DAVIS ENERGY GROUP INCORPORATED

Development of an Integrated Residential Heating, Ventilation, Cooling, and Dehumidification System for Residences

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Presented to:	Terry Logee, DOE
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EXECUTIVE SUMMARY

The Need and the Opportunity

Codes such as ASHRAE 90.2 and the International Energy Conservation Code, and programs such as Energy Star and Builders Challenge, are causing new homes to be built to higher standards than ever before. As a result of tighter, better insulated building envelopes, sensible cooling loads in new homes are going down, but indoor air quality prerogatives are causing ventilation rates and moisture (latent cooling) loads to increase in humid climates. Conventional air conditioners are unable to provide the low (less than 50%) sensible heat ratios that are needed to efficiently cool and dehumidify homes. Since conventional thermostats respond only to indoor temperature, reduced sensible loads result in fewer cooling system operating hours, and therefore less dehumidification potential. Stand-alone dehumidifiers can be effective at moisture removal, but add heat to the indoor space that must be removed by air conditioners.

The project team saw an opportunity to develop a system that is at least as effective as a conventional air conditioner plus dehumidifier, removes moisture without increasing the sensible cooling load, reduces equipment cost by using the conventional air conditioner compressor instead of a second compressor, and simplifies installation. Such a system would seamlessly integrate heating, cooling, dehumidification, and fresh air ventilation functions under one control system and user interface.

Project Overview

Prime contractor Davis Energy Group led a team in developing and demonstrating an Integrated Heating, Ventilation, Cooling, and Dehumidification (I-HVCD) system under the DOE Small Business Innovation Research (SBIR) program. The original I-HVCD proof-of-concept work was completed in a Phase I project that ran from July 2003 to April 2004. Phase II project activities were completed from July 2004 through December 2007 and included:

- Mechanical Design and System Prototyping
- Controls Design and Testing
- Laboratory Testing
- Field Testing in a Gainesville, Florida house
- Commercialization Activities

Technology Description

Key components of the prototype I-HVCD system developed under this project include a an evaporator coil assembly (including reheat coil, refrigerant control valves, and sensors), return and outdoor air damper, and controls (including control board, communicating thermostat/humidistat, and outdoor air temperature sensor). These are used in conjunction with conventional components that include a variable speed air handler or furnace, and a two-stage condensing unit.

I-HVCD controls enable the system to operate in three distinct cooling modes to respond to different weather conditions and indoor temperature and relative humidity levels. When sensible cooling loads are relatively high, the system operates similar to a conventional system but has the capability to vary the air velocity over the evaporator to provide normal or enhanced dehumidification. In the second mode, the compressor is operated only at first stage, airflow is further reduced, and the reheat coil adds some heat to the supply air using refrigerant subcooling. In the third mode, which is activated by the humidity sensor, the reheat coil adds sufficient heat to maintain the supply air temperature close to the return air temperature (0% sensible cooling or 100% latent cooling). Additional features include the capability to provide ventilation cooling and fresh air ventilation using the outside air damper and variable speed fan.

Project Outcomes

The key objectives of the Phase II project were to develop a pre-production version of the system and to demonstrate its capability to efficiently maintain indoor temperature and relative humidity under all expected operating conditions in an actual house. The system was developed and successfully demonstrated in the laboratory and subsequently underwent field-testing at a newly constructed 3,080 ft^2 house in Gainesville, Florida. Field testing began in 2006 with monitoring of a "conventional best practices" base case system that included a two stage air conditioner and Energy Star dehumidifier. In September 2007, the cooling coil and controls were then exchanged with I-HVCD components to facilitate the comparison of test results.

Both systems maintained uniform indoor temperatures, but indoor humidity control was considerably better with the I-HVCD system. The daily variation from average indoor humidity conditions was less than 2% for the I-HVCD relative to the 5-7% for the base case system. Data showed that the energy use of the two systems was comparable, but additional system optimization using existing and future test results can be used to improve and fully establish potential I-HVCD energy performance.

Preliminary installed cost estimates suggest that production costs for the current I-HVCD integrated design would likely be lower than for competing systems that would include a high efficiency air conditioner, dehumidifier, and fresh air ventilation system. Value engineering and product refinements can be used to further reduce costs.

Project Benefits

This project verified that a system can be packaged for production scale market distribution that can be installed with air conditioning equipment from a variety of manufacturers, will be relatively familiar to HVAC technicians and retrofittable[1], and is cost competitive with other systems that provide similar control of indoor temperature and relative humidity. Project research also confirmed that the system can provide precise indoor temperature and humidity control under a variety of climate conditions, while delivering cooling at sensible heat ratios ranging from 0 to 90% and while using no more energy than the best available air conditioner and dehumidifier technologies. The integration of functions ensures thorough distribution of dehumidified air through

[1] Only two refrigerant lines are required between indoor and outdoor components.

common ductwork, simplifies the provision of fresh air ventilation, and reduces the amount of equipment that must be maintained, and ultimately replaced, by the homeowner.

Work completed under this project represents a significant step towards product commercialization. When commercialized, the I-HVCD will join the one or two systems that may have similar capabilities, and will provide a cost-competitive alternative for HVAC contractors who are married to particular equipment brands. Improved indoor humidity control and fresh air ventilation are system attributes that will become increasingly important in the years ahead as building envelopes improve and sensible cooling loads continue to fall. Technologies like I-HVCD will be instrumental in meeting goals set by Building America and other programs to reduce energy use while improving the indoor environment.

Next Steps

The following steps are needed to bring the product to commercialization:

1. Value engineering to reduce costs and improve performance.
2. Addition of zoning capability to controls to improve marketability.
3. Fabrication and testing of additional prototypes.
4. Identification of a manufacturer.

The I-HVCD has the distinct advantage of functioning with cooling equipment built by a variety of manufacturers. Initial efforts to interest large air conditioner manufacturers has shown interest, but no firm commitments, and the preferred path to market may be to employ a boutique manufacturer that markets to HVAC contractors. Distribution of the results of this work will improve opportunities for attracting manufacturers. However, additional sources of funding are needed to prepare the product for this final step.

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1. Background

Standard residential vapor compression cooling systems provide a mix of sensible and latent cooling. Sensible cooling is represented by a reduction in temperature of the return air while latent cooling reflects the energy removed due to condensation of water vapor at the cold evaporator coil. Most residential vapor compression equipment operates at a sensible heat ratio, or SHR[2], of about 0.75 to 0.85, depending upon both indoor and outdoor conditions. In typical humid climates this balance of sensible to latent cooling leads to indoor temperatures in the ideal range of 74-78°F, but indoor relative humidity often exceeds the 50-55% "comfort" when latent cooling loads are high. Advanced two-stage vapor compression systems typically operate longer run cycles (at low compressor capacity) allowing for more latent cooling, but still do not provide adequate dehumidification under very low sensible load conditions.

New homes being built in compliance with advanced energy efficiency initiatives such as Energy Star, LEED for Homes, and Building America, have considerably lower sensible cooling loads than homes built just five years ago. This is due to improved construction practices and diagnostic testing (tighter envelopes and duct systems), as well as greater implementation of advanced technologies that favorably impact building cooling loads (e.g. low solar heat gain coefficient glazing, energy efficient appliances and lighting, and attic radiant barriers). 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, either transported through the building envelope or delivered through mechanical outdoor air ventilation. Other studies have looked at how long it takes a vapor compression system to start condensing moisture from an initially dry evaporator coil[3] (Shirey and Henderson, 2004). Although wet coils at system startup are typical during mid-day operation, dry coil startup may occur during low sensible load periods when latent cooling demands are often higher. With air conditioners commonly oversized (Proctor et al, 1995), system short cycling further reduces latent cooling capacity.

The dehumidification performance of residential vapor compression cooling equipment is dependent upon many factors including climate, equipment sizing and selection (e.g. sizing of indoor and outdoor components, number of cooling stages), integrity of the vapor barrier, installation issues (duct location/insulation/leakage, HVAC supply airflow, refrigerant charge, etc.), occupant thermostat control behavior, as well as the type and magnitude of internal and outdoor sources of moisture. All these factors play a role in contributing to indoor comfort as shown in Figure 1.

Two fairly common approaches to improving humidity control are lowering the thermostat setpoint (increasing compressor run time and total airflow volume across the cold evaporator coil) and the addition of free-standing, or in some cases integrated[4], dehumidifiers. The first approach results in increased cooling energy use (and cooling load, as the indoor-to-outdoor temperature difference is increased), clammy indoor conditions, and the greater likelihood of condensation and mold forming on cool interior surfaces. Dehumidifiers are often used in humid climates to control indoor humidity, but since heat from their condenser is added to indoor air, they add to

[2] Sensible cooling capacity divided by total cooling capacity
[3] 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
[4] Free-standing dehumidifiers are relatively inexpensive, but also inefficient. Integrated units are integrated with the duct system and deliver dehumidified air (and in some cases outdoor ventilation air) to the supply ducts. These units are considerably more expensive, but also more efficient.

the sensible cooling load and therefore increase energy consumption. In addition, controls on these units are typically imprecise resulting in unnecessary energy use.

Figure 1: Interrelated Factors Affecting Indoor Comfort

A preferred approach to maintaining indoor comfort would be to optimally and efficiently dehumidify indoor air based on current indoor conditions and the comfort requirements of the homeowners. Additional desirable attributes of an advanced residential HVAC system would include fresh air ventilation and the ability to dehumidify when sensible cooling loads are small or nonexistent. Fresh air ventilation is receiving increasing interest from code officials and consumers as they become educated on issues related to indoor air quality and mold, and as code bodies adopt ASHRAE 62.2. For such advanced systems to be economically viable in the marketplace they must be easily installed, have user-friendly controls, and provide a performance and/or cost advantage relative to conventional system alternatives. This SBIR Phase II Project report presents development activities related to an Integrated Heating, Ventilation, Cooling, and Dehumidification (I-HVCD) system. The design intent of the I-HVCD system is to efficiently provide optimal temperature and humidity control under a wide range of operating conditions.

2. Characterizing the Problem

Concern about indoor air quality, indoor humidity, and mold has increased over the past ten to fifteen years. With a national trend towards tighter, more efficient houses, and an interest in mechanical ventilation for improved indoor air quality, humidity problems have increased. A number of recent technical papers have described the problem of humidity control in high efficiency, tightly constructed homes. The following excerpts of papers and reports present relevant findings from experts in this field:

> "However, it has been noted that some houses built under this program (*Building America*) in the hot and humid climate and equipped with a dedicated ventilation system were reported to have longer periods of elevated interior relative humidity (RH>60%) relative to conventional houses without dedicated ventilation systems (Rudd 2003)" (Moyer et al, 2004)
>
> "Because of the airtight envelope, well-shaded low solar heat gain windows and continuous mechanical ventilation the thermostat would not call for sensible cooling until after the RH rose above what would be considered acceptable in some situations." (Christian, 2005)
> *Note: Data collected at the Zero Energy test house in Tennessee indicated that for approximately 15% of the year, indoor RH exceeded 60%.*
>
> "While both houses were similar in size, total energy consumed for the Energy-efficient Reference house was less than half that of the Standard Reference house. However, because of the reduced sensible heat gain, and the resultant reduction in cooling system operation, humidity control performance in the energy-efficient house was inferior" (Rudd et al, 2005)
>
> "Incorrect use of vapor barriers is leading to an increase in moisture-related problems." (Lstiburek, 2004)
>
> "Otherwise, in more humid regions of the Midwest and South, set up systems to operate at high latent capacities using indoor airflows in the 300 to 350 cfm per ton range." (Kurtz, 2003)
>
> "During the cooling season, keep the indoor dew point below 55°F (*approximately 50% RH for cooling setpoint of 75°F*." (Lstiburek, 2002F)
>
> "Cycling results in a warmer coil temperature with less latent capacity than a colder coil. Hence, just when latent removal capability may be needed most, it is least available." (Hourahan, 2004)
>
> "Henderson's research suggests that until the runtime exceeds 40% of each hour, the moisture removal of a conventional cooling unit is so small as to be negligible." (Harriman, 2002)

As building scientists monitor HVAC systems in humid climates, the reality of these problems has become increasingly clear. To gain a better understanding of cooling loads and outdoor moisture sources in humid climates, a computer modeling study was completed for several climates. A 1,761 ft^2 prototype house[5] was modeled with the MICROPAS[6] simulation program using full-year hourly weather data from Houston, Tallahassee, Philadelphia, and Denver.

Figure 2 plots calculated outdoor air humidity ratios for the four locations based on the data contained in the hourly full-year weather files. The humidity ratio shown for each month was calculated based on monthly average outdoor dry and wet bulb temperatures. A target indoor comfort condition of 75°F and 50% RH is shown as a reference comfort level. The graph demonstrates how climate may impact indoor humidity levels. In Denver, outdoor conditions are, on average, drier than indoor conditions for all of the summer months, indicating that interior moisture will migrate outdoors and mechanical ventilation will generally result in drier indoor air. In contrast, Houston and Tallahassee represent very humid climates where moisture will generally migrate from outdoors to indoors and mechanical ventilation would contribute significantly to indoor relative humidity.

