New Approach to Energy Savings For Rooftop AC

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New Approach to Energy Savings For Rooftop AC

By Detlef Westphalen, Ph.D., Member ASHRAE

Two of the most important HVAC industry issues are energy efficiency and latent capacity. ANSI/ASHRAE/IESNA Standard 90.1-1999, *Energy Standard for Buildings Except Low-Rise Residential Buildings,* set new standards for the minimum energy efficiency ratios (EERs) of unitary air-conditioning equipment in the commercial capacity range. The U.S. Energy Policy Act of 1992 stipulated that these proposed energy levels would become mandatory, subject to a review by the U.S. Department of Energy. Even though that review is not complete, availability and sales of units with higher efficiencies have increased to reflect the changes in Standard 90.1.

Prior to 1999, a standard efficiency 10-ton (35 kW) unit had an EER of 9.0. Today, a unit compliant with Standard 90.1-1999 has an EER of at least 10.1. However, EER is not the full story regarding seasonal energy use of an air-conditioning unit.

The industry also has established an integrated part load value (IPLV), a weighted average of efficiencies for each capacity stage, which is intended to be a better indication of seasonal energy use. However, even the IPLV is not the full story regarding seasonal energy use. Since tests for both EER and IPLV involve 100% recirculating operation, these numbers do not capture the impact of outdoor air and the different ways that outdoor air can be conditioned. The air-conditioning unit discussed in this paper incorporated design features intended to reduce seasonal energy use—while it achieved good EER (10.8) and IPLV (14.1), it saved significant energy in a field test as compared with a conventional unit with the same EER.

In the area of latent capacity, outdoor ventilation rates, which are prescribed by ANSI/ASHRAE Standard 62, *Ventilation for Acceptable Indoor Air Quality,* have been increasing in many cases. This trend has increased the proportion of the cooling load associated with moisture removal (the latent load) in non-arid climates. The reductions in sensible air-conditioning loads that result from using more efficient lighting, improved building materials, and other factors have contributed to increased importance of the latent load.

Conventional unitary air-conditioning equipment has a sensible heat ratio

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The Florida Solar Energy Center's Building Science Laboratory, site of the field test of the prototype rooftop unit.

(SHR) that is typically in the range 70% to 75% at standard ARI rating conditions. In many operating scenarios, the equipment latent capacity is insufficient to adequately dehumidify the air. Since air-conditioning units generally are controlled by thermostats, which respond to sensible loads, the result is excess humidity in the building space. This situation is exacerbated by the tendency of the condensed moisture that remains on the evaporator coil after the compressors cycle off to re-evaporate since the fan must run continuously to provide the required minimum outdoor airflow.[1]

A number of technology options are available to treat moisture loads that exceed the levels typically handled by standard unitary equipment. The available options and those being developed represent a range of sophistication and efficiency. They include reduced airflow, electric reheat, hot gas reheat, hot liquid reheat,[2] heat pipes,[3] evaporator bypass, passive energy recovery (passive desiccants, membrane), variable air volume (VAV), active desiccant systems (solid and liquid), and Cromer cycle.[4] The system design described in this article is an innovative combination of passive energy recovery and VAV intended to be cost effective, energy efficient, and flexible in a broad range of applications requiring low to medium outdoor airflow rates (i.e., offices, retail, etc.).

The project described here addresses improvement of energy efficiency and performance of unitary air-conditioning equipment, an equipment category that is used for cooling in roughly three-quarters of commercial building floor space. TIAX was awarded a DOE contract to develop the proposed energy-efficient air conditioner, with Aaon as the manufacturing partner. While the project's main focus was delivering maximum energy savings benefits at a minimum cost premium, the increasing importance of moisture control in all commercial air-conditioned environments made the focus on this issue also necessary.

Analysis Phase

The project started with examination of energy and cost characteristics of options for reducing energy use. These options included improvement in all key components (compressors, indoor blower, condenser fan, blower and fan motors, heat exchangers), use of a zero-superheat variable expansion valve, total energy recovery for outdoor air, and mechanical subcooling.

The costs and energy savings impacts of each of the design improvement options were calculated. Assumptions and results of this analysis are discussed in the project final report[7] and also in the online version of this article at ashrae.org.

