Improvement of Integrated Energy Efficiency and Latent Cooling Capability by Refrigeration Cycle Variation with Evaporator Coil Optimization in R-410a Unitary Equipment

Michael K. West, Ph.D., P.E.
Member ASHRAE

Richard Combes, Ph.D., P.E.
Associate Member ASHRAE

ABSTRACT

Variation of the refrigeration circuit and evaporator coil arrangement from conventional unitary air-conditioning equipment has been demonstrated to improve energy efficiency and dehumidification performance in lab and field tests. This paper will present the field test methodology and results. The ensuing cycle modifications allowed further optimization of tube circuiting, face area, and fin density of the evaporator coil in combination with a controllable return air bypass to achieve a 15% to 25% improvement in field-estimated IEER (Integrated Energy Efficiency Ratio, weighted Btuh per Watt according to ANSI/AHRI Standard 340/360-2007). Analysis using the DOE/ORNL Heat Pump Design Model and other software identified that refrigerant condition through the evaporator coil can be controlled to increase phase change heat transfer by maximizing liquid refrigerant fraction.

In two rigorously instrumented seasonal field tests of dual-compressor package units, the resultant component and control modifications reduced energy consumption by at least 15% relative to identical unmodified equipment. The modifications also enhanced the capability of the equipment to meet latent loads, independent of sensible load. In hot & humid climate tests, control of the modified units was optimized to meet latent cooling load without reheat while reducing seasonal electric energy consumption. The paper will also present modification requirements, which indicate a minimal increase in first cost would provide worthwhile energy savings along with enhanced dehumidification.

INTRODUCTION

The topic at hand is improvement of energy efficiency and dehumidification performance of unitary DX air-conditioners, an equipment type that cools roughly three-quarters of commercial and two-thirds of government building

Michael K. West, Ph.D., P.E., is Principal-CEO at Advantek Consulting Engineering, Inc., Melbourne, Florida. Richard Combes, Ph.D., P.E., is Lead Mechanical Engineer at Advantek Consulting Engineering, Inc., Fripp Island, South Carolina.

floor space. The choice of a particular DX package unit is guided by performance characteristics and component reliability, along with serviceability and cost. This paper presents field test results demonstrating the advantages offered by application of a simple refrigeration cycle variant with attendant equipment modifications (U.S. patent 6,427,454), and suggests that this approach to improving catalog IEER ratings, as well as installed performance, has significant potential.

When a valuation of alternate air-conditioning technology is performed, system operation at a wide spectrum of environmental conditions must be examined. The high ambient temperature region attracts particular attention, where performance is most demanded and valued by end customers. Any refrigeration system suffers performance degradation in high ambient temperature environments, particularly when operation temperatures are extended to as high as 135°F (57.2°C). R-410A equipment exhibits a marked performance deficiency at high ambient conditions in comparison to legacy R-22 equipment. The modifications addressed in this paper require significantly less effort and investment towards performance parity at high ambient temperatures, where conventional approaches become thermodynamically prohibitive. When retrofitted to legacy R-22 equipment, the modifications can achieve either an increase in capacity or a significant reduction in energy use, along with an increase in dehumidification capability.

Implementing the most cost effective dehumidification approach is critical to reducing facility energy costs. Dehumidification at part load accounts for an estimated one-third of cooling energy consumption in the Southern region of the U.S., a quarter of the cooling energy in the Northeast and Midwest regions according to rough estimates (Dieckman, 2009). The Southern region accounts for more than 60% of the total U.S. cooling energy consumption, while the Northeast and Midwest regions account for 22%. In total, dehumidification at part load accounts for at least 25% of total U.S. cooling energy consumption. Cooling equipment that can efficiently meet latent load independently of sensible loads can in practice realize the potential energy savings from improved building envelopes and equipment.