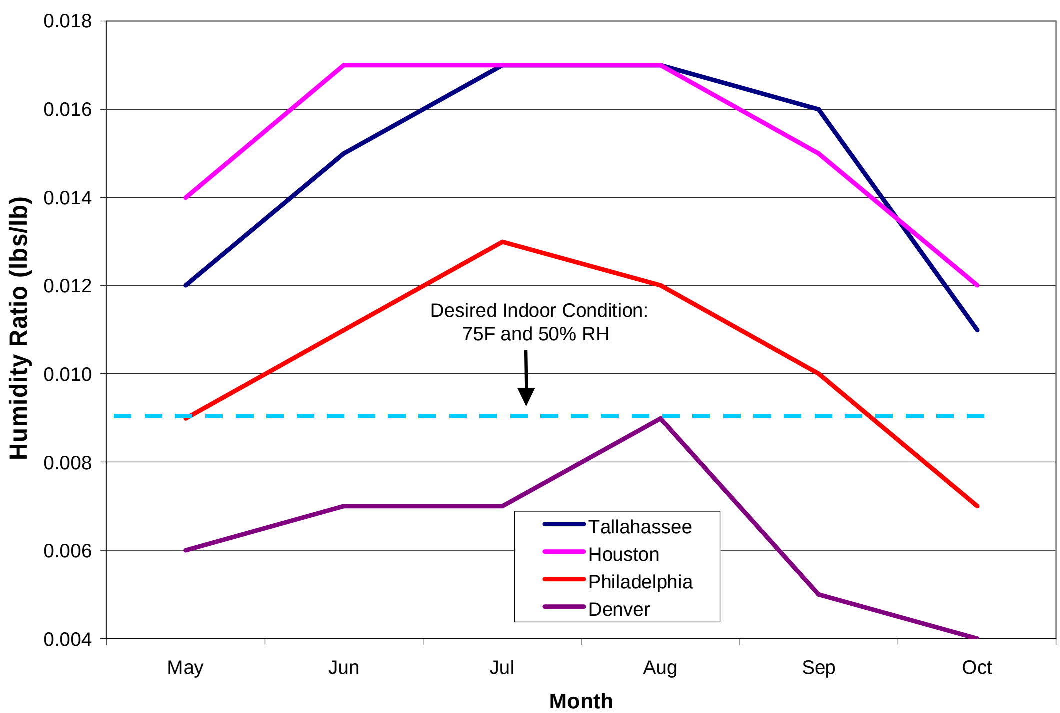

Figure 2: Average Monthly Outdoor Air Humidity Ratio

Figure 3 plots hourly Houston sensible cooling loads for the 1,761 ft^2 prototype as a function of outdoor dew point temperature. Peak sensible loads of slightly over 30,000 Btu/hour occur at dew points ranging from the upper 60's to 75°F. Under these design conditions an air conditioning system operating at a Sensible Heat Ratio (SHR) of 0.8 would provide about 7,000 Btu/hour of latent cooling, equal to about 2.9 liters of condensate removal <u>under steady-state operating conditions</u>. Indoor humidity control is most problematic in the region of the graph denoted by the red arrows. These conditions have a low sensible load, but a significant latent

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[5] This house features low Solar Heat Gain glazing and other efficiency measures that represent leading edge construction practice. Homes with these characteristics will demonstrate lower sensible cooling loads than typical homes and therefore represents cases where indoor humidity control is of greater concern.
[6] MICROPAS is the most widely recognized simulation model for Title 24 energy compliance in California.

load due to the high outdoor moisture levels. A low sensible load implies equipment cycling with little or no latent cooling potential. A conventional cooling system, even a two-stage unit, cannot adequately respond to these conditions and maintain indoor humidity levels.

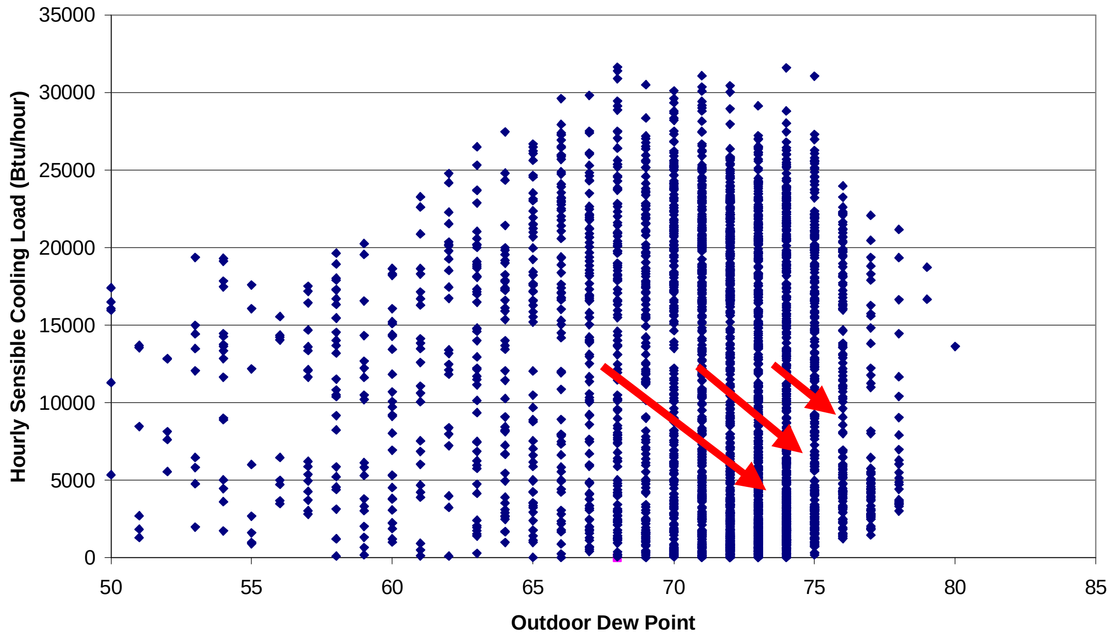

Figure 3: Hourly Sensible Load as a Function of Outdoor Dew Point (Houston)

Table 1 summarizes output from the MICROPAS hourly simulations in the Houston, Tallahassee, and Philadelphia climates. Data for each location is shown both for the full cooling season and for those hours with calculated sensible loads of less than 6,000 Btu/hour. In all three climates the average full-season hourly cooling load was slightly under one ton (11.2 and 11.4 kBtu/hour), although the number of cooling load hours varied from 1,180/year in Philadelphia to 3,300/year in Houston. Seven to nine percent of the total annual cooling load occurred in hours where the total sensible load was less than 6,000 Btu/hour. Interestingly in the humid Houston and Tallahassee climates, the average outdoor dew point temperature in the low sensible cooling load regime was more than 1°F higher than the seasonal average, highlighting the difficulty in providing sufficient latent cooling under very low sensible load conditions[7].

The MICROPAS analyses indicates that the most important design condition for a humid climate comfort system is not the standard ASHRAE 1% outdoordesign condition, but a condition that is likely to occur during summer night thunderstorms and other cool but humid weather events. A system meeting the following design conditions would need to operate at 0% SHR (100% latent cooling):

Outdoor: 70°F, 90% RH
Indoor: 76°F, 55% RH

[7] Not quantified in this analysis is the situation where indoor humidity is unacceptable but sensible loads are non-existent (humid nights with temperatures in the 70's).

Table 1: Projected Cooling Load Summary

Location	Number of hours	Sensible Load (kBtu)	Average Hourly Load (kBtu/hr)	Outdoor Dew Point Temperature (°F)*
Houston, TX				
Full season	3,230	36,800	11.4	70.0 °
Hours w/ Load< 0.5 tons	1,320	3,300 (9%)	2.5	71.3 °
Tallahassee, FL				
Full season	2,660	30,300	11.4	68.4 °
Hours w/ Load< 0.5 tons	875	2,200 (7%)	2.5	69.6 °
Philadelphia, PA				
Full season	1,180	13,270	11.2	64.1 °
Hours w/ Load< 0.5 tons	375	981 (7%)	2.6	63.6 °

* Load weighted average

3. Project Objectives and Overview

The first phase of this DOE-sponsored SBIR project proved the concept that an air conditioner could operate through a wide range of conditions and deliver supply air at a temperature and relative humidity appropriate for the SHR. The goal of the second phase of the project was to design, develop, and demonstrate the performance of a fully integrated HVAC system that can provide optimal sensible and latent cooling performance, ventilation cooling, fresh air ventilation, and heating. The intended Phase 2 outcome was to develop a product that is nearly ready for commercialization, and that could be handed off to a major HVAC equipment manufacturer.

The development effort focused on producing an efficient, low-cost design that is controlled such that it responds to changing indoor conditions. The premise of this project is that an integrated appliance can provide sensible heat ratios as low as zero without rejecting unnecessary heat to indoors, can precisely regulate both indoor temperature and relative humidity, and can consume less energy than conventional two-stage air conditioners with add-on dehumidifiers. The product should also be packaged such that it would not deviate significantly from conventional systems in terms of components and installation requirements.

Phase II project activities were completed during the time period extending from July 2004 through December 2007. (A brief description of the Phase II project team is included in Appendix A.) The original Phase II completion target of July 2006 was extended twice: first, to provide for two summer seasons of field monitoring, and second, to accommodate delays in completing development of the I-HVCD controller. The original Phase I proof-of-concept work was completed in the period from July 2003 to April 2004.

Specific project objectives by topical area include:

I-HVCD System Mechanical Design and Development

1. Evaluate need for two-stage condensing unit and applicability of advanced variable speed compressors
2. Develop refrigeration system configuration with component cost and performance flexibility being key criteria
3. Determine subcooling coil design and circuiting for optimal performance
4. Integrate heating, ventilation cooling, and fresh air ventilation operation

Controls Design and Testing

5. Develop prototype control system design that utilizes both indoor temperature and RH as control inputs capable of triggering cooling/ dehumidification operation.
6. Provide for dehumidification operation under situations where no sensible cooling is required.
7. Test and verify control logic operation in both lab and field environments.

System Laboratory Testing

8. Verify that I-HVCD prototype performance meets design expectations and that the system can reliably operate in all modes.
9. Document the impact of refrigerant control and supply airflow variations on sensible and latent cooling capacity under varying return air conditions common to humid climates.

System Field Testing

10. Monitor best current practice HVAC system performance in a new home located in a humid climate for a full cooling season
11. Monitor I-HVCD performance in the same home the following summer and contrast performance.

Commercialization Activities

12. Evaluate preliminary economics relative to base case best practice
13. Based on acceptable field performance, pursue commercialization activities with a major HVAC equipment manufacturer

4. Design and Test Methodology

Phase II of this SBIR project comprised mechanical and refrigeration system design, control development and testing, laboratory and field testing of the refrigeration components and controls, and commercialization activities. The following report sections describe project efforts in each of these areas.

4.1. Determining I-HVCD Design Conditions and Capabilities

U.S. climate data were reviewed to identify system performance requirements. Performance parameters were defined for the range of climates where there is a market need for the I-HVCD product and used to establish sensible heat ratio and latent cooling capacity ranges and targets. It was concluded that the product should have the capability to adapt to the most humid conditions, exemplified by the Houston climate, as well as to hot-dry conditions that can occasionally occur in humid climates. As previously pointed out, it was concluded that the product should be able to satisfy SHR's ranging from 0 to the upper range of conventional air conditioners.

The I-HVCD product is a derivative of a ventilation cooling system that has proven to be effective at reducing cooling energy use in dry climates. It was of interest to determine the extent to which ventilation cooling should be retained in the I-HVCD design. Ventilation cooling is intuitively less beneficial in humid climates than in dry climates since the diurnal temperature variation is smaller and higher outdoor humidity corresponds to higher enthalpy of outside air. Prior dry climate studies (Springer 2004) showed that the optimal ventilation rate for ventilation cooling is about 0.6 cfm per square foot of conditioned floor area, which is about the same air volume typically used to distribute air conditioned air in residential systems. The large dampers required to process this volume of outside air adds significantly to the system cost. Given the limited value of ventilation cooling in humid climates it was determined that the system should deliver ample air for fresh air ventilation with some ventilation cooling contribution when conditions are appropriate. This change would reduce the required duct/damper size from about 20" (for a 2800 ft^2 house) to about 10".

To evaluate the potential of ventilation cooling in various U.S. climates, hourly computer simulations were completed on an 1860 ft^2 prototype house using the DOE2 building simulation model. Simulations were completed with enthalpy ventilation control to insure that the outdoor air enthalpy was lower than indoor air during ventilation operation. Results of this analysis are provided in Section 5.1.

The Phase I concept was to build on the prior development of a dry climate integrated night ventilation cooling system (NightBreeze™) using the system's variable speed fan and programmable control technology to develop a humid climate system that would regulate sensible heat ratio by varying the airflow rate through the evaporator coil[8]. The lower potential for ventilation cooling in humid climates shifted the emphasis to providing fresh air ventilation as a first priority, and ventilation cooling as an additional feature that could be used during dry periods.

4.2. Developing the Mechanical Design

4.2.1. Design Strategy

Preliminary evaluations of the Phase I design concluded that reduced coil airflow and refrigerant subcooling alone would still provide a higher SHR than desired under low sensible load and high relative humidity conditions. This led to a design that allows partial condensing of refrigerant to occur in the subcooling coil.

Other key mechanical system design considerations included:

1. Development of a subcooling coil design that would achieve required reheat performance, while minimizing both refrigerant and airflow pressure drops.
2. Designing 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 facilitate installation and servicing of refrigeration hardware
4. Compatibility with most two-stage condensing units and variable speed furnaces

Other facts became clear in the course of the Phase I evaluations. First, variable or two-stage compressor operation would be essential to provide the capacity modulation necessary to allow for longer operating cycles and therefore greater dehumidification potential. Based on the limited availability and high cost of variable speed condensers, the decision was made to design the system for use with two-speed condensers. Second, variable speed blower fan operation would be critical for controlling evaporator coil temperature. Also, dehumidification under conditions with little or no sensible cooling load would require that the system provide varying degrees of reheat capacity.

4.2.2. Optimizing the Subcooling Coil Design

The subcooling coil (as shown in Figure 4) provides reheating of supply air during I-HVCD operating modes where a lower SHR is desired. The subcooling coil recovers heat from the liquid or liquid/vapor refrigerant mix as it is supplied from the condenser to the expansion valve

[8] Development of the NightBreeze ventilation cooling system is described in Springer, 2004.

and evaporator coil, and reheats air from the evaporator. Three primary factors were taken into consideration in sizing and configuring the subcooling coil:

Reheat capacity: the coil should deliver about 1 ton of heating at an airflow of 500 cfm (~170 cfm/ton) when the condensing unit is in a 70°F outdoor environment.