Based on the initial energy and cost analyses, we developed a design configuration incorporating the best design options:

- Increased heat exchanger size to achieve an EER of at least 10.3, consistent with the ASHRAE 90.1-1999 requirement for 10-ton electric-heat rooftop units;
- Variable air volume using an induction motor and inverter;

- Energy recovery wheel (ERW); and
- Economizer.

The design (called Configuration 1) is summarized in *Table 1*. We also tested two additional configurations. The first of these, Configuration 2, used a tandem scroll compressor set including one variable-speed compressor. The second, Configuration 3, used a microchannel condenser and a tandem compressor set consisting of two single-speed compressors.

We used aggressive interlacing in the design of the fin/round-tube heat exchangers to achieve highly efficient part-load performance with a separate refrigerant circuit serving each compressor. This design approach involves spreading out the circuits serving each compressor so that most of the heat exchanger's fin surface is used when just one compressor operates. Since this approach is not possible with microchannel heat exchangers, we used the tandem compressor set for Configuration 3, allowing refrigerant to flow during single-compressor operation through the entire condenser but only through one of the two conventional evaporator circuits.

	High-Efficiency Design (Configuration 1)
Compressors	Two 5-ton ZP54 scroll
Heat Exchangers: Type	**Fin/Round Tube**
Condenser:	
Face Dimensions, in. (mm)	46 x 72 (1168 × 1829)
Area, ft^2 (m^2)	23 [2.14]
Depth, in. (mm)	1.8 [46] (2 Rows)
Evaporator:	
Face Dimensions	40 × 42 (1016 × 1067)
Area	11.7 (1.09)
Depth	2.7 (69) (3 Rows)
Target COP [EER (Btu/W-h)]	10.3 (3.02)
Expansion Device	Variable Expansion Valve
Energy Recovery Wheel	Airxchange ERC-3020[1]
Main Blower: Wheel	Acme 185 BC Plenum Fan
Motor	Three-Phase 208V 3 hp Motor
Variable Frequency Drive	Baldor 15J Minidrive
Exhaust Blower	Acme 150 BC Plenum Fan Three-Phase 208V 1 hp Motor
Condenser Fans (2)	Revcor KH2404-29 Three-Phase 208V hp Motor
Dampers: Model Number	Ruskin CD60
Face Dimensions, in. (mm):	
	34 × 7.5 (864 × 191)
Outdoor	36 × 20 (914 × 508)
Return	36 × 14 (914 × 356)
Damper Actuators	Belimo LM245R (Proportional)

1. 30-in. wheel size, design airflow 2,000 cfm at 1 in. w.c. pressure drop.

Table 1: Energy efficient AC unit design summary.

The cost of the energy-efficient design was calculated using the same methodology we used to evaluate design options. We estimated the manufacturing cost increase for the unit as $1,200, a 40% increase over the baseline 9 EER unit. However, the capacity enhancement for the ERW was estimated for a small office building with about 20% outdoor air to range from 15% to 35%. Hence, the net cost premium for this application ranges from 5% to 20% depending on climate. Cost premium as compared with a current Standard 90.1-1999-compliant unit was not evaluated but would be expected to be less.

VAV Control

The preliminary analysis showed that the energy impact of VAV is considerable, supporting our decision to incorporate it in our design. However, a concern regarding implementation of conventional VAV approaches was that controlling the supply air temperature by staging two single-speed compressors would have been quite coarse, resulting in supply temperature fluctuations. Other refrigerant-side control approaches, such as hot-gas bypass and suction line throttling, were considered undesirable because they did not meet our focus on energy. Airside control involving air bypass to the return was rejected for the same reason.

We developed a new approach to VAV that uses modulation of the blower rather than of cooling to control the supply air temperature. *Figure 1* illustrates this approach, called "reverse VAV." It reverses the way in which air volume is adjusted. Under conventional control, space temperature sensors affect the throttling of terminal box valves, and the blower is modulated to deliver the required flow while maintaining desired duct pressure. The cooling coil capacity is adjusted (through compressor cycling or modulation, or chilled water valve modulation) to control the supply air temperature.

