Equation 7*, this may be rewritten as:

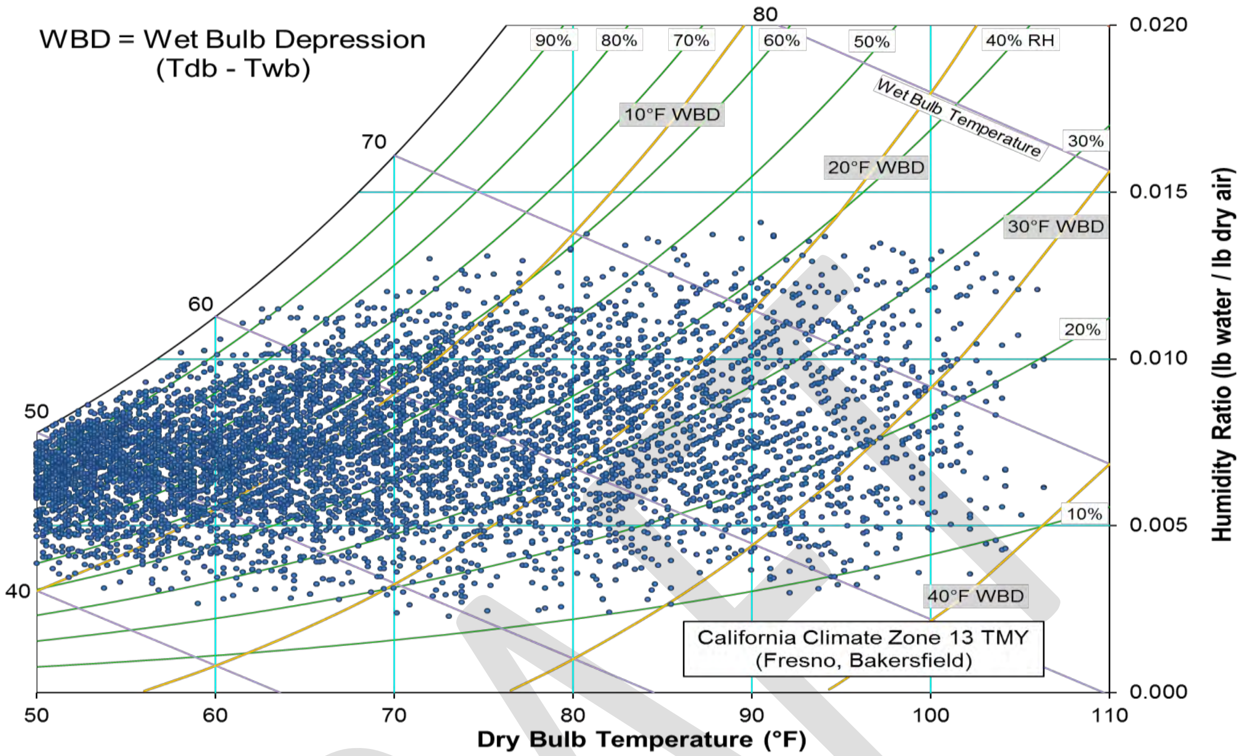
GPH = 60 × CFM / *V* × EE × ($W_{sat@Twb\ in}$ − W_{in}) / 8.33 Equation 9

Over a broad range of common wet bulb temperatures, the change in humidity ratio in *Equation 9* is directly proportional to the wet bulb depression by a factor of about 4330 °F/(lbW/lbA) (e.g. a decrease in temperature of 20°F along a line of constant wet bulb temperature would produce a humidity ratio rise of 0.0046 lbW/lbA (20/4330)). With this concept in mind, the water consumption of a pre-cooler due to evaporation is thus proportional to the product of:

- the airflow rate (CFM)
- the wet bulb depression (°F), and
- the evaporative effectiveness of the pre-cooler

The opportunity for applying evaporative cooling is fairly large in California, particularly in the hottest climates. Figure 35 shows a psychrometric chart with the hourly data points from the Typical Meteorological Year (TMY) climatic data base from Title-24 for Climate Zone 13, which covers most of the southern San Joaquin Valley. This data shows that when temperatures rise above 90°F, there will likely be better than a 20°F wet-bulb depression available for evaporative cooling.

FIGURE 35: PSYCHROMETRIC CHART OF CLIMATE ZONE 13 TMY HOURLY DATA



Tap water usually contains some level of dissolved solids, which will become concentrated in a system that uses a recirculation loop, and will need to be purged either by means of a continuous bleed on the loop or by periodic operation of a drain valve. This is a necessary water quality maintenance loss, but is still water that is not constructively cooling the air. For systems that use a once-through spray, overspray that is carried away by the wind or drains out before being evaporated is likewise not contributing to the air cooling. The ratio of the calculated water evaporation rate to the water supply or makeup flow rate is defined as the water utilization efficiency. Better systems will keep this ratio close to 100%. This factor could potentially be greater than 100% if the system can somehow capture the condensate from the evaporator coil. (For reference, throughout these pre-cooler tests, which were conducted with a constant return air condition of 80°Fdb/67°Fwb, the condensate collection rate was about 3 gallons per hour.)

Another concern about the dissolved solids in tap water regards liquid droplets directly contacting the condenser coil. While in the short term, this can result in enhanced heat transfer from the coil through direct evaporation, over time this could cause in a buildup of solids on the coil that can be very difficult to remove, with a resulting restriction in airflow. A wet coil could also encourage biological growth or corrosion between the fins and tubes. Thus, it is important in the design of an evaporative pre-cooler to allow for periodic inspection of the coil and easy replacement of the evaporative media, particularly in areas with hard water.

The first step in testing the performance of add-on evaporative pre-coolers was to produce a new performance map for the test RTU. This was necessary because during the economizer testing the second stage circuit leaked and needed to be recharged and that can have a significant effect on performance. The previous performance mapping (shown as the dotted lines in *Figure 15*) was also done with the economizer dampers removed, while the new mapping was done with the latest damper assembly installed (the unit with the opposed blades), but with return damper in the fully open position and the outside air intake sealed with plastic. The results are shown in ***Figure 36***, with solid lines representing the new map and dashed lines showing the previous map. Comparing the two sets, the power is about the same at high outside air temperatures but higher than before at low temperatures. Conversely, the capacity is about the same at the low temperature point, but lower than before at the high outside temperature. Much of the difference can be attributed to the airflow restriction caused by the damper, as the measured airflow dropped from 2,950 CFM in the previous mapping to 2,850 CFM in the new mapping. For reference, the actual values of the performance metrics are listed in ***Table 6***.

Figure 36: New RTU Benchmark Performance Map

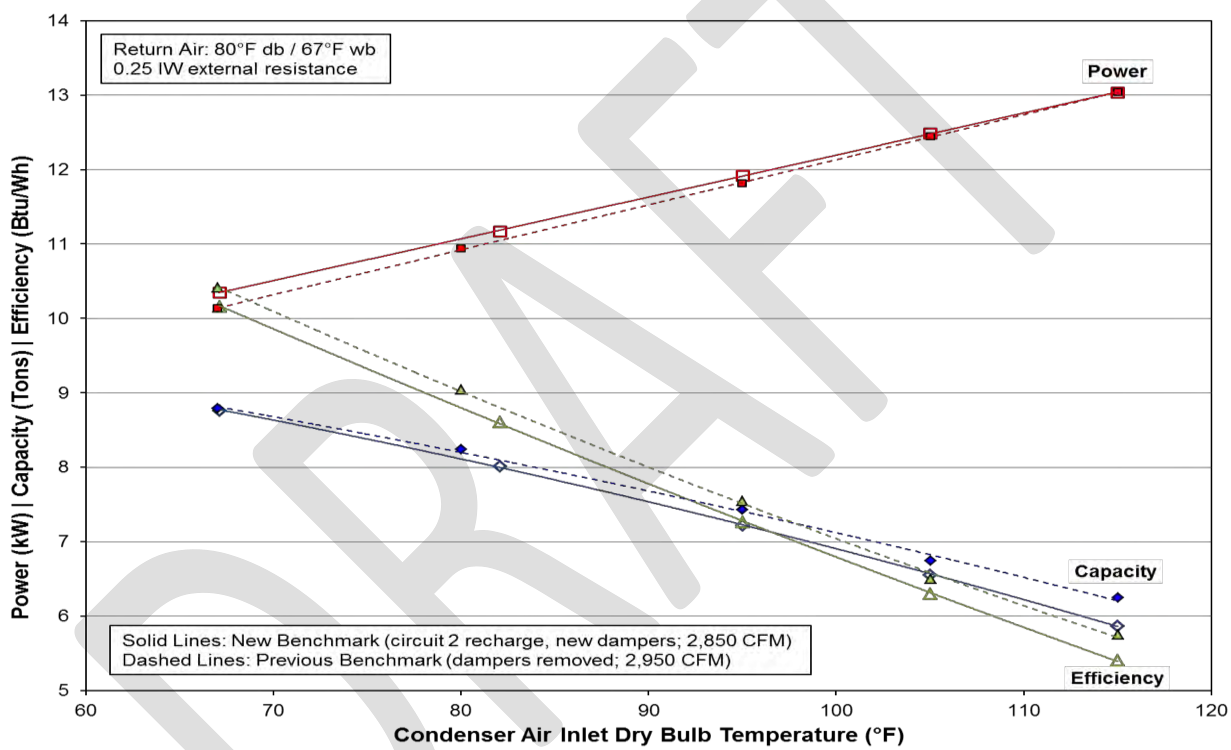


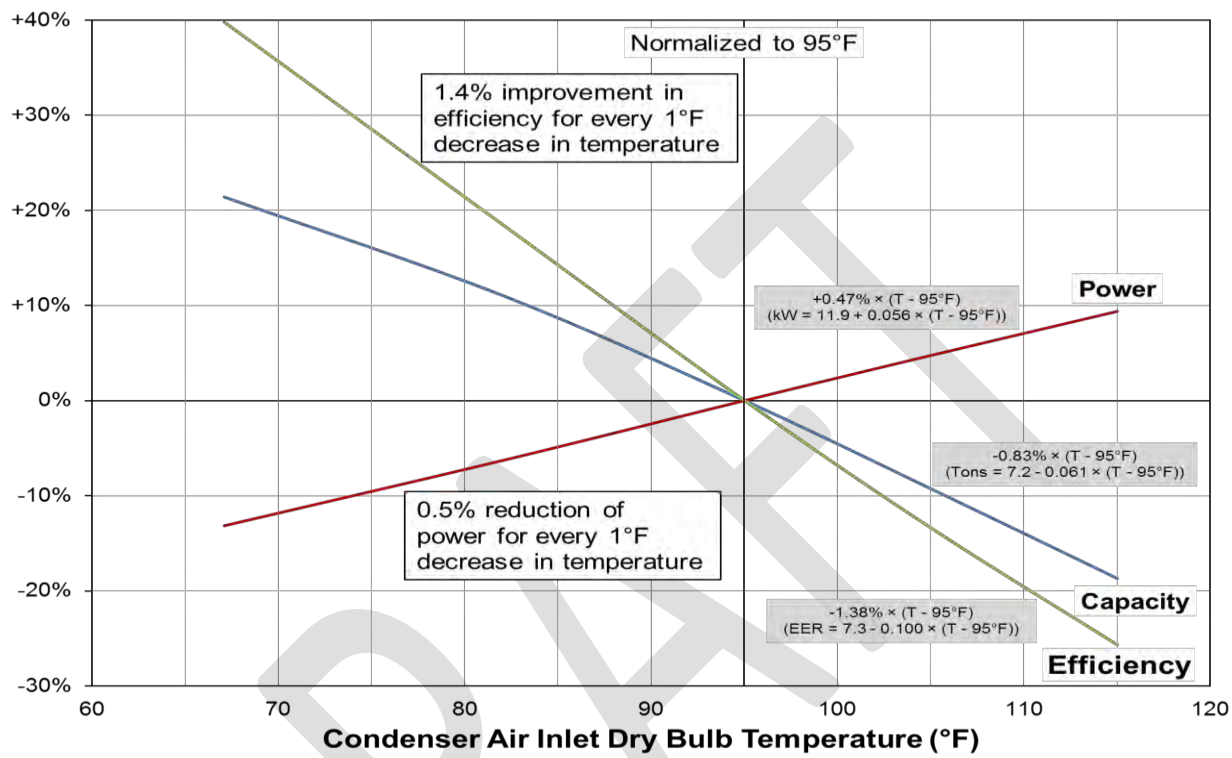
Table 6: New RTU Benchmark Performance Results

Outside Air Tdb (°F)	Condenser Airflow (CFM)	Face Velocity (ft/s)	Condenser Temp Rise (°F)	CAP^{BASE} Capacity (Tons)	P^{BASE} Power (kW)	EER^{BASE} Efficiency (Btu/Wh)	Relative to 95°F Capacity	Relative to 95°F Power	Relative to 95°F Efficiency
67	6,600	5.37	20.6	8.77	10.35	10.16	+21%	-13%	+40%
82	6,567	5.34	19.9	8.02	11.17	8.61	+11%	-6%	+18%
90				*7.54**	*11.63**	*7.77**	*+4%*	*-2%*	*+7%*
95	6,523	5.30	19.6	7.22	11.92	7.27	0%	0%	0%
100				*6.91**	*12.19**	*6.80**	*-4%*	*+2%*	*-7%*
105	6,502	5.29	19.2	6.55	12.49	6.30	-9%	+5%	-13%
115	6,476	5.26	18.6	5.87	13.04	5.40	-19%	+9%	-26%
Average:	6,533	5.31	19.6						

* Interpolated

The results shown in *Figure 36* can be normalized to a specific temperature to show the relative effect on the system performance from changes in outside temperature. This is shown in ***Figure 37***, with a normalization temperature selected as the AHRI Standard rating condition of 95°F. This chart shows that for this system there is about a half of a percent rise in power consumption for every 1°F in temperature rise, and about a 1.4% decrease in system efficiency.

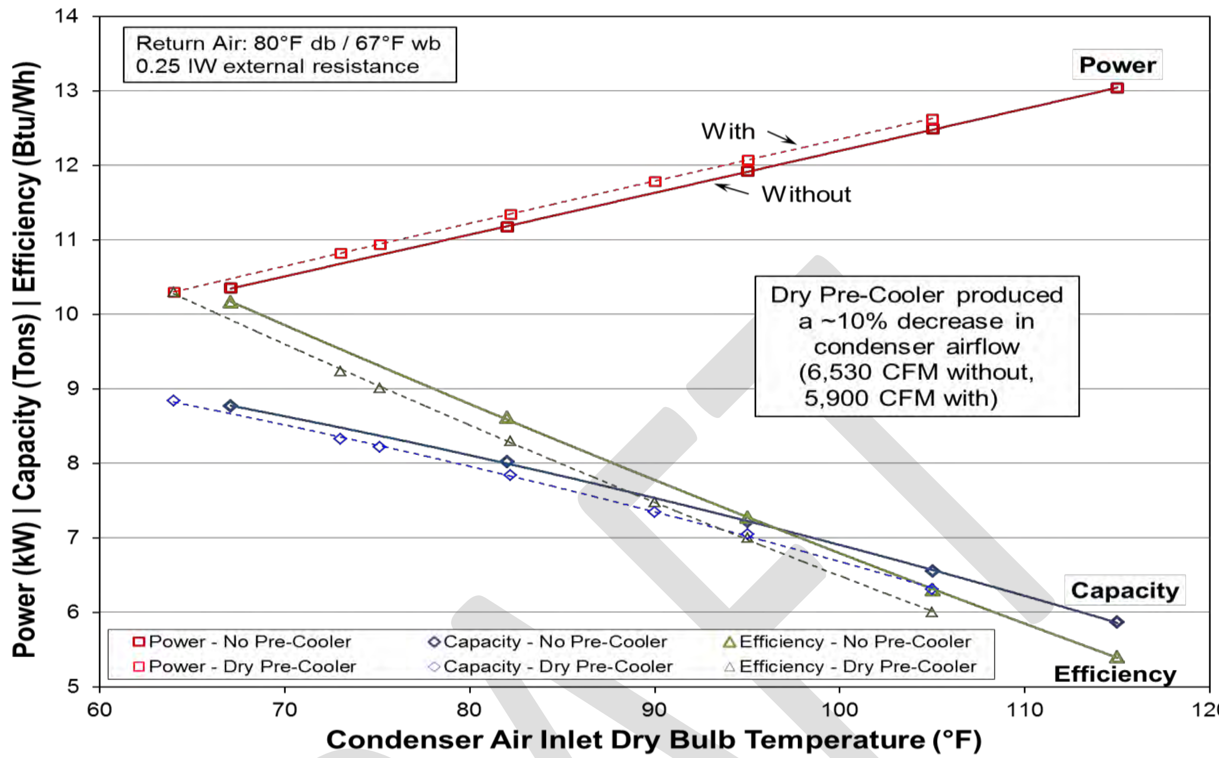
FIGURE 37: NEW RTU BENCHMARK RELATIVE PERFORMANCE TREND



Another factor of concern with the addition of an evaporative pre-cooler is the resistance that it adds to the condenser airflow[9]. A reduction in condenser airflow reduces its heat rejection capability and can cause higher compressor discharge pressures and a resulting decrease in efficiency. This effect is demonstrated in ***Figure 38***, which shows the performance mapping of the system with one of the test pre-coolers installed but dry so that it was not affecting the air temperature. This particular pre-cooler produced about a 10% reduction in the condenser airflow when dry. When wet, the airflow was reduced by about another percentage point, but the performance improvement from the temperature reduction masked the performance penalty from the airflow reduction.

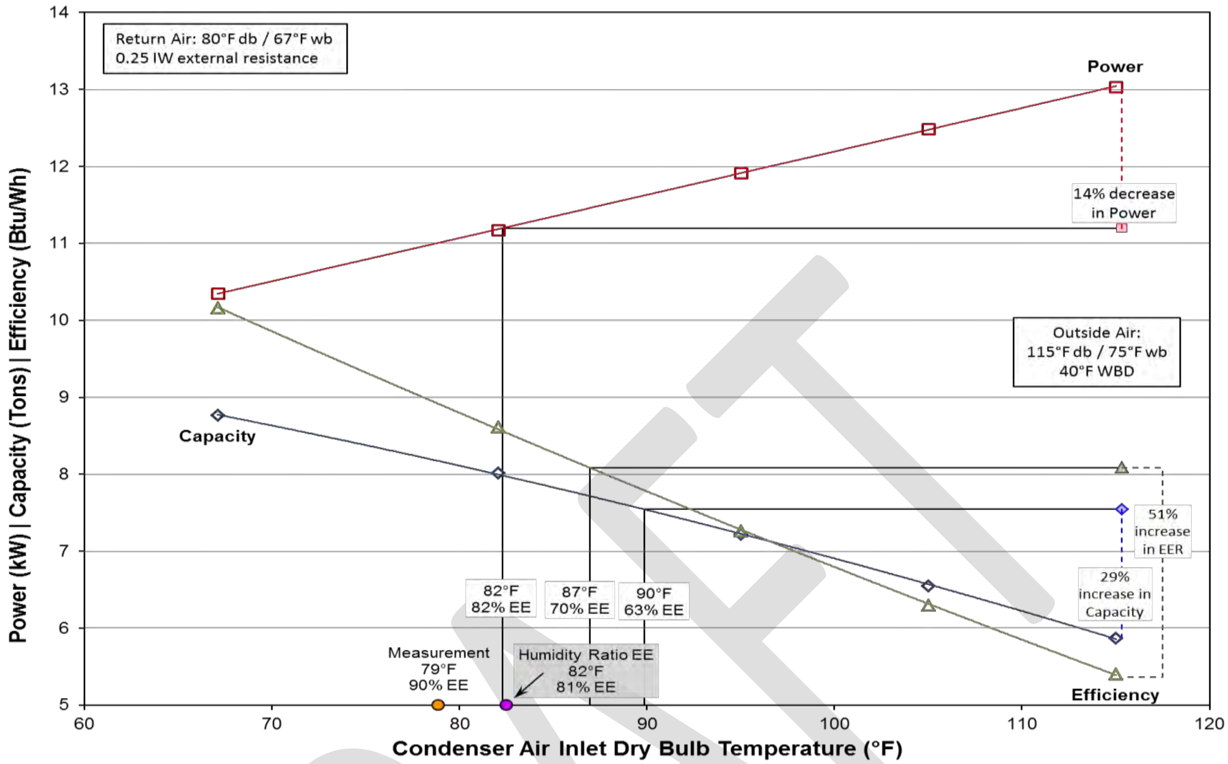
[9] The measurement of condenser air flow was made using a nozzle chamber airflow measurement apparatus or "code tester" attached to the condenser fan discharge by ducting, with a booster fan set to maintain zero static pressure at the connection to the test unit.

FIGURE 38: CHANGE IN PERFORMANCE FROM CONDENSER AIRFLOW RESTRICTION



Another method of determining the evaporative effectiveness of the pre-cooler is to compare the performance metrics (power, capacity and efficiency) as measured with the pre-cooler in operation against the benchmark performance map without the pre-cooler. In other words, at what condenser inlet temperature would the baseline unit have produced the same level of performance? This method is demonstrated graphically in ***Figure 39***. This example is an extreme case with a wet bulb depression of 40°F to emphasize the effects. This example also demonstrates that there are other factors involved (such as the airflow restriction) that can affect the performance metrics unequally. The unit with the pre-cooler showed a 14% decrease in power consumption from the map at the same outdoor dry bulb temperature of 115°F, and this corresponds to the power the baseline RTU would have used at 82°F. Calculating the effectiveness using this number results in 82% ((115-82)/40). However, using the cooling capacity map, the equivalent capacity occurs at an air intake temperature of 90°F, resulting in an effectiveness of only 63%. The mapping of the unit efficiency (either EER or COP) is a compromise between the other two, and the corresponding efficiency of the baseline unit occurs at a temperature of 87°F for an effectiveness of 70%. For reference, the evaporative effectiveness as calculated by *Equation 7* using the change in humidity ratio is included, and produced a result very close to the mapping of unit power at an effectiveness of 81%. Despite the potential measurement inaccuracies, a single sensor was installed between the pre-cooler and the condenser coil for a direct measurement of temperature. The indicated temperature of 79°F produces an effectiveness of 90% using *Equation 6*, but this is likely affected by some water droplet carry-over. The evaporation effectiveness values determined by each of these five methods are included in tables in each section describing the performance of each of the four tested pre-coolers.

FIGURE 39: EVALUATION OF EVAPORATIVE EFFECTIVENESS USING RTU PERFORMANCE MAP



The testing of the four subject evaporative pre-coolers was done on a consistent basis as much as possible. Four primary outside air conditions were included for all of the units, including the dry bulb and wet bulb temperature pairings (in °F) of 82/73 (warm, humid), 90/64 (WCC western climate average), 95/75 (AHRI Standard), and 105/73 (WCC western climate peak). As testing progressed, this was expanded to include 100/70 and 115/75 (AHRI maximum) for some of the systems. Other than that, the test RTU was operated in the same manner as for the baseline AHRI Standard performance tests, with the return air held at 80/67, and the external resistance on the unit set to 0.25 IW. The economizer was set with the return damper open and the outside damper closed and covered in plastic sheeting (with one exception). The return duct booster fan was turned on and its damper adjusted to maintain zero static pressure at the test unit return air intake to reduce leakage.

PRE-COOLER #1

The first pre-cooler evaluated consists of a continuous pumped recirculation loop that feeds multiple spray nozzles that wet a polymer media. The spray nozzles and media are sandwiched between inlet pre-filters and a drift elimination screen so that airborne debris does not get in and water droplets do not get out to the coil. The face area of the pre-cooler is larger than the face area of the condenser coil to lower the velocity through the pre-cooler and improve the contact time with the water. Water that is not evaporated returns through PVC pipe to a pump box external to the pre-cooler for recirculation. The pump box also contains a float valve fed from a standard water faucet to maintain the water level. With the pump box lower than the pre-cooler, it could easily be adapted to capture water condensed from the evaporator coil, so long as an overflow drain is also installed.

Water quality is maintained by a bleed valve off of the return pipe and set manually. The installation of the system on the test RTU was performed by a product representative contractor.

This system is not usually applied to an air conditioner of this small capacity, as their typical application is on systems of 20-tons or more. The problem with this is that the smallest size pump specified for these systems at 1/3-hp is really too big for this application, resulting in excessive auxiliary power use. The pump is triggered by an air temperature thermostat with an interconnection to the condenser fan so that it can only operate when the fan is on.

FIGURE 40: PRE-COOLER #1
(LEFT: PRE-FILTERS REMOVED SHOWING NOZZLES, RIGHT: COMPLETE SYSTEM UNDER TEST)

Because of the large face area of the pre-cooler, the air velocity passing through it was only 3.1 ft/s, compared with the bare condenser face velocity of 5.3 ft/s. Despite the low face velocity, the pre-cooler still imparted a condenser airflow rate loss of about 8%; some of which may be due to the added ducting needed to mate the pre-cooler to the RTU.

The performance metrics measured for the RTU with the pre-cooler installed have been graphed along with the benchmark performance map for the RTU in ***Figure 41***. All of the tested conditions showed an improvement in RTU power, and all but the test with the lowest wet bulb depression showed an improvement in capacity. The effect of the oversized circulation pump is indicated by the total power symbols representing the sum of the RTU power and the average pump power of 440W. Its effect was enough that the total system power actually increased over the baseline for the lowest wet bulb depression case. The charted values are also listed in ***Table 7***. The last columns in this table are the percent electrical savings estimates over the baseline from this product both in terms of demand (kW) savings from steady-state operation, and energy (kWh) that takes into account the capacity increase and should allow the RTU to operate less frequently to meet the same load.

FIGURE 41: RTU PERFORMANCE WITH PRE-COOLER #1

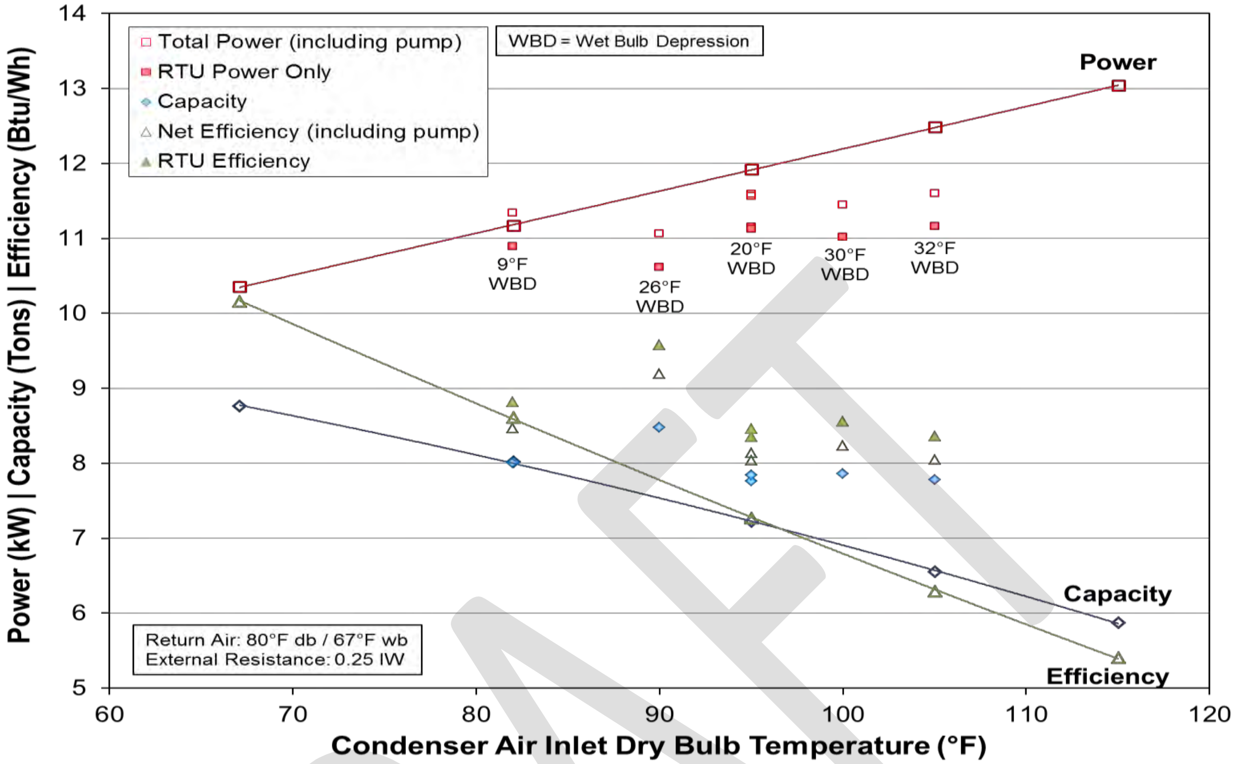


FIGURE 42: RELATIVE CHANGE IN RTU PERFORMANCE WITH PRE-COOLER #1

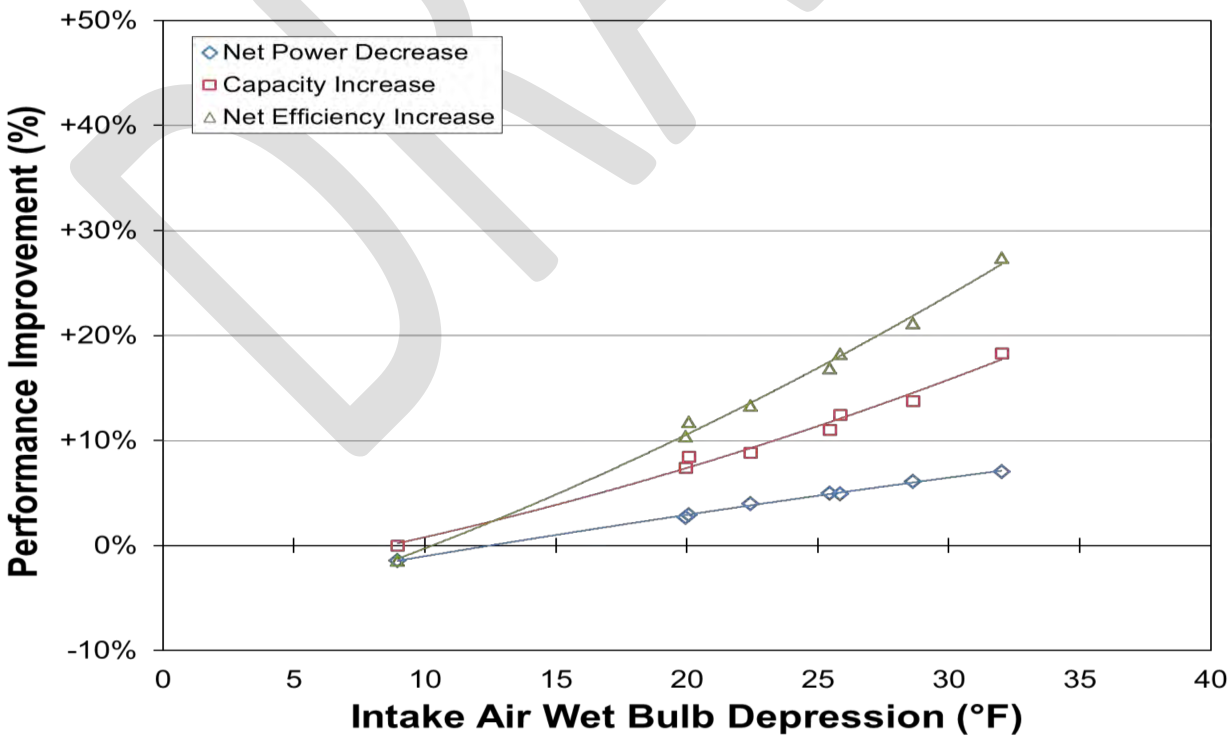


TABLE 7: RTU PERFORMANCE METRICS WITH PRE-COOLER #1

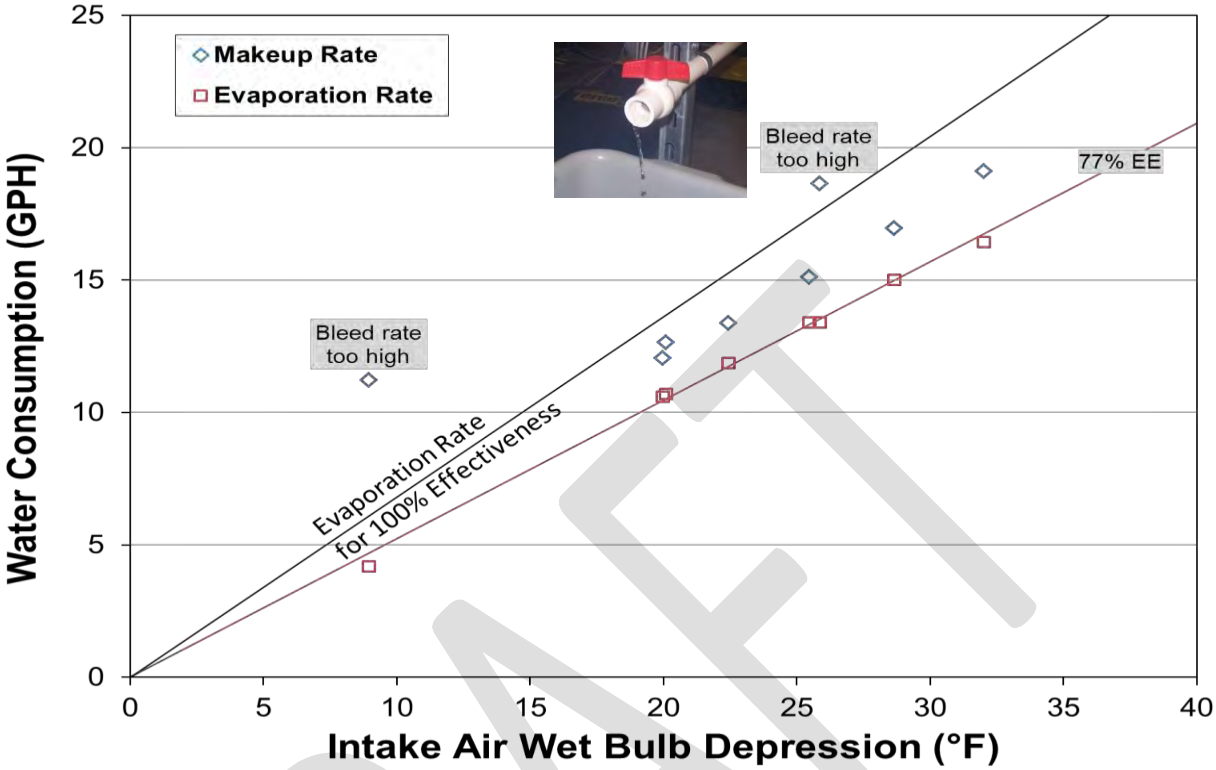
DB	WB	WBD	Condenser Airflow (CFM)	Face Velocity (ft/s)	Pre-Cooler & Condenser Temp Rise	CAP^{WET} Capacity (Tons)	P^{WET} Power (kW)	COP^{WET}	EER^{WET} Efficiency (Btu/Wh)	Relative to Baseline $\%CAP^{inc}$	Relative to Baseline $\%P^{inc}$	Relative to Baseline $\%EER^{inc}$	Pump Power (kW)	Net Electrical Savings Demand	Net Electrical Savings Energy
82	73	9	5,972	3.09	15.3	8.00	10.89	2.58	8.82	-0%	-2%	+2%	0.45	-2%	2%
90	64	26	5,965	3.09	3.1	8.48	10.62	2.81	9.58	+12%	-9%	+23%	0.44	5%	19%
95	75	20	5,944	3.07	6.5	7.80	11.14	2.46	8.41	+8%	-7%	+16%	0.44	3%	14%
100	71	29	5,942	3.07	-0.1	7.86	11.02	2.51	8.56	+14%	-10%	+26%	0.43	6%	21%
105	73	32	5,939	3.07	-2.9	7.77	11.16	2.45	8.36	+19%	-11%	+33%	0.43	7%	25%
	Averages:		5,952	3.08											
			-9%	Reduction from baseline											

The relative change in the three performance metrics from the baseline are shown in ***Figure 42*** as a function of the outside air wet bulb depression. The power values in this figure are the total values that include the pump power (which as noted previously is higher than it needs to be for this application), and this also affects the efficiency. The data is derived from *Figure 41* by taking the individual data points and dividing by the curves representing the performance map. The resulting trends show an increasing improvement in performance with increasing wet bulb depression for all of the metrics.

Figure 43 examines the water consumption rate of the pre-cooler in gallons per hour, which is again plotted as a function of the intake air wet bulb depression. The figure includes the water make-up flow rate as measured with a flow meter on the fill line, and the evaporation rate calculated from the measured rise in humidity ratio between the intake and RTU exhaust multiplied by the airflow rate. As was discussed earlier, the evaporation rate is a function of the product of the airflow rate, the wet bulb depression, and the evaporative effectiveness. Since the airflow rate through the unit is fairly constant, plotting the evaporation rate as a function of the wet bulb depression produces a straight line that is an indication of the average value of the evaporative effectiveness as determined by *Equation 7*. For this system, the slope of the line represents an average evaporative effectiveness of 77%. Also included in the figure is a trend line showing the evaporation rate that would be necessary to achieve 100% effectiveness, or the maximum rate of water evaporation that the RTU's condenser airflow could theoretically hold.

This system had a manually-set ball valve to bleed off some of the circulating water for maintaining water quality. The initial setting provided a steady trickle of water into a catch basin on a scale to measure its flow. After the first couple tests, it was decided that this bleed rate was too high and it was reduced from about 6.4 GPH down to 1.7 GPH. The excessively high bleed rate points are noted in the figure.

