Unitary HVAC Equipment: Performance Optimization Strategy and Field Tests

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ABSTRACT

Although often garnering scant attention, commercial unitary HVAC systems, such as rooftop air conditioners, are estimated to consume 0.88 quads of energy annually, or about 46% of commercial building cooling site energy consumption, and are used to cool over 60% of all commercial space in the U.S. The as-installed energy efficiency of unitary systems can be half that of central systems, and the efficiency gap widens as systems age due to maintainability issues. When tuning systems, energy engineers and service technicians use indirect indicators of equipment performance and make adjustments according to manufacturer guidelines and standard field practice, which varies with their level of experience. Growing numbers of unitary systems combined with shrinking budgets result in deferred maintenance, and long-term operation of equipment at degraded levels. Energy efficiency is a metric that must be measured to be optimized. This paper reports on field testing of continuous sensing of operating energy efficiency to control unitary equipment operating parameters, provide remote fault detection diagnostics, and support maintainability. Optimization systems were installed on package units at three sites in diverse climate locations: Cape Canaveral, FL; Mojave Desert, CA; and Beaufort, SC. The systems utilize a relational control strategy to continuously maximize the ratio of cooling delivered versus power consumed as operating conditions vary over a day and across seasons, and as components degrade over time. Condenser fan speed, supply airflow, evaporator temperature, outside airflow, and refrigerant charge were continuously adjusted by the system to maintain a state of optimized operation. The systems successfully detected and attempted to compensate for faults such as low refrigerant charge or condenser coil fouling, and reported operating EER, pressures, temperatures, and efficiency degradation to service technicians in an actionable way. Analysis of resulting data from the field tests shows considerable unitary energy efficiency gains and maintenance improvements can be obtained cost effectively.

INTRODUCTION

Commercial unitary HVAC systems, or rooftop air conditioners, are used to cool over 60% of U.S commercial floor area (DOE, 2013). Rooftop units are also available in heat pump models as an alternative to fuel gas or electric resistance heating. In total, they consumed 0.88 quads of energy annually, or about 46% of commercial building cooling primary energy consumption in 2010 (DOE, 2011). About 170,000 new unitary systems sized 8-tons (28 kW) and larger are installed annually in the U.S. and there are over 1.6 million units in service that were installed since 2005 (AHRI, 2015).

Rooftop air conditioners (RTUs) have been identified as a high priority target for energy savings in buildings

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by ACEEE (Sachs, 2009). The as-installed energy efficiency of unitary systems can be half that of central plants and the efficiency gap widens as systems age due to maintainability issues (Little, 2001 and EPRI, 1997). The U. S. Department of Energy (DOE) teamed with American Society for Heating, Refrigeration and Air Conditioning Engineers (ASHRAE) and the Retail Industry Leaders Association (RILA) to launch the Advanced RTU Campaign, started in May 2013. The Campaign "…is a recognition and guidance program designed to encourage building owners and operators to take advantage of savings opportunities from high efficiency RTUs" (Advanced RTU, 2015). The Campaign is based on the premise that both installed and new RTUs are excellent targets for improved energy efficiency and significant energy savings.

Technology is needed that can increase the energy efficiency and maintainability of unitary equipment to be comparable with central plants. Package units are typically selected in applications where low cost and ease of maintenance are paramount, so technology advances must be cost effective and enhance maintainability. Most unitary models have fixed operating parameters, such as constant speed fans, which differs markedly from variable speed central plants that can be twice as energy efficient. The actual energy efficiency of a unit that's been in operation for several years could be degraded 10 to 40% from its like-new condition, although it might appear to be performing adequately to occupants and service technicians, usually because units are oversized. Efficiency degradation is largely invisible using currently available diagnostic tools, so a system that measures energy efficiency is a step forward.

A versatile diagnostic & control technology that is analogous to central plant optimal control was field tested under the DoD's ESTCP program in order to measure the potential reduction in energy consumption. The technology continuously monitors and maximizes the in-situ energy efficiency ratio (EER) of package units as operating conditions change, and detects and reports faults while automatically minimizing energy consumption. Since unitary equipment is often kept in operation past its economic life, EER measurement would enable facility engineers to present a truer economic justification for major service or replacement of equipment that might otherwise continue to be operated at poor efficiency levels. Economic life is the point where replacing the equipment would pay for itself in operational savings. Formally, it is defined as the time period over which an asset's NPV is maximized. Economic life can be less than absolute physical life for reasons of technological obsolescence, physical deterioration, or product life cycle.