In the reverse approach, the compressor plant operation (staging or modulation, depending on the compressor) is adjusted based on space temperature, and the blower responds by adjusting airflow to maintain a desired supply air temperature. One of the striped arrows in *Figure 1* shows that terminal box zone control can still be adjusted on the basis of space temperature. The second striped arrow illustrates the adjustment of airflow to respond to space humidity conditions. In practice, this can be done by adjusting the desired supply air temperature setpoint. For example, the setpoint would be allowed to modulate from 7.2°C (45°F) to 15.6°C (60°F). When the space needs more dehumidification and less sensible cooling, the setpoint would be reduced. The blower speed would drop to reduce airflow so that the lower setpoint can be achieved. When the space requires less dehumidification, the supply air setpoint is increased, leading to an increase in airflow. This approach allows decoupling of sensible and latent control of the space, allowing for use of reduced airflow to enhance dehumidification when it is appropriate, but operating with high airflow when there is less moisture load.

The benefits of reverse VAV, particularly in unitary systems, are as follows:

• *Cost Savings*. Blower speed rather than compressor modulation is used to set the supply air temperature. Options for modulating blower speed are available off the shelf at reasonable cost and are already part of any VAV installation. This avoids the additional cost of modulation for the compressor. The approach allows for implementation of VAV without the need for expensive VAV terminal units. Better zone control is possible with the use of conventional VAV terminal units, but could also be provided with less expensive zone dampers, or, depending on zone diversity, with no dampers.

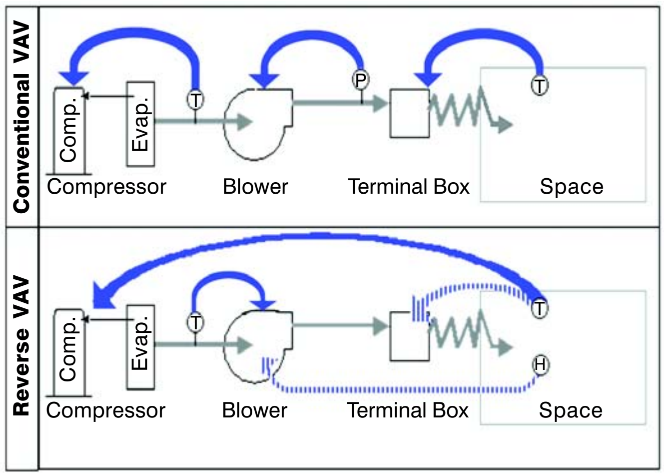

Figure 1: Contrast of conventional VAV and reverse VAV.

• *Humidity Control*. Most conventional systems only provide temperature control for the building, but this approach attempts to control both temperature and humidity. The airflow rate is modulated to better match the sensible heat ratio (SHR) required by the conditioned space.

• *Ease of System Upgrade.* The reduced zone airflow control requirements allow this VAV system to be retrofit into buildings that currently have constant air volume (CAV) systems without the need for the entire air-distribution system to be modified. This represents a significant market opportunity because it allows much easier system upgrade when an old inefficient unit is replaced.

Energy Recovery Wheel Integration

The air-conditioning unit design was intended for applications with moderate percentages of outdoor air. The energy recovery wheel (ERW) was selected with a design flow of 2,000 cfm (944 L/s), up to about half of the unit's maximum total airflow. While this is a higher flow than would be considered moderate, the selection allows for a broader range of system applicability and it results in reduced blower power input. The wheel incurs 1 in. w.c. (249 Pa) of pressure drop at the design flow rate.

Although the concept of recovering energy from exhaust air to precondition incoming outdoor air is straightforward, integrating the ERW into the unit can be complex. Some of the issues are illustrated in *Figure 2*. First, an exhaust blower is required to draw the exhaust air through the wheel, since the wheel's pressure drop is not negligible. The pressure drop on the outdoor air side of the ERW can be provided by the main blower or by an additional outdoor air blower. We used the main blower in this project. This may require that throttling occur in the return dampers to ensure that equal pressures of outdoor and return air enter the mixing box, depending on the pressure in the return plenum.