FIGURE 43: WATER CONSUMPTION RATE WITH PRE-COOLER #1



Another way of looking at the water consumption is relating it to the energy savings resulting from the system. The basic method is to take the continuous water consumption rates shown in *Figure 43* and divide them by the difference between the baseline RTU power and the sum of the measured RTU power with the pre-cooler and the added pump power at the same outside temperature indicated in *Figure 41*, as follows:

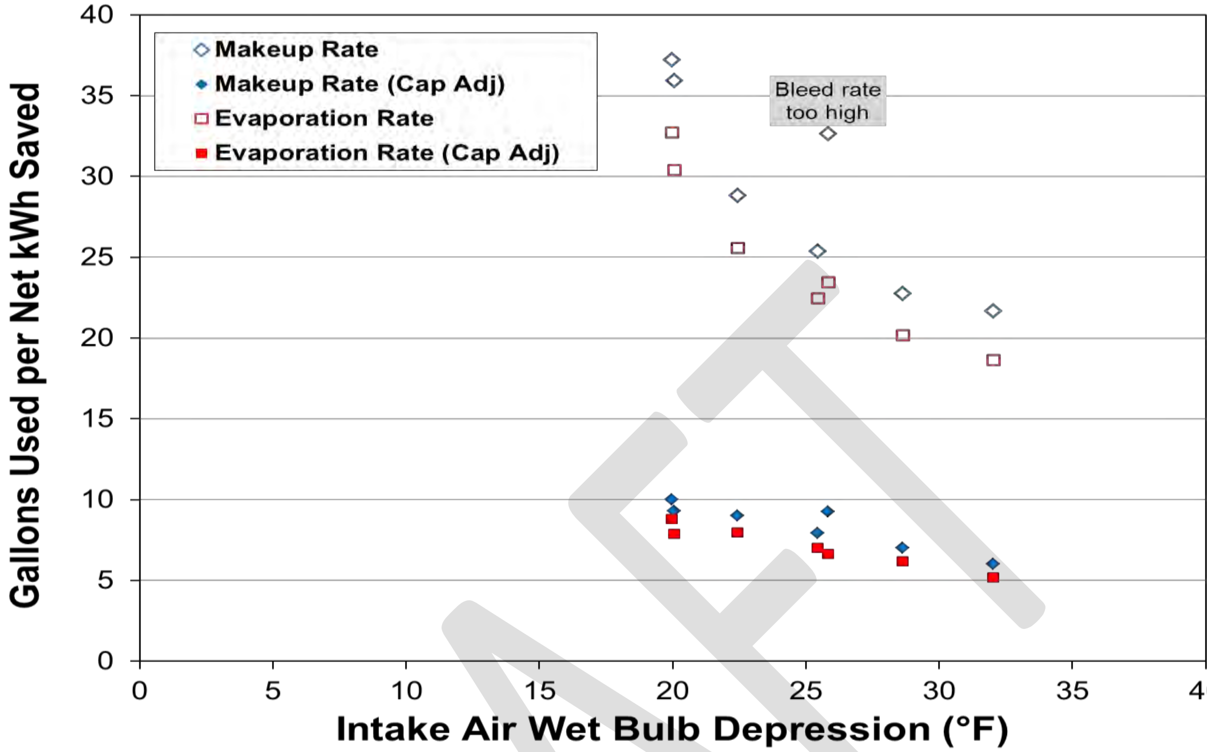
Gallons/kWh = GPH / ($kW_{Baseline}$ – ($kW_{Pre\text{-}cooled}$ + kW_{Pump})) Equation 10

This value could be looked at as the water cost for demand savings (GPH/kW). What this equation doesn't account for is the improvement in capacity provided by the pre-cooler, since this would allow the system to operate less frequently to meet the same load. The water and system energy would then be reduced by the same capacity ratio, with the assumption that the pre-cooler system would also shut off as the RTU cycles off. The formula that captures the capacity improvement can be simplified to:

Gallons/kWh = GPH / ($Tons_{Pre\text{-}cooled}$ × $(kW/Ton)_{Baseline}$ – ($kW_{Pre\text{-}cooled}$ + kW_{Pump})) Equation 11

Thus in this form, the effective baseline power is the power that the RTU would have had to consume to provide the same capacity as the pre-cooled system by using the baseline unit efficiency at the same outside temperature. In ***Figure 44***, the resulting values using the Equation 10 are shown using open symbols, and those using the Equation 11 are shown with filled symbols (identified as "Cap Adj" for capacity adjusted).

FIGURE 44: WATER USAGE PER UNIT OF ENERGY SAVED WITH PRE-COOLER #1



At low wet bulb depressions, the water consumption and energy savings become lower. As the divisor in this formulation, low energy savings can cause the result to get huge as it approaches zero. Negative savings, where the added pump power and airflow restriction results in a larger power usage with the pre-cooler than for the baseline (as is the case for the lowest wet bulb depression case for this system) are not included in the figure. Understanding the water efficiency allows for determining the set point above which the pre-cooler will operate in order to produce the most economical benefit.

The five methods of evaluating the evaporative effectiveness of the pre-cooler as illustrated in *Figure 39* are listed in ***Table 8*** for this product, and each section lists the corresponding pre-cooler outlet temperature. For the physical temperature measurement, this system has a relatively open and long duct between the wetted pad and the condenser coil, so the measurement of air temperature in this duct should not be particularly influenced by either. This is still only a single point measurement from a probe inserted through the duct wall, and may not represent an adequate average. The effectiveness as determined from the change in humidity ratio through the system corresponds to the trend of evaporation rate in *Figure 43*, since they are calculated from the same measurements. Finally, the temperatures and effectiveness values that are figured from the relative change from the baseline RTU performance tests are given, with the value from the efficiency curve listed in the middle because it is a combination of the power and capacity and the results tend to fall in between; effectively being a compromise.

TABLE 8: EVAPORATIVE EFFECTIVENESS AND OUTLET TEMPERATURES FOR PRE-COOLER #1

			Measured Temperature Out		Humidity Ratio Effectiveness		No Pre-Cooler Power Curve		No Pre-Cooler Efficiency Curve		No Pre-Cooler Capacity Curve	
DB	WB	WBD	°F	EE	°F	EE	°F	EE	°F	EE	°F	EE
82	73	9	76.1	66%	76.0	67%	76.9	57%	79.8	24%	81.9	1%
90	64	26	74.4	60%	70.5	75%	72.0	69%	72.6	67%	73.1	65%
95	75	20	78.8	81%	79.3	78%	81.2	69%	83.8	56%	85.5	48%
100	71	29	80.8	67%	78.0	77%	79.1	73%	82.3	62%	84.6	54%
105	73	32	80.9	75%	80.7	76%	81.7	73%	84.3	65%	86.0	60%

PRE-COOLER #2

Evaporative coolers not only cool the air that passes through them, they also cool the circulating water. With a continuous recirculation system, the water temperature can also approach the entering air wet bulb temperature. Pre-cooler #2 utilizes this capability by including an air-to-water heat exchanger that is used to cool the ventilation air drawn into the RTU through the economizer. This system configuration reduces the cooling load required for ventilation and also increases the time that the system can operate in economizing mode.

This is one technique of indirect evaporative cooling (IEC) where water that is cooled by evaporation to air is used to cool another air stream. Since the indirect coil can at best cool the air to the entering water temperature, and that water temperature is limited to the air wet bulb temperature, the coil cannot change the humidity ratio of the air passing through it, as that would require a coil temperature below the entering air dew point temperature. In addition, the evaporation process is no longer adiabatic as it now receives water warmed by the evaporator coil. On a psychrometric chart, this means that the process through the direct evaporative cooler will follow a line slightly steeper than the constant wet bulb line when there is energy absorbed by the indirect coil.

The direct evaporative cooling section is similar to Pre-cooler #1 in that it extends the condenser air out to a larger area in order to reduce the face velocity and increase the water contact time. Instead of a spray system, this unit uses a thick rigid cellulose media fed by a water distribution header at the top. Water from a sump at the bottom of the unit is pumped first through the indirect coil and then to the header, where it flows by gravity down over the media, which spreads the water out over a large surface area for evaporation. The flow rate is relatively low and with few restrictions, so the pumping power requirement is also low. The face velocity of Pre-cooler #2 at 4.2 ft/s was between that of the uncovered condenser and Pre-cooler #1, but the airflow through the condenser was only reduced by an average of 3% from the baseline due to the wide spaces in the evaporative media.

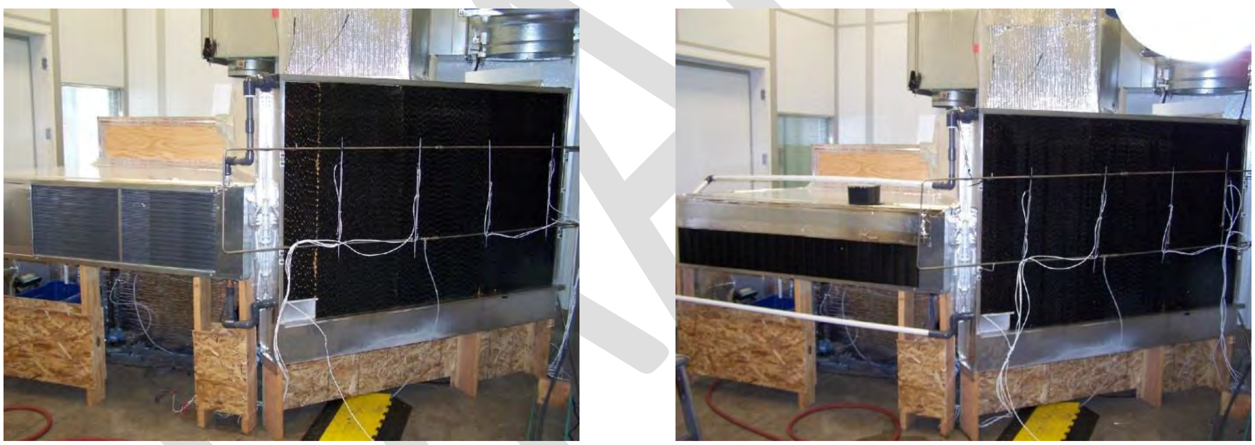
Like Pre-cooler #1, this system was installed on the test RTU by a product representative contractor. It would normally have been installed with a bleed valve in the circulation loop to maintain water quality while the pump was operating, but as a short term test and because of the low mineral content of the local supply, this step was skipped. This means that the make-up water meter will only measure the evaporation rate.

The circulation pump is controlled by a remote thermostat, but without an interconnection with the RTU condenser fan. Thus, it would operate whenever the ambient temperature exceeded the set point, even if the RTU was not running. This would allow water to flow through the indirect coil with the potential for cooling the outside air when the RTU is just being used for ventilation with the compressors and condenser fan off. However, the cooling effect would be minimal because the heat rejection capability of the direct evaporative section is poor if there is no air being forced through it. There may be some optimization potential by tying together the operation of the condenser fan with the circulation pump to extend the range that the system can cool with just ventilation and keeping the compressors off, but this was not explored.

The testing was done with the indirect coil attached to the opposed blade damper assembly examined in Phase 3a. The addition of the indirect coil creates a significant complication in testing because it adds the variable of outside air intake airflow as controlled by the damper position. System capacity is always measured relative to the return or conditioned space temperature and humidity and not the mixed air entering the evaporator coil. Thus, if the combination of temperature and humidity in the air leaving the indirect coil produces an enthalpy that is higher or lower than the return air, then the capacity will be decreased or increased (respectively) relative to the case where no outside air is used.

Another variable was the change out of the indirect coil part way into the testing. The original system included the manufacturer's standard metallic coil, but they wanted an opportunity to evaluate a lower cost polymer coil. Some testing overlap was done to provide a direct comparison, but there was not a complete doubling of the testing conditions for the two coils. Pre-cooler #2 with the two different indirect coils is shown in ***Figure 45***.

FIGURE 45: PRE-COOLER #2 (LEFT: ORIGINAL METALLIC COIL, RIGHT: POLYMER COIL)



In order to provide a direct comparison to the other pre-cooler systems, a series of tests were run that would minimize the airflow through the indirect coil and thus minimize its effect on the direct section. As was done with the other systems, the booster fan on the return duct was operated and its damper set to maintain zero static pressure at the return intake of the RTU. The return damper was fully open and the outside air damper was fully closed, but the indirect coil attached to the outside air intake was not sealed off with plastic as was done with the other systems. This still allows for a small amount of leakage, which will have an effect on the capacity. This arrangement corresponds to the solid lines in *Figure A-4*, which suggests that the outside air leakage rate should be less than 10%, particularly considering the added flow resistance from the coil. Even with no airflow through the indirect coil, it will still absorb some heat from the air surrounding it since it is exposed to the outside air, and this reduces the effectiveness of the pre-cooler slightly.

Once again, the measured performance metrics have been plotted alongside the baseline performance map in ***Figure 46***. The power and efficiency values are plotted twice to show the effect of the added 120W of circulating pump power. The case with the lowest wet bulb depression still showed a decrease in unit power even with the pump, but its measured capacity was slightly below the baseline, mainly because of the high humidity outside air leaking in. The relative change from the baseline is shown in ***Figure 47***, and the values for both figures are listed in ***Table 9***.

FIGURE 46: RTU PERFORMANCE WITH PRE-COOLER #2

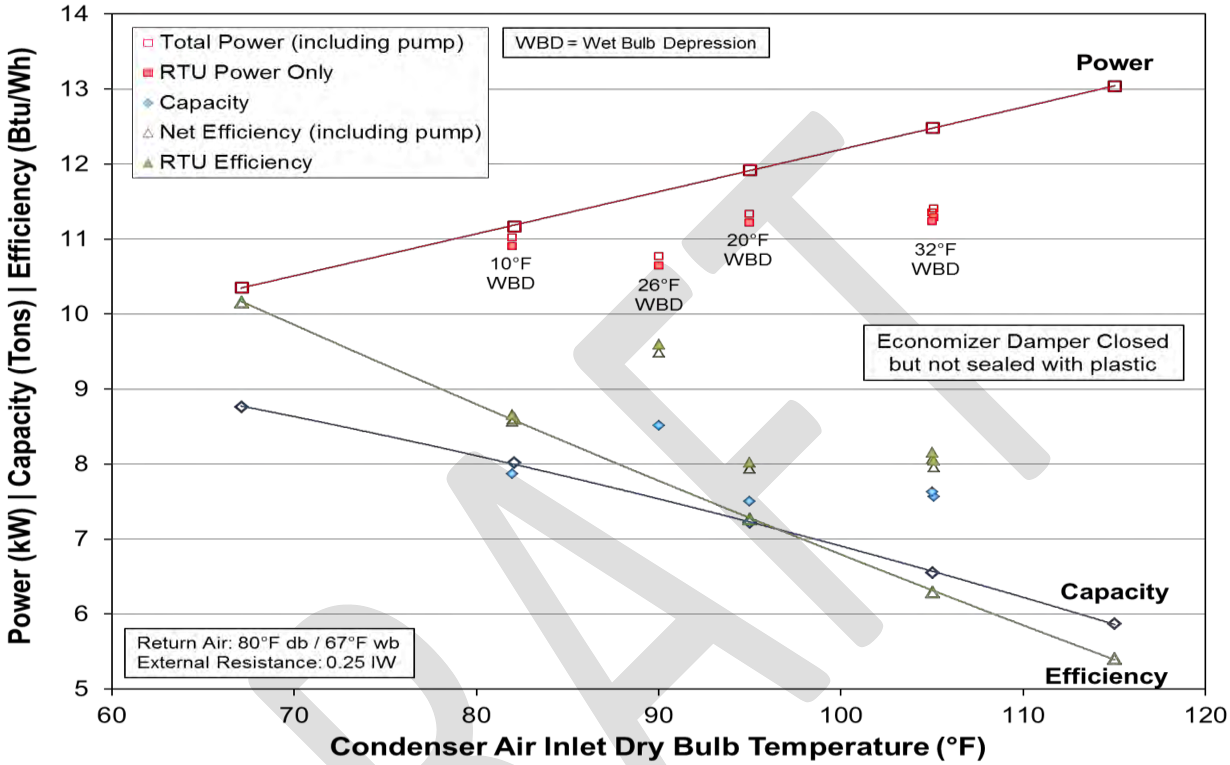


FIGURE 47: RELATIVE CHANGE IN RTU PERFORMANCE WITH PRE-COOLER #2

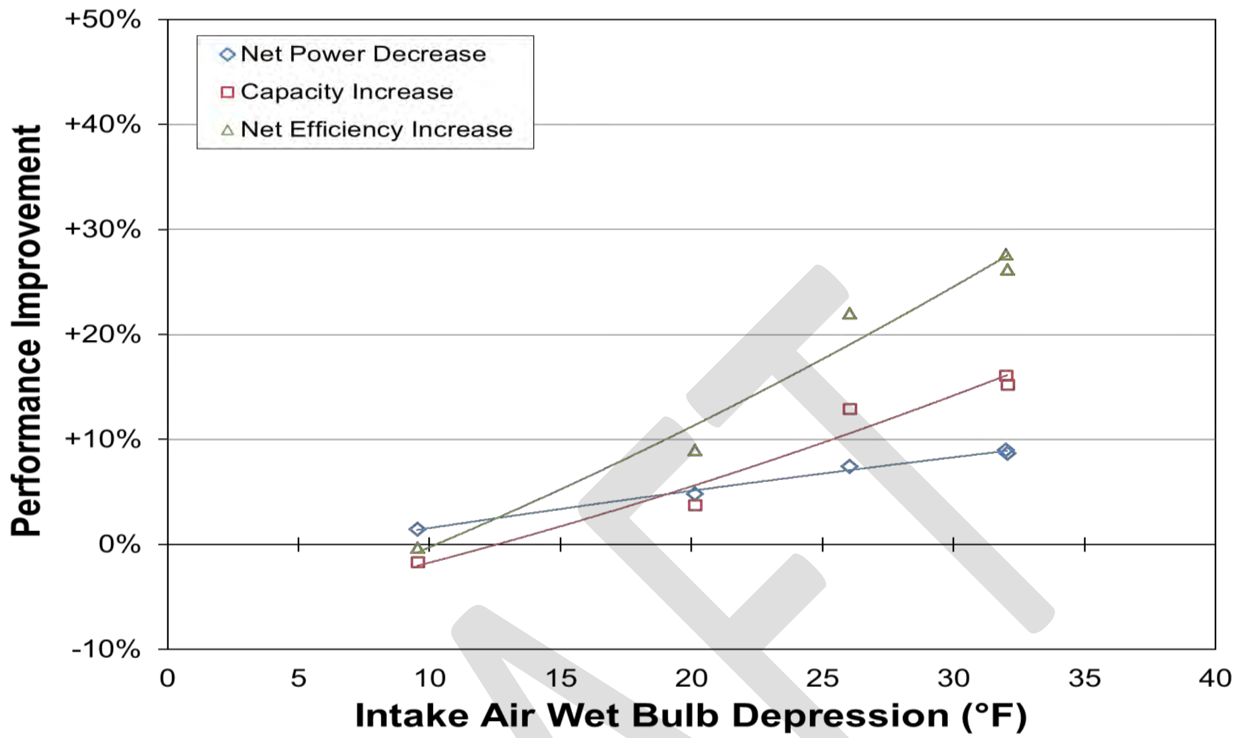


TABLE 9: RTU PERFORMANCE METRICS WITH PRE-COOLER #2

DB	WB	WBD	Condenser Airflow (CFM)	Face Velocity (ft/s)	Pre-Cooler & Condenser Temp Rise	CAP^{WET} Capacity (Tons)	P^{WET} Power (kW)	COP^{WET}	EER^{WET} Efficiency (Btu/Wh)	Relative to Baseline %CAP^{inc}	%P^{inc}	%EER^{inc}	Pump Power (kW)	Net Electrical Savings Demand	Energy
82	73	9	6,313	4.24	14.8	7.87	10.90	2.54	8.66	-2%	-2%	+1%	0.12	1%	1%
90	64	26	6,314	4.24	3.2	8.51	10.65	2.81	9.59	+13%	-8%	+23%	0.12	7%	19%
95	75	20	6,291	4.22	7.4	7.50	11.22	2.35	8.02	+4%	-6%	+10%	0.12	5%	9%
105	73	32	6,306	4.23	-1.8	7.60	11.26	2.37	8.10	+16%	-10%	+29%	0.12	9%	22%
	Average:		6,306	4.23											
			-3%	Reduction from Baseline											

As this system did not have a bleed installed, the make-up water flow rate was roughly equal to the calculated evaporation rate as shown in ***Figure 48*** (allowing for measurement error). The water evaporation rate as measured on the air side in points towards an average evaporative effectiveness of 73%. The chart shows that a wet bulb depression of 20°F would be a reasonable start for pre-cooler operation. The effectiveness values calculated by the other methods are shown in ***Table 11***.

FIGURE 48: WATER CONSUMPTION RATE WITH PRE-COOLER #2

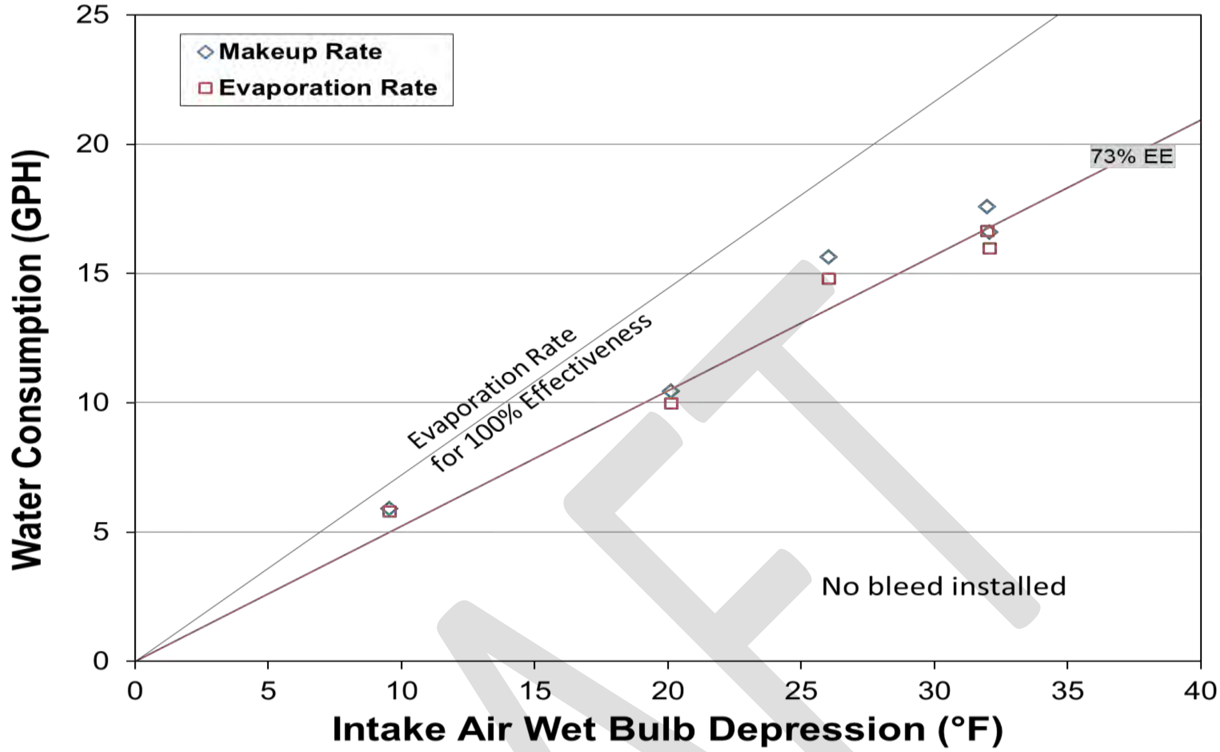


FIGURE 49: WATER USAGE PER UNIT OF ENERGY SAVED WITH PRE-COOLER #2

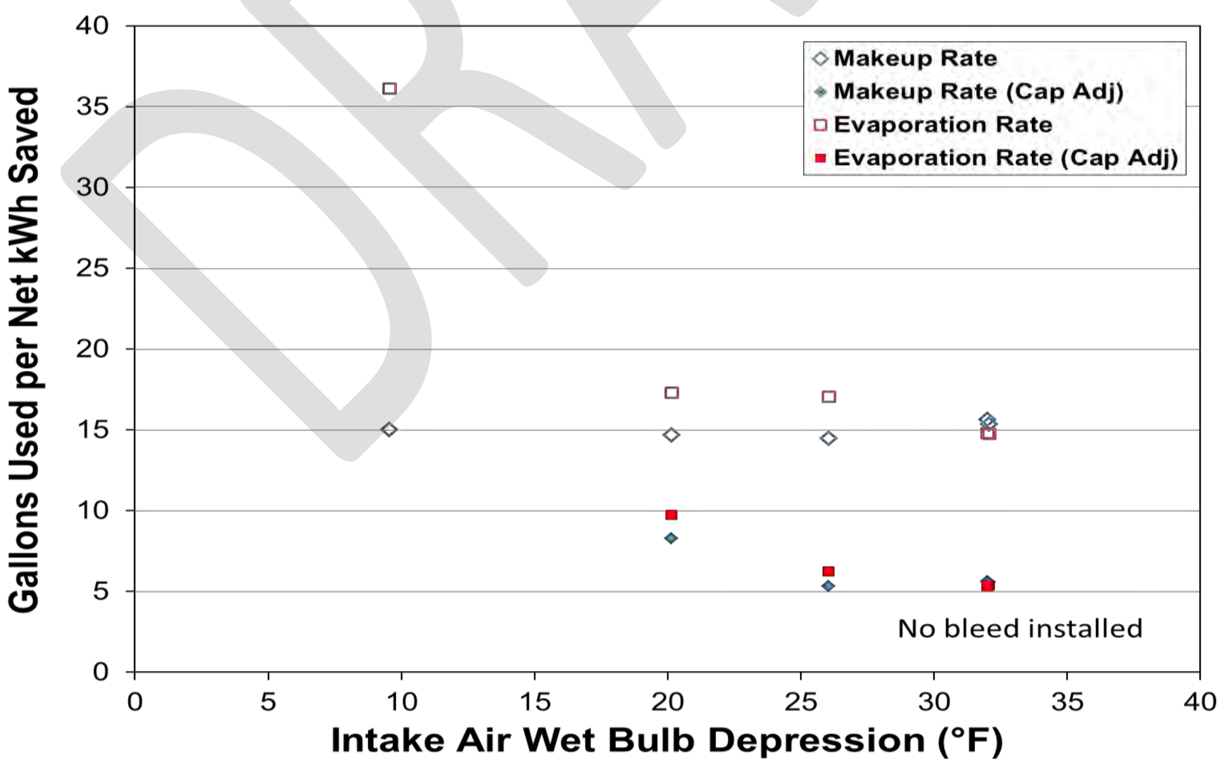


TABLE 10: EVAPORATIVE EFFECTIVENESS AND OUTLET TEMPERATURES FOR PRE-COOLER #2

			Measured Temperature Out		Humidity Ratio Effectiveness		No Pre-Cooler Power Curve		No Pre-Cooler COP Curve		No Pre-Cooler Capacity Curve	
DB	WB	WBD	(°F)	EE	(°F)	EE	(°F)	EE	(°F)	EE	(°F)	EE
82	73	9	74.9	77%	74.1	85%	77.0	53%	81.4	7%	84.4	-25%
90	64	26	69.9	77%	69.6	78%	72.4	68%	72.5	68%	72.4	68%
95	75	20	79.8	76%	78.9	80%	82.6	62%	87.6	37%	90.6	22%
105	73	32	79.9	79%	82.4	71%	83.4	68%	87.4	55%	89.5	49%

For the next step of testing on this product, the return duct booster fan was turned off and its damper fully opened, and the static pressure at the return intake to the RTU was allowed to float with the friction of the return duct. The supply outlet static pressure was held to 0.25 IW. With the economizer damper set for 100% return air, the frictional pressure loss through the return duct was around 0.3 IW. The outside air fraction should then follow a trend similar to the dashed lines in *Figure A-4*, although with some reduction in the outside air flow caused by the restriction through the indirect coil.

For each of the outside air condition set points, the outside air damper was set to six different positions: 0%, 6%, 20%, 33%, 46%, and 100% of the actuator range. The intermediate stages were selected because they would produce outside air fractions of about 20%, 30%, 40% and 50%. Of the four methods of determining the outside air fraction described in Phase 3a, only Method 1 by direct measurement of the return airflow provided a reasonable result. Methods 3 and 4 were unavailable because the evaporator coil was in operation, and this affects the supply air temperature and humidity. Method 2 using the mixed air temperature was unacceptable most of the time because the temperature of the air leaving the indirect coil was often too close to that of the return air.

The results from this phase of testing are shown in ***Table 11***. Reading the table from left to right, the first columns describe the condition of the outside air. Added to the usual values of dry and wet bulb temperatures and wet bulb depression are the humidity ratio (W) and enthalpy (H) of the air. At the bottom of the table are the corresponding values for the return air for reference. The reason for including these values is that when the outside air humidity ratio is higher than that of the return air, it will often result in a decrease in the measured capacity unless there is a significant decrease in temperature.

Following those columns are descriptors of the ventilation air intake through the indirect coil. The first column is the outside air damper position, as measured based on its operating range of movement. This is followed by the calculation of outside air fraction as calculated from the measurements of return and supply airflow rates using *Equation 2*, even with the increasing uncertainty in the return air measurement as the outside air damper opens and the return airflow decreases. Although it was never measured directly, an estimation has been made as to what the apparent capacity of the RTU would have been if the same fraction of unconditioned outside air was brought in, similar to what is shown in *Figure 16*. This assumes the same actual capacity as the baseline, but cooling a blend of return and outside air instead of just return air. One interesting outcome of this calculation is the case with the outside air at 95°Fdb and 75°Fwb and with the economizer open for 100% outside air, where the RTU would not have been able to condition this amount of outside air to below the enthalpy of the return air, so the apparent capacity is negative. A functioning economizer should never operate like this normally.

The next section of the table presents the measured performance of the test RTU and their values relative to the baseline performance map without any outside air intake. Since for these latest tests there was an allowed pressure drop in the return duct resulting in a lower pressure in the mixed air plenum and more leakage flow through the indirect coil, the values listed here with the outside air damper closed (0% damper position) do not correspond to the results given previously in *Table 9* which had a fixed return pressure of zero relative to the outside. In the net savings section, the demand savings are basically the negative of the relative change in electric consumption from the previous section, but adjusted for the added pump power. The energy savings columns take into account the change in capacity for determining the duration that the system would run to meet the same load. The first column shown uses measured the capacity relative to the baseline RTU without any outside air. The second column uses the estimated apparent capacity of the baseline RTU with outside air. This second value is usually a larger number, except for the case where the enthalpy of the outside air is less than the return air and the simple economizer mode without pre-conditioning would still provide a capacity benefit.

The next section (in Part 2 of the table) first looks at the performance of the indirect coil. The first column is the air dry bulb temperature leaving the coil, as measured by an array of four thermocouples located upstream of the damper. From this measurement has been calculated a wet bulb effectiveness for the indirect coil using *Equation 6*. These two values are a function of the airflow through the indirect coil, resulting in lower values at higher flows; although the value with the outside air damper closed can also be high because of the mostly stagnant air between them being warmed by conduction to the outside. The next column is the enthalpy of the air leaving the coil based on the measured temperature and the same humidity ratio as the outside air. The values that are higher than the return air have been bolded to indicate that they will still decrease the RTU capacity from the baseline having no outside air.

The last section of the table looks at the evaporative effectiveness of the condenser air pre-cooler section similar to what is in *Table 10*, but for only three of the five methods. It did not make sense to map to the baseline capacity curve or to the efficiency curve that is calculated from it when the ventilation air is impacting the capacity.

The tests done with the polymer indirect coil instead of the metallic coil have been highlighted with a gray background. Of particular interest are the overlapping tests done at the outside conditions of 95°Fdb/75°Fwb, where the effectiveness of the indirect coil shows a slightly higher result for the metallic coil under the same operating conditions. Although, in perspective, this only represents between a 1 to 4°F difference in outlet temperature depending on the airflow. The results for the system with the polymer coil with an outside air condition of 90°Fdb/64°Fwb showed a markedly higher increase in RTU efficiency than any of the other tests, particularly with the outside air damper wide open. Much of this result is because this outside air condition has the lowest humidity ratio of all, but also because the indirect coil is absorbing less energy and thus having less impact on the performance of the direct pre-cooler section.

TABLE 11: PERFORMANCE WITH INDIRECT COOLING OF OUTSIDE AIR AND PRE-COOLER #2

Outside Air					RTU Performance			Economizer			RTU Performance								
Temperatures			W	H	Baseline Metrics			Damper	RA Flow	Capacity	New Metrics			Relative to Baseline (No OA)			Net Savings		
DB (°F)	WB (°F)	WBD (°F)	(lbW/ lbA)	(Btu/ /lbA)	Capacity (Tons)	Power (kW)	EER (Btu/Wh)	Position	% OA	with OA (Tons)	Capacity (Tons)	Power (kW)	EER (Btu/Wh)	Capacity	Power	Efficiency	Demand	Energy No OA	Energy With OA
82	73	9	**0.0155**	**36.7**	8.02	11.17	8.61	0%	14%	7.26	7.45	10.70	8.35	-7%	-4%	-3%	3%	-4%	5%
								6%	22%	6.87	7.32	10.77	8.16	-9%	-4%	-5%	2%	-7%	8%
								20%	31%	6.41	7.21	10.82	7.99	-10%	-3%	-7%	2%	-9%	13%
								33%	40%	5.95	7.28	10.83	8.07	-9%	-3%	-6%	2%	-8%	20%
								46%	46%	5.62	7.16	10.98	7.83	-11%	-2%	-9%	1%	-11%	22%
								100%	100%	2.82	6.10	11.05	6.62	-24%	-1%	-23%	0%	-31%	54%
90	64	26	0.0069	29.3	7.54	11.63	7.77	0%	19%	8.04	8.63	10.34	10.02	+15%	-11%	+29%	10%	18%	16%
								6%	22%	8.11	8.92	10.34	10.35	+18%	-11%	+33%	10%	21%	18%
								20%	31%	8.36	9.37	10.34	10.87	+24%	-11%	+40%	10%	25%	20%
								33%	40%	8.60	9.69	10.33	11.26	+29%	-11%	+45%	10%	27%	20%
								46%	47%	8.79	9.92	10.39	11.47	+32%	-11%	+47%	10%	29%	20%
								100%	100%	10.20	11.20	10.21	13.15	+49%	-12%	+69%	11%	38%	19%
95	75	20	**0.0144**	**38.7**	7.22	11.92	7.27	0%	18%	5.92	6.85	11.23	7.32	-5%	-6%	+1%	5%	0%	18%
									17%	5.98	6.72	11.09	7.27	-7%	-7%	-0%	6%	-1%	16%
								6%	21%	5.73	6.85	11.29	7.28	-5%	-5%	+0%	4%	-1%	20%
									22%	5.64	6.59	11.14	7.10	-9%	-7%	-2%	6%	-3%	19%
								20%	30%	5.05	6.69	11.36	7.06	-7%	-5%	-3%	4%	-4%	27%
									32%	4.93	6.41	11.22	6.85	-11%	-6%	-6%	5%	-7%	27%
								33%	39%	4.38	6.61	11.35	6.99	-9%	-5%	-4%	4%	-5%	36%
									41%	4.26	6.07	11.27	6.46	-16%	-5%	-11%	4%	-14%	33%
								46%	46%	3.90	6.40	11.38	6.75	-11%	-5%	-7%	3%	-9%	41%
									47%	3.80	5.98	11.41	6.28	-17%	-4%	-14%	3%	-17%	39%
								100%	100%	-0.03	4.99	11.36	5.27	-31%	-5%	-28%	4%	-39%	100%
									100%	-0.03	4.92	11.49	5.14	-32%	-4%	-29%	3%	-43%	100%
100	70	30	0.0090	**34.0**	6.91	12.19	6.80	0%	19%	6.46	7.78	10.76	8.68	+13%	-12%	+28%	11%	21%	26%
								6%	22%	6.39	7.93	10.80	8.81	+15%	-11%	+30%	10%	22%	28%
								20%	32%	6.15	8.15	10.91	8.97	+18%	-11%	+32%	10%	23%	32%
								33%	42%	5.91	8.29	10.94	9.10	+20%	-10%	+34%	9%	25%	35%
								46%	51%	5.70	8.28	10.94	9.08	+20%	-10%	+34%	9%	24%	38%
								100%	100%	4.55	8.12	11.04	8.83	+18%	-10%	+30%	9%	22%	49%
105	73	32	0.0103	**36.7**	5.87	13.04	5.40	0%	20%	4.84	7.29	11.05	7.92	+24%	-15%	+47%	14%	31%	39%
								6%	24%	4.64	7.34	11.13	7.91	+25%	-15%	+46%	14%	31%	42%
								20%	35%	4.05	7.19	11.15	7.74	+22%	-14%	+43%	14%	29%	48%
								33%	45%	3.53	7.28	11.13	7.85	+24%	-15%	+45%	14%	30%	55%
								46%	53%	3.16	7.10	11.23	7.58	+21%	-14%	+40%	13%	28%	59%
								100%	100%	0.73	6.79	11.44	7.12	+16%	-12%	+32%	11%	23%	90%
80	67	13	0.0114	31.8	*<- Return Air*														

<- Indicates polymer heat exchanger rather than metallic Net Savings Includes Pump Power

Outside Air					Economizer		Indirect Coil Outlet				Evaporative Effectiveness					
Temperatures			W	H	Damper	RA Flow	T	Effect-	Enthalpy		Measured		Humidity Ratio		No Pre-Cooler	
DB	WB	WBD	(lbW/	(Btu/	Position	% OA	DB	iveness		MAT	Temperature Out		Effectiveness		Power Curve	
(°F)	(°F)	(°F)	lbA)	/lbA)			(°F)	(%)	(Btu/lbA)	(°F)	°F	EE	°F	EE	°F	EE
82	73	9	**0.0155**	**36.7**	0%	14%	75	79%	**34.9**	80	75	75%	75	73%	73	93%
					6%	22%	74	84%	**35.0**	79	75	73%	75	73%	75	79%
					20%	31%	74	83%	**35.0**	79	75	73%	75	75%	76	70%
					33%	40%	75	80%	**34.5**	78	75	78%	74	79%	76	68%
					46%	46%	75	74%	**35.0**	78	75	74%	74	82%	78	38%
					100%	100%	77	52%	**35.4**	78	75	73%	74	88%	80	25%
90	64	26	0.0069	29.3	0%	19%	76	56%	25.8	80	70	77%	70	78%	67	88%
					6%	22%	70	76%	24.3	79	70	77%	69	80%	67	89%
					20%	31%	72	70%	24.6	79	70	77%	69	82%	67	88%
					33%	40%	73	65%	25.2	79	70	76%	68	84%	67	89%
					46%	47%	74	60%	25.5	78	70	76%	70	76%	68	85%
					100%	100%	77	51%	26.1	79	70	75%	69	81%	65	97%
95	75	20	**0.0144**	**38.7**	0%	18%	79	81%	**34.6**	80	80	77%	79	78%	83	61%
						17%	83	60%	**35.7**	81	80	75%	80	75%	80	73%
					6%	21%	78	84%	**34.4**	80	80	77%	79	78%	84	55%
						22%	81	70%	**35.4**	81	80	74%	80	75%	81	69%
					20%	30%	78	86%	**34.4**	80	80	77%	79	80%	85	49%
						32%	81	69%	**35.3**	81	80	75%	80	75%	83	62%
					33%	39%	79	80%	**34.7**	80	80	77%	79	82%	85	50%
						41%	82	63%	**35.5**	82	80	74%	79	78%	84	57%
					46%	46%	80	77%	**34.8**	80	80	77%	78	83%	86	47%
						47%	83	60%	**35.8**	83	80	74%	80	78%	86	44%
					100%	100%	83	60%	**35.6**	83	80	76%	77	87%	85	49%
						100%	84	54%	**36.0**	84	80	74%	79	81%	88	37%
100	70	30	0.0090	**34.0**	0%	19%	80	65%	29.2	80	76	79%	78	74%	75	85%
					6%	22%	75	84%	27.9	80	76	79%	78	73%	75	83%
					20%	32%	75	83%	27.9	80	77	78%	77	77%	77	76%
					33%	42%	78	75%	28.5	80	77	78%	76	79%	78	75%
					46%	51%	78	73%	28.7	80	77	77%	76	81%	78	74%
					100%	100%	85	50%	30.4	85	77	76%	75	84%	79	69%
105	73	32	0.0103	**36.7**	0%	20%	79	81%	30.3	80	80	78%	82	73%	80	80%
					6%	24%	79	83%	30.2	80	80	79%	81	75%	81	75%
					20%	35%	79	81%	30.5	80	80	77%	81	76%	81	74%
					33%	45%	81	75%	30.7	81	80	79%	80	79%	81	75%
					46%	53%	82	71%	31.0	82	80	78%	79	80%	83	69%
					100%	100%	89	52%	**32.5**	89	80	77%	78	85%	87	58%

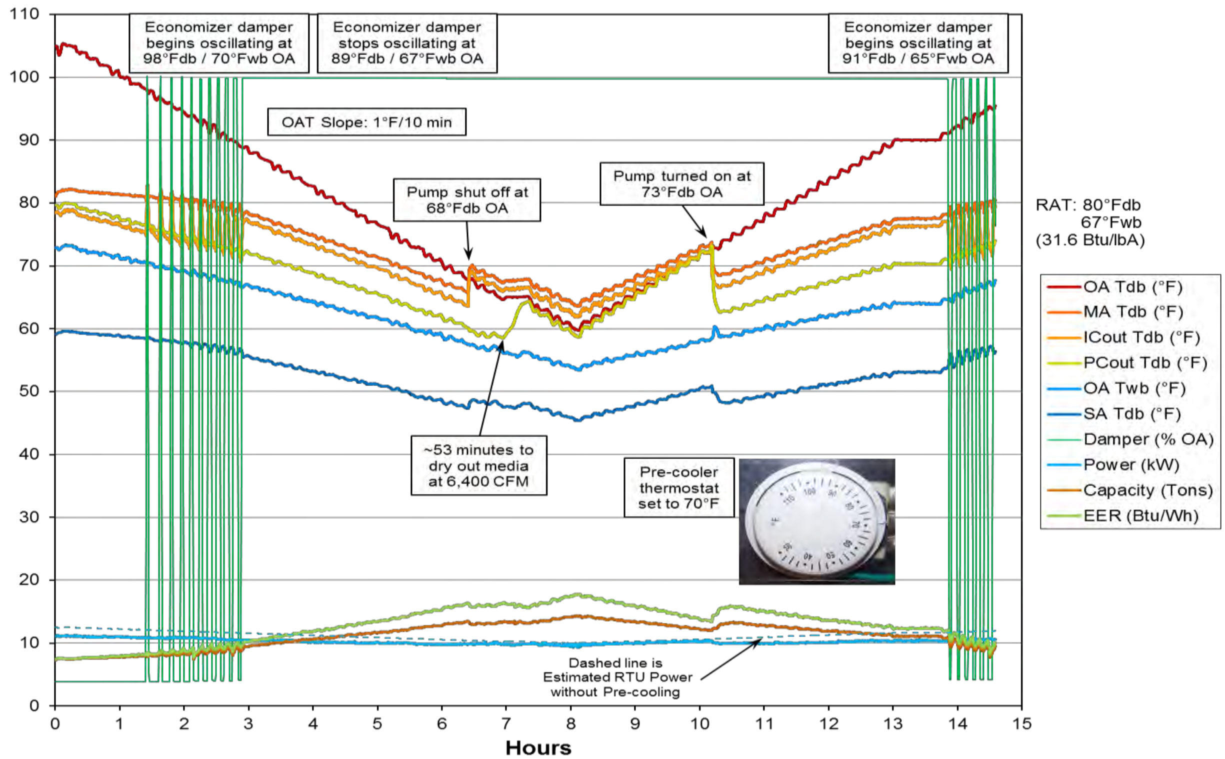
<- Indicates polymer heat exchanger rather than metallic

The results show that the two outside air conditions that had a humidity ratio higher than the return air did produce a decrease in the capacity of the RTU, but under these conditions the outside air damper would likely remain at its minimum opening if properly controlled by the economizer, and the decrease is still less than the unit would experience without the indirect coil. In order to examine the how this system would perform with a varying outside condition, a so-called "V" test from Phase 3b was conducted. This test was done with the Economizer #2 controller with the original OAT and MAT sensors in place. The controller was programmed for economizer changeover at an outside air temperature of 75°F (per Title-24 for this climate zone) and also to not turn off the second compressor stage when it went into economizing mode. The thermostat for the pre-cooler was set to operate the pump at an air temperature above 70°F.