TECHNOLOGY DESCRIPTION

Three field test units were equipped with technology that utilizes a relational control strategy to continuously maximize the ratio of cooling delivered to total power consumed as operating conditions vary over a day and across seasons, and as components degrade over time. This strategy accounts for field effects on system efficiency, such as duct system design, voltage variations, and outdoor airflow. It also responds to the variations in operating conditions, such as ambient temperature and space relative humidity, which all units experience. The control functions supplement the existing factory-supplied unitary controls, which continue to provide basic functionality such as compressor cycling in response to thermostat calls and high- and low-pressure safety protection. Failure of the added technology would result in the unit reverting to the basic factory controls. However, the microprocessor controller is capable of complete unit control and could replace the factory controls for greater cost effectiveness.

The microprocessor controller performs continuous EER and IEER calculations based on the difference between the enthalpy of the refrigerant at the entrance and exit of the cooling coil, since increase in the enthalpy of the refrigerant is balanced by an equal loss of heat from the air being cooled. Control processing is augmented via connection to a cloud server. The user interface is mirrored on a cloud server viewable from anywhere there is an internet connection, while the controller operates stand-alone. The enthalpy difference is calculated from refrigerant enthalpies, which are calculated from measured refrigerant temperatures and pressures. The rate of heat transport is calculated from the refrigerant mass flow rate, which is calculated from the refrigerant volume flow rate and density, which in turn is calculated from sensed refrigerant velocity, temperature, and pressure. The true RMS power demand is calculated by sampling the sensed input voltage and current sine waves. Finally, EER is calculated as the rate of heat transport divided by the power input, and provided as a Btuh per Watt display. The cooling being delivered and the power consumed are also displayed on the controller screen.

The microprocessor software compares sensor readings and derived calculated values against normal logged ranges to detect operating faults, reports them, and compensates by adjusting operation. For example, a sensed delta-T between the high-pressure saturation temperature and ambient temperature beyond the normal range indicates possible condenser coil fouling, which is shown as an alarm on the controller screen, and compensated for by increasing condenser fan speed until a maintenance technician corrects the condition. Sensors include high-side and low-side refrigerant pressure, liquid refrigerant temperature, vapor refrigerant temperature, refrigerant flow rate, ambient and entering air temperature and humidity, leaving air temperature, and supply air temperature.

The controller automatically adjusts condenser fan speed, supply airflow, leaving air temperature, supply air temperature, refrigerant charge level, and damper positions to minimize energy usage and compensate for faults as needed, while meeting space temperature, humidity and air quality needs. For example, if space humidity drifts outside the setpoint range, the cooling coil leaving air temperature is adjusted accordingly; and once the humidity setpoint range is satisfied the temperature is adjusted relationally along with other parameters to maximize the amount of cooling delivered versus power consumed. Supply airflow is adjusted correspondingly, for example, more restrictive duct work with higher pressure losses will tend towards lower airflow settings, or in other installations the system will take advantage of free flowing ductwork to provide more airflow and increased energy efficiency.