In many operating scenarios, conventional equipment latent capacity is inadequate. Since most air-conditioning units are controlled by thermostats, which respond to sensible loads only, the result is high indoor humidity or use of energy intensive reheat. This situation is exacerbated by condensed humidity that remains on the evaporator coil after the compressors cycle off and re-evaporates, since the blower must run constantly to meet ventilation requirements. In other operating scenarios such as desert climates, the reduced need for latent capacity is not utilized to improve energy efficiency. This situation exists because current unitary equipment design is necessarily compromised across all climate zones into a single product line.

While for many years, the ideals of higher energy efficiency and higher moisture removal were thought to be at odds with each other, more recent developments show that's not always the case. Also to be considered, to strive for minimum cost and inventory, manufacturers design their unitary equipment with the same bill of material for all their geographical sales regions. Compromise on energy efficiency and moisture removal for all regions is the name of the game; Florida, California, Texas, and Illinois equipment are the same even though the electricity rates and moisture loads are drastically diverse. . Current energy efficiency specifications for new unitary air conditioning and heat pump systems establish Energy Efficiency Ratios (EERs) of 9.7 to 14.0, depending on system capacity. However, the substantial base of installed unitary systems has an EER of 9.0 or less, dependent on system condition and maintenance history. In humid climates, such as the two field sites, control of relative humidity (RH) and indoor air quality (IAQ) in conditioned spaces is often less than satisfactory with unitary systems due to relatively warm coil leaving air temperatures and frequent compressor cycling. Their limited dehumidification capability results in a limit on the amount of fresh outside air a unit can condition of about 20% of the total airflow.

EQUIPMENT

Conceptually, the cycle variation and component modifications that were field tested appear simple; and they are, in fact, simple to recognize as diagrammed in Figure 1. The interaction of the variation and modifications produce an improvement that is greater than the sum of their individual effects. Application required many design decisions, including the selection of optimum evaporator configuration & airflow, refrigerant inventory management, and trade-offs between increased performance vs. increased cost. Rigorous field testing in commercial equipment resulted in an optimized configuration that provides documented, cost-effective improvements in energy efficiency and dehumidification, without sacrificing occupant comfort or equipment reliability & maintainability.

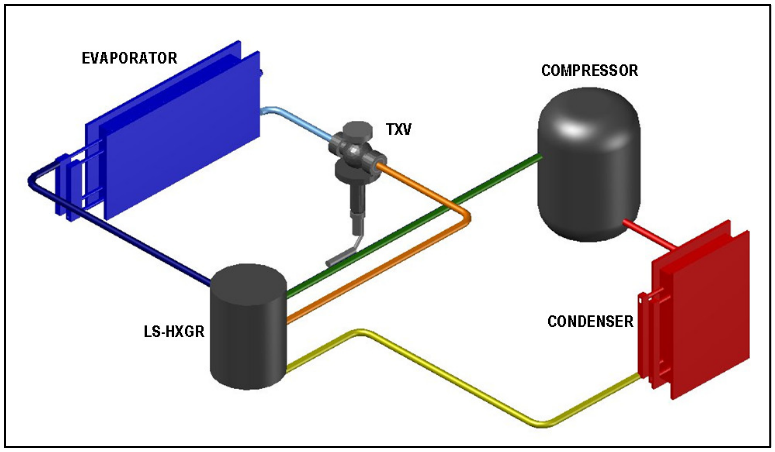

Figure 1. Component layout is identical to that for a conventional air-conditioning cycle with the addition of a liquid suction heat exchanger accumulator (LS-HXGR).

Since heat transfer in boiling is much greater than heat transfer in either subcooled liquid or vapor refrigerant, it is desirable to maximize the presence of liquid refrigerant at the boiling point in the evaporator coil. Vapor increases in temperature as it absorbs heat, while evaporating liquid does not, which results in a colder bulk coil temperature and increased latent capacity. By placing an accumulator in the refrigerant circuit downstream of the evaporator coil, the author's cycle variation results in a higher fraction of liquid refrigerant at or near the boiling point in the coil, increasing heat transfer. At the inlet, refrigerant subcooling to near the evaporator saturation temperature results in half the vapor fraction at the exit of the expansion device. At the outlet, refrigerant flows to a high-effectiveness heat exchange accumulator, which evaporates the ten to twenty percent mass fraction of liquid as it moves towards the compressor.