Airflow pressure drop: the coil should be designed to minimize the pressure drop when the system is operating at full airflow (1200 cfm). A design target of 0.12" static pressure was specified. Fin spacing has the largest impact on pressure drop.

Refrigerant pressure drop: the coil should be circuited to provide sufficient flow area to minimize the refrigerant pressure drop through the reheat coil. Refrigerant pressure drop degrades performance by increasing the condensing temperature, thereby reducing operating efficiency.

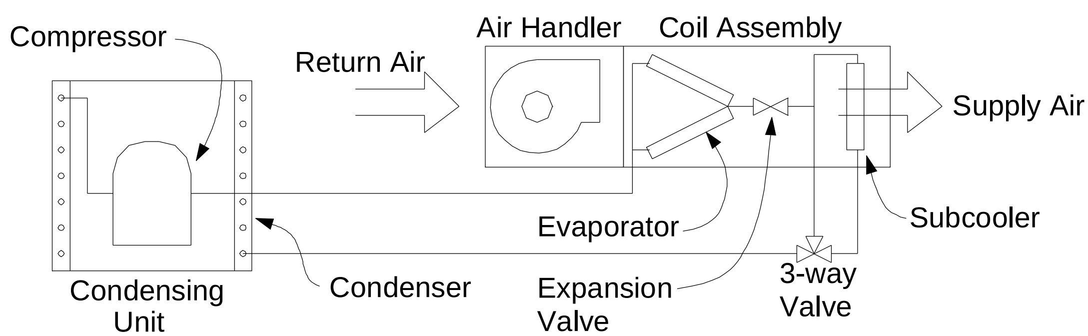

Figure 4: Basic I-HVCD System Schematic Developed in Phase I

The Heatcraft coil design software was used to iterate on various coil design parameters to arrive at the preferred coil design. The coil program calculates sensible and latent cooling capacities and supply air temperatures given the specifications of the coil and airflow rate. Coil characteristics that were varied include number of rows, fin spacing, fin type and thickness, tube size, number of circuits, evaporator temperature, and airflow across the coil. The subcooling coil was sized to accommodate about 30% of the condensing capacity for a 3 ton condensing unit. After selecting the optimal design we completed additional analysis of the preferred coil types using an online evaluation model available from ORNL [9]. This model was used to identify performance impacts of the preferred coil design[10].

An additional concern in the subcooling coil design was ensuring uniform leaving air temperatures from the coil. Since partial condensing operation sends a hot liquid/vapor refrigerant mixture to the coil, it is important that the coil circuiting delivers heat uniformly across the face of the coil to maximize heat transfer efficiency and provide a uniform supply air temperature profile leaving the coil. If the temperature profile is uneven, some supply ducts may get cooler air and others may get warmer air.

4.2.3. Refrigerant Distribution, Control, and Packaging of Components

The outcome of the first phase of the SBIR project was a prototype system that met the hoped-for functional performance, but that was not in a form that would serve as a production prototype. In

[9] ORNL (see http://www.ornl.gov/~wlj/hpdm/MarkVI.html).
[10] Unfortunately this model does not have the capability to simulate a system with a subcooling coil.

the second project phase substantial attention was given to the packaging of components into a coil case that is not significantly larger than a standard cased evaporator coil, and that includes all necessary refrigerant piping and valving, as well as the evaporator and reheat coil. This work (by Jim Phillips) included designing the refrigerant distribution piping and connections between the various components, including the evaporator, subcooling coil, solenoid valves, check valves, thermostatic expansion valve, and refrigerant receiver. Because of the pressure limitations of available receivers, the higher operating pressures for R-410a, and space constraints, it was necessary to custom fabricate an accumulator that is part of the coil assembly. A total of four solenoid valves were included, requiring special provisions to enable charging the multiple, isolated refrigerant passages.

4.3. Controls Design and Development

4.3.1. Preliminary Development

I-HVCD controls built on the original NightBreeze control system that was developed in conjunction with RCS/ZTECH beginning in the late 1990's. The first step in development of the control was to prepare a functional specification that defined I/O and other requirements, and that described control functions and capabilities. Hardware was then eveloped to meet these requirements.

4.3.2. Hardware Description

The controller receives temperature and relative humidity data, setpoints, and other information from the wall display unit (WDU) and other sensors, and contains all of the logic and outputs. Controller outputs include conventional 24VAC 'W'and 'Y' heating and cooling outputs, as well as a pulse width modulation (PWM) signal to control the indoor fan and a dry contact output to control the condensing unit fan. The WDU and controller were developed specifically for this project and are pictured in Figures 5 and 6. Embedded C language was used to program both the WDU and controller. A four-wire communications bus links the WDU and CCU. A block diagram of the control components is provided in Figure 7.

Figure 5: Prototype Wall Display Unit

Figure 6: Prototype Central Control Unit

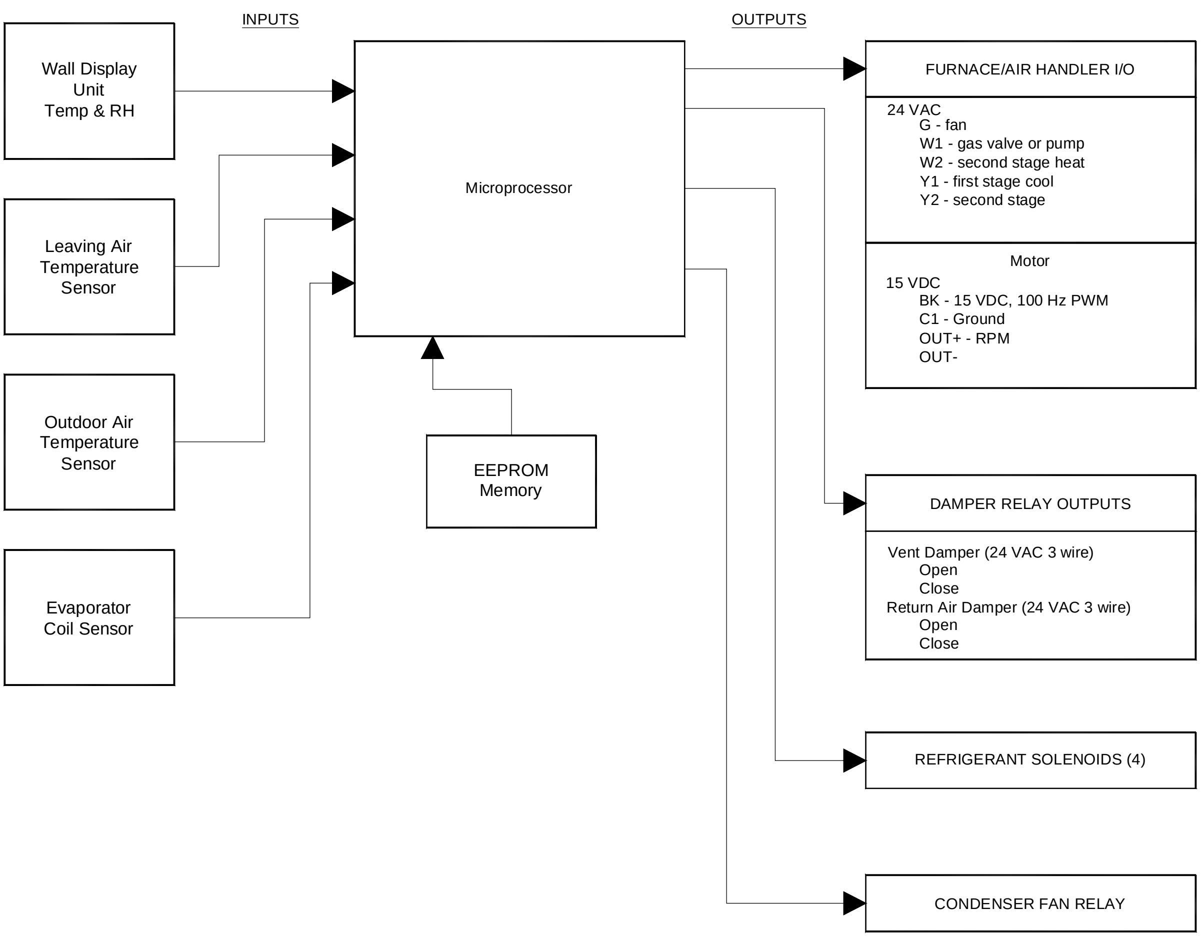

Figure 7: Block Diagram of Control System

RCS/ZTECH produced the custom central control unit (CCU) and prototype wall display unit (WDU or "thermostat"). The CCU includes the following inputs:

- Indoor temperature (via a temperature/RH sensor located in the WDU)[11]
- Indoor RH (via the WDU temperature/RH sensor)
- Outdoor temperature thermistor
- Leaving air temperature (thermistor for sensing air temperature entering supply plenum)
- Evaporator coil temperature (clamp-on thermistor for sensing evaporator coil temperature)

CCU outputs include:

24 VAC Heating/Cooling System Control

- W_{low} and W heating signals for first and second stage furnace heating (air handlers use only the W_{low} output to operate the hydronic pump and vary fan speed with heating demand)
- Y1/Y2 cooling signals for first and second stage condensing unit operation
- G fan signal. Used to control furnace fans for air conditioning
- Outside air/return air damper (3-wire)

Fan Motor

ECM motor outputs include:

- G "run" signal, 15 VDC
- BK variable speed signal, 15 VDC, 100 Hz, pulse width modulated[12]
- C2 15 VDC common

Refrigerant Controls

- Four refrigerant solenoid valves for controlling refrigerant flow path
- Condenser fan relay (for cycling condenser fan during maximum dehumidification operation

Display

An LCD display screen is used to display temperatures, show operating status, and other information. "Soft" buttons are labeled appropriate to their function under the different viewing screens. Four LED lamps on the left side of the wall display unit are used to indicate system status:

LED1 (green) – Fan on, no heating or cooling call.
LED2 (green) – Call for cooling and/or dehumidification
LED3 (green) –Call for heating
LED4 (red) – Service needed
All green LED's off – no operation

4.3.3. Firmware Development

A series of design review meetings were conducted to evaluate a wide range of alternative control strategies, and to define specific control routines. A control approach was selected that provides different control modes for varying indoor temperature and humidity conditions, and that also provides a smooth transition between each mode. Upon finalizing this preliminary design

[11] The temperature/RH sensor is a Sensiron SHT11 with 2% accuracy
[12] Furnace motors are switched to "vspd" mode by applying a 100 Hz PWM signal and to "tstat" mode by applying a 60 Hz sign wave signal.

concept, the I/O requirements were specified, enabling RCS/ZTECH to proceed with hardware development.

Concurrent with hardware development, DEG developed a menu tree for the WDU, and, in collaboration with refrigeration and controls programmer, flow charts and state diagrams to define the control logic and algorithms for programming into the controller. Once finalized, control algorithms were incorporated into a LabVIEW program for initial laboratory testing with the prototype coil assembly. The LabVIEW program drove mechanical relays that operated the solenoid driven refrigerant valves, fans, and condensing unit. System performance was monitored and modifications to the control logic were implemented and tested. Finalized algorithms and flow charts were completed and provided to Harlan Strickland for coding, testing, and implementation with the prototype I-HVCD control. The final step in the controls development process involved laboratory testing of the complete "field ready" controls with the refrigeration hardware prior to delivery to the Florida field test site. Nearly two years were required to complete design, development, and testing of the controls.

4.4. I-HVCD Mechanical System Laboratory Testing Methodology

4.4.1. Laboratory Test Setup and Equipment

The test chamber at Davis Energy Group's facility (shown in Figure 8) was configured to provide controlled environmental conditions for testing the prototype I-HVCD system. The test chamber was heated by a heat pump air handler ("HP" in Figure 8) and hydronic coil connected to a 180,000 Btu/hour instantaneous gas water heater. An evaporative cooler ("EC") located inside the test chamber provided both a source of moisture and cooling, in response to the datalogger controlling environment conditions. A datalogger was used to control both the hydronic circulating pump and the evaporative cooler pump. Both the heat pump fan and the evaporative cooler fan operated continuously during the tests to thoroughly mix test chamber air. If chamber humidity levels were lower than the target level, the datalogger would activate the evaporative cooler pump, providing moisture and cooling to the chamber. If chamber temperatures started to fall, the hydronic pump would be energized until chamber conditions stabilized. This setup was capable of maintaining chamber conditions within 1.5° and 3% of the target conditions for all tests completed.

The I-HVCD condensing unit was installed inside a louvered box. Changing the position of the louvers regulated condensing unit operating pressure. Fully opening the louvers would introduce cool air from the open warehouse, reducing the air temperature. Since most of the testing was conducted during mild outdoor conditions, this approach was adequate in maintaining the desired (and reasonably constant) condensing unit environment. Refrigerant lines were equipped with pressure gauges and surface-mount temperature sensors to aid in identifying the operating conditions at various key points in the system.

Table 2 specifies the test equipment used in monitoring the prototype I-HVCD system in the laboratory. Two factory calibrated Vaisala temperature and relative humidity sensors were installed in the supply and return airflow paths. A TrueFlow airflow measurement grid was installed downstream of the evaporator and subcooling coils to measure system supply airflow. 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 Data Electronics DT-50 datalogger. The DT-50 was programmed to scan all sensors channels on 10-second intervals and log data at one-minute intervals. A sloped condensate pan, positioned under the evaporator coil, facilitated complete draining of condensate. A graduated cylinder was used to record condensate collected during each twenty minute timed test.