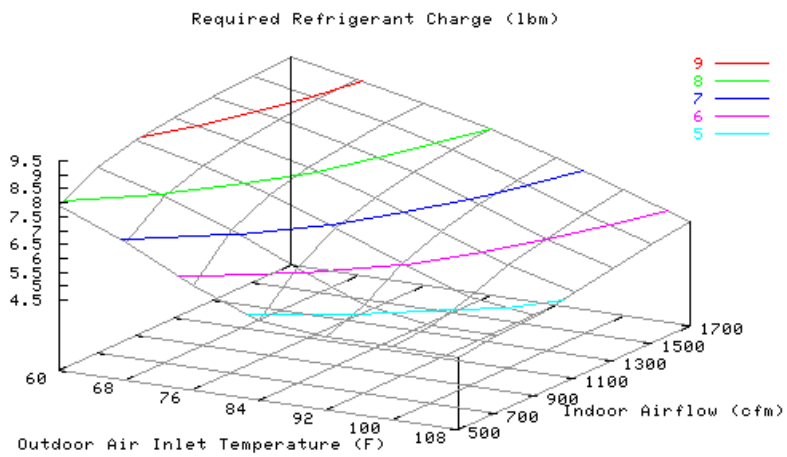

Figure 1 Results from ORNL Mark VII modeling of a prototype 4-ton package unit showing how optimum refrigerant charge level (z-axis and colored contours) varies with condenser outdoor air inlet temperature (x-axis) and indoor supply airflow (y-axis).

The technology addresses EER degradation due to refrigerant leaks in a straightforward manner. Service technicians sometimes address minor refrigerant leaks by adding refrigerant during seasonal service visits. It is difficult and time-consuming to locate a small leak, which is usually not repairable without the labor-intensive procedure of recovering, evacuating, and recharging a system. Systems are on occasion intentionally overcharged to compensate for pinhole leaks. Unfortunately, repeated topping off over time can result in drift of the mixture proportion in blended refrigerants, for example, more R-125 than R-32 could escape from a leaking R410A condenser coil, since R-125 condenses first. The controller detects a refrigerant charge imbalance, and adjusts the charge accordingly by flowing refrigerant into or out of a receiver. The controller also responds to an overcharge condition, as ambient air temperature and airflow affects optimal refrigerant charge level. Simulation results show that energy efficiency is increased by reducing charge as ambient temperature rises, and as evaporator coil airflow is reduced.

DATA ANALYSIS

Data logging of the cooling delivered versus power consumed enables calculation of a field-measured operational IEER (Integrated Energy Efficiency Ratio) calculated using the formulas published in ANSI/AHRI Standard 340/360 (AHRI, 2007), which is displayed on the controller screen. Trending of operational IEER can quantify long-term degradation of energy efficiency when compared against as-installed values. Regression

calculations are performed to obtain linear relations for power used and cooling delivered versus ambient temperature, which characterizes most of the variation in operating conditions – entering air conditions typically have much less variation than ambient temperature. The regression yields equations of two lines in the form y=mx+b that are

$$\text{Power} = m_p * \text{OAT} + b_p \quad \text{and} \quad \text{Cooling} = m_c * \text{OAT} + b_c \qquad \text{(1a and 1b)}$$

From equations 1a and 1b, Power and Cooling are calculated at the four standard rating temperatures: OAT = 90, 81.5, 68, 65 F, giving four values of Power and Cooling. Then EER is calculated at the four temperatures for substitution into the formula for IEER defined by Section 6.2.2 of ANSI/AHRI Standard 340/360-2007,

$$\text{IEER} = 0.02 * \text{EER}(95) + 0.617 * \text{EER}(81.5) + 0.238 * \text{EER}(68) + 0.125 * \text{EER}(65) \quad (2)$$

While operational IEER is not directly equivalent to published IEER ratings measured under tightly controlled laboratory conditions, it can be valuable for comparisons over time and between systems.

FIELD TESTING

Test systems were installed at three sites in diverse climate locations: Cape Canaveral, FL; Mojave Desert, CA; and Beaufort, SC. The Florida and South Carolina sites are located at humid and temperate ends of the ASHRAE hot & humid climate region with mild and cool winters, respectively. The Florida site has a winter heating season limited to a few days of below normal temperatures and requires dehumidification nearly all year. The South Carolina site has a 4-month heating season, during which no cooling is needed and heat is provided by a gas burner. The California site has the high ambient temperatures and low humidity of the hot & arid Mojave Desert, requiring more aggressive refrigerant management than the Florida and South Carolina sites, especially due to the low critical temperature of R410A as compared with R22. The Mojave site has a 4-month heating season.

Figure 2 Unitary equipment at field test sites, left to right: South Carolina, Florida, California.

