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Creating Energy Independence

Improving Best Air Conditioner Efficiency by 20-30% through a High Efficiency Fan and Diffuser Stage Coupled with an Evaporative Condenser Pre-Cooler

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Acronyms

AC – Air conditioning

AHRI – Air-Conditioning, Heating, and Refrigeration Institute

ARI – Air-Conditioning and Refrigeration Institute

EER – Energy efficiency ratio

FSEC – Florida Solar Energy Center

HVAC – Heating, ventilation, and air conditioning

MHlab – Manufactured housing lab

NREL – National Renewable Energy Laboratory

OEM – Original equipment manufacturer

PV – Photovoltaic

PVC – Polyvinyl chloride

SEER – Seasonal energy efficiency ratio

Executive Summary

The Florida Solar Energy Center (FSEC) conducted a research project to improve the best residential air conditioner condenser technology currently available on the market by retrofitting a commercially-available unit with both a high efficiency fan system and an evaporative pre-cooler. The objective was to integrate these two concepts to achieve an ultra-efficient residential air conditioner design. The project produced a working prototype that was 30% more efficient compared to the best currently-available technologies; the peak the energy efficiency ratio (EER) was improved by 41%. Efficiency at the Air-Conditioning and Refrigeration Institute (ARI) standard B-condition which is used to estimate seasonal energy efficiency ratio (SEER), was raised from a nominal 21 Btu/Wh to 32 Btu/Wh

Background/Prior Work

Some 87% of homes in the United States have AC (Residential Energy Consumption Survey 2009). It remains the most energy intensive use of electricity for homes that have mechanical space cooling and is the largest factor accounting for utility peak electrical generation demand during summer.

Air conditioner energy efficiency is measured as SEER and peak consumption as EER, both of which have units of cooling Btu's per watt hour. Minimum air conditioner efficiency for manufacture and sale in the United States is now 13 Btu/Wh (Central Air Conditioning 2012). To date, the best AC condenser technology utilizes inverter-driven rotary compressors and has efficiencies around 21 Btu/Wh. In the United States, this type of system is marketed under the IQ Drive brand name by Nordyne Corporation.

This study involved the acquisition and modification of two such units. One unit was installed in an environmental test laboratory for baseline performance testing and prototype design, while the second unit was used for field testing.

High Efficiency Condenser Fans

Air-cooled condensers in residential AC systems employ finned-tube construction to transfer heat from the refrigerant to the outdoor air. Electrically-powered fans draw air across the outdoor condensing coils to remove refrigerant heat. Methods to reduce air source system fan power and the condenser inlet air temperature offer potential methods to improve system operating efficiencies for very efficient homes.

FSEC has done significant work in both the design of high efficiency fans and exhaust diffusers for residential air conditioners. This research has shown the ability to reduce air-cooled system fan energy by more than 30% through the use of optimized fans with pressure recovery in the updraft section of the fan stage (Parker and Sherwin 2005). Figure 1 shows the original prototype fan assembly and schematic diagram.

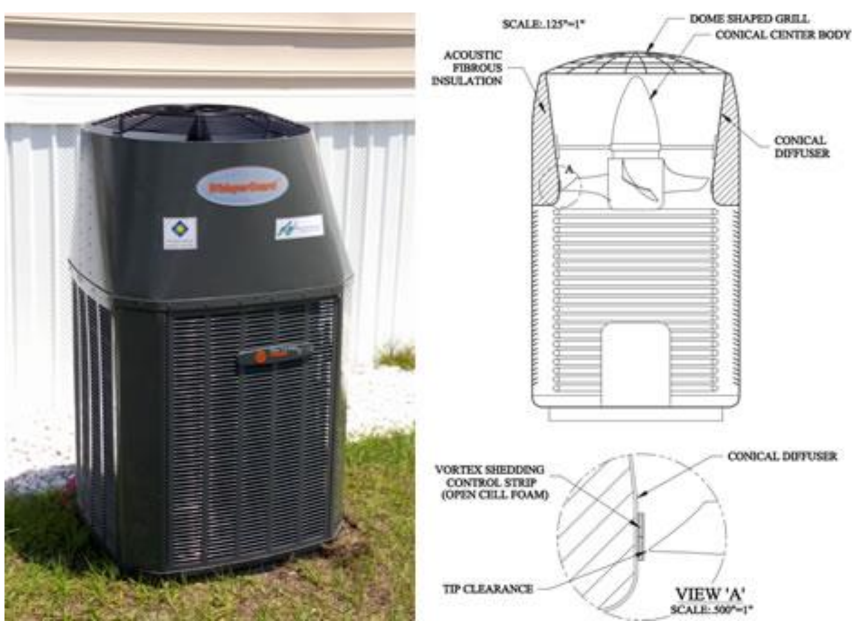

Figure 1. Developed high efficiency condenser fan and assembly

From 2003-2005, FSEC tested potential enhancements to the outdoor unit condenser fan by altering its shape and aerodynamic characteristics. Optimized fan blades were designed via a numerical flow simulation and fabricated using stereo lithography. After several months of testing, the research produced a fan exhibiting greatly superior air moving efficiency compared with conventional stamped metal blades. The evaluation was performed on a standard 3-ton, straight-cool AC condenser. Measurements were made of condenser air flow, motor power, sound levels, and condenser cabinet pressures. The developed prototype fan substituted on the original condenser reduced condenser fan electric power by 25% (48 watts) with slightly higher condenser air flow. Air moving efficiency (cfm/watt) was increased by 35%. The technology is patented.

In the summer of 2008, a field test was performed by substituting the improved diffuser and fan system top onto an existing condensing unit at FSEC's manufactured housing lab (MHLab). This facility is a conventional three bedroom, two bath manufactured home located on the FSEC campus. Nearly one year of baseline performance data on the original condensing unit was available prior to the retrofit. After the change, comparison to the previous baseline data indicated a 70 to 100-watt drop in the condenser fan motor assembly power (*original blade, standard motor, standard top*: -16.2 Pa cavity pressure (average), 238 Volts, 0.8 Amps = 190 watts; *5-bladed efficient fan, electronically-commutated motor, elongated diffuser*: -16.0 Pa pressure (average), 238 Volts, 0.4-0.5 Amps = 95 - 120 watts). This reduction was consistent with what was measured in the lab previously. Long term data on the MHLab change out verified the maximum condenser fan power to be about 70 watts lower than previously measured. In addition to lower fan motor power, condenser air flow was measured to be slightly greater. The data in Figure 2 shows a difference in pre- and post-retrofit power due to the condenser fan change out.

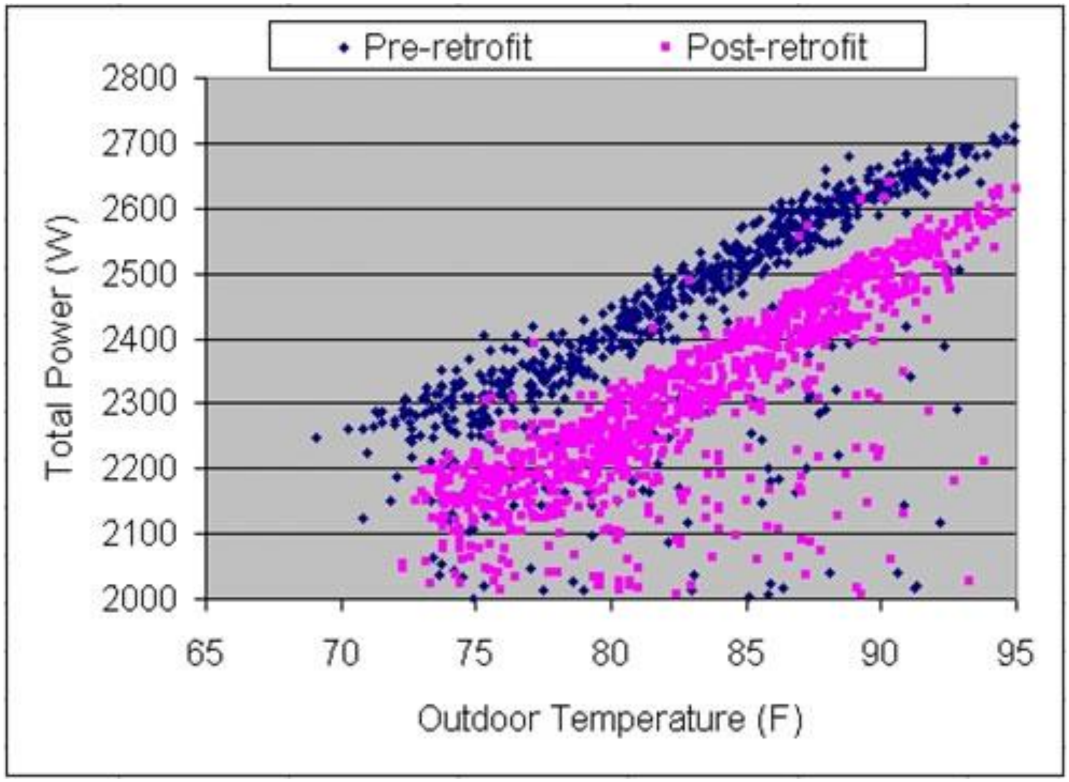

Figure 2. 15-minute AC power demand pre and post fan retrofit.

The data indicates approximately a 70-watt reduction at low outdoor air temperature and a 100-watt difference at high outdoor air temperature. The increased air flow resulted in a corresponding decrease in condenser head pressures, which impacted compressor power use. In any case, there was approximately a 4% reduction to total AC system power through use of the improved fan system alone.

Evaporatively Pre-cooled Condensers

The second element proposed to achieve greater efficiencies was the addition of an evaporative pre-cooler stage for the condenser. Evaporatively-cooled condensers have long been used in larger commercial AC systems to reduce effective condensing temperatures and improve

AC efficiency. Commercial air conditioners larger than about 150 tons typically rely on cooling towers to cool water and remove heat generated from the compression cycle of the chiller.