We mitigated the energy penalty associated with return damper throttling by using VAV. However, because the unit operates with VAV, the positions of the return and outdoor dampers must be modulated as the main blower speed modulates to ensure the constant flow of outdoor air through the

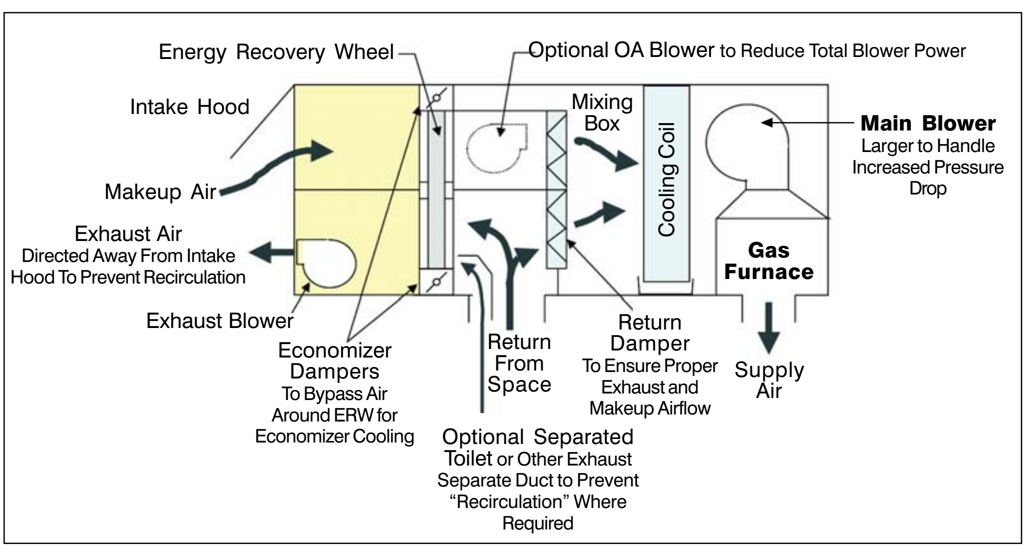

Figure 2: Integration of an energy recovery wheel in an air-conditioning unit.

Unit Configuration	1	2	3
Compressors	Two 5-ton SS	Tandem, VS	Tandem, Two 5-ton SS
Condenser	Conventional	Conventional	Microchannel
Evaporator	Conventional	Conventional	Conventional
Temperatures, °F (°C)			
Condensing	117/113 (47/45)[1]		109 (43)
Evaporating	51/51 (10.6/10.4)[1]		51 (10.3)
Capacity Rating Point			
Total Capacity, Btu/h (W)	119,000 (34 900)	112,000 (32 900)	113,000 (33 100)
Sensible Capacity, Btu/h (W)	86,000 (25 200)		77,000 (22 600)
Sensible Heat Ratio	72%		68%
Power Input (W)	11,010	12,100	10,460
EER, Btu/h-W (COP [W/W])	10.8 (3.16)	9.3 (2.72)	10.8 (3.16)
IPLV Btu/h-W (W/W)			
Staged Condenser Fans	13.1 (3.84)	13.8 (4.04)	
Constant Condenser Fans	12.8 (3.75)		14.1 (4.13)
1. The two temperatures represent the two refrigeration loops.			

Table 2: Capacity, EER, and IPLV test results.

	Conventional Unit		High Efficiency Prototype	
	Standard Unit at 3,600 cfm	Tested Unit at 4,500 cfm	Tested Unit, With Hot Liquid Reheat, at 4,500 cfm	Configuration 3 (Microchannel Condenser)
Net Capacity, Btu/h (W)	131,000 (38 400)	132,300 (38 800)	125,500 (36 800)	113,000 (33 100)[1]
Sensible Heat Ratio	64%	69%	60%	68%
COP, Btu/h-W (EER)	10.8 (3.16)	10.7 (3.14)	10.3 (3.02)	10.8 (3.16)
IPLV, Btu/h-W (W/W)	11.6 (3.40)	Not known	Not known	14.1 (4.13)
Curb Footprint, L × W, in. (m)	86 × 84 (2.2 × 2.1)		128 × 50 (3.3 × 1.3)	
Max. Dimensions, in. (m)	86 × 84 × 45 (2.2 × 2.1 × 1.1)		128 × 87 × 52 (3.3 × 2.2 × 1.3)	
Weight, lb (kg)	1,725 (784)		1,782 (810)	
Evaporator				
Face L × W × D, in. (m)	70 × 36 × 3 (1.8 x 0.9 × 0.076)		42 × 40 × 2.7 (1.1 × 1.0 × 0.069)	
Face Area, ft^2 (m^2)	17.5 (1.63)		11.7 (1.08)	
Condenser				
Face L x W, in. (m)	82 × 40 (2.1 × 1.0)		72 × 46 (1.8 × 1.2)	
Face Area, ft^2 (m^2)	21.7 (2.01)		23 (2.14)	
Number of Condenser Fans	3		2	
Fan Blade Dia., in. (m)	22 (0.56)		24 (0.61)	
1. Not including impact of energy recovery wheel.				