At the Florida site, equipment is a dual-compressor 8¼-ton (29 kW) R410A unit with electric heat installed 2012, which serves an electronics laboratory with tight ±1 deg-F (±0.56 deg-C) space temperature and humidity 45±5%rh requirements. The South Carolina site has a dual compressor 20-ton (70 kW) gas heat unit installed 2003; this is a legacy R22 system that serves a retail store. The California equipment is a dual-compressor, 12½-ton (44 kW) R410A heat pump installed 2010, which serves a classroom building. The diagnostic controller system has capability to simultaneously optimize many operating parameters using a relational control algorithm, including supply air temperature setpoint, supply fan airflow, cooling coil temperature setpoint, bypass damper position, condenser fan speed, fresh air damper position, and refrigerant charge level. Many of these parameters are not controllable on most single zone unitary models. Accordingly, the test units were also retrofitted with variable speed supply and condenser fan drives, and a bypass damper with actuator in order to test simultaneous optimization of several parameters. Supply airflow is optimized to maximize sensed EER, rather than typical duct static pressure based VAV control – test units were single zone, not VAV systems. Note that relational control of these variable components was used to maximize operating EER, rather than conventional VAV control to meet a static pressure setpoint.

Baseline performance measurements were taken over the first cooling season of the project to benchmark energy efficiency before installation of the diagnostic controller technology, which was installed between the two cooling seasons of the project. Performance during the second cooling season was compared against the benchmark. Metrics used to measure performance are field-measured EER (Energy Efficiency Ratio = Btu/hr cooling / total unit Watts] and IEER (Integrated Energy Efficiency Ratio) ; cooling season electric kWh consumed – both actual and normalized to cooling degree-day and heating degree-day (CDD and HDD) weather data for adaptation to other climate locations; IAQ via space relative humidity, temperature, and carbon dioxide levels and the fraction of occupied hours which these levels are deemed acceptable; and maintenance costs and the number and severity of unplanned or emergency maintenance interventions, if any.

Web-based 45-channel data loggers at each site were used to collect averaged data at 1-minute intervals continuously throughout the project period. Dependent system-level variables measured are: System power demand (kW) and energy consumption (kWh); system cooling delivered in terms of both sensible and latent (Btuh); and occupied space air temperature (F), relative humidity (%RH), and carbon dioxide level (ppm) differential with respect to ambient carbon dioxide level. Dependent component-level variables measured are: compressor and fan electric power (Watts), refrigerant pressures and temperatures at the inlet and outlet of the compressor (psig and F); refrigerant flow rate (gpm); refrigerant charge (lbm); coil air face velocity (fpm).

A propagation of error analysis was performed on typical data sets using a sensitivity analysis technique, to quantify the error in the IEER measurement from n^{th}-order uncertainty in the temperature, humidity, pressure, flow, and power inputs. The accuracy of the measured operational IEER is generally ±5% or ±0.6 Btuh/Watt via this sensitivity/propagation of error uncertainty analysis. Because comparisons were performed using data from the same sensors installed in the same positions, and the same equations, calibrations and correlations were used in the analysis, the same uncertainty in the baseline field operational IEER equally applies to the test systems' operational IEER. Thus, the field measured IEER values are directly comparable to each other with better certainty than comparisons with values obtained from other sources, such as factory ratings obtained under laboratory conditions.