While it has long been known that an indirect, evaporatively-cooled condenser system can improve condenser performance, there are a host of issues associated with long term reliability that has precluded their use with residential systems. These include condenser fouling, water system and pump/valve reliability, potential for water biological growth, freeze protection, and water consumption.

Even in hot and humid climates, evaporative pre-coolers can provide large performance advantages. Goswami, Mathur, and Kulkarni (1993) at the University of Florida did a careful and detailed evaluation of the impact of an indirect evaporative pre-cooler for the condenser. The study found about a 20% reduction in overall energy use in realistic testing in Florida's difficult climate. Air was pre-cooled by an average of 14°F before reaching the coil, with about 8.7 gallons per hour of water used by the evaporative media. The media pads used added 0.025 inches of water column to the static pressure faced by the fan. Another assessment with a different arrangement in Maryland showed a 15% improvement to SEER from an evaporatively-cooled condenser (Hwang et al. 2000). Based on a psychometric evaluation, we would expect that the potential savings in drier climates with much lower dew points are perhaps 30% with even larger peak reductions.

In an effort to demonstrate the potential of an evaporative pre-cooler stage, FSEC conducted a short term experiment in the summer of 2008. The setup was very crude, consisting of misting nozzles and double weave insect screening draped over a condensing unit. The system power use was sub-metered, and the AC return and supply temperatures were monitored.

The infrared images in Figure 3 show the impact of no evaporative cooling versus evaporative cooling. Although the misting nozzles were not properly spaced, the images show depict substantially cooler condenser coil temperatures as evidenced by the absence of yellow and green shading in the evaporatively cooled section on the right hand side.

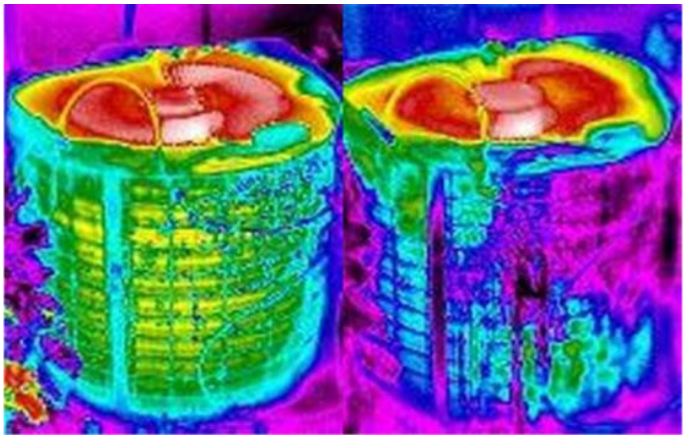

Figure 3. Infrared thermographs of AC condenser without (left)and with evaporative pre-cooler (right). Color is proportional to temperature.

Although the test configuration was impractical (problems with water use, distribution, valving, and scaling), the demonstration revealed a 7% drop in measured air conditioner power (150-200 watts at 91^{o}F with a coincident 74^{o}F dew point) and a 13% improvement in sensible cooling capacity (supply temperature dropped by 1.5^{o}F). This suggested a 20% overall improvement in air conditioner efficiency in Florida's August climate. Lower evaporator temperatures also led to enhanced dehumidification performance.

Evaporative Cooling Performance During Peak Demand Conditions

Evaporative pre-coolers offer the potential to substantially improve the summer peak design performance of air conditioners. This is ideal both for utilities concerned with summer peak demand as well as for zero energy buildings with a need for very high efficiencies. Peak design day weather data shows that such a scheme will provide most performance improvement during the hottest weather either in dry or humid climates. For instance, with a 95°F air temperature in a humid climate with a dew point around 74°F, the maximum achievable wet bulb temperature to the condenser from evaporative cooling would be about 80°F – a change of 15 degrees. However, at 3 AM, with the same dew point temperature and an 80°F dry bulb temperature, the temperature depression to the condenser would be only about 4 degrees.

Conversely, in a hot dry climate, similar to what might be seen in much of the arid southwest, a 105°F outdoor dry-bulb temperature with a 55°F dew point temperature would provide a 72°F wet-bulb point temperature to the condenser – a 33°F drop in the heat rejection temperature. An 80°F nighttime temperature at 3 AM with the same low dew point would produce a 65°F wet-bulb temperature for heat rejection. Thus, in both humid climates and dry climates, evaporatively-cooled condensers offer the potential to strongly reduce AC power and improve cooling capacity during a utilities' on-peak period.

Commercial Systems and Limitations

Within unitary-sized systems, several attempts have been made to commercialize evaporative-cooled systems – the AC2 system in the 1990s and more recently the Freus AC system. In California's drier climate, energy savings of 36-40% were demonstrated relative to SEER 10 equipment. Average seasonal EERs of 14-15 Btu/Wh were observed (Davis Energy Group 1998). An evaluation by Proctor Engineering Group of the AC2 system showed somewhat more modest savings and some potential problems with scaling of the condenser heat exchanger surfaces with the AC2 system. This work has shown up to 35% reduction in measured space cooling energy for a given cooling capacity in dry climates (Eberling 1998).

More recently, the AC2 system has been further developed into the *Freus* air conditioner. However, initial testing done by the National Renewable Energy Laboratory (NREL) and Davis

Energy Group has shown only modestly better performance from this system compared with more modern air-cooled equipment (Steven Winters and Associates 2006). Part of the missing advantage has been the lack of lower energy use fan and blower motors in the Freus system compared with the air-cooled systems and the very large condenser surface area in the newer high-SEER units, which use two-speed operation to achieve their very low energy use in low-speed operation (Kutchner, Eastment, Hancock, and Reeves 2006). As of 2013, the Freus unit is no longer manufactured.

Specific Design Objectives

Development of a working prototype to achieve the project objective of increasing the performance of the best available heat pump AC system by 20-30% required the following:

1) Baseline testing of a commercially-available high performance heat pump system

The best performing split system heat pump system on the market is the *Westinghouse IQ Drive SEER 24* system using inverter-controlled, variable-frequency, rotary-driven compressors (Westinghouse up to 25.5 SEER iQ Drive® Air Conditioner 2014). The unit is manufactured by Nordyne Corporation and licensed to Westinghouse and other select brand names. Using these systems and providing lower day and nighttime condenser inlet temperatures should result in extremely low-power consumption as well as ultra-quiet operation with the media pads in place. The objective was to achieve up to SEER 30 with the system being proposed for research. Baseline testing would be used to verify the manufacturer’s performance data and document any improvements achieved as a result of this study.

2) Implement an advanced fan and diffuser assembly to maintain/augment air flow

As discussed earlier, FSEC demonstrated the increased energy performance of an advanced fan and diffuser technology compared to original equipment manufacturer (OEM) configurations on standard AC condensing units in previous research efforts. The primary objective in utilizing that combination on this project was to realize the increased performance on the IQ drive system while meeting manufacturers' original condenser air flow rates with the addition of evaporative media. A series of diffuser geometries and fan blade configurations were tested on the IQ drive and a best performance combination selected.

3) Choose known evaporative cooling system from commercial sector.

The use of evaporative cooling technologies in commercial/industrial settings is well established. Issues with condenser fouling and system reliability have been barriers to the use of evaporative systems in the residential market. It was proposed to adapt commercial/industrial evaporative cooling technologies to the residential prototype. Munters Corporation was identified as a potential partner and resource to achieve this objective. Through their Humidicool Division, this company has many years of experience tailored to developing and adapting evaporative pre-cooler condensers for commercial applications.

FSEC Environmental Chambers for Equipment Testing

FSEC's on-site environmental facilities are capable of testing air conditioners and heat pumps with cooling/heating capacities up to 3.5 tons. The facility is comprised of an indoor chamber, an outdoor chamber, and a computerized control room. The environmental chamber's indoor and outdoor conditions are maintained automatically with a laboratory-grade data acquisition and control system. Full automation allows complete flexibility for parametric testing of the air conditioner according to the prescribed test procedure for a particular test.

The indoor environment contains the indoor air handler section of the air conditioner. Proportional/integral/derivative (PID) feedback, control loops are used to maintain indoor chamber temperature and humidity using electric heaters and a steam humidifier. The outdoor chamber houses the condenser section of the heat pump. The outdoor chamber temperature is controlled using a variable-speed, chilled-water system supplied by a 9-ton chilled-water unit. Two commercial refrigeration units augment the chilled-water system and allow for winter testing of heat pumps down to 40°F.

The control room houses a data acquisition and control system. It is programmed to monitor instrumentation output and control psychometric chamber temperature and humidity conditions. The test chamber/apparatus is shown schematically in Figure 4.