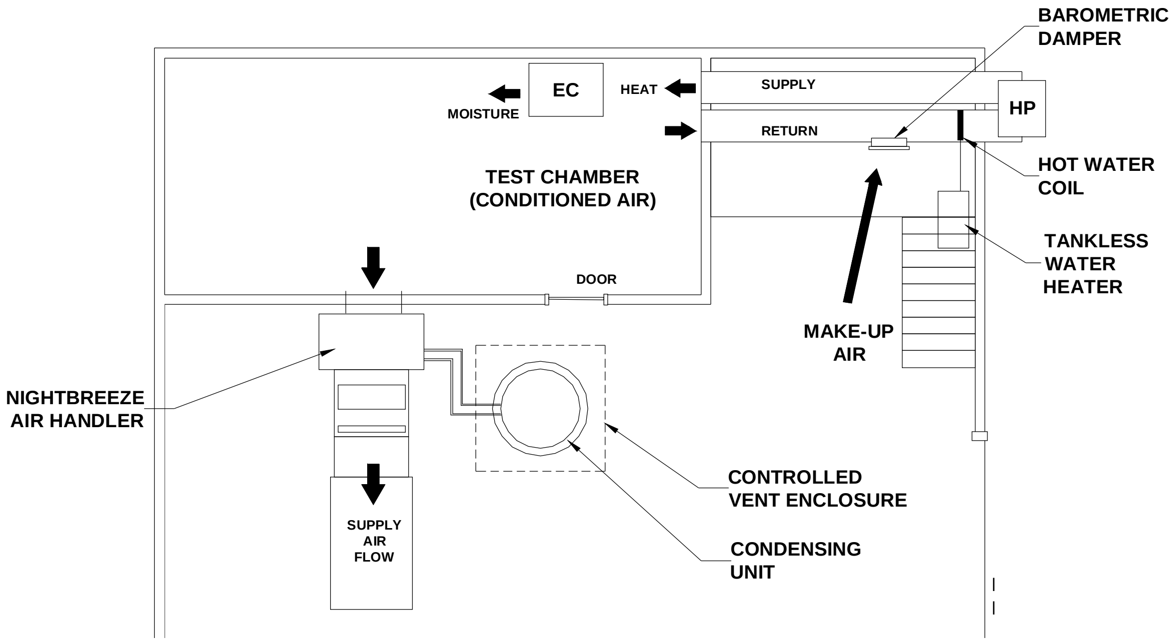

Figure 8: Test Chamber Configuration

Table 2: Monitoring Equipment Specifications

Type	Measurement	Mfg./Model	Signal	Span	Accuracy
Temperature and RH	Supply and return air	Vaisala HMD60	4-20 mA output	50-130°F 20-90%	±0.5°F ±2%
Flow Grid	Airflow rate	Energy Conservatory TrueFlow airflow meter	Pressure differential	500-2100 cfm	±7%
Power Monitor	Condensing unit and air handler power	Rochester Instruments PM-1006-240-K0-D-0	4 pulses per Watt-hour	n/a	±0.5%

4.4.2. Prototype Test Configuration

The SBIR Phase I report ("Development of an Integrated Heating, Ventilation, Cooling, and Dehumidification System" – April 2004) details the initial testing of the original Phase I prototype system, including a variable speed air handler, a single stage condensing unit, evaporator coil, and reheat coil. Early in Phase II the original prototype coil assembly was re-tested with a two-stage condensing unit instead of the single speed condenser used in Phase I. The initial Phase II testing included the following components:

- Amana RSG48C2 two-speed condensing unit (nominal four ton capacity)
- NightBreeze air handler model # NB10-2-120A (1 hp blower motor)
- Goodman CAPF042C4 3 row evaporator coil, 15 fins/inch, raised lance fins
- 5 circuit, 2 row Super Radiator subcooling coil (10 fins/inch, raised lance fins)

Upon completion of the packaged coil assembly, further testing was conducted using the same test configuration, substituting the up-flow prototype packaged coil for the horizontally mounted original prototype. To conform to the size of the evaporator coil, a slightly smaller reheat coil was used with the second prototype design.

Figure 9 shows the location of the supply and return airstream temperature sensors and the TrueFlow airflow measurement grid. A DataTaker model DT-50 datalogger was used to continuously monitor system performance as well as provide environmental control for the test chamber. A thermocouple grid was installed to average multiple temperature readings of the air leaving the evaporator coil. Supply airflow readings and refrigerant pressures were recorded at the start, middle, and end of each test.

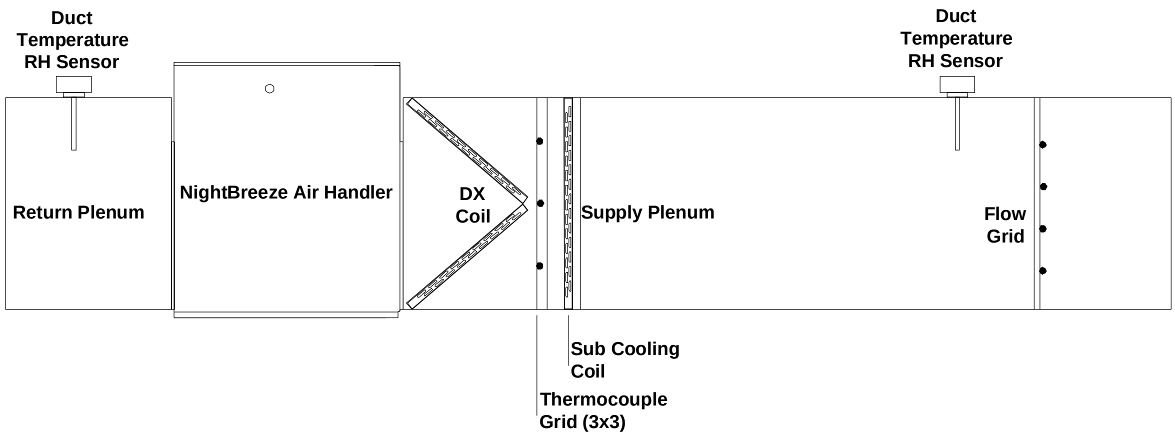

Figure 9: I-HVCD Test Configuration

4.4.3. Test Conditions

A matrix of test conditions was developed to evaluate the performance of the system under a variety of specific conditions of "indoor" temperature and relative humidity, condensing temperatures and pressures (reflecting outdoor conditions), air flow rate, refrigerant charge, operating mode, and other parameters. The following procedures were followed for each set of conditions:

1. *Turn on evaporative cooler and chamber supply fan to initiate chamber air mixing.*
2. *Set up datalogger program for test.* Modify the datalogger program to set the chamber target temperature and relative humidity for the specified test. Download the program and allow the chamber temperature and RH to approach the target conditions.
3. *Activate the air handler fan.* Once chamber conditions have approached within 3°F of the test target temperature, activate the system in cooling mode and set the air handler airflow desired for the particular test.
4. *Allow chamber conditions and condenser environment to stabilize.* Allow the DT-50 to cycle heat and moisture supply to the test chamber until stable environmental conditions are achieved. Adjust condenser "chamber" to maintain desired condenser air temperature (or refrigerant condensing pressure). Operate the system for at least 10 minutes before recording data to allow the refrigerant system to achieve steady state conditions and to allow the evaporator coil to be fully wetted.
5. *Start test after conditions have been stable for five minutes.* Empty condensate catch bucket. Record time when test begins. Run steady state test for 20 minutes logging temperatures and relative humidity at one-minute intervals. At the end of 20 minutes,

record the volume of condensate collected. (If at any time conditions deviate from the tolerances, restabilize chamber conditions and repeat the test.)

6. *Document test results.* Record and file both manually taken data (from gauges) and electronically logged data.

Results from testing prompted modifications to some test conditions to zero in on particular performance parameters. The following data were recorded from each test:

1. Airflow measured by TrueFlow grid at start and end of test (CFM_{TF})
2. High and low side refrigerant pressures at start, middle, and end of test
3. Condensate volume collected during test (VOL_C in ounces)
4. One minute interval supply and return air temperature and RH, and air handler and condensing unit average power demand
5. High and low side refrigerant pressures at start and end of test

4.4.4. Analysis of Test Data

The sensible cooling rate was calculated using one-minute supply and return air temperatures and average airflow. Latent cooling was determined from the volume of condensate collected. Operating efficiency was calculated by dividing total cooling by the sum of the condensing unit and air handler demand. These calculations are defined in Equations 1-3.

Eqn 1: $Q_{sensible}$ (Btu/hour) = 1.08 x CFM x ($T_{supply} - T_{return}$)

Eqn 2: Q_{latent} (Btu/hour) = 211.02 x VOL_C

Where 211.02 = 1080.84 (Btu/lb latent heat of condensation at 45°F) times 8.33 pounds per gallon times 0.00781 gallons per ounces times 3.0 (three 20 minute intervals per hour).

Eqn 3: EER (Btu/Watt-hour) = ($Q_{sensible}$ + Q_{latent}) / Average System Demand (W-hrs)

4.5. Field Test Methodology

Davis Energy Group collaborated with subcontractor Steven Winter Associates (SWA) to secure a new home monitoring site that could accommodate field monitoring for two full summer seasons. Using climate and construction schedule criteria, a Gainesville, Florida site was selected from various projects that SWA was managing under the Building America program. The 3,080 ft^2 two-story house (floor plan and site photos in Appendix B) was to be monitored using a base case system during 2006, and with the I-HVCD prototype in 2007. The 2006 base case system was selected to represent “best available practice” HVAC components that would establish a performance level for comparison to the I-HVCD. The installed base case HVAC system featured the following components:

- Two-stage 3 ton Amana RSG condensing unit (nominal 16 SEER, 12 EER)
- Takagi T-KD20 gas tankless water heater for space and domestic hot water heating
- NightBreeze variable speed hydronic fan coil unit with evaporator coil
- Thermastor UltraAire UA-100V integrated dehumidification and fresh air ventilation system (rated at 100 pints/day and 2.51 liters/kWh).

Product literature on the condensing unit, NightBreeze unit, and UltraAire unit can be found in Appendix C.

The principal goal of the 2006 monitoring was to document best practice performance in terms of cooling delivered, energy consumed, indoor/outdoor conditions, and overall operating efficiency. The monitoring strategy developed and implemented by SWA allowed for collection of the following data:

- cooling energy delivered (latent, sensible, and total)
- cooling energy consumed (air conditioning and supplemental dehumidification)
- overall efficiency (effective EER)
- indoor and outdoor conditions (temperature and relative humidity)

All sensors were scanned on a 15 second time interval, and data were summed or averaged, and stored in datalogger memory on 10 minute intervals. Although datalogger memory was sufficient to store several months of data, the loggers were downloaded and reviewed on a weekly basis using telephone modems to insure data integrity. Downloaded data were saved to a central computer and screened using automated software to verify that the collected data was within reasonable sensor range. Monitoring points for evaluating performance are shown in Table 3.

Table 3: Installed Monitoring Sensors

Abbrev.	*Description*	*Location*	*Sensor*	*Part Number*
ATRH1	Ambient (OA intake) air temperature / RH	outdoor	Humirel 2500	HTM2500-ND
			+ radiation shield	41303
ATRH2	Indoor air temperature / RH	1st floor	Vaisala temp/RH	HMW40Y
ATRH3	Indoor air temperature / RH	2nd floor	Vaisala temp/RH	HMW40Y
ATRH4	Return air plenum temperature / RH	return plenum	Vaisala temp/RH	HMD40Y
ATRH5	Supply air plenum temperature / RH	supply plenum	Vaisala temp/RH	HMD60Y
ATRH6	Supply air plenum temperature / RH	supply plenum	Vaisala temp/RH	HMD60Y
ATRH7	Dehumidifier outlet air temperature / RH	mechanical room	Vaisala temp/RH	HMD40Y
P1	Compressor power	electrical panel	CCS Wattnode	WNA-1P-240-P
			CCS current transformer	CTT-0300-030
			CCS current transformer	CTT-0300-030
P2	Air handler unit power	electrical panel	CCS Wattnode	WNA-1P-240-P
			CCS current transformer	CTT-0300-005
P3	Dehumidifier power	electrical panel	CCS Wattnode	WNA-1P-240-P
			CCS current transformer	CTT-0300-015
FR1	Return plenum flow rate	return plenum	AMC Fan-e	custom order
			Setra pressure transducer	2641-0R1WD-11-T1-C
C1	Condensate (rain gauge)	air conditioner	Texas Electronics	TR-525USW-R3
C2	Condensate (rain gauge)	dehumidifier	Texas Electronics	TR-525USW-R3

Sensible cooling capacity was calculated on 15-second intervals using Equation 1. Latent cooling was determined for both the air conditioner and the dehumidifier based on both the volume of condensate collected[13] and the enthalpy difference between supply and return air. Since there is a time lag between condensate flow and the start of cooling operation, total cooling (latent plus sensible) and overall EER[14] were calculated on a daily basis. Enthalpy calculations were used to calculate total cooling capacity based on system airflow and supply/return air temperature and relative humidity. One shortcoming of enthalpy-based calculations of latent cooling is that the method does not accurately account for re-evaporation of condensate during the system off-cycle.

Figure 10 shows the configuration of the mechanical closet at the Gainesville house during the 2006 base case test. The UltraAire dehumidifier was ducted both to the return plenum and to a 10” outdoor air duct. The unit provided 70 cfm of outdoor air to satisfy ventilation requirements.

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[13] The latent heat of condensation is equal to 1080.8 Btu/lb at a temperature of 45°F.
[14] Overall EER equals total cooling in Btu’s divided by total energy consumed in Watt-hours.

When the unit operated in response to a dehumidification call, return air was dried and heated by the dehumidifier, and then delivered to the supply plenum.

In 2007 the existing Amana cooling coil was replaced by the I-HVCD coil assembly, NightBreeze controls were replaced with prototype I-HVCD controls, and the dehumidifier was removed. Figure 11 represents the I-HVCD configuration. The return air damper was designed to operate in tandem with the outdoor air damper, so that when the system enters outside air ventilation mode (for fresh air ventilation or ventilation cooling), 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. For ventilation cooling the fan speed increases to about 400 cfm.

