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Field Test and Performance Verification: Integrated Active Desiccant Rooftop with Heat Pump Capability

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ABSTRACT:

FIELD VERIFICATION OF AN IADR WITH HEAT PUMP CAPABILITY

This report summarizes the results from an investigation that involved field performance verification of integrated active desiccant rooftop (IADR) systems designed to incorporate an advanced heat pump capability. Three such systems were involved, one installed to condition the Oak Ridge National Laboratory's (ORNL) Cooling, Heating, and Power (CHP) laboratory facility, one installed at a South Florida chain pharmacy, and one designed as a split system and installed within the SEMCO research and development test facility in Columbia, Missouri.

The IADR system was described in detail in a previous report entitled *Desiccant Based Combined Systems: Integrated Active Desiccant Rooftop Hybrid System Development and Testing* (ORNL/SUB-01-4000010402). Some of the unique design enhancements integrated into this system, such as extended coil surfaces, variable speed compressors, and direct digital controls (DDC), provided an excellent opportunity for integrating a high-efficiency heat-pump option with the ability to process high outdoor air quantities at unusually low ambient conditions.

All the important operating points associated with these systems were monitored remotely and trended for more than one heating season. The hybrid system was able to maintain the space temperature as desired while delivering the outdoor air ventilation rate required by ASHRAE Standard 62. The overall energy efficiency of the system, measured over a wide range of operating conditions, was found to be very high.

A novel defrost cycle was developed that minimizes the frequency of the cycle and avoids the delivery of cold air to the occupied space associated with traditional heat pump systems.

1. BACKGROUND

As described in depth by the previous report, *Desiccant Based Combined Systems: Integrated Active Desiccant Rooftop Hybrid System Development and Testing* (ORNL/SUB-01-4000010402), the IADR product was developed to independently control both temperature and humidity delivered to an occupied space, during the cooling season, while accommodating up to 100% outdoor air. The system has the unique advantage of being able to process any percentage of outdoor air and adjusting to the load requirements of the space by varying the sensible heat ratio (SHR). The need for improved cooling season humidity control (dehumidification), especially when high outdoor air volumes are required, is essential for facilities in hot and humid climates designed to meet the ventilation requirements set by ASHRAE 62 and the energy-efficiency requirements set by ASHRAE 90.1.

During the heating season, the IADR system will often be required to act as a heating unit as well. This is especially true if the system is installed as a dedicated outdoor air system (DOAS) where it operates with all outdoor air. Traditionally, this heating is accomplished through the use of either an indirect gas-fired heat exchanger or electric-resistance heat. Each method presents a performance disadvantage, especially when high outdoor air volumes are required.

Indirect gas-fired options typically do not offer the modulation necessary to accommodate high outdoor air percentages. This presents two potential problems. One is the introduction of very warm air during moderate outdoor air conditions. Since minimum run times are required to protect the life of the heat exchanger, this can result in a wide fluctuation in space temperature. The other is that during the heating season design days, undesirably cold air can be “dumped” to the space when the heater is cycled off but the supply fan continues to run as required to deliver the necessary ventilation air.

The use of electric-resistance heat has the obvious disadvantage of extreme operating costs due to its very low operating efficiency. At typical energy costs, a BTU of electric-resistance heat will cost approximately twice as much as the indirect-fired gas heat approach assuming an 80% efficient heat exchanger. Unless a high-end electric heating control method [i.e., silicon-controlled rectifier (SCR)] is employed to modulate the heating output, the electric-resistance heating approach can result in the same dumping of cold air associated with the indirect-gas approach.

Another important market-driven factor is that since hot and humid climates have, by definition, relatively mild heating seasons, electrical utilities have been extremely successful convincing their customers to design around all electric facilities (i.e., use electric resistance heat) in exchange for favorable electric rates. Although the heating season is short, the heating cost per BTU is extremely high with this heating method so annual energy costs can be significant. This is supported by the research findings resulting from a DOE-sponsored demonstration site. Figure 1 shows the IADR system installed at a major chain pharmacy in Florida where the IADR’s heat pump performance has been verified. This is discussed in more detail in Sect. 2.2.

Fig. 1. The IADR system installed at a major chain pharmacy where the heat pump performance has been verified.

To address the need for heating by the IADR and to accomplish a stated development objective of high operating energy efficiency, the necessary parts were combined with novel control capabilities to integrate a heat pump cycle into the refrigeration portion of the hybrid system. This option offers an excellent fit to the hot and humid climates most likely to employ the IADR since they inherently have moderate heating seasons (heating design > 20°F).

1.1 IADR Design for Dehumidification Benefits Heat Pump Integration

Since the IADR hybrid system includes an advanced vapor compression cooling section, only a few additional parts are required to complete the infrastructure for an effective air-cooled heat pump system. A simplified heat pump piping schematic is provided as Fig. 2 to show the major components required for heat pump operation.

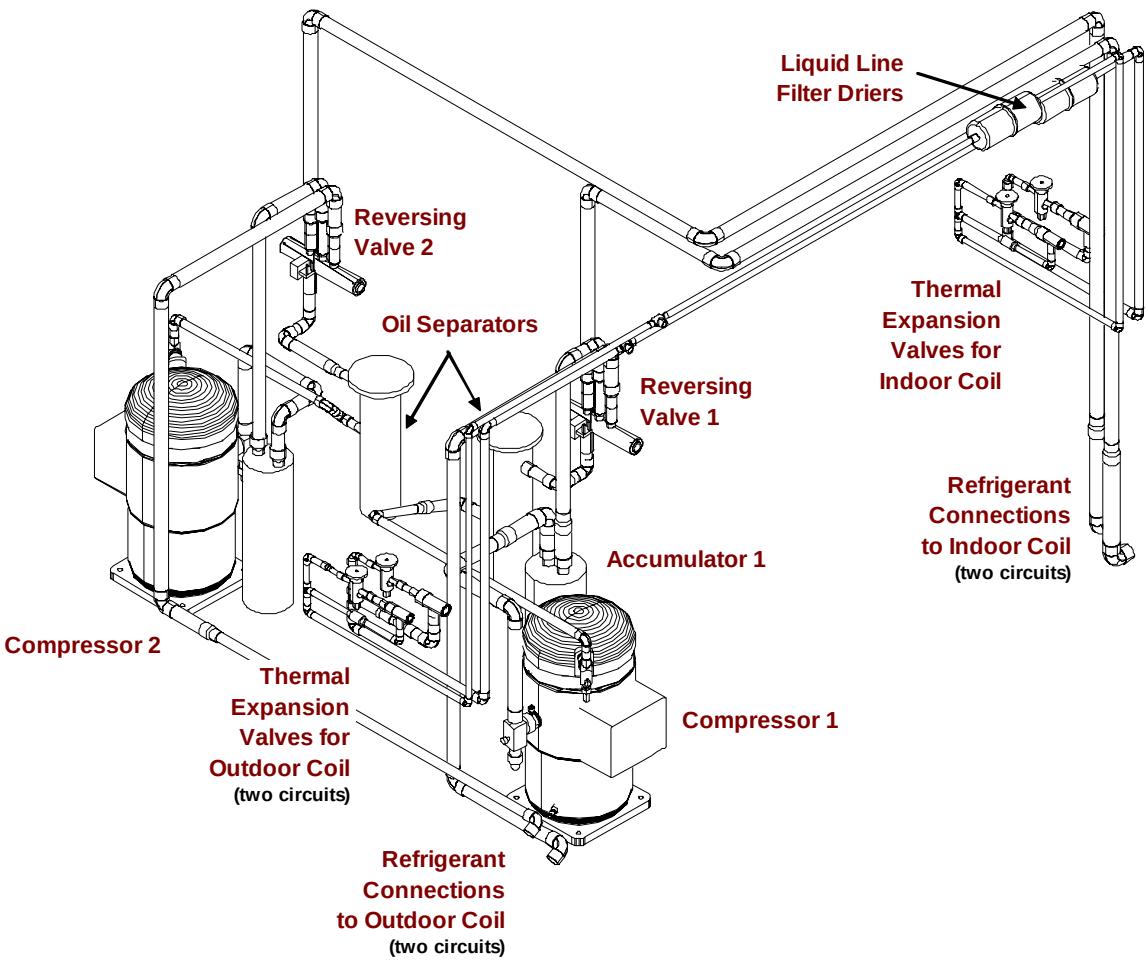

Fig. 2. A schematic showing the major components required by the heat pump cycle integrated into the IADR hybrid system (outdoor and indoor coils are not shown).

To accommodate 100% outdoor air operation in the dehumidification/cooling mode, the cooling side components (coils, compressors, etc.) designed into the IADR have a much greater capacity per cubic foot per minute (cfm) of airflow processed than traditional packaged equipment. Traditional packaged systems are designed for approximately 400 cfm/ton of cooling output while the IADR is designed for approximately 175 cfm/ton. This design approach results in the size of the evaporator coil, condenser coil, condenser airflow, and compressor capacity being large relative to the airflow delivered. This design approach is highly beneficial for heat pump operation.

1.1.1 Compressor Capacity

In addition to the high ratio of compressor tons to airflow processed, a high heating output results because all the energy output during heat pump mode is translated into temperature increase. During cooling mode, part of the energy produced is allocated to condensing moisture (latent load). This latent portion is significant (say 50%) if all outdoor air is being processed. This is not the case in heat pump mode.

Another factor enhancing performance in heat pump mode is the compressor heat, which degrades cooling capacity but increases heat output during heat pump mode.

1.1.2 Outdoor Coil and Airflow Capacity

During heat pump operation, the outdoor air coil becomes the “cooling coil” while the indoor coil becomes the “heating coil.” Since the heating performance is

directly affected by the difference between refrigerant temperature and ambient air temperature, and since ambient air temperature can be quite low during the heating season, it is advantageous to keep this temperature difference to a minimum. This is accomplished by maximizing the outdoor coil surface area and operating with the highest possible outdoor airflow.