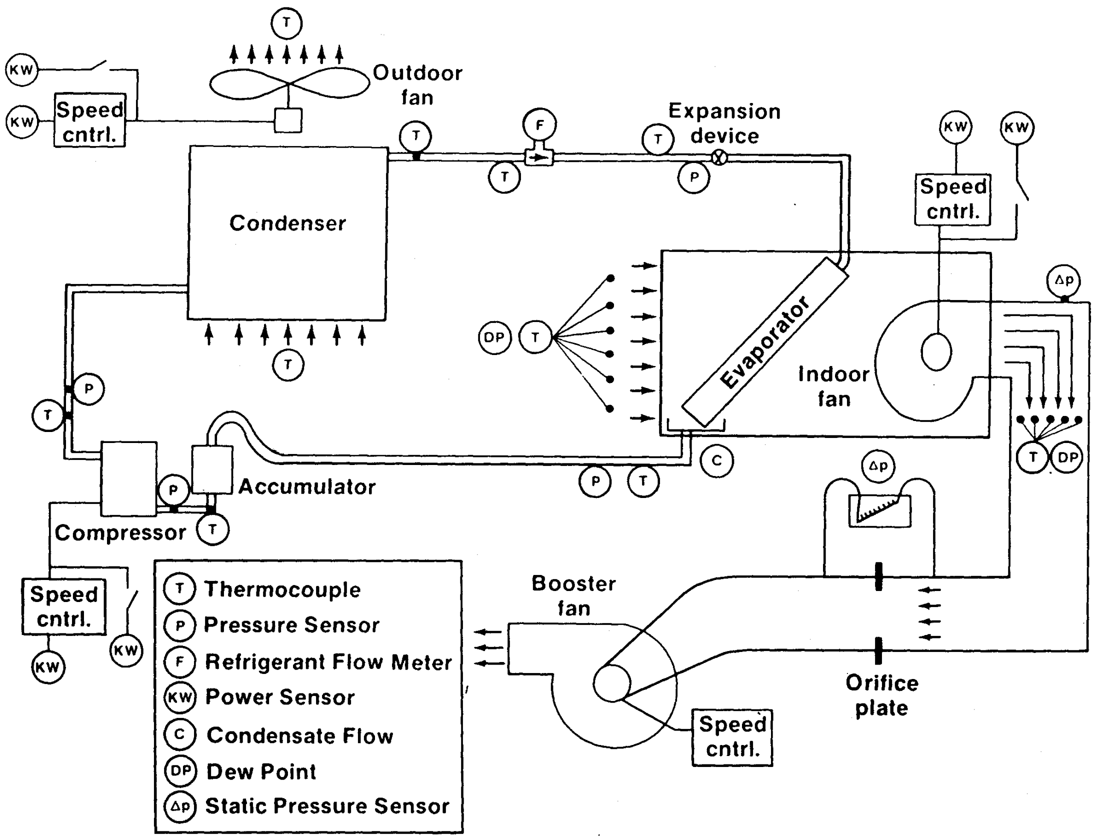

Figure 4. Schematic of psychrometric chambers/air conditioner testing apparatus at FSEC

Prototype Development

As previously stated, the prototype development consisted of adding a diffuser and fan assembly along with an evaporative pre-cooling stage to an existing heat pump system. Producing a high performance prototype utilizing readily available parts and components was paramount to this objective. While the diffuser was manufactured from raw materials, both the fan and evaporative cooling section were obtained from commercial sources. The task at hand became evaluating the various combinations of those components for optimal performance.

Figures 5 and 6 depict the original OEM fan (upper left metal-stamped blade) along with other configurations being readied for evaluation.

Figure 5. Fans in evaluation in FSEC machine shop

Figure 6. Fabrication of 7-degree divergent diffuser

Extensive testing was conducted on alternative fan and diffuser designs. Four separate fans were obtained from Multi-Wing Corporation, and two divergent diffuser designs were evaluated. A specific fan design and diffuser was settled upon after extensive testing of several configurations. The superior configuration was a 9-degree divergent diffuser with a 4-bladed, 40-degree pitch fan from Multi-Wing Corporation (Figure 7). A cone-shaped fairing located after the fan was determined to be slightly superior to an annular design. Figure 8 shows the selected fan in the chosen system diffuser.

Figure 7. Nine-degree divergent diffuser under test

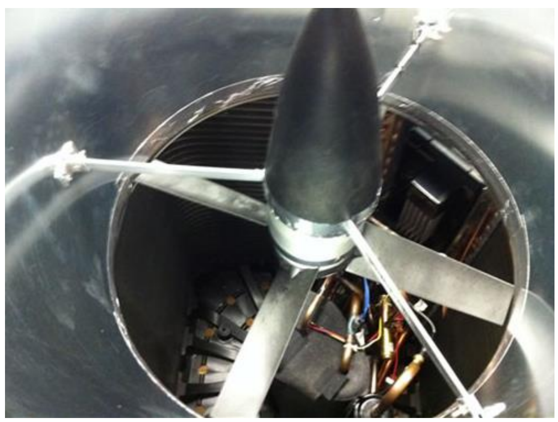

Figure 8. 40-degree pitch propeller with 9-degree divergent diffuser and conical center after body

Evaporative Pre-Cooler Design

The evaporative pre-cooler design was comprised of three parts: media selection, water delivery method, and water resource management. Two pre-cooler designs were evaluated throughout the course of the project. These were labeled Phase I and Phase II prototypes with the latter being the final tested unit and suitable for use in the field study (Figure 9).

Figure 9. Phase II prototype

The media pad selected was the CELdek® brand pad from Munters Corporation (Munters EPCC Products for Commercial Condensers 2013). The pads are composed of a specialty-grade paper treated with a resin coating to inhibit mold growth. Various thicknesses were evaluated to find the best evaporative performance for the fan flows experienced while not appreciably increasing static pressures within the condenser cavity.

As shown in Figure 10, testing was performed to examine the temperature drop of the air as it passed through the wet media. Air flow rates were adjusted to approximate 140 fpm to replicate the velocities seen in the heating, ventilation, and air conditioning (HVAC) laboratory on the OEM IQ drive system.

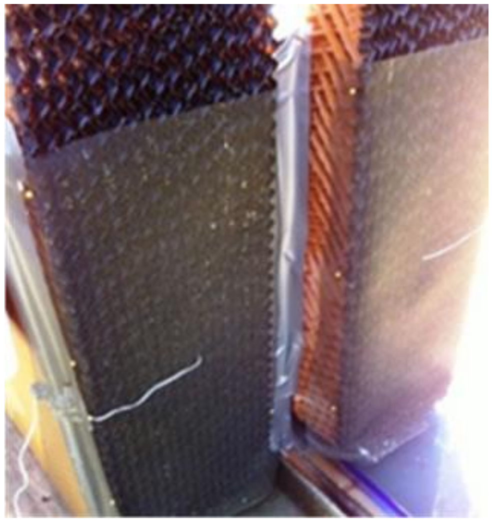

Figure 10. Simultaneous outdoor testing of two differing thicknesses of evaporative media pads

These tests showed that at the minimum condenser fan rotational speed, the air temperature dropped by approximately 3.5°F before being introduced to the condenser coil at an 82°F outdoor condition. Considering the chamber dry and wet-bulb temperatures, this represents a 36% cooling effectiveness for the evaporative media. Evaluating the catalog data from Munters as well as a series of tabletop experiments of the two media depths, it was determined that improvements in cooling effectiveness of up to 55% could be obtained by changing to a design with 4-6” evaporative media. This was anticipated to produce temperature drops of 2-3°F further than we have obtained thus far and potentially improve obtained efficiencies even further. Figure 11 shows measured results in a tabletop experiment conducted in April 2012 showing the superior performance of 4” versus 2” pads.

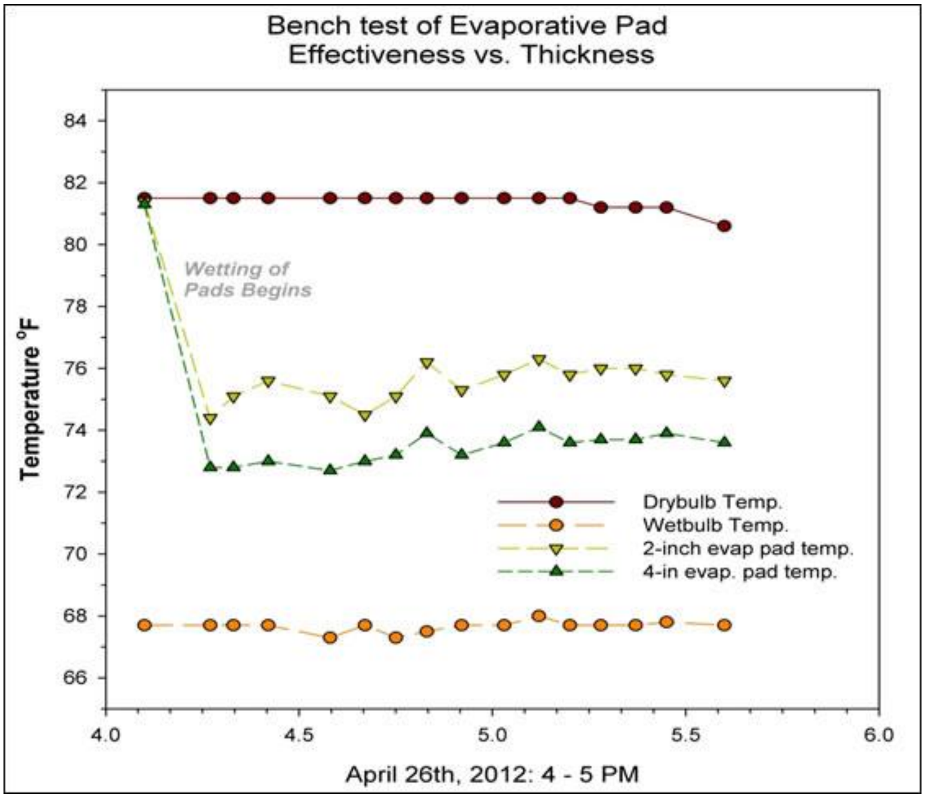

Figure 11. Measured cooling effectiveness of 2" versus 4" evaporative cooling pads conducted on April 26, 2012. Note approximately 2°F cooler temperatures achieved

Based on the test observations, modifications were made for the media pad configuration and the 6" thickness media pad selected (Figure 12).