Most significantly, constraints are released on other equipment design parameters with the ability to manage refrigerant conditions bestowed by the cycle variation. Additional subcooling increases enthalpy difference in the evaporator. The enthalpy difference increase combined with zero superheat and the greater heat transfer coefficient allows selection of a smaller evaporator coil for higher face velocity and reduced face area (which can lower cost). The reduced face area combined with the ability to tolerate liquid at the evaporator coil exit allows evaporator bypass, which greatly improves latent performance. Further gains are achieved by allowing return air to bypass the evaporator through a modulating damper that dynamically varies the latent capacity to match the load sensible heat ratio – the modified unit is a variable sensible heat ratio machine with a fully protected compressor inlet along with an increase in energy efficiency.

Additionally, performance improvements were realized with air-side controls modifications that take advantage of the cycle variant's full potential. Constant air volume (CAV) blowers continuously bring outdoor air (OA) into the building to meet ventilation requirements. This introduces a continuous flow of moisture that, when combined with re-evaporated condensate left on the evaporator coil during compressor off cycles, can lead to excessive indoor humidity levels. To address this issue, whenever compressors are not energized (1) the controls are configured to open the evaporator bypass damper fully, and (2) the OA damper opens fully and the blower speed slows to provide a pre-set flow of fresh air.

Finally, the control methodology was altered from CAV to a sequence developed by the authors dubbed variable volume constant temperature (VVCT). As the name suggests, VVCT varies airflow to meet a constant supply air temperature setpoint, or a constant differential between supply air and return air temperature, as evaporator and condenser entering air conditions change and compressors cycle. Colder setpoints become practical with addition of the cycle variant's increased subcooling and zero superheat capabilities, down to the frost point (32F, 0C) if needed. When the space calls for more dehumidification and less sensible cooling, the bypass damper is opened and the setpoint is reduced. When the space requires less dehumidification, the bypass damper is closed and supply air setpoint is increased, leading to an increase in airflow, up to the condensate carryover point (550 fpm, 2.8 m/s) if needed. This decoupling of sensible and latent control allows reduced airflow to enhance dehumidification when it is appropriate, but operation with high airflow through an interlaced coil when there is less moisture load.

Figure 2a and 2b. New 8½-ton (30 kW) dual-circuit unit installed at Florida test site (left), and retrofitted 20-ton (70 kW) dual-circuit unit at South Carolina test site.

Cost is one of the most critical factors in valuation and justification of alternate technology. Costs associated with application of the variant cycle are related to the liquid suction heat exchanger accumulator, bypass damper, controls, copper piping, and refrigerant charge. Figure 3 gives an example of cost premium for application to a 10-ton (35 kW) unit. The 16% estimated cost increase for the components is partially offset by a reduced size evaporator coil. The added cost will be economically recovered in the field tests through reduced operational energy costs in 4 years at the South Carolina site and in 2.6 years at the Florida site.

Component	**Manufactured Cost**	**Sales Cost Premium**	**Cost increase % of Unit Cost**
Base 10-ton RTU	$3,000	$5,000	
LS-HXGR accumulator	$110	$180	3.6%
Evaporator bypass damper	$170	$280	5.6%
Reduced size cooling coil	($90)	($150)	-3.0%
Controls	$190	$310	6.2%
Copper pipe	$15	$30	0.6%
Refrigerant	$30	$50	1.0%
Other	$60	$100	2.0%
Component Total	$485	$800	16.0%
TOTAL	**$3,485**	**$5,580**	

Figure 3. Estimated manufactured cost premium for application of cycle variant to a baseline 10-ton (35 kW) package unit.