High outdoor coil surface area and low temperature differentials are also important to minimize the likelihood and duration of frost formation on the outdoor coil. When the outdoor ambient humidity content is above the dew point of the outdoor coil surface, and if this surface temperature is below freezing, frost can slowly accumulate on the coil surface. By reversing the refrigeration flow and temporarily reverting back to cooling, the outdoor air coil heats up and melts the frost. Since cold air leaves the indoor coil when heating is actually needed, it is advantageous to minimize both the frequency and duration of this defrost cycle.

2. MULTIPLE INSTALLATION SITES USE TO OBTAIN PERFORMANCE VERIFICATION

Three separate IADR systems were investigated at different locations to benchmark the heating performance resulting from the integration of a heat pump option. The first system installed to condition the ORNL CHP laboratory in Oak Ridge, Tennessee, provided an excellent opportunity to observe heating performance at high outdoor air percentages and in a relatively cold environment.

The second site selected was an existing pharmacy in St. Petersburg, Florida, to provide data from a hot, humid climate during continuous actual operating conditions.

The third system was installed within the SEMCO test laboratory in Columbia, Missouri, allowing a wide range of tests to be completed since airflows and indoor coil entering conditions could be modulated as desired. Some basic details are provided for each of these installations.

2.1 ORNL's CHP Test Lab

An IADR system with a nominal cooling capacity of 12.5 tons was installed to condition the CHP Laboratory at ORNL. The system was operated to provide approximately 3200 cfm of supply airflow, which comprised between 50% and 100% outdoor air, depending on the facility's pressurization needs. A small fume hood exhausted approximately 1000 cfm, and the building envelope was not well sealed. This new system was installed during July 2004, and it has been operated to handle the outdoor air sensible and latent cooling loads, all of the space latent load, and most of the space sensible load. An existing split system and hot water space heaters supplement the capacity of the IADR when needed. The 99% heating design condition for this location is 19°F according to the 2001 ASHRAE Fundamentals. Traditionally, heat pumps would not be successfully applied to facilities with this climate, especially if high outdoor air percentages were required.

Fig. 3. Photos of the IADR installed at the ORNL CHP laboratory.

2.2 Major Pharmacy Verification Site in South Florida

An IADR system with a nominal cooling capacity of 17.5 tons was installed to condition a facility owned and operated by a major pharmacy chain in St. Petersburg Florida. The system was operated to provide approximately 5,000 cfm of supply airflow of which 1800 cfm was outdoor air used to pressurize the facility. It also met the ASHRAE 62 criteria requirement for a ground level retail facility, which requires outdoor air at the rate of .3 cfm/square foot of sales area. At startup, the pressurization within this facility was measured and the airflow quantity was maintained at .015” of positive pressure. Before the installation of the IADR system, all the outdoor air dampers on the existing five previous rooftop units had been locked in the closed position in an attempt to improve space humidity control. The dampers had also been closed to minimize the winter heating cost associated with the electric resistance heat mounted in each unit.

This new system was installed September 2004 and has been operated to handle the outdoor air sensible and latent cooling loads, all of the space latent load, and approximately 40% of space sensible load. Three additional rooftop units that operate without outdoor air supplement the sensible cooling and heating capacity of the IADR as needed.

The 99% heating design condition for this location is 47°F according to the 2001 ASHRAE Fundamentals. This moderate heating climate makes this site an excellent candidate to employ an effective heat pump technology since, as will be reported later, it is approximately four times more efficient than the electric resistance heating approach previously used.

One of the interesting observations made as part of this pilot investigation was that the heating balance point—the outdoor temperature at which the facility calls for heating— was determined to be an unusually moderate 64°F. This resulted from two contributing factors: first, the amount of outdoor air flow required to pressurize the space is high relative to the total supply air flow needed to heat the space; second, the large open cooler sections along approximately half of one wall of the store (see Fig. 4) act to lower the indoor temperature.

Fig. 4. Photo of the IADR installed at the chain pharmacy and the pharmacy's cooler section.

Two years of annual electric consumption was analyzed for this facility once this heating balance point was observed. As a testament to the inefficiency of the electric resistance heat, it was discovered that the electrical use was essentially the same (within 5%) during February as it was during August. This was unexpected since (1) St. Petersburg has the most humid climate in the country and the air conditioning load is therefore extreme, and (2) as mentioned previously, no outdoor air had been introduced by the rooftop units. Had it been, it is likely that the heating season electric consumption would have exceeded the peak cooling season amount.

Figure 5 provides actual field data showing how the space enters the heating mode once the outdoor air temperature reaches approximately 64°F. Note how the supply air temperature becomes very consistent once the outdoor air reaches approximately 56°F. This results from the variable-speed compressor being controlled to supply a uniform supply-air discharge temperature. With outdoor air below 56°F, the space load reaches the minimum load needed to allow for continuous modulation of the variable speed compressor (approximately 3 tons). The control system operates the IADR system as a lead unit allowing it to supply the majority of the space-heating needs. This resulted in a significant reduction in what the owner would have paid to heat the facility with electric-resistance heat had a similar amount of ventilation air been provided.

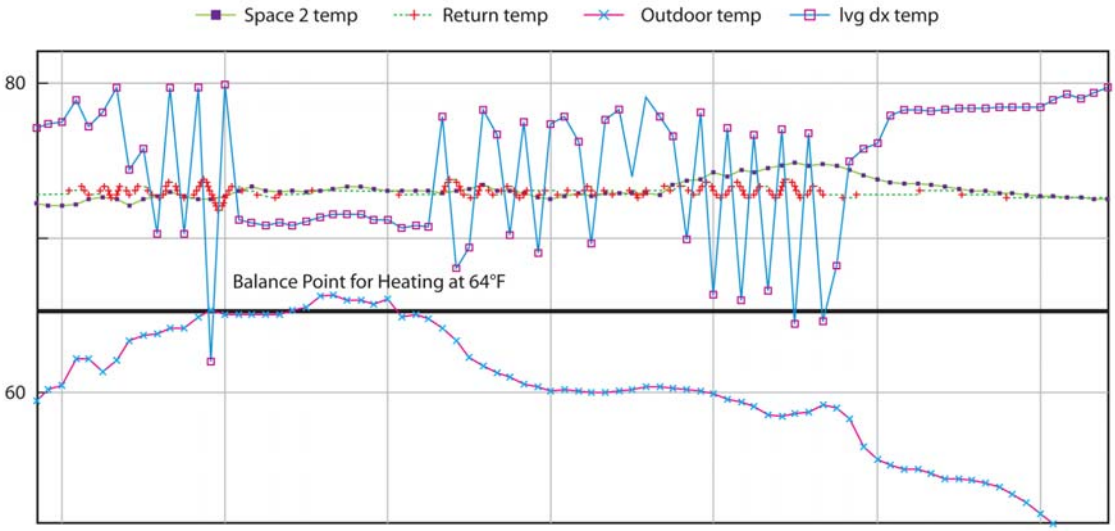

Fig. 5. Sample of actual field performance data during the heating mode operation of the IADR system.

2.3 SEMCO Inc. Research and Development Test Laboratory

An IADR system having a nominal cooling capacity of 12.5 tons was installed to help condition the SEMCO R&D Air Test Laboratory in Columbia, Missouri. The system is operated to provide approximately 3000 cfm of supply airflow, which is comprised of approximately 50% outdoor air.

This system has been operating since November 2004 and has provided an excellent opportunity to test a variety of operating conditions for the IADR system's heat pump capability. The 99% heating design condition for Columbia, Missouri, is 5°F according to the 2001 ASHRAE Fundamentals. This provided the most extreme heating season operating conditions of the three sites investigated. As shown in Fig. 6, the IADR system at the SEMCO test laboratory differed from the systems installed at the other two sites—the SEMCO system was built as a "split system," with the main system located indoors but the condensing section located outdoors.

The extreme outdoor air temperatures and consistent cold conditions that existed for this system due to its location offered the most effective site for optimizing the defrost control algorithms and methods. The application of air-side heat pump technology used to condition a high percentage of outdoor air, as was the case for this test site, is highly unusual and provided an excellent challenge to the performance capabilities of the IADR heat pump integration.

Fig. 6. Photos of the IADR system installed within (main unit) and outside (condensing portion) of the SEMCO R&D Test Laboratory located in Columbia Missouri.

3. PERFORMANCE DOCUMENTED FOR THE IADR HEAT PUMP OPTION

The three sites discussed in the previous section were all operated over at least one heating season to allow data to be collected and archived. Over 45 state points were continuously monitored for each of the systems investigated (see sample instrumentation list in Appendix A). The on-board DDC control system also served as the data acquisition system to archive these data. In addition to the state points monitored by way of digital or analog inputs to the controller, an additional 40 pieces of information relative to the various frequency inverters employed were monitored by way of modbus communications.

The data provided by the monitoring and testing of these three systems were analyzed to optimize operational and control functions and to provide the performance data as summarized by the following sections.

3.1 Initial Modeling with the ORNL Heat Pump Performance Model

ORNL has considerable expertise in heat pump modeling. The laboratory has developed the ORNL Heat Pump Design Simulation model, which has become an industry standard used by many of the large heat pump manufacturers for product development and optimization.

As part of this development program, ORNL was provided the design parameters used for all aspects of the IADR heat pump design. More than 70 input parameters are required for this analysis. This modeling provided various performance parameters of interest including projected heating and cooling capacity, power consumption, coefficient of performance values, supply temperatures, suction temperatures, discharge temperatures, and sub-cooling values.

Two of the more interesting pieces of modeling data provided by the analysis are the projected heating capacity and supply air temperature. Figure 7 shows a summary of the modeled heating capacity output anticipated as a function of outdoor ambient temperature. Shown in yellow is the average of actual data measured as part of the testing completed at ORNL. Excellent agreement was obtained proving the effectiveness of the ORNL HPDM simulation model. The results of this testing were very favorable to the IADR since a high amount of heating output was available even at the very low ambient temperatures experienced at the ORNL test location.