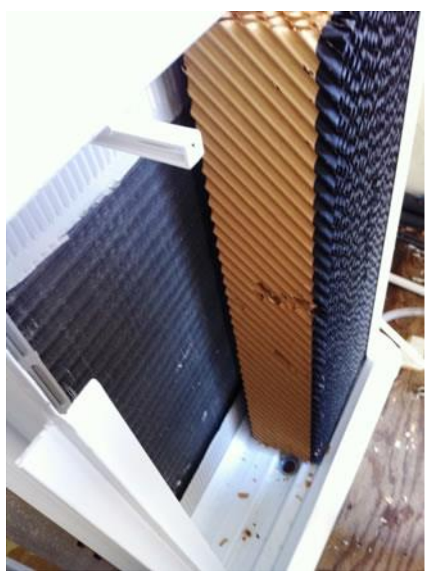

Figure 12. Showing the 6" evaporative media pads being installed in the water collection trough of the Phase II prototype

Water management is an important consideration for evaporative cooling systems. Within this study, media pad wetting and fluid circulation were investigated. Based on testing in early 2012 (Figure 13), the system by which the evaporative pads were wetted had to be altered from a spray-on system with nozzles to a drip system with headers above the tops of the pads. The rationale for the spray system was to take advantage of the evaporation of the atomized water in addition to the moisture deposited on the media pads. Issues with water waste from overspray and the potential of incomplete wetting of the media pads forced this approach to be abandoned. The Phase II prototype employed a drip method through a continuous header positioned over the media pads. Pad wetting was accomplished through direct connection to the laboratory water supply lines; therefore, no pumping energy was included in the performance data.

Figure 13. Preliminary Phase I prototype under test in HVAC laboratory

Baseline Testing

Testing of the unit (Figure 14) in baseline condition was conducted to map the performance of the unaltered heat pump system. Over a hundred tests were conducted showing system sensible and latent cooling as well as machine power at differing operating points.

Figure 14. Nordyne IQ drive AC system prior to start of baseline testing

Specific results for all tested configurations were reproduced, and a sample of such data is shown in Figure 15. It should be noted that under normal conditions, the IQ drive systems operate using a “smart” thermostat that senses load and adjusts accordingly. In order to perform the laboratory tests, the heat pump was placed in a manual, override mode as per the manufacturer’s recommendation. It was tested at three factory-determined speeds: minimum, intermediate, and high speed at four differing indoor conditions: 80°F dry bulb with 62, 67, and 72°F wet-bulb temperatures as well as a test at 75°F indoors at 63°F wet-bulb.

Tests using min/interm, or max speed to measure Nominal Capacity

Speed	Outdoor Temperature			70°F				80°F				90°F		
	Indoor Tdb (F)	Indoor Twb (F)		Total Capacity	Sensible Capacity	kW	EER	Total Capacity	Sensible Capacity	kW	EER	Total Capacity	Sensible Capacity	kW
Minimum	80	62	Manuf	16	16	0.54	29.63	15.1	15.1	0.65	23.23	14.4	14.4	0.78
			Base	16.3	16.3	0.5	32.5	15.5	15.4	0.62	25.02	14.5	14.5	0.76
			Cond											
			Evap											
	80	67	Manuf	16.9	13.6	0.51	33.14	15.8	13.1	0.65	24.31	14.7	12.6	0.78
			Base	17.1	14.4	0.49	34.88	16.				15.1	13.5	0.76
			Cond									14.7	13.5	0.74
			Evap									2.65%	0.00%	2.63
	80	72	Manuf	17.7	11.1	0.5	35.4	16.5	10.7	0.62	26.61	15.5	10.2	0.76
			Base	18.8	10.9	0.46	41.32	17.8	10.5	0.59	30.2	16.6	10.1	0.74
			Cond											
			Evap											
	75	63	Manuf	14.5	13.3	0.56	25.89	13.7	12.9	0.67	20.45	13	12.5	0.8
			Base	16.1	14.1	0.52	31.05	15.3	13.7	0.63	24.26	14.2	13.3	0.77
			Cond	15.8	14	0.51	31.02							
			Evap	1.86%	0.71%	1.92%	0.10%							

Al indications are that Run 2 had 10cfm lower airflow (diff blower spd)

Figure 15. Sample of baseline test data taken in the project

As shown in Figure 16, there was good reproducibility between manufacturer claims and measured performance in the laboratory.

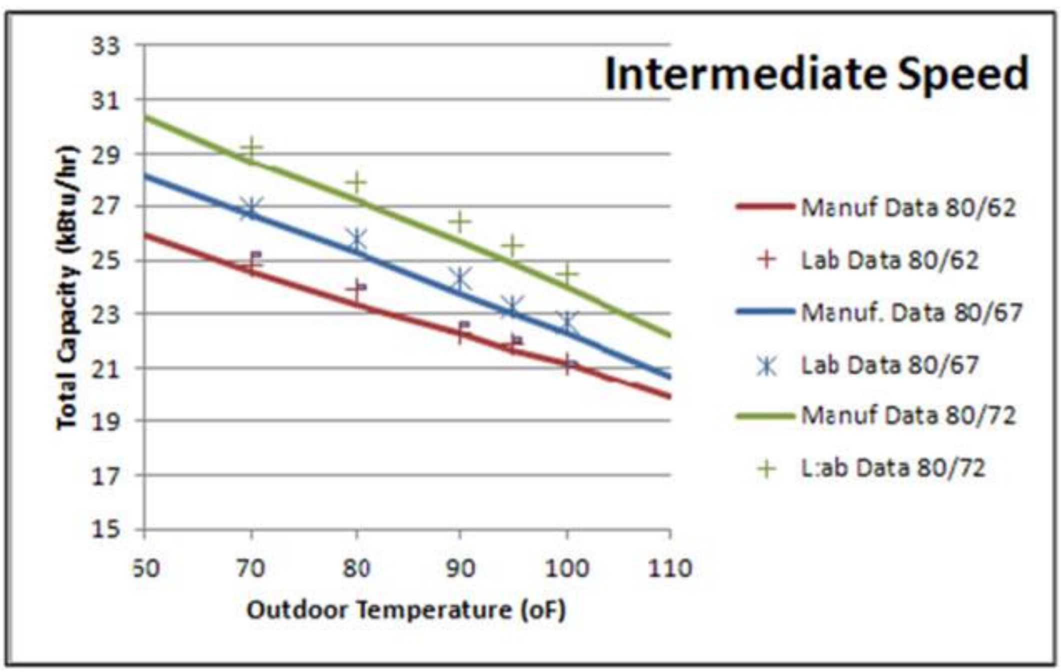

Figure 16. Manufacturer's data for system total cooling capacity against measured results in FSEC HVAC laboratory

The correspondence seen between the manufacturer's data and measured performance provided assurance that the indicated change in performance due to the evaporative cooling system and better condenser fan design could be reliably evaluated.

Laboratory Test of Condenser with Both Improved Fan Assembly and Evaporative Pre-Cooler

The fundamental objective of the project was to show a 20-30% improvement in performance of a best available air conditioner/heat pump through the use of a high efficiency condenser fan and diffuser coupled with an evaporative condenser pre-cooler. The improvement would be demonstrated by documenting the efficiency gains of the prototype over the OEM unit in controlled laboratory testing. All performance testing was accomplished using AHRI Standard 210/240-2008 test conditions.

The laboratory tests of the prototype produced very favorable results. Some two dozen tests were completed. In an environmental chamber, baseline tests with the fan assembly and media pads were performed at outdoor temperatures of 80°F, 82°F, 87°F, and 95°F. Tests were completed with the dry and wetted pads to evaluate the evaporative performance of the system versus the fan system alone. A set of duplicate tests were conducted at 80°F and verified the favorable performance seen in the initial tests. Some difficulty was experienced in controlling the outdoor chamber moisture levels during the experiments, but the chamber dew points were recorded so influences could be evaluated. The higher than desired moisture levels that could be achieved made for a conservative estimate of results.

Table 1 summarizes laboratory testing of the fabricated Phase I unit with a preliminary evaporative cooler with 2" evaporative media and a headered, top-of-pad wetting system.

Table 1. Laboratory Testing of Advanced Evaporatively-Cooled Condenser
January – April 2012

Speed	Case	Total Capacity (kBTU/hr)	Sensible Capacity (kBTU/hr)	kW	EER (Btu/Wh)	Outdoor Air Dew Point (F)	Efficiency Increase %
Outdoor Air 80°F							
MIN	Base	15.8	13.1	0.65	24.31		
	Evap.	17.3	14.4	0.54	31.98	67.66	+32%
	Repeat	16.7	14.3	0.54	30.82	65.74	+27%
	Dry	16.4	14.2	0.57	28.74		
INT	Base	25.3	17.8	1.18	21.44		
	Evap.	26.0	18.6	1.08	24.17	66.51	+13%
	Repeat	27.0	19.7	1.07	25.28	67.54	+18%
	Dry	25.7	19.2	1.18	22.91		
MAX	Base	37.7	24.5	2.23	16.91		
	Evap.	38.3	24.6	1.97	19.41	66.37	+15%
	Repeat	39.1	25.5	1.97	19.80	66.83	+17%
	Dry	39.5	23.2	2.05	19.27		
Outdoor Air 82°F (ARI "B" Condition)							
MIN	Base	15.58	13.0	0.676	23.05		
	Evap.	16.90	14.4	0.553	30.59	65.74	+33%
INT	Base	25.0	18.20	1.240	20.16		
	Evap.	26.4	19.50	1.102	23.97	69.97	+19%
MAX	Base	37.36	24.36	2.29	16.31		
	Evap.	38.30	25.10	2.05	18.61		+14%
Outdoor Air 87°F							
MIN	Base	15.03	12.75	0.741	20.28		
	Evap.	15.90	13.90	0.619	25.78	71.87	27%
INT	Base	24.25	17.95	1.313	18.47		
	Evap.	25.30	18.90	1.174	21.58	71.09	17%
MAX	Base	36.51	24.01	2.44	14.96		
	Evap.	37.20	24.60	2.15	17.37	72.43	15%
Outdoor Air 95°F							
MIN	Base	14.2	12.3	0.85	16.71		
	Evap.	16.5	11.2	0.70	23.65	75.54	+41%
	Base	23.0	17.5	1.48	15.54		
INT	Evap.	24.0	18.2	1.33	18.07	71.66	+16%
	Base	36.0	12.6	2.69	13.38		
	Evap.	36.3	12.1	2.35	15.41	77.05	+15%

Of particular note is the performance of the prototype operated in minimum speed at the 82°F outdoor temperature; at this condition, the unit achieved an EER 30.6 Btu/Wh (80°F indoor condition with a 67°F wet-bulb). This represents the ARI "B" condition that is often used to rate the SEER performance of single-speed AC equipment. Figure 17 shows the test conditions evaluated in the laboratory.