The result of the first run of the "V" test produced an interesting result, and one that needs to be considered when setting up an economizer controller to work with an indirect coil. The outside air temperature was initially set to 105°F and then ramped down at a rate of 1°F every 10 minutes to a temperature of 60°F, and which point the ramp reversed and the outside temperature was raised back to 95°F where the test was concluded. The outside air humidity was uncontrolled and allowed to float with the ambient humidity ratio (which ranged between 0.0067 and 0.0106). ***Figure 50*** shows the trend for this test with most of the same measurements as in *Figure 31*, with the addition of the outside air wet bulb temperature and the outlet dry bulb temperatures from the indirect coil (ICout) and from the direct pre-cooler (PCout). The outside air temperature sensor for the economizer was still situated upstream of the damper, and thus downstream of the indirect coil. As the test started, the damper was in its minimum open position allowing a small amount of air to pass through the coil, with a relatively large drop in temperature. Eventually, the economizer temperature sensor detected that the air temperature coming off the indirect coil was below its threshold and it opened the damper. This increased the airflow that passed through the coil and consequently shrunk the drop in temperature. The economizer then detected that the air temperature had moved back above the threshold and put the damper back to its minimum. Thus began a series of damper oscillations until the outside air temperature conditions dropped to a point where the outlet temperature stayed below the transition threshold with full airflow.

As the outside air temperature continued to decrease, it eventually reached the point where the pre-cooler thermostat shut off the pump. This had the immediate effect of raising the temperature at the indirect coil outlet and subsequently the mixed air temperature, with some lag behind the trend of the outside air temperature due to the thermal mass of the coil. (This test was conducted with the more massive metallic coil.) The direct pre-cooler section, however, continued to provide some air cooling for the condenser for nearly another hour as the media dried out. Once the trend reversed and the outside temperature was raised, the thermostat eventually restarted the pump and the cooling effect to both the condenser air and the ventilation air were immediate. The temperature ramp up also reached a point that triggered the same damper oscillation observed during the ramp down. This oscillation did not have much effect on the cooling capacity of the RTU because the indirect coil outlet temperature was fairly close to the return air temperature, so the change in the mixed air temperature was small. The oscillations would have had a greater impact if the economizer had been set to its default mode of operation that would have shut down the second stage compressor when the damper opened, and started it back up when it closed. This would add to the wear-and-tear of the compressor possibly shortening its life, in addition to the extra wear-and-tear on the dampers and actuator. Understanding these effects can allow controls to be implemented to minimize oscillation or "hunting."

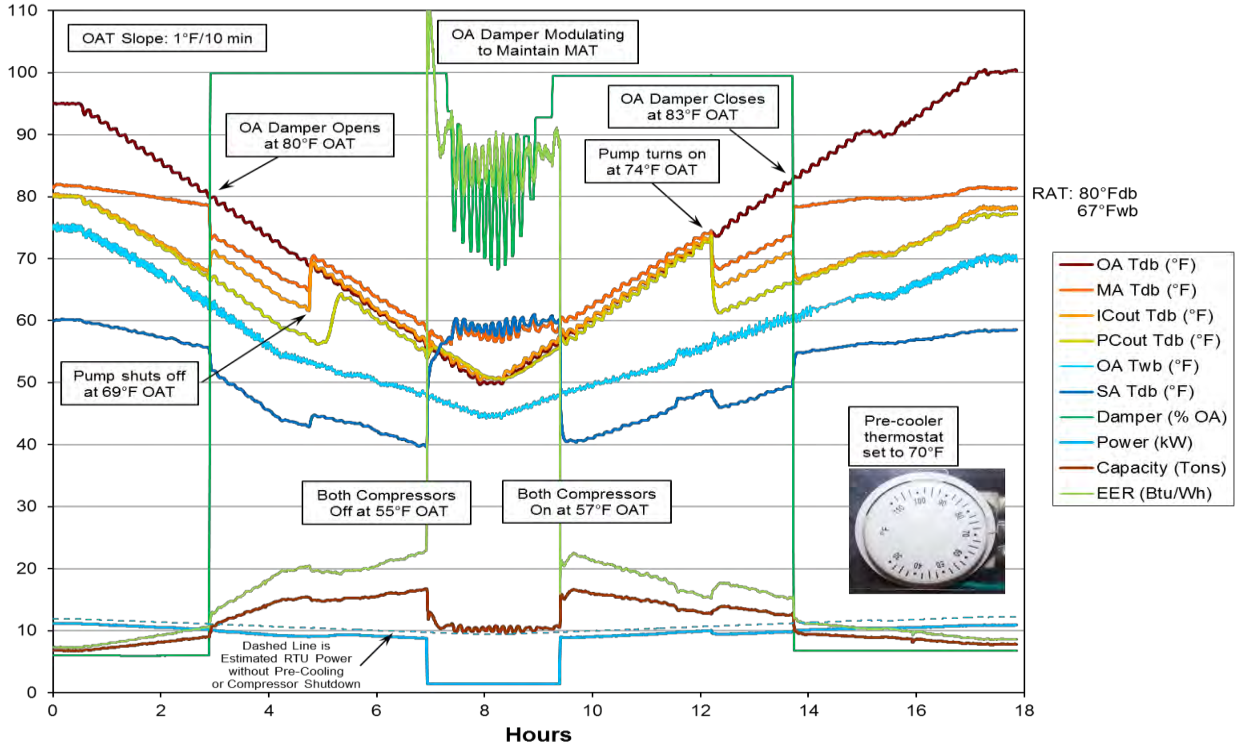
FIGURE 50: FIRST ECONOMIZER "V" TEST WITH PRE-COOLER #2 (METALLIC INDIRECT COIL)



A second "V" test was then conducted where the outside air temperature sensor for the economizer was moved into the actual outside air, and thus it would not see the changes in temperature coming off of the indirect coil. To compensate, the economizer changeover set point was raised by 5°F to 80°F (same as the return air) in anticipation of the cooling done by the coil. The second stage compressor shutdown was left disabled, but the compressor lock-out temperature set point was raised from the default 32°F to 55°F because it would be expected to provide cool enough air at this point without refrigeration.

The trends from this second test are shown in ***Figure 51***. This was done after the indirect coil swap, so the trends are slightly different. The outside air temperature ramp began at 95°F and was reduced at a rate of 1°F every 10 minutes to a low of 50°F before reversing and eventually reaching 100°F. When the outside air temperature reached 80°F, the outside air damper opened as programmed into the economizer. Despite the high transition temperature, the air cooled by the indirect coil was several degrees cooler than the outside air resulting in a lower mixed air temperature and an increased capacity. With the outside air sensor no longer affected by the indirect coil, the system did not experience the damper oscillations displayed in the previous test. Once again, the circulating pump shut off at an outside temperature of 69°F although the pre-cooler continued to provide a cooling effect for about 35 minutes more before it dried out. At this point, the RTU operated basically as it would without any of the pre-cooler components, other than the added airflow restrictions. When the outside temperature reached 55°F, both compressors shut down as programmed. When the outside temperature dropped below 53°F, the damper began modulating to maintain the proper mixed air temperature, and again displayed the "hunting" effect seen previously in *Figure 30*, which is related to the mixed air temperature sensor placement.

FIGURE 51: SECOND ECONOMIZER "V" TEST WITH PRE-COOLER #2 (POLYMER INDIRECT COIL)



The conclusion from these tests show that when an indirect coil is used, the economizer controls need to be thoroughly understood and adjusted properly.

PRE-COOLER #3

Like Pre-cooler #1, Pre-Cooler #3 (***Figure 52***) uses a spray system with polymer media. Unlike Pre-cooler #1, in this system the spray is intermittent and once-through rather than continuous with recirculation. Temperature and humidity instruments pass information back to a microcomputer control system, and the computer then determines the amount of water that can just be evaporated by direct injection into the air, and controls the spray frequency and duration accordingly. The polymer media is primarily there to prevent water droplets from being carried over to the condenser coil, and any excess water that is not evaporated drains out of the bottom of the modules.

The pre-cooler modules also attach directly to the sides of the condenser coil openings of the RTU rather than with additional ducting like the first two pre-coolers. The modules consist of a painted metal box sized for the specific condenser it will be attached to. At the front, a screen prevents the intake of large airborne debris from being drawn into the wet section. With a face area very close to that of the condenser, the face velocity through the pre-cooler was about the same as that of the condenser at 5.2 ft/s; the highest of all four pre-cooler systems. The screen and wet media produced about an 8% reduction in condenser airflow.

Figure 52: Pre-Cooler #3
(Pre-screens removed. Right: Dual control valves)

Between the screens and the evaporative media, several nozzles of specific flow rates and spray patterns are arranged around the top and sides of the box. The two modules tested with this system each had nine nozzles. The nozzles in each module are connected by tubing back to a common control valve. The system was originally configured with one control valve for both modules, but was later reconfigured with two valves: one per module. The valves and the microcomputer draw their power directly from the 24VAC transformer inside the RTU, so the minimal power that they draw is included in the recorded RTU input power. Upstream of the control valves, the water supply line connected to an optional pressure booster pump, since a higher water pressure at the nozzles is expected to produce smaller droplets that should evaporate quicker. The booster pump is set to maintain a specific water pressure, and is attached to a small expansion tank. With the intermittent flow through the nozzles and the small storage volume, the operation of the booster pump was also intermittent. The maximum power recorded for the pump was about 330W, although with the usual half hour tests it averaged less than half of this, and varied with the water injection rate. (Pump power was measured separately from the RTU power for all of the pre-coolers.)

The unit was installed by a product representative contractor, who stayed on-site through the testing to help diagnose operational issues (including facilitating the valve change). In addition, the microcomputer had the ability to communicate operational information back to the manufacturer for their review, and factory representatives provided useful feedback through the testing. In addition to the mentioned measures of ambient temperature and humidity, the system also had a temperature sensor at the pre-cooler outlet, and it sent these measures back along with valve operational information.

The results from the testing are displayed in the usual assortment of figures and tables. ***Figure 53*** has the RTU performance data recorded with the pre-cooler plotted against the baseline performance map, and ***Figure 54*** shows the relative change in performance between them. The values from each of these figures are listed in ***Table 12***. The air-side measured evaporation and make-up water consumption rates are shown in ***Figure 55***, and the gallons consumed per kWh saved are shown in ***Figure 56***. With relatively low power savings through the tests creating a small denominator, the results in this figure are more scattered than for the others. Finally the five calculated values of evaporative effectiveness are listed in ***Table 13***. Some of the measured outlet temperatures in this table are struck out because it was discovered during the valve change that the sensor probe had been in contact with the condenser and thus raising its reading.

FIGURE 53: RTU PERFORMANCE WITH PRE-COOLER #3

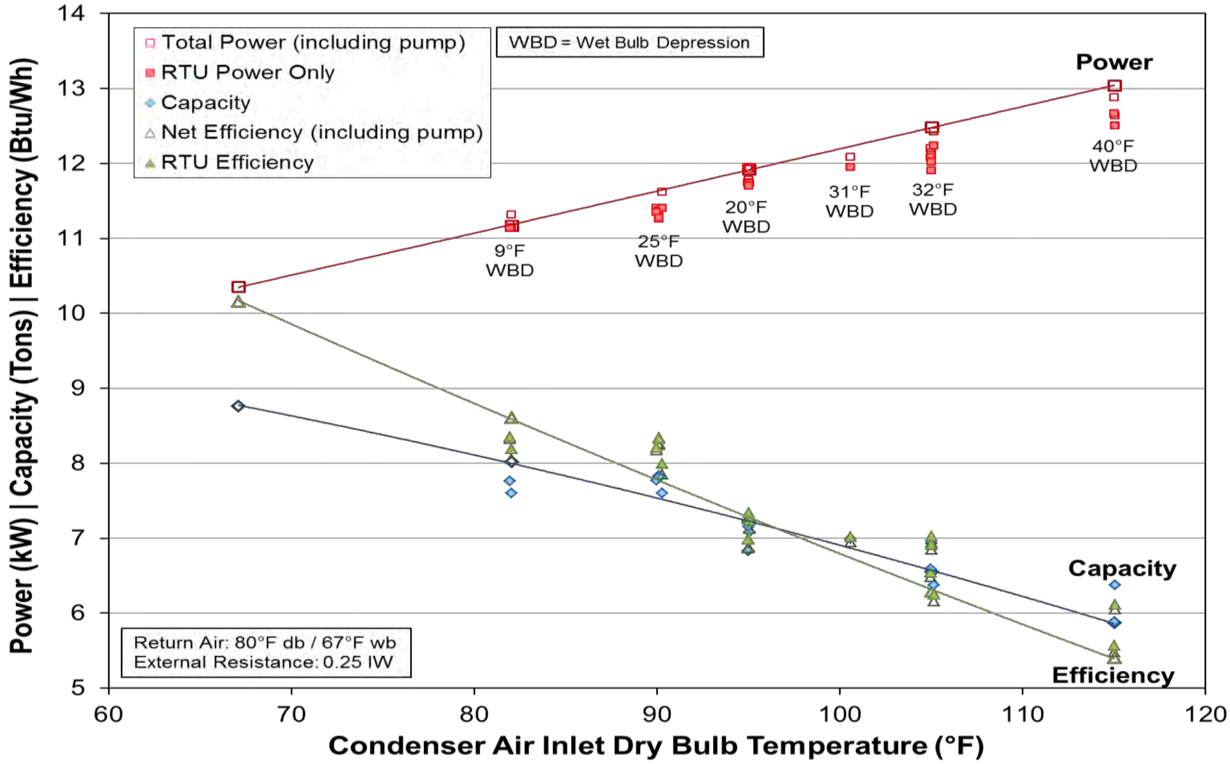


FIGURE 54: RELATIVE CHANGE IN RTU PERFORMANCE WITH PRE-COOLER #3

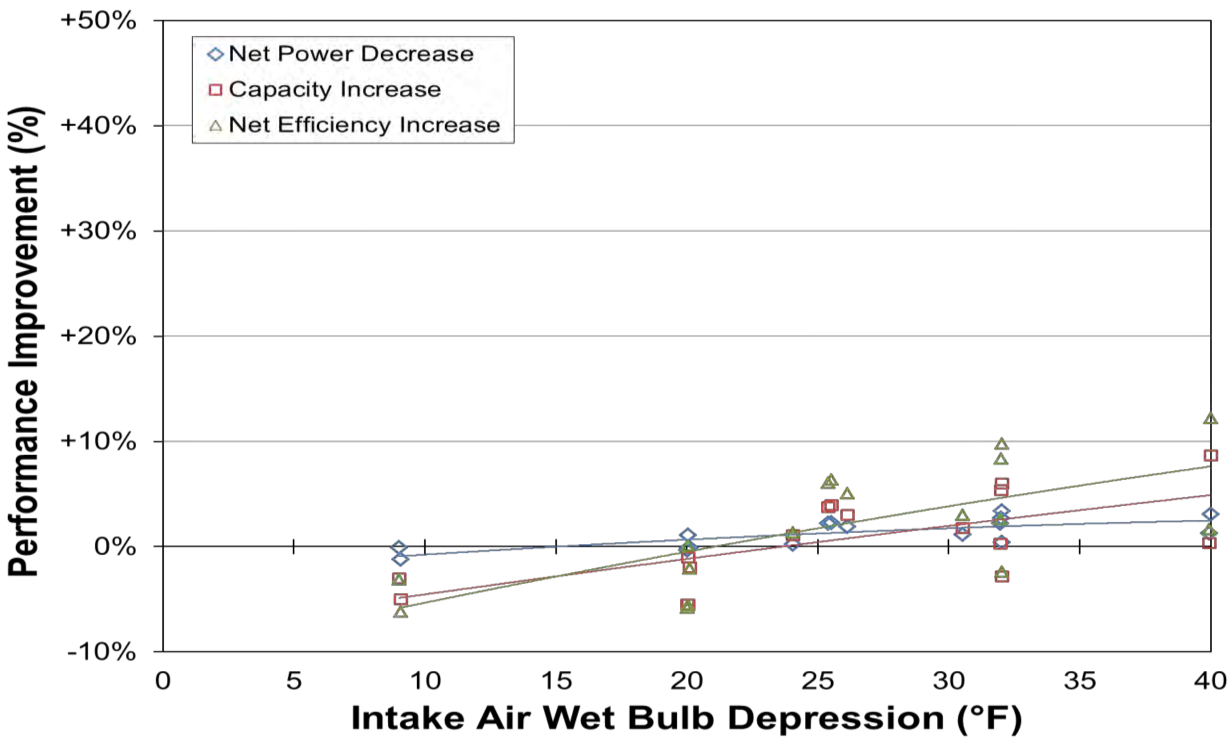


TABLE 12: RTU PERFORMANCE METRICS WITH PRE-COOLER #3

DB	WB	WBD	Control Valves	Condenser Airflow (CFM)	Face Velocity (ft/s)	Pre-Cooler & Condenser Temp Rise	CAP^{WET} Capacity (Tons)	P^{WET} Power (kW)	COP^{WET}	EER^{WET} Efficiency (Btu/Wh)	Relative to Baseline $\%CAP^{inc}$	$\%P^{inc}$	$\%EER^{inc}$	Pump Power (kW)	Net Electrical Savings Demand	Energy
82	73	9	Single	6,041	5.27	19.1	7.60	11.13	2.40	8.19	-5%	-0%	-5%	0.19	-1%	-5%
			Dual	6,029	5.26	19.2	7.76	11.15	2.45	8.35	-3%	-0%	-3%	0.04	0%	-3%
90	64	26	Dual	6,025	5.26	14.7	7.77	11.35	2.41	8.21	+3%	-2%	+6%	0.05	2%	5%
95	75	20	Single	6,016	5.25	15.9	6.92	11.74	2.07	7.07	-4%	-2%	-3%	0.20	0%	-3%
			Dual	5,995	5.23	15.6	7.16	11.70	2.15	7.34	-1%	-2%	+1%	0.08	1%	1%
100	70	30	Dual	6,044	5.28	14.0	6.99	11.95	2.06	7.02	+1%	-2%	+3%	0.14	1%	3%
105	73	32	Single	5,996	5.23	13.0	6.48	12.16	1.87	6.40	-1%	-3%	+2%	0.15	1%	2%
			Dual	5,905	5.15	9.2	6.97	11.90	2.06	7.02	+6%	-5%	+12%	0.15	3%	10%
115	75	40	Single	5,967	5.21	11.5	5.88	12.67	1.63	5.57	+0%	-3%	+3%	0.21	1%	3%
			Average:	6,002	5.24											
				-8%	Reduction from Baseline											

Pump cycles, so this number is the average through the 30-minute test

FIGURE 55: WATER CONSUMPTION RATE WITH PRE-COOLER #3

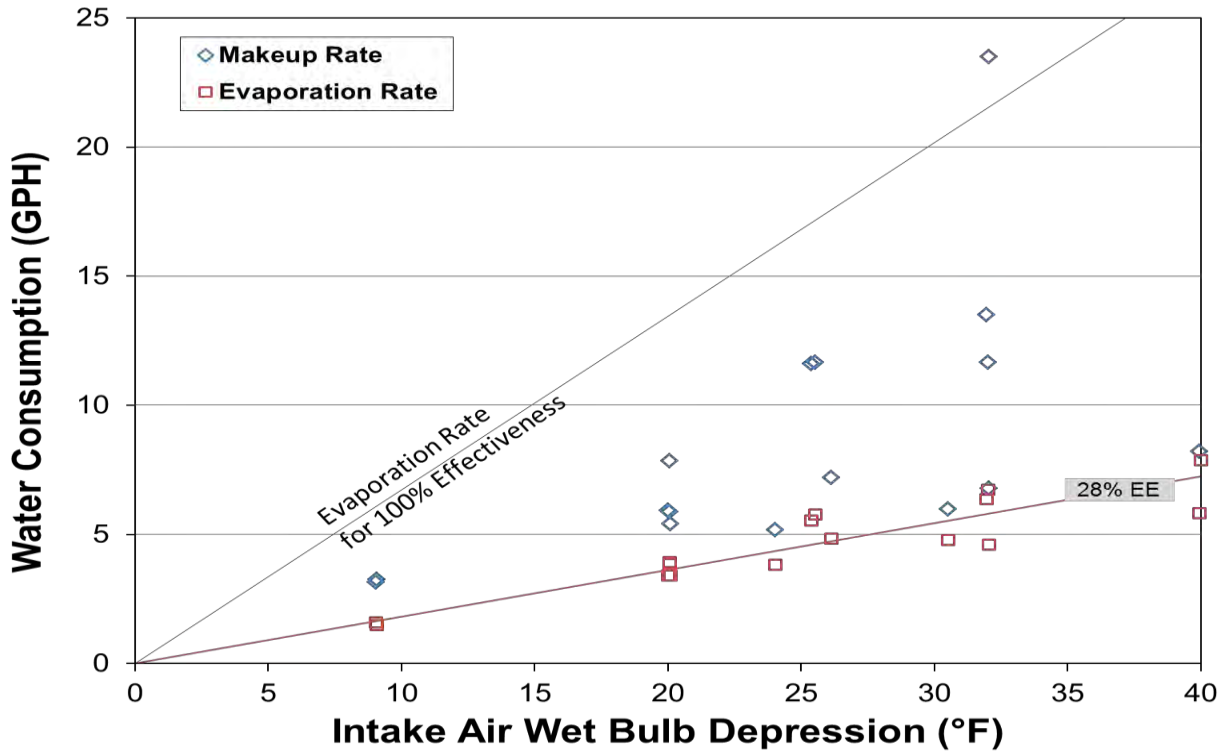


FIGURE 56: WATER USAGE PER UNIT OF ENERGY SAVED WITH PRE-COOLER #3

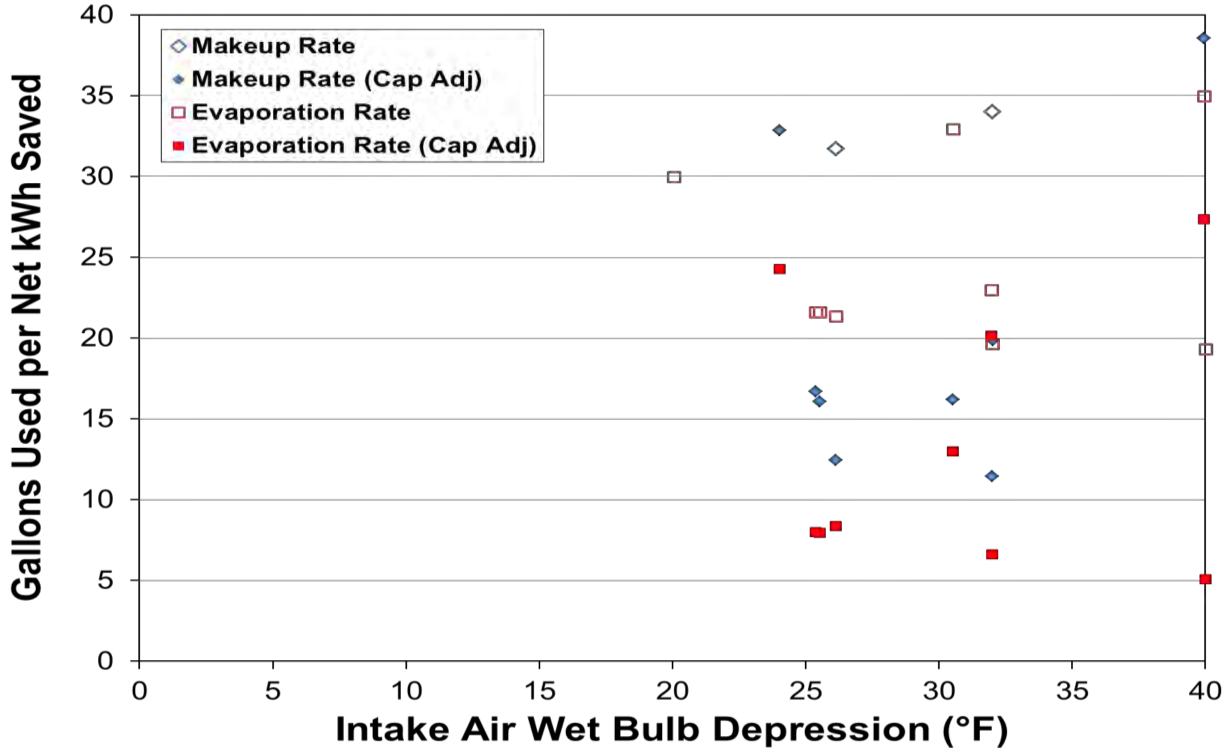


TABLE 13: EVAPORATIVE EFFECTIVENESS AND OUTLET TEMPERATURES FOR PRE-COOLER #3

				Measured Temperature Out		Humidity Ratio Effectiveness		No Pre-Cooler Power Curve		No Pre-Cooler COP Curve		No Pre-Cooler Capacity Curve	
DB	WB	WBD	Control Valves	(°F)	EE	(°F)	EE	(°F)	EE	(°F)	EE	(°F)	EE
82	73	9	Single	~~83.4~~	~~-15%~~	79.9	24%	81.1	10%	85.9	-42%	88.9	-75%
			Dual	79.7	26%	79.7	26%	81.4	7%	84.3	-25%	86.2	-46%
90	64	26	Dual	83.2	26%	82.9	27%	85.0	19%	85.7	17%	86.1	15%
95	75	20	Single	~~95.0~~	~~0%~~	89.8	26%	91.9	16%	97.1	-11%	99.8	-24%
			Dual	88.9	30%	89.2	29%	91.2	19%	94.4	3%	96.1	-6%
100	70	30	Dual	93.8	21%	93.5	22%	95.6	15%	97.7	8%	98.7	4%
105	73	32	Single	~~103.7~~	~~4%~~	96.9	25%	99.4	18%	104.2	3%	106.3	-4%
			Dual	92.9	38%	92.5	39%	94.8	32%	97.6	23%	99.1	19%
115	75	40	Single	~~110.0~~	~~13%~~	117.5	-6%	108.4	17%	113.0	5%	114.7	1%

Note: In initial configuration, our temperature sensor was contacting the condenser coil.

As the test results show, the performance of this pre-cooler sample was disappointing relative to the others, particularly to the manufacturer who had seen significant improvements from several field installations. The issue seems to be a lower than expected evaporation rate, possibly because of insufficient contact time or droplets that are too large. The high air velocity through the media is also a factor. In addition to the valve change (which did not appear to change the situation), the spray rate was adjusted higher several times, which was accomplished remotely by the manufacturer. The only significant effect that this had was an increase in the water consumption rate, as demonstrated by the difference between the make-up flow and the evaporation rate shown in *Figure 55*. The evaporative media that stops the droplets also does not appear to hold on to much of the water, and the excess just sloughs off and drains out the bottom. This may be a temporary phenomenon since polymers tend to be hydrophobic when new, and this water repellant nature tends to degrade over time from age or the buildup of solids. A different media that captures, spreads out, and thus increases the contact time of the water spray with the air would be recommended. The booster pump pressure may also need to be raised so that the spray nozzles can produce smaller droplets. Finally the higher face velocity compared to the other systems will likely decrease evaporative effectiveness.

PRE-COOLER #4

Pre-cooler #4 (***Figure 57***), like Pre-cooler #3, is made up of two modules that attach directly to the condenser air intake without additional ductwork. It also uses a 12-inch thick cellulose evaporative media with an overhead water distributor media like Pre-cooler #2, but without the indirect coil. The water basin in each module is connected by a hose, and single circulating pump in one of them supplies water to the distribution headers in both. Because the pump only needs to lift a small flow of water to the top of the pads, it requires very little power at an average measured value of 37 watts. The media is slightly larger than the condenser air opening and the face velocity is slightly less at 4.9 ft/s. Because of the relatively high velocity and media that was thicker than any of the others, it had the largest impact on the airflow rate through the condenser with a reduction of 11%.

FIGURE 57: PRE-COOLER #4

The pre-cooler includes the most sophisticated water maintenance system of the four, but it created complications to the testing. Rather than a float valve in the water basin, the fill uses a solenoid valve activated by a level switch, so the water make-up flow is intermittent. One of the reasons for this is that it is designed to empty the basin at a programmable interval of inactivity, thus not leaving a pan of water to stagnate, and it needs to lock out the fill when it does this. The release of water to remove solids buildup is also done intermittently through a solenoid valve. The operation of this valve is described as being controlled by a conductivity sensor in the water basin and thus would be sensitive to the evaporation rate. However, in testing, the blowdown appeared to occur at a regular timed interval of 65 minutes with a discharge averaging 3 gallons.

Once again, the results from the testing are displayed in the usual collection of figures and tables. ***Figure 58*** has the RTU performance data on the baseline performance map, and ***Figure 59*** shows the relative change in performance between them. The values from each of these figures are listed in ***Table 14***. The air-side measured evaporation and make-up water consumption rates are shown in ***Figure 60***, and the values of water consumed per unit of energy saved are shown in ***Figure 61***. Lastly, the five calculated values of evaporative effectiveness are listed in ***Table 15***.

FIGURE 58: RTU PERFORMANCE WITH PRE-COOLER #4

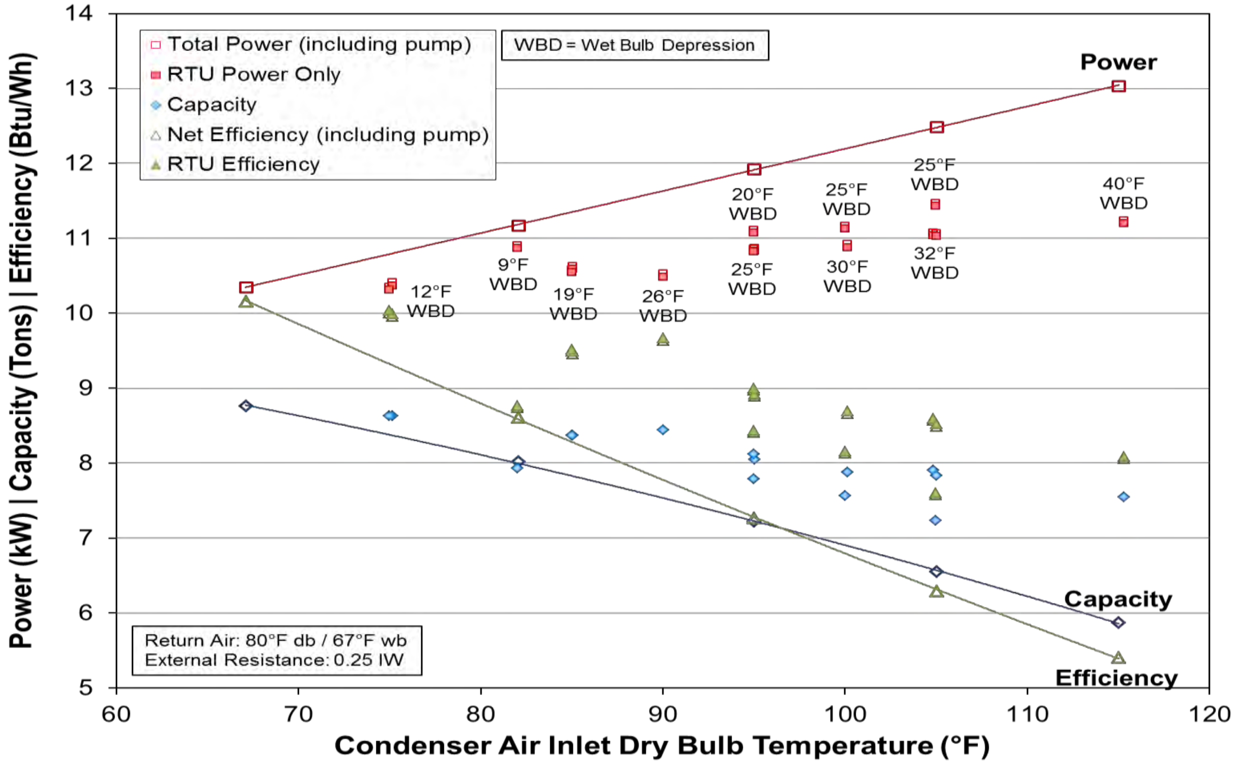


FIGURE 59: RELATIVE CHANGE IN RTU PERFORMANCE WITH PRE-COOLER #4

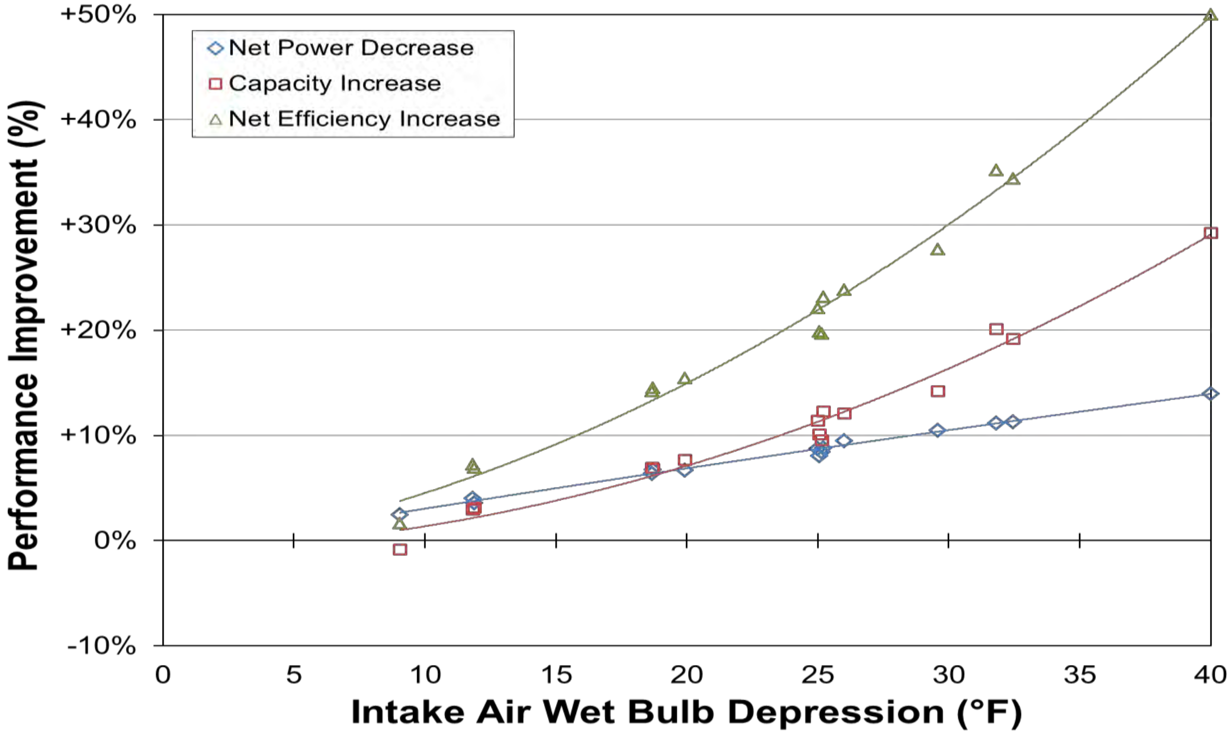


TABLE 14: RTU PERFORMANCE METRICS WITH PRE-COOLER #4

DB	WB	WBD	Condenser Airflow (CFM)	Face Velocity (ft/s)	Pre-Cooler & Condenser Temp Rise	CAP^{WET} Capacity (Tons)	P^{WET} Power (kW)	COP^{WET}	EER^{WET} Efficiency (Btu/Wh)	Relative to Baseline %CAP^{inc}	%P^{inc}	%EER^{inc}	Pump Power (kW)	Net Electrical Savings Demand	Energy
82	73	9	5,883	4.98	14.9	7.93	10.86	2.57	8.76	-1%	-3%	+2%	0.04	2%	2%
90	64	26	5,847	4.95	1.1	8.45	10.49	2.83	9.67	+12%	-10%	+24%	0.04	10%	20%
95	75	20	5,830	4.94	5.8	7.79	11.08	2.47	8.44	+8%	-7%	+16%	0.04	7%	14%
100	71	30	5,837	4.94	-2.2	7.88	10.88	2.55	8.69	+14%	-11%	+28%	0.04	10%	22%
105	73	32	5,816	4.92	-4.4	7.87	11.03	2.51	8.56	+20%	-12%	+36%	0.04	11%	26%
115	75	40	5,786	4.90	-11.4	7.55	11.20	2.37	8.08	+29%	-14%	+50%	0.04	14%	33%
		Average:	5,833	4.94											
			-11%	Reduction from Baseline											