Table 3: Comparison of conventional and new design units.

ERW. The pressure drop across the ERW is used as an indicator of flow rate. As blower speed drops, the outdoor damper must open more and the return damper must throttle more, since the return duct pressure drop gets lower while ERW pressure drop remains constant.

Another issue is how to incorporate economizing. A large amount of outdoor air must flow during economizing operation, which would incur excessive pressure drop across the ERW. Therefore dampers are needed to allow air bypass around the ERW for both the outdoor air and the exhaust air. An additional possible operating mode occurs when the ambient air is too warm to use for economizing but not warm enough so that compressor power savings are greater than blower power increase when using the ERW. We did not incorporate this operational mode in the tested prototype.

Laboratory Testing

We conducted laboratory testing primarily to evaluate capacity, EER, and IPLV for the conventional configuration as compared with the variable-speed and microchannel configurations. The unit configurations and the test results are summarized in *Table 2*. The conventional unit design, with an EER of 10.8, exceeded the 10.3 target. The high IPLV levels result from the aggressive heat exchanger interlacing. Note that IPLV was better with staged condenser fans. This is also the result of the condenser interlacing, which allows the refrigerant to transfer heat to all of the reduced condenser airflow during part load.

Configuration 2, which used the variable-speed tandem compressor, had poor EER but good IPLV. This is because the variable-speed compressor efficiency was very good at low speed but very poor at high speed. For the ARI capacity rating point, the variable speed compressor's input power was about 1 kW higher than that of the single-speed compressor of the tandem set. Configuration 3, with the microchannel condenser, had reduced capacity with EER equal to that of Configuration 1. The use of a microchannel condenser and conventional evaporator required the use of a receiver for the tested unit to avoid pressure fluctuations associated with charge mismatch of the heat exchangers. When operating with a receiver, the hot liquid has nearly zero subcooling, which reduced capacity, but the condenser is used entirely for condensing, which helped further reduce the condensing temperature and compressor power input. The IPLV for the microchannel configuration was very high, due to the use of the tandem compressor set for this configuration. Use of sepa-