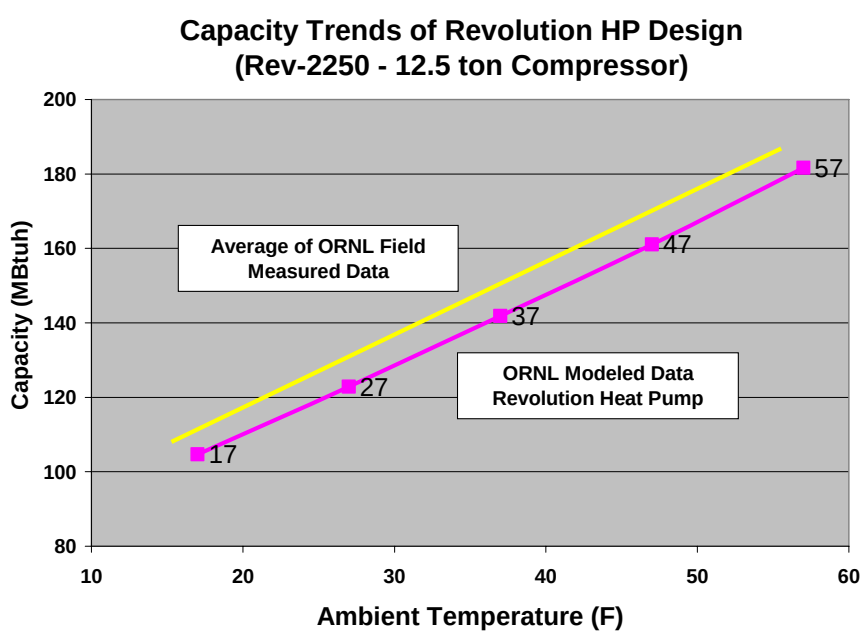

Fig. 7. Actual test data showing heating capacity output as a function of ambient temperature compared with that projected for the IADR heat pump option using the ORNL HPDM simulation model. Based on 3600 cfm of supply airflow and a full recirculation mode (no outdoor air).

Figure 8 shows the projected supply air temperature leaving the IADR system with heat pump integration when operated as a 100% outdoor air system. The supply air temperature is shown as a function of the outdoor temperature and is based on a

Revolution 2250 system operated at its 2250 cfm nominal airflow. These results are based on the ORNL HPDM model and also agree well with actual test data.

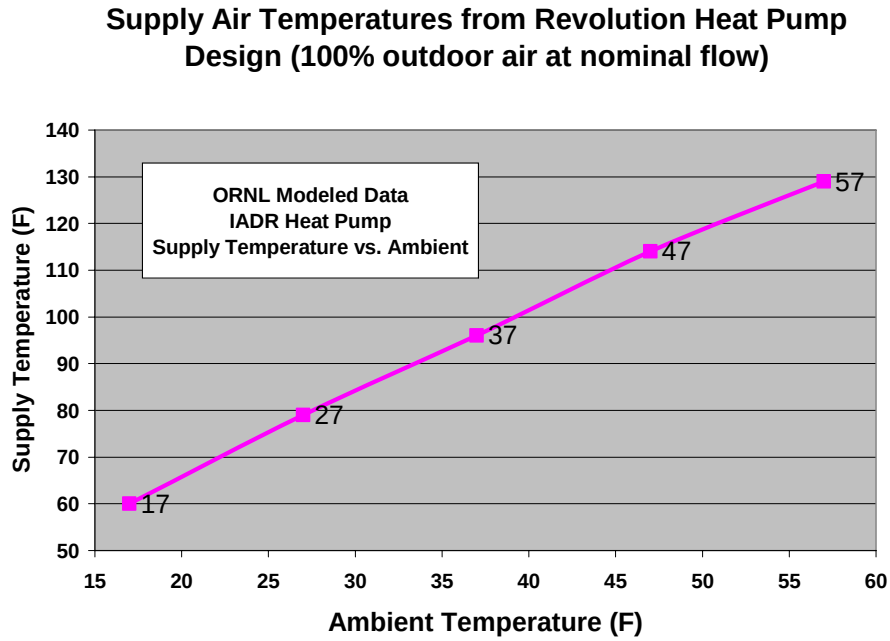

Fig. 8. Projected supply air temperature delivered by IADR heat pump option when operated as a 100% outdoor air system based on the ORNL HPDM simulation model. The Revolution 2250 unit serves as the basis of the analysis and is operated at the nominal 2250 cfm airflow.

These data are interesting in that typical heat pump units are not used for systems in cool climates or those involving high outdoor air percentages. The fact that the IADR system is capable of preheating outdoor air as cold as 20°F to approximately 65°F demonstrates how this system differs from conventional systems and opens up opportunities for users to benefit from the elevated energy efficiency associated with this technology in the heating mode (this is discussed in detail later).

3.2 Heating Data Collected at Pilot Installation Sites

Large amounts of data were collected to map the performance of the heat pump integration into the IADR system. The three sites described previously were all fitted with full instrumentation that was continuously monitored with the resulting data archived. Figures 9a–9c summarize data collected at ORNL from November 2004 though March 2005.

These data show a wide variety of outdoor air percentages, outdoor temperatures, compressor modulation, and delivered heating capacity. The electrical energy consumption was monitored for each component within the system and the heating output was measured by equating the supply airflow and the associated temperature increase.

Good agreement was found between the modeled results and those measured. For example, the 10:45 data point collected on November 26 shows approximately 2,400 cfm of outdoor air being heated from 38°F to 87°F, resulting in a heating capacity output of 124,500 Btu/hour. The compressor was operating at approximately 80%. The ORNL modeling predicts an output of 142,000 Btu/hour at 100% compressor capacity.

Revolution Heat Pump Performance Analysis at ORNL CHP Lab

Time	OA Temp	Airflow (cfm)	Entering Coil Temp	Leaving Coil Temp	Heating BTUs Delivered	Compressor HZ	Compressor % Full Capacity	Total KW Input[3]	BTUs Delivered per KW Input	Energy Cost[4] per million btus
100% Outdoor air										
11/26/2004										
9:30 AM	47	2411	49.8	92.4	110925	35	58%	7.2	**15323**	$4.57
10:15 PM	40	2400	40.9	83.2	109642	41.4	69%	7.9	**13905**	$5.03
10:35 PM	38.7	2402	39.1	86	121666	46.9	78%	8.8	**13771**	$5.08
10:45 PM	38.5	2389	38.5	86.8	124620	48	80%	9.3	**13386**	$5.23
0% Outdoor air										
11/27/2004										
8:46 AM	41	2407	70.6	99.3	74607	25.6	43%	6.5	**11478**	$6.10
40% Outdoor air										
9:42 AM	46	2399	68.8	99.5	79541	24.6	41%	6.7	**11872**	$5.90
60% Outdoor air										
11/28/2004										
7:30 AM	43.1	2411	59.2	93.8	90094	29	48%	6.8	**13249**	$5.28
100% Outdoor air with flow increased by 12.5 % to 2700 cfm										
11/28/2004										
8:13 AM	44	2697	49.9	83.9	99034	29.6	49%	6.2	**15867**	$4.41
8:25:00 AM [1]	44	2700	47.8	82.7	101768	32.9	55%	6.5	**15707**	$4.46
8:30:00 AM [1]	44.3	2706	47.1	85.1	111054	35.3	59%	6.8	**16395**	$4.27
8:40:00 AM [1]	44.4	2721	46.8	86.5	116666	36.9	62%	7.1	**16528**	$4.24
8:58:00 AM [2]	45.1	2680	46.9	88	118960	37.7	63%	7.1	**16696**	$4.19
50% Outdoor air with flow back to 2400 cfm and 75 deg set point										
12/3/2004										
9:05 PM	36.8	2408	61.8	91.6	77499	27.5	46%	6.7	**11571**	$6.05
9:41 PM	36.1	2406	61.8	87.1	65742	22	37%	5.8	**11363**	$6.16
10:22 PM	35	2400	52.5	94.8	109642	50	83%	10.3	**10686**	$6.55
11:52 PM	34	2400	53.4	90.6	96422	36	60%	7.6	**12687**	$5.52

Fig. 9a. Heat pump operation field test data from the ORNL CHP lab installation site.

Revolution Heat Pump Performance Analysis at ORNL CHP Lab

30% Outdoor air, 2400 cfm and 75 deg set point										
12/5/2004										
8:40 AM	36	2413	71.2	98.1	70102	24	40%	5.6	**12518**	$5.59
50% Outdoor air, 2400 cfm and 75 deg set point										
12/5/2004										
9:06 AM	38.3	2395	64	90.6	68804	24.2	40%	5.5	**12510**	$5.60
9:06 AM	38.4	2402	63.2	91.6	73674	25.87	43%	6.05	**12178**	$5.75
50% Outdoor air, 2400 cfm and 75 deg set point										
12/12/2004										
8:30 AM	39.8	2385	62.8	91.9	74956	23.9	40%	6.2	**12090**	$5.79
50% Outdoor air, 2400 cfm and 75 deg set point										
12/14/2004										
8:20 AM	30.7	2444	57.9	100	111124	53	88%	11.1	**10011**	$6.99
10% Outdoor air, 2400 cfm and 75 deg set point										
12/20/2004										
8:28 AM	19.9	2415	61	102.4	107979	50	83%	10.73	**10063**	$6.96
8:41 AM	22	2413	61.5	102.7	107369	49.3	82%	10.43	**10294**	$6.80
10% Outdoor air, 2400 cfm and 75 deg set point										
12/20/2004										
8:53 AM	38.2	2413	71.5	105.5	88605	29.3	49%	7.4	**11974**	$5.85
Note: Fan KW increases due to flow increase and impacts the btu/kw values										
50% Outdoor air, 3000 cfm and 75 deg set point										
1/30/2005										
11:33 AM	40.7	2999	64.2	82.9	60568	19.9	33%	4.8	**12618**	$5.55
30% Outdoor air, 3000 cfm and 75 deg set point										
2/10/2005										
4:45 PM	36.1	3073	60.2	85.7	84630	28.5	48%	6.6	**12823**	$5.46
								5.2	**16275**	
6:05 PM	35	3021	60	85.3	82546	29.7	50%	6.4	**12898**	$5.43
								5.1	**16185**	
0% Outdoor air, 3000 cfm and 75 deg set point (space temp closed damper)										
2/11/2005										
8:00 PM	28	3030	63.8	89.9	85410	40.2	67%	8.3	**10290**	$6.80
								6.9	**12378**	
8:23 PM	29.8	3020	64.4	91.1	87085	36	60%	8.3	**10492**	$6.67
								6.9	**12621**	
50% Outdoor air, 3000 cfm and 75 deg set point										
2/11/2005										
12:50 PM	43	2955	63.3	81	56488	19.9	33%	5.2	**10863**	$6.44
50% Outdoor air, 3000 cfm and 75 deg set point										
2/13/2005										
7:30 AM	41.9	3019	62.8	82.5	64232	19.9	33%	5.5	**11679**	$5.99

Fig. 9b. Heat pump operation field test data from the ORNL CHP lab installation site.