FIGURE 60: WATER CONSUMPTION RATE WITH PRE-COOLER #4

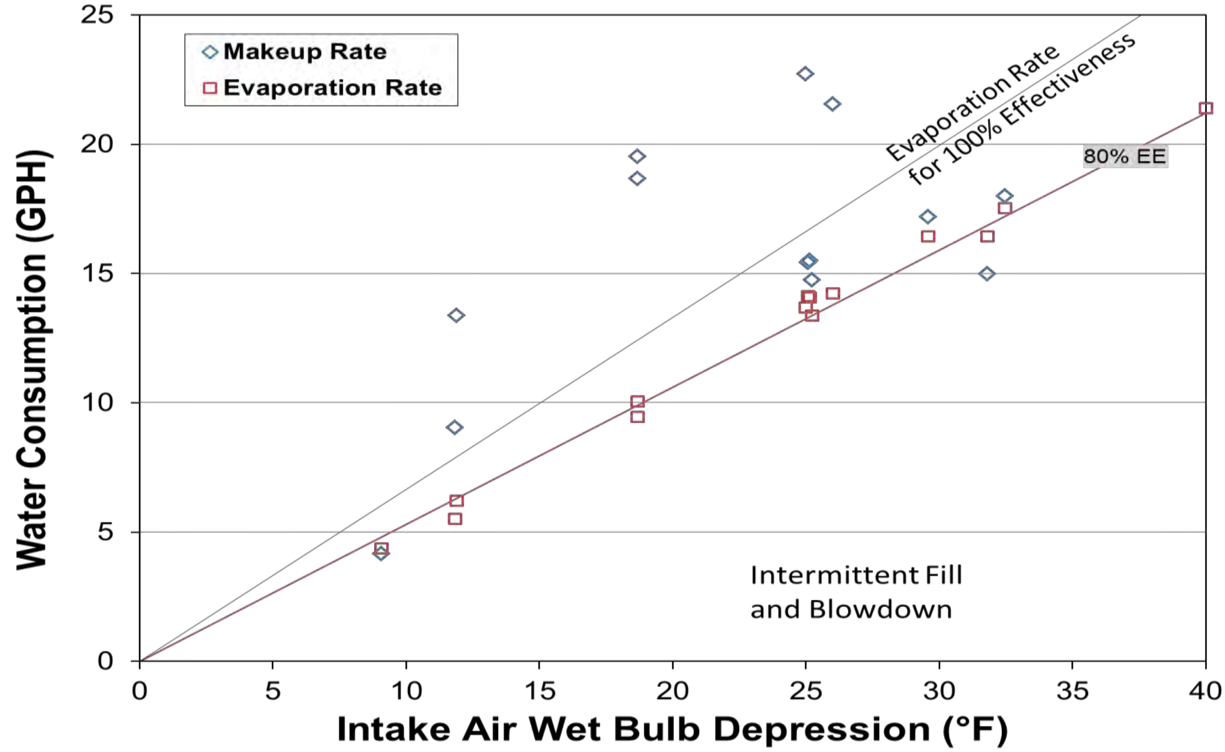


FIGURE 61: WATER USAGE PER UNIT OF ENERGY SAVED WITH PRE-COOLER #3

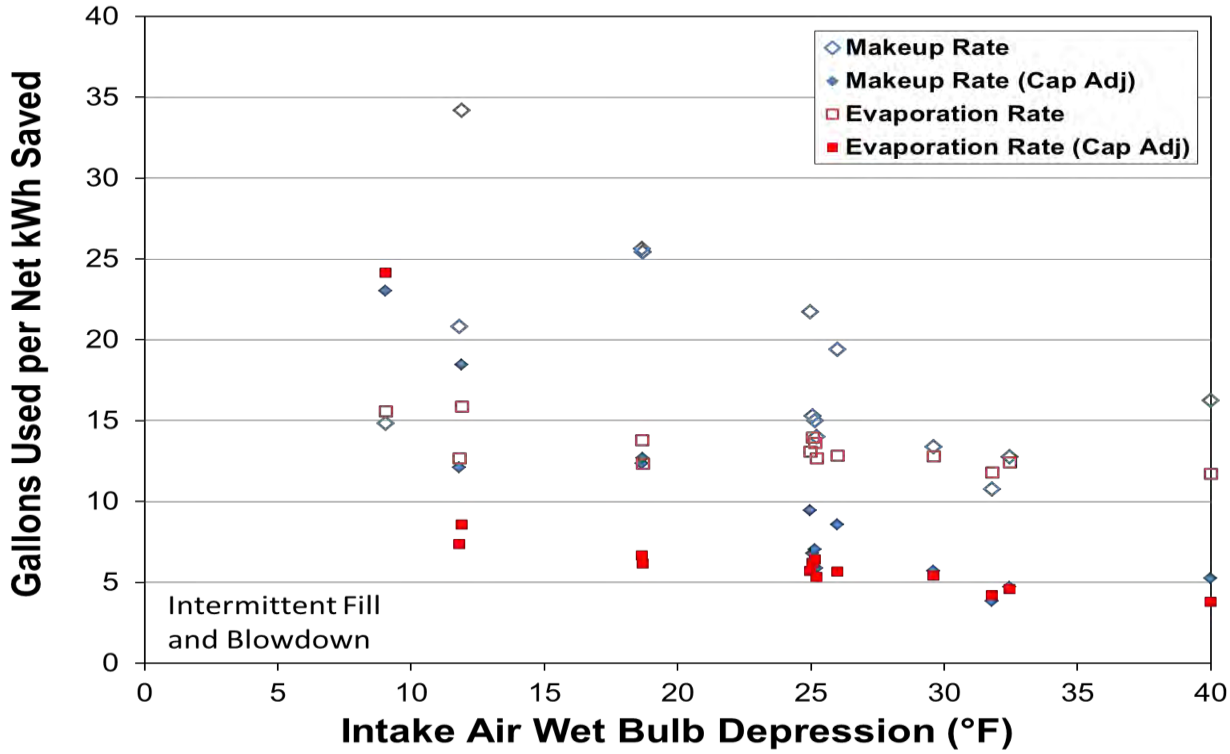


TABLE 15: EVAPORATIVE EFFECTIVENESS AND OUTLET TEMPERATURES FOR PRE-COOLER #4

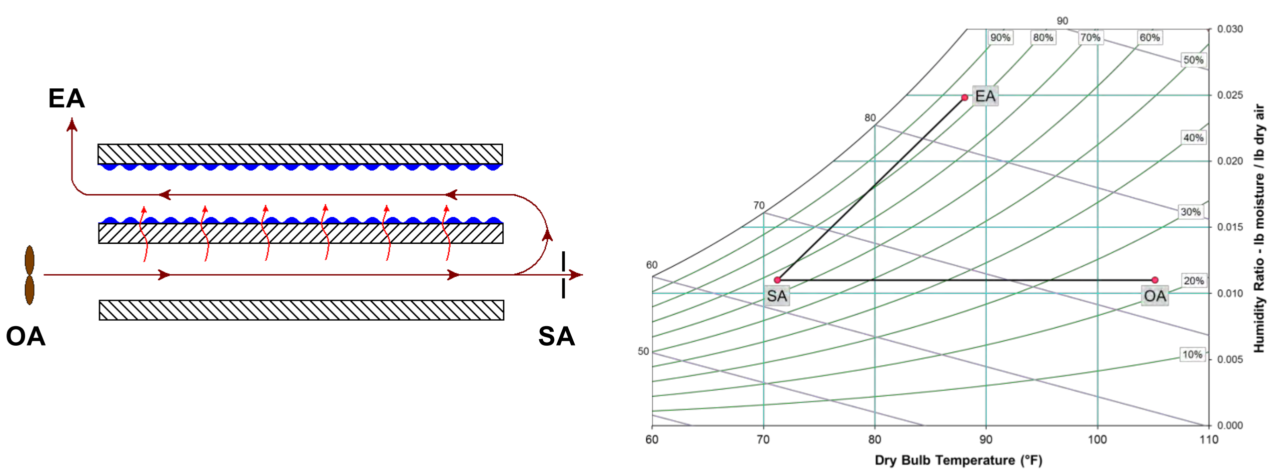
			Measured Temperature Out		Humidity Ratio Effectiveness		No Pre-Cooler Power Curve		No Pre-Cooler COP Curve		No Pre-Cooler Capacity Curve	
DB	WB	WBD	(°F)	EE	(°F)	EE	(°F)	EE	(°F)	EE	(°F)	EE
82	73	9	74.1	88%	75.6	71%	76.3	63%	80.3	19%	83.2	-12%
90	64	26	66.7	90%	68.8	81%	69.6	79%	71.8	70%	73.7	63%
95	75	20	76.8	91%	84.6	52%	80.1	75%	83.5	58%	85.7	46%
100	71	30	73.5	90%	75.4	83%	76.6	79%	81.0	64%	84.1	54%
105	73	32	75.9	90%	79.2	80%	79.4	80%	82.3	70%	83.7	66%
115	75	40	78.9	90%	82.5	81%	82.3	82%	87.0	70%	89.8	63%

PHASE 5: INDIRECT EVAPORATIVELY COOLED VENTILATION AIR

Pre-cooler #4 from the previous section was provided as a package along with an indirect evaporative cooler to pre-treat the ventilation air drawn in through the economizer. This arrangement is similar to what was done with Pre-cooler #2, but with separate independent systems. This indirect evaporative cooler is usually intended for stand-alone use, but as the submission for the Western Cooling Challenge, is being applied as an enhancement to a rooftop unit.

Indirect evaporative coolers (IECs) take advantage of the cooling potential of evaporating water but without increasing the moisture content of the air supplied to the space. Air that is cooled by evaporation is used to cool a second air stream by heat exchange. This IEC uses a unique flow arrangement to achieve maximum temperature drop from this process. Outside air is drawn in using a variable speed fan where it is forced through the dry side of the heat exchanger. As it exits the heat exchanger, a fixed damper that which is set during the installation creates backpressure, which causes part of the air flow to return through the wetted side of the heat exchanger to exhaust back outside. The remaining flow is supplied to the RTU through the outside air intake of the economizer, at a roughly 55%/45% split between the supply and exhaust. The process through the dry side of the heat exchanger does not change the humidity ratio of the air, but it does result in a reduction in both the dry and wet bulb temperatures. With sufficient heat exchange surface, an IEC of this configuration can produce an outlet supply temperature below the entering wet bulb temperature and thus an evaporative effectiveness greater than 100%. The limiting temperature for this type of process is actually the entering air dew point temperature.

FIGURE 62: DESCRIPTION OF A REVERSE FLOW INDIRECT EVAPORATIVE COOLER



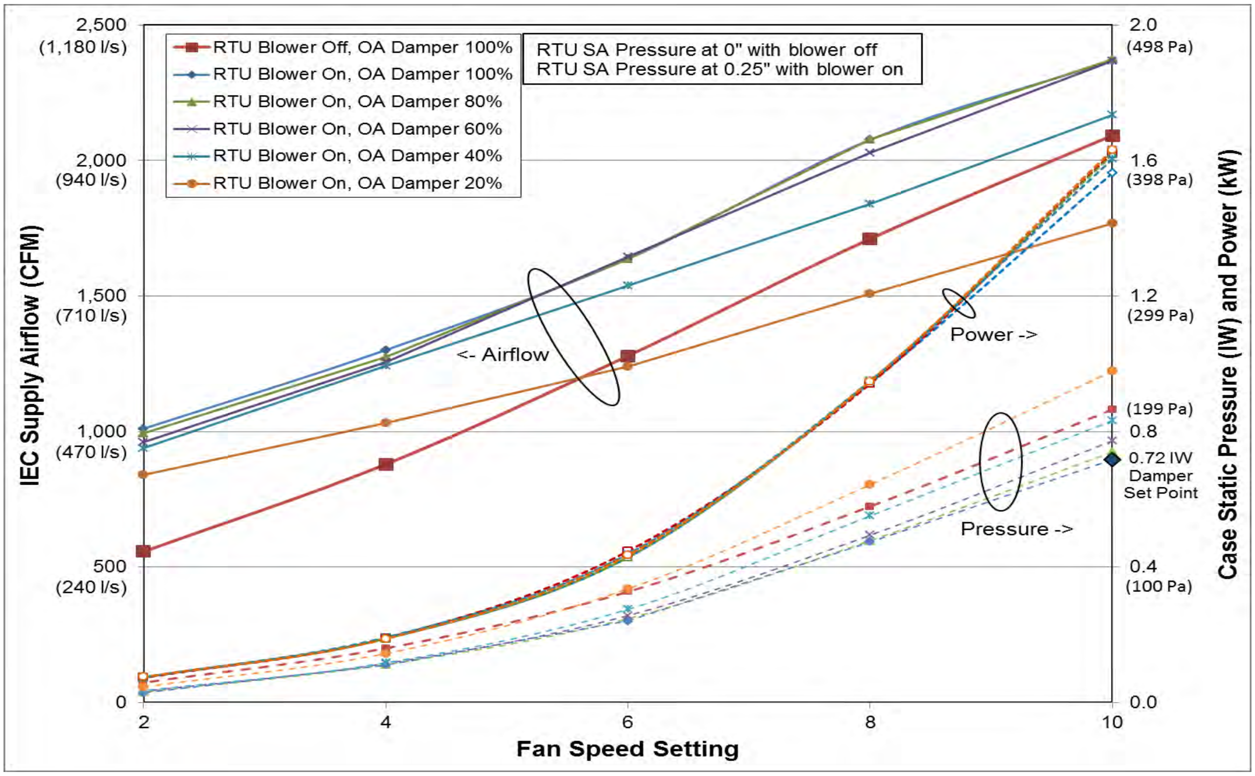
Connection of the IEC to the RTU was done similar to what is shown in *Figure 2*. A duct was fabricated to connect the supply air discharge of the IEC to the outside air intake of the economizer on the RTU. In the middle of this duct was placed an averaging pitot tube array to measure the IEC supply air flow, similar to the one used for measuring the return air flow to the RTU. Between the flow station and the economizer was installed a barometric damper that would allow outside air to enter the duct if the IEC blower was off and the RTU was on and accessing outside air, thus bypassing the flow restriction of the idle IEC. The economizer controller was disabled and a direct voltage signal was provided to the damper actuator to set the dampers to the desired position. (In field applications, the economizer damper control would need a fairly sophisticated control system to coordinate operation with the IEC.) To help remove the humidified air exhausting from the unit, a vent hood was placed a short distance above the exhaust discharge but without a direct duct connection that would affect its operation. Temperature sensors were placed in the outdoor air intake, the supply and exhaust air outlets, and in the water basin. Static pressure sensors were attached to the case of the IEC upstream of the backpressure damper, and at the flow station along with a pair of differential pressure sensors measuring the velocity pressure in parallel.

The first step in testing of this unit was the setting of the fixed damper. The instruction from the manufacturer was to set the pressure at the outlet of the heat exchanger (upstream of the damper) to 190 Pa (0.76 IW) with the IEC fan at full speed, the RTU blower operating, and with the economizer outside air damper 100% open. Although it was not specified in the testing plan, throughout the testing of this unit the supply static pressure of the RTU was held to 0.25 IW when its blower was on and 0 IW when it was off (IEC blower operation only), and the pressure drop in the return duct was uncompensated for by the return air booster fan. So, with the RTU blower on, its supply static pressure set at 0.25 IW, the IEC blower on at full speed, and with its heat exchanger wetted, the fixed damper was adjusted until about 0.76 IW was measured at the pressure transmitter attached to the case. (The adjustment was not very precise, and the final pressure indication was actually around 0.72 IW.)

Following the setting of the fixed damper, a series of short duration tests (10 minute averages) was conducted to map the airflow performance of the IEC; the results of which are shown in ***Figure 63***. This involved measuring the case static pressure (the same measurement point as used for setting the damper position), the unit power, and the airflow measured with the pitot array as functions of the fan speed setting and the position of the outside air damper of the RTU. The variable speed fan of the IEC can be controlled in 10 steps from off to full speed, and the tests were run at the even speed increments (2, 4, 6, 8 and 10). The outside air damper was also varied in even 20% steps from 20% to 100% open. The majority of the tests were conducted with the RTU blower on, except for one sequence which was run with the RTU blower turned off and the outside air damper fully open to demonstrate operation of the IEC alone.

The results show a fairly linear trend of airflow as a function of the fan speed, a second-order trend for pressure, and cubic trend for power. As the economizer damper is closed, pressure builds in the duct from the IEC, and this results in more airflow being diverted through the exhaust path of the IEC and less to the supply. However, this does not affect the total airflow through the IEC by much, resulting in little change in power.

Figure 63: Fan Mapping of Indirect Evaporative Cooler



The testing plan for the performance of the IEC was put together by the WCEC based on a design of experiments to refine the number of configuration variables that the system now involves. The plan initially put forth desired values of airflow for the supply, return, and outside air intake of the RTU, based on nominal values for the RTU supply airflow (3,000 CFM) and the IEC supply airflow (2,500 CFM). The testing plan requested three tests with the IEC operating alone feeding into the RTU with the OA damper fully open at 2500 CFM (100%), 1800 CFM (72%) and 1020 CFM (41%). From the fan mapping tests, fan speed settings of 10, 8 and 5 were selected to reach these flows, and no other adjustments were made to dampers or booster fans to hold the airflow constant. For the tests conducted with the IEC and the RTU operating in concert, the testing plan requested a mix of IEC conditioned air and space return air, and this required a combination of adjustments to the IEC speed and the RTU OA damper position. One pair of tests requested 1/3 of the RTU supply airflow be supplied by the IEC with the other two thirds being return air. A mix approximating this was obtained at an IEC fan speed setting of 5 with the OA damper at 20% open. Another pair of tests requested 5/6 of the supply airflow from the IEC (actually its full output) and the remaining 1/6 be return air. The settings to achieve this were an IEC fan speed of 10 with the OA damper at 75%.

As demonstrated in the earlier testing, there is leakage through the economizer dampers and RTU case, so the airflow measured at the supply of the IEC did not match up precisely with the airflow measured at the RTU supply. In fact, if the RTU blower was not operating, some of the airflow delivered by the IEC through the RTU's outside air intake would find exit paths around the return air damper to go to either the conditioned space through the return duct or exit back outside through the barometric relief damper or other case leaks.

The summary of test scenarios is listed in ***Table 16***, with just the fan speed and damper position settings used rather than the requested airflow rates. Scenarios 10 and 12 had been done previously, as #12 is the RTU baseline testing per AHRI Standard procedures, and #10 is the testing of the RTU with the pre-cooler performed in the previous phase for Pre-Cooler 4. It was not certain at the beginning whether Scenario 8 could be conducted without adjusting the internal controls of the RTU, but it was eventually discovered that if the RTU was started with its blower and one compressor on, the blower could be turned off if the IEC was providing the airflow through the coil. (If not, the compressor would eventually switch off due to high pressure on the suction line.) For Scenario 11, the barometric damper on the duct connecting the IEC to the RTU was propped open to minimize any potential flow pulled through the IEC due to low pressure in the connecting duct.

TABLE 16: LIST OF TESTING SCENARIOS FOR COMBINED RTU, IEC AND PRE-COOLER TESTING

Test Scenario	Description	IEC Fan Speed	RTU OA Damper	RTU Blower	RTU Com-pressors	Pre-Cooler
1	IEC Only	5	100%	*Off*	0	Off
2	IEC Only	8	100%	*Off*	0	Off
3	IEC Only	10	100%	*Off*	0	Off
4	IEC+RTU	5	20%	On	2	Off
5	IEC+RTU	10	75%	On	2	Off
6	IEC+RTU+PC	5	20%	On	2	*On*
7	IEC+RTU+PC	10	75%	On	2	*On*
8	IEC+1C+PC	10	100%	*Off*	*1*	*On*
9	RTU+PC	0	20%	On	2	*On*
10	RTU+PC	0	0%	On	2	*On*
11	RTU Only	0	20%	On	2	Off
12	RTU Only	0	0%	On	2	Off

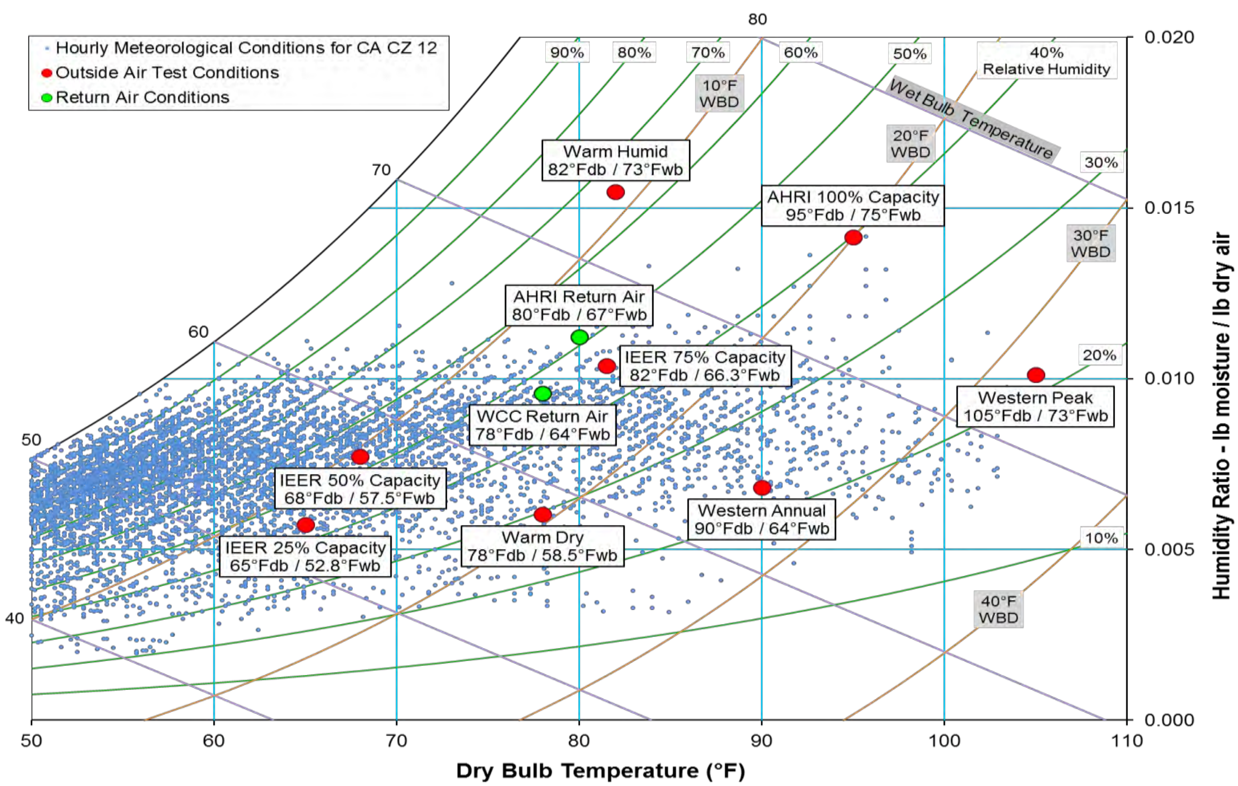
Within each of the twelve test scenarios are a specified set of outside conditions for the RTU and IEC specified for the Western Cooling Challenge. For the majority of them, the space return air was held to 78°F dry bulb and 64°F wet bulb; except for scenarios 10 and 12 which used the AHRI Standard 80°F DB and 67°F WB. These outside condition set points are listed in ***Table 17*** and are shown placed on a psychrometric chart in ***Figure 64***. In the background of this figure has been added the hourly conditions from the Title-24 typical meteorological year for California Climate Zone 12 (middle of the Central Valley, and for San Ramon) to provide a basis for comparison.

TABLE 17: RANGE OF TEST CONDITIONS USED FOR WESTERN COOLING CHALLENGE

	T_{DB} (°F)	T_{WB} (°F)	Corresponds to
Return Air	78	64	WCC Return Air
Outside Air Conditions	105	73	WCC "Peak" *
	95	75	ANSI/AHRI 340/360 "Nominal" & IEER 100% *
	90	64	WCC "Annual", Hot-Dry Average Annual *
	82	73	"Warm Humid" *
	81.5	66.3	ANSI/AHRI 340/360 IEER 75%
	78	58.5	"Warm Dry"
	68	57.5	ANSI/AHRI 340/360 IEER 50%
	65	52.8	ANSI/AHRI 340/360 IEER 25%

* Test condition was also used in the pre-cooler testing

FIGURE 64: WCC TEST CONDITIONS ON A PSYCHROMETRIC CHART

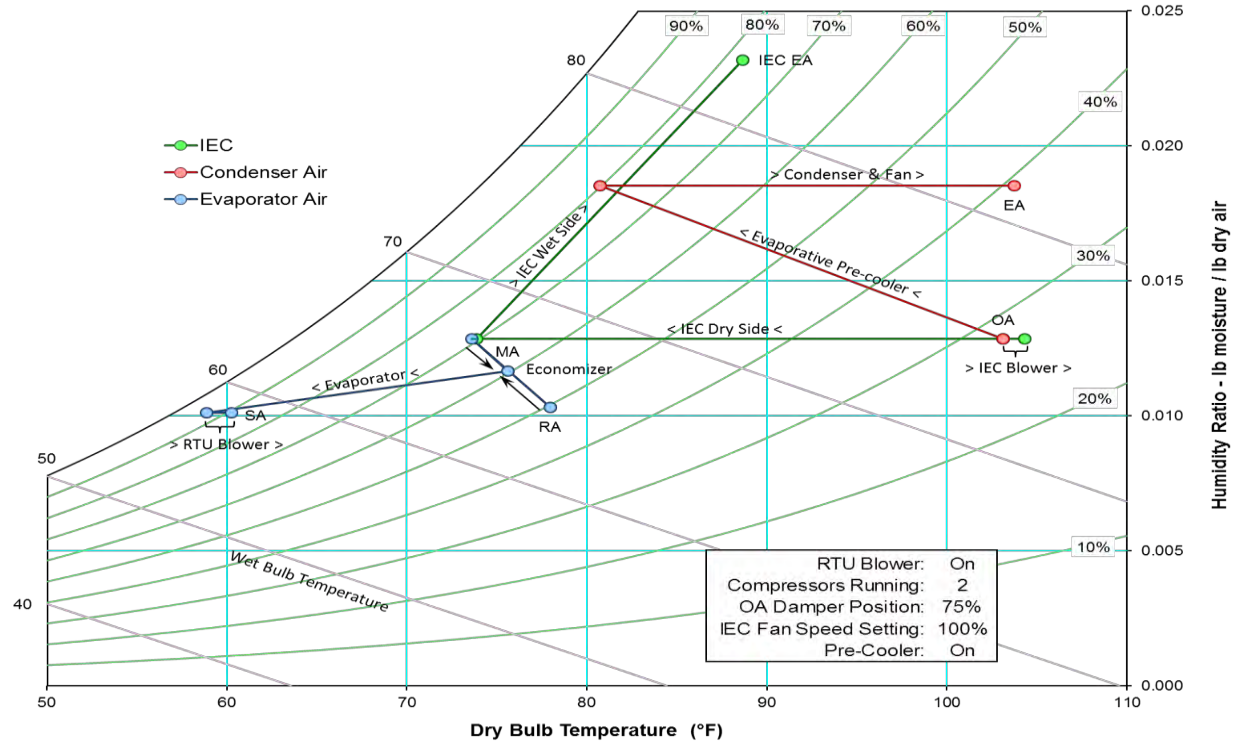


Unfortunately, not all of the test conditions could be achieved; specifically, those with low humidity ratios (less than 0.010 lb/lb). The outdoor test room conditioning system lacks sufficient capacity to dehumidify the air significantly, and the space is usually limited (on the low side) to whatever the humidity ratio of the outside air might be. (Higher humidities can be achieved using a steam humidifier.) Since most of the tests were done in the summer months, humidity ratios were typically at their highest of the year since warm air can hold more moisture. There were a few opportune periods when an "offshore flow" weather pattern created hot and dry conditions, and these were taken advantage of when they occurred including for several test repeats when the earlier test was far from the desired set points. The ability of the room to exhaust the air humidified by the IEC and the pre-cooler, particularly when operating together, is also apparently limited. This also resulted in conditions that were more humid than desired, but still sufficiently stable through the half-hour test periods.

This phase of testing brought together a combination of the various systems examined as phases of this study. To gain a little better understanding of the air conditioning process as it exists with this arrangement, the path of several airflows from one of the many test results have been drawn on a psychrometric chart in ***Figure 65***. Following the red path: outside air (OA) passes through the direct evaporative pre-cooler along a line of constant wet bulb temperature (dropping from 103°F to 81°F), before passing through the condenser coil and fan and heating back up to a temperature only slightly warmer than the outside air as it exhausts from the RTU (EA). Since the condenser "sees" a temperature 22°F lower than the actual outside temperature, its performance is improved as demonstrated during the Phase-4 pre-cooler tests. Following the green path: outside air is also pulled into the IEC where it picks up a little heat from the blower before it is cooled without any change in its moisture content to a temperature less than the wet bulb temperature of the outside air (105% evaporative effectiveness in this case), at which point the flow splits with about 45% of the intake flow passing through the counterflow wetted section of the IEC before it exhausts to the outside, warm and very humid. (The temperature of this exhaust is still less than the outside air and could have also been used for condenser pre-cooling if a separate dedicated system was not used, although it would provide less benefit due to both a higher temperature and a lower flow rate.) Following the blue line: the other 55% of the IEC cooled air is blended with return air (RA) in the economizer mixed air plenum (MA), and from there it is cooled through the RTU evaporator coil and picks up a couple degrees from the indoor blower before exiting as the supply air (SA) to the space. In this example, the temperature of the IEC cooled air is less than the return air, but its humidity ratio is higher; and this resulted in an air enthalpy that is also higher[10]. Thus, the total capacity of the RTU is less than what it would be without the IEC and without any outside air intake, although its sensible capacity is higher, which is of primary importance in hot/dry climates. The total capacity is still considerably better than it would be if the RTU were drawing in the same amount of outside air without pre-conditioning.

[10] Enthalpy lines run approximately parallel to the wet bulb temperature lines.

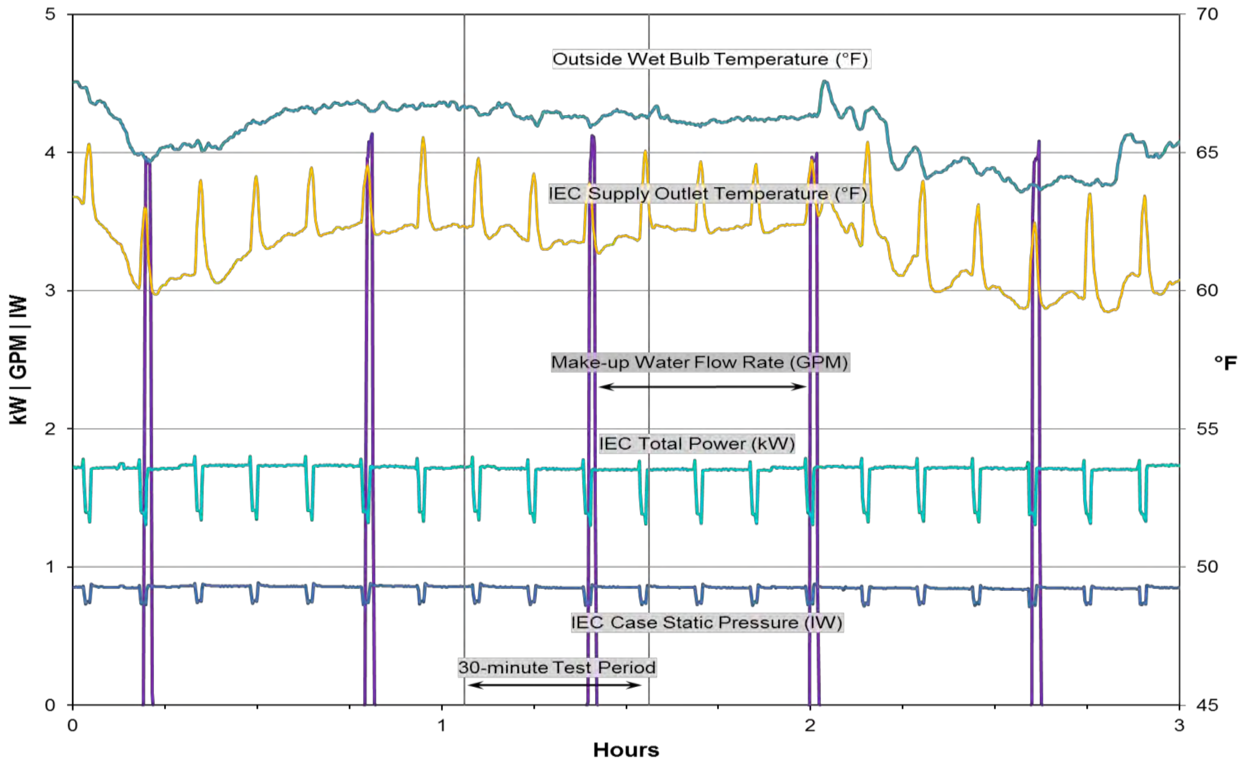
Figure 65: Example Psychrometric Diagram of RTU + IEC + Pre-Cooler



The water management system in this unit is similar to the one used in Pre-cooler #4 (as they are from the same manufacturer) in that it utilizes conductivity sensors to decide when to conduct a blowdown from the sump by opening a solenoid valve. The water level in the sump is also managed with a solenoid valve and a level sensor, and is kept closed during the blowdown sequence, which would not be possible with a float valve. Perhaps because of the locally good quality water, a blowdown while operating was never observed through the course of testing. The only time that a discharge occurred was over the weekends, since the system is programmed to drain the sump after a long period of inactivity.

In addition to the intermittent operation of the water fill and blowdown, the operation of the circulating pumps is also intermittent rather than continuous. Also, when the pumps are activated, the blower slows down to allow the media to be wetted and then speeds back up when the pump shuts off. This results in performance trends similar to the example shown in ***Figure 66***. This was a three hour recording period at a sample rate of every 10 seconds, out of which a single 30-minute test point was extracted (starting just after the one hour mark). In the trend of IEC power, the activation of the pump is indicated by a short increase, followed by a large drop as the blower speed is reduced. The speed reduction is also reflected in a drop in static pressure measured upstream of the backpressure damper. When the pump shuts off, there is a brief drop in power before the blower speeds back up. The outlet supply temperature is also affected by the pump operation, and shows a rise as the wetted media reaches the temperature of the water circulated from the sump. This provides an explanation as to why the pump operation is intermittent as it demonstrates that it is better able to achieve supply temperatures below the entering wet bulb temperature by only being periodically wetted. In this example, the pump cycles are occurring at an even interval of about every 9 minutes and last for about 70 seconds. In the background, the make-up water flows are occurring at a regular interval of about every 36 minutes and last for about 80 seconds admitting 6.7 gallons of water.

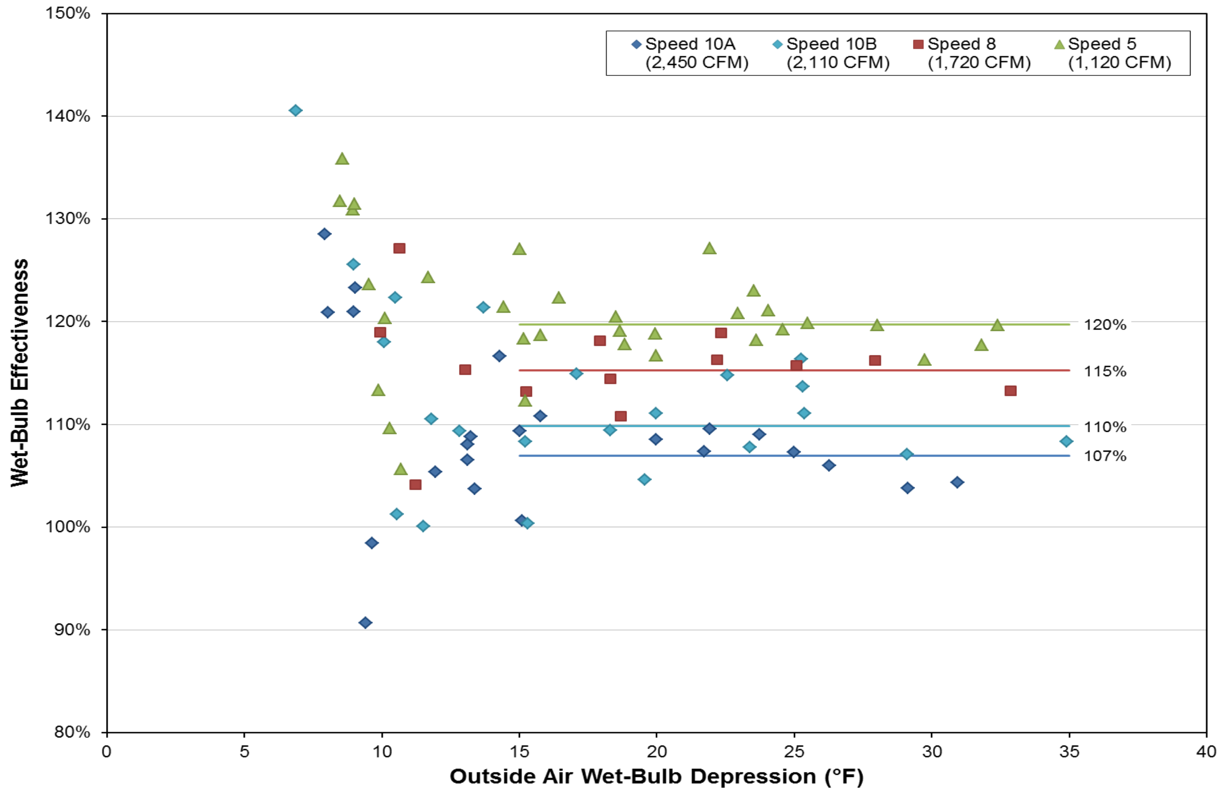
FIGURE 66: EXAMPLE OF PERFORMANCE TRENDS WITH INTERMITTENT OPERATIONS



The consequence of this intermittency is that the test operating tolerances specified by ASHRAE Standard 37 for outlet temperatures cannot be met, and the uncertainty in the test results is elevated. Where the 30-minute test window is drawn will have an effect on the derived averages and totals depending on how many of the periodic events are captured. This type of testing challenge will occur more frequently as equipment is made that uses controls to optimize performance to level which were not possible before digital computer controls.

Figure 67 is a plot of the IEC effectiveness, defined as the temperature reduction from the outside air to the IEC supply air divided by the outside air wet-bulb depression (similar to Equation 6), as a function of the wet-bulb depression and grouped by various flow rates and blower speeds. For Speed 10, two airflow rates are presented, which depended on whether the RTU blower was operating (A is on, B is off). The results show an increase in effectiveness with decreasing airflow. Above a wet-bulb depression of about 15°F, the effectiveness values appear to flatten out for each flow rate. Below 15°F, the denominator in the ratio is becoming so small as to accentuate small deviations in the supply temperature reduction, leading to a more scattered pattern. The key point of this chart is to show that for the vast majority of test scenarios, the effectiveness was greater than 100%, meaning it was supplying air at a temperature less than the outside air wet-bulb temperature.