PROCEDURE

Demonstration units were assembled for field tests at either end of the hot humid climate zone, Cape Canaveral Air Force Station at Patrick Air Force Base in Florida (Figure 2a), and Marine Corps Air Station Beaufort in South Carolina (Figure 2b). The demonstration units are fully instrumented on both the airflow process and refrigerant cycle, and the real time data is updated every 1-minute and can be accessed remotely on the web. In Florida, the demonstration unit is a new 8½-ton (30 kW) dual circuit unit with 9 kW electric reheat, which replaced a 7½-ton (26.3 kW) packaged single compressor unit manufactured 5/1999 with 28 kW electric reheat. The unit is controlled by a stand-alone, web enabled thermostat-humidistat. In South Carolina, the demonstration is a field-retrofit of an existing 20-ton (70 kW) dual circuit unit with gas heat manufactured 10/2003. The Air Force building served by the unit is an instrumentation laboratory with tight 24/7 temperature control requirements. The Marine Corps building is the Base Exchange, which is similar to a large commercial retail store, with night time shutdown. The unit is controlled by the central base EMCS, which includes temperature and humidity sensors.

Metrics used to gauge performance are field-measured EER and field-estimated IEER, cooling season electric kWh; actual tracked installation materials and labor costs versus realized electric savings; 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 maintenance interventions, if any.

Each of the 45 sensors on each package unit was bench calibrated and then verified in the field after installation. Sensor calibrations were checked periodically, and a few suspect sensors were recalibrated or replaced. Dependent system-level variables continuously measured are: Unit power (kW) and energy consumption (kWh), total and sensible cooling delivered (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 continuously measured are: compressor and fan electric (amps), refrigerant pressures and temperatures at the inlet and outlet of the compressor (psig and F); refrigerant flow rate (gpm); coil air face velocity (fpm), inter-component air and refrigerant temperatures (F); and control signals status and voltages. The independent variable is a binary change of status 'with' versus 'without' the cycle variant modifications. Background independent variables are ambient temperature (F), humidity (%RH), carbon dioxide level (ppm), occupancy status, and time of day / day of week.

Four operating conditions satisfy the full load and part-load IEER rating criteria set forth in ANSI/AHRI Standard 340/360-2007 and the IEER of each unit configuration was calculated accordingly. To obtain performance at the specified A, B, C, and D rating conditions of 95F, 81.5F, 68F, and 65F ambient and 80/67F db/wb entering air, linear interpolation between field-measured data points was used as indicated acceptable by section 6.2.2 of the Standard: "EER is determined by plotting the tested EER vs. the percent load and using straight line segments to connect the actual performance points. Linear interpolation is used to determine the EER at 75%, 50% and 25% net capacity." The field-estimated IEER was then calculated according to section 6.2.2 of ANSI/AHRI Standard 340/360-2007 by the formula,

$$IEER = (0.020 \cdot A) + (0.617 \cdot B) + (0.238 \cdot C) + (0.125 \cdot D)$$

where A, B, C, and D are the calculated, interpolated, field-measured EER (Btuh per Watt) at each rating point. A propagation of error analysis was performed using a sensitivity analysis technique, to quantify how the error in the IEER estimation can be apportioned to uncertainty in the temperature, humidity, pressure, flow, and power inputs. The accuracy of this field-estimated IEER is ±5% or ±0.6 Btuh/Watt via this sensitivity/propagation of error uncertainty analysis. However, because all calculations are performed using data from the same sensors installed in the same positions, and the same equations, formulae calibrations and correlations were used in the analysis and comparison of the data, the same uncertainty in the baseline field-estimated IEER equally applies to the modified unit's field-estimated IEER in the same direction (high or low). Thus, the field-estimated 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.

RESULTS

Data collected at 1-minute sampling rate was analyzed to evaluate the performance of each component of the package units, as well as total system performance, in various operating modes and conditions. End results are summarized in Figure 6. Data sets for the standard unit span from January thru July, and for the modified units from July thru December 2012. The authors found a significant increase in the field-measured EER at 95F 80/67 db/wb and field-estimated IEER and humidity control performance of both units along with decreased energy consumption, while maintaining or improving comfort and ventilation levels.

