Development of an Integrated Residential Humid Climate HVAC System

Marc Hoeschele and David Springer, Davis Energy Group

ABSTRACT

Standard residential vapor compression cooling systems provide a mix of sensible cooling (lowering indoor temperature) and latent cooling (removing moisture at the evaporator coil). These systems operate at a sensible heat ratio[1] (SHR) of about 0.75 to 0.85, depending upon indoor and to a lesser degree, outdoor conditions. In humid climates this often leads to indoor humidity exceeding typical comfort levels of 55-60%. This is especially true in energy efficient homes where sensible loads are reduced through the implementation of measures such as low solar heat gain coefficient glazing, improved building envelopes, and use of energy efficient appliances and lighting. Some advanced vapor compression cooling systems provide two-stage cooling or lower the airflow through the evaporator coil to increase dehumidification, but this control strategy may not improve comfort under low sensible load conditions, leading homeowners to further reduce their cooling setpoint or install a dehumidifier.

This project involved the design, construction, and testing of an advanced appliance that provides space conditioning, fresh air ventilation, and integrated dehumidification. The system was designed to respond to changing indoor temperature and humidity conditions by varying the supply air temperature, humidity, and airflow level. During conditions with elevated indoor relative humidity and no sensible cooling loads, supply airflow can be reduced to as low as 150 cfm/nominal ton, and a second refrigerant coil can be used to reheat supply air. Preliminary results from a Florida field test site are encouraging in terms of improved comfort and overall performance, although additional laboratory and field testing is needed to fully validate system performance.

Introduction

New homes being built in compliance with advanced energy efficiency programs such as Energy Star, LEED for Homes, and Building America, perform at levels significantly surpassing homes built just five years ago. Improved construction practices, diagnostic testing, and greater implementation of advanced cooling load reduction technologies contribute to significantly lower sensible loads (Christian, 2005). A lower sensible load reduces the ability of the vapor compression air conditioning system to remove the average 14 to 15 pounds per day (6.7 liters per day) of internally generated moisture (Christian, 1994) as well as any moisture addition from outdoor air. Prior field studies have evaluated the efficiency and effectiveness of various ventilation (Moyer et al, 2004) and dehumidification strategies (Rudd et al, 2005) in humid climates. Other studies have looked at how long it takes a vapor compression system to start condensing moisture from an initially dry evaporator coil[2] (Shirey and Henderson, 2004). Although wet coils at startup are typical during mid-day operation, dry coil startup may occur

---

[1] the sensible cooling capacity divided by the total (sensible + latent) cooling capacity

[2] A minimum of 11.5 minutes before condensed water vapor starts to fall from the evaporator coil, depending upon evaporator coil design, HVAC supply airflow, and indoor conditions

during low sensible load periods. Air conditioner oversizing (Proctor et al, 1995) further reduces latent cooling capacity by reducing the length of operating cycles.

Two common approaches to improving humidity control are lowering the thermostat setpoint and the use of free-standing, or in some cases integrated[3], dehumidifiers. The first approach may result in increased loads and energy use, clammy indoor conditions, and greater likelihood of mold forming on cool interior surfaces. Dehumidifiers add condenser heat to indoors, resulting in increased energy consumption.

A preferred strategy to maintaining indoor comfort is to optimally and efficiently dehumidify indoor air based on current indoor conditions and the comfort requirements of the homeowners. Additional desirable system attributes include fresh air ventilation to improve indoor air quality. For advanced cooling, dehumidification, and ventilation systems to become viable in the marketplace they must be easily installed, have user-friendly controls, and provide performance and/or cost advantages relative to conventional system alternatives. Although similar technical approaches have been used in commercial HVAC systems, and to a lesser extent in residential systems, the I-HVCD design approach incorporates a greater degree of control flexibility and system features, and also can integrate with two-stage condensing units from a variety of manufacturers.

Work presented in this paper was completed under DOE's Small Business Innovation Research program. The initial proof-of-concept development activities for the Integrated Heating, Ventilation, Cooling, and Dehumidification (I-HVCD) system were completed under a Phase I award, with project activities reported here completed under a subsequent Phase II award.