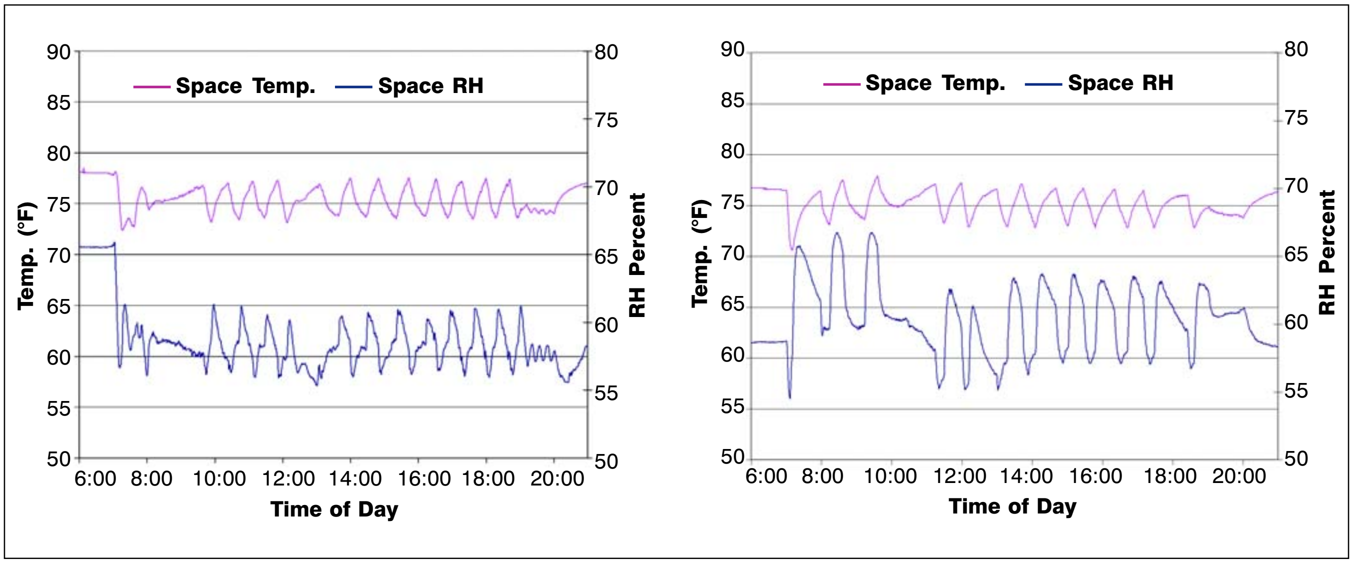

Figure 3: Comparison of prototype (left) and conventional unit maintenance of space conditions.

rate compressor loops with microchannel heat exchangers would have resulted in less impressive IPLV, since the available microchannel technology did not allow interlacing.

Field Testing

We conducted a field test at Florida Solar Energy Center's (FSEC's) Building Science Laboratory to compare the performance of the prototype rooftop unit with a conventional unit that uses hot liquid reheat, which is an energy-efficient approach to enhancing latent capacity. We chose a conventional unit with capacity greater than 10 tons to demonstrate the capacity enhancement of the ERW. The units are compared in *Table 3.*

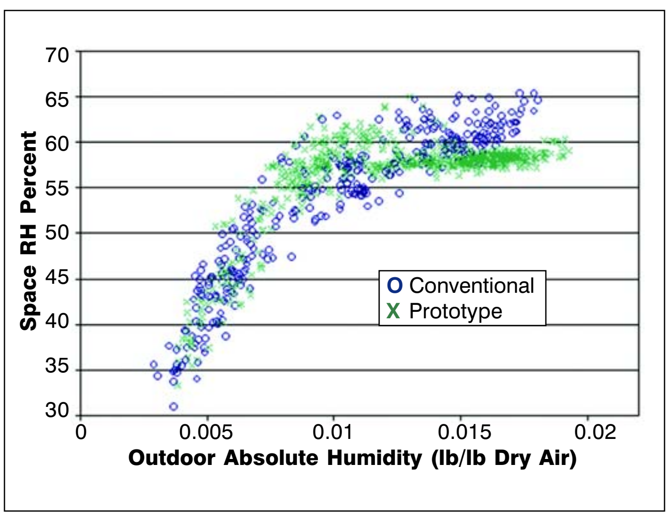

Figure 4: Humidity control comparison.

We connected both units to a common ducting system serving the building, with shutoff dampers that would allow switching between the two units. The conventional unit was set up to cycle the hot liquid reheat based on a space humidistat. The facility is a highly instrumented test building with simulated internal loads that allowed careful comparison of the performance and energy use of the two units. The Florida location was chosen not only because of the FSEC facility, but also to allow testing of the dehumidification capabilities of the new design. The test building was set up to simulate a small office building typical of Florida construction. The outdoor ventilation flow was 840 cfm (396 L/s), and the units were set up for a 4,500 cfm (2124 L/s) maximum total airflow. Estimated total peak load (including ductwork thermal losses) was 32 100 W (109,400 Btu/h) with a 64% SHR. We established a schedule for activation of the loads and operation of the HVAC equipment to represent the office application. Testing was conducted between August 2002 and January 2003.

The cooling performance of the two units was comparable. *Figure 3* shows the performance on two days with similar ambient conditions. Since both units had two single-speed compressors, they both provided cycling control of the space temperature. The humidity control of the prototype unit was tighter, due to the ability to adjust airflow rate to respond to dehumidification needs. Even so, humidity fluctuations were not entirely avoided. The airflow modulation control was set up with moderate-speed response to avoid instability. Hence, the system does not immediately reduce airflow and bring space humidity back down. The latent performance of the prototype unit was not sufficient to prevent rise of the space relative humidity during single-compressor operation for the most humid conditions. It is anticipated that further optimization of the control system would help alleviate these fluctuations.