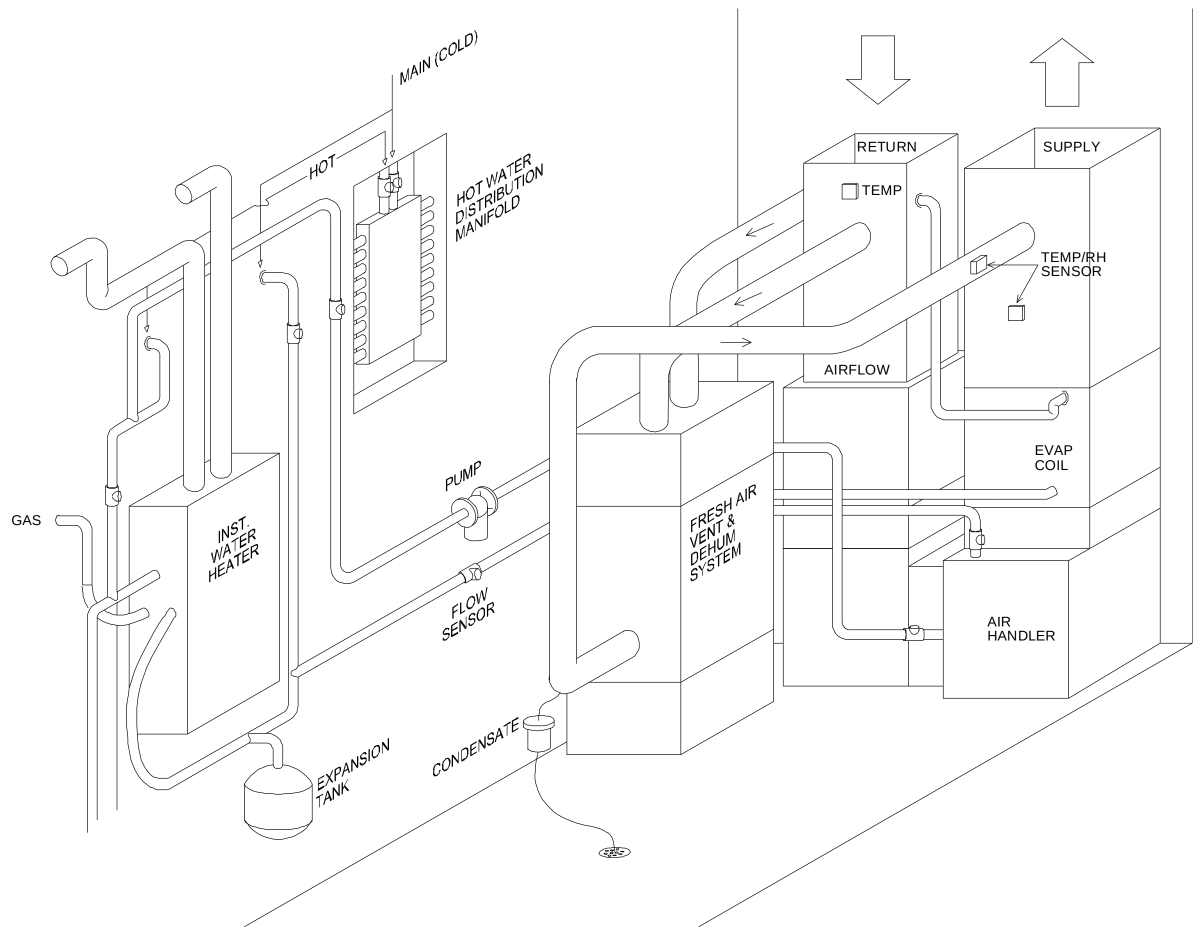

Figure 10: Base Case HVAC System Configuration

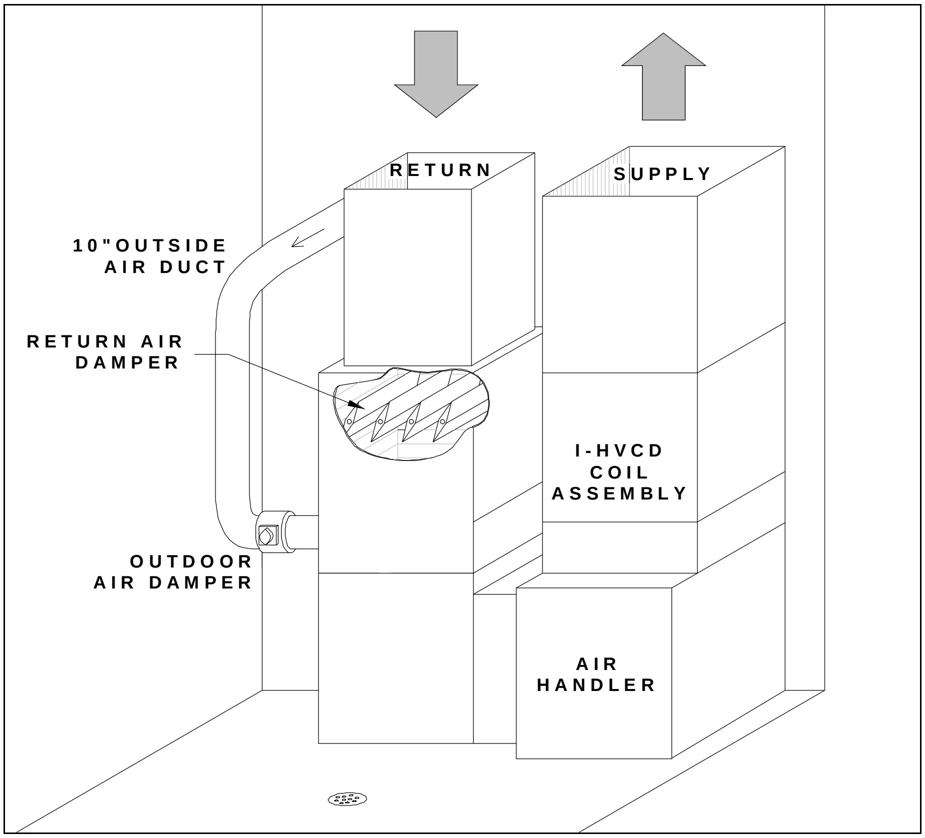

Figure 11: I-HVCD Mechanical System Layout

5. Evaluation and Test Results

5.1. Ventilation Cooling Evaluation

Table 4 summarizes the average electric rate[15], base case annual cooling energy use, projected savings (kWh and cost), and simple payback (based on a $900 incremental cost for the ventilation cooling components). Only four of the 16 locations modeled are projected to have simple paybacks of less than 20 years using the 0.6 cfm per square foot ventilation system sizing. Sacramento, El Paso, and Tucson all have fairly dry climates and are well-suited for ventilation cooling. Surprisingly Houston also was projected to have a 12 year simple payback despite notoriously humid summer conditions. High annual cooling loads in Houston offer significant ventilation cooling savings potential during the shoulder cooling seasons (spring and fall). For the other twelve locations the most favorable economics were projected to be in Jacksonville, Florida with a 21-year simple payback. The relatively poor projected economics[16] suggest that it is difficult to justify the added cost of full-sized ventilation cooling dampers and ducting that can provide the 0.6 cfm per square foot ventilation level.

[15] Edison Electric Institute average 2004 statewide electric rates
[16] It should be noted that rising electric rates and increased production volumes could conceivably result in the paybacks shown in Table 2 being cut in half.

Table 4: Projected Cooling Savings with Full-Size Ventilation System

Location	Average Electric Rate ($/kWh)	Annual Base Cooling kWh	Estimated Vent Cooling Savings kWh/year	Cost	Simple Payback (yrs)
Sacramento, CA	$0.14	1554	537	$75	12
Chicago, IL	$0.09	1227	186	$17	54
Ft Wayne, IN	$0.08	1106	167	$13	72
Salt Lake City, UT	$0.08	2149	397	$30	30
Tucson, AZ	$0.09	7309	945	$85	11
Philadelphia, PA	$0.11	2351	279	$29	31
Jacksonville, FL	$0.09	5703	475	$43	21
Albany, NY	$0.14	688	152	$21	42
El Paso, TX	$0.09	5910	988	$89	10
Bismarck, ND	$0.08	618	214	$16	56
Atlanta, GA	$0.08	3554	398	$30	30
Houston, TX	$0.09	7830	803	$72	12
Raleigh, NC	$0.09	3105	446	$40	22
St. Louis, MO	$0.08	3517	390	$29	31
Memphis, TN	$0.06	5109	500	$30	30
Denver, CO	$0.09	684	223	$20	45

5.2. LabVIEW Testing

The proprietary control logic developed by Davis Energy Group and refrigeration engineer Jim Phillips was designed to provide precise control of refrigeration system operation in response to indoor temperature and humidity variations. The logic was developed to optimally transition from “typical” vapor compression system operation (low or high speed condensing unit with indoor fan supplying about 400 cfm/ton) to modes with decreasing sensible heat ratios.

The first step in the control process involves incrementally reducing the supply airflow to lower the SHR from 0.8. If indoor RH could not be adequately maintained without overcooling, reheating of supply air would be added, also in a staged manner. Through this control sequencing, the system SHR would range from 0.8 to zero. Before the control logic algorithms and flow chart were provided to our controls programmer Harlan Strickland, the algorithms were coded and tested into LabVIEW. LabVIEW is a graphical tool that can be used for development and testing of control application and measurement programs. Figure 12 depicts one of the LabVIEW display screens developed for the testing. System control inputs are shown on the right hand side of the screen, and outputs are shown on the left hand side. The LabVIEW routines were rigorously tested, first with the computer, and later with I-HVCD prototype refrigeration hardware.

5.3. I-HVCD Laboratory Test Results

I-HVCD laboratory test results demonstrated that the system is capable of exceeding the moisture removal goals established in the original (Phase I) SBIR proposal. Tests verified that reducing the airflow rate across the optimized evaporator coil from a nominal 400 cfm per ton to 200 cfm per ton significantly increases moisture removal at the indoor coil. By utilizing the subcooling coil to reheat the cool air exiting the evaporator coil, the overall system sensible heat ratio (SHR) could be further reduced. In partial condensing mode operation the system was shown to essentially operate as an efficient dehumidifier with little or no thermal impact on the conditioned space (supply air temperature equal to return air temperature).

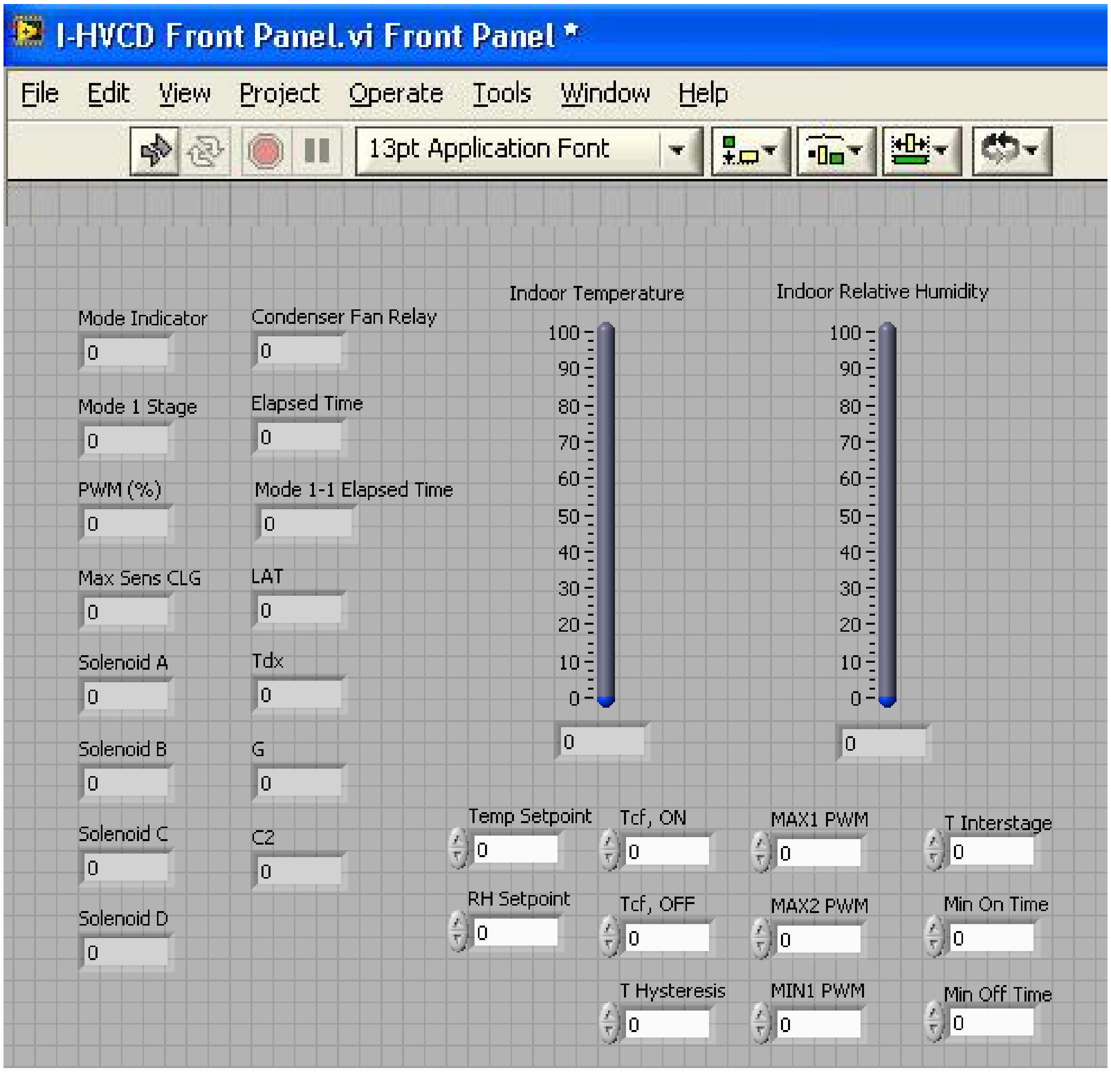

Figure 12: Sample LabVIEW Control Test Panel

Table 5 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[17]. 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[18].