Project Objectives

The goal of this project was to design, develop, and demonstrate the performance of an integrated HVAC system that can optimally satisfy sensible and latent loads under varying conditions, as well as providing fresh air ventilation and nighttime ventilation cooling. The desired project outcome was to develop a product that is nearly ready for commercialization. The design goal was to develop an integrated appliance with the following attributes:

- Achieve low SHR's without rejecting unnecessary heat to indoors
- Precisely regulate both indoor temperature and relative humidity
- Provide fresh air ventilation and optionally night ventilation cooling
- Consume less energy than conventional two-stage air conditioner system with add-on dehumidifiers
- Package components so that they vary little from conventional components and are familiar to installers

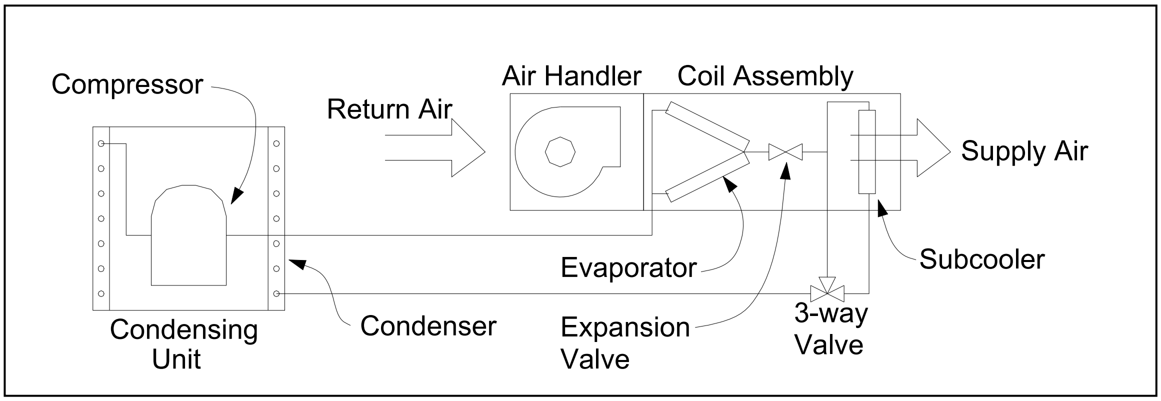
Project activities focused on refrigeration system design, controls development, and laboratory and field testing. A discussion of each of these areas follows.

[3] Free-standing dehumidifiers commonly available through retail stores are relatively inexpensive, but also inefficient. Integrated units are typically tied in to the duct system and deliver dehumidified air (and in some cases dehumidified outdoor air) to the supply ducts. These units are considerably more expensive, but also more efficient.

Refrigeration System Design

In evaluating equipment and design options for the I-HVCD system, several facts became clear. First, variable or two-stage compressor operation would be essential in providing the necessary capacity modulation to allow for longer operating cycles and therefore greater dehumidification potential. The decision was made to use two-speed condensing units based on the limited availability and high cost of variable speed condensers. Second, variable speed blower fan operation is critical for controlling evaporator coil temperature. Also, dehumidification under conditions with low sensible cooling loads would require the system to provide varying degrees of re-heat capability. Adding a sub-cooling coil downstream of the evaporator (as shown in Figure 1) provides this re-heat capability[4].

Figure 1: Basic I-HVCD System Configuration



The initial I-HVCD concept was to build on the control platform developed for a dry climate integrated night ventilation cooling system, by adding refrigeration modifications and sub-cooling coil re-heating to achieve lower SHRs. The lower potential for nighttime ventilation cooling in humid climates shifted the I-HVCD emphasis to providing fresh air ventilation as a first priority, and ventilation cooling as a secondary feature that could be used during less humid summer periods.

Preliminary laboratory testing of the Figure 1 design concluded that reduced coil airflow and supply air reheating at the sub-cooling coil still resulted in a higher than desired SHR under low sensible load conditions. This result led to a design modification that allows for partial condensing of refrigerant in the subcooling coil, allowing for a greater degree of supply air reheating. The subcooling coil recovers heat from liquid refrigerant (or a liquid and vapor mix), reheating air leaving the evaporator. Three primary factors were taken into consideration in designing the subcooling coil:

- *Re-heat capacity*: coil should deliver about 1 ton of heating at an airflow of 500 cfm (about 170 cfm/ton for the nominal three ton condensing unit)
- *Airflow Pressure drop:* the coil design should minimize the pressure drop when operating at full airflow of about 1,200 cfm (design target of 0.12” static pressure).

[4] Not all refrigerant valves and components are shown in Figure 1.

- *Refrigerant pressure drop*: coil circuiting should provide sufficient flow area to minimize refrigerant pressure drop. Refrigerant pressure drop degrades performance by increasing condensing pressure and temperature, thereby reducing operating efficiency.

Other key mechanical system design considerations included:

1. Development of a subcooling coil design that would achieve required re-heat performance, while minimizing both refrigerant and airflow pressure drops.
2. Refrigeration circuiting that would provide optimal performance flexibility (i.e. staged control of reheat), as well as reduced cost and increased reliability.
3. Packaging of components to minimize space requirements and facilitate installation and servicing of refrigeration hardware.