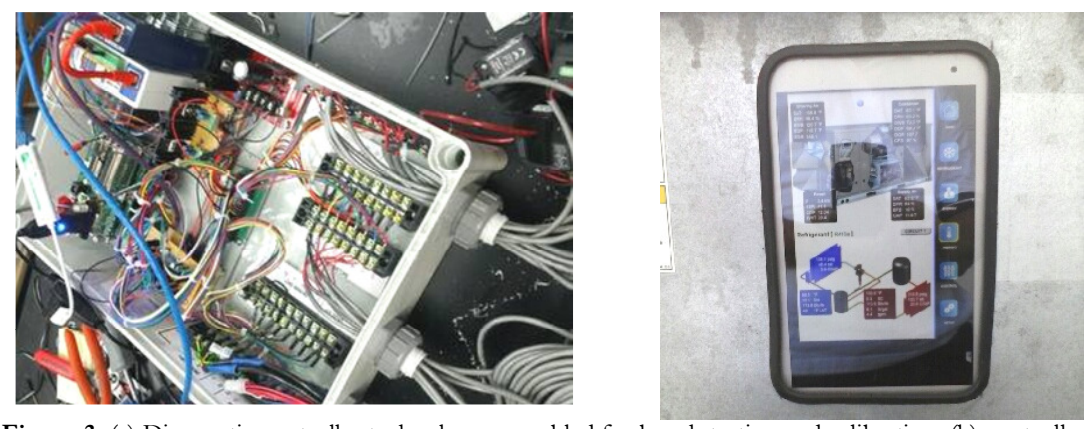

Figure 3 (a) Diagnostic controller technology assembled for bench testing and calibration, (b) controller touch screen panel as installed as the Cape Canveral, FL test site. The touch screen is accessible remotely via web.

RESULTS

Data collected over two cooling seasons was analyzed to evaluate the performance of each of the three package units. Data sets for benchmark performance span from summer 2014, and for the units after installation of the diagnostic controller from summer 2015. The authors found a significant increase in the field-measured operating IEER of the three test units along with decreased energy consumption, while maintaining or improving comfort and ventilation levels. On a few occasions, the controller identified and alerted faults, including a failed condenser fan motor and low refrigerant charge. The controllers adjusted operating parameters to maximize performance, as shown in Figure 4. Sensor inputs are filtered to remove transients caused by compressor starts /stops and controller set point initializations. The retail store at the South Carolina site does not require a tight humidity setpoint, so a band

setting between too dry and too humid was entered into the controller. Steady-state damper position signal plotted in Figure 4a shows how the controller modulates the bypass damper open as space humidity rises, and modulates closed in the setpoint range of 50% to 60%rh. The California site required no dehumidification whatsoever.

Steady state condenser fan speed signal plotted in Figure 4b shows the controller response versus outdoor ambient temperature. The South Carolina system benefited from the unit's intertwined condenser coil circuiting, which allowed the condenser fan to run at approximately 40-60%-speed when one compressor was energized. With two compressors energized, there was fan energy savings at ambient temperatures below 80F (27C) down to approximately 80%-speed at the coldest ambient temperature during the test period of 57.2F (14C). The other two package units have face-split condenser coils, so fan energy savings was less.

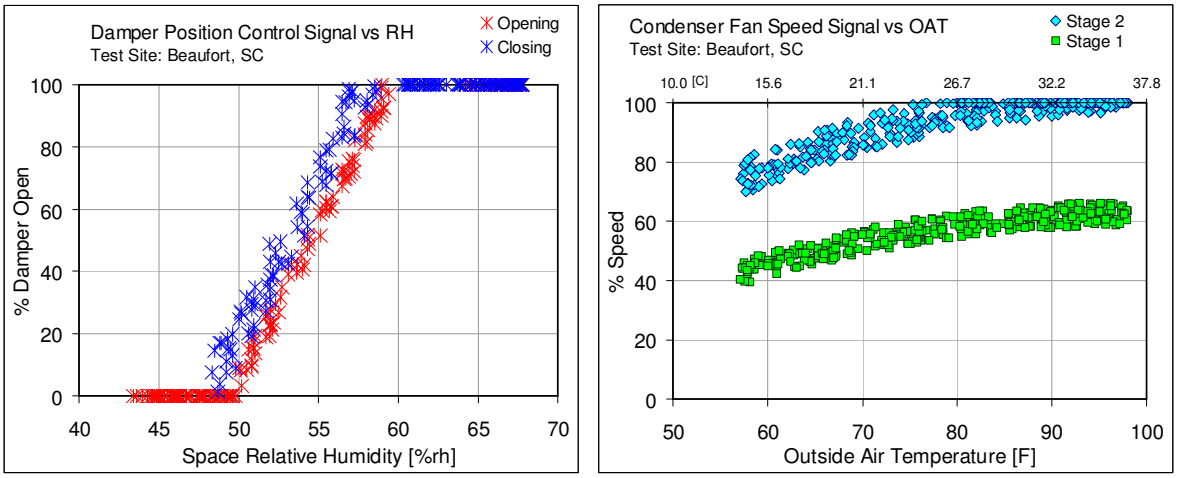

Figure 4 (a) Damper position opened according to rise in space relative humidity, and (b) condenser fan responded to ambient temperature to minimize total system power while satisfying comfort set points.