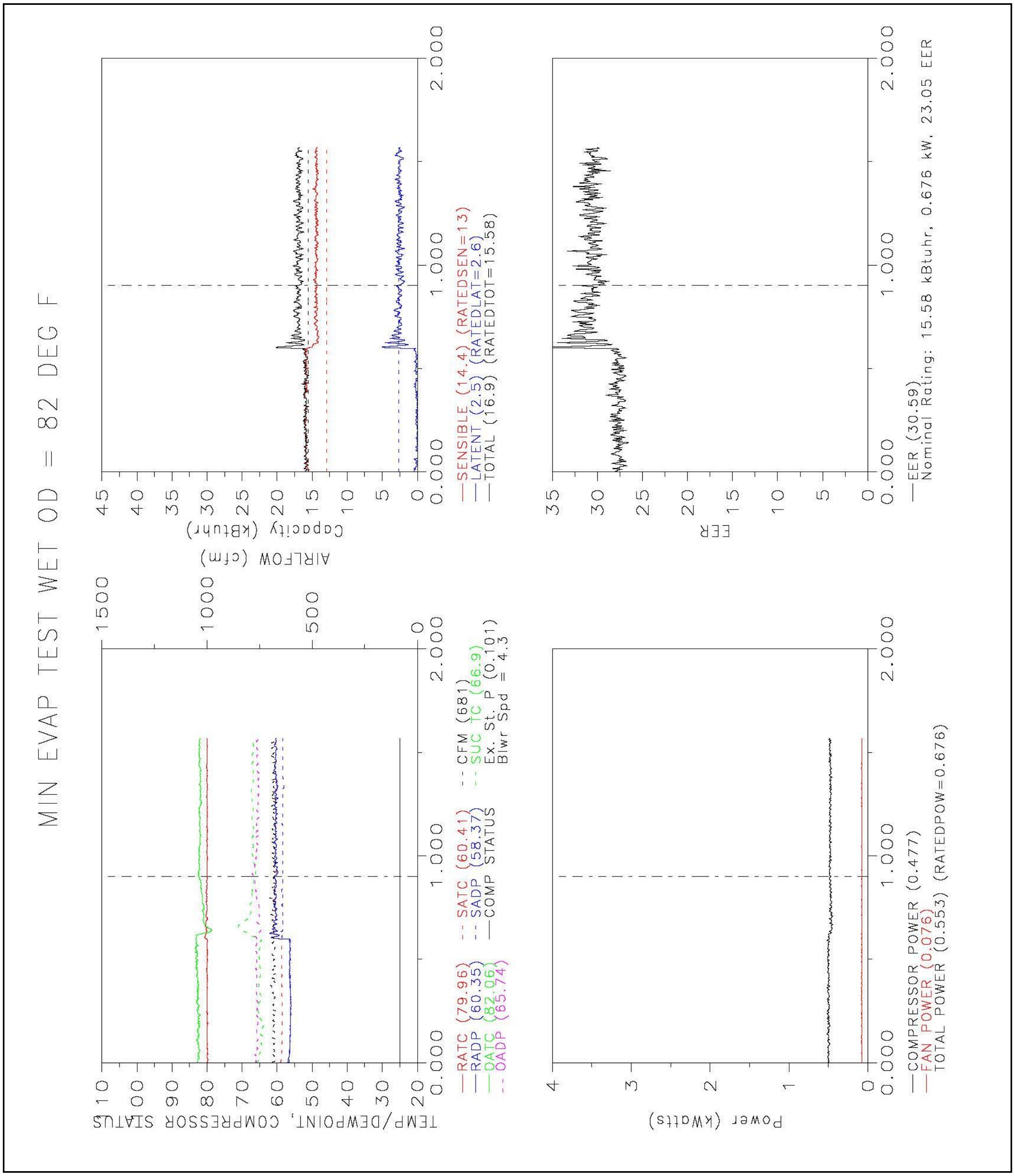

Figure 17. Performance of the Phase I prototype at the 82°F ARI "B" condition

An EER 30.6 Btu/Wh represents a 33% increase in efficiency compared to the standard IQ Drive unit and met the research goal of the project.

Tests at the 80°F outdoor conditions reached an EER at minimum speed of up to 32.0 Btu/Wh, which was a 32% increase in rated performance. At the 95°F outdoor condition, representing the hottest summer weather, performance at minimum speed increased from a baseline EER of 16.7 Btu/Wh to 23.7 – an increase in efficiency of more than 41%. As expected, performance at the higher intermediate and maximum fan speeds were lower, showing improvements in cooling efficiency from 13%-19% above the baseline data.

Testing was also done in a “dry” configuration (media pads not wetted) to evaluate improvements produced by the diffuser, alterations to reduce condenser cabinet pressure drop, and the improved fan assembly. These results, shown in Table 1, indicated that 7%-18% of the measured efficiency increase was from the improved air flow characteristics of the improved condenser fan and diffuser assembly, depending on operating speed. Increases were highest at the lower speeds.

As previously discussed, two prototypes were developed over the course of the laboratory testing. The Phase II prototype represented the unit that would be installed into the field. Conceptually the same, the second unit featured improvements in the evaporative pad wetting system and used 6” media pads. This unit was laboratory tested at the same conditions as the Phase I unit. Table 2 shows the performance of the baseline, Phase I, and Phase II prototypes compared with the manufacturer’s published data. Figure 18 shows the laboratory test results at 82°F outdoor conditions.

Table 2. Laboratory Testing of Advanced Evaporatively-Cooled Condenser January - April 2012

Speed	Case	Total Capacity (kBTU/hr)	Sensible Capacity (kBTU/hr)	kW	EER (Btu/Wh)	Outdoor Air Dew Point (F)	Efficiency Increase%
Outdoor Air 80°F							
MIN	Base	15.8	13.1	0.65	24.31		
	Evap.	17.3	14.4	0.54	31.98	67.66	+32%
	Repeat	16.7	14.3	0.54	30.82	68.74	+33%
	Dry	16.4	14.2	0.57	28.74		
INT	Base	25.3	17.8	1.18	21.44		
	Evap.	26.0	18.6	1.08	24.17	66.51	+13%
	Repeat	27.0	19.7	1.07	25.28	67.54	+18%
	Dry	25.7	19.2	1.18	22.91		
MAX	Base	37.7	24.5	2.23	16.91		
	Evap.	38.3	24.6	1.97	19.41	66.37	+15%
	Repeat	39.1	25.5	1.97	19.80	67.83	+23%
	Dry	39.5	23.2	2.05	19.27		
Outdoor Air 82°F (ARI "B" Condition)							
MIN	Base	15.58	13.0	0.676	23.05		
	Evap.	16.90	14.4	0.553	30.59	65.74	+33%
	Phase II	16.9	14.5	0.525	32.26	69.51	+40%
INT	Base	25.0	18.20	1.240	20.16		
	Evap.	26.4	19.50	1.102	23.97	69.97	+19%
	Phase II	26.2	19.5	1.053	24.92	71.61	+24%
MAX	Base	37.36	24.36	2.29	16.31		
	Evap.	38.30	25.10	2.05	18.61		+14%
	Phase II	38.1	25.0	1.965	19.41	71.61	+19%
Outdoor Air 87°F							
MIN	Base	15.03	12.75	0.741	20.28		
	Evap.	15.90	13.90	0.619	25.78	71.87	27%
INT	Base	24.25	17.95	1.313	18.47		
	Evap.	25.30	18.90	1.174	21.58	71.09	17%
MAX	Base	36.51	24.01	2.44	14.96		
	Evap.	37.20	24.60	2.15	17.37	72.43	15%
Outdoor Air 95°F							
MIN	Base	14.2	12.3	0.85	16.71		
	Evap.	16.5	11.2	0.70	23.65	75.54	+41%
	PhII	15.9	13.6	0.70	22.71	78.63	+36%
INT	Base	23.0	17.5	1.48	15.54		
	Evap.	24.0	18.2	1.33	18.07	71.66	+16%
	PhII	24.6	18.6	1.29	19.01	77.59	+22%
MAX	Base	36.0	12.6	2.69	13.38		
	Evap.	36.3	12.1	2.35	15.41	77.05	+15%
	Phase II	38.8	28.0	2.31	16.78	78.54	+25%