Controls Development

I-HVCD controls were built on the original ventilation cooling control platform that was first developed in the late 1990's (Springer et al, 2005). The hardware consists of a communicating wall display unit (or thermostat), a central control board, and remote sensors (outdoor air, leaving air, indoor temperature & RH, and evaporator coil surface temperature). The control board, connected to the wall display unit (WDU), receives input data from the WDU and other sensors. Controller outputs include conventional 24VAC heating/cooling outputs, a pulse width modulation signal to provide control of a variable speed indoor air handler fan, and a dry contact output to control the condensing unit fan.

Concurrent with hardware development, a WDU menu tree (describing how the wall display interacts with the user), flow charts, and state diagrams were developed to define the control logic and algorithms for programming into the controller. These algorithms were incorporated into a LabVIEW program that was used to drive mechanical relays operating refrigerant solenoid valves, fans, and the condensing unit. System operation was closely observed and modifications to the control logic were implemented and tested. Final algorithms and flow charts were then provided to the software developer for coding, testing, and implementation with the prototype I-HVCD control. The final step in the controls development process involved laboratory testing of the "field ready" controls and I-HVCD hardware prior to delivery to the Florida field test site.

Laboratory Testing

Prototype I-HVCD refrigeration systems were tested in our laboratory facility. Instrumentation included factory calibrated temperature and relative humidity sensors (supply and return air), true RMS power monitors, and a flow plate airflow measurement grid (located downstream of the subcooling coil and upstream of the supply air sensor)[5]. Power monitors were installed to record air handler and condensing unit power. The duct temperature/RH sensors and power monitors were continuously monitored by a programmable datalogger. The datalogger scanned all sensors on 10-second intervals and logged data at one-minute intervals. A sloped

---

[5] Accuracy of the temperature sensors were ±0.5°F, RH ±2.0% from 0-90% RH, ±3% > 90%, power monitors ±1.0%, airflow ±7% with a digital manometer.

condensate pan facilitated draining of condensate from the evaporator coil into a graduated cylinder.

Table 1 compiles data from eight tests, all of which represent exactly 20 minutes of steady-state operation at return conditions of 76°F and 50% RH[6]. The unit was operated in one of three modes: *Standard, subcooling, or partial condensing*. In "standard" operation, I-HVCD controls allow the condensing unit to operate in either first or second compressor stage. In subcooling and partial condensing modes, the I-HVCD operates only in first stage cooling to extend the cooling cycle duration for maximum dehumidification potential. The three modes are achieved by controlling the refrigerant solenoid valves, compressor staging, a condenser fan relay, and supply airflow levels[7].

Table 1 sensible and total cooling is reported for both "I-HVCD" and "Cooling", with the former entry including the reheat impact of the subcooling coil, while "Cooling" represents the temperature measurement immediately downstream of the evaporator coil. SHR is calculated two ways: first by dividing the sensible cooling by the total cooling (sensible plus latent) for the AC only ("SHRac"), and second by dividing the sensible cooling by the total cooling for the combined AC and dehumidifier ("SHRttl"). "Moisture Removal Efficiency" is calculated based on the total volume of condensate removed divided by the sum of the energy consumed by the dehumidifier and the energy consumed by the air conditioner in performing the latent cooling. Analogous to the SHR calculation, the term is derived in two ways: first based on both total combined AC and dehumidifier energy use, and second in terms of "latent kWh", defined as the sum of dehumidifier energy use and the fraction of total cooling energy use consumed for latent cooling[8].

The I-HVCD SHR ranges from close to 0.8 in Test 1 to 0.5 in subcooling tests 4 and 5, all the way down to 0.02 in partial condensing mode operation in Test 8. The last row in Table 1 reports condensate removal per kBtu of sensible cooling delivered. The last section of Table 1 summarizes moisture removal characteristics of the system under the varying operating modes and supply airflow levels. These results highlight the ability of the I-HVCD system to adjust performance based on indoor conditions and the occupant's desired temperature and RH setpoints. The difference between standard operation in Test 1 and partial condensing Test 8 clearly demonstrates the range of cooling and dehumidification performance achievable with the I-HVCD system.

---

6 Average chamber conditions for all eights were within 1°F and 1% RH of the target conditions, although short-term fluctuations during the test of up to 2°F and 3% RH were experienced.

7 "Std"= standard 1st or 2nd stage cooling operation, with airflow variation allowable in "low" stage. "Subcool"= subcooling operation with 1st stage compressor and supply airflow variation. "PartCond"= partial condensing = increased reheat at the subcooling coil with either partial or full condensing.