Revolution Heat Pump Performance Analysis at ORNL CHP Lab

70% Outdoor air, 3000 cfm and 75 deg set point										
2/13/2005										
7:45 AM	42	3031	55.5	77.2	71035	23.9	40%	4.96	**14321**	$4.89
7:51 AM	42	3029	54.3	78.8	80147	25.9	43%	5.77	**13890**	$5.04
7:57 AM	42	3027	53.8	79.7	84671	26.3	44%	5.98	**14159**	$4.94
80% Outdoor air, 3000 cfm and 75 deg set point										
2/13/2005										
8:35 AM	42.9	3002	50.8	77.1	85269	26.3	44%	5.88	**14501**	$4.83
8:45 AM	43	3002	51	76.5	82675	25.6	43%	5.58	**14816**	$4.72
100% Outdoor air, 3000 cfm and 75 deg set point										
2/13/2005										
9:35 AM	43	3006	45.4	79.9	112004	36.7	61%	7.2	**15556**	$4.50
9:45 AM	43	2997	45.2	83.1	122673	41	68%	7.6	**16141**	$4.34
9:55 AM	43	3005	45	85.6	131763	44.3	74%	8.9	**14805**	$4.73
50% Outdoor air, 3000 cfm and 75 deg set point										
2/13/2005										
7:45 AM	38.4	3018	61.7	86.5	80834	28.4	47%	6.3	**12831**	$5.46
50% Outdoor air, 3000 cfm and 75 deg set point										
3/2/2005										
7:47 AM	27.4	3005	56.7	91.6	113264	49.6	83%	10	**11326**	$6.18
8:01 AM	28.1	3002	58	92.8	112827	48.3	81%	9.7	**11632**	$6.02
8:21 AM	28.4	3007	58.7	92.9	111067	47.8	80%	8.62	**12885**	$5.43
70% Outdoor air, 3000 cfm and 75 deg set point										
3/2/2005										
9:02 AM	30.9	3005	47.3	89.1	135658	60	100%	11.2	**12112**	$5.78
9:14 AM	31.1	3007	47.4	89.3	136073	60	100%	11.2	**12149**	$5.76
20% Outdoor air, 3000 cfm and 75 deg set point										
3/2/2005										
9:52 AM	37.1	3005	71.9	91.8	64583	22.6	38%	5.5	**11742**	$5.96
40% Outdoor air, 3000 cfm and 75 deg set point										
3/2/2005										
10:52 AM	38	3005	67	87	64908	22.6	38%	5.9	**11001**	$6.36
								Average	**14515**	**$5.52**

1. Raised space set point to 78 deg to increase compressor HZ output and to eliminate compressor cycling
2. Steady state condition, no cycling - good data point
3. The KW shown corrects for 5% inverter inefficiency
4. The cost of operation is based on an electrical cost of $.07/KWH

Fig. 9c. Heat pump operation field test data from the ORNL CHP lab installation site.

3.3 Contrasting Indirect Gas-Fired Heating and the IADR Heat Pump

The heat pump option offers two distinct advantages when compared to indirect gas-fired heating (which the IADR also can provide). Both were documented as part of this research.

The first advantage is the stability of the delivered air temperature from the system. Figure 12 shows the stability of the supply air leaving the IADR when in heat pump mode versus the temperature fluctuation associated with a common gas-fired furnace. This figure shows the indirect gas-fired heater operating with minimal outdoor air and thereby under the best possible conditions for temperature stability. Had a higher outdoor air percentage been involved, two stages of heat would have been required, and the temperature being delivered would have fluctuated much more. The stability of the delivered temperature associated with the heat pump is independent of the outdoor air percentage (with the exception of the defrost mode, which will be discussed later) due to the exceptional control capabilities offered by the variable-speed compressor.

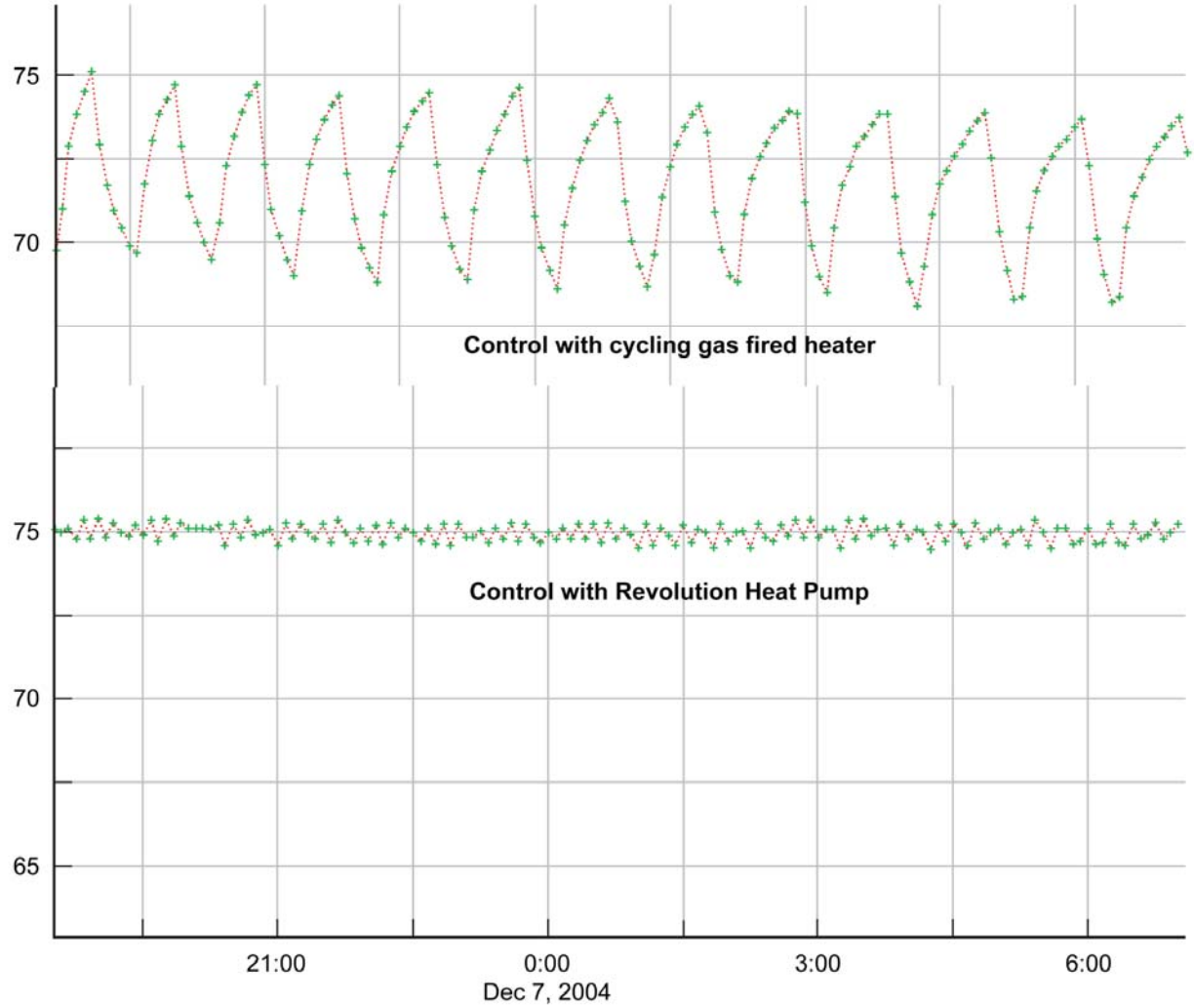

Fig. 10. Field data showing the improved stability of the supply air temperature associated with the heat pump approach as compared to a typical indirect gas-fired heater.

The second advantage is that operating costs are lower. This is particularly true now, when natural gas prices are at record levels and the higher cost of primary energy has not yet been fully reflected in electric rates. As shown by Figs. 9 through 11, the average cost for creating 1,000,000 Btus of heating output using the IADR heat pump option is between $5.50 and $6.00 at ORNL assuming an electrical cost of $.07/KWH. Indirect-fired gas, assuming a 78% efficient heat exchanger and gas costing $8/million Btu results in a delivered cost of approximately $10.2/million Btu.

Hot and humid climates have, by definition, relatively moderate cooling seasons. As a result, they are excellent candidates for using this heat pump integration since the heating loads can be easily accommodated while offering the end user the opportunity for significant energy savings.

As previously reported for the pharmacy site in Florida, it is very common for buildings constructed in hot and humid climates to be heated with electric-resistance heat. Using the heat pump approach reduces the cost of heating from what would be $20.5/million Btu to $6/million BTU, a 71% reduction. As shown by the pharmacy example discussed previously, using the electric resistance heating option will result in very high heating costs despite the moderate heating climate.