The instrument laboratory served by the Florida unit requires temperature and humidity control within ±1 deg-F and ±5%rh at setpoints of 72F (22.2 C) and 60%rh, and has a large flow of fresh air. To satisfy these setpoints, both the standard unit and the unit it replaced required prodigious use of electric reheat, 9 kW and 28 kW respectively, accounting for one-third to one-half of system energy use. The modified unit did not: controls called for reheat only 0.5% of the run hours, which accounted for 1.3% of the system energy use, while staying within range of setpoint at least 80% of the total hours, including unoccupied and downtime hours. In addition to these reheat savings, the modified unit efficiency was 27.2% improved over the standard unit from field-estimated IEER 12.5 to field-estimated IEER 15.9 (Integrated Energy

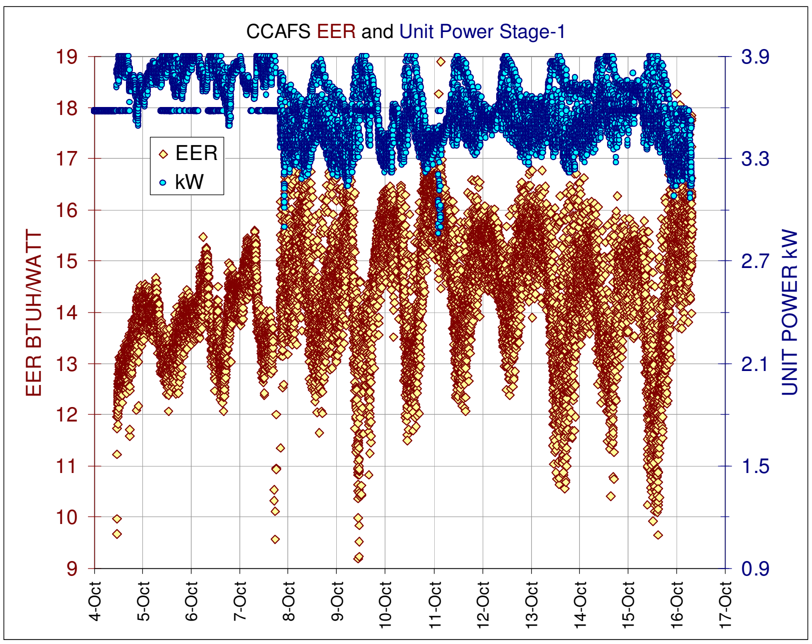

Figure 4: Plot showing field-measured EER and total unit power of Stage-1 (one compressor energized) for a 2-week period at the beginning of October 2012.

Efficiency Ratio, weighted Btuh per Watt calculated using ANSI/AHRI Standard 340/360-2007) with the unit in maximum dehumidification mode almost continuously. Figure 4 shows field-measured EER and total unit kW for a two week period in early October. Sampling at 1-minute intervals of space temperature, humidity and CO_2 was used to calculate comfort level via predicted mean vote (PMV). Dissatisfaction with comfort level was less than 1% of the time: 0.5% warm and 0.4% cool, as listed in Figure 5. Temperature and humidity were in the ASHRAE Standard 55 defined comfort zone 81% of the time. Ventilation was acceptable according to ASHRAE Standard 62 defined

COMFORT CONDITIONS Space Occupancy	CCAFS, FL Laboratory	MCASB, SC Retail
Temperture Setpoint	72	74
Humidity Setpoint	55-65%	50-60%
Median Space Temperature	71.3	74.0
Median Space Humidity	63.7%	49.1%
Median Space CO_2 ppm	418	406
Time in Comfort Zone	81%	67%
Time PMV Satisfied	99.9%	70.1%
Time Ventilation Adequate	100%	100%

Figure 5. Comfort set points and measured values.