8 Calculated by multiplying the total AC cooling energy use by (1-SHRttl).

Table 1: Laboratory Test Results Summary

Test	1	2	3	4	5	6	7	8
Operating Mode	Std	Subcool	Std	Subcool	Subcool	PartCond	PartCond	PartCond
Compressor Stage	High	High	Low	Low	Low	Low	Low	Low
Evaporator Airflow (cfm)	1509	1509	546	546	429	373	546	348
Condensing Temperature (F)*	107	107	101	101	79	80	94	100
Qlatent (Btu/hour)	8556	12406	8485	11265	11836	10410	10267	8984
Qtotal (Btu/hour) - IHVCD		35907		22313	23968	15942	13478	9147
Qtotal (Btu/hour) - Cooling	38412	46264	24259	29342	29285	25748	26622	22710
Power (kW)	3.5	3.5	2.1	2.0	1.7	1.7	1.8	2.1
Cooling EER	11.0	13.4	11.8	14.4	16.8	15.2	14.5	10.9
I-HVCD EER		10.4		11.0	13.8	9.4	7.3	4.4
Cooling SHR (SHRac)	0.78	0.73	0.65	0.62	0.60	0.60	0.61	0.60
I-HVCD SHR (SHRttl)		**0.65**		**0.50**	**0.51**	**0.35**	**0.24**	**0.02**
Average Supply Air Temp (F)	58.4	61.9	50.2	57.8	49.9	62.3	71.2	76.6
Average Supply Air RH	87%	71%	91%	61%	61%	36%	40%	23%
Evaporator Temperature (F)*	46	46	42	38	30	30	40	31
Moisture Removal Summary								
Rate (liters/hr)	**3.6**	**5.2**	**3.6**	**4.7**	**5.0**	**4.4**	**4.3**	**3.8**
Efficiency (liters/total kWh)	**1.0**	**1.5**	**1.7**	**2.3**	**2.9**	**2.6**	**2.3**	**1.8**
Efficiency (liters/latent kWh)	**4.7**	**4.2**	**4.9**	**4.7**	**6.0**	**4.0**	**3.1**	**1.8**
Rate (liters/sensible kBtu)	**0.12**	**0.22**	**0.23**	**0.43**	**0.41**	**0.79**	**1.35**	**23.11**

** based on refrigerant pressure*

Field Testing

A key aspect of the project was to field test the prototype I-HVCD refrigeration hardware and control system in a humid climate. A 3,038 ft^2, two-story, energy efficient new home in Gainesville, Florida was secured for the 2006 and 2007 summer seasons. The original goal was to complete two years of field testing in an occupied house. Unfortunately complications with the homeowner (ultimately leading to foreclosure) resulted in the house being unoccupied during the full two-year period.

The field monitoring strategy was to monitor the base case system during 2006 and the I-HVCD prototype in 2007. The base case system was selected to represent "best available practice" HVAC design that would establish a performance level for comparison to I-HVCD. The installed base case HVAC system featured a three ton two-stage condensing unit (nominal 16 SEER, 12 EER), a variable speed hydronic fan coil unit with DX coil, and a whole house dehumidifier[9] (rated at 100 pints/day and 2.51 liters/kWh) that can also provide scheduled outdoor ventilation air (70 cfm).

The installed monitoring system allowed for collection of cooling energy delivered (latent, sensible, and total), cooling energy consumed (air conditioning and supplemental dehumidification), overall efficiency (effective EER[10]), and temperature and RH conditions (indoors, both 1st and 2nd floor, and outdoors). All sensors were scanned on a 15-second interval, and 10-minute interval data were stored in datalogger memory.

Summer 2006 base case monitoring extended from May 1st to October 29th. The summer was divided into three operational periods that featured fairly comparable indoor conditions and

[9] The 240 cfm unit pulls air either from the return plenum or a 10" outdoor air duct, dehumidifies and reheats the air, and exhausts the air to the supply plenum.
[10] Total cooling delivered divided by combined air conditioner and dehumidifier energy use.

system control strategies[11]. The first period (May 1st to July 30th) was characterized by conventional two-stage cooling operation, little outdoor air ventilation[12], and limited supplemental dehumidification provided by the integrated dehumidifier. In the second phase (July 31st to October 1st), the dehumidifier digital control was installed and 70 cfm of fresh air ventilation were provided for most of the period. The digital control was set to maintain indoor humidity at 50%. The third phase (October 2nd to the 29th) was implemented at the very end of the summer to assess how a lower (45% RH) setpoint would improve on the observed average indoor RH which exceeded the Phase 2 50% setting[13].