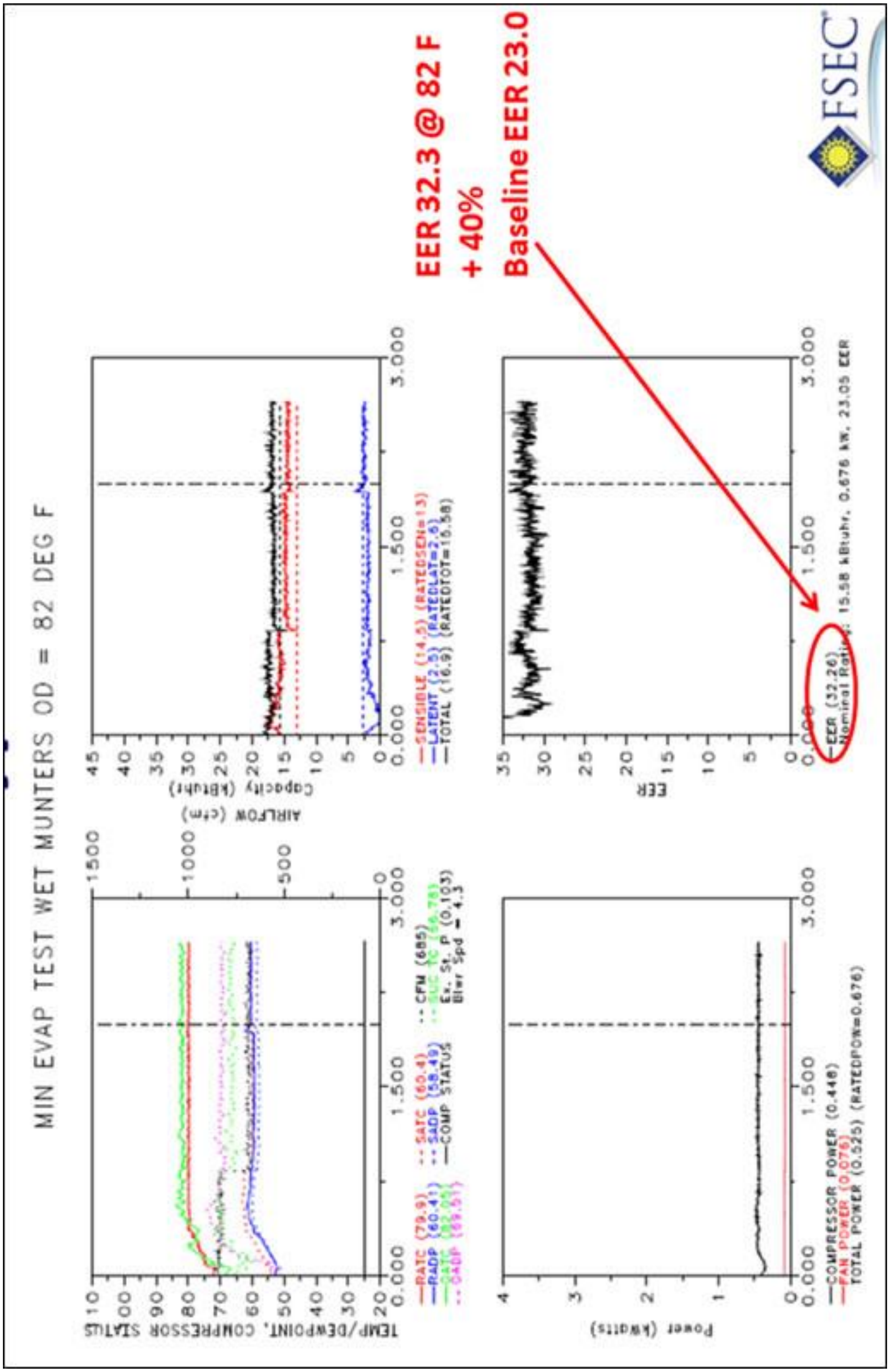

Figure 18. Test of Phase II prototype at minimum speed with 82°F outdoor conditions

Field Monitoring

The Phase II prototype was installed in the field in early 2013 (Figure 19). Project funding was depleted shortly thereafter, which directly impacted the ability to monitor, maintain, and improve the installed system. However, despite this shortcoming, data was obtained that allowed for a limited analysis of the field unit. The data was comprised of interior space conditions, HVAC energy use, evaporative media temperatures, ambient outdoor conditions, and water consumption. The home was monitored during an entire cooling season. In order to gauge the impact of the evaporative pre-cooler, the unit was initially installed without the evaporative section. Midway through the cooling season, the evaporative pre-cooler was installed. The diffuser and divergent cone were not installed in the field test unit.

Figure 19. Field Installed Phase II prototype

At its core, the prototype unit is the IQ Drive OEM unit with the addition of an advanced fan stage and evaporative pre-cooler. When installed in a typical residential setting, as was done with the field unit, the prototype operates in accordance with the control strategies set by the manufacturer or, more specifically, by the logic programmed into the IQ Drive thermostat. This operational mode is different than the manual mode used in the laboratory testing. Manual mode is not plausible in the field setting, as it forces the unit to operate uninterrupted and does not respond to temperature set points. The normal operation of the IQ Drive is to continuously adjust system capacity to meet load. The dynamic operational nature of the field unit made direct comparisons to laboratory data problematic. In any event, the overall performance improvements of the modified unit relative to the OEM were documented. Average energy used to cool the 1,200 square foot home at the height of summer is approximately 13 kWh/day. This was about half the air conditioner electricity consumption prior to installation of the unit. A demand profile for the prototype is shown in Figure 20. Performance with wet and dry media pads is also shown.

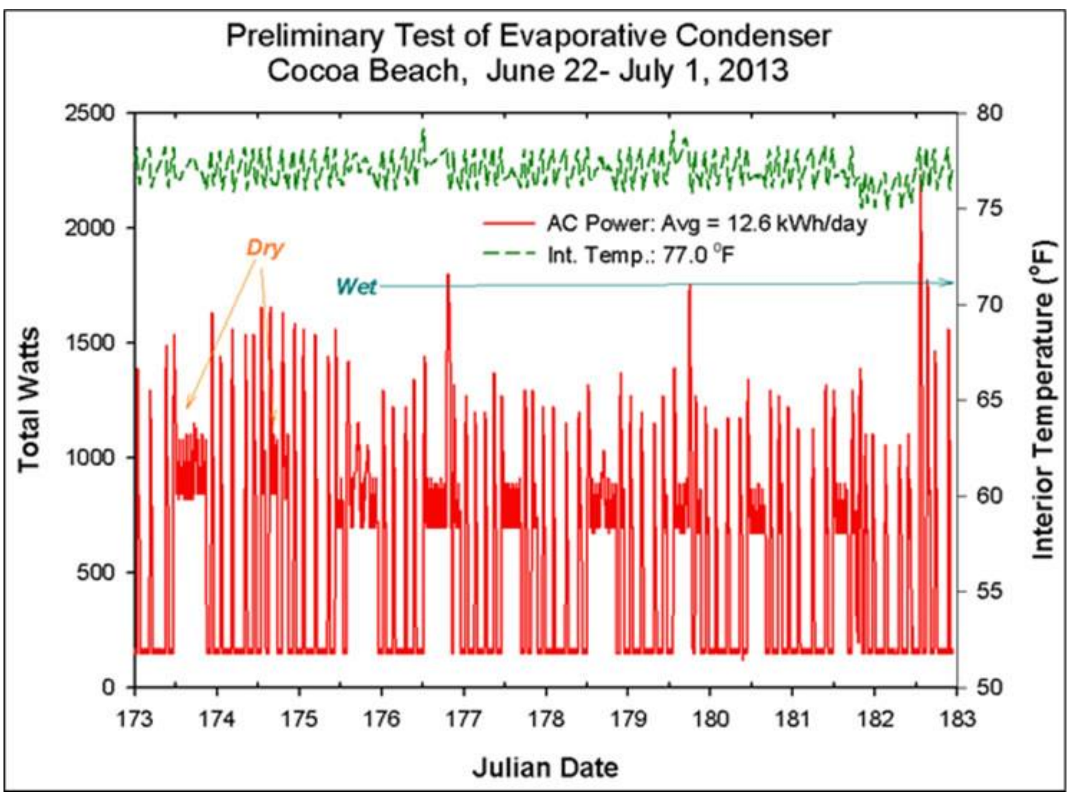

Figure 20. Field test demand profile

Measured Water Consumption

One barrier to the use of evaporative cooling in the residential market is water consumption. The Phase II prototype used a pumping system to re-circulate water from a sump comprised of a continuous ring of 8” polyvinyl chloride (PVC) piping up to another PVC manifold over the top of the media pads. Water would trickle out of the manifold and onto the evaporative pads, which were seated in a gutter assembly. The gutter assembly was affixed to the 8” PVC where the water would drain into the piping. A tub reservoir to maintain water levels was connected to the sump ring. The tub contained a float valve supplied by the city water supply. Water consumption was measured using a pulse-initiating flow meter (100 pulses/gallon) recorded at five-minute intervals.

Figure 21 shows the measured water consumption over the monitoring period from September 14th - 17th during which the circulation system operated continuously to maintain saturated media pads. Superimposed on the plot of ambient temperatures and the temperatures after the cooling pads is the measured water consumption every five minutes. Note that the water draw is not constant as the reservoir held approximately 30 gallons and after it was filled would only re-fill when the float valve called for more water. This did not occur often during nighttime hours or even daytime hours after a major fill. The measured water consumption averaged 5.6 gallons per hour (135 gallons per day) during the period.