FIELD TEST RESULTS	Florida site		South Carolina site	
Location	Cape Canvaeral AFS		Marine Corps AS	
Unit GPS Coordinates	28.433282, -80.583266		32.461092, -80.723941	
Unit Configuration	**Standard**	**Modified**	**Standard**	**Modified**
Refrigerant	R-410a		R-22	
Nominal Tons	8.5		20	
Expansion Type	TXV		Orifice	TXV
Evaporator Coil	4 row/15 fpi	5 row/14 fpi	4 row/15 fpi	4 row/16 fpi
Circuiting	Face split	Interlaced	Interlaced	Interlaced
Evaporator Face Area sqft	11.1	8.6	26.0	20.3
Average Face Velocity FPM	263	357	283	200
Performance				
Cooling Capacity Tons	**7.3**	**8.7**	**20.8**	**20.2**
Reheat kWh per day	8.9	1.1	-	-
SHR @ 81.5F ambient	0.70	0.65	0.72	0.74
SHR Decrease Percent	6.2%		-2.8%	
Average coil airflow CFM	2922	3068	7350	4064
Supply Air F	66.2	61.2	57.3	58.2
Average coil static in-wc	0.11	0.13	0.21	0.09
EER @ 95F ambient	11.3	14.6	10.6	11.7
Field Estimated IEER	**12.5**	**15.9**	**11.2**	**13.4**
Percent Increase IEER	27.2%		19.4%	

Figure 6. Comparison of Modified unit field test results versus Standard unit factory performance. Listed "EER" is field-measured at standard 95F ambient, 80/67F coil entering dry/wet bulb temperature. Listed "IEER" is field-estimated using the IEER calculation procedure set forth in ANSI/AHRI Standard 340/360-2007, as explained above.

CO_2 level 100% of the time.

An important condition to be noted together with the Florida data is the frequent compressor cycling, which is well known to reduce energy efficiency. The tight temperature limits of the lab space required a 1 degree-F deadband to be set on the controller, with no limit on the number of compressor cycles per hour. This is in contrast to a more typical application, such as the South Carolina site, which allows 2-degree swings in space temperature above and below the setpoint and a 10 cycle limit. Compressors in the South Carolina unit at had a cycle period of 20 minutes to hours, compared with just a few minutes for the compressors in the Florida unit. These observations together with the ORNL Mark 7 modeling predictions lead the investigators to believe cycling significantly limited the measured energy efficiency improvement at the Florida site. The field-estimated IEER of the standard un-modified unit was 5% less than the factory rating of 13.2.

The Base Exchange retail store served by the South Carolina unit required a bit less dehumidification than was being provided by the standard unit to limit humidity to 60%rh, and reheat controls were not provided nor were they needed. The modified unit efficiency was 19.4% improved over the standard unit from IEER 11.2 to field-estimated IEER 13.4 with the unit varying its dehumidification capacity to match the space load. EER at 95F 80/67 db/wb increased by 10.4% from the standard unit's 10.6 to the modified unit at 11.7 Btuh/Watt. The bypass damper modulated as expected according to programming of the base EMCS in response to the deviation of space relative humidity from setpoint. Dissatisfaction with comfort level was less than 29.9% of the time, mostly too warm as expected with a night temperature setpoint of 80F (26.7 F) for 8 hours/day. Temperature and humidity were in the ASHRAE Standard 55 defined comfort zone 67% of the time. Ventilation was acceptable according to ASHRAE Standard 62 defined CO_2 level 100% of the time. Median temperature was 74.0 F (23.3) and median humidity was 49.1% as measured in the space with calibrated sensors.

Compressor operation is significantly cooler in the modified units, which might result in longer compressor life as well as sustainability of improved efficiency. Compressor discharge superheat averaged 47 to 67 degrees-F (26 to 37 degrees-C) cooler than the baseline units. Compressor discharge temperatures were reduced from closer to 200 F (93 C) down to closer to 100 F (37 C), depending on ambient conditions.