Table 2 summarizes performance data for each of the three phases. "AC Total cooling" is defined as the sum of the sensible and latent cooling by the air conditioner based on 1,080 Btu/lb of condensate collected during the period. "Overall EER" was determined based on the total AC cooling divided by the sum of the energy consumed (condensing unit, air handler, and dehumidifier) as shown below.

$$\text{Overall EER (Btu/W-hr)} = \frac{\text{Total AC cooling (kBtu)} + \text{Dehumidifier Latent cooling (kBtu)}}{(\text{(AC kWh} + \text{Air Handler kWh} + \text{Dehumidifier kWh)} \times 3.413)}$$

Table 2: 2006 Summer Data Summary

Parameter	Full Season	Phase 1	Phase 2*	Phase 3
Average Outdoor Temperature (°F)	79.3	80.2	80.9	72.7
Average Outdoor Dew Point (°F)	71.0	70.8	74.6	63.3
Average 1st Floor Temperature (°F)	75.2	73.7	77.2	75.4
Average 1st Floor RH (%)	49.7	48.9	54.5	41.4
AC Total Cooling (kBtu)	20,069	10,951	6,982	2,136
AC Latent Cooling (kBtu)	4,123	2,227	1,833	61
Condensing Unit kWh	1685	995	574	116
Air Handler kWh	403	190	111	102
Dehumidifier kWh	545	63	231	250
Overall EER (Btu/W-hr)	7.6	8.8	7.6	4.6
AC Condensate Volume (liters)	1748	944	777	26
Dehumidifier Condensate Volume (liters)	586	3	189	394
SHRac	0.79	0.80	0.74	0.97
SHRttl	0.73	0.80	0.67	0.53
Moisture Removal Efficiency (liters/latent kWh)	2.4	3.1	2.4	1.4

"*" includes 6.5 days during which thermostat was accidentally turned off

For the full six-month 2006 monitoring period, the base case system operated at an average EER of 7.6 and consumed a total of 2,633 kWh (21% dehumidifier). Indoor conditions in the unoccupied house averaged ~ 75°F and 50% relative humidity (the nominal setpoints maintained through both the 2006 and 2007 monitoring period). The average combined cooling

[11] Conditions were not perfectly controlled during each of the three phases as the HVAC contractor made modifications to the system through summer. The separation into three phases tries to capture similar operating characteristics.

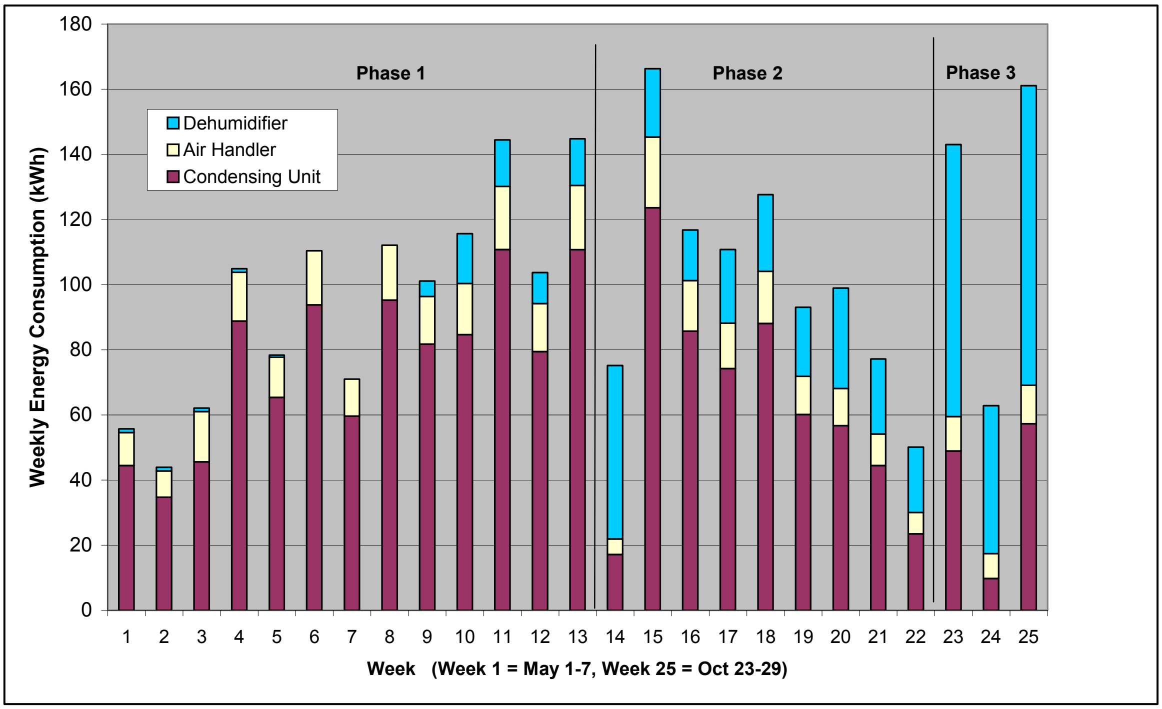
[12] The HVAC contractor was still completing the control installation.