3.4 Managing the Defrost Cycle

One of the main barriers to heat pump use, especially where high outdoor-air percentages are required, is the need for defrost cycles. When operating in the heating mode, the heat pump essentially “flips” the normal cooling cycle so that the outdoor coil acts as the evaporator and the indoor coil becomes the condenser. As the outdoor coil cools the outdoor air, the coil surface temperature frequently drops below 32°F. If the outdoor dew point is above the coil surface temperature, moisture in the outdoor will be condensed, and frost will begin to form. As the frost builds, airflow across the coil will slowly decrease while the frost begins to degrade the transfer characteristics of the coil. The net result is reduced heating capacity and an increased need for the compressor to work harder to meet the desired condition. Therefore, this frost formation must be carefully managed; frost must be removed before a significant degradation in performance occurs (see Fig. 11).

Fig. 11. Photo of outdoor coil of the IADR installed at the ORNL site just before the defrost mode is initiated.

Eliminating frost formation is easily done. By switching back to “cooling mode” through the activation of a “reversing valve,” the outdoor coil once again becomes the condensing coil, heats up, and quickly melts any frost that has formed. The problem during defrost occurs at the

indoor coil. When in defrost mode, the indoor coil cools the supply air to very low temperatures, which is problematic, especially if outdoor air is being delivered by the system. As a result, most heat pump systems currently rely on inefficient electric-resistance heat to operate during the defrost cycle to isolate the occupied space from the cold air leaving the coil.

The IADR incorporates the capability of using the gas-fired regeneration burner and the variable-speed active desiccant wheel to provide an effective indirect gas-fired heating source. A novel approach was devised to anticipate the need for a defrost cycle, then bring on the indirect-fired gas heat for a short duration to maintain an acceptable supply-air temperature. In this way, the defrost control algorithm optimized as part of this research was used to isolate the space from the cold air leaving the indoor coil during the defrost mode.

Figure 12 shows how this control methodology functioned. In this figure, the temperature leaving the indoor coil, the supply temperature to the space, and the temperature entering the regeneration site of the active desiccant wheel are shown. The increase in the temperature entering the desiccant wheel reflects the burner being initiated. When the burner and regeneration fan are not energized, the temperature shown reflects the internal system temperature since dampers are closed. Outdoor air is used for regeneration of the desiccant wheel. This figure shows how the exiting temperature from the coil becomes extremely cold during a typical defrost cycle (approximately 34°F) and how the initiation of the gas burner isolates this from the space (air supplied at 68°F not 34°F).

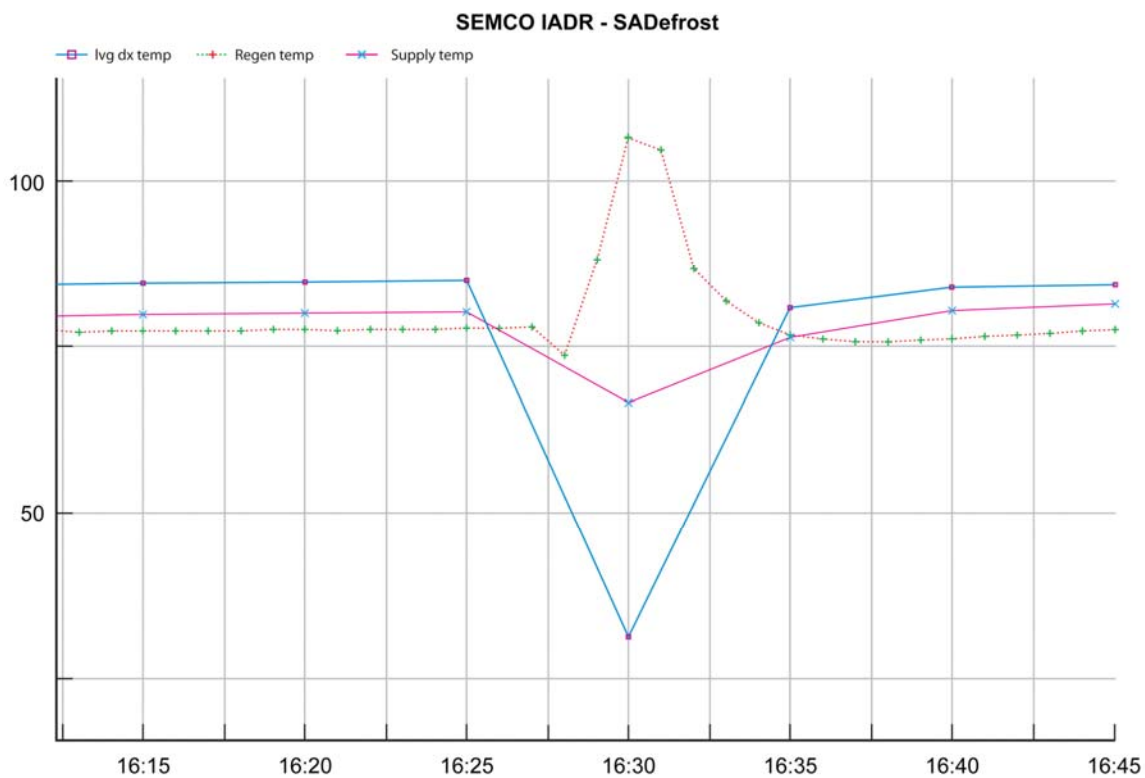

Fig. 12. Data showing a typical defrost cycle and how the gas fired heating capability is used to provide a consistent supply air temperature despite the extremely cold air leaving the indoor coil.

This graphic also shows the typical duration of a well-designed defrost cycle. As shown, the defrost cycle only involved a 10-minute time frame. One optimization that was incorporated as a result of these data was to better anticipate the defrost cycle. In doing so the gas burner would be initiated several minutes earlier so that the -air temperature would remain even more stable. A second optimization that was incorporated into future systems was an increased capacity in the

regeneration heater so that it was sufficient to provide a consistent supply-air temperature even during extreme days. This is discussed further in the technical article “Performance Analysis of Integrated Rooftop with Heat Pump During the Heating Mode” included in Appendix B..

3.5 Optimizing Performance with Advanced DDC Control Algorithms

It would be impossible to operate the heat pump cycle integrated into the IADR system without proven, advanced control algorithms. These control algorithms far exceed the capacity of standard, prepackaged control systems. As a result, a significant portion of the research involved determining the best way to control the heat pump cycle, how to manage the defrost cycle, and which control state points are needed.

Parameters evaluated as part of the defrost algorithm include outdoor air ambient temperature, outdoor dew point, pressure loss across the outdoor coil, outdoor air coil suction temperature, outdoor air coil suction pressure, minimum defrost time, and minimum time between defrost cycles. By monitoring these parameters carefully and having the unique control flexibility offered by the variable-speed compressor, an effective defrost cycle was optimized to provide the maximum heating output with the minimum amount of time operated in the defrost mode.

This feature is critical to the objective of using the IADR as a high outdoor air percentage unit (often at 100% outdoor air) in both moderate and hot and humid climates. Also, by minimizing the amount of time in the defrost cycle, the significant decrease in cost of operation over even gas-fired systems is recognized, which quickly justifies the added cost associated with advanced controls.

4. CONCLUSIONS AND RECOMMENDED FUTURE WORK

These research/demonstration installations involving the integrated vapor compression – active desiccant hybrid system (IADR) with heat pump integration were successful. All original objectives were met, the measured performance matched or exceeded the modeled expectations, and all end users are very satisfied with the results. Attractive heating costs relative to either gas or electric-resistance heating were documented as a result of this system’s high operating efficiency. Owners applying this technology could expect to reduce the cost associated with heating by approximately 40% and 71% compared to indirect-gas and electric-resistance heating respectively.

This design approach will be duplicated and implemented as part of a much larger, newly constructed high school in Georgia, scheduled to be completed in 2006. This project will apply numerous IADR systems that include the heat pump option. These systems will be applied as both dedicated outdoor air systems and complete variable-air-volume (VAV) systems. This opportunity will allow the performance of the IADR and the integrated heat pump cycle to be further optimized, since all systems will be equipped with instrumentation and data trending.

Four of the systems will also be combined with on-site power generation to form an effective CHP site, using the heat of rejection from the engine to regenerate the active desiccant wheels. This heat will also be used for space heating with the heat pump being used for back-up heat as needed.

Further work that would be highly advantageous would be additional demonstration sites located in extreme climates where the heat pump option would satisfy the vast majority of the heating hours with the integral indirect gas capability handling the more infrequent extreme conditions.

ACKNOWLEDGEMENTS

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APPENDIX A

Sample instrumentation list

Outdoor temp
Outdoor RH
Outdoor airflow
Return temp
Return RH
Return airflow
Supply temp
Supply RH
Supply airflow
Entering DX temp
Entering DX RH
Leaving DX temp
Leaving DX RH
Regeneration temp in
Regeneration temp out
System on/off
Supply fan on/off
Compressor on/off
Compressor VFD
Condenser fan on/off
Condenser fan VFD
Regenerator fan on /off
Burner power on/off
Reversing valve (heat pump)
Building pressure
Condenser coil air pressure loss
Desiccant wheel regeneration side delta pressure
Leaving condenser air temp
Starter panel temp
Suction pressure
Condensing pressure
Vapor temp leaving compressor
Electric power to supply fan
Electric power to regeneration fan
Electric power to compressor
Electric power to condenser fan
Electric power to desiccant wheel
Outdoor damper position
Return damper position
Bypass damper position
Natural gas valve position
Supply fan VFD
Regeneration fan VFD
Desiccant wheel VFD

APPENDIX B

PERFORMANCE ANALYSIS OF INTEGRATED ACTIVE DESICCANT ROOFTOP AIR-CONDITIONING SYSTEM OPERATING IN HEATING MODE

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PERFORMANCE ANALYSIS OF INTEGRATED ACTIVE DESICCANT ROOFTOP AIR-CONDITIONING SYSTEM OPERATING IN HEATING MODE

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ABSTRACT

This study describes the performance study of a novel Integrated Active Desiccant-Vapor Compression Hybrid Rooftop (IADR) at Oak Ridge National Laboratory (ORNL) in a heat pump mode. The tests were performed at two different ratios of outdoor/return air. Analysis of performance characteristics under each operating mode, including heating capacity and energy efficiency ratio, are given. Results are also presented while the unit is undergoing defrost. Comparison between the experimental performance of IADR unit and the calculated performance of other commercially available heat pump systems at comparable operating conditions has been done.