[17] 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.
[18] "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.

Table 5 sensible and total cooling is reported for both “I-HVCD” and “Cooling”, with the “I-HVCD” 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[19].

The I-HVCD SHR ranges from close to 0.8 in Test 1 to 0.5 in subcooling tests 4 and 5, down to 0.02 in partial condensing mode operation in Test 8. The I-HVCD EER falls off with diminishing SHR as reheating increases, especially at levels below 0.5. The last section of Table 1 summarizes moisture removal characteristics of the system under the varying operating modes and supply airflow levels, with the last row reporting condensate removal per kBtu of sensible cooling delivered. 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.

Table 5: 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*

One control feature of the I-HVCD system is the ability to control the condenser fan during partial condensing operation. In partial condensing mode, the system operates to dehumidify the air but not to further cool the space. By controlling the condenser fan, the level of heat rejection to the subcooling coil can be affected. With the condenser fan on, more heat will be rejected at

[19] Calculated by multiplying the total AC cooling energy use by (1-SHRttl).

the condensing unit; with the condenser fan off, virtually all condenser heat rejection will occur at the subcooling coil. Figure 13 plots supply air temperature variations[20] as the condenser fan is shut off. When the condenser fan is shut off, the supply air temperature initially rises at a rate of ~1 to 1.5°F per minute. The I-HVCD's leaving air temperature sensor allows for control of the condenser fan relay to ensure that the supply air temperature varies within ±5°F of the indoor temperature.

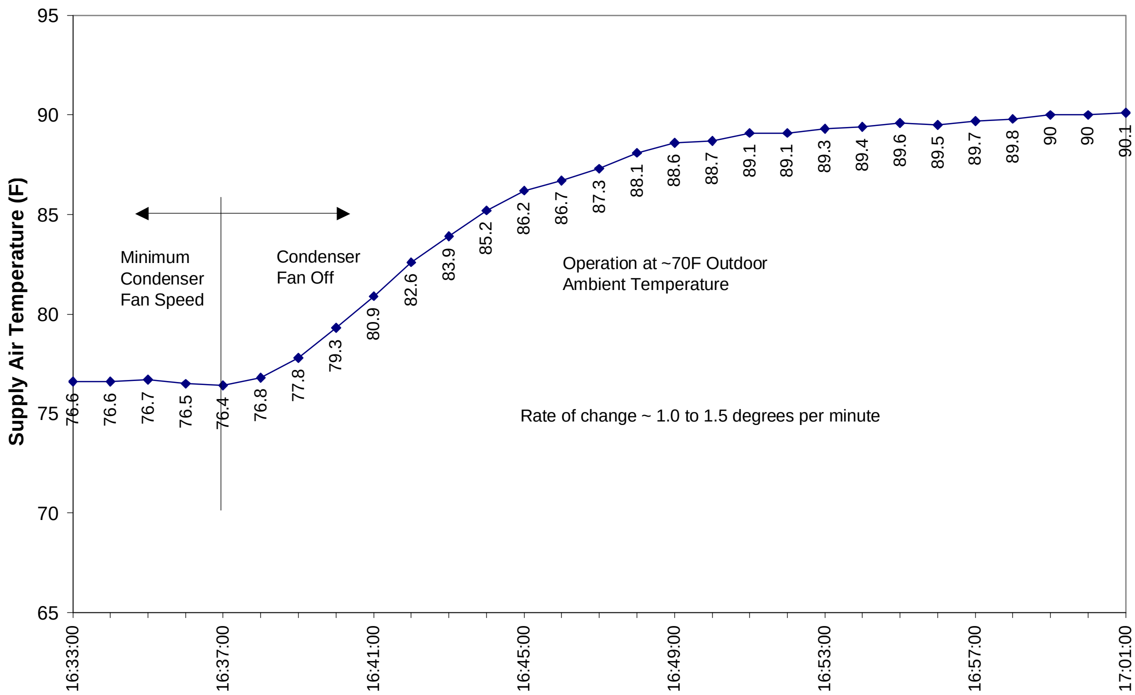

Figure 13: Impact of Condenser Fan Cycling on Supply Air Temperature

Figure 14 shows the prototype I-HVCD coil assembly (evaporator coil, subcooling coil, and refrigerant valves) installed at DEG's testing facility. The front sheet metal access panel of the unit was temporarily replaced with a plexiglass window to allow observation of condensate and/or frost[21] formation on the evaporator coil. Upon completion of the lab testing, the plexiglass window was removed and the prototype coil assembly and controls package was shipped to Florida for installation at the field test site.

[20] at ambient temperatures of about 70°F surrounding the condensing unit
[21] Since the I-HVCD controls allow airflow as low as 150 cfm/ton, lab testing evaluated whether coil icing would be a problem under varying conditions.

Figure 14: I-HVCD Prototype Coil Assembly

5.4. 2006 Base Case Field Test Results

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 system control strategies[22]. The first period (May 1st to July 30th) was characterized by conventional two-stage cooling operation, little outdoor air ventilation[23], 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[24].

Table 6 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.

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

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

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

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

Table 6: 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 (dehumidifier represents 21% of the total). 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 system + dehumidifier SHR 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 SHRttl 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 installed dehumidifier has a rated nominal efficiency of 2.51 liters/kWh.

Figure 15 plots weekly energy consumption for each of the energy consuming components[25]. 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.

5.5. 2007 I-HVCD Field Test Results

The original project goal was to field monitor the I-HVCD system for the full 2007 summer. Unfortunately the complexities of completing and testing the prototype control system delayed shipment of the I-HVCD coil assembly and controls until early September 2007. (Although Florida weather provides significant warm and humid conditions well into October, the truncated monitoring period resulted in a shortened window for system performance review and a limited ability to assess and modify system control functions.) Installation of the I-HVCD prototype system in the Florida home was completed on September 13, 2007 with the help of Davis Energy

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

Group staff and the local HVAC contractor (Mark Hurm and Co.). Steven Winter Associates engineers were also on hand to modify the monitoring system to accommodate the installed I-HVCD hardware. Installed HVAC components included the replacement indoor coil assembly, I-HVCD controls (wall display unit and control board located at the air handler), and a condenser fan relay connected to the I-HVCD CCU and the electrical circuit powering the condenser fan.

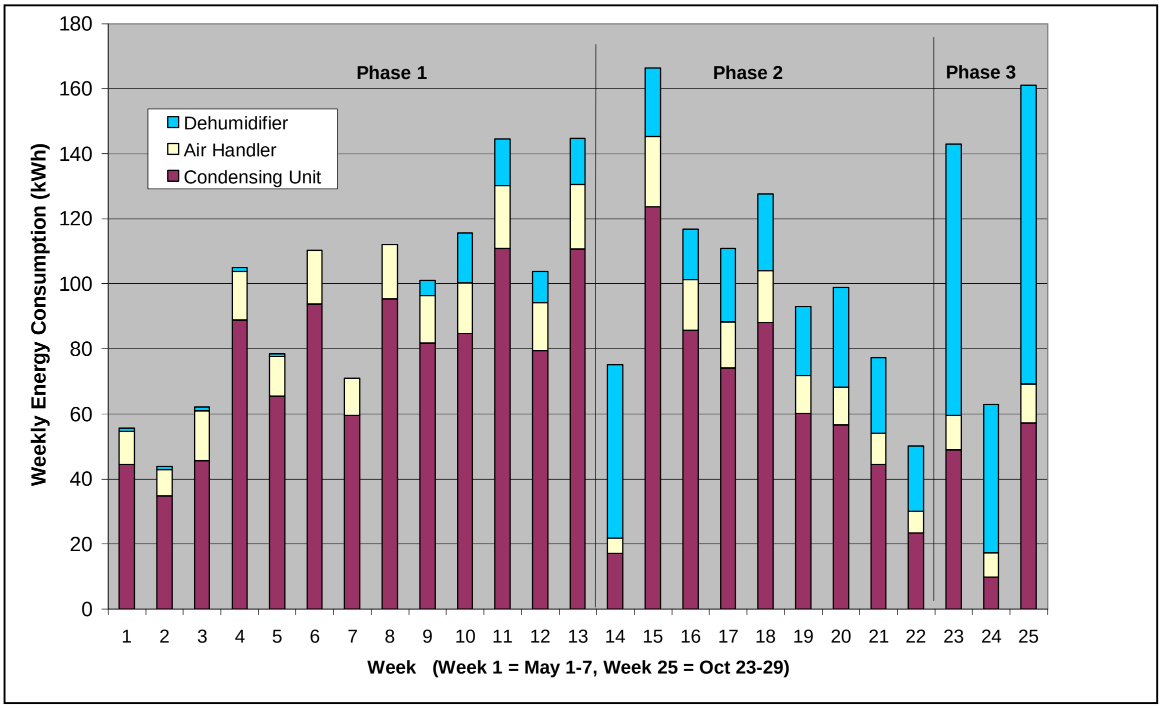

Figure 15: Weekly 2006 Base Case Energy Consumption

The one component that was unavailable for installation was the return air damper. The HVAC contractor had procured a damper, but the damper actuator motor was positioned such that the motor conflicted with other system hardware in the mechanical closet. The contractor attempted to get a substitute damper, but the non-standard nature of the request delayed procurement to late October. The absence of the damper precluded testing of full I-HVCD system performance since outdoor air could not be delivered to the space. This was unfortunate both in terms of fully demonstrating fresh air ventilation performance[26], but also in providing a direct comparison to the 2006 data where fresh air ventilation was being provided for roughly half the summer. Despite this shortcoming, detailed cooling and dehumidification operating data was collected from September 13th through early November, although most of the cooling/dehumidification operation ceased by mid-October. Figure 16 characterizes weather during the period from September 13th through October 17th by plotting outdoor daily temperatures maximums and minimums. Outdoor maximums reached or exceeded 90°F on seven days.

[26] Fresh air ventilation control logic involves closing the return damper and opening the outdoor damper and operating the fan at low speed. This can occur during either heating or cooling operation, or at 45 minutes past the hour, if no HVAC operation occurred that hour. Monitoring data clearly show fan operation beginning at 45 minutes past the hour, indicating proper fan operation despite the lack of the dampers.

The I-HVCD system operated reliably throughout the test period. Detailed review of the one-minute interval data indicated the need for further evaluation of system operating characteristics to determine how existing control algorithms should be modified. One aspect in particular relates to how supply fan speed control is implemented. The control algorithm looks at how close the indoor RH is to the user-defined target RH (or if it exceeds the target) and proceeds to adjust supply airflow, and optionally add reheat at the subcooling coil. Evaluating the field monitoring data suggests that the supply airflow range likely should be modified. Other issues that require additional field-testing include determining the transition points for when reheating is initiated, and at what point reheating transitions to the partial condensing mode when SHR's approach zero. Since it is less efficient than the normal sensible cooling operation (at 400 cfm/ton), reheat operation should be initiated as little as possible, while maintaining acceptable indoor comfort.

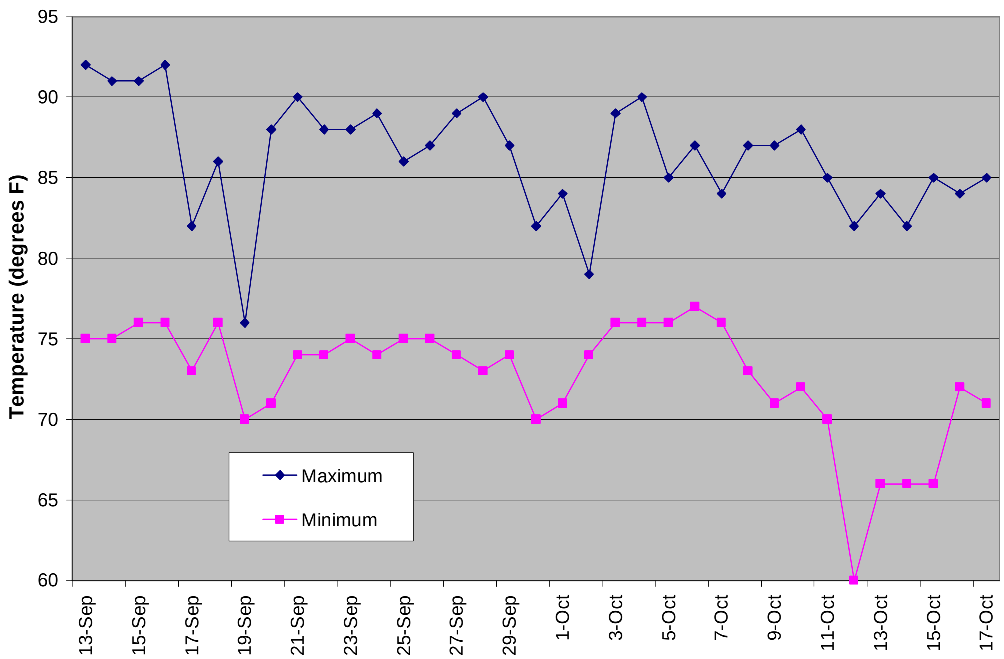

Figure 16: Recorded Daily Outdoor Temperature Extremes

Table 7 compares outdoor and indoor monitored conditions for the base case monitoring (Phases 1 – 3) and the September/October 2007 I-HVCD monitoring. Phase 1 operation (May 1 – July 30, 2006) was represented by typical two-stage AC operation with little supplemental dehumidification or fresh air ventilation provided by the UltraAire system (the control system was not fully installed). Phase 2 operation (July 31 – October 1, 2006) has indoor comfort settings of ~75°F cooling setpoint, 50% RH on the UltraAire system, and fresh air ventilation set at 60 cfm[27]. Phase 3 includes data from the end of 2006 season (October 2nd – 29th) and the beginning of the 2007 season (June 18th – July 15th). During this time the UltraAire RH setpoint was lowered to 45% to reduce the average indoor RH level. As shown in Table 7, the maximum indoor humidity levels in Phases 1 – 3 show a significant variation from the average, in contrast to I-HVCD. Improved indoor humidity control appears to be a clear I-HVCD system benefit.