[13] Typically the dehumidifier would cycle on at about 60% RH and turn off when indoor conditions reached about 48% RH.

system + dehumidifier sensible heat ratio was calculated to be 0.73. The SHRac was lowest during the more humid Phase 2 period. During Phase 3 when the dehumidifier was performing most of the condensate removal, SHRac rose to 0.97. The lower Phase 3 RH setpoint contributed to a significant increase in dehumidifier operation and a corresponding reduction in overall cooling efficiency to 4.6 EER. The Phase 3 SHR was 0.53. During Phase 3 operation, the dehumidifier had a moisture removal efficiency of 1.58 (394 liters divided by 250 kWh), relative to a 1.4 for the combined AC and dehumidifier. As noted, the dehumidifier has a rated efficiency of 2.51 liters/kWh.

Figure 2 plots weekly energy consumption for each of the energy consuming components[14]. Phase 1 data show a fairly standard breakdown between condensing unit and air handler energy use; Phase 2 demonstrates increased dehumidifier operation; and Phase 3 shows predominantly dehumidifier operation.

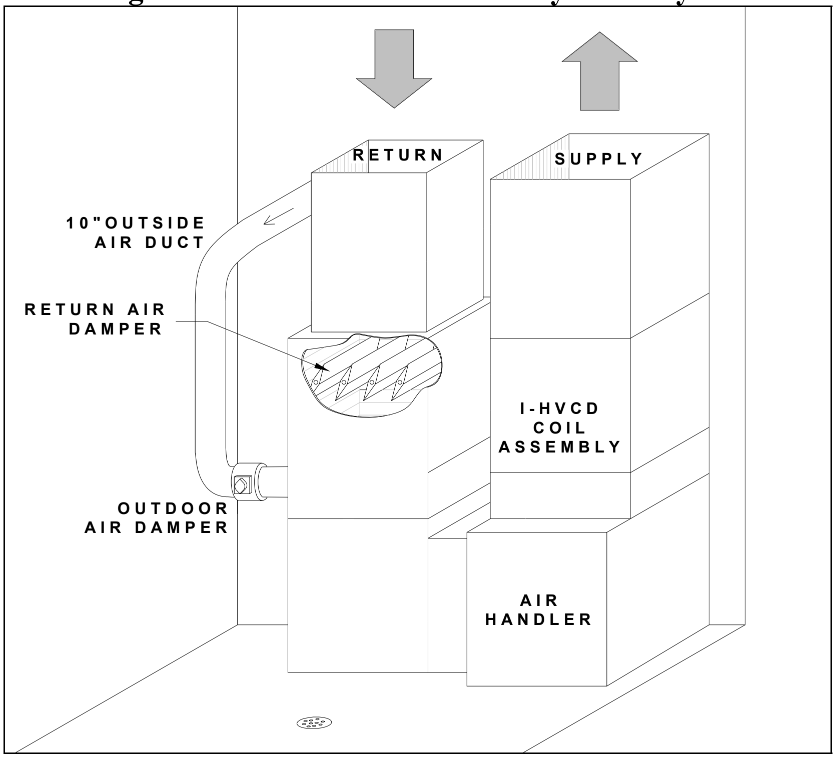
Figure 2: Weekly 2006 Base Case Energy Consumption



The plan for 2007 was to replace the existing evaporator coil with the prototype I-HVCD coil assembly, install the I-HVCD controls, disable the existing dehumidifier, and reroute the outdoor air duct as shown in Figure 3. The I-HVCD return air damper was designed to operate in tandem with the outdoor air damper: when the system enters outside air ventilation mode, the return air damper closes and the outdoor air damper opens, thereby delivering 100% outdoor air to the house. For fresh air ventilation the fan runs at minimum speed, supplying about 200 cfm for a fraction of each hour to meet the 70 cfm average outdoor air requirement.

[14] Note that low energy consumption during Week 14 was due to system being turned off at the thermostat.

Figure 3: I-HVCD Mechanical System Layout



Due to delays in finalizing and lab testing the I-HVCD control system, the Florida installation did not occur until the second week of September 2007. Difficulties with obtaining the correct return air damper prevented outdoor air system operation, making it difficult to compare results directly with the 2006 base case results with outdoor air ventilation. The 2007 I-HVCD data did however demonstrate proper control cycling of the damper relay.

Table 3 presents a sample of the full I-HVCD daily summary data collected from the September 13th system start-up through the October 17th monitoring end date. Over the entire fall 2007 I-HVCD monitoring period, the unit consumed a total of 433 kWh and removed a total of 200 liters of condensate while maintaining average first floor conditions of 75.1° and 44.6% RH. Applying the same methodology as for the base case system, a "Moisture Removal Efficiency/ latent kWh" of 2.6 liters/kWh is calculated. The most striking result in Table 3 is the typically small daily variation in indoor RH from average to maximum[15] (<2% typical) relative to the ± 5-7% daily variation observed during 2006 base case monitoring.