Measured operational EER is compared with-versus-without the controller in Figure 5. The EER of the test units with the optimized controller tends to be more consistent, we think due to the controller continuously tuning operating parameters as ambient and entering air conditions vary. Also, as the system reaches steady-state in the few minutes after a compressor is energized, particularly at the Florida site which experiences shorter compressor cycles due to the tight space temperature band of ±1 degF, the controller mitigates low EER after compressor start by slowly ramping up the condenser fan and blower speeds as cooling energy becomes available. The result is a 14.7% average efficiency increase of the South Carolina test unit, and a 22.1% increase in average operating efficiency of the Florida unit.

A similar trend exists in EER data from the California test unit, except that measured system efficiency was higher during mid-day as shown in Figure 6. The California unit's measured operational efficiency was influenced by the fresh air intake temperature, which tended to raise the evaporating temperature during the hot and very dry desert afternoons, even though there was actually less cooling being delivered to the space due to the warmer supply air temperature. In any case, the optimizing controller compensated by reducing blower speed and refrigerant charge level, and significantly increased energy efficiency by 30.3% on average, in part due to high ambient operation.

The efficiency analysis results are summarized in Table 1, which lists the measured versus rated cooling capacity and IEER, the benchmarked IEER measured at the start of testing, the baseline IEER and EER over the first (2014) cooling season, and the optimized IEER and EER from the test data with the optimizing controls installed (2015), with the average measured efficiency increase.

Results of economic analysis of the three test systems are shown in Table 2. The analysis was performed using PNNL's RTUCC (Rooftop Unit Comparison Calculator) software, which simulates annual energy costs based on the EER test results using BIN weather data. Of note is the California system had the largest increase in average EER and operationally measured IEER as a percentage, and the largest annual energy savings at 40%. Conversely, the

South Carolina system had the lowest measured improvement in EER and IEER (Table 1), and was the bottom performer at 27% annual energy savings (Table 2). The results illustrate how EER and IEER measurements do not entirely characterize actual annual operating costs. The divergence between efficiency measurement and annual energy consumption is likely due to differences in economizer usage, occupancy schedule, and of course, climate. It appears the Florida system benefited from the optimizing controller due to a large dehumidification requirement and short compressor cycles, resulting in 37% annual energy savings. The savings-to-investment ratio (SIR) was best for the California system, despite the low $/kWh rate.

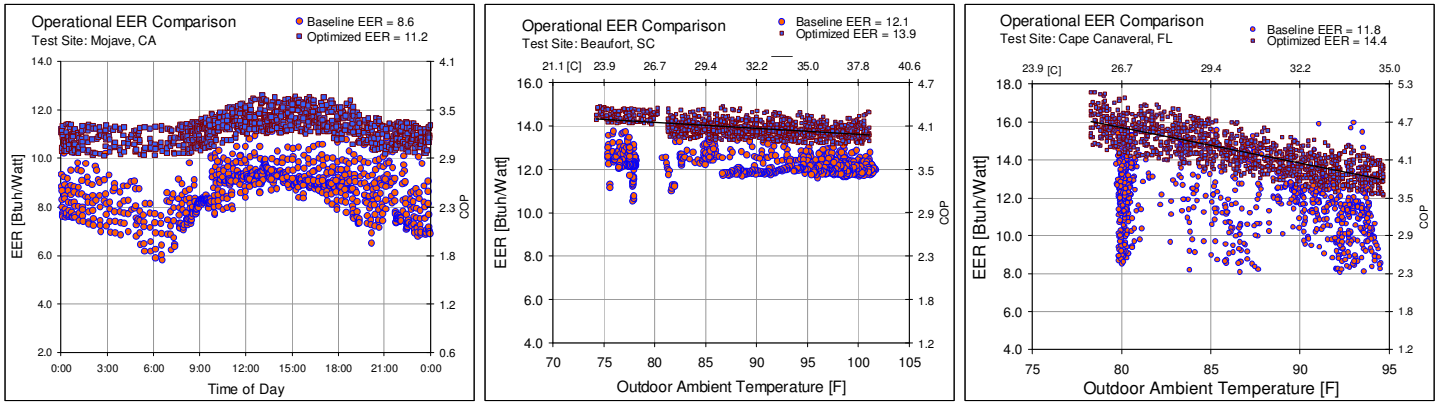

Figure 5 Measured operational EER comparison of Optimized against Baseline from the (a) Mojave, CA test unit (b) Beaufort, SC test unit and (c) the Cape Canaveral, FL test unit with optimizing controller installed(typical summer day 2015) against the baseline EER before the controls were installed (typical summer day 2014).