FIGURE 67: IEC EFFECTIVENESS AS A FUNCTION OF OUTSIDE WET-BULB DEPRESSION AND AIRFLOW



APPENDIX SUMMARY

A summary of the results from the large number of tests with this system is given in the Appendix. This data can be used in the further analysis of the performance of this compressor-less cooling system and in the planning of additional lab or field research. ***Table A-8*** presents the set point values already given in *Table 16* and *Table 17* (labeled as "Target") along with the actual values achieved. The rows are grouped by the test scenarios given in *Table 16* and sub-divided by the test conditions given in *Table 17*. As noted previously, the outside room wet-bulb temperature could not always be controlled due to the local climate conditions. In the section describing the airflow rates, "SA Rate" refers to the supply airflow leaving the RTU as measured by the indoor room code tester, and "RA Rate" refers to the space return airflow to the RTU as measured by the pitot tube array. When the IEC unit was operating, the "OA Rate" refers to the supply airflow from the IEC as measured by its pitot tube array, and the "Net" column refers to the difference between the supply airflow and the sum of the return and IEC airflows. When the IEC is not operating, the "OA Rate" is the net difference between the supply and return airflows, and the "Net" column is blanked out.

For the first three scenarios with the IEC operating alone but feeding into the RTU, the Net airflow values are all negative as an indication of air that is not being delivered through the supply air path and either entering the space through the return duct, or escaping back outside via the barometric damper or other leaks in the RTU case. Some of the difference represented by the net value can also be attributed to measurement error, as the pitot tube array systems have a lower accuracy than the code tester, particularly when the less than optimum ducting is considered. This demonstrates that if the application allows for the IEC operating alone, more needs to be done than simply turning off the RTU blower. This may justify the use of ducting that bypasses the RTU.

The three tables following (*Table A-9*, *Table A-10*, and *Table A-11*) include the exact same set of measured performance metrics, but are then compared against a different reference baseline system. To accomplish this, a more detailed model of the performance of the RTU as a stand-alone system needed to be developed beyond what is described in *Figure 36*. Specifically, this model includes the effect of differences in the return air condition from the 80°Fdb/67°Fwb standard. Performance metrics from the data used in *Figure 36* were combined with other test results (particularly those from Scenario 11) to derive bi-quadratic equations for capacity, power, and sensible heat ratio as functions of the outside air dry-bulb temperature and mixed air wet-bulb temperature, similar to those used in the California Title-24 Alternative Calculation Manual[11].

The first block of columns in the tables present the measured performance metrics for the combination of system components, including total and sensible cooling capacity (referenced to the space return air condition and using the RTU supply airflow measurement), the total power for all systems, and the resulting total and sensible system energy efficiency ratios (EERs). The last column is the total make-up water (MUW) measured for the IEC and pre-cooler normalized to an hourly rate (GPH). This number is only for reference as it does not represent a real consumption rate due to the intermittent nature of the fills for both systems, which may or may not have been captured in the 30-minute test window.

The next section presents the metrics for the IEC, beginning with the wet-bulb effectiveness. Following this, three values of cooling capacity are given. The first is the cooling effect to the intake air (OA Cap), based on the measured IEC supply flow rate and the difference between the outside and supply temperature. This capacity is totally sensible cooling since there is only a temperature reduction with no change in humidity ratio through the IEC. The next two measures of capacity use the space return air as the reference instead of the outside air, and are calculated for both total and sensible cooling effect. In all of the test scenarios, the sensible cooling (RA C-s) resulted in a positive value, indicating that the IEC supply temperature was always less than the return temperature of the space. However, there are a few instances where the total cooling effect (RA C-t) is negative because the humidity ratio of the outside air and hence the IEC supply air is greater than that of the space, creating a latent heat gain that is greater than the sensible cooling. (This effect was shown in Figure 65, which is from Test 7b on the list.) This latent heating load is usually ignored in building cooling analyses that involve high outside air supply since it will normally be pushed through the space and exhausted back outside and thus not present an extra load to a DX cooling coil. This is also an unusual condition for a hot/dry climate.

The next column in this section is the measured IEC power, which is almost all fan power except for the intermittent operation of the pump. The last column is an estimation of the water evaporation rate, which is based on an assumed 55%/45% split between the supply air and the exhaust air, as given by the manufacturer (thus, the exhaust airflow is approximately 82% of the measured supply airflow). An energy balance on the process says that the heat removed from the total intake airflow to bring it to the supply conditions must equal the exhaust flow multiplied by the enthalpy rise from the supply to the exhaust.

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[11] http://www.energy.ca.gov/2012publications/CEC-400-2012-006/CEC-400-2012-006-CMF-REV.pdf

This energy balance was used to determine the exhaust enthalpy, from which the exhaust humidity ratio was calculated using the measured exhaust temperature. The estimated evaporation rate is then the exhaust airflow rate multiplied by the difference in humidity ratio between the exhaust and supply (or intake). Test scenarios where the IEC was not used are blanked out.

The next section is a brief summary of the pre-cooler consumption rates of power (in Watts) and water (GPH). (A more detailed analysis of this system was given in Phase 4 for Pre-Cooler #4.) The water consumption is again just an estimate of the evaporation rate based on air measurements and does not include the blowdown water for maintenance, and the power is the average power consumption of the pump. The evaporation rate for the pre-cooler is usually much higher than the IEC because of the larger relative airflow rates, although the humidity ratio change through the IEC exhaust is usually larger. Test scenarios that do not involve the pre-cooler are again blanked out.

In ***Table A-9***, the baseline comparison is against the RTU as if it were not using any outside air. This is almost the same performance map as shown in *Figure 36*, except for the adjustment for the lower return air wet-bulb temperature used during most of the tests (64°F versus 67°F). The lower wet-bulb temperature means a lower latent cooling load, and as a result a higher sensible heat ratio and a lower total cooling capacity. The first three columns are the values of total and sensible cooling capacity and power derived from the performance model bi-quadratic equations. From these are calculated the total and sensible energy efficiency ratios (capacity divided by power). The last three columns are the percent savings of the combined system performance as compared to this reference baseline. The demand savings just looks at the difference in total power between the two systems. The other two columns provide energy savings that take into account differences in capacity as well as power, for both total and sensible cooling. Thus, if one system has a greater capacity, it will be able to run for less time to satisfy the same load, and thus use less energy even if the demand values are the same. Mathematically, these percent savings values are one minus the ratio of the baseline EER to the modified system EER. Negative values in these columns mean that the modified system uses more energy than the baseline, which usually involves situations where outside air with a high humidity ratio is being brought in through the IEC. For the few situations where the modified system does not provide any cooling relative to the return air (usually with the IEC operating alone with humid air), the savings value is replaced with "#N/A" since the baseline system would provide cooling while the modified system would not.

To add some more fairness to the comparison, ***Table A-10*** models the baseline system as if it were drawing in enough outside air to represent 25% of the supply air. 25% was chosen because the damper leakage testing in Phase 3 showed that there would be 15-20% outside air provided with the dampers closed, and this allows for the damper to be cracked open to a "minimum" position. The same modeled bi-quadratic equations are applied to calculate capacities and power, but this time using the mixed air condition made up of a blend of 75% return air and 25% outside air for the entering wet-bulb temperature. The capacities resulting from these equations are then referenced back to the return air condition to obtain the apparent cooling capacities, and these are what are shown in the table along with the EERs derived from them. With a 25% outside air fraction, the savings for the modified system are improved from the case with no outside air in most scenarios.

In the final comparison, the baseline in ***Table A-11*** determines what the apparent performance of the RTU would be if it were to bring in the exact same outside air fraction as the modified system. The first column in this set describes what this fraction is, which is an averaged value of three outside air fraction values derived from using (respectively) the IEC flow, the RA flow(Phase 3a Method 1), and the mixed air temperature methods (Phase 3a Method 2). There are a couple cases in this set where the baseline system was unable to provide any sensible cooling relative to the return air, meaning the supply temperature was higher than the return temperature after having to cool down a large fraction of outside air.

The actual savings potential of the combined system will be a function of the control scenario of when it would operate in its widely various modes, and for how many hours in the year that the conditions apply to put it into each mode. This involves modeling the performance of both the modified system and the baseline system with a simulated building load in different climates, or obtaining field test data. The Western Cooling Efficiency Center has conducted a field study of a similar system to the modified unit, except without the pre-cooler, and the results of this study may be reviewed in Reference 9. The WCEC will also be taking the results from these laboratory tests to analyze the qualifications of this combined system for their Western Cooling Challenge, and will be producing their own report with annualized energy savings built upon these steady-state test results.

RECOMMENDATIONS

This report presented a comprehensive examination of several technologies that can alter and potentially improve the performance of a packaged rooftop air conditioner. The results from these tests provide support for the continuation or inclusion of these technologies into energy efficiency incentive programs and compliance options for CEC Title 24.

- The testing results have shown that the Quality Maintenance service can provide a measurable benefit even with minimal system adjustments. This result justifies the continuation of the Quality Maintenance program.
- The results from the economizer and pre-cooler testing will need to be incorporated into building simulation programs like eQUEST[12] and CBECC-Com[13] to develop models of annualized system performance.
- Economizers are required by Title-24 in most commercial RTUs and study results can be used to support retrofits of existing units and replacement of controllers.
- The main barrier for adoption of combinations of these technologies is the need for a control protocol for optimizing system performance to the climate where the system is installed.

As with most experimentation, the results from these tests raise more questions that require further study. One of the more crucial areas as demonstrated in the last phase is how to best control a variety of system components to provide the required level of cooling for the least cost in energy, and in some cases water. There is interest in the HVAC community towards developing an annualized or load-based performance method of test (MOT) and metrics for RTUs that takes into account the intake of outside air, whether required or when it is actually beneficial in reducing the refrigeration requirement. It is hoped that the results from these tests can be used as a step towards development of the new metric.

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[12] http://www.doe2.com/equest/

[13] http://www.energy.ca.gov/title24/2013standards/2013_computer_prog_list.html

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APPENDICES

TABLE A-1: MEASUREMENTS AND INSTRUMENTATION FOR LABORATORY TESTS

Measurement	Instrument	Make	Accuracy
Barometric Pressure	Multi-function weather station on roof of building	Vaisala WTX520	±0.007 PSIA (±50 Pa)
Return air dry-bulb temperature	Average of four fast-response RTDs inserted through wall of duct attached to test unit return	Burns Engineering	±0.2°F
Return air dew-point temperature	Chilled mirror dew point sensor	General Eastern Hygro-M2+	±0.36°F
Return air static pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the duct entering the unit	Rosemount 3051C	±0.04% of span (-3 to 3 IW)
Return air flow rate	Full duct averaging pitot tube array with low range differential pressure transmitter (0.1 IW span)	Air Monitor Fan-E Flow Station and Veltron II transmitter	±0.1% of span
Supply air discharge dry-bulb temperature	Average of six fast-response RTDs inserted through wall of duct attached to test unit return.	Burns Engineering	±0.2°F
Supply air discharge dew-point temperature	Chilled mirror dew point sensor	General Electric Optica	±0.36°F
Supply air static pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the duct leaving the unit	Rosemount 3051C	±0.04% of span (-3 to 3 IW)
Supply-return differential pressure	Pressure transmitter connected between the supply and return manifolds.	Rosemount 3051C	±0.04% of span (-4 to 4 IW)
Supply airflow station upstream static pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the flow box upstream of the nozzle partition	Rosemount 3051C	±0.04% of span (-1 to 3 IW)
Supply airflow station differential pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the flow box on both sides of the nozzle partition	Rosemount 3051C	±0.04% of span (0 to 4 IW)
Supply airflow station dry bulb temperature	Single fast-response RTD upstream of nozzles	Burns Engineering	±0.2°F
Outside air/condenser intake dry-bulb temperature	Average of eight fast-response resistance temperature detectors (RTDs) arrayed across the condenser air intake.	Burns Engineering	±0.2°F
Outside air dew-point temperature	Chilled mirror dew point sensor	General Eastern Hygro-M4	±0.36°F
Outside air wet-bulb temperature	Average of four fast-response resistance temperature detectors (RTDs) each enclosed in a wetted wick and with a blower for air movement.	Burns Engineering	±0.2°F
Outside air intake at Economizer	Average of 4 Type-T thermocouples	Therm-X	±0.5°F
Mixed air plenum dry-bulb temperature at filters	Average of 12 Type-T thermocouples attached to filter supports. Bendable 48" averaging RTD	Therm-X Minco S457PE	±0.5°F ±0.25%
Mixed air plenum static pressure	Pressure transmitter attached to port through side of RTU into mixed air plenum.	Rosemount 3051C	±0.04% of span (-2 to 1 IW)

Exhaust air dry-bulb temperature	Average of four fast-response RTDs inserted through wall of duct attached to test unit exhaust	Burns Engineering	±0.2°F
Exhaust air dew-point temperature	Chilled mirror dew point sensor	General Eastern Hygro-M2	±0.36°F
Exhaust air static pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the duct leaving the unit	Rosemount 3051C	±0.04% of span (-2 to 2 IW)
Exhaust airflow station upstream static pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the flow box upstream of the nozzle partition	Rosemount 3051C	±0.04% of span (-2 to 2 IW)
Exhaust airflow station differential pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the flow box on both sides of the nozzle partition	Rosemount 3051C	±0.04% of span (0 to 4 IW)
Exhaust airflow station dry bulb temperature	Single fast-response RTD upstream of nozzles	Burns Engineering	±0.2°F
Compressor suction pressures (2 circuits)	Pressure transmitter attached to compressor suction (vapor) line Schrader valve	Rosemount 3051C	±0.04% of span (0 to 300 psig)
Condenser outlet pressures (2 circuits)	Pressure transmitter attached to liquid line Schrader valve	Rosemount 3051C	±0.04% of span (0 to 400 psig)
Refrigerant temperatures (compressor suction and discharge, condensed liquid before and after filter/drier)	Type-T thermocouples (10 total) clamped to outside of refrigerant tubing, coated with thermal paste and wrapped in insulation	Therm-X	±0.5°F
Water supply flow	Positive displacement water meter with analog output for flow rate and pulse output for totalization	Badger M25	±1.5% of scale
Water supply temperature	Single fast-response RTD inserted into supply line	Burns Engineering	±0.2°F
Water basin temperature	Single fast-response RTD in recirculation basin near pump intake	Burns Engineering	±0.2°F
Water discharge	Catch basin on electronic scale	Measuretek	±0.2 lb
Unit Supply Power, Voltage and Current	3-element true-RMS power meter with outputs for total power, 3-phase voltage and 3-phase current	Yokogawa 2533	±0.2% of reading ±0.1% f.s.
Sub-components Line Current	Clamp-on current transmitter on one leg of the power feeding each of two compressors and two fans	NK Technologies ATR1	±1% of f.s. (±0.2 A)
Blower/Fan Speeds	Optical tachometers (2) reading reflective tape on indoor blower and condenser fan.	Monarch ACT-1B	±1 RPM or 0.005% of reading
Temperature Calibration Reference Standard	Electronic thermometer	Fluke 1502A	±0.015°F
High Pressure Calibration Reference Standard	Pressure calibration standard	Condec UPC5200	±0.05% of f.s.
Low Pressure Calibration Reference Standard	Precision manometer	Dwyer 1430	±0.00025 I.W.

FIGURE A-1. LINKED PARALLEL BLADE DAMPER OUTSIDE AIR TEST WITH 0.25 IW SUPPLY STATIC PRESSURE

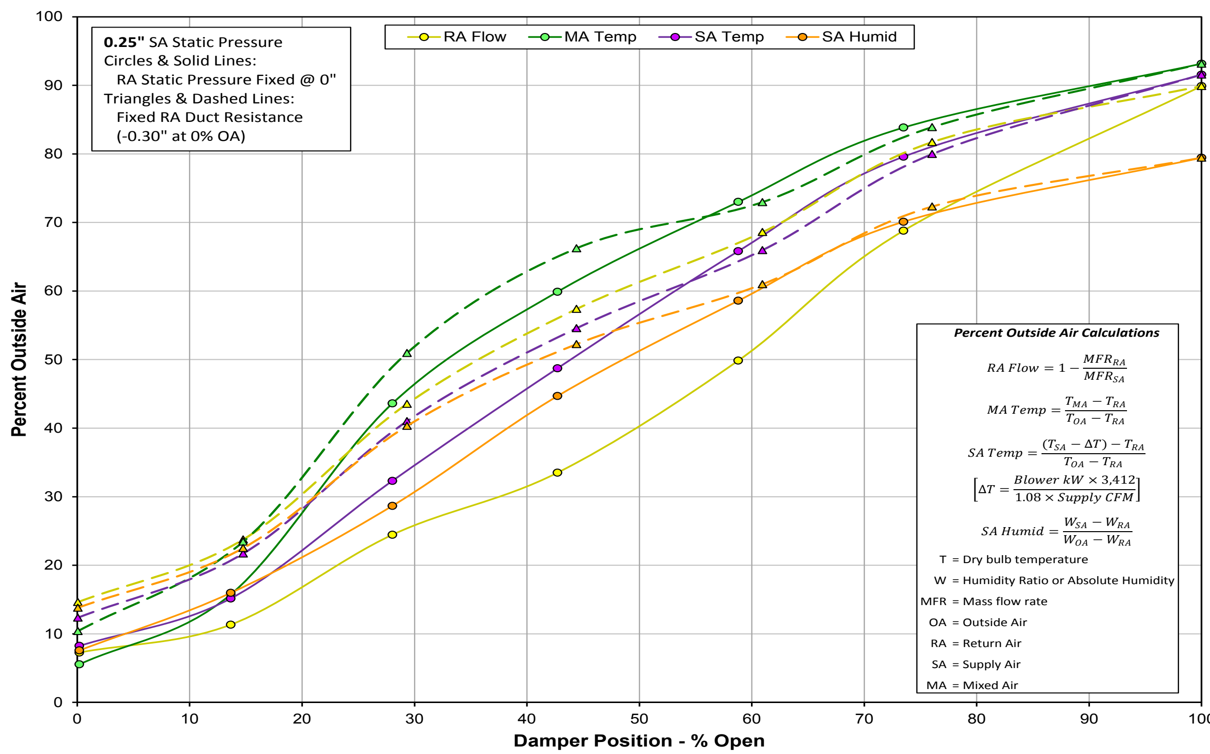

FIGURE A-2. LINKED PARALLEL BLADE DAMPER OUTSIDE AIR TEST WITH 0.50 IW SUPPLY STATIC PRESSURE

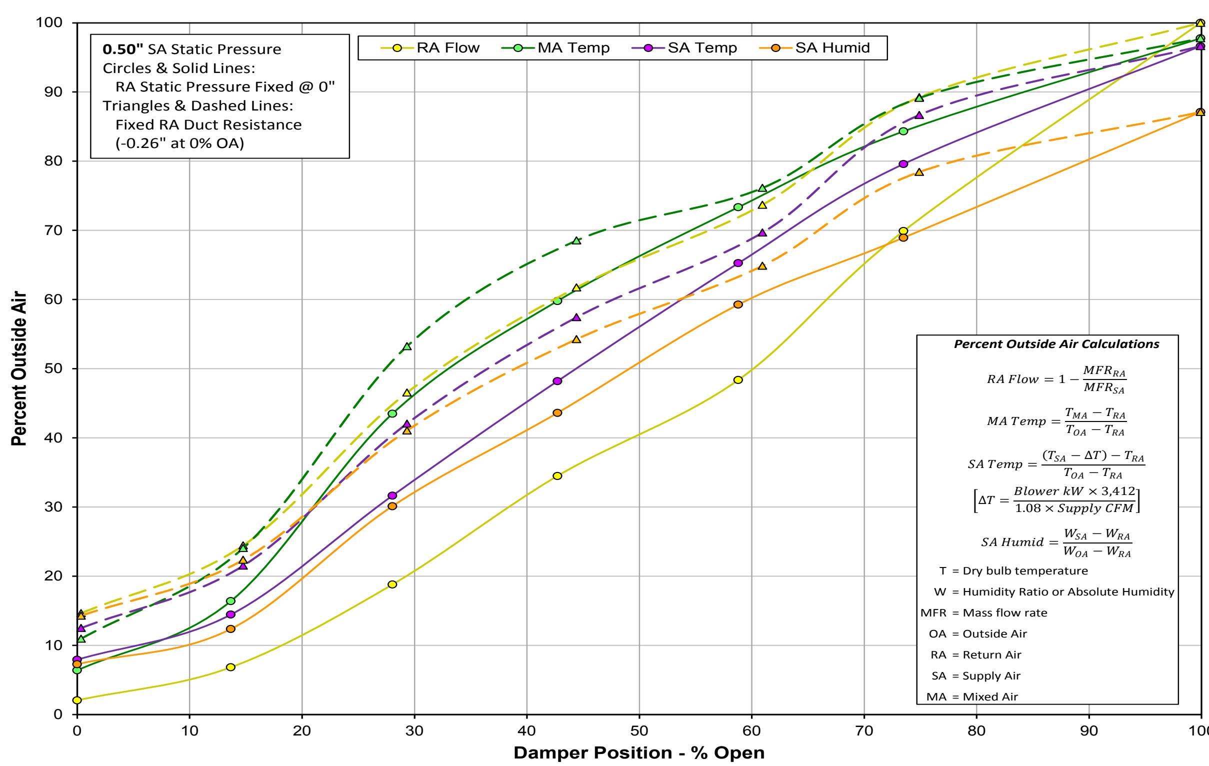

FIGURE A-3. LINKED PARALLEL BLADE DAMPER OUTSIDE AIR TEST WITH 0.75 IW SUPPLY STATIC PRESSURE

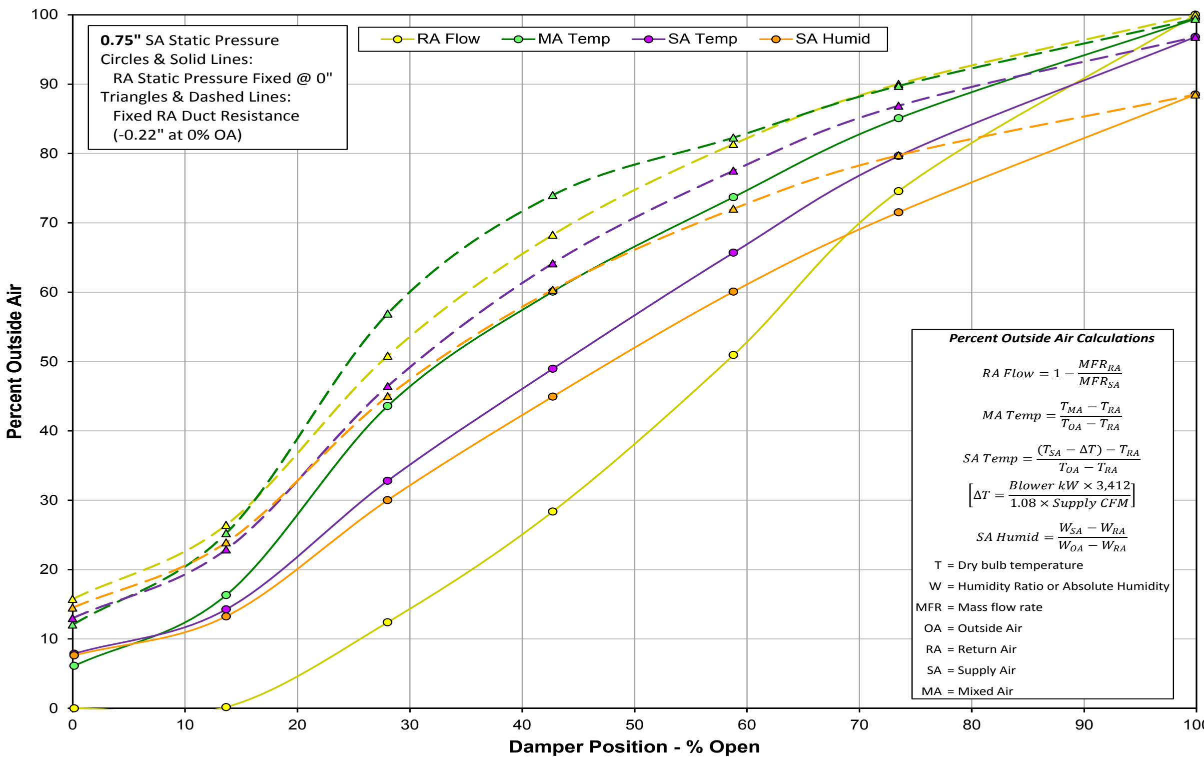

FIGURE A-4: GEARED OPPOSED BLADE DAMPER OUTSIDE AIR TEST WITH 0.25 IW SUPPLY STATIC PRESSURE

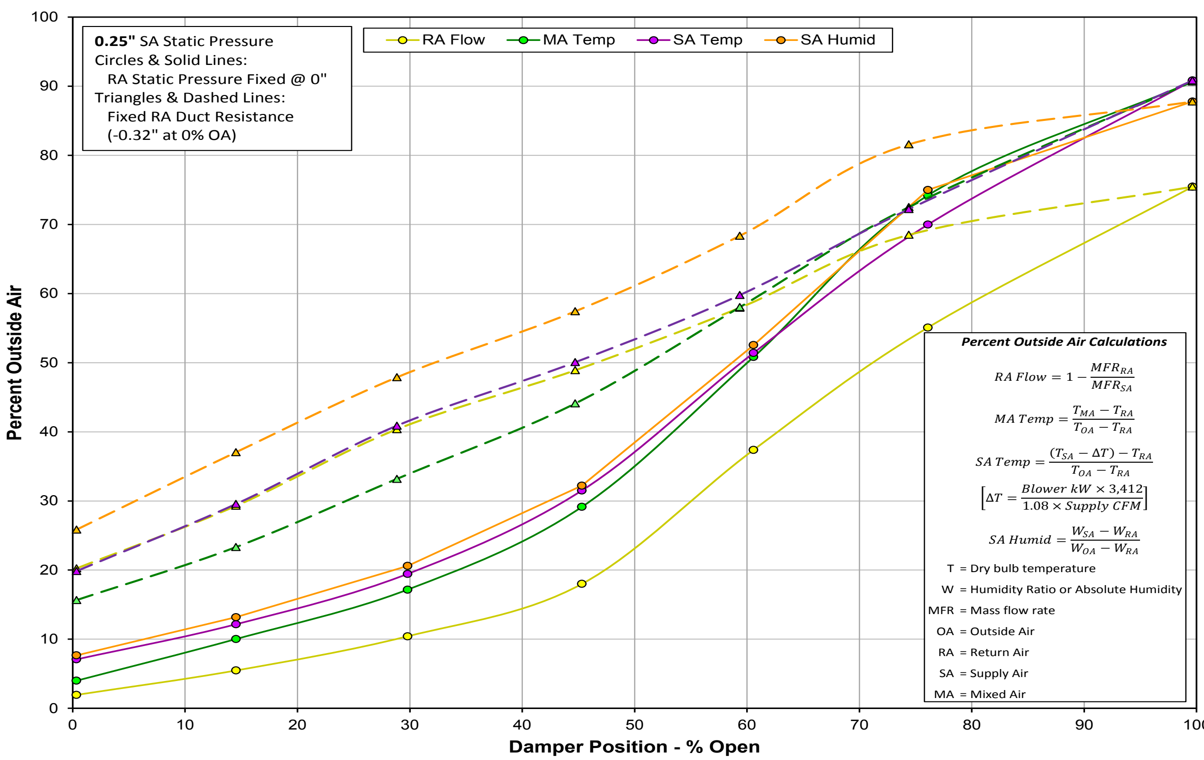

Figure A-5: Geared Opposed Blade Damper Outside Air Test with 0.50 IW Supply Static Pressure

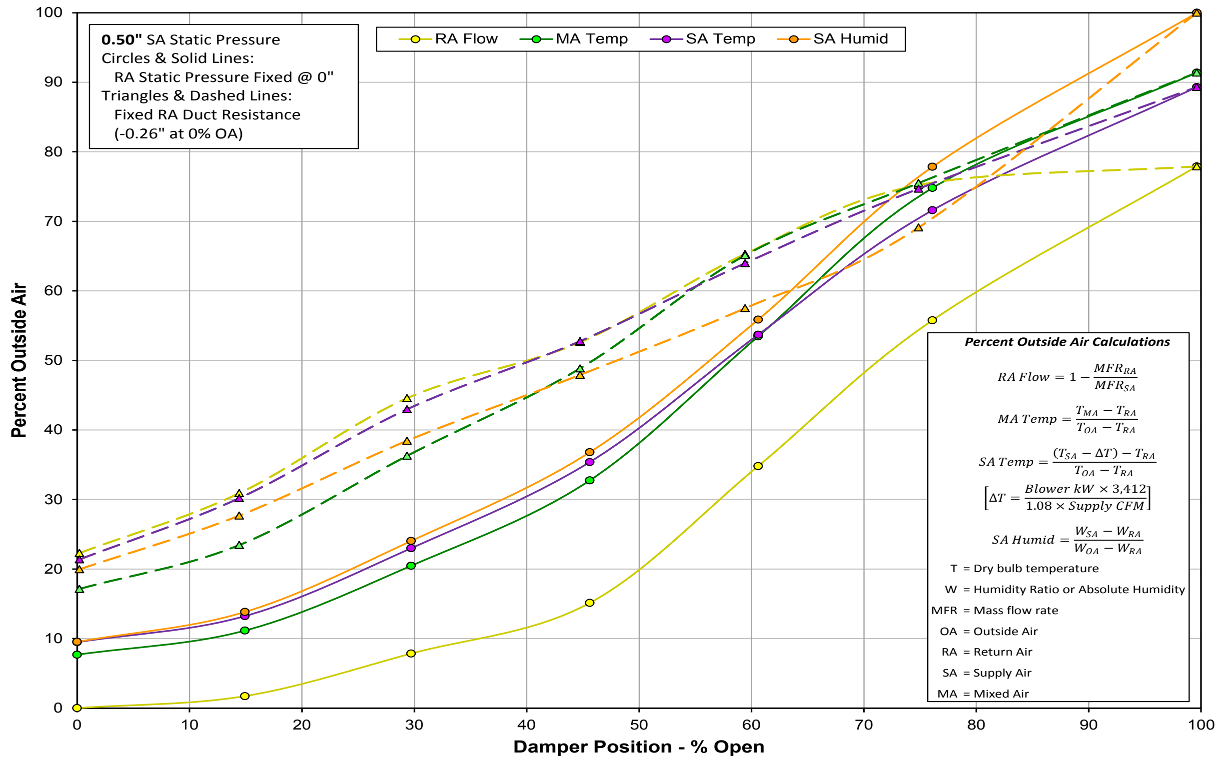

Figure A-6: Geared Opposed Blade Damper Outside Air Test with 0.75 IW Supply Static Pressure

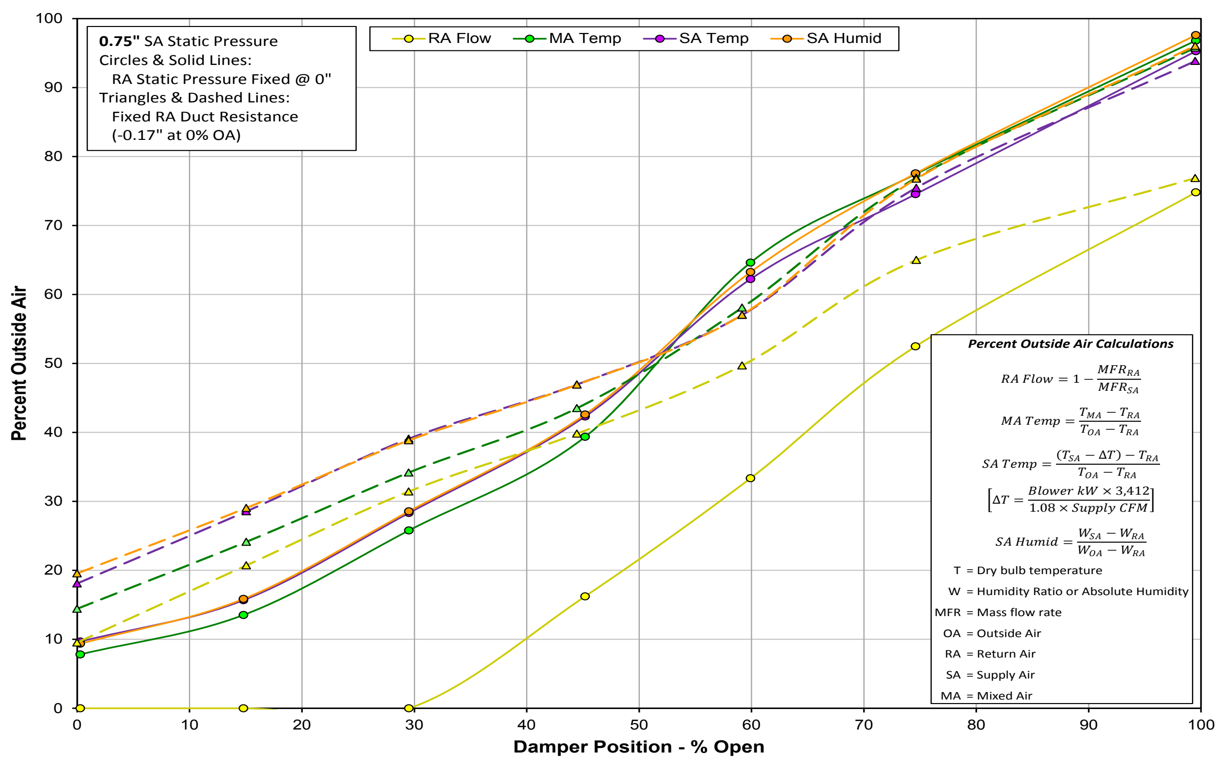

TABLE A-2: MIXED AIR PLENUM TEMPERATURE DISTRIBUTION DURING LINKED PARALLEL BLADE DAMPER OUTSIDE AIR TEST WITH 0.25 IW SA PRESSURE (RA PRESSURE - LEFT SIDE: FRICTION LOSS, RIGHT SIDE: FIXED ZERO)

Maximum - Minimum: 4.3°F

78.6	78.4	78.9
78.7	77.0	78.5
75.7	76.7	76.9
75.4	74.8	74.6

Thermocouple Average: **77.0°F**
Long RTD: **76.4°F**
Return Air: **75.2°F**
Damper Position: **0%**
Outside Air: **94.5°F**
Outside Air Fraction: **9%**

Supply Static Pressure: 0.25 " w.g.
Return Static Pressure: -0.30 " w.g.
Mixed Air Static Pressure: -0.52 " w.g.
Supply Air Flow Rate: 2,410 CFM

Maximum - Minimum: 2.7°F

77.2	77.0	77.5
77.3	76.3	77.1
75.4	76.1	76.2
75.3	74.9	74.8

Thermocouple Average: **76.3°F**
Long RTD: **75.8°F**
Return Air: **75.2°F**
Damper Position: **0%**
Outside Air: **95.5°F**
Outside Air Fraction: **5%**

Supply Static Pressure: 0.25 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.32 " w.g.
Supply Air Flow Rate: 2,760 CFM

Maximum - Minimum: 9.2°F

81.8	82.3	82.5
81.8	79.9	83.7
76.3	79.4	79.5
75.1	74.6	74.5

Thermocouple Average: **79.3°F**
Long RTD: **78.1°F**
Return Air: **74.9°F**
Damper Position: **15%**
Outside Air: **94.6°F**
Outside Air Fraction: **22%**

Supply Static Pressure: 0.26 " w.g.
Return Static Pressure: -0.24 " w.g.
Mixed Air Static Pressure: -0.47 " w.g.
Supply Air Flow Rate: 2,480 CFM

Maximum - Minimum: 6.8°F

80.1	80.3	79.8
80.3	78.4	81.1
76.2	78.4	78.4
74.9	74.5	74.3

Thermocouple Average: **78.1°F**
Long RTD: **77.4°F**
Return Air: **74.9°F**
Damper Position: **14%**
Outside Air: **95.6°F**
Outside Air Fraction: **15%**

Supply Static Pressure: 0.25 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.30 " w.g.
Supply Air Flow Rate: 2,790 CFM

Maximum - Minimum: 12.9°F

86.5	86.0	84.8
87.1	87.6	86.9
88.6	88.6	89.1
76.3	78.6	78.5

Thermocouple Average: **84.9°F**
Long RTD: **84.3°F**
Return Air: **75.0°F**
Damper Position: **29%**
Outside Air: **94.7°F**
Outside Air Fraction: **50%**

Supply Static Pressure: 0.29 " w.g.
Return Static Pressure: -0.12 " w.g.
Mixed Air Static Pressure: -0.34 " w.g.
Supply Air Flow Rate: 2,620 CFM

Maximum - Minimum: 13.3°F

85.2	85.4	83.6
87.0	85.0	86.7
87.4	88.1	89.1
75.8	75.7	76.3

Thermocouple Average: **83.8°F**
Long RTD: **82.6°F**
Return Air: **74.9°F**
Damper Position: **28%**
Outside Air: **95.6°F**
Outside Air Fraction: **43%**

Supply Static Pressure: 0.26 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.26 " w.g.
Supply Air Flow Rate: 2,830 CFM

Maximum - Minimum: 11.8°F

Thermocouple Average: **88.1°F**

Long RTD: **88.4°F**

87.2	85.9	86.5
90.3	91.3	88.0
92.4	91.3	90.6
80.5	84.1	89.5

Damper Position: **44%**

Outside Air: **94.8°F**

Outside Air Fraction: **66%**

Return Air: **75.3°F**

Supply Static Pressure: 0.31 " w.g.
Return Static Pressure: -0.05 " w.g.
Mixed Air Static Pressure: -0.25 " w.g.
Supply Air Flow Rate: 2,710 CFM

Maximum - Minimum: 12.3°F

Thermocouple Average: **87.2°F**

Long RTD: **87.4°F**

86.5	85.1	85.7
89.9	90.6	87.3
92.1	91.5	90.8
81.5	79.8	85.0

Damper Position: **43%**

Outside Air: **95.4°F**

Outside Air Fraction: **59%**

Return Air: **75.3°F**

Supply Static Pressure: 0.26 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.23 " w.g.
Supply Air Flow Rate: 2,830 CFM

Maximum - Minimum: 8.4°F

Thermocouple Average: **89.5°F**

Long RTD: **90.8°F**

86.1	86.1	87.4
88.1	91.9	87.8
93.1	92.3	92.7
85.4	89.0	93.8

Damper Position: **61%**

Outside Air: **95.1°F**

Outside Air Fraction: **72%**

Return Air: **74.8°F**

Supply Static Pressure: 0.33 " w.g.
Return Static Pressure: 0.01 " w.g.
Mixed Air Static Pressure: -0.18 " w.g.
Supply Air Flow Rate: 2,780 CFM

Maximum - Minimum: 7.1°F

Thermocouple Average: **89.7°F**

Long RTD: **90.9°F**

86.1	86.3	87.8
88.8	91.8	88.2
93.1	92.9	93.0
88.5	87.1	93.2

Damper Position: **59%**

Outside Air: **95.4°F**

Outside Air Fraction: **72%**

Return Air: **74.9°F**

Supply Static Pressure: 0.26 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.18 " w.g.
Supply Air Flow Rate: 2,860 CFM

Maximum - Minimum: 6.8°F

Thermocouple Average: **91.3°F**

Long RTD: **92.5°F**

87.7	88.1	88.9
88.8	92.5	89.8
93.4	94.1	94.5
91.2	91.8	94.2

Damper Position: **76%**

Outside Air: **94.6°F**

Outside Air Fraction: **83%**

Return Air: **75.1°F**

Supply Static Pressure: 0.33 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.14 " w.g.
Supply Air Flow Rate: 2,820 CFM