A qualitative effort was made to compare overall energy efficiency between base case and I-HVCD operation. Figure 4 plots daily total energy consumption (condensing unit, air handler, and dehumidification system) for the three base case operating phases[16] and the I-HVCD phase. Interestingly, Phases 1 and 2 indicate very similar total daily energy use trends as a function of average outdoor temperature. Phase 3 with a lower RH setpoint (7-13% lower average RH than Phases 1 and 2) resulted in energy consumption roughly double that of Phases 1 and 2. I-HVCD monitored energy consumption falls somewhere between Phase 3 and Phases 1

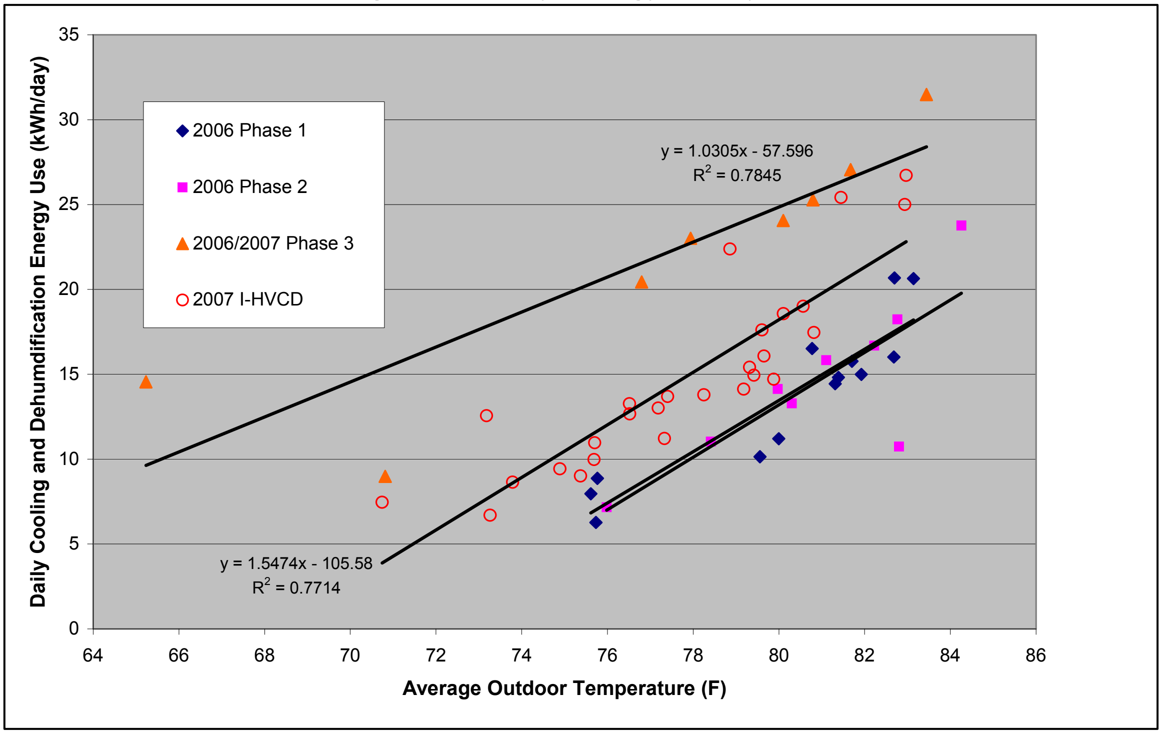
[15] September 13th shows a larger variation since this was the I-HVCD startup day and the system had been off.
[16] Base case data was logged and reported in weekly spreadsheet files. Each base case data point represents a full weeks worth of operation. Phase 3 data includes 2007 data using the Phase 3 thermostat settings.

& 2[17], suggesting that I-HVCD performance is clearly within the range of expected best practice system performance. Clearly more testing is needed to develop a better understanding of I-HVCD system performance.