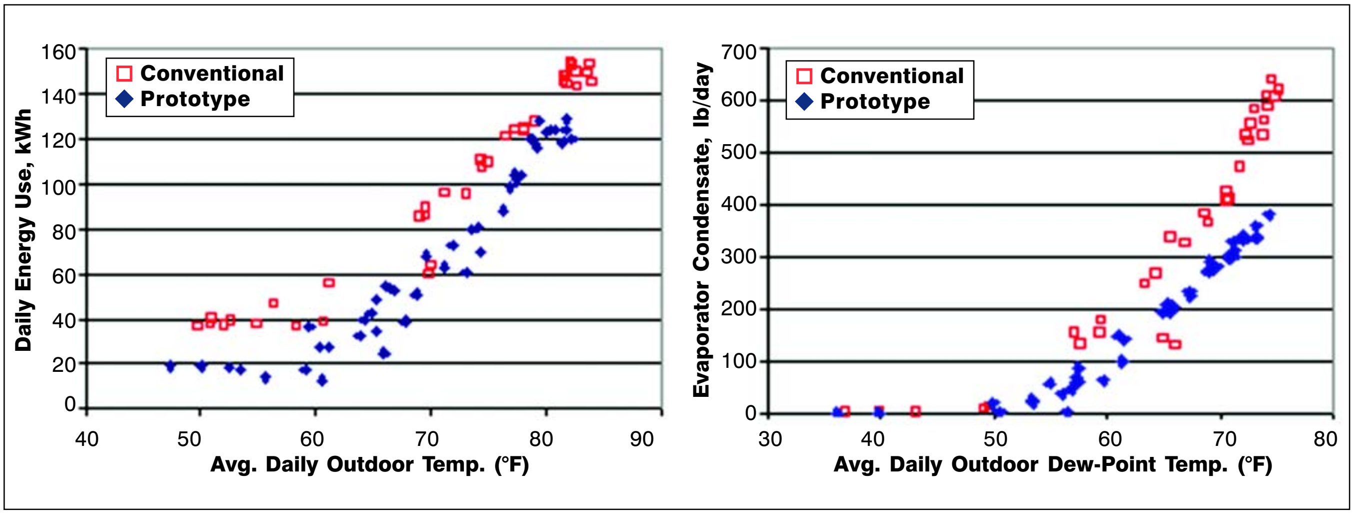

Figure 5: Daily energy use and evaporator condensate comparison.

Figure 4 compares humidity control of the two units. It shows that the prototype unit provided better dehumidification (space relative humidity is reduced by 3% to 5%) in warmer conditions when two compressors were likely to be operating. For less severe conditions, in which a single compressor was likely to be operating, the prototype unit humidity control was not as good, reflecting the part-load issues discussed above. For less severe conditions, humidity levels were lower, reflecting added dehumidification provided by the outdoor ventilation air.

Energy use of the two units is shown in *Figure 5*. For the test period, the energy use of the prototype unit was about 25% lower than that of the Conventional unit. While this might be expected based on the IPLV ratings of the units (14.1 vs. 11.6), the IPLV doesn't tell the whole story, since it is measured with the unit operating at full airflow in all-return mode. The part-load efficiency of the refrigeration circuit was probably not as good in the field as the 14.1 IPLV suggests, because of the tendency of the unit's airflow to be reduced at part load to maintain latent performance. However, use of the ERW is not incorporated in the IPLV, and the ERW is clearly a factor in reducing the energy use of the prototype unit. Daily measurements of condensate collected from the units' evaporator drip pans, shown in *Figure 5,* show the significant latent load contribution of the ERW.

Conclusions

The energy-efficient rooftop unit design took advantage of the most cost-effective design options, resulting in better performance than that of a conventional 10.8 EER rooftop unit while using about 25% less energy. This work illustrates the point that EER is not the full story regarding system energy use. The design options that contribute most to the unit's energy savings are VAV, an energy recovery wheel, heat exchanger interlacing, and the microchannel condenser. The use of a new reverse-VAV approach for system control provided good humidity control during the field test and offers a way to use VAV in small-to-medium size unitary systems. The applied cost premium for the unit, as compared with current typical air-conditioning units, is expected to average about 10% if the unit is manufactured in sufficient quantity. This system development addresses the current industry emphasis on improved humidity control and reduced energy use without significantly increasing cost. In addition, the work has helped illustrate the individual benefits of the key technologies incorporated into the design.

Acknowledgments

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