Maximum - Minimum: 7.1°F

Thermocouple Average: **91.7°F**

Long RTD: **92.9°F**

87.9	88.6	89.2
89.3	93.0	90.4
94.0	94.8	95.0
91.9	91.9	94.7

Damper Position: **73%**

Outside Air: **95.2°F**

Outside Air Fraction: **83%**

Return Air: **74.8°F**

Supply Static Pressure: 0.27 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.14 " w.g.
Supply Air Flow Rate: 2,910 CFM

Maximum - Minimum: 3.8°F

Thermocouple Average: **92.7°F**

Long RTD: **93.3°F**

90.8	91.7	92.1
90.9	94.4	93.0
94.6	94.2	94.6
92.1	91.4	93.3

Damper Position: **100%**

Outside Air: **94.5°F**

Outside Air Fraction: **91%**

Return Air: **74.8°F**

Supply Static Pressure: 0.35 " w.g.
Return Static Pressure: 0.03 " w.g.
Mixed Air Static Pressure: -0.09 " w.g.
Supply Air Flow Rate: 2,890 CFM

TABLE A-3: MIXED AIR PLENUM TEMPERATURE DISTRIBUTION DURING LINKED PARALLEL BLADE DAMPER OUTSIDE AIR TEST WITH 0.50 IW SA PRESSURE (RA PRESSURE - LEFT SIDE: FRICTION LOSS, RIGHT SIDE: FIXED ZERO)

Maximum - Minimum: 4.5°F

Thermocouple Average: **77.2°F**

Long RTD: **76.2°F**

79.0	78.6	79.1
78.9	77.0	78.8
76.0	77.0	77.0
75.3	74.7	74.6

Damper Position: **0%**

Outside Air: **94.7°F**

Outside Air Fraction: **11%**

Return Air: **75.0°F**

Supply Static Pressure: 0.50 " w.g.
Return Static Pressure: -0.26 " w.g.
Mixed Air Static Pressure: -0.42 " w.g.
Supply Air Flow Rate: 2,100 CFM

Maximum - Minimum: 2.7°F

Thermocouple Average: **76.3°F**

Long RTD: **75.6°F**

77.5	77.1	77.5
77.6	76.2	77.2
75.7	76.1	76.2
75.2	74.9	74.9

Damper Position: **0%**

Outside Air: **95.4°F**

Outside Air Fraction: **7%**

Return Air: **75.0°F**

Supply Static Pressure: 0.50 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.23 " w.g.
Supply Air Flow Rate: 2,430 CFM

Maximum - Minimum: 9.3°F

Thermocouple Average: **79.8°F**

Long RTD: **78.2°F**

82.4	82.8	83.2
82.3	80.2	84.2
76.9	79.8	79.9
75.4	74.9	74.9

Damper Position: **15%**

Outside Air: **94.7°F**

Outside Air Fraction: **24%**

Return Air: **75.0°F**

Supply Static Pressure: 0.52 " w.g.
Return Static Pressure: -0.21 " w.g.
Mixed Air Static Pressure: -0.38 " w.g.
Supply Air Flow Rate: 2,140 CFM

Maximum - Minimum: 6.0°F

Thermocouple Average: **78.3°F**

Long RTD: **77.3°F**

80.5	80.1	80.3
80.8	77.9	80.8
77.6	78.1	78.5
75.2	74.9	74.8

Damper Position: **14%**

Outside Air: **95.5°F**

Outside Air Fraction: **16%**

Return Air: **75.0°F**

Supply Static Pressure: 0.51 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.22 " w.g.
Supply Air Flow Rate: 2,460 CFM

Maximum - Minimum: 13.1°F

Thermocouple Average: **85.4°F**

Long RTD: **84.7°F**

87.1	86.3	85.3
87.3	87.8	87.4
89.0	88.9	89.7
76.6	79.7	79.4

Damper Position: **29%**

Outside Air: **94.8°F**

Outside Air Fraction: **53%**

Return Air: **75.0°F**

Supply Static Pressure: 0.56 " w.g.
Return Static Pressure: -0.11 " w.g.
Mixed Air Static Pressure: -0.27 " w.g.
Supply Air Flow Rate: 2,230 CFM

Maximum - Minimum: 13.4°F

Thermocouple Average: **83.7°F**

Long RTD: **82.4°F**

85.0	85.2	83.8
86.9	84.2	86.7
87.5	87.5	89.1
76.2	75.7	76.5

Damper Position: **28%**

Outside Air: **95.3°F**

Outside Air Fraction: **43%**

Return Air: **75.0°F**

Supply Static Pressure: 0.51 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.19 " w.g.
Supply Air Flow Rate: 2,470 CFM

Maximum - Minimum: 10.9°F

Thermocouple Average: **88.4°F**

Long RTD: **88.9°F**

87.6	86.2	87.1
90.5	91.4	88.0
92.4	91.3	90.9
81.5	85.7	88.6

Damper Position: **44%**

Outside Air: **94.8°F**

Outside Air Fraction: **68%**

Return Air: **75.0°F**

Supply Static Pressure: 0.58 " w.g.
Return Static Pressure: -0.06 " w.g.
Mixed Air Static Pressure: -0.20 " w.g.
Supply Air Flow Rate: 2,280 CFM

Maximum - Minimum: 11.6°F

Thermocouple Average: **87.0°F**

Long RTD: **87.0°F**

86.5	85.1	85.4
89.9	90.6	87.0
91.9	91.5	90.6
80.4	80.3	84.2

Damper Position: **43%**

Outside Air: **95.3°F**

Outside Air Fraction: **59%**

Return Air: **75.0°F**

Supply Static Pressure: 0.50 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.17 " w.g.
Supply Air Flow Rate: 2,460 CFM

Maximum - Minimum: 6.7°F

Thermocouple Average: **89.9°F**

Long RTD: **91.2°F**

86.7	86.7	87.9
88.3	91.9	88.4
92.8	92.3	92.7
87.4	90.4	93.4

Damper Position: **61%**

Outside Air: **94.8°F**

Outside Air Fraction: **75%**

Return Air: **75.0°F**

Supply Static Pressure: 0.60 " w.g.
Return Static Pressure: -0.02 " w.g.
Mixed Air Static Pressure: -0.14 " w.g.
Supply Air Flow Rate: 2,320 CFM

Maximum - Minimum: 6.6°F

Thermocouple Average: **89.7°F**

Long RTD: **90.6°F**

86.3	86.5	87.8
88.8	91.6	88.2
93.0	92.7	92.9
88.5	87.1	92.7

Damper Position: **59%**

Outside Air: **95.1°F**

Outside Air Fraction: **73%**

Return Air: **75.0°F**

Supply Static Pressure: 0.51 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.14 " w.g.
Supply Air Flow Rate: 2,480 CFM

Maximum - Minimum: 5.2°F

Thermocouple Average: **92.5°F**

Long RTD: **93.4°F**

89.7	89.9	90.4
90.4	93.4	91.2
94.1	94.7	94.8
93.0	93.8	94.8

Damper Position: **75%**

Outside Air: **94.8°F**

Outside Air Fraction: **88%**

Return Air: **75.0°F**

Supply Static Pressure: 0.61 " w.g.
Return Static Pressure: -0.03 " w.g.
Mixed Air Static Pressure: -0.10 " w.g.
Supply Air Flow Rate: 2,350 CFM

Maximum - Minimum: 6.9°F

Thermocouple Average: **91.8°F**

Long RTD: **92.8°F**

88.1	88.7	89.4
89.5	93.0	90.4
94.0	94.8	95.0
91.8	91.8	94.6

Damper Position: **73%**

Outside Air: **95.1°F**

Outside Air Fraction: **83%**

Return Air: **74.8°F**

Supply Static Pressure: 0.52 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.10 " w.g.
Supply Air Flow Rate: 2,510 CFM

Maximum - Minimum: 2.1°F

Thermocouple Average: **94.3°F**

Long RTD: **94.5°F**

93.0	93.3	93.4
93.0	94.8	93.9
95.1	95.1	95.1
94.7	94.9	95.0

Damper Position: **100%**

Outside Air: **94.9°F**

Outside Air Fraction: **97%**

Return Air: **75.0°F**

Supply Static Pressure: 0.62 " w.g.
Return Static Pressure: -0.02 " w.g.
Mixed Air Static Pressure: -0.06 " w.g.
Supply Air Flow Rate: 2,380 CFM

TABLE A-4: MIXED AIR PLENUM TEMPERATURE DISTRIBUTION DURING LINKED PARALLEL BLADE DAMPER OUTSIDE AIR TEST WITH 0.75 IW SA PRESSURE (RA PRESSURE - LEFT SIDE: FRICTION LOSS, RIGHT SIDE: FIXED ZERO)

Maximum - Minimum: 4.9°F

Thermocouple Average: **77.3°F**

Long RTD: **76.3°F**

79.3	78.9	79.4
79.2	77.1	79.1
76.1	77.0	77.1
75.4	74.7	74.6

Damper Position: **0%**

Outside Air: **94.7°F**

Outside Air Fraction: **12%**

Return Air: **75.0°F**

Supply Static Pressure: 0.75 " w.g.
Return Static Pressure: -0.22 " w.g.
Mixed Air Static Pressure: -0.32 " w.g.
Supply Air Flow Rate: 1,670 CFM

Maximum - Minimum: 3.0°F

Thermocouple Average: **76.0°F**

Long RTD: **75.6°F**

77.6	76.7	76.4
77.4	75.9	76.5
75.8	75.6	75.4
75.1	74.8	74.6

Damper Position: **0%**

Outside Air: **94.4°F**

Outside Air Fraction: **5%**

Return Air: **75.0°F**

Supply Static Pressure: 0.76 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.15 " w.g.
Supply Air Flow Rate: 2,020 CFM

Maximum - Minimum: 9.7°F

Thermocouple Average: **79.9°F**

Long RTD: **78.3°F**

82.5	83.1	83.7
82.4	80.4	84.6
77.2	79.8	79.8
75.7	74.9	75.0

Damper Position: **14%**

Outside Air: **94.7°F**

Outside Air Fraction: **25%**

Return Air: **75.0°F**

Supply Static Pressure: 0.77 " w.g.
Return Static Pressure: -0.19 " w.g.
Mixed Air Static Pressure: -0.28 " w.g.
Supply Air Flow Rate: 1,700 CFM

Maximum - Minimum: 5.9°F

Thermocouple Average: **78.0°F**

Long RTD: **77.1°F**

80.2	79.5	80.0
80.6	77.5	80.1
78.0	77.7	77.9
75.1	74.9	74.7

Damper Position: **14%**

Outside Air: **94.5°F**

Outside Air Fraction: **15%**

Return Air: **75.0°F**

Supply Static Pressure: 0.77 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.14 " w.g.
Supply Air Flow Rate: 2,020 CFM

Maximum - Minimum: 11.7°F

Thermocouple Average: **86.2°F**

Long RTD: **85.6°F**

87.9	87.1	86.2
88.0	88.4	87.9
89.3	89.2	89.6
77.9	81.4	81.1

Damper Position: **28%**

Outside Air: **94.8°F**

Outside Air Fraction: **56%**

Return Air: **75.0°F**

Supply Static Pressure: 0.80 " w.g.
Return Static Pressure: -0.11 " w.g.
Mixed Air Static Pressure: -0.20 " w.g.
Supply Air Flow Rate: 1,740 CFM

Maximum - Minimum: 12.9°F

Thermocouple Average: **83.5°F**

Long RTD: **82.4°F**

84.8	84.7	83.9
86.9	83.7	86.5
87.4	86.7	88.6
76.2	75.7	76.4

Damper Position: **28%**

Outside Air: **94.6°F**

Outside Air Fraction: **43%**

Return Air: **75.2°F**

Supply Static Pressure: 0.76 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.13 " w.g.
Supply Air Flow Rate: 2,020 CFM

Maximum - Minimum: 8.1°F

Thermocouple Average: **89.6°F**

Long RTD: **90.0°F**

88.8	87.7	88.2
91.2	91.7	88.9
93.0	91.7	91.2
84.9	87.9	89.8

Return Air: **75.0°F**

Damper Position: **43%**

Outside Air: **94.8°F**

Outside Air Fraction: **73%**

Supply Static Pressure: 0.82 " w.g.
Return Static Pressure: -0.07 " w.g.
Mixed Air Static Pressure: -0.14 " w.g.
Supply Air Flow Rate: 1,780 CFM

Maximum - Minimum: 11.8°F

Thermocouple Average: **86.6°F**

Long RTD: **86.7°F**

86.1	84.7	85.8
89.6	89.6	86.8
91.4	90.6	90.4
80.7	79.6	84.1

Return Air: **75.0°F**

Damper Position: **43%**

Outside Air: **94.7°F**

Outside Air Fraction: **59%**

Supply Static Pressure: 0.76 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.12 " w.g.
Supply Air Flow Rate: 2,020 CFM

Maximum - Minimum: 5.5°F

Thermocouple Average: **91.2°F**

Long RTD: **92.1°F**

88.6	88.4	89.3
89.8	92.6	89.7
93.2	93.2	92.9
90.4	91.9	93.9

Return Air: **75.0°F**

Damper Position: **59%**

Outside Air: **94.8°F**

Outside Air Fraction: **82%**

Supply Static Pressure: 0.84 " w.g.
Return Static Pressure: -0.05 " w.g.
Mixed Air Static Pressure: -0.10 " w.g.
Supply Air Flow Rate: 1,800 CFM

Maximum - Minimum: 6.6°F

Thermocouple Average: **89.3°F**

Long RTD: **90.3°F**

86.3	86.2	87.3
88.3	91.4	87.7
92.8	92.0	92.4
87.7	87.3	92.4

Return Air: **74.8°F**

Damper Position: **59%**

Outside Air: **94.7°F**

Outside Air Fraction: **73%**

Supply Static Pressure: 0.76 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.09 " w.g.
Supply Air Flow Rate: 2,030 CFM

Maximum - Minimum: 4.9°F

Thermocouple Average: **92.6°F**

Long RTD: **93.4°F**

89.9	90.1	90.8
90.8	93.4	91.4
94.0	94.6	94.7
93.0	94.0	94.8

Return Air: **75.0°F**

Damper Position: **73%**

Outside Air: **94.8°F**

Outside Air Fraction: **89%**

Supply Static Pressure: 0.86 " w.g.
Return Static Pressure: -0.04 " w.g.
Mixed Air Static Pressure: -0.07 " w.g.
Supply Air Flow Rate: 1,820 CFM

Maximum - Minimum: 5.9°F

Thermocouple Average: **91.9°F**

Long RTD: **92.8°F**

88.8	89.0	89.6
90.0	93.1	90.5
94.1	94.5	94.7
91.9	92.1	94.3

Return Air: **75.4°F**

Damper Position: **73%**

Outside Air: **94.8°F**

Outside Air Fraction: **85%**

Supply Static Pressure: 0.76 " w.g.
Return Static Pressure: -0.01 " w.g.
Mixed Air Static Pressure: -0.07 " w.g.
Supply Air Flow Rate: 2,050 CFM

Maximum - Minimum: 1.3°F

Thermocouple Average: **94.7°F**

Long RTD: **94.6°F**

94.2	94.2	94.2
94.0	94.8	94.5
95.3	95.2	95.2
94.8	95.0	95.1

Return Air: **76.3°F**

Damper Position: **100%**

Outside Air: **94.9°F**

Outside Air Fraction: **99%**

Supply Static Pressure: 0.87 " w.g.
Return Static Pressure: -0.02 " w.g.
Mixed Air Static Pressure: -0.04 " w.g.
Supply Air Flow Rate: 1,840 CFM

Table A-5: Mixed Air Plenum Temperature Distribution during Geared Opposed Blade Damper Outside Air Test with 0.25 IW SA Pressure (RA Pressure - Left Side: Friction Loss, Right Side: Fixed Zero)

Maximum - Minimum: 7.0°F

82.0	81.8	78.1
81.4	78.0	78.5
78.8	76.5	75.0
76.6	76.0	75.6

Thermocouple Average: **78.1°F**
Long RTD: **76.6°F**
Return Air: **75.0°F**
Damper Position: **0%**
Outside Air: **95.6°F**
Outside Air Fraction: **15%**

Supply Static Pressure: 0.25 " w.g.
Return Static Pressure: -0.32 " w.g.
Mixed Air Static Pressure: -0.38 " w.g.
Supply Air Flow Rate: 2,640 CFM

Maximum - Minimum: 3.4°F

78.4	76.8	76.0
77.4	75.5	75.7
76.0	75.4	75.0
75.5	75.3	75.2

Thermocouple Average: **76.0°F**
Long RTD: **75.4°F**
Return Air: **75.0°F**
Damper Position: **0%**
Outside Air: **99.9°F**
Outside Air Fraction: **4%**

Supply Static Pressure: 0.26 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.12 " w.g.
Supply Air Flow Rate: 3,020 CFM

Maximum - Minimum: 11.2°F

84.2	86.3	78.9
83.9	79.1	79.7
81.1	77.0	75.1
78.4	76.6	76.7

Thermocouple Average: **79.7°F**
Long RTD: **77.5°F**
Return Air: **75.1°F**
Damper Position: **15%**
Outside Air: **95.5°F**
Outside Air Fraction: **23%**

Supply Static Pressure: 0.27 " w.g.
Return Static Pressure: -0.25 " w.g.
Mixed Air Static Pressure: -0.31 " w.g.
Supply Air Flow Rate: 2,710 CFM

Maximum - Minimum: 6.9°F

81.9	80.5	77.9
78.8	77.5	78.3
76.4	75.8	75.0
76.4	75.8	76.0

Thermocouple Average: **77.5°F**
Long RTD: **76.3°F**
Return Air: **75.0°F**
Damper Position: **15%**
Outside Air: **100.0°F**
Outside Air Fraction: **10%**

Supply Static Pressure: 0.26 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.10 " w.g.
Supply Air Flow Rate: 3,040 CFM

Maximum - Minimum: 15.4°F

88.9	90.5	84.5
86.0	79.9	86.7
79.5	76.1	75.1
79.9	77.2	76.6

Thermocouple Average: **81.7°F**
Long RTD: **79.1°F**
Return Air: **75.1°F**
Damper Position: **29%**
Outside Air: **95.3°F**
Outside Air Fraction: **33%**

Supply Static Pressure: 0.28 " w.g.
Return Static Pressure: -0.17 " w.g.
Mixed Air Static Pressure: -0.26 " w.g.
Supply Air Flow Rate: 2,770 CFM

Maximum - Minimum: 11.8°F

86.8	86.2	79.9
79.7	78.6	82.8
76.4	76.2	75.0
76.8	76.3	77.0

Thermocouple Average: **79.3°F**
Long RTD: **77.3°F**
Return Air: **75.0°F**
Damper Position: **30%**
Outside Air: **100.1°F**
Outside Air Fraction: **17%**

Supply Static Pressure: 0.26 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.12 " w.g.
Supply Air Flow Rate: 2,980 CFM

Maximum - Minimum: 18.5°F

Thermocouple Average: **83.9°F**

Long RTD: **81.9°F**

93.7	92.7	88.6
90.3	83.4	89.7
79.6	79.0	75.2
79.4	77.9	77.4

Return Air: **75.2°F**

Damper Position: **45%**

Outside Air: **95.3°F**

Outside Air Fraction: **43%**

Supply Static Pressure: 0.29 " w.g.
Return Static Pressure: -0.11 " w.g.
Mixed Air Static Pressure: -0.25 " w.g.
Supply Air Flow Rate: 2,800 CFM

Maximum - Minimum: 16.7°F

Thermocouple Average: **82.4°F**

Long RTD: **79.4°F**

92.4	92.2	87.9
84.2	79.5	89.4
77.0	77.0	75.7
77.5	77.1	78.6

Return Air: **75.2°F**

Damper Position: **45%**

Outside Air: **99.8°F**

Outside Air Fraction: **29%**

Supply Static Pressure: 0.26 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.17 " w.g.
Supply Air Flow Rate: 2,920 CFM

Maximum - Minimum: 16.3°F

Thermocouple Average: **86.8°F**

Long RTD: **84.0°F**

94.8	94.5	93.7
93.3	87.9	92.2
84.7	85.1	78.7
79.2	78.5	78.6

Return Air: **75.5°F**

Damper Position: **59%**

Outside Air: **95.2°F**

Outside Air Fraction: **57%**

Supply Static Pressure: 0.30 " w.g.
Return Static Pressure: -0.04 " w.g.
Mixed Air Static Pressure: -0.23 " w.g.
Supply Air Flow Rate: 2,850 CFM

Maximum - Minimum: 21.4°F

Thermocouple Average: **88.0°F**

Long RTD: **84.8°F**

99.4	98.7	98.4
95.5	86.1	95.5
82.5	82.0	78.0
79.9	79.2	81.3

Return Air: **75.0°F**

Damper Position: **61%**

Outside Air: **100.6°F**

Outside Air Fraction: **51%**

Supply Static Pressure: 0.24 " w.g.
Return Static Pressure: -0.02 " w.g.
Mixed Air Static Pressure: -0.23 " w.g.
Supply Air Flow Rate: 2,880 CFM

Maximum - Minimum: 17.0°F

Thermocouple Average: **89.9°F**

Long RTD: **87.0°F**

94.9	95.0	94.9
95.1	94.2	94.8
92.8	92.5	88.5
79.4	78.2	78.1

Return Air: **76.6°F**

Damper Position: **74%**

Outside Air: **95.1°F**

Outside Air Fraction: **72%**

Supply Static Pressure: 0.30 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.20 " w.g.
Supply Air Flow Rate: 2,900 CFM

Maximum - Minimum: 21.1°F

Thermocouple Average: **93.4°F**

Long RTD: **89.4°F**

99.8	99.7	99.7
99.5	97.2	99.6
96.1	95.5	95.2
79.9	78.6	79.4

Return Air: **75.0°F**

Damper Position: **76%**

Outside Air: **99.8°F**

Outside Air Fraction: **74%**

Supply Static Pressure: 0.24 " w.g.
Return Static Pressure: -0.02 " w.g.
Mixed Air Static Pressure: -0.22 " w.g.
Supply Air Flow Rate: 2,920 CFM

Maximum - Minimum: 10.1°F

Thermocouple Average: **93.2°F**

Long RTD: **92.6°F**

95.7	95.4	95.0
96.6	96.6	95.3
94.9	93.2	90.6
90.0	86.5	88.0

Return Air: **77.1°F**

Damper Position: **100%**

Outside Air: **96.6°F**

Outside Air Fraction: **83%**

Supply Static Pressure: 0.31 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.15 " w.g.
Supply Air Flow Rate: 2,920 CFM

Table A-6: Mixed Air Plenum Temperature Distribution during Geared Opposed Blade Damper Outside Air Test with 0.50 IW SA Pressure (RA Pressure - Left Side: Friction Loss, Right Side: Fixed Zero)

Maximum - Minimum: 6.6°F

Thermocouple Average: **79.4°F**

Long RTD: **77.8°F**

82.7	82.5	79.5
82.9	79.3	79.9
80.4	78.0	76.2
78.2	77.2	76.8

Damper Position: **0%**

Outside Air: **95.4°F**

Outside Air Fraction: **17%**

Return Air: **76.2°F**

Supply Static Pressure: 0.50 " w.g.
Return Static Pressure: -0.26 " w.g.
Mixed Air Static Pressure: -0.30 " w.g.
Supply Air Flow Rate: 2,290 CFM

Maximum - Minimum: 5.2°F

Thermocouple Average: **76.9°F**

Long RTD: **75.9°F**

80.2	78.4	77.1
78.7	76.5	76.9
77.0	76.0	75.0
76.2	75.7	75.6

Damper Position: **5%**

Outside Air: **100.1°F**

Outside Air Fraction: **8%**

Return Air: **75.0°F**

Supply Static Pressure: 0.50 " w.g.
Return Static Pressure: -0.01 " w.g.
Mixed Air Static Pressure: -0.09 " w.g.
Supply Air Flow Rate: 2,660 CFM

Maximum - Minimum: 10.5°F

Thermocouple Average: **80.7°F**

Long RTD: **78.6°F**

84.6	86.8	80.1
84.7	80.3	80.5
82.0	78.1	76.3
79.6	77.8	77.7

Damper Position: **14%**

Outside Air: **95.5°F**

Outside Air Fraction: **23%**

Return Air: **76.3°F**

Supply Static Pressure: 0.52 " w.g.
Return Static Pressure: -0.20 " w.g.
Mixed Air Static Pressure: -0.25 " w.g.
Supply Air Flow Rate: 2,340 CFM

Maximum - Minimum: 7.9°F

Thermocouple Average: **77.7°F**

Long RTD: **76.3°F**

82.8	81.2	77.5
79.1	77.6	78.8
76.5	75.8	74.9
76.5	75.8	76.3

Damper Position: **15%**

Outside Air: **100.1°F**

Outside Air Fraction: **11%**

Return Air: **74.9°F**

Supply Static Pressure: 0.50 " w.g.
Return Static Pressure: -0.01 " w.g.
Mixed Air Static Pressure: -0.08 " w.g.
Supply Air Flow Rate: 2,650 CFM

Maximum - Minimum: 14.6°F

Thermocouple Average: **83.1°F**

Long RTD: **80.4°F**

89.8	91.0	86.6
88.0	81.3	87.6
81.3	77.9	76.4
81.4	78.8	78.3

Damper Position: **29%**

Outside Air: **95.4°F**

Outside Air Fraction: **36%**

Return Air: **76.4°F**

Supply Static Pressure: 0.53 " w.g.
Return Static Pressure: -0.14 " w.g.
Mixed Air Static Pressure: -0.21 " w.g.
Supply Air Flow Rate: 2,380 CFM

Maximum - Minimum: 12.6°F

Thermocouple Average: **80.2°F**

Long RTD: **77.8°F**

87.8	87.1	83.8
80.5	77.9	86.3
76.5	76.2	75.2
76.8	76.5	77.3

Damper Position: **30%**

Outside Air: **100.1°F**

Outside Air Fraction: **20%**

Return Air: **75.0°F**

Supply Static Pressure: 0.50 " w.g.
Return Static Pressure: -0.01 " w.g.
Mixed Air Static Pressure: -0.11 " w.g.
Supply Air Flow Rate: 2,590 CFM

Maximum - Minimum: 17.5°F

Thermocouple Average: **85.5°F**

Long RTD: **83.5°F**

94.3	93.3	90.5
92.0	85.2	91.2
81.4	81.3	76.8
81.4	79.7	78.8

Damper Position: **45%**

Outside Air: **95.2°F**

Outside Air Fraction: **48%**

Return Air: **76.4°F**

Supply Static Pressure: 0.54 " w.g.
Return Static Pressure: -0.10 " w.g.
Mixed Air Static Pressure: -0.21 " w.g.
Supply Air Flow Rate: 2,390 CFM

Maximum - Minimum: 18.2°F

Thermocouple Average: **83.2°F**

Long RTD: **80.1°F**

93.9	92.9	90.4
86.1	79.5	90.7
77.3	76.9	75.7
78.1	77.3	79.0

Damper Position: **46%**

Outside Air: **99.9°F**

Outside Air Fraction: **33%**

Return Air: **75.0°F**

Supply Static Pressure: 0.50 " w.g.
Return Static Pressure: -0.01 " w.g.
Mixed Air Static Pressure: -0.14 " w.g.
Supply Air Flow Rate: 2,530 CFM

Maximum - Minimum: 14.9°F

Thermocouple Average: **88.7°F**

Long RTD: **85.3°F**

95.0	94.9	94.6
94.1	89.7	93.5
89.1	88.1	83.9
80.7	80.1	80.1

Damper Position: **59%**

Outside Air: **95.0°F**

Outside Air Fraction: **65%**

Return Air: **76.8°F**

Supply Static Pressure: 0.55 " w.g.
Return Static Pressure: -0.07 " w.g.
Mixed Air Static Pressure: -0.19 " w.g.
Supply Air Flow Rate: 2,420 CFM

Maximum - Minimum: 20.4°F

Thermocouple Average: **88.3°F**

Long RTD: **85.2°F**

99.2	98.5	98.4
95.8	87.1	96.1
83.1	83.0	78.8
79.5	79.3	81.3

Damper Position: **61%**

Outside Air: **100.0°F**

Outside Air Fraction: **53%**

Return Air: **75.0°F**

Supply Static Pressure: 0.50 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.17 " w.g.
Supply Air Flow Rate: 2,490 CFM

Maximum - Minimum: 15.9°F

Thermocouple Average: **90.6°F**

Long RTD: **87.7°F**

94.9	95.1	95.1
95.1	94.4	95.1
93.9	93.2	90.7
80.5	79.4	79.2

Damper Position: **75%**

Outside Air: **95.1°F**

Outside Air Fraction: **75%**

Return Air: **77.0°F**

Supply Static Pressure: 0.57 " w.g.
Return Static Pressure: -0.02 " w.g.
Mixed Air Static Pressure: -0.15 " w.g.
Supply Air Flow Rate: 2,460 CFM

Maximum - Minimum: 21.0°F

Thermocouple Average: **93.8°F**

Long RTD: **90.0°F**

100.1	100.0	100.1
99.9	97.8	99.9
96.8	96.2	95.6
80.5	79.1	79.8

Damper Position: **76%**

Outside Air: **100.1°F**

Outside Air Fraction: **75%**

Return Air: **75.3°F**

Supply Static Pressure: 0.49 " w.g.
Return Static Pressure: -0.02 " w.g.
Mixed Air Static Pressure: -0.18 " w.g.
Supply Air Flow Rate: 2,530 CFM

Maximum - Minimum: 5.9°F

Thermocouple Average: **93.0°F** **90% OA**

Long RTD: **92.4°F** **86% OA**

94.2	94.1	93.9
94.8	94.8	94.1
94.8	94.2	92.3
89.9	89.3	88.9

Damper Position: **100%**

Outside Air: **94.8°F**

Return Air: **77.8°F**

Supply Static Pressure: 0.57 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.11 " w.g.
Supply Air Flow Rate: 2,480 CFM

TABLE A-7: MIXED AIR PLENUM TEMPERATURE DISTRIBUTION DURING GEARED OPPOSED BLADE DAMPER OUTSIDE AIR TEST WITH 0.75 IW SA PRESSURE (RA PRESSURE - LEFT SIDE: FRICTION LOSS, RIGHT SIDE: FIXED ZERO)

Maximum - Minimum: 10.3°F

Thermocouple Average: **74.4°F**

Long RTD: **72.2°F**

80.3	78.1	74.7
79.3	74.0	74.5
75.0	72.4	70.0
72.6	71.4	70.3

Damper Position: **0%**

Outside Air: **100.1°F**

Return Air: **70.0°F**

Outside Air Fraction: **14%**

Supply Static Pressure: 0.75 " w.g.
Return Static Pressure: -0.17 " w.g.
Mixed Air Static Pressure: -0.21 " w.g.
Supply Air Flow Rate: 1,910 CFM

Maximum - Minimum: 5.6°F

Thermocouple Average: **72.9°F**

Long RTD: **71.6°F**

76.4	74.5	72.8
74.7	72.5	73.3
72.7	72.0	70.8
71.6	71.5	71.8

Damper Position: **0%**

Outside Air: **100.0°F**

Return Air: **70.6°F**

Outside Air Fraction: **8%**

Supply Static Pressure: 0.75 " w.g.
Return Static Pressure: -0.01 " w.g.
Mixed Air Static Pressure: -0.05 " w.g.
Supply Air Flow Rate: 2,240 CFM

Maximum - Minimum: 16.3°F

Thermocouple Average: **77.3°F**

Long RTD: **73.8°F**

84.6	86.4	76.5
83.3	75.8	77.2
79.4	73.6	70.1
75.2	72.8	72.9

Damper Position: **15%**

Outside Air: **100.0°F**

Return Air: **70.1°F**

Outside Air Fraction: **24%**

Supply Static Pressure: 0.76 " w.g.
Return Static Pressure: -0.12 " w.g.
Mixed Air Static Pressure: -0.16 " w.g.
Supply Air Flow Rate: 1,940 CFM

Maximum - Minimum: 9.7°F

Thermocouple Average: **75.1°F**

Long RTD: **73.1°F**

81.1	79.5	76.5
75.7	74.0	78.5
73.0	72.3	71.4
72.7	72.6	73.3

Damper Position: **15%**

Outside Air: **100.0°F**

Return Air: **71.2°F**

Outside Air Fraction: **14%**

Supply Static Pressure: 0.75 " w.g.
Return Static Pressure: -0.01 " w.g.
Mixed Air Static Pressure: -0.05 " w.g.
Supply Air Flow Rate: 2,200 CFM

Maximum - Minimum: 22.4°F

Thermocouple Average: **80.5°F**

Long RTD: **76.5°F**

91.3	92.7	85.6
85.8	76.9	88.7
77.7	72.7	70.4
77.3	73.9	73.5

Damper Position: **30%**

Outside Air: **100.1°F**

Return Air: **70.4°F**

Outside Air Fraction: **34%**

Supply Static Pressure: 0.77 " w.g.
Return Static Pressure: -0.08 " w.g.
Mixed Air Static Pressure: -0.13 " w.g.
Supply Air Flow Rate: 1,960 CFM

Maximum - Minimum: 16.9°F

Thermocouple Average: **79.1°F**

Long RTD: **76.0°F**

89.2	87.3	86.3
80.5	75.6	87.7
74.1	73.5	72.3
74.4	73.9	74.8

Damper Position: **30%**

Outside Air: **100.0°F**

Return Air: **71.9°F**

Outside Air Fraction: **26%**

Supply Static Pressure: 0.75 " w.g.
Return Static Pressure: -0.01 " w.g.
Mixed Air Static Pressure: -0.08 " w.g.
Supply Air Flow Rate: 2,130 CFM

Maximum - Minimum: 27.2°F

Thermocouple Average: **83.1°F**

Long RTD: **80.0°F**

97.5	95.8	90.7
91.9	81.7	92.1
76.7	75.9	70.3
75.9	74.5	73.8

Damper Position: **44%**

Outside Air: **100.0°F**

Return Air: **70.0°F**

Outside Air Fraction: **44%**

Supply Static Pressure: 0.78 " w.g.
Return Static Pressure: -0.04 " w.g.
Mixed Air Static Pressure: -0.12 " w.g.
Supply Air Flow Rate: 1,970 CFM

Maximum - Minimum: 22.2°F

Thermocouple Average: **83.5°F**

Long RTD: **80.2°F**

96.1	94.7	92.1
88.8	80.0	91.7
77.0	76.1	73.9
77.3	76.3	78.2

Damper Position: **45%**

Outside Air: **100.0°F**

Return Air: **72.8°F**

Outside Air Fraction: **39%**

Supply Static Pressure: 0.75 " w.g.
Return Static Pressure: -0.01 " w.g.
Mixed Air Static Pressure: -0.11 " w.g.
Supply Air Flow Rate: 2,060 CFM

Maximum - Minimum: 24.4°F

Thermocouple Average: **87.5°F**

Long RTD: **83.4°F**

99.7	99.1	98.5
96.9	88.3	96.2
84.1	84.5	75.6
75.8	75.3	76.3

Damper Position: **59%**

Outside Air: **100.2°F**

Return Air: **69.9°F**

Outside Air Fraction: **58%**

Supply Static Pressure: 0.78 " w.g.
Return Static Pressure: -0.01 " w.g.
Mixed Air Static Pressure: -0.11 " w.g.
Supply Air Flow Rate: 1,980 CFM

Maximum - Minimum: 20.9°F

Thermocouple Average: **90.9°F**

Long RTD: **87.2°F**

100.0	99.5	99.6
98.6	92.0	98.1
89.1	89.1	86.4
79.1	79.2	79.6

Damper Position: **60%**

Outside Air: **100.0°F**

Return Air: **74.2°F**

Outside Air Fraction: **65%**

Supply Static Pressure: 0.75 " w.g.
Return Static Pressure: -0.01 " w.g.
Mixed Air Static Pressure: -0.12 " w.g.
Supply Air Flow Rate: 2,030 CFM

Maximum - Minimum: 24.6°F

Thermocouple Average: **93.3°F**

Long RTD: **89.6°F**

99.9	99.9	99.9
99.9	98.9	99.8
98.2	97.7	95.0
78.0	75.4	76.0

Damper Position: **75%**

Outside Air: **99.9°F**

Return Air: **71.0°F**

Outside Air Fraction: **77%**

Supply Static Pressure: 0.78 " w.g.
Return Static Pressure: -0.01 " w.g.
Mixed Air Static Pressure: -0.11 " w.g.
Supply Air Flow Rate: 1,990 CFM

Maximum - Minimum: 21.7°F

Thermocouple Average: **94.2°F**

Long RTD: **90.7°F**

100.0	100.0	100.0
100.0	98.8	99.8
98.6	97.7	96.7
80.4	78.3	79.6

Damper Position: **75%**

Outside Air: **100.0°F**

Return Air: **74.4°F**

Outside Air Fraction: **77%**

Supply Static Pressure: 0.75 " w.g.
Return Static Pressure: -0.01 " w.g.
Mixed Air Static Pressure: -0.11 " w.g.
Supply Air Flow Rate: 2,050 CFM

Maximum - Minimum: 4.4°F

Thermocouple Average: **99.1°F**

Long RTD: **99.1°F**

99.9	99.6	99.3
100.1	100.1	99.3
100.1	100.1	99.5
96.7	97.6	95.6

Damper Position: **99%**

Outside Air: **100.1°F**

Return Air: **77.4°F**

Outside Air Fraction: **96%**

Supply Static Pressure: 0.78 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.08 " w.g.
Supply Air Flow Rate: 1,990 CFM

Maximum - Minimum: 3.4°F

Thermocouple Average: **99.2°F**

Long RTD: **99.3°F**

99.9	99.8	99.6
99.9	99.9	99.4
99.9	99.9	99.2
97.5	97.8	96.5

Damper Position: **100%**

Outside Air: **99.9°F**

Return Air: **78.7°F**

Outside Air Fraction: **97%**

Supply Static Pressure: 0.75 " w.g.
Return Static Pressure: 0.00 " w.g.
Mixed Air Static Pressure: -0.08 " w.g.
Supply Air Flow Rate: 2,050 CFM