Table 1. Overall Measured Efficiency Gains and Summary of IEER / EER Analysis

Data Summary		Cooling Tons		Efficiency IEER		Operational IEER		Average EER		Efficiency Gain	
Site	CDD-2014	Data	Rated	Rated	Benchmark	Baseline	Optimized	Baseline	Optimized	IEER	EER
Beaufort, SC	2627	21.5	20.8	11.2	11.8	12.4	14.3	12.1	13.9	15.1%	14.7%
Mojave, CA	3225	8.9	12.5	10.7	9.5	7.8	10.6	8.6	11.2	36.6%	30.3%
Cape Canaveral, FL	3633	7.2	8.1	13.2	13.9	13.4	16.4	11.8	14.4	21.9%	22.1%

Table 2. Economic Comparison Results from Annual Simulations

	Beaufort, SC		Mojave, CA		Cape Canaveral, FL	
Annual Comparison	Baseline	Optimized	Baseline	Optimized	Baseline	Optimized
Energy Used kWh	34,219	24,943	39,169	23,358	28,112	17,846
Energy Cost	$3,422	$2,494	$2,664	$1,588	$1,743	$1,106
Rate per kWh	$0.100		$0.068		$0.062	
Energy Saved	**27%**		**40%**		**37%**	
Net Present Value	$3,412		$4,672		$929	
Payback Years	5.3		4.5		8.1	
Rate of Return	15.9%		20.1%		6.9%	
SIR	**1.8**		**2.0**		**1.2**	

The diagnostic controller hardware was found to be somewhat complex and the project team experienced moderate difficulty retrofitting sensors and wiring into the existing HVAC package units in the field. The $4,500 controller cost used in the economic calculations includes hardware and sensors, not installation labor. It would be more in line with assembly in a controlled setting or factory, where the system could be installed without the time pressure of an occupied space in need of cooling. Total field retrofit cost might be twice this amount or more, depending on local factors. Most of the controller components have performed robustly with no maintenance issues as of the writing of this paper.

CONCLUSION

Field testing of three units with the optimizing controller installed resulted in measured cooling efficiency increases of 14.7 to 30.3%, and simulated annual energy savings of 27 to 40%. The results show considerable variability in savings potential depending on the application and climate. Economic indicators were positive for all three applications, giving payback period ranging from 4.5 to 8.1 years and savings-to-investment ratio (SIR) of 1.2 to 2.0. Installation of the technology on the larger 12½ (44 kW) and 20-ton (70 kW) package units produced correspondingly more energy savings, shorter payback period, and higher SIR than the 8¼-ton (29 kW) package unit.

ACKNOWLEDGMENTS

Thanks to the many knowledgeable and dedicated people who made this work possible, including Dr. Jim Galvin, Peter Knowles, and Travis Michalke of the DoD Environmental Security Technology Certification Program (ESTCP), Neil Tisdale and Bill Rogers of Marine Corps Air Station Beaufort, Hossam Kassab of Army Ft. Irwin, Mike Manning of Cape Canaveral Air Force Station, Jason Zareva of Trane, Ted Cherubin and Michael Taras of Carrier.

NOMENCLATURE

ACEEE – American Council for an Energy-Efficient Economy
b_p and b_c - intercept of liner regression of power versus OAT and cooling delivered vs OAT
m_p and m_c - slope of liner regression of power versus OAT and cooling delivered vs OAT
ESTCP – Environmental Security Technology Certification Program
OAT - outdoor air temperature
CDD - cooling degree days
HDD - heating degree days
EER – energy efficiency ratio
IEER – integrated energy efficiency ratio
SIR – savings to investment ratio
ORNL – Oak Ridge National Laboratory
PNNL – Pacific Northwest National Laboratory

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