INTRODUCTION

More than 95% of all commercial buildings in the United States are conditioned with some type of packaged equipment – mainly rooftop units. This equipment is compact and reliable, but it is often unable to accommodate the increased quantities of outdoor air, which must be continuously delivered to be in compliance with major building codes and the ASHRAE 62 Ventilation Standard [1]. Requirements put on U.S. equipment and appliance manufacturers to meet minimum efficiency standards for their products have been one of the most effective energy-saving policies instituted by Congress and the Department of Energy.

Based on the requirements mentioned above, there is a demand for compact, highly efficient, and cost competitive air conditioning units that can bring in varying amounts of outdoor air for cooling, heating, dehumidifying, and ventilation. The novel Integrated Active Desiccant-Vapor Compression Hybrid Rooftop (IADR) fully meets these requirements. The IADR technology combines the strengths of an advanced direct expansion (DX) cooling or heating cycle, utilizing variable speed compressors and optimal control strategies, with the unique dehumidification capability offered by an active desiccant wheel. It can be applied as a dedicated outdoor air system handling 100% outdoor air, or as total conditioning system, handling both outdoor air and space cooling or heating loads.

NOMENCLATURE

Abbreviations:

DX	= direct expansion
EER	= energy efficiency ratio
IADR	= Integrated Active Desiccant-Vapor Compression Hybrid Rooftop
CHP	= Combined Cooling, Heating, and Power

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Variables:

G	= volumetric air flow rate, scfm or m^3/min
h	= enthalpy, Btu/lb or kJ/kg
Q	= heat input, heating output, Btu/h or kW
W	= electric power, kW
ρ	= density of air, lb/ft^3 or kg/m^3

EQUIPMENT

A general view and schematic of the IADR unit installed at ORNL site showing major components and flow diagram are given in Figures 1 and 2. The vapor compression cycle of the unit uses R-22 refrigerant. Fresh outdoor air enters the system at point "A", where it is mixed with return air from the laboratory building (point "B"), combining to produce the total air flow that enters the DX coil. The ratio of outdoor-to-total air flow is controlled by outdoor air and return air dampers. In the cooling mode of operation a portion of the cool saturated air leaving the DX coil passes through the active desiccant wheel where it is dried to a very low dewpoint, and warmed by the latent heat of vaporization from water adsorbed onto the desiccant wheel surface.

The remaining portion of the air cooled by the DX coil is bypassed around the desiccant wheel through a bypass damper. Mixing of warm, very dry air after the wheel with the cool, moderately dry bypass air leaving the DX coil provides supply air at the temperature and humidity condition required by the space. The conditioned supply air is delivered to the laboratory building via duct perforated on one side (Figure 3).

Figure 1. General View of the IADR Unit at ORNL Site.

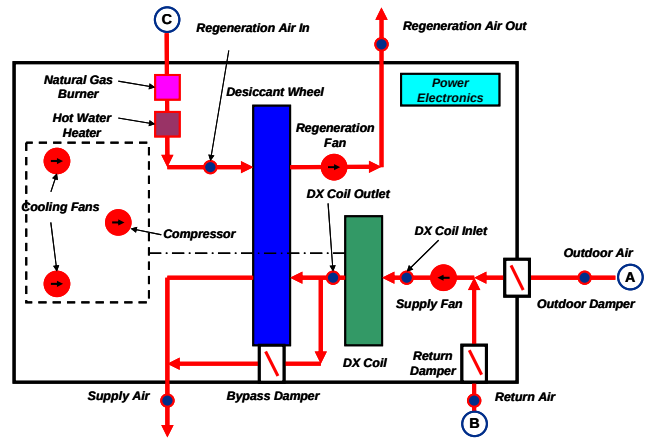

Figure 2. Schematic Diagram of the IADR Unit.

Figure 3. View of Air Supply Duct inside the Laboratory Building.

Air required for desiccant regeneration is heated either by a modulating natural gas-fired burner or by hot water coil (point "C", Figure 2). Upon regeneration of the active desiccant wheel, the regeneration air is exhausted from the unit.

The pattern of operation in the heating (heat pump) mode is similar to the cooling mode; the difference is in the desiccant stream function. The desiccant feature of the unit with its provisions for thermal regeneration provides a means to resolve some primary weakness associated with heat pump technology. During extremely cold weather or during defrost cycles many heat pumps do not have the capacity or lose control to effectively warm the conditioned space. The traditional way to overcome this capacity loss is the use of inefficient electric resistance heaters. At these times the IADR unit can energize the regeneration burner and increase the rotation speed of

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desiccant wheel, changing it from dehumidification device into a highly efficient heat exchanger. This provides additional effective heating of the supply air stream.

Instead of the use of the natural gas burner, the regeneration stream of this IADR unit can be heated by optional hot water heater, i.e. this unit can be part of combined cooling, heating and power (CHP) system, using waste heat for desiccant dehumidification purposes.

The unit at ORNL is fully instrumented in order to calculate various performance parameters. The data collection, analysis, and storage are performed with the Internet-based Web Control system [2]. The major test instrumentation and sensor measurement accuracies used for this analysis are given in Table 1.

Table 1. The Major Test Instrumentation and Measurement Accuracies during IADR Testing

Measurement	Sensor	Range	Accuracy
Temperature	Thermistor	-67 to 302 °F	± 0.4 °F (32 to 158 °F)
Relative Humidity	Humidity sensor	10 to 90 % RH at 14 to 140 °F	± 3% RH
Air flow	Piezo ring pressure transducer	0 to 3,600 cfm	± 5%
Natural gas flow	High-precision Coriolis sensor	0 to 5 lb/h	±2.0%
Electric power	Watt transducer	0 to 20 kW	±0.5% of full scale

DISCUSSION OF TEST DATA

The IADR at ORNL works to maintain a building set-point temperature condition of 72 ^{o}F over the wide range of outdoor temperatures (27-55 ^{o}F) in the heat pump mode. These test baseline conditions were performed at a supply air flowrate of 3,000 cfm and two outdoor air ratios: 50% and 0%. Also, some discussion of the IADR operation in a defrost mode as compared to normal operation at the same outdoor temperature is given for the 50% outdoor air/50% return air operation condition. All the data points are averaged values over a 2-3 hour test run with minimum change in outdoor temperature. A large sampling time interval is especially needed in an analysis of defrost operation or during relatively high outdoor temperatures, when "on-off" cycling operation of natural gas burner or compressor occurs.

IADR Performance at 50% Outdoor Air, 50% Return Air

Figure 4 shows dependence between outdoor temperature and the following temperatures: air entering the DX coil, supply air, and return air. The IADR unit was capable of maintaining the required laboratory space temperature of 72 ^{o}F very well over the outdoor temperatures ranging from 27 to 55 ^{o}F. All the temperature dependencies follow linear pattern, and the data scatter for both return and DX entering temperatures is much less then for supply temperature. The reason for this is that, apart from the outdoor temperature, the entering temperature is mainly influenced by return temperature and percentage of outdoor air: both values are almost constant during these tests. In addition to the above-mentioned parameters, supply air temperature is influenced by the heating load needed to maintain the required laboratory space set-point temperature. Since the laboratory building is not an environmental chamber, the initial temperature inside it at the same outdoor temperature would largely depend on diurnal daily conditions. For example, at the same outdoor temperature of 47 ^{o}F, the supply temperature during night conditions was almost 4 ^{o}F higher than during daytime conditions; coil entering temperature being almost the same in both cases. As a result, heating capacity produced by the heat pump in the first case is by almost 11 kBtu/h higher, than that in the second case. This is why a better way to do an analysis of the performance parameters is by a comparison to supply temperature, which is the method used in this paper.

The dependence of supply heating capacity generated by the IADR unit in heating mode versus supply temperature is shown in Figure 5. Supply heating capacity was calculated with the following equation:

$$Q_{supply} = \rho \cdot G \cdot \left(h_{supply} - h_{entry}\right) \quad (1)$$

where ρ is the density of air at standard condition, G is the volumetric flow rate of supply air, h_{supply} and h_{entry} are the supply air and DX entering air enthalpies, which are calculated from temperature and relative humidity data measured at corresponding air locations.

Supply capacity basically increases linearly with supply temperature. The maximum supply heating capacity at 50% outdoor air observed during these tests was around 100 kBtu/h. Figure 6 shows dependence of total electric power consumed by IADR unit versus supply temperature: at the maximum supply temperature observed during these tests and 50% outdoor air it reached 8 kW. Increases in compressor operating frequency with supply temperature are shown in Figure 7. The maximum frequency observed during the heating mode with defrost is slightly below 50 Hz.

One of the major performance parameters of the IADR unit is energy efficiency ratio (EER), which is the ratio of supply heating capacity to electrical and thermal power consumed by the unit. It is calculated with two options: using total power consumed by the IADR unit, and without the electric power consumption of the supply fan. The latter option, although does not give a complete picture of the relative use of power, can be useful while comparing the IADR performance

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with other conventional heat pump applications, where the supply fan power because of variability on the static supply pressure is usually taken out of consideration.

In both cases the variation of EER versus supply temperature is the same (Figure 8): it gradually increases and reaches maximum (14 and 17 Btu/W with and without account for the supply fan power) when supply temperature reaches around 82-83 °F. That corresponds to outdoor temperature of around 35-40 °F. With further increase in supply temperature EER values start to decrease. The difference between EER calculated with and without the supply fan power is the largest at lower supply temperatures (or higher outdoor temperature), especially when power consumed by supply fan (constant value) becomes larger than power needed for refrigeration process (compressor power and condenser fan power).