TABLE A-8: WCC TEST RESULTS TABLE – SET POINTS & ACTUAL VALUES

			OATdb (°F)		OATwb (°F)		RATdb (°F)		RATwb (°F)		IEC Fan	RTU	RTU	OA	SA Rate		RA Rate		OA Rate		
Test #	Description	Climate	Target	Actual	Target	Actual	Target	Actual	Target	Actual	Set Point	Blower	Comps	Damper	Target	Actual	Target	Actual	Target	Actual	Net
1a	IEC Only	Warm Dry	78	78	58.5	60.1	78	78	64	64	5	Off	0	100%	1020	716	0	0	1020	1078	-362
1b	IEC Only	Western Peak	105	105	73	73	78	78	64	64	5	Off	0	100%	1020	695	0	0	1020	1110	-416
1c	IEC Only	AHRI 100% Capacity	95	95	75	76	78	78	64	64	5	Off	0	100%	1020	728	0	0	1020	1110	-382
1d	IEC Only	IEER 75% Capacity	81.5	81.5	66.3	66.7	78	78	64	64	5	Off	0	100%	1020	686	0	0	1020	1091	-405
1e	IEC Only	Warm Humid	82	82	73	73	78	78	64	64	5	Off	0	100%	1020	711	0	0	1020	1085	-374
1f	IEC Only	IEER 50% Capacity	68	68	57.5	57.8	78	78	64	64	5	Off	0	100%	1020	663	0	0	1020	1045	-381
1g	IEC Only	IEER 25% Capacity	65	65	52.8	53.4	78	78	64	64	5	Off	0	100%	1020	677	0	0	1020	1091	-414
1h	IEC Only	Western Annual	90	90	64	67	78	78	64	64	5	Off	0	100%	1020	733	0	0	1020	1112	-379
2a	IEC Only	Warm Dry	78	78	58.5	61.2	78	78	64	64	8	Off	0	100%	1800	1110	0	0	1800	1701	-591
2b	IEC Only	Western Peak	105	105	73	74	78	78	64	64	8	Off	0	100%	1800	1131	0	0	1800	1734	-603
2c	IEC Only	AHRI 100% Capacity	95	95	75	76	78	78	64	64	8	Off	0	100%	1800	1149	0	0	1800	1739	-589
2d	IEC Only	IEER 75% Capacity	81.5	80.8	66.3	66.9	78	78	64	64	8	Off	0	100%	1800	1122	0	0	1800	1725	-603
2e	IEC Only	Warm Humid	82	82	73	73	78	78	64	64	8	Off	0	100%	1800	1133	0	0	1800	1731	-599
2f	IEC Only	IEER 50% Capacity	68	68	57.5	57.9	78	78	64	64	8	Off	0	100%	1800	1088	0	0	1800	1686	-597
2g	IEC Only	IEER 25% Capacity	65	65	52.8	54.4	78	78	64	64	8	Off	0	100%	1800	1082	0	0	1800	1711	-630
2h	IEC Only	Western Annual	90	90	64	68	78	78	64	64	8	Off	0	100%	1800	1143	0	0	1800	1737	-594
3a	IEC Only	Warm Dry	78	78	58.5	61.2	78	78	64	64	10	Off	0	100%	2500	1374	0	0	2500	2110	-735
3b	IEC Only	Western Peak	105	105	73	75	78	78	64	64	10	Off	0	100%	2500	1388	0	0	2500	2113	-726
3c	IEC Only	AHRI 100% Capacity	95	95	75	76	78	78	64	64	10	Off	0	100%	2500	1403	0	0	2500	2125	-723
3d	IEC Only	IEER 75% Capacity	81.5	81.5	66.3	68.1	78	78	64	64	10	Off	0	100%	2500	1392	0	0	2500	2122	-730
3e	IEC Only	Warm Humid	82	82	73	73	78	78	64	64	10	Off	0	100%	2500	1399	0	0	2500	2110	-711
3f	IEC Only	IEER 50% Capacity	68	68	57.5	57.8	78	78	64	64	10	Off	0	100%	2500	1369	0	0	2500	2100	-731
3g	IEC Only	IEER 25% Capacity	65	65	52.8	54.6	78	78	64	64	10	Off	0	100%	2500	1348	0	0	2500	2100	-752
3h	IEC Only	Western Annual	90	90	64	67	78	78	64	64	10	Off	0	100%	2500	1386	0	0	2500	2112	-726
4a	IEC+RTU	Warm Dry	78	78	58.5	62	78	78	64	64	5	On	2	20%	3000	2619	1980	1606	1020	1143	-130
4b	IEC+RTU	Western Peak	105	105	73	73	78	78	64	64	5	On	2	20%	3000	2631	1980	1577	1020	1132	-78
4c	IEC+RTU	AHRI 100% Capacity	95	96	75	75	78	79	64	64	5	On	2	20%	3000	2682	1980	1755	1020	1040	-112
4d	IEC+RTU	IEER 75% Capacity	81.5	81.5	66.3	66.4	78	78	64	64	5	On	2	20%	3000	2646	1980	1610	1020	1131	-95
4e	IEC+RTU	Warm Humid	82	82	73	73	78	78	64	64	5	On	2	20%	3000	2639	1980	1663	1020	1103	-127
4f	IEC+RTU	IEER 50% Capacity	68	68	57.5	57.9	78	78	64	64	5	On	2	20%	3000	2661	1980	1684	1020	1130	-153
4g	IEC+RTU	IEER 25% Capacity	65	65	52.8	56.5	78	78	64	64	5	On	2	20%	3000	2640	1980	1675	1020	1133	-168
4h	IEC+RTU	Western Annual	90	90	64	66	78	78	64	64	5	On	2	20%	3000	2639	1980	1593	1020	1155	-109
5a	IEC+RTU	Warm Dry	78	78	58.5	62	78	78	64	64	10	On	2	75%	3000	2772	500	748	2500	2462	-437
5b	IEC+RTU	Western Peak	105	105	73	74	78	78	64	65	10	On	2	75%	3000	2778	500	640	2500	2479	-341
5c	IEC+RTU	AHRI 100% Capacity	95	98	75	76	78	79	64	64	10	On	2	75%	3000	2796	500	992	2500	2466	-662
5d	IEC+RTU	IEER 75% Capacity	81.5	81.5	66.3	66.5	78	78	64	64	10	On	2	75%	3000	2788	500	733	2500	2452	-397
5e	IEC+RTU	Warm Humid	82	82	73	73	78	78	64	64	10	On	2	75%	3000	2770	500	961	2500	2433	-624
5f	IEC+RTU	IEER 50% Capacity	68	68	57.5	58.4	78	78	64	64	10	On	2	75%	3000	2774	500	802	2500	2455	-482
5g	IEC+RTU	IEER 25% Capacity	65	65	52.8	57.0	78	78	64	64	10	On	2	75%	3000	2800	500	934	2500	2484	-618
5h	IEC+RTU	Western Annual	90	90	64	65	78	78	64	64	10	On	2	75%	3000	2780	500	680	2500	2482	-381

Test #	Description	Climate	OATdb (°F)		OATwb (°F)		RATdb (°F)		RATwb (°F)		IEC Fan	RTU	RTU	OA	SA Rate		RA Rate		OA Rate		
			Target	Actual	Target	Actual	Target	Actual	Target	Actual	Set Point	Blower	Comps	Damper	Target	Actual	Target	Actual	Target	Actual	Net
6a	IEC+RTU+PC	Warm Dry	78	78	58.5	63	78	78	64	64	5	On	2	20%	3000	2629	1980	1608	1020	1149	-128
6b	IEC+RTU+PC	Western Peak	105	105	73	75	78	78	64	64	5	On	2	20%	3000	2639	1980	1611	1020	1134	-106
6c	IEC+RTU+PC	AHRI 100% Capacity	95	95	75	75	78	78	64	64	5	On	2	20%	3000	2697	1980	1733	1020	1082	-118
6d	IEC+RTU+PC	IEER 75% Capacity	81.5	81.5	66.3	66.3	78	78	64	64	5	On	2	20%	3000	2623	1980	1599	1020	1134	-110
6e	IEC+RTU+PC	Warm Humid	82	82	73	73	78	78	64	64	5	On	2	20%	3000	2639	1980	1635	1020	1125	-121
6f	IEC+RTU+PC	IEER 50% Capacity	68	68	57.5	58.2	78	78	64	64	5	On	2	20%	3000	2635	1980	1630	1020	1121	-116
6g	IEC+RTU+PC	IEER 25% Capacity	65	65	52.8	56.5	78	78	64	64	5	On	2	20%	3000	2622	1980	1631	1020	1159	-167
6h	IEC+RTU+PC	Western Annual	90	90	64	66	78	78	64	64	5	On	2	20%	3000	2631	1980	1608	1020	1135	-112
7a	IEC+RTU+PC	Warm Dry	78	78	58.5	63.7	78	78	64	64	10	On	2	75%	3000	2779	500	737	2500	2474	-433
7b	IEC+RTU+PC	Western Peak	105	105	73	76	78	78	64	65	10	On	2	75%	3000	2779	500	733	2500	2484	-438
7c	IEC+RTU+PC	AHRI 100% Capacity	95	95	75	75	78	78	64	64	10	On	2	75%	3000	2802	500	886	2500	2478	-562
7d	IEC+RTU+PC	IEER 75% Capacity	81.5	81.4	66.3	66.3	78	78	64	64	10	On	2	75%	3000	2762	500	629	2500	2457	-323
7e	IEC+RTU+PC	Warm Humid	82	82	73	73	78	78	64	64	10	On	2	75%	3000	2782	500	905	2500	2454	-577
7f	IEC+RTU+PC	IEER 50% Capacity	68	68	57.5	58.6	78	78	64	64	10	On	2	75%	3000	2778	500	740	2500	2459	-421
7g	IEC+RTU+PC	IEER 25% Capacity	65	65	52.8	57.1	78	78	64	64	10	On	2	75%	3000	2800	500	827	2500	2483	-510
7h	IEC+RTU+PC	Western Annual	90	90	64	68	78	78	64	64	10	On	2	75%	3000	2782	500	684	2500	2492	-394
8a	IEC+1C+PC	Warm Dry	78	78	58.5	64.3	78	78	64	64	10	On	1	100%	2500	1318	0	0	2500	2115	-797
8b	IEC+1C+PC	Western Peak	105	105	73	76	78	78	64	65	10	On	1	100%	2500	1345	0	0	2500	2112	-767
8c	IEC+1C+PC	AHRI 100% Capacity	95	95	75	75	78	78	64	64	10	On	1	100%	2500	1357	0	0	2500	2121	-763
8d	IEC+1C+PC	IEER 75% Capacity	81.5	81.5	66.3	66.2	78	78	64	64	10	On	1	100%	2500	1334	0	0	2500	2095	-761
8e	IEC+1C+PC	Warm Humid	82	82	73	73	78	78	64	64	10	On	1	100%	2500	1354	0	0	2500	2116	-762
8g	IEC+1C+PC	IEER 25% Capacity	65	65	52.8	58.1	78	78	64	64	10	On	1	100%	2500	1285	0	0	2500	2088	-803
8h	IEC+1C+PC	Western Annual	90	90	64	67	78	78	64	64	10	On	1	100%	2500	1343	0	0	2500	2113	-771
9a	RTU+PC	Warm Dry	78	78	58.5	62.2	78	78	64	64	Off	On	2	20%	3000	2592	1980	1681	1020	911	
9f	RTU+PC	IEER 50% Capacity	68	68	57.5	57.5	78	78	64	64	Off	On	2	20%	3000	2570	1980	1714	1020	856	
9g	RTU+PC	IEER 25% Capacity	65	65	52.8	56.2	78	78	64	64	Off	On	2	20%	3000	2587	1980	1693	1020	893	
9h	RTU+PC	Western Annual	90	90	64	68	78	78	64	64	Off	On	2	20%	3000	2594	1980	1643	1020	951	
10a	RTU+PC	Western Peak	105	105	73	73	80	80	67	67	Off	On	2	0%	3000	2869	3000	2869	0	0	
10b	RTU+PC	AHRI 100% Capacity	95	95	75	75	80	80	67	67	Off	On	2	0%	3000	2866	3000	2883	0	-16	
10c	RTU+PC	Warm Humid	82	82	73	73	80	80	67	67	Off	On	2	0%	3000	2821	3000	2858	0	-37	
10d	RTU+PC	Western Annual	90	90	64	64	80	80	67	67	Off	On	2	0%	3000	2856	3000	2864	0	-8	
11a	RTU Baseline	Western Peak	105	105	73	73	78	78	64	64	Off	On	2	20%	3000	2668	1980	1864	1020	804	
11b	RTU Baseline	AHRI 100% Capacity	95	95	75	69	78	78	64	64	Off	On	2	20%	3000	2647	1980	1854	1020	793	
11c	RTU Baseline	Western Annual	90	90	64	68	78	78	64	64	Off	On	2	20%	3000	2642	1980	1847	1020	796	
11d	RTU Baseline	Warm Humid	82	82	73	65	78	78	64	64	Off	On	2	20%	3000	2639	1980	1860	1020	779	
11e	RTU Baseline		75	75		63	78	78	64	64	Off	On	2	20%	3000	2873	1980	1966	1020	907	
11f	RTU Baseline		73	73		64	78	78	64	64	Off	On	2	20%	3000	2872	1980	1971	1020	901	
11g	RTU Baseline		64	64		56	78	78	64	64	Off	On	2	20%	3000	2880	1980	1995	1020	886	
12a	RTU Baseline	Western Peak	105	105		69	80	80	67	67	Off	On	2	0%	3000	2850	3000	2863	0	-13	
12b	RTU Baseline	AHRI 100% Capacity	95	95		66	80	80	67	67	Off	On	2	0%	3000	2849	3000	2874	0	-25	
12c	RTU Baseline	Western Annual	90	90		64	80	80	67	67	Off	On	2	0%	3000	2852	3000	2876	0	-24	
12d	RTU Baseline	Warm Humid	82	82		61	80	80	67	67	Off	On	2	0%	3000	2851	3000	2885	0	-34	
	RTU Baseline	IEER 75% Capacity	81.5	81.5		61	80	80	67	67	Off	On	2	0%	3000	2852	3000	2876	0	-24	
	RTU Baseline	Warm Dry	78	78		60	80	80	67	67	Off	On	2	0%	3000	2852	3000	2876	0	-24	
12e	RTU Baseline		75	75		59	80	80	67	67	Off	On	2	0%	3000	2852	3000	2876	0	-24	
12f	RTU Baseline		73	73		58	80	80	67	67	Off	On	2	0%	3000	2852	3000	2876	0	-24	
	RTU Baseline	IEER 50% Capacity	68	68		56	80	80	67	67	Off	On	2	0%	3000	2852	3000	2876	0	-24	
	RTU Baseline	IEER 25% Capacity	65	82		61	80	80	67	67	Off	On	2	0%	3000	2851	3000	2885	0	-34	
12g	RTU Baseline		64	64		55	80	80	67	67	Off	On	2	0%	3000	2852	3000	2876	0	-24	

Table A-9: WCC Test Results Table – Performance Metrics & Comparison to 0% OA Baseline

			Combined System Metrics						IEC Metrics						Pre-Cooler		Baseline Comparison - 0% OA							
			Total	Sens	Power	EER -T	EER-S	MUW	EE	OA Cap	RA C-t	RA C-s	Power	Evap	Power	Evap	Total	Sensible	Power	EER -T	EER-S	% Savings		
Test #	Description	Climate	(Tons)	(Tons)	(kW)	(Btu/Wh)	(Btu/Wh)	(GPH)	(%)	(Tons)	(Tons)	(Tons)	(kW)	(GPH)	(W)	(GPH)	(Tons)	(Tons)	(kW)	(Btu/Wh)	(Btu/Wh)	Demand	Total E	Sens E
1a	IEC Only	Warm Dry	1.97	1.29	0.5	45.3	29.7	0.0	120%	2.13	3.32	2.14	0.30	3.6			7.51	5.25	10.5	8.6	6.0	95.0	81.0	79.7
1b	IEC Only	Western Peak	0.11	0.47	0.5	2.4	10.4	0.0	120%	3.76	0.75	1.09	0.31	6.8			6.18	4.80	12.2	6.1	4.7	95.6	-151.3	54.6
1c	IEC Only	AHRI 100% Capacity	-0.66	0.43	0.5	-14.9	9.7	13.0	127%	2.57	-1.20	0.83	0.31	4.6			6.61	4.94	11.6	6.9	5.1	95.4	#N/A	47.1
1d	IEC Only	IEER 75% Capacity	0.37	0.77	0.5	8.6	17.6	0.0	119%	1.75	1.12	1.40	0.30	3.0			7.39	5.21	10.7	8.3	5.8	95.1	3.9	66.9
1e	IEC Only	Warm Humid	-1.23	0.37	0.5	-27.6	8.3	13.1	124%	1.07	-1.82	0.68	0.30	1.9			7.33	5.19	10.7	8.2	5.8	95.0	#N/A	30.6
1f	IEC Only	IEER 50% Capacity	1.52	1.22	0.5	34.8	27.8	0.0	110%	1.11	2.85	2.03	0.30	1.9			8.05	5.45	9.9	9.8	6.6	94.7	71.9	76.2
1g	IEC Only	IEER 25% Capacity	2.64	1.64	0.5	60.3	37.5	6.1	124%	1.45	4.38	2.73	0.30	2.4			8.23	5.52	9.7	10.2	6.9	94.6	83.1	81.7
1h	IEC Only	Western Annual	1.22	0.94	0.5	28.0	21.7	7.1	120%	2.88	2.30	1.68	0.31	5.0			6.89	5.04	11.2	7.4	5.4	95.3	73.6	75.1
2a	IEC Only	Warm Dry	2.71	1.92	1.1	28.6	20.3	0.0	118%	3.12	4.59	3.13	0.92	5.5			7.52	5.26	10.5	8.6	6.0	89.2	70.0	70.3
2b	IEC Only	Western Peak	-0.35	0.66	1.2	-3.6	6.6	14.2	113%	5.44	0.58	1.31	0.96	10.0			6.20	4.81	12.2	6.1	4.7	90.2	#N/A	28.5
2c	IEC Only	AHRI 100% Capacity	-1.48	0.56	1.2	-15.2	5.8	12.6	119%	3.67	-2.13	1.03	0.95	6.7			6.66	4.96	11.5	6.9	5.2	89.9	#N/A	11.0
2d	IEC Only	IEER 75% Capacity	0.37	1.24	1.2	3.9	12.9	0.0	113%	2.50	1.44	2.06	0.93	4.4			7.42	5.22	10.7	8.3	5.9	89.2	-115.1	54.5
2e	IEC Only	Warm Humid	-1.99	0.61	1.2	-20.5	6.3	0.0	119%	1.65	-3.01	1.02	0.93	3.0			7.31	5.18	10.7	8.2	5.8	89.1	#N/A	8.3
2f	IEC Only	IEER 50% Capacity	2.41	1.98	1.1	25.8	21.2	0.0	104%	1.66	4.50	3.19	0.90	2.8			8.06	5.46	9.9	9.8	6.6	88.6	62.0	68.6
2g	IEC Only	IEER 25% Capacity	4.00	2.55	1.1	42.6	27.1	0.0	127%	2.11	6.30	4.16	0.91	3.6			8.23	5.52	9.7	10.2	6.9	88.3	76.0	74.7
2h	IEC Only	Western Annual	1.45	1.38	1.2	14.9	14.1	0.0	116%	4.16	2.81	2.31	0.97	7.5			6.89	5.04	11.2	7.4	5.4	89.5	50.4	61.8
3a	IEC Only	Warm Dry	2.80	2.20	1.8	18.2	14.3	0.0	105%	3.52	5.67	3.52	1.63	6.3			7.52	5.26	10.5	8.6	6.0	82.4	52.7	57.8
3b	IEC Only	Western Peak	-1.14	0.63	1.9	-7.2	4.0	13.3	108%	6.33	0.16	1.26	1.69	11.7			6.15	4.79	12.2	6.1	4.7	84.3	#N/A	-19.6
3c	IEC Only	AHRI 100% Capacity	-2.10	0.62	1.9	-13.4	3.9	0.0	115%	4.32	-2.76	1.11	1.67	8.0			6.65	4.96	11.5	6.9	5.2	83.7	#N/A	-31.5
3d	IEC Only	IEER 75% Capacity	-0.14	1.34	1.9	-0.9	8.5	0.0	108%	2.84	0.78	2.17	1.67	5.1			7.36	5.20	10.7	8.2	5.8	82.4	#N/A	31.7
3e	IEC Only	Warm Humid	-2.49	0.75	1.9	-16.1	4.9	0.0	118%	1.99	-3.71	1.22	1.62	3.6			7.30	5.18	10.7	8.2	5.8	82.7	#N/A	-18.6
3f	IEC Only	IEER 50% Capacity	3.03	2.48	1.8	19.9	16.3	0.0	100%	2.03	5.64	3.92	1.60	3.5			8.06	5.46	9.9	9.8	6.6	81.5	50.8	59.3
3g	IEC Only	IEER 25% Capacity	4.79	3.08	1.8	31.5	20.3	0.0	122%	2.45	7.47	4.96	1.61	4.2			8.22	5.52	9.7	10.2	6.9	81.2	67.6	66.3
3h	IEC Only	Western Annual	2.14	1.80	1.9	13.7	11.5	9.8	114%	5.13	4.02	2.88	1.67	9.2			6.91	5.04	11.2	7.4	5.4	83.3	46.0	53.3
4a	IEC+RTU	Warm Dry	9.13	6.36	10.8	10.2	7.1	0.0	122%	2.09	2.74	2.03	0.31	3.6			7.50	5.25	10.5	8.6	6.0	-2.5	15.8	15.4
4b	IEC+RTU	Western Peak	5.58	4.72	12.5	5.4	4.5	0.0	118%	3.68	0.70	1.06	0.32	6.6			6.17	4.80	12.2	6.1	4.7	-2.8	-13.7	-4.5
4c	IEC+RTU	AHRI 100% Capacity	5.12	4.87	12.2	5.0	4.8	13.7	120%	2.21	-1.18	0.71	0.32	4.0			6.64	4.95	11.6	6.8	5.1	-5.2	-36.5	-7.1
4d	IEC+RTU	IEER 75% Capacity	7.02	5.76	11.2	7.5	6.2	0.0	118%	1.88	1.02	1.43	0.31	3.3			7.32	5.19	10.7	8.2	5.8	-4.3	-8.9	6.0
4e	IEC+RTU	Warm Humid	5.93	5.05	11.4	6.2	5.3	0.0	131%	1.11	-1.80	0.76	0.31	2.0			7.32	5.19	10.7	8.2	5.8	-6.2	-31.1	-9.0
4f	IEC+RTU	IEER 50% Capacity	9.50	6.85	10.2	11.2	8.1	0.0	120%	1.49	2.95	2.18	0.30	2.6			8.06	5.46	9.9	9.8	6.6	-3.4	12.3	17.6
4g	IEC+RTU	IEER 25% Capacity	10.47	7.09	10.0	12.6	8.5	13.5	136%	1.05	3.39	2.57	0.31	1.7			8.21	5.51	9.7	10.2	6.8	-3.5	18.8	19.5
4h	IEC+RTU	Western Annual	8.28	5.90	11.4	8.7	6.2	10.7	121%	3.09	2.42	1.76	0.31	5.4			6.90	5.04	11.2	7.4	5.4	-1.8	15.2	13.1
5a	IEC+RTU	Warm Dry	10.27	7.13	12.0	10.2	7.1	12.8	111%	3.87	5.02	3.84	1.63	6.8			7.52	5.25	10.5	8.6	6.0	-15.0	15.8	15.3
5b	IEC+RTU	Western Peak	4.55	4.42	14.3	3.8	3.7	23.8	104%	6.34	0.16	1.35	1.71	11.4			6.27	4.83	12.2	6.2	4.7	-17.3	-61.3	-28.2
5c	IEC+RTU	AHRI 100% Capacity	3.35	4.46	14.1	2.9	3.8	15.4	110%	4.79	-3.22	1.25	1.68	8.8			6.59	4.94	11.7	6.7	5.0	-19.8	-135.4	-32.7
5d	IEC+RTU	IEER 75% Capacity	7.43	6.12	12.6	7.1	5.8	13.7	109%	3.83	1.81	2.72	1.64	6.8			7.34	5.19	10.7	8.2	5.8	-17.8	-16.3	0.0
5e	IEC+RTU	Warm Humid	4.84	4.80	13.1	4.4	4.4	12.0	123%	2.22	-4.14	1.53	1.64	3.9			7.31	5.18	10.7	8.2	5.8	-21.7	-83.7	-31.4
5f	IEC+RTU	IEER 50% Capacity	10.57	7.45	11.5	11.0	7.8	0.0	98%	2.39	5.49	4.11	1.62	3.9			8.06	5.46	9.9	9.8	6.6	-16.9	10.8	14.4
5g	IEC+RTU	IEER 25% Capacity	12.45	8.32	11.2	13.3	8.9	0.0	121%	1.64	6.65	5.31	1.67	2.6			8.18	5.50	9.6	10.2	6.8	-16.6	23.4	23.0
5h	IEC+RTU	Western Annual	9.57	6.62	12.7	9.0	6.2	13.8	107%	5.87	5.35	3.30	1.64	10.4			6.90	5.04	11.2	7.4	5.4	-13.4	18.3	13.6

			Combined System Metrics						IEC Metrics						Pre-Cooler		Baseline Comparison - 0% OA							
			Total	Sens	Power	EER -T	EER-S	MUW	EE	OA Cap	RA C-t	RA C-s	Power	Evap	Power	Evap	Total	Sensible	Power	EER -T	EER-S	% Savings		
Test #	Description	Climate	(Tons)	(Tons)	(kW)	(Btu/Wh)	(Btu/Wh)	(GPH)	(%)	(Tons)	(Tons)	(Tons)	(kW)	(GPH)	(W)	(GPH)	(Tons)	(Tons)	(kW)	(Btu/Wh)	(Btu/Wh)	Demand	Total E	Sens E
6a	IEC+RTU+PC	Warm Dry	9.48	6.60	10.1	11.3	7.8	29.1	127%	2.01	2.29	1.96	0.32	3.4	38	7.3	7.52	5.26	10.5	8.6	6.0	3.5	23.5	23.1
6b	IEC+RTU+PC	Western Peak	6.08	4.93	11.3	6.5	5.2	30.5	116%	3.38	-0.22	0.77	0.33	6.1	38	16.1	6.27	4.83	12.2	6.2	4.7	7.6	4.8	9.4
6c	IEC+RTU+PC	AHRI 100% Capacity	5.63	5.14	11.3	6.0	5.5	66.6	119%	2.18	-1.34	0.65	0.32	3.9	38	11.0	6.65	4.96	11.5	6.9	5.2	2.3	-15.4	5.7
6d	IEC+RTU+PC	IEER 75% Capacity	7.39	5.88	10.5	8.4	6.7	8.3	112%	1.76	1.00	1.35	0.32	3.1	38	9.5	7.34	5.19	10.7	8.2	5.8	1.6	2.4	13.1
6e	IEC+RTU+PC	Warm Humid	6.27	5.19	11.0	6.8	5.7	5.3	131%	1.15	-1.82	0.77	0.31	2.0	38	3.9	7.34	5.19	10.8	8.2	5.8	-2.5	-19.9	-2.5
6f	IEC+RTU+PC	IEER 50% Capacity	9.74	6.83	9.8	11.9	8.3	13.2	110%	1.24	2.78	2.06	0.31	2.1	38	5.5	8.03	5.45	9.9	9.7	6.6	0.4	17.9	20.6
6g	IEC+RTU+PC	IEER 25% Capacity	10.78	7.21	9.6	13.4	9.0	25.8	132%	1.01	3.38	2.59	0.31	1.6	38	3.6	8.20	5.51	9.7	10.2	6.8	0.2	24.1	23.8
6h	IEC+RTU+PC	Western Annual	8.85	6.19	10.4	10.2	7.1	14.6	118%	2.81	2.10	1.62	0.31	4.9	38	12.9	6.89	5.04	11.2	7.4	5.4	7.2	27.7	24.5
7a	IEC+RTU+PC	Warm Dry	10.48	7.26	11.5	10.9	7.6	7.6	117%	3.73	3.86	3.67	1.66	6.5	38	6.3	7.52	5.25	10.5	8.6	6.0	-10.0	21.1	20.4
7b	IEC+RTU+PC	Western Peak	4.66	4.44	13.1	4.3	4.1	32.0	104%	5.97	-1.54	0.89	1.71	10.8	38	16.0	6.33	4.85	12.2	6.2	4.8	-7.0	-45.3	-16.8
7c	IEC+RTU+PC	AHRI 100% Capacity	4.23	4.63	13.0	3.9	4.3	23.8	109%	4.48	-3.42	1.08	1.69	8.2	38	11.3	6.65	4.96	11.5	6.9	5.2	-12.6	-77.2	-20.6
7d	IEC+RTU+PC	IEER 75% Capacity	7.26	6.07	12.0	7.2	6.1	9.4	101%	3.33	1.70	2.53	1.65	5.9	38	8.4	7.34	5.19	10.7	8.2	5.8	-12.5	-13.7	3.8
7e	IEC+RTU+PC	Warm Humid	5.03	4.89	12.7	4.8	4.6	5.3	121%	2.23	-4.24	1.49	1.68	4.0	38	3.8	7.31	5.18	10.7	8.2	5.8	-18.5	-72.1	-25.4
7f	IEC+RTU+PC	IEER 50% Capacity	10.45	7.41	11.2	11.2	7.9	4.3	91%	2.12	5.15	3.89	1.61	3.5	38	4.9	8.06	5.46	9.9	9.8	6.6	-14.0	12.0	16.0
7g	IEC+RTU+PC	IEER 25% Capacity	12.99	8.55	10.9	14.3	9.4	11.7	129%	1.71	6.70	5.38	1.67	2.7	38	2.9	8.19	5.50	9.6	10.2	6.8	-13.1	28.7	27.2
7h	IEC+RTU+PC	Western Annual	8.72	6.24	12.1	8.6	6.2	11.3	107%	4.97	2.30	2.56	1.68	8.8	38	11.7	6.91	5.05	11.2	7.4	5.4	-7.8	14.5	12.9
8a	IEC+1C+PC	Warm Dry	5.06	3.64	6.3	9.6	6.9	9.0	121%	3.20	2.99	3.14	1.66	5.7	38	6.2	7.52	5.26	10.5	8.6	6.0	39.8	10.6	13.0
8b	IEC+1C+PC	Western Peak	1.67	2.08	7.1	2.8	3.5	28.7	107%	5.12	-1.21	0.93	1.69	9.4	38	16.1	6.33	4.85	12.2	6.2	4.8	42.4	-117.9	-34.2
8c	IEC+1C+PC	AHRI 100% Capacity	1.38	2.11	7.0	2.4	3.6	10.2	111%	3.65	-2.79	1.08	1.72	6.6	38	10.2	6.65	4.96	11.5	6.9	5.2	39.1	-192.6	-43.3
8d	IEC+1C+PC	IEER 75% Capacity	3.62	3.02	6.6	6.6	5.5	10.8	100%	2.70	1.62	2.25	1.65	4.9	38	10.1	7.34	5.19	10.7	8.2	5.8	38.7	-24.3	-5.3
8e	IEC+1C+PC	Warm Humid	1.83	2.32	6.9	3.2	4.0	14.9	126%	1.95	-3.62	1.36	1.69	3.6	38	3.3	7.30	5.18	10.7	8.2	5.8	35.5	-157.5	-44.0
8g	IEC+1C+PC	IEER 25% Capacity	5.97	4.09	5.9	12.1	8.3	3.2	141%	1.67	4.82	4.31	1.63	2.9	38	2.2	8.17	5.49	9.6	10.2	6.8	38.2	15.4	17.1
8h	IEC+1C+PC	Western Annual	4.54	3.30	6.5	8.4	6.1	36.9	108%	4.43	3.34	2.60	1.68	7.9	37	13.1	6.89	5.04	11.2	7.4	5.4	42.4	12.8	12.2
9a	RTU+PC	Warm Dry	8.44	5.97	9.9	10.2	7.2	10.6							38	7.8	7.52	5.26	10.5	8.6	6.0	5.6	15.9	17.0
9f	RTU+PC	IEER 50% Capacity	9.35	6.45	9.6	11.7	8.1	4.9							38	5.55	8.06	5.46	9.9	9.8	6.6	3.1	16.5	18.0
9g	RTU+PC	IEER 25% Capacity	9.92	6.69	9.4	12.7	8.5	6.0							38	3.93	8.22	5.52	9.7	10.2	6.9	2.7	19.4	19.8
9h	RTU+PC	Western Annual	7.16	5.27	10.4	8.3	6.1	18.2							38	11.92	6.92	5.05	11.2	7.4	5.4	7.4	10.5	11.2
10a	RTU+PC	Western Peak	7.87	5.28	11.1	8.5	5.7	0.0							37	17.02	6.78	4.92	12.5	6.5	4.7	11.5	23.7	17.4
10b	RTU+PC	AHRI 100% Capacity	7.79	5.33	11.1	8.4	5.8	0.0							38	6.91	7.23	5.07	11.9	7.3	5.1	6.7	13.4	11.2
10c	RTU+PC	Warm Humid	7.93	5.39	10.9	8.7	5.9	0.0							38	4.38	7.82	5.29	11.1	8.4	5.7	1.9	3.3	3.6
10d	RTU+PC	Western Annual	8.45	5.54	10.5	9.6	6.3	0.0							38	14.25	7.44	5.15	11.6	7.7	5.3	9.2	20.1	15.7
11a	RTU Baseline	Western Peak	4.44	4.15	12.4	4.3	4.0										6.17	4.80	12.2	6.1	4.7	-2.1	-41.9	-18.0
11b	RTU Baseline	AHRI 100% Capacity	5.70	4.73	11.7	5.8	4.8										6.65	4.96	11.5	6.9	5.2	-1.8	-18.8	-6.6
11c	RTU Baseline	Western Annual	6.34	5.04	11.4	6.7	5.3										6.90	5.04	11.2	7.4	5.4	-1.6	-10.4	-1.6
11d	RTU Baseline	Warm Humid	7.28	5.51	10.9	8.0	6.1										7.31	5.18	10.7	8.2	5.8	-1.6	-2.0	4.5
11e	RTU Baseline		8.14	6.15	10.6	9.2	7.0										7.67	5.31	10.3	8.9	6.2	-2.9	3.0	11.2
11f	RTU Baseline		8.05	6.14	10.5	9.2	7.0										7.78	5.35	10.2	9.2	6.3	-3.4	0.1	9.9
11g	RTU Baseline		9.98	7.03	9.9	12.2	8.6										8.28	5.54	9.6	10.3	6.9	-2.5	14.9	19.2
12a	RTU Baseline	Western Peak	6.55	4.75	12.5	6.3	4.6										6.79	4.92	12.5	6.5	4.7	0.0	0.0	0.0
12b	RTU Baseline	AHRI 100% Capacity	7.22	5.06	11.9	7.3	5.1										7.23	5.07	11.9	7.3	5.1	0.0	0.0	0.0
12c	RTU Baseline	Western Annual	7.54	5.19	11.6	7.8	5.4										7.54	5.19	11.6	7.8	5.4	0.0	0.0	0.0
12d	RTU Baseline	Warm Humid	8.02	5.43	11.2	8.6	5.8										7.82	5.29	11.1	8.4	5.7	0.0	0.0	0.0
	RTU Baseline	IEER 75% Capacity	8.03	5.42	11.2	8.6	5.8										8.03	5.42	11.2	8.6	5.8	0.0	0.0	0.0
	RTU Baseline	Warm Dry	8.22	5.51	11.0	9.0	6.0										8.22	5.51	11.0	9.0	6.0	0.0	0.0	0.0
12e	RTU Baseline		8.38	5.60	10.8	9.3	6.2										8.38	5.60	10.8	9.3	6.2	0.0	0.0	0.0
12f	RTU Baseline		8.48	5.65	10.7	9.5	6.4										8.48	5.65	10.7	9.5	6.4	0.0	0.0	0.0
	RTU Baseline	IEER 50% Capacity	8.73	5.80	10.4	10.1	6.7										8.73	5.80	10.4	10.1	6.7	0.0	0.0	0.0
12d	RTU Baseline	IEER 25% Capacity	8.02	5.43	11.2	8.6	5.8										7.82	5.29	11.1	8.4	5.7	0.0	0.0	0.0
12g	RTU Baseline		8.92	5.92	10.2	10.5	7.0										8.92	5.92	10.2	10.5	7.0	0.0	0.0	0.0

Table A-10: WCC Test Results Table – Performance Metrics & Comparison to 25% OA Baseline