Table 3: I-HVCD Sample Performance (Daily Averages & Totals)

Date	Outdoor Dry Bulb Temp (F)	Outdoor Dew Point Temp (F)	1st Floor Temp (F)	1st Floor RH (%)	1st Floor RH Max (%)	2nd Floor Temp (F)	2nd Floor RH (%)	2nd Floor Max RH (%)	Condensate Removal (liters)	Ttl Daily Energy (kWh)
13-Sep	85.3	77.0	77.7	42.2	48.4	78.2	44.3	48.3	16.3	33.2
14-Sep	81.4	76.9	76.2	41.5	42.2	76.2	44.5	46.2	11.3	25.4
15-Sep	83.0	78.0	75.8	41.9	42.9	76.2	45.6	48.2	10.8	26.7
16-Sep	82.9	77.3	75.7	42.5	43.2	76.2	46.2	47.6	10.4	25.0
17-Sep	77.3	75.3	76.0	43.1	45.6	76.2	46.1	47.9	6.9	11.2
11-Oct	76.5	68.5	73.9	46.6	48.3	73.5	51.0	52.7	7.1	13.3
12-Oct	70.7	59.5	73.6	46.9	48.8	72.8	51.0	52.7	0.6	7.5
13-Oct	73.8	67.3	73.7	48.3	49.6	73.1	52.3	54.0	2.6	8.6
14-Oct	73.2	67.8	73.6	47.4	50.6	72.9	51.3	53.6	4.7	12.6
15-Oct	75.4	69.2	73.9	48.4	50.3	73.5	52.2	54.2	4.0	9.0

Figure 4: Daily Energy Use by Mode



Economics

A key consideration for an advanced space conditioning system such as I-HVCD is cost: both first cost and operating cost. Although it is difficult to estimate the cost of the I-HVCD unit in production, the incremental cost should be significantly less than the $3,500 installed base case system cost for the dehumidifier/fresh air ventilation system, controls, and outdoor air

[17] The lack of a fresh air ventilation load during I-HVCD monitoring reduced the latent load on the system.

damper. The I-HVCD saves the cost of a refrigeration system, equipment cabinet, controls, some ducting, and installation costs. Although 2007 performance results are not conclusive, they suggest that the system should provide comparable operating costs relative to competing advanced system options.

Summary

Project results can be summarized as follows:

1. The system provides better indoor RH control than an advanced base case system alternative.
2. The I-HVCD has comparable moisture removal efficiency to the best dehumidifiers, while remaining thermally neutral to the space (no overcooling of indoor air or unnecessary heating during "latent only" operation).
3. The refrigeration hardware and controls are compatible with two-stage condensing units available from various HVAC manufacturers.

Next Steps

The development work completed in this project represents a significant step in moving the technology to the marketplace. The field results were encouraging, although the shortened monitoring period and lack of fresh air ventilation resulted in an incomplete demonstration of I-HVCD capabilities. To further move the technology forward, , the following steps need to be undertaken:

1. Test additional I-HVCD systems in the laboratory and in occupied houses in both hot-humid and mixed-humid climates.
2. Perform a value engineering review of the I-HVCD system to identify cost reduction opportunities.
3. Work with manufacturers to commercialize the I-HVCD technology. Two-zone control capability is needed to increase the marketability of the I-HVCD system.
4. Develop a testing standard that can be used to rate performance of both combined dehumidification/AC systems and integrated systems.

Acknowledgements

The U.S. Department of Energy's Small Business Innovation Research Program supported this work under Contract No DE-FG02-03ER83636. We would like to thank the support of our project subcontractors including Jim Phillips of The Energy Savers, Srikanth Puttagunta of Steven Winter Associates, Harlan Strickland, and Michael Kuhlmann of Residential Control Systems.

References

Christian, J.E. 1994. “Chapter 8: Moisture Sources:” Moisture Control in Buildings, ASTM Manual 18~~.~~ ed., H. Trechsel. West Conshocken, PA: ASTM 176-182.

Christian, J. 2005~~,~~. “Ultra-Low Energy Residences”, *ASHRAE Journal,* January 2005. Vol 47, #1, 20-26.

Moyer, N., D. Chasar, D. Hoak, and S. Chandra. 2004.~~,~~ “Assessing Six Residential Ventilation Techniques in Hot and Humid Climates,” ~~–~~*Proceedings of the 2004 ACEEE Summer Study on Energy Efficiency in Buildings.* ~~-~~ Washington, D.C.: American Council for an Energy-Efficient Economy.

Proctor, J., Z. Katsnelson, and B. Wilson. 1995.~~,~~ “Bigger is Not Better: Sizing Air Conditioners Properly”, Home Energy Magazine, May/June 1995.

Rudd, A.F., J.W. Lstiburek, P. Eng, and K. Ueno.~~,~~ 2005.~~,~~ “Residential Dehumidification Systems Research for Hot-Humid Climates”, NREL/SR-550-36643. Prepared for National Renewable Energy Laboratory under subcontract number ADC-1-34069-00.

Shirey, D.B. and H. Henderson. 2004.~~,~~ “Dehumidification at Part Load”, *ASHRAE Journal,* April 2004. Vol. 46, #4, 42-47.

Springer, D., L. Rainer, and W. Dakin. 2005. “Development and Testing of an Integrated Residential Night Ventilation Cooling System”, ASHRAE Transactions, vol. 111, pt.2: 501-510.