Comparisons between normal heating and heating during

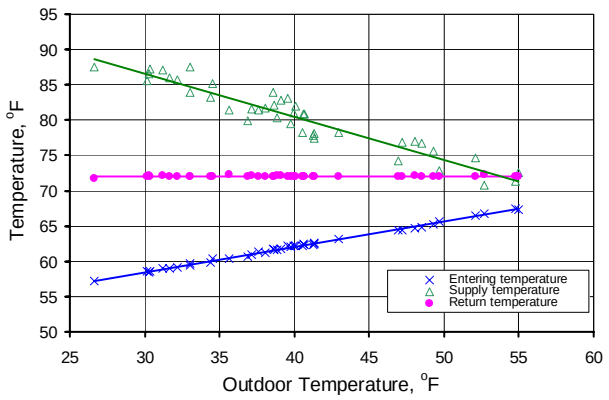

Figure 4. Effect of Outdoor Temperature on DX Coil Entering, Supply and Return Temperature at 50% Outdoor Air.

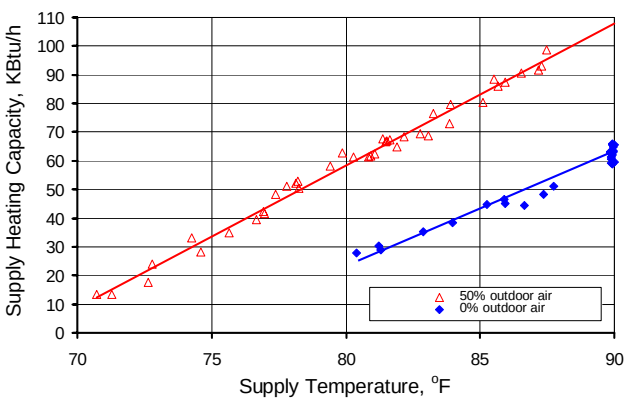

Figure 5. Effect of Supply Temperature on Heating Capacity at 50% and 0% of Outdoor Air

defrost modes of the IADR operation were performed at similar outdoor temperatures of 30.5 °F. A picture of the frosted condenser coil is shown in Figure 9, and some of the results of this analysis are presented in Figures 10 and 11. During a

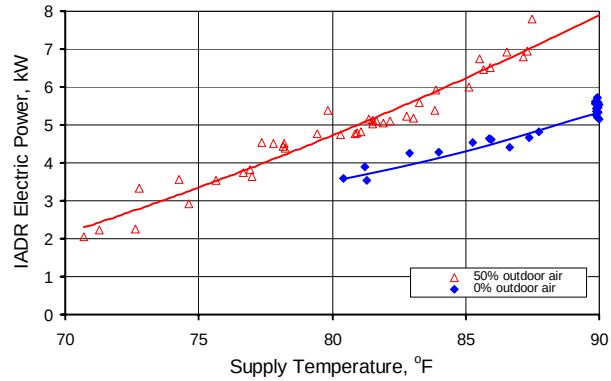

Figure 6. Effect of Supply Temperature on Electric Power Consumed by IADR at 50% and 0% Outdoor Air.

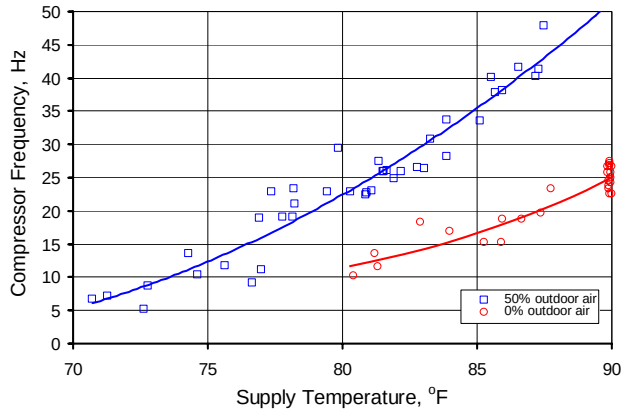

Figure 7. Effect of Supply Temperature on Compressor Speed at 50% and 0% Outdoor Air

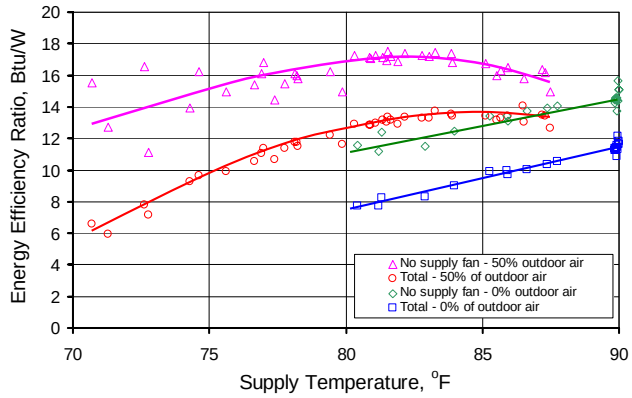

Figure 8. Effect of Supply Temperature on EER of IADR at 50% and 0% Outdoor Air.

monitored defrost over a three-hour period the unit went into a defrosting cycle for three minutes every 22-25 minutes. This operational mode called for activation of natural gas regeneration burner, regeneration fan, and desiccant wheel motor at full speed to prevent dumping of cold air into the space during defrost. This affected all performance parameters

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of the IADR unit as compared to the "non-defrost" mode of operation: supply heating capacity decreased from 90 to 70 kBtu/h, supply temperature decreased from 87 to 80 ^{o}F, EER fell from 13 to 8 Btu/W, total (electrical + thermal) power consumption increased from 7 to 9 kW. But the most important factor is that during defrost mode the unit was not able to hold

Figure 10. Effect of Defrost Mode on DX Coil Entering Temperature, Supply Temperature, Return Temperature, and Supply Heating Capacity at 50% Outdoor Air.

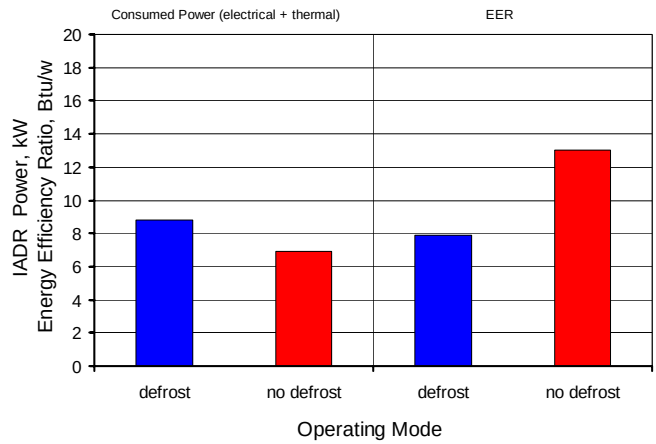

Figure 11. Effect of Defrost Mode on Power Consumption and EER of the IADR Unit at 50% Outdoor Air.

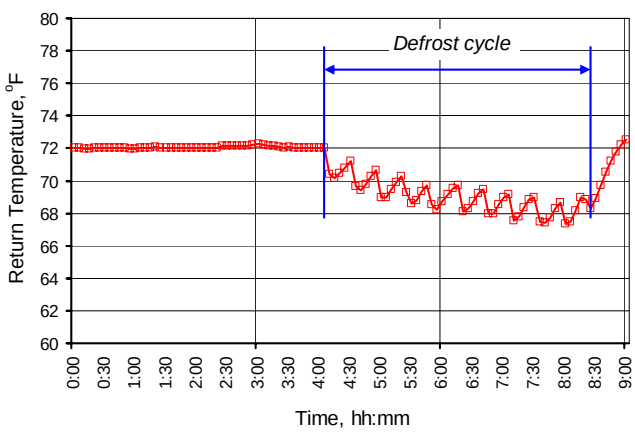

Figure 12. Real-Time Dependence of Return Temperature during Normal and Defrost Operation at 50% Outdoor Air.

the laboratory space set-point temperature of 72 ^{o}F, which dropped below 70 ^{o}F. Figure 12 shows real-time behavior of return air temperature during normal and defrost operation. It drops to as low as 68 ^{o}F. The major reason for this temperature decrease is that the natural gas burner at ORNL was sized primarily for desiccant wheel regeneration and the heating requirement needed to maintain the building set-point temperature was greater than the regeneration requirement. On the basis of this testing the capacity of the regeneration burner was increased so that the delivered supply air temperature during defrost can be the same as that provided by the heat pump.

Use of this desiccant thermal reactivation system in this manner, to prevent "cold-blow" during defrost, suggests that it could also be used as a means to augment or replace the vapor compression heat pumping function of the unit on very cold (< 0 ^{o}F) days when the efficiency of the reverse-Rankine cycle for heat pumping becomes inefficient and electric resistance heating is used to augment heat pump performance.

Figure 9. View of Defrosted Condenser Coil.

Comparisons between experimentally determined EER values for the IADR and for different types of conventional heat pump equipment are shown in Table 2. The conventional equipment data are derived from manufacturer's datasheets by extrapolation to match IADR test outdoor and DX coil entering temperatures. IADR EER data are given without including the supply fan power consumption, since the EER evaluation of the conventional equipment usually does not include supply fan power. The data in this table show that over the range of outdoor temperatures studied, EER of IADR is significantly higher than those for conventional heat pump equipment. The highest EER improvement is observed at lower outdoor temperatures (7.1-7.8 Btu/W), and the lowest – at higher outdoor temperatures (2.5-3.4 Btu/W).

Comparisons can also be made to the consensus agreement recently worked out between U.S. air conditioning manufacturers represented by the Air-Conditioning and

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Refrigeration Institute and energy efficiency advocates represented by the American Council for an Energy Efficient Economy that would raise the energy efficiency ratio (EER) of the most common air conditioning equipment from 8.9 to 11.2 Btu/W by 2010 [3].