			Combined System Metrics						IEC Metrics						Pre-Cooler		Baseline Comparison - 25% OA							
			Total	Sens	Power	EER -T	EER-S	MUW	EE	OA Cap	RA C-t	RA C-s	Power	Evap	Power	Evap	Total	Sensible	Power	EER -T	EER-S	% Savings		
Test #	Description	Climate	(Tons)	(Tons)	(kW)	(Btu/Wh)	(Btu/Wh)	(GPH)	(%)	(Tons)	(Tons)	(Tons)	(kW)	(GPH)	(W)	(GPH)	(Tons)	(Tons)	(kW)	(Btu/Wh)	(Btu/Wh)	Demand	Total E	Sens E
1a	IEC Only	Warm Dry	1.97	1.29	0.5	45.3	29.7	0.0	120%	2.13	3.32	2.14	0.30	3.6			7.55	5.18	10.4	8.7	6.0	95.0	80.7	79.8
1b	IEC Only	Western Peak	0.11	0.47	0.5	2.4	10.4	0.0	120%	3.76	0.75	1.09	0.31	6.8			6.22	4.50	12.5	6.0	4.3	95.7	-146.9	58.5
1c	IEC Only	AHRI 100% Capacity	-0.66	0.43	0.5	-14.9	9.7	13.0	127%	2.57	-1.20	0.83	0.31	4.6			6.59	4.78	11.9	6.6	4.8	95.5	#N/A	50.5
1d	IEC Only	IEER 75% Capacity	0.37	0.77	0.5	8.6	17.6	0.0	119%	1.75	1.12	1.40	0.30	3.0			7.38	5.18	10.8	8.2	5.8	95.2	4.6	67.3
1e	IEC Only	Warm Humid	-1.23	0.37	0.5	-27.6	8.3	13.1	124%	1.07	-1.82	0.68	0.30	1.9			7.27	5.22	11.1	7.9	5.7	95.2	#N/A	32.1
1f	IEC Only	IEER 50% Capacity	1.52	1.22	0.5	34.8	27.8	0.0	110%	1.11	2.85	2.03	0.30	1.9			8.11	5.47	9.7	10.0	6.8	94.6	71.1	75.6
1g	IEC Only	IEER 25% Capacity	2.64	1.64	0.5	60.3	37.5	6.1	124%	1.45	4.38	2.73	0.30	2.4			8.35	5.49	9.4	10.7	7.0	94.4	82.3	81.3
1h	IEC Only	Western Annual	1.22	0.94	0.5	28.0	21.7	7.1	120%	2.88	2.30	1.68	0.31	5.0			6.88	4.87	11.3	7.3	5.2	95.4	73.8	76.1
2a	IEC Only	Warm Dry	2.71	1.92	1.1	28.6	20.3	0.0	118%	3.12	4.59	3.13	0.92	5.5			7.64	5.20	10.4	8.8	6.0	89.1	69.2	70.4
2b	IEC Only	Western Peak	-0.35	0.66	1.2	-3.6	6.6	14.2	113%	5.44	0.58	1.31	0.96	10.0			5.94	4.25	12.5	5.7	4.1	90.5	#N/A	38.4
2c	IEC Only	AHRI 100% Capacity	-1.48	0.56	1.2	-15.2	5.8	12.6	119%	3.67	-2.13	1.03	0.95	6.7			6.28	4.64	11.9	6.3	4.7	90.2	#N/A	19.4
2d	IEC Only	IEER 75% Capacity	0.37	1.24	1.2	3.9	12.9	0.0	113%	2.50	1.44	2.06	0.93	4.4			7.34	5.18	10.8	8.2	5.8	89.3	-111.5	55.2
2e	IEC Only	Warm Humid	-1.99	0.61	1.2	-20.5	6.3	0.0	119%	1.65	-3.01	1.02	0.93	3.0			6.97	5.18	11.0	7.6	5.6	89.4	#N/A	10.9
2f	IEC Only	IEER 50% Capacity	2.41	1.98	1.1	25.8	21.2	0.0	104%	1.66	4.50	3.19	0.90	2.8			8.30	5.57	9.7	10.3	6.9	88.4	60.2	67.3
2g	IEC Only	IEER 25% Capacity	4.00	2.55	1.1	42.6	27.1	0.0	127%	2.11	6.30	4.16	0.91	3.6			8.58	5.64	9.4	10.9	7.2	88.0	74.3	73.5
2h	IEC Only	Western Annual	1.45	1.38	1.2	14.9	14.1	0.0	116%	4.16	2.81	2.31	0.97	7.5			6.80	4.78	11.3	7.2	5.1	89.6	51.4	64.1
3a	IEC Only	Warm Dry	2.80	2.20	1.8	18.2	14.3	0.0	105%	3.52	5.67	3.52	1.63	6.3			7.72	5.19	10.4	8.9	6.0	82.2	50.9	57.9
3b	IEC Only	Western Peak	-1.14	0.63	1.9	-7.2	4.0	13.3	108%	6.33	0.16	1.26	1.69	11.7			5.69	4.09	12.5	5.5	3.9	84.7	#N/A	0.6
3c	IEC Only	AHRI 100% Capacity	-2.10	0.62	1.9	-13.4	3.9	0.0	115%	4.32	-2.76	1.11	1.67	8.0			6.07	4.54	11.9	6.1	4.6	84.2	#N/A	-16.7
3d	IEC Only	IEER 75% Capacity	-0.14	1.34	1.9	-0.9	8.5	0.0	108%	2.84	0.78	2.17	1.67	5.1			7.17	5.14	10.8	7.9	5.7	82.5	#N/A	33.1
3e	IEC Only	Warm Humid	-2.49	0.75	1.9	-16.1	4.9	0.0	118%	1.99	-3.71	1.22	1.62	3.6			6.78	5.16	11.0	7.4	5.6	83.2	#N/A	-14.9
3f	IEC Only	IEER 50% Capacity	3.03	2.48	1.8	19.9	16.3	0.0	100%	2.03	5.64	3.92	1.60	3.5			8.42	5.63	9.7	10.5	7.0	81.2	47.6	57.2
3g	IEC Only	IEER 25% Capacity	4.79	3.08	1.8	31.5	20.3	0.0	122%	2.45	7.47	4.96	1.61	4.2			8.73	5.72	9.4	11.1	7.3	80.6	64.7	64.1
3h	IEC Only	Western Annual	2.14	1.80	1.9	13.7	11.5	9.8	114%	5.13	4.02	2.88	1.67	9.2			6.82	4.70	11.3	7.2	5.0	83.4	47.0	56.7
4a	IEC+RTU	Warm Dry	9.13	6.36	10.8	10.2	7.1	0.0	122%	2.09	2.74	2.03	0.31	3.6			7.81	5.19	10.4	9.0	6.0	-3.1	11.8	15.8
4b	IEC+RTU	Western Peak	5.58	4.72	12.5	5.4	4.5	0.0	118%	3.68	0.70	1.06	0.32	6.6			4.85	3.30	12.5	4.7	3.2	-0.3	12.8	29.9
4c	IEC+RTU	AHRI 100% Capacity	5.12	4.87	12.2	5.0	4.8	13.7	120%	2.21	-1.18	0.71	0.32	4.0			4.91	3.99	12.0	4.9	4.0	-1.9	2.3	16.4
4d	IEC+RTU	IEER 75% Capacity	7.02	5.76	11.2	7.5	6.2	0.0	118%	1.88	1.02	1.43	0.31	3.3			6.96	5.01	10.8	7.8	5.6	-3.6	-2.8	9.8
4e	IEC+RTU	Warm Humid	5.93	5.05	11.4	6.2	5.3	0.0	131%	1.11	-1.80	0.76	0.31	2.0			5.90	5.04	11.0	6.4	5.5	-3.2	-2.8	-3.0
4f	IEC+RTU	IEER 50% Capacity	9.50	6.85	10.2	11.2	8.1	0.0	120%	1.49	2.95	2.18	0.30	2.6			8.93	5.95	9.7	11.1	7.4	-5.2	1.1	8.6
4g	IEC+RTU	IEER 25% Capacity	10.47	7.09	10.0	12.6	8.5	13.5	136%	1.05	3.39	2.57	0.31	1.7			9.24	6.14	9.4	11.7	7.8	-5.8	6.6	8.3
4h	IEC+RTU	Western Annual	8.28	5.90	11.4	8.7	6.2	10.7	121%	3.09	2.42	1.76	0.31	5.4			6.65	4.35	11.3	7.1	4.6	-1.3	18.6	25.4
5a	IEC+RTU	Warm Dry	10.27	7.13	12.0	10.2	7.1	12.8	111%	3.87	5.02	3.84	1.63	6.8			7.79	5.23	10.4	9.0	6.0	-15.5	12.4	15.4
5b	IEC+RTU	Western Peak	4.55	4.42	14.3	3.8	3.7	23.8	104%	6.34	0.16	1.35	1.71	11.4			4.76	3.23	12.5	4.6	3.1	-14.3	-19.5	16.5
5c	IEC+RTU	AHRI 100% Capacity	3.35	4.46	14.1	2.9	3.8	15.4	110%	4.79	-3.22	1.25	1.68	8.8			4.72	3.86	12.1	4.7	3.8	-15.9	-63.1	-0.3
5d	IEC+RTU	IEER 75% Capacity	7.43	6.12	12.6	7.1	5.8	13.7	109%	3.83	1.81	2.72	1.64	6.8			6.95	5.01	10.8	7.7	5.6	-17.0	-9.4	4.2
5e	IEC+RTU	Warm Humid	4.84	4.80	13.1	4.4	4.4	12.0	123%	2.22	-4.14	1.53	1.64	3.9			5.80	5.03	11.0	6.3	5.5	-18.4	-41.8	-24.0
5f	IEC+RTU	IEER 50% Capacity	10.57	7.45	11.5	11.0	7.8	0.0	98%	2.39	5.49	4.11	1.62	3.9			8.91	5.98	9.7	11.0	7.4	-18.9	-0.2	4.6
5g	IEC+RTU	IEER 25% Capacity	12.45	8.32	11.2	13.3	8.9	0.0	121%	1.64	6.65	5.31	1.67	2.6			9.19	6.19	9.4	11.7	7.9	-18.9	12.2	11.5
5h	IEC+RTU	Western Annual	9.57	6.62	12.7	9.0	6.2	13.8	107%	5.87	5.35	3.30	1.64	10.4			6.77	4.29	11.3	7.2	4.6	-13.1	20.1	26.7

			Combined System Metrics						IEC Metrics						Pre-Cooler		Baseline Comparison - 25% OA							
			Total	Sens	Power	EER -T	EER-S	MUW	EE	OA Cap	RA C-t	RA C-s	Power	Evap	Power	Evap	Total	Sensible	Power	EER -T	EER-S	% Savings		
Test #	Description	Climate	(Tons)	(Tons)	(kW)	(Btu/Wh)	(Btu/Wh)	(GPH)	(%)	(Tons)	(Tons)	(Tons)	(kW)	(GPH)	(W)	(GPH)	(Tons)	(Tons)	(kW)	(Btu/Wh)	(Btu/Wh)	Demand	Total E	Sens E
6a	IEC+RTU+PC	Warm Dry	9.48	6.60	10.1	11.3	7.8	29.1	127%	2.01	2.29	1.96	0.32	3.4	38	7.3	7.67	5.24	10.5	8.8	6.0	3.3	21.7	23.1
6b	IEC+RTU+PC	Western Peak	6.08	4.93	11.3	6.5	5.2	30.5	116%	3.38	-0.22	0.77	0.33	6.1	38	16.1	4.67	3.31	12.6	4.5	3.2	10.3	31.2	39.7
6c	IEC+RTU+PC	AHRI 100% Capacity	5.63	5.14	11.3	6.0	5.5	66.6	119%	2.18	-1.34	0.65	0.32	3.9	38	11.0	4.90	4.03	11.9	4.9	4.1	5.4	17.6	25.8
6d	IEC+RTU+PC	IEER 75% Capacity	7.39	5.88	10.5	8.4	6.7	8.3	112%	1.76	1.00	1.35	0.32	3.1	38	9.5	7.01	5.02	10.8	7.8	5.6	2.3	7.4	16.6
6e	IEC+RTU+PC	Warm Humid	6.27	5.19	11.0	6.8	5.7	5.3	131%	1.15	-1.82	0.77	0.31	2.0	38	3.9	5.94	5.05	11.1	6.4	5.5	0.4	5.7	3.1
6f	IEC+RTU+PC	IEER 50% Capacity	9.74	6.83	9.8	11.9	8.3	13.2	110%	1.24	2.78	2.06	0.31	2.1	38	5.5	8.84	5.91	9.7	10.9	7.3	-1.2	8.1	12.4
6g	IEC+RTU+PC	IEER 25% Capacity	10.78	7.21	9.6	13.4	9.0	25.8	132%	1.01	3.38	2.59	0.31	1.6	38	3.6	9.21	6.14	9.4	11.7	7.8	-2.0	12.9	13.2
6h	IEC+RTU+PC	Western Annual	8.85	6.19	10.4	10.2	7.1	14.6	118%	2.81	2.10	1.62	0.31	4.9	38	12.9	6.57	4.35	11.3	7.0	4.6	7.8	31.6	35.2
7a	IEC+RTU+PC	Warm Dry	10.48	7.26	11.5	10.9	7.6	7.6	117%	3.73	3.86	3.67	1.66	6.5	38	6.3	7.56	5.25	10.5	8.7	6.0	-10.1	20.6	20.4
7b	IEC+RTU+PC	Western Peak	4.66	4.44	13.1	4.3	4.1	32.0	104%	5.97	-1.54	0.89	1.71	10.8	38	16.0	4.57	3.24	12.6	4.3	3.1	-3.8	-1.6	24.3
7c	IEC+RTU+PC	AHRI 100% Capacity	4.23	4.63	13.0	3.9	4.3	23.8	109%	4.48	-3.42	1.08	1.69	8.2	38	11.3	4.83	4.00	11.9	4.9	4.0	-9.1	-24.7	5.8
7d	IEC+RTU+PC	IEER 75% Capacity	7.26	6.07	12.0	7.2	6.1	9.4	101%	3.33	1.70	2.53	1.65	5.9	38	8.4	6.97	5.01	10.8	7.8	5.6	-11.7	-7.4	7.8
7e	IEC+RTU+PC	Warm Humid	5.03	4.89	12.7	4.8	4.6	5.3	121%	2.23	-4.24	1.49	1.68	4.0	38	3.8	5.79	5.03	11.0	6.3	5.5	-15.2	-32.5	-18.4
7f	IEC+RTU+PC	IEER 50% Capacity	10.45	7.41	11.2	11.2	7.9	4.3	91%	2.12	5.15	3.89	1.61	3.5	38	4.9	8.88	5.99	9.7	11.0	7.4	-15.8	1.5	6.3
7g	IEC+RTU+PC	IEER 25% Capacity	12.99	8.55	10.9	14.3	9.4	11.7	129%	1.71	6.70	5.38	1.67	2.7	38	2.9	9.20	6.20	9.5	11.7	7.9	-15.3	18.4	16.3
7h	IEC+RTU+PC	Western Annual	8.72	6.24	12.1	8.6	6.2	11.3	107%	4.97	2.30	2.56	1.68	8.8	38	11.7	6.27	4.34	11.4	6.6	4.6	-6.6	23.3	25.9
8a	IEC+1C+PC	Warm Dry	5.06	3.64	6.3	9.6	6.9	9.0	121%	3.20	2.99	3.14	1.66	5.7	38	6.2	7.51	5.26	10.5	8.6	6.0	39.9	10.9	13.0
8b	IEC+1C+PC	Western Peak	1.67	2.08	7.1	2.8	3.5	28.7	107%	5.12	-1.21	0.93	1.69	9.4	38	16.1	5.81	4.11	12.6	5.5	3.9	44.2	-93.8	-10.4
8c	IEC+1C+PC	AHRI 100% Capacity	1.38	2.11	7.0	2.4	3.6	10.2	111%	3.65	-2.79	1.08	1.72	6.6	38	10.2	6.08	4.55	11.9	6.1	4.6	41.0	-159.0	-27.5
8d	IEC+1C+PC	IEER 75% Capacity	3.62	3.02	6.6	6.6	5.5	10.8	100%	2.70	1.62	2.25	1.65	4.9	38	10.1	7.22	5.12	10.8	8.0	5.7	39.1	-21.6	-3.1
8e	IEC+1C+PC	Warm Humid	1.83	2.32	6.9	3.2	4.0	14.9	126%	1.95	-3.62	1.36	1.69	3.6	38	3.3	6.78	5.16	11.0	7.4	5.6	37.3	-132.4	-39.4
8g	IEC+1C+PC	IEER 25% Capacity	5.97	4.09	5.9	12.1	8.3	3.2	141%	1.67	4.82	4.31	1.63	2.9	38	2.2	8.46	5.76	9.5	10.7	7.3	37.2	11.0	11.7
8h	IEC+1C+PC	Western Annual	4.54	3.30	6.5	8.4	6.1	36.9	108%	4.43	3.34	2.60	1.68	7.9	37	13.1	6.76	4.71	11.3	7.2	5.0	42.9	15.0	18.5
9a	RTU+PC	Warm Dry	8.44	5.97	9.9	10.2	7.2	10.6							38	7.8	7.77	5.23	10.4	8.9	6.0	5.1	12.6	17.0
9f	RTU+PC	IEER 50% Capacity	9.35	6.45	9.6	11.7	8.1	4.9							38	5.55	8.94	5.92	9.7	11.1	7.3	1.3	5.6	9.3
9g	RTU+PC	IEER 25% Capacity	9.92	6.69	9.4	12.7	8.5	6.0							38	3.93	9.27	6.13	9.4	11.8	7.8	0.4	6.9	8.9
9h	RTU+PC	Western Annual	7.16	5.27	10.4	8.3	6.1	18.2							38	11.92	6.41	4.40	11.4	6.8	4.6	8.3	18.0	23.4
10a	RTU+PC	Western Peak	7.87	5.28	11.1	8.5	5.7	0.0							37	17.02	5.84	3.31	12.7	5.5	3.1	12.9	35.4	45.4
10b	RTU+PC	AHRI 100% Capacity	7.79	5.33	11.1	8.4	5.8	0.0							38	6.91	5.82	4.06	12.2	5.7	4.0	9.0	32.0	30.6
10c	RTU+PC	Warm Humid	7.93	5.39	10.9	8.7	5.9	0.0							38	4.38	6.75	5.15	11.3	7.1	5.4	3.8	18.1	8.1
10d	RTU+PC	Western Annual	8.45	5.54	10.5	9.6	6.3	0.0							38	14.25	7.92	4.49	11.5	8.3	4.7	8.5	14.2	25.8
11a	RTU Baseline	Western Peak	4.44	4.15	12.4	4.3	4.0										4.96	3.30	12.4	4.8	3.2	0.1	-11.5	20.7
11b	RTU Baseline	AHRI 100% Capacity	5.70	4.73	11.7	5.8	4.8										5.89	4.02	11.7	6.0	4.1	-0.3	-3.7	14.9
11c	RTU Baseline	Western Annual	6.34	5.04	11.4	6.7	5.3										6.37	4.37	11.3	6.7	4.6	-0.6	-0.9	12.7
11d	RTU Baseline	Warm Humid	7.28	5.51	10.9	8.0	6.1										7.14	4.96	10.8	8.0	5.5	-1.2	0.7	8.9
11e	RTU Baseline		8.14	6.15	10.6	9.2	7.0										7.79	5.49	10.3	9.1	6.4	-3.1	1.3	8.0
11f	RTU Baseline		8.05	6.14	10.5	9.2	7.0										7.72	5.68	10.2	9.1	6.7	-3.2	1.1	4.5
11g	RTU Baseline		9.98	7.03	9.9	12.2	8.6										9.57	6.29	9.4	12.3	8.0	-5.1	-0.8	6.0
12a	RTU Baseline	Western Peak	6.55	4.75	12.5	6.3	4.6										6.47	3.32	12.6	6.2	3.2	0.9	2.1	30.8
12b	RTU Baseline	AHRI 100% Capacity	7.22	5.06	11.9	7.3	5.1										7.43	4.10	11.9	7.5	4.1	-0.3	-3.3	18.7
12c	RTU Baseline	Western Annual	7.54	5.19	11.6	7.8	5.4										15.71	2.93	10.3	18.3	3.4	-13.0	-135.6	36.3
12d	RTU Baseline	Warm Humid	8.02	5.43	11.2	8.6	5.8										8.74	5.13	10.9	9.6	5.6	-2.2	-11.4	3.6
	RTU Baseline	IEER 75% Capacity	8.03	5.42	11.2	8.6	5.8										16.78	3.41	9.7	20.8	4.2	-15.1	-140.6	27.6
	RTU Baseline	Warm Dry	8.22	5.51	11.0	9.0	6.0										17.23	3.59	9.4	21.9	4.6	-16.1	-143.3	24.5
12e	RTU Baseline		8.38	5.60	10.8	9.3	6.2										17.62	3.73	9.2	22.9	4.9	-16.9	-145.9	22.0
12f	RTU Baseline		8.48	5.65	10.7	9.5	6.4										17.88	3.82	9.1	23.6	5.1	-17.5	-147.7	20.5
	RTU Baseline	IEER 50% Capacity	8.73	5.80	10.4	10.1	6.7										18.54	4.04	8.7	25.5	5.5	-19.1	-152.9	17.1
12d	RTU Baseline	IEER 25% Capacity	8.02	5.43	11.2	8.6	5.8										8.74	5.13	10.9	9.6	5.6	-2.2	-11.4	3.6
12g	RTU Baseline		8.92	5.92	10.2	10.5	7.0										19.08	4.19	8.4	27.1	6.0	-20.4	-157.6	14.7

Table A-11: WCC Test Results Table – Performance Metrics & Comparison to Same % OA Baseline

			Combined System Metrics						IEC Metrics						Pre-Cooler		Baseline Comparison - Same % OA								
			Total	Sens	Power	EER -T	EER-S	MUW	EE	OA Cap	RA C-t	RA C-s	Power	Evap	Power	Evap	%	Total	Sensible	Power	EER -T	EER-S	% Savings		
Test #	Description	Climate	(Tons)	(Tons)	(kW)	(Btu/Wh)	(Btu/Wh)	(GPH)	(%)	(Tons)	(Tons)	(Tons)	(kW)	(GPH)	(W)	(GPH)	OA	(Tons)	(Tons)	(kW)	(Btu/Wh)	(Btu/Wh)	Demand	Total E	Sens E
1a	IEC Only	Warm Dry	1.97	1.29	0.5	45.3	29.7	0.0	120%	2.13	3.32	2.14	0.30	3.6			100	7.69	4.87	10.0	9.2	5.8	94.8	79.7	80.4
1b	IEC Only	Western Peak	0.11	0.47	0.5	2.4	10.4	0.0	120%	3.76	0.75	1.09	0.31	6.8			100	6.35	2.84	13.4	5.7	2.5	96.0	-133.7	75.7
1c	IEC Only	AHRI 100% Capacity	-0.66	0.43	0.5	-14.9	9.7	13.0	127%	2.57	-1.20	0.83	0.31	4.6			100	6.53	3.15	13.2	5.9	2.9	96.0	#N/A	70.6
1d	IEC Only	IEER 75% Capacity	0.37	0.77	0.5	8.6	17.6	0.0	119%	1.75	1.12	1.40	0.30	3.0			100	7.35	5.08	11.0	8.0	5.5	95.2	6.7	68.5
1e	IEC Only	Warm Humid	-1.23	0.37	0.5	-27.6	8.3	13.1	124%	1.07	-1.82	0.68	0.30	1.9			100	7.12	4.65	12.1	7.1	4.6	95.6	#N/A	44.6
1f	IEC Only	IEER 50% Capacity	1.52	1.22	0.5	34.8	27.8	0.0	110%	1.11	2.85	2.03	0.30	1.9			100	8.32	5.27	9.2	10.9	6.9	94.3	68.8	75.3
1g	IEC Only	IEER 25% Capacity	2.64	1.64	0.5	60.3	37.5	6.1	124%	1.45	4.38	2.73	0.30	2.4			100	8.83	4.73	8.7	12.2	6.5	93.9	79.8	82.6
1h	IEC Only	Western Annual	1.22	0.94	0.5	28.0	21.7	7.1	120%	2.88	2.30	1.68	0.31	5.0			100	6.87	4.34	11.5	7.2	4.5	95.4	74.3	79.0
2a	IEC Only	Warm Dry	2.71	1.92	1.1	28.6	20.3	0.0	118%	3.12	4.59	3.13	0.92	5.5			100	7.99	4.98	10.1	9.5	5.9	88.8	67.0	70.9
2b	IEC Only	Western Peak	-0.35	0.66	1.2	-3.6	6.6	14.2	113%	5.44	0.58	1.31	0.96	10.0			100	5.16	1.76	13.5	4.6	1.6	91.2	#N/A	76.4
2c	IEC Only	AHRI 100% Capacity	-1.48	0.56	1.2	-15.2	5.8	12.6	119%	3.67	-2.13	1.03	0.95	6.7			100	5.17	2.56	13.2	4.7	2.3	91.1	#N/A	59.8
2d	IEC Only	IEER 75% Capacity	0.37	1.24	1.2	3.9	12.9	0.0	113%	2.50	1.44	2.06	0.93	4.4			100	7.11	5.02	11.0	7.8	5.5	89.5	-100.8	57.4
2e	IEC Only	Warm Humid	-1.99	0.61	1.2	-20.5	6.3	0.0	119%	1.65	-3.01	1.02	0.93	3.0			100	5.97	4.50	12.1	5.9	4.5	90.3	#N/A	29.1
2f	IEC Only	IEER 50% Capacity	2.41	1.98	1.1	25.8	21.2	0.0	104%	1.66	4.50	3.19	0.90	2.8			100	9.05	5.66	9.2	11.8	7.4	87.8	54.2	65.0
2g	IEC Only	IEER 25% Capacity	4.00	2.55	1.1	42.6	27.1	0.0	127%	2.11	6.30	4.16	0.91	3.6			100	9.74	5.44	8.8	13.4	7.5	87.1	68.6	72.5
2h	IEC Only	Western Annual	1.45	1.38	1.2	14.9	14.1	0.0	116%	4.16	2.81	2.31	0.97	7.5			100	6.53	3.93	11.6	6.8	4.1	89.9	54.4	71.1
3a	IEC Only	Warm Dry	2.80	2.20	1.8	18.2	14.3	0.0	105%	3.52	5.67	3.52	1.63	6.3			100	8.35	4.90	10.1	9.9	5.8	81.7	45.4	59.1
3b	IEC Only	Western Peak	-1.14	0.63	1.9	-7.2	4.0	13.3	108%	6.33	0.16	1.26	1.69	11.7			100	4.33	1.08	13.6	3.8	1.0	85.9	#N/A	75.8
3c	IEC Only	AHRI 100% Capacity	-2.10	0.62	1.9	-13.4	3.9	0.0	115%	4.32	-2.76	1.11	1.67	8.0			100	4.33	2.19	13.2	3.9	2.0	85.7	#N/A	49.1
3d	IEC Only	IEER 75% Capacity	-0.14	1.34	1.9	-0.9	8.5	0.0	108%	2.84	0.78	2.17	1.67	5.1			100	6.60	4.87	11.2	7.1	5.2	83.1	#N/A	38.6
3e	IEC Only	Warm Humid	-2.49	0.75	1.9	-16.1	4.9	0.0	118%	1.99	-3.71	1.22	1.62	3.6			100	5.23	4.41	12.1	5.2	4.4	84.6	#N/A	10.1
3f	IEC Only	IEER 50% Capacity	3.03	2.48	1.8	19.9	16.3	0.0	100%	2.03	5.64	3.92	1.60	3.5			100	9.55	5.89	9.2	12.5	7.7	80.1	37.3	52.8
3g	IEC Only	IEER 25% Capacity	4.79	3.08	1.8	31.5	20.3	0.0	122%	2.45	7.47	4.96	1.61	4.2			100	10.34	5.79	8.8	14.2	7.9	79.2	55.1	61.0
3h	IEC Only	Western Annual	2.14	1.80	1.9	13.7	11.5	9.8	114%	5.13	4.02	2.88	1.67	9.2			100	6.54	3.66	11.5	6.8	3.8	83.6	50.0	66.8
4a	IEC+RTU	Warm Dry	9.13	6.36	10.8	10.2	7.1	0.0	122%	2.09	2.74	2.03	0.31	3.6			36	7.94	5.17	10.4	9.1	6.0	-3.4	10.1	16.0
4b	IEC+RTU	Western Peak	5.58	4.72	12.5	5.4	4.5	0.0	118%	3.68	0.70	1.06	0.32	6.6			33	4.44	2.80	12.6	4.2	2.7	0.4	20.9	40.9
4c	IEC+RTU	AHRI 100% Capacity	5.12	4.87	12.2	5.0	4.8	13.7	120%	2.21	-1.18	0.71	0.32	4.0			30	4.58	3.79	12.1	4.5	3.8	-1.2	9.4	21.2
4d	IEC+RTU	IEER 75% Capacity	7.02	5.76	11.2	7.5	6.2	0.0	118%	1.88	1.02	1.43	0.31	3.3			35	6.82	4.94	10.8	7.6	5.5	-3.3	-0.4	11.3
4e	IEC+RTU	Warm Humid	5.93	5.05	11.4	6.2	5.3	0.0	131%	1.11	-1.80	0.76	0.31	2.0			34	5.42	4.97	11.2	5.8	5.3	-2.2	6.5	-0.5
4f	IEC+RTU	IEER 50% Capacity	9.50	6.85	10.2	11.2	8.1	0.0	120%	1.49	2.95	2.18	0.30	2.6			35	9.29	6.14	9.6	11.6	7.7	-6.0	-3.6	4.9
4g	IEC+RTU	IEER 25% Capacity	10.47	7.09	10.0	12.6	8.5	13.5	136%	1.05	3.39	2.57	0.31	1.7			35	9.67	6.38	9.4	12.4	8.2	-6.7	1.4	3.8
4h	IEC+RTU	Western Annual	8.28	5.90	11.4	8.7	6.2	10.7	121%	3.09	2.42	1.76	0.31	5.4			35	6.55	4.06	11.3	7.0	4.3	-1.1	20.1	30.5
5a	IEC+RTU	Warm Dry	10.27	7.13	12.0	10.2	7.1	12.8	111%	3.87	5.02	3.84	1.63	6.8			75	8.33	5.16	10.3	9.7	6.0	-16.7	5.3	15.7
5b	IEC+RTU	Western Peak	4.55	4.42	14.3	3.8	3.7	23.8	104%	6.34	0.16	1.35	1.71	11.4			74	1.84	-0.30	13.2	1.7	-0.3	-8.4	56.2	#N/A
5c	IEC+RTU	AHRI 100% Capacity	3.35	4.46	14.1	2.9	3.8	15.4	110%	4.79	-3.22	1.25	1.68	8.8			70	1.37	1.42	12.9	1.3	1.3	-9.0	55.5	65.4
5d	IEC+RTU	IEER 75% Capacity	7.43	6.12	12.6	7.1	5.8	13.7	109%	3.83	1.81	2.72	1.64	6.8			74	6.19	4.63	10.9	6.8	5.1	-15.3	4.0	12.8
5e	IEC+RTU	Warm Humid	4.84	4.80	13.1	4.4	4.4	12.0	123%	2.22	-4.14	1.53	1.64	3.9			71	3.05	4.46	11.6	3.1	4.6	-12.2	29.4	-4.1
5f	IEC+RTU	IEER 50% Capacity	10.57	7.45	11.5	11.0	7.8	0.0	98%	2.39	5.49	4.11	1.62	3.9			75	10.59	6.93	9.4	13.5	8.9	-22.7	-23.0	-14.1
5g	IEC+RTU	IEER 25% Capacity	12.45	8.32	11.2	13.3	8.9	0.0	121%	1.64	6.65	5.31	1.67	2.6			73	11.17	7.41	9.1	14.7	9.8	-23.3	-10.7	-9.9
5h	IEC+RTU	Western Annual	9.57	6.62	12.7	9.0	6.2	13.8	107%	5.87	5.35	3.30	1.64	10.4			75	6.51	2.78	11.3	6.9	2.9	-12.5	23.5	52.8

			Combined System Metrics						IEC Metrics						Pre-Cooler		Baseline Comparison - Same % OA								
			Total	Sens	Power	EER -T	EER-S	MUW	EE	OA Cap	RA C-t	RA C-s	Power	Evap	Power	Evap	%	Total	Sensible	Power	EER -T	EER-S	% Savings		
Test #	Description	Climate	(Tons)	(Tons)	(kW)	(Btu/Wh)	(Btu/Wh)	(GPH)	(%)	(Tons)	(Tons)	(Tons)	(kW)	(GPH)	(W)	(GPH)	OA	(Tons)	(Tons)	(kW)	(Btu/Wh)	(Btu/Wh)	Demand	Total E	Sens E
6a	IEC+RTU+PC	Warm Dry	9.48	6.60	10.1	11.3	7.8	29.1	127%	2.01	2.29	1.96	0.32	3.4	38	7.3	36	7.74	5.23	10.4	8.9	6.0	3.1	21.0	23.1
6b	IEC+RTU+PC	Western Peak	6.08	4.93	11.3	6.5	5.2	30.5	116%	3.38	-0.22	0.77	0.33	6.1	38	16.1	32	4.21	2.85	12.7	4.0	2.7	11.1	38.4	48.6
6c	IEC+RTU+PC	AHRI 100% Capacity	5.63	5.14	11.3	6.0	5.5	66.6	119%	2.18	-1.34	0.65	0.32	3.9	38	11.0	31	4.51	3.80	12.0	4.5	3.8	6.1	24.7	30.5
6d	IEC+RTU+PC	IEER 75% Capacity	7.39	5.88	10.5	8.4	6.7	8.3	112%	1.76	1.00	1.35	0.32	3.1	38	9.5	35	6.87	4.94	10.8	7.6	5.5	2.5	9.4	18.0
6e	IEC+RTU+PC	Warm Humid	6.27	5.19	11.0	6.8	5.7	5.3	131%	1.15	-1.82	0.77	0.31	2.0	38	3.9	34	5.41	4.96	11.2	5.8	5.3	1.4	15.0	5.8
6f	IEC+RTU+PC	IEER 50% Capacity	9.74	6.83	9.8	11.9	8.3	13.2	110%	1.24	2.78	2.06	0.31	2.1	38	5.5	36	9.19	6.10	9.7	11.4	7.6	-2.0	3.8	8.9
6g	IEC+RTU+PC	IEER 25% Capacity	10.78	7.21	9.6	13.4	9.0	25.8	132%	1.01	3.38	2.59	0.31	1.6	38	3.6	36	9.68	6.41	9.4	12.4	8.2	-2.9	7.6	8.6
6h	IEC+RTU+PC	Western Annual	8.85	6.19	10.4	10.2	7.1	14.6	118%	2.81	2.10	1.62	0.31	4.9	38	12.9	35	6.44	4.08	11.3	6.8	4.3	8.0	33.1	39.4
7a	IEC+RTU+PC	Warm Dry	10.48	7.26	11.5	10.9	7.6	7.6	117%	3.73	3.86	3.67	1.66	6.5	38	6.3	75	7.64	5.24	10.5	8.8	6.0	-10.3	19.6	20.4
7b	IEC+RTU+PC	Western Peak	4.66	4.44	13.1	4.3	4.1	32.0	104%	5.97	-1.54	0.89	1.71	10.8	38	16.0	71	1.29	-0.28	13.4	1.2	-0.3	2.2	73.0	#N/A
7c	IEC+RTU+PC	AHRI 100% Capacity	4.23	4.63	13.0	3.9	4.3	23.8	109%	4.48	-3.42	1.08	1.69	8.2	38	11.3	71	1.50	1.76	12.7	1.4	1.7	-2.6	63.5	60.9
7d	IEC+RTU+PC	IEER 75% Capacity	7.26	6.07	12.0	7.2	6.1	9.4	101%	3.33	1.70	2.53	1.65	5.9	38	8.4	76	6.24	4.62	10.9	6.9	5.1	-10.2	5.4	16.1
7e	IEC+RTU+PC	Warm Humid	5.03	4.89	12.7	4.8	4.6	5.3	121%	2.23	-4.24	1.49	1.68	4.0	38	3.8	72	2.94	4.45	11.7	3.0	4.6	-9.0	36.3	1.0
7f	IEC+RTU+PC	IEER 50% Capacity	10.45	7.41	11.2	11.2	7.9	4.3	91%	2.12	5.15	3.89	1.61	3.5	38	4.9	75	10.53	6.97	9.4	13.4	8.9	-19.5	-20.4	-12.4
7g	IEC+RTU+PC	IEER 25% Capacity	12.99	8.55	10.9	14.3	9.4	11.7	129%	1.71	6.70	5.38	1.67	2.7	38	2.9	75	11.22	7.46	9.1	14.8	9.8	-19.7	-3.4	-4.4
7h	IEC+RTU+PC	Western Annual	8.72	6.24	12.1	8.6	6.2	11.3	107%	4.97	2.30	2.56	1.68	8.8	38	11.7	75	4.98	2.86	11.6	5.1	2.9	-4.1	40.6	52.4
8a	IEC+1C+PC	Warm Dry	5.06	3.64	6.3	9.6	6.9	9.0	121%	3.20	2.99	3.14	1.66	5.7	38	6.2	100	7.47	5.27	10.5	8.5	6.0	40.0	11.6	13.0
8b	IEC+1C+PC	Western Peak	1.67	2.08	7.1	2.8	3.5	28.7	107%	5.12	-1.21	0.93	1.69	9.4	38	16.1	100	4.23	0.66	13.9	3.6	0.6	49.4	-28.1	83.9
8c	IEC+1C+PC	AHRI 100% Capacity	1.38	2.11	7.0	2.4	3.6	10.2	111%	3.65	-2.79	1.08	1.72	6.6	38	10.2	100	4.37	2.23	13.2	4.0	2.0	46.7	-68.4	43.6
8d	IEC+1C+PC	IEER 75% Capacity	3.62	3.02	6.6	6.6	5.5	10.8	100%	2.70	1.62	2.25	1.65	4.9	38	10.1	100	6.87	4.86	11.0	7.5	5.3	40.3	-13.4	3.9
8e	IEC+1C+PC	Warm Humid	1.83	2.32	6.9	3.2	4.0	14.9	126%	1.95	-3.62	1.36	1.69	3.6	38	3.3	100	5.24	4.41	12.1	5.2	4.4	42.6	-64.4	-9.0
8g	IEC+1C+PC	IEER 25% Capacity	5.97	4.09	5.9	12.1	8.3	3.2	141%	1.67	4.82	4.31	1.63	2.9	38	2.2	100	9.35	6.37	9.0	12.4	8.5	34.2	-3.0	-2.3
8h	IEC+1C+PC	Western Annual	4.54	3.30	6.5	8.4	6.1	36.9	108%	4.43	3.34	2.60	1.68	7.9	37	13.1	100	6.38	3.67	11.5	6.6	3.8	44.1	21.4	37.8
9a	RTU+PC	Warm Dry	8.44	5.97	9.9	10.2	7.2	10.6							38	7.8	32	7.85	5.22	10.4	9.0	6.0	5.0	11.7	17.0
9f	RTU+PC	IEER 50% Capacity	9.35	6.45	9.6	11.7	8.1	4.9							38	5.55	27	9.02	5.96	9.7	11.2	7.4	1.1	4.6	8.6
9g	RTU+PC	IEER 25% Capacity	9.92	6.69	9.4	12.7	8.5	6.0							38	3.93	24	9.23	6.11	9.5	11.7	7.7	0.5	7.4	9.2
9h	RTU+PC	Western Annual	7.16	5.27	10.4	8.3	6.1	18.2							38	11.92	22	6.47	4.48	11.3	6.8	4.7	8.2	17.1	21.9
10a	RTU+PC	Western Peak	7.87	5.28	11.1	8.5	5.7	0.0							37	17.02	0	6.78	4.92	12.5	6.5	4.7	11.5	23.7	17.4
10b	RTU+PC	AHRI 100% Capacity	7.79	5.33	11.1	8.4	5.8	0.0							38	6.91	0	7.23	5.07	11.9	7.3	5.1	6.7	13.4	11.2
10c	RTU+PC	Warm Humid	7.93	5.39	10.9	8.7	5.9	0.0							38	4.38	0	7.82	5.29	11.1	8.4	5.7	1.9	3.3	3.6
10d	RTU+PC	Western Annual	8.45	5.54	10.5	9.6	6.3	0.0							38	14.25	0	7.44	5.15	11.6	7.7	5.3	9.2	20.1	15.7
11a	RTU Baseline	Western Peak	4.44	4.15	12.4	4.3	4.0										17	5.34	3.78	12.3	5.2	3.7	-0.6	-21.0	8.3
11b	RTU Baseline	AHRI 100% Capacity	5.70	4.73	11.7	5.8	4.8										17	6.13	4.32	11.7	6.3	4.4	-0.8	-8.4	8.1
11c	RTU Baseline	Western Annual	6.34	5.04	11.4	6.7	5.3										17	6.53	4.58	11.3	6.9	4.9	-0.9	-3.9	8.2
11d	RTU Baseline	Warm Humid	7.28	5.51	10.9	8.0	6.1										15	7.21	5.04	10.8	8.0	5.6	-1.3	-0.3	7.2
11e	RTU Baseline		8.14	6.15	10.6	9.2	7.0										19	7.76	5.45	10.3	9.1	6.4	-3.0	1.7	8.8
11f	RTU Baseline		8.05	6.14	10.5	9.2	7.0										18	7.74	5.60	10.2	9.1	6.6	-3.3	0.8	5.9
11g	RTU Baseline		9.98	7.03	9.9	12.2	8.6										18	9.22	6.09	9.4	11.7	7.7	-4.4	3.5	9.5
12a	RTU Baseline	Western Peak	6.55	4.75	12.5	6.3	4.6										0	6.79	4.92	12.5	6.5	4.7	0.0	0.0	0.0
12b	RTU Baseline	AHRI 100% Capacity	7.22	5.06	11.9	7.3	5.1										0	7.23	5.07	11.9	7.3	5.1	0.0	0.0	0.0
12c	RTU Baseline	Western Annual	7.54	5.19	11.6	7.8	5.4										0	7.54	5.19	11.6	7.8	5.4	0.0	0.0	0.0
12d	RTU Baseline	Warm Humid	8.02	5.43	11.2	8.6	5.8										0	7.82	5.29	11.1	8.4	5.7	0.0	0.0	0.0
	RTU Baseline	IEER 75% Capacity	8.03	5.42	11.2	8.6	5.8										0	8.03	5.42	11.2	8.6	5.8	0.0	0.0	0.0
	RTU Baseline	Warm Dry	8.22	5.51	11.0	9.0	6.0										0	8.22	5.51	11.0	9.0	6.0	0.0	0.0	0.0
12e	RTU Baseline		8.38	5.60	10.8	9.3	6.2										0	8.38	5.60	10.8	9.3	6.2	0.0	0.0	0.0
12f	RTU Baseline		8.48	5.65	10.7	9.5	6.4										0	8.48	5.65	10.7	9.5	6.4	0.0	0.0	0.0
	RTU Baseline	IEER 50% Capacity	8.73	5.80	10.4	10.1	6.7										0	8.73	5.80	10.4	10.1	6.7	0.0	0.0	0.0
12d	RTU Baseline	IEER 25% Capacity	8.02	5.43	11.2	8.6	5.8										0	7.82	5.29	11.1	8.4	5.7	0.0	0.0	0.0
12g	RTU Baseline		8.92	5.92	10.2	10.5	7.0										0	8.92	5.92	10.2	10.5	7.0	0.0	0.0	0.0