[27] The UltraAire control was not properly setup for weekend operation until September 12, 2006. Prior to this date, weekend fresh air ventilation and humidistat control was disabled.

Figures 17 and 18 highlight the relative humidity control characteristics of the two systems in more detail. Figure 17 plots base case 15-minute data for the seven-day period of August 21 – 27, 2006. This plot clearly demonstrates the RH variations observed with the base case system as the average indoor RH ranged from 48 to 62%. The UltraAire dehumidifier/ventilation air system provided ventilation air during the non-weekend days of the week[28]. Three dehumidification cycles are clearly showed on the graph with durations ranging from approximately 2.5 to 4.5 hours. During these cycles, the average indoor RH would drop 10-12%.

Table 7: Comparison of Base Case vs. I-HVCD Monitored Operating Conditions

Mode	Average Outdoor Temp	Average Outdoor RH	Average Indoor Temp	Average Indoor RH	Maximum Indoor Temp	Maximum Indoor RH
1	80.2°	73.0%	74.3°	48.2%	83.2°	62.6%
2	80.9°	81.2%	77.4°	53.5%	85.3°*	64.0%
3	77.1°	79.5%	75.7°	41.8%	77.2°	58.1%
I-HVCD	78.1°	86.4%	75.1°	46.4%	82.1°**	54.2%

"*" thermostat was turned off for ~ 6 days
"**" peak temperature and RH occurred when I-HVCD prototype was first turned on (September 13th)

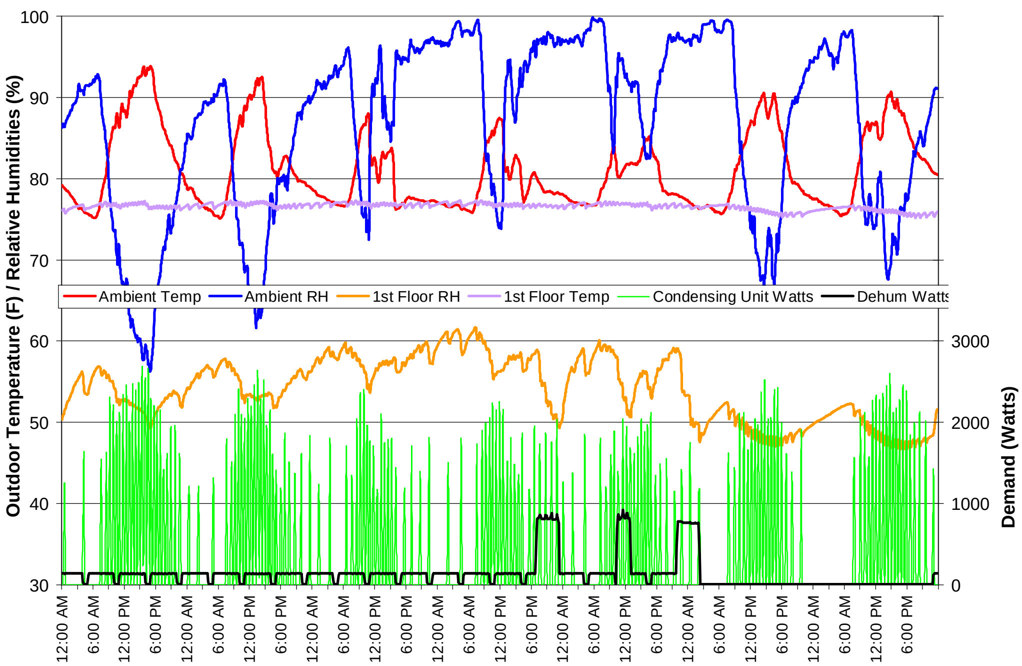

Figure 17: Base Case Operation for the Week of August 21-27, 2006

Figure 18 plots one-minute interval I-HVCD data from September 19, 2007. The weather this day was characterized as a tropical storm day (very mild and humid). The I-HVCD system operated infrequently (relative to Figure 17 base case operation) due to the low sensible cooling loads. The I-HVCD system maintained a perfectly flat indoor temperature profile and controlled indoor RH fluctuations to a maximum deviation of 2% by both modulating supply airflow and also performing reheating.

[28] Later in the summer the control was reprogrammed to provide ventilation air seven days a week.

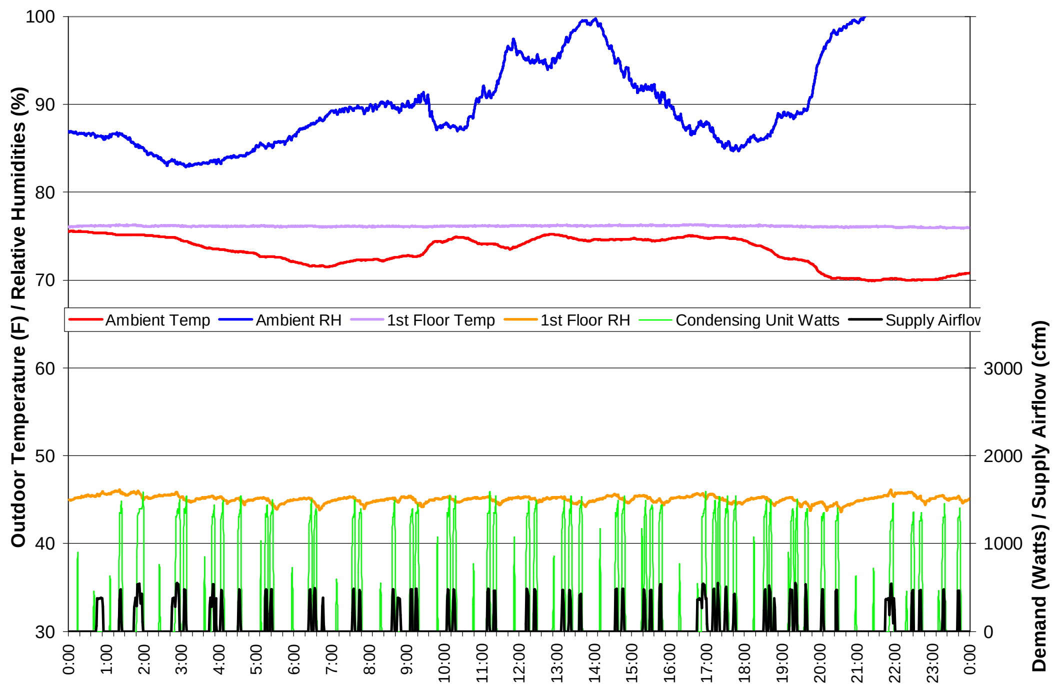

Figure 18: I-HVCD Operation (September 19, 2007)

Figures 19 and 20 provide additional one-minute interval I-HVCD data. Figure 19 plots an operating cycle on October 10th when outdoor temperatures were in the mid-80's. The I-HVCD starts operating in first stage cooling (blue condensing unit kW and airflow lines shown on right Y-axis). After twelve minutes of operation the unit transitions to 2nd stage cooling delivering the nominal 1200 cfm of airflow. During the cycle, indoor RH is close to 50% at the beginning and decreases a few percent during the cooling cycle.

Figure 20 depicts an early morning cooling/dehumidification cycle on October 14th. Outdoor temperatures are in the upper 60's, while indoor temperatures are in the mid-70's with RH slightly over 50%. The I-HVCD unit starts in first stage cooling with supply air temperatures approaching 60° at which time the unit transitions to full reheat mode to avoid overcooling. In reheat mode the system operates less efficiently but delivers neutral air (~75°F) to the space. By reheating the air, supply air RH is reduced from 75% to roughly 40% and supply airflow is continually reduced from 470 to 340 cfm. By the end of the cycle the airflow increases slightly as the thermostat senses that the indoor RH target has been satisfied. During the full operating cycle indoor temperature did not change, while indoor RH was reduced by ~3%.

Table 8 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 (full data included in Appendix D). 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 8 is the typically small daily variation in indoor RH from average to maximum[29] (<2% typical) relative to the ± 5-7% daily variation observed during 2006 base case monitoring.

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[29] September 13th shows a larger variation since this was the I-HVCD startup day and the house had been unconditioned for more than a day.

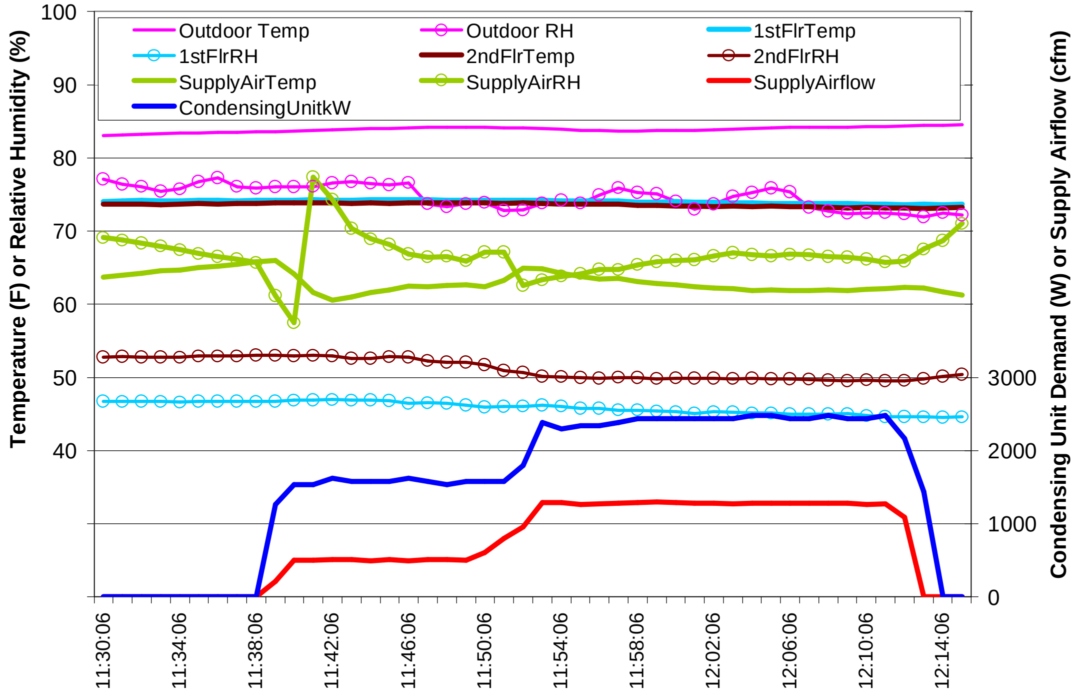

Figure 19: I-HVCD Operation (October 10, 2007)

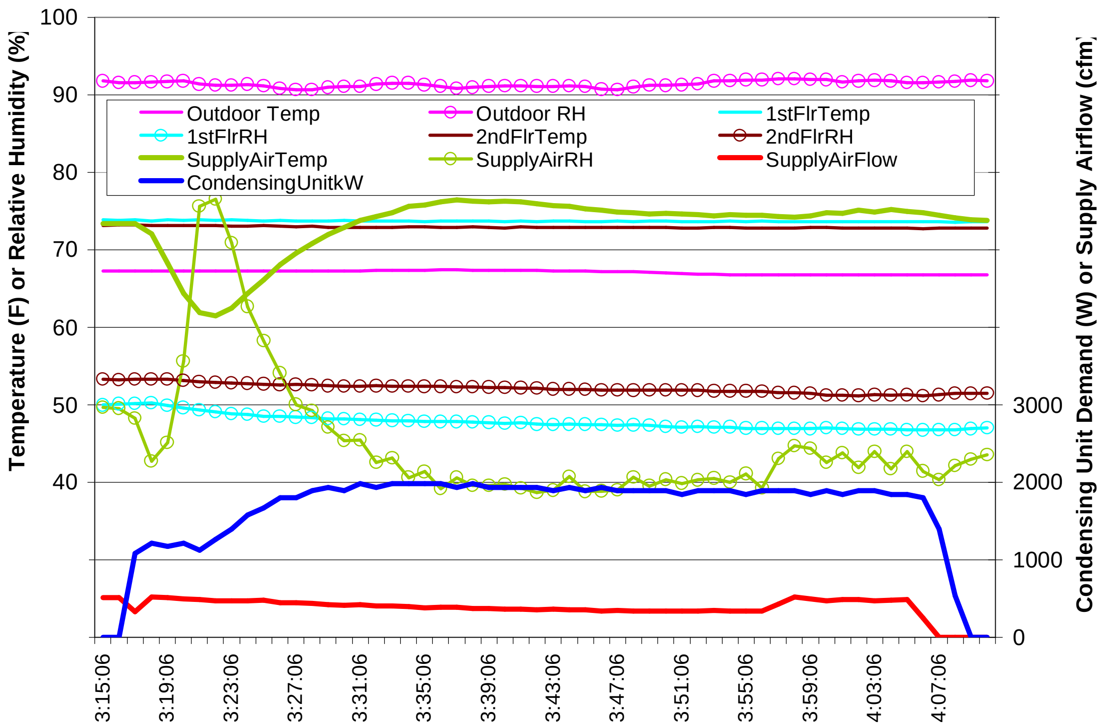

Figure 20: I-HVCD Operation (October 14, 2007)

Table 8: 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

A qualitative effort was made to compare overall energy efficiency between base case and I-HVCD operation. Figure 21 plots daily total energy consumption (condensing unit, air handler, and dehumidification system) for the three base case operating phases[30] 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 (6-12% lower average RH than Phases 1 and 2) resulted in energy use roughly double that of Phases 1 and 2. I-HVCD monitored energy consumption falls somewhere between Phase 3 and Phases 1 & 2[31], suggesting that I-HVCD performance is clearly within the range of expected best practice system performance. Clearly more testing is needed to better understand I-HVCD system performance.