IADR Performance at 0% Outdoor Air, 100% Return Air

This mode of IADR operation is similar to the way conventional residential heat pumps perform, when there is virtually no entry of outdoor air in the cycle. As a result, DX coil entering temperature is almost the same as return temperature (Figure 13). This coil entering temperature is much higher as compared to the 50% outdoor air case. This, in theory, should be a beneficial factor in heating the conditioned space, provided that the building is properly sealed and there is no uncontrolled ingress of outdoor air. But in this specific laboratory building approximately 1,000 cfm of air was being exhausted via a laboratory fume hood, creating a negative building pressure and allowing for the same amount of unconditioned cold outdoor air to enter the building. This is why higher supply air temperatures are needed in this case to maintain required space temperature. But the supply air temperature of the IADR unit was limited to 90 ^{o}F in order to provide safe operation of the compressor in accordance with recommendations of compressor manufacturers [4]. As it can be seen from Figure 14, the maximum condensing temperature of 95 ^{o}F (which corresponds to supply temperature of 90 ^{o}F) is based on the most extreme evaporating temperature of -10 ^{o}F, which is far less than was observed during these tests (20-30 ^{o}F). As a result, at outdoor temperatures below 45 ^{o}F the lab space temperature went down to 70 ^{o}F (Figure 15) as a consequence of the depressurization of the building (0% outdoor air) and a constraint on the maximum condensing temperature permitted by the IADR control scheme.

The IADR unit under these 0% outdoor air conditions was unable to successfully maintain the required laboratory space set-point temperature, as it was in 50% outdoor air case. The compressor speed at maximum available load at 0% outdoor air was only less than half of its rated frequency, i,e. 25-27 Hz

Table 2. EER Comparison: IADR vs Conventional Heat Pump Equipment at 50% Outdoor Air

Outdoor temp, ^{o}F	DX coil entering temp, ^{o}F	IADR EER, Btu/W	Conventional equipment EER, Btu/W				
			2,400 cfm 72 kBtu/h	2,400 cfm 72 kBtu/h	2,250 cfm 60 kBtu/h	5,000 cfm 150 kBtu/h	5,000 cfm 150 kBtu/h
34.4	59.8	17.5	10.4	10.3	9.8	9.9	9.7
36.9	60.7	15.9	10.6	10.5	9.9	10.1	9.8
40.0	62.2	16.8	10.8	10.7	10.1	10.3	10.0
43.0	63.2	15.8	11.2	11.1	10.4	10.6	10.4
47.2	64.4	16.1	12.1	12.0	11.5	11.1	11.4
49.3	65.3	14.9	12.2	12.1	11.7	11.2	11.5
52.1	66.4	16.3	12.4	12.3	11.8	11.4	11.6
55.0	67.7	16.5	12.6	12.5	12.0	11.5	11.8

Table 3. EER Comparison: IADR vs Conventional Heat Pump Equipment at 0% Outdoor Air

Outdoor temp, ^{o}F	DX coil entering temp, ^{o}F	IADR EER, Btu/W	Conventional equipment EER, Btu/W				
			2,400 cfm 72 kBtu/h	2,400 cfm 72 kBtu/h	2,250 cfm 60 kBtu/h	5,000 cfm 150 kBtu/h	5,000 cfm 150 kBtu/h
35.9	70.9	13.7	9.5	9.4	9.2	9.0	8.7
37.8	70.7	14.4	9.7	9.6	9.3	9.2	8.9
39.0	70.2	14.5	9.9	9.8	9.5	9.4	9.1
40.8	70.7	14.4	10.0	9.9	9.6	9.5	9.2
42.8	71.7	14.6	10.3	10.1	9.8	9.6	9.4
43.7	71.4	15.1	10.5	10.4	10.0	9.8	9.7
44.1	72.0	13.4	10.5	10.4	10.1	9.8	9.7
45.3	72.2	12.0	10.8	10.7	10.4	9.9	10.0
46.6	72.2	11.5	11.1	11.0	10.8	10.1	10.3
47.8	72.3	12.2	11.3	11.2	11.0	10.3	10.5
48.2	72.2	12.5	11.3	11.2	11.0	10.3	10.5
52.8	72.6	12.4	11.7	11.6	11.4	10.7	10.9

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(Figure 7). Therefore, the limiting condensing temperature of the IADR unit should be modified to reflect the actual evaporating temperature. This can be easily implemented with this unit because all of its operating and control algorithms are performed with a direct digital control programmable logic board, which can be reprogrammed to accept and employ improved operating and control logic suited to the specific building application.

The pattern of the remaining major performance parameters of the IADR at 0% outdoor air plotted against supply temperature (supply heating capacity, electric power consumption, EER) is basically the same as those for 50% outdoor air (Figures 5, 6, and 8). The only exception is IADR efficiency: over the range of supply air temperatures studied (80-90 ^{o}F), the IADR increased and reached his maximum EER values at a maximum supply temperature of 90 ^{o}F, There is no efficiency extremum, like in 50% outdoor air case.

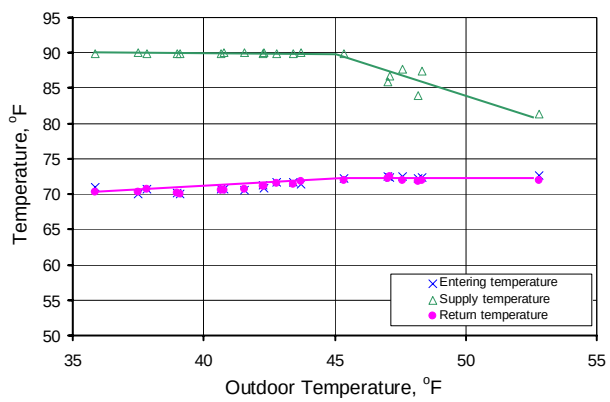

Figure 13. Effect of Outdoor Temperature on DX Coil Entering, Supply and Return Temperature at 0% Outdoor Air.

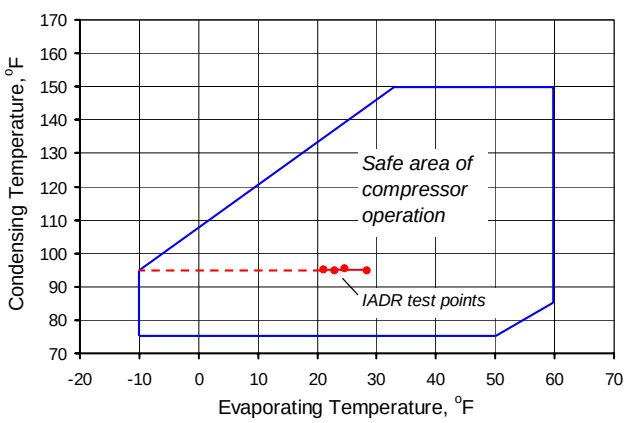

Figure 14. R-22 Operating Envelope [4].

The comparison of test results with 50% and 0% outdoor air shows the significance of building pressurization. If air conditioning system closed outdoor dampers result in negative building pressurization and the infiltration of unconditioned outdoor air, interior temperature and humidity set points are lost during heating and cooling seasons.

As in the previous case with 50% outdoor air, EER values for the IADR are higher than EER of conventional heat pump equipment (Table 3). Although at 0% outdoor air, this improvement is slightly less than that seen at 50% outdoor air (Table 2). But the 0% outdoor case is more typical for residential heat pump equipment. The EER improvement is also more noticeable at lower outdoor temperatures (4.7-5.5 Btu/W). At higher outdoor temperatures it decreases to 0.4-1.2 Btu/W.

CONCLUSIONS

A performance analysis of the novel Integrated Active

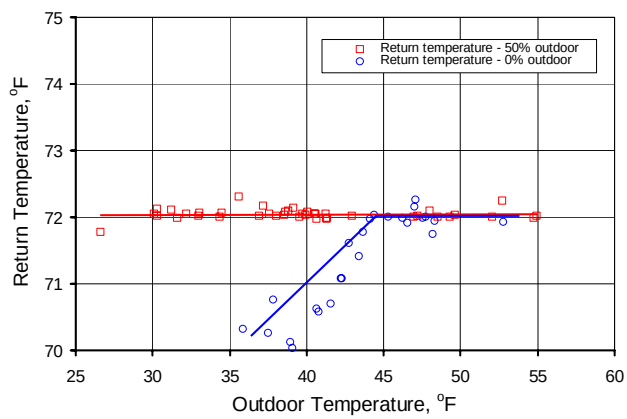

Figure 15. Effect of Outdoor Temperature on Return (Space) Temperature at 50 and 0% Outdoor Air.

Desiccant-Vapor Compression Hybrid Rooftop (IADR) at ORNL in a heat pump mode was performed at two outdoor/supply air ratios: 50 and 0%. The tests revealed clear advantage of variable speed compressors and blower motors over "on-off" cycling equipment. The EER of the IADR unit can be 7.1-7.8 Btu/W higher than that for conventional heat pump equipment. The defrost cycle decreases EER and to some extent the ability to hold a space temperature, but the use of desiccant regeneration system as a booster heater during defrost cycle prevents significant reduction of space temperature. On the basis of this testing the capacity of the gas burner used for desiccant regeneration was increased so that the supply air temperatures can be the same as that provided by the heat pump during normal operation. For this specific building application, slight building pressurization obtained by pulling in 50% outdoor air, resulted in better control of indoor set point at more extreme ambient conditions. When outdoor air is brought through the IADR unit, as opposed to cracks and leaks in the building, much more heating is provided, and better set-point control is obtained. Selection of the optimum heating mode

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should be based on both ambient and building conditions. If the building is well pressurized, there is no artificial extraction of air from the building, then the 0% outdoor air heating mode might be preferable.

The future study of the IADR unit will be focused on its performance in cooling and dehumidification modes, for both stand-alone and CHP-based operation.

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