CONCLUSIONS

The proposed technology successfully addresses the two different realities needed from a major unitary product improvement, namely energy efficiency and dehumidification, while simultaneously addressing conflicting technical and cost issues. The bottom line is equipment that provides a step improvement in energy efficiency, while also improving the ability of the air conditioning equipment to efficiently remove moisture and have its sensible heat ratio (SHR) better follow the load's SHR was successfully demonstrated in a cost-effective way. If a number of models of different capacities share an identical chassis, the variant cycle modifications could be a useful technique to achieve desired vastly different performance targets from the same platform, providing good competitive leverage for a wide range of climates.

Perhaps more importantly to facility managers, the modifications were field retrofit to R-410a equipment and legacy R-22 equipment with excellent results. At most facilities, the installed numbers of existing units far exceeds the number that will be replaced in the coming years, and these older units are more in need of performance improvements.

AKNOWLEDGMENTS

Thanks to the many people who helped make this work possible, including Dr. Jim Galvin and Dr Jeffery Marqusee of the DoD Environmental Security Technology Certification Program (ESTCP), Neil Tisdale and Bill Rogers of Marine Corps Air Station Beaufort, Mike Manning of Cape Canaveral Air Force Station, Jason Zareva of Trane, Ted Cherubin and Michael Taras of Carrier, Kevin Riley of Indyne Corporation, and Christopher Cook of Booze Allen Hamilton.

BIBLIOGRAPHY

Phil Baggett, John Murphy, Paul Solberg, and Justin Wieman. "Energy-Saving Strategies for Rooftop VAV Systems" Trane, Engineers Newsletter Live, 2010.

Michael West, Ph.D, P.E. "Low-Cost Dedicated Outdoor-Air Systems" HPAC Engineering, July 2010.

John Dieckmann, Kurtis McKenney, and James Brodrick, Ph.D. "Vapor Compression, Cromer Cycle: Energy-Efficient Dehumidification" ASHRAE Journal, August 2009.

Michael Y. Feng Ph.D., Kurt W. Roth, Ph.D., Detlef Westphalen, Ph.D., and James Brodrick, Ph.D. "Packaged Rooftop Units: Automated Fault Detection and Diagnostics" ASHRAE Journal, April 2005.

C. K. Rice, The ORNL Modulating Heat Pump Design Tool – ORNL/CON-343, OAK RIDGE NATIONAL LABORATORY, U.S. Department of Energy.

Michael F. Taras. "Is Economizer Cycle Justified for AC Applications" ASHRAE Journal, July 2005.

Stanley A. Mumma, Ph.D., PE., and Jae-Weon Jeong, Ph.D. "Field Experience Controlling a Dedicated Outdoor Air System" ASHRAE Journal, 2005.

Michael F. Taras. "Is Reheat: Which Concept is Best" ASHRAE Journal, December 2004.

Steve P Kavanaugh, Ph.D. "Limitations of SEER for Measuring Efficiency" ASHRAE Journal, July 2002.

Mark Breuker, Todd Rossi, Ph.D., and Jim Braun, Ph.D. "Smart Maintenance for Rooftop Units" ASHRAE Journal, November 2000.

ACRONYMS

IEER – Integrated Energy Efficiency weighted Btuh per Watt according to ANSI/AHRI Standard 340/360-2007
DX – Direct Expansion
DOE/ORNL – U.S. Department of Energy/Oak Ridge National Laboratory
EER – Energy Efficiency Ratio
HCFC – Hydrochlorofluorocarbon
HFC – Hydrofluorocarbon
AHRI – Air-Conditioning, Heating and Refrigeration Institute
R-value – Resistance to heat flow, used in insulation
SHR – Sensible Heat Ratio
%RH - % Relative Humidity
NIST CYCLE_D – National Institute of Science and Technology Vapor Compression Cycle Design software
COP – Coefficient of Performance
p-h – pressure enthalpy diagram
TXV – thermal expansion valve
IPLV – Integrated Part Load Value
FPM – feet per minute
CFM – cubic feet per minute
LS-HXGR – liquid suction heat exchanger accumulator