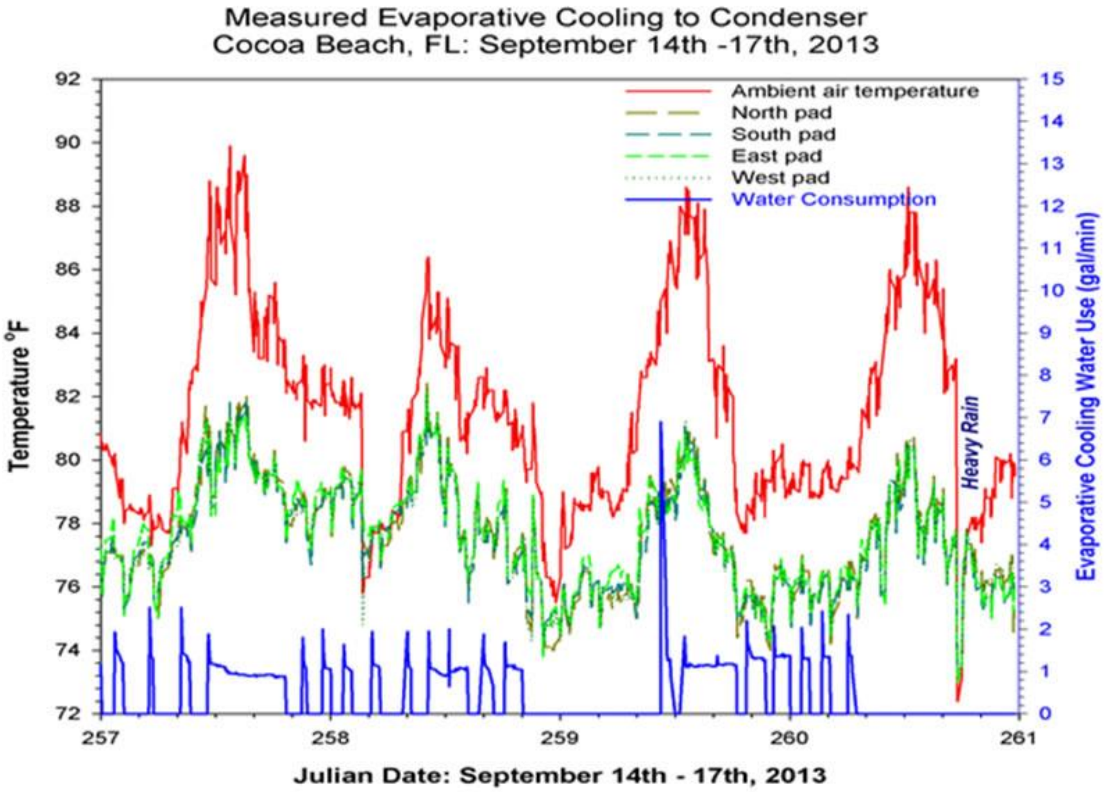

Figure 21. Evaporative cooler water demand

Examining the data, it was found that restricting pumping to the hours from 9 AM to midnight could reduce this consumption by as much as 30 gallons per day. Such control would provide little reduction in system performance since little evaporative cooling potential exists outside of this time frame. This would also have the advantage of drying out the pads each day

On the other hand, with a fully developed embodiment of the invention, the system water sump would have to be purged at least once a week such that even with best management, water consumption would still reach approximately 2,000 gallons per month. Current local water rates are approximately $1.00 per 1000 gallons such that this would not be an excessive economic reduction in the attractiveness of the potential energy savings.

There are three other strategies considered to reduce water use: 1) collection and utilization of rain water 2) use of local gray water, and 3) piping of evaporator condensate to the system sump. Although considered, these were not examined within the scope of our study.

In addition to water consumption, the parasitic pump power used for water circulation remained an issue. Various pumping and control strategies were investigated. The final design used a 20-watt DC fountain pump powered by two 10-watt solar photovoltaic (PV) panels. However, due to available sunlight, that system only operated from approximately 10:30 AM to 4:30 PM.

Testing revealed reliable operating experience with the wetting system during the entire summer. Data showed that the unit runs in low speed during wetted operation (as seen in the data in Figure 20). Average water use has averaged about 135 gallons per day, although some of this was due to leakage from the system sump.

Field Data Performance of the Evaporative Cooling Apparatus

Figure 22 shows the measured performance of the evaporative cooling system at the field test site during a week long period from September 11th - 18th, 2013. During this period, the system was changed from a solar PV pumped system to one with continuous pumping. This change occurred on Julian Day 256 (September 13th). The sole change was the addition of a small AC to DC transformer used to power the 20-watt pump as opposed to the two 10-watt PV panels used previously.

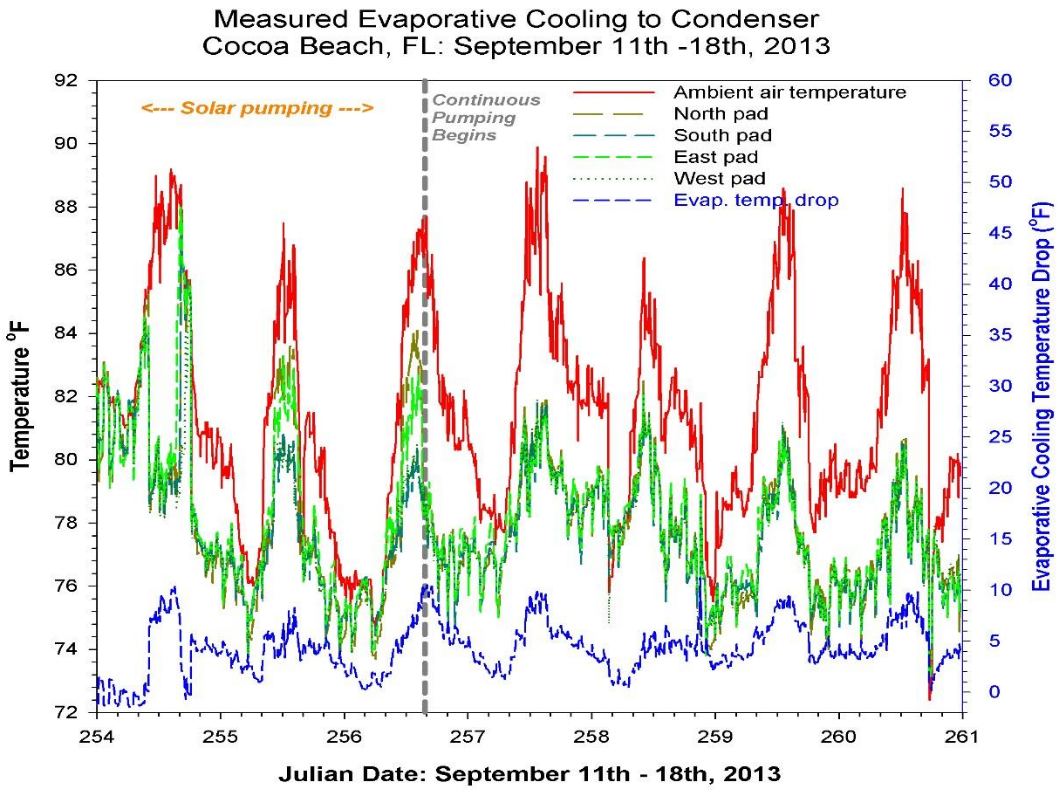

Figure 22. Evaporative media performance profile.

The plot shows the ambient air temperature each five minutes during the data collection as well as the air temperature immediately on the condenser side of the evaporative pads on each of the four sides. Little difference is seen in the individual 6" pads, although clearly, cooling performance is improved with continuous pumping, particularly in morning and evening hours. The average temperature difference between the average pad temperatures and that of the ambient air is shown in blue.

During a 24-hour period, the average temperature drop from ambient air temperature to that before the condenser was 3.8°F with solar pumping and 4.6°F with continuous pumping. However, these numbers include the nighttime hours. If the analysis is confined to the period between 7 AM and midnight, the temperature differences for the condenser are larger: 4.9°F for the solar pumped segment versus 5.6°F for the continuous pumped segment. These represent significant improvements in the operating environment for the condenser during the operating period where the building experiences most of its cooling load. It is also noteworthy that little advantage of the evaporative cooling system is seen between midnight and 7 AM in Central Florida's summer climate. This may mean that pumping could be suspended during these hours with more advanced controls.

To roughly see how evaporative cooling impacted performance in the field experiment, an eleven-day period at the end of May until early June with no evaporative cooling was compared to another period with continuous evaporative cooling of the condenser in early September. Figures 23 and 24 depict system power plotted against the ambient outdoor temperature conditions with the evaporative cooling system turned off and on.

A comparison of Figures 23 and 24 shows that the power demand of the unit with evaporative cooling is much lower and consistent than without. Also of note is that this reduction in power consumption was observed despite the ambient conditions being warmer in the post period (82.1°F) versus the pre period (78.3°F). Thus, any bias from weather is reducing the indicated savings. We compared periods in the 5-minute data when compressors were operating in the two plots. Table 3 summarizes the two periods.

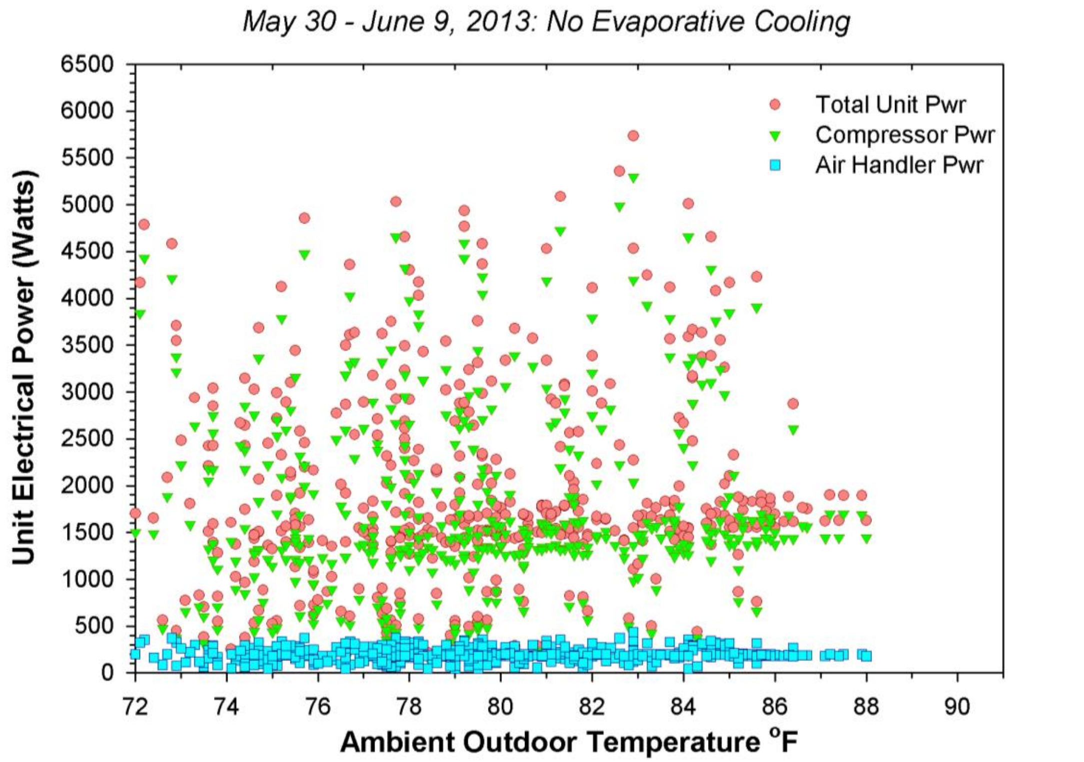

Figure 23. Performance of the Nordyne Variable-Speed Unit with No Evaporative Cooling from May 30th- June 9th, 2013

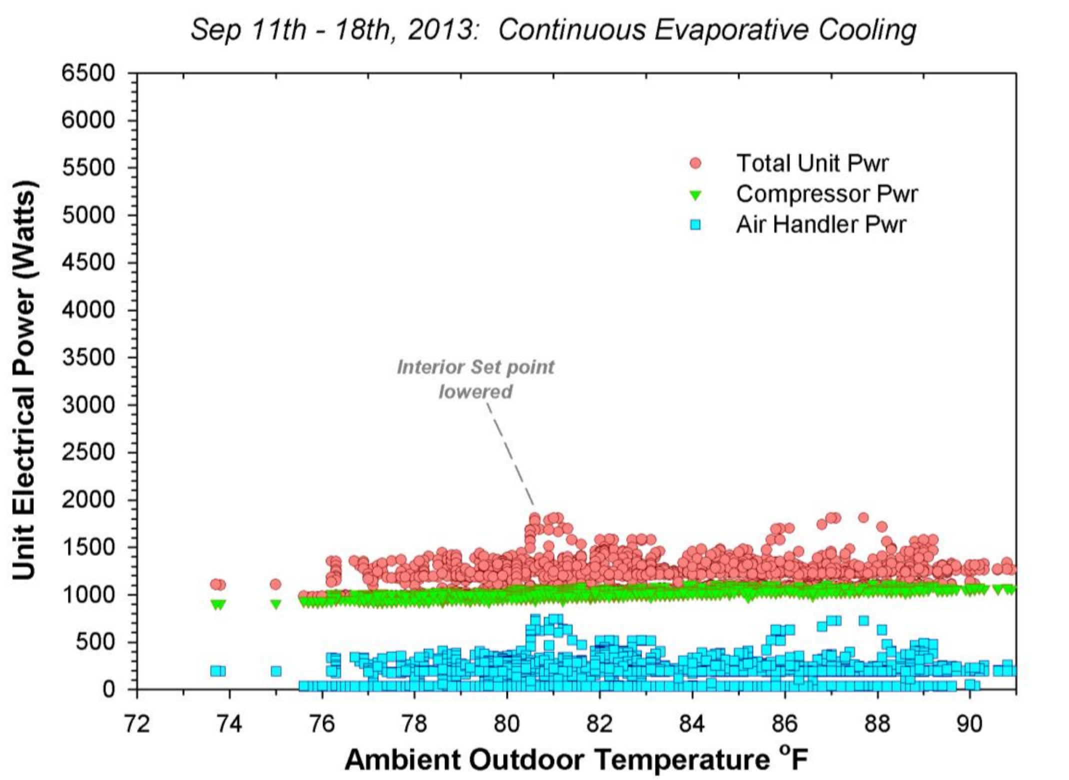

Figure 24. Performance of the Nordyne Variable-Speed AC Unit with Continuous Evaporative Cooling from September 11th - 18th, 2013

Table 3. Influence of Evaporative Cooling and Unit Power Performance in Summer 2013

Parameter	No Evaporative Cooling May 30 - June 9, 2013	Continuous Evaporative Cooling September 11th - 18th, 2013
Ambient Air Temperature (°F)	78.3	82.1
Ambient Air Relative Humidity (%)	75.4	73.3
Total Unit Power (W)	1957	1154
Compressor Power (W)	1759	1008
AHU Power (W)	198	146
Interior Temperature (°F)	77.1	76.8
Interior Relative Humidity (%)	50.5	47.0

Table 3 shows that not only was total unit power 41% lower when the unit was operating in evaporative cooling mode (less favorable outdoor conditions), but that it was also able to maintain lower interior sensible conditions and reduced humidity (50.5% versus 47.0%).

Conclusions

Two prototype evaporatively-cooled condensers with an advanced fan state were produced. The first Phase I test prototype, with 2” evaporative pads and an advanced fan state achieved an EER 30.6 Btu/Wh at the ARI B-test condition (a 33% improvement in the nameplate EER for the unit). Testing at 95°F outdoor temperature and an 80°F/67°F indoor dry-bulb/wet-bulb condition achieved an EER of 23.6, which was a 40% improvement in performance over compared to the manufacturer's data for the unit (16.7 Btu/Wh).

The second Phase II prototype featured a revised pad arrangement and diffuser pressure recovery stage designed to reduce pressure drop in the fan stage. Tests showed that during dry conditions, without evaporative cooling, cooling-related efficiency was increased by 3-5%. The second prototype also used a thicker 6” evaporative pads and a more effectual water distribution system. With the evaporative system operating at low speed, the new prototype achieved an EER of 32.3 Btu/Wh at the 82^{o}F ARI B condition. This was 6% better than the performance with the earlier prototype and 40% better than the nameplate manufacturer's data for the system.

Performance under peak conditions was 22.7 Btu/Wh at minimum speed as opposed to 16.7 Btu/Wh in the baseline configuration — a 36% improvement in performance. Capacity was 15.9 kBtu at minimum speed; performance improved 22% at the intermediate speed at this temperature, again pointing to the importance of reducing high-speed operation when evaporative cooling is available.

In summary, laboratory tests of a best available AC system with an advanced fan diffuser stage coupled with evaporative pre-cooler could reach a performance of 32 Btu/Wh at conditions typically used to rate single-speed systems for SEER (see Figure 25). Capacity was increased by evaporative cooling, and power use of the compressor was lower. Efficiency improvements without evaporative cooling, realized by the better fan and diffuser stage, were measured at 9-11%. Efficiency improvements at peak summer conditions were large: 36-41% increase at 95^{o}F. Our laboratory and field data for this system suggests that seasonal energy efficiency improvements of 30% are available from such systems — even greater in arid climates. Peak reductions at highest summer temperatures are even larger, suggesting that promoting such systems would be attractive for utilities that experience peak demand constraints.

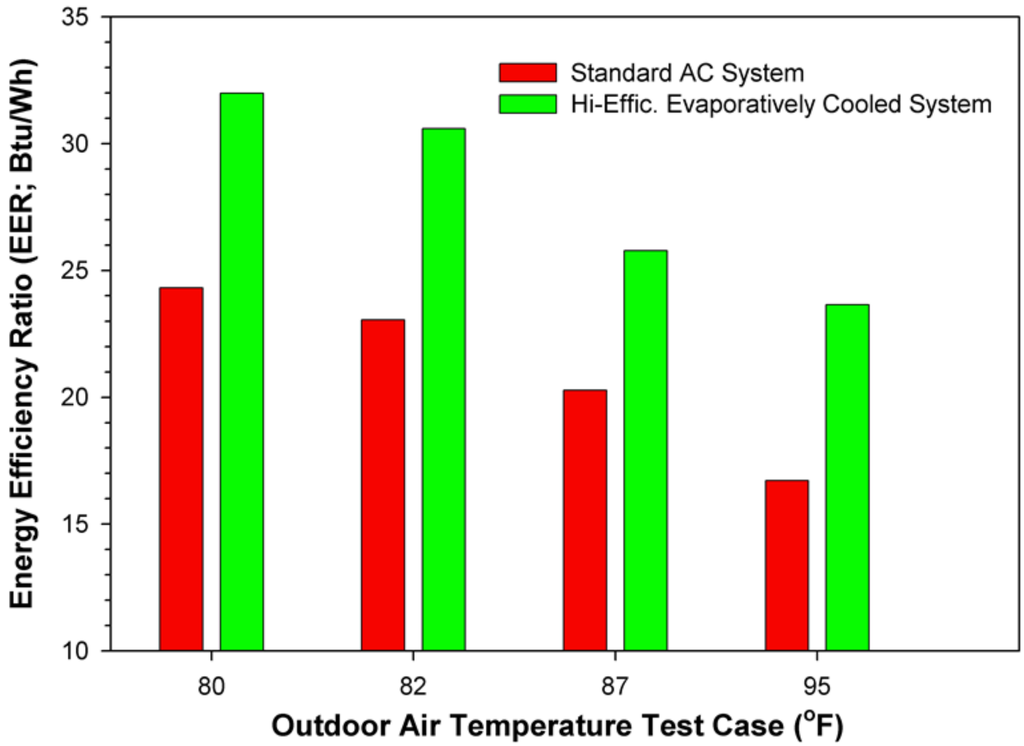

Figure 25. Laboratory measured change in Energy Efficiency Ratio (EER) in standard vs. Phase II prototype evaporatively-cooled unit.

Field testing of this system revealed reliable operation during the summer of 2013 in an occupied home in Cocoa Beach, Florida and very low cooling energy consumption. Measured energy use only averaged about 13 kWh/day during the hottest part of summer. Comparisons of field data with and without evaporative cooling showed that the prototype unit used 41% less electrical energy when the evaporative cooler was operating and also maintained lower interior temperature at reduced indoor humidity levels. Several potential system improvements were discovered during operation, including prolonged system operation at low speed during part load conditions, periodic drain of the system sump, and use of solar DC PV panels to augment pumping energy. Also, the prototype would benefit from the inclusion of smart controls for water management as well as further investigations into alternate water sources.

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