5.6. Preliminary I-HVCD Economics

In addition to system performance, a key consideration for an advanced space conditioning system such as I-HVCD is cost, both first and operating. Based on the information collected from the Gainesville field test site, a preliminary cost comparison was developed assuming a two-stage condensing unit with variable speed air handler (or furnace) as the base case system. Table 9 provides an installed cost comparison for the base case system installed at the Madera house and the I-HVCD system. The reported base case costs were obtained from the installing contractor and include a 30% markup on components. I-HVCD costs are rough estimates based on actual component costs[32], if and when the technology becomes commercialized. Cost projections indicate a small potential cost savings for the I-HVCD system. Further refinement of these costs will occur as the product moves closer to the commercialization stage.

[30] 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.
[31] The lack of a fresh air ventilation load during I-HVCD monitoring reduced the latent load on the system.
[32] I-HVCD control costs are based on the currently available NightBreeze2 control ($1395) plus an estimated additional $100 for the RH sensor and added outputs on the control board. Current control costs will come down significantly as product volumes increase.

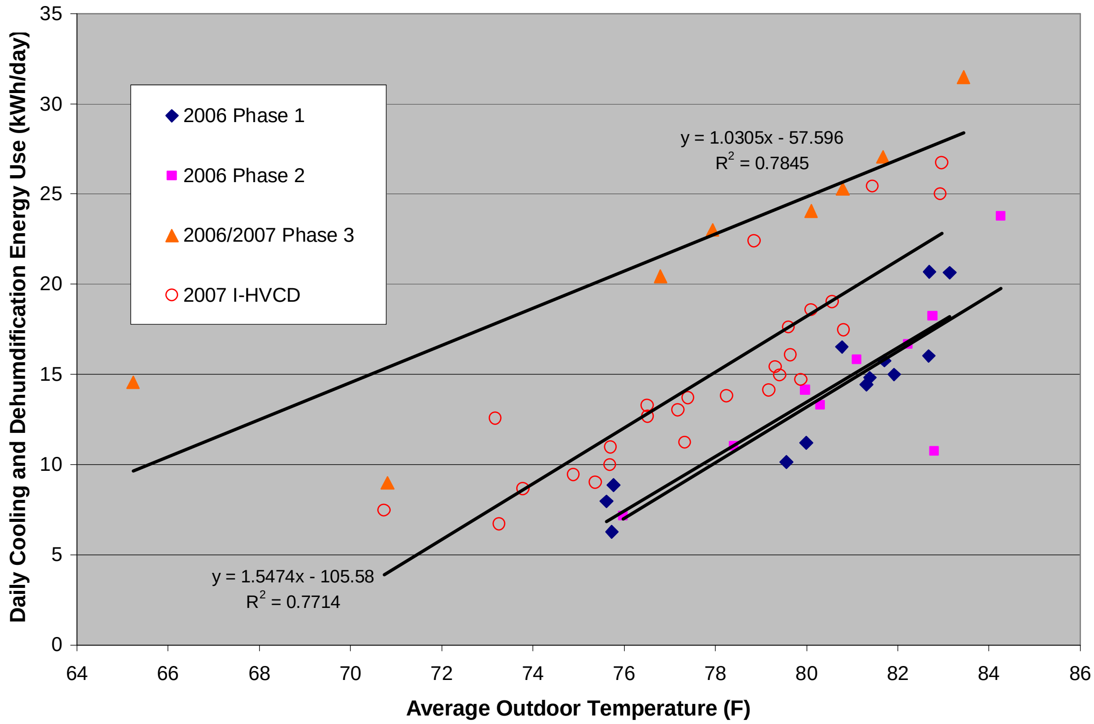

Figure 21: Daily Energy Use by Mode

Table 9: Projected Installed Cost Comparison

	Base Case System	I-HVCD
Dehumidifier/Fresh Air Ventilation System	$2,350	n/a
I-HVCD Coil Assembly	n/a	$700
Other Components (controls, damper, etc.)	$620	$1,950
Labor	$500	$500
Total	$3,470	$3,150

In terms of operating costs, the preliminary results shown in Figure 21 suggest that the I-HVCD system should consume approximately the same amount of energy as the base case system monitored at the Gainesville house, while offering significantly improved RH control[33]. We feel that there is room for improvement in I-HVCD operating efficiency by optimizing the control transitions from Mode 1 to the less efficient Modes 2 and 3. More field experience is needed with the I-HVCD system to better understand where these transitions should occur. A final I-HVCD benefit not quantified in the field monitoring is the ability of the system to deliver night

[33] Improved RH control should translate to a higher indoor temperature setpoint for equivalent comfort, resulting in reduced sensible loads.

ventilation cooling during favorable outdoor conditions. Night pre-cooling will generate some level of savings that weren't captured in the Florida monitoring.

6. Commercialization Activities

DEG collaborated informally with Amana Heating & Air Conditioning throughout both Phases I and II of the I-HVCD SBIR project. Amana, one of the largest manufacturers of residential and small commercial HVAC equipment, has expressed continued interest in keeping abreast of project developments. Amana's involvement in the project has included donation of two-stage condensing units and evaporator coils for I-HVCD development, and engineering support on technical issues such as details related to motor programming. Amana has indicated an interest in this technology including possibly manufacturing the system. These discussions will continue after Amana has reviewed the final report.

Other commercialization efforts have included preliminary discussions with Aprilaire, a Midwest-based manufacturer of whole house dehumidifiers. Contacted at the recent ASHRAE 2008 Winter Meeting in New York, Aprilaire expressed interest in the technology and would be a viable alternative manufacturing partner.

Also at the ASHRAE Winter Meeting, design engineers were queried about the need for a whole house ventilation system that is integrated with heating, air conditioning, and ventilation. There was broad support for such a product, particularly from design engineers working in the southern U.S. They underscored the need for dehumidification on mild days to avoid discomfort, as well as an improved strategy to addressing indoor air quality and mold problems. Similarly, a HVAC distributor from Florida, queried at the GreenBuild conference in November 2007, underscored the need for such a product and was enthusiastic about distributing it.

In sum, it is clear that the market perceives a need for I-HVCD technology. The benefits for home occupants include comfort, improved indoor air quality, potential health benefits from enhanced IAQ, and energy savings. The capability to manufacture the product is clearly available, and interest has been shown by at least two manufacturers. DEG's involvement with Building America provides a direct connection to leading research teams and homebuilders who are struggling to solve the problem of maintaining humidity control in thermally efficient new houses. Contacts have been initiated with the design community as well as the marketing and distribution community. The outlook is positive, and to move these discussions to a more meaningful stage, additional testing and product refinement is needed, so that more definitive results can be shared with air conditioning manufacturers.

7. Conclusions

The key objective of the Phase II SBIR project was to develop and demonstrate an integrated appliance that can efficiently and more economically provide indoor temperature and relative humidity <u>under all expected operating conditions</u>, while providing fresh air ventilation for improved indoor air quality. The prototype I-HVCD system provided impressive control of indoor temperature and RH relative to the base case system monitored in 2006. Unfortunately time and budget constraints, combined with the Florida test house going into foreclosure, precluded successful demonstration of ventilation cooling and fresh air ventilation operation. To thoroughly verify and demonstrate the I-HVCD technology, more work needs to be completed to ensure that the control logic and refrigeration system performance are optimized. This effort represents a necessary step in the commercialization process before a manufacturer would commit to pursuing the technology.

Project successes include the following:

I-HVCD System Mechanical Design and Development

Using the fully variable speed supply fan and controls from the NightBreeze system, DEG developed an integrated cooling, ventilation, and dehumidification system that can operate under an extended range (SHR from 0 to 0.8) and should be compatible with most two-state condensing units on the market.

The I-HVCD prototype system developed in this project includes the following:

- NightBreeze variable speed air handler
- Two-stage condensing unit (with relay for controlling condenser fan)
- Custom evaporator coil assembly that includes evaporator coil, subcooling coil, four refrigerant solenoid valves, thermal expansion valve, and two control input sensors (leaving air temperature and evaporator coil surface temperature)
- Return and outdoor air damper
- Outdoor air temperature sensor
- I-HVCD custom controls

Controls Design and Testing

A control system design was developed by DEG in conjunction with their refrigeration engineer and software developer. Control algorithms were first tested in LabView and then coded for the I-HVCD microprocessor. A WDU with RH sensor and a hardware control board were designed and developed. Firmware and hardware were tested in the lab and then final controls were installed in the Florida test house. Initial monitoring indicates reliable performance, but more testing and evaluation is needed to fully optimize the control.

I-HVCD Laboratory Testing

A prototype I-HVCD system was tested at DEG's shop facility to assess system performance under varying modes of operation. The system was operated in all expected cooling/ dehumidification modes. Table 10 summarizes the SHR and relative moisture removal capability of each of the modes during steady state operation. The moisture removal rate is relative to the 2nd stage cooling/ high airflow case or "conventional air conditioner" performance. The lab test results clearly demonstrate the ability of the system to operate in response to a variety of indoor conditions, and to effectively dehumidify without providing significant cooling to indoors.

Table 10: Lab Test Result Summary

Test Case	SHR	Moisture Removal Rate
2nd stage cooling/ high airflow	0.78	1.0 [34]
1st stage cooling/ low airflow	0.65	1.9
Subcooling (Mode 2)	0.50	3.6
Partial condensing (Mode 3)	0.02	193

[34] The rate of moisture removal for this case was 0.12 liters per kBtu of sensible cooling. Other modes of operation are reported relative to this performance level.

System Field Testing

Field testing was completed at the Gainesville, Florida Building America test site during the 2006 summer (base case operation) and the 2007 summer (both base case and I-HVCD operation). Monitoring results in the unoccupied house indicate that both systems did a good job in maintaining uniform indoor temperatures. RH control was considerably better with the I-HVCD system with typical daily variations in indoor RH of <2% vs. 5-7% for the base case system. Improved humidity control should translate to higher indoor temperature setpoints.

The absence of a return air damper in the Gainesville I-HVCD installation precluded testing of fresh air ventilation and ventilation cooling logic[35] and comparative performance testing relative to base case operation. Preliminary monitoring results suggest that I-HVCD energy use is roughly comparable to the base case "best practice" system. This is impressive given that there are certainly opportunities to use the current field data (and future field monitoring data) to improve system performance.

Commercialization Activities

Preliminary performance and installed cost estimations suggest that the current I-HVCD design can provided equivalent operating efficiency and improved indoor comfort at a lower first cost than a competing conventional system alternative that provides cooling, dehumidification, and fresh air ventilation. We feel confident that additional system modifications and value engineering will improve these results.

Despite the fact that the prototype I-HVCD system demonstrated in Gainesville shows significant promise, additional work needs to be completed before the product has been sufficiently refined to garner industry from major industry partners. Our goal was to reach that point in this project, but it was not achieved. Contacts have been made with several industry leaders, but more work needs to be completed to develop a more compelling performance picture that would generate stronger interest from potential industry partners.

A patent application (number 11/115,188) was submitted for the I-HVCD system on April 27, 2005.

8. Next Steps

The I-HVCD development work completed in this project represents a significant step in moving the technology to the marketplace. Prototype refrigeration hardware and a controls package were developed, tested, and demonstrated in the field. The field results were encouraging although a shortened monitoring period without demonstration of the fresh air ventilation (and night vent cooling) system operation resulted in incomplete demonstration of I-HVCD capabilities. With project successes in mind, the following steps need to be undertaken to bring the product to commercialization:

a. Fabricate additional coil assemblies and control packages and install the components in monitored Building America projects in various humid climates. These monitored installations could collect detailed short time interval data (such as the one minute interval data collected at the Gainesville site) to provide high resolution data to identify and resolve control performance issues.

[35] Since much of this functionality was preexisting in the NightBreeze control, there is no reason to expect significant problems in the control logic.

b. Perform a value engineering review of the I-HVCD system to identify cost reduction opportunities. Reducing system cost is a critical step in increasing system marketability.

c. Control firmware development is a time-consuming and detailed process that entails repeated cycles of field testing and control firmware revision. In this project the prototype I-HVCD control was initially tested by the controls subcontractor prior to being tested for four weeks at DEG's shop facility. The control and refrigeration hardware were then shipped to Florida for installation in the test house. The field testing lasted about two months and provided initial feedback on control performance. More field experience is needed to fully debug controller and I-HVCD system performance. We envision two summers of system monitoring at multiple field sites to fully test, refine, and optimize the I-HVCD control.

d. Develop two-zone control capability. Much of the new construction market is moving to two or more air delivery zones. Adding two-zone capability to I-HVCD would greatly increase system marketability.

e. Work with manufacturers to move the I-HVCD technology to commercialization. The completion of this project and final report represents a major first step in demonstrating the I-HVCD technology. The above mentioned steps are needed to develop a compelling case to HVAC manufacturers that the technology has merit and is marketable. Additional funding is needed to achieve this final pre-commercialization demonstration and verification step.

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Appendix A:

Project Team

Davis Energy Group

Davis Energy Group (DEG), the project lead, is a mechanical engineering firm that has operated in Davis since 1981. DEG's primary focus is on improving the energy and resource efficiency of buildings. In addition to product development and demonstration, DEG's services include HVAC design, building energy systems evaluation, monitoring and verification, building and appliance standards work, and energy and green program administration and support. DEG's staff currently totals 21 employees, including 13 engineering professionals.

The Energy Savers

Jim Phillips, the owner of The Energy Savers, has worked in the refrigeration industry for over 45 years. He has extensive experience designing advanced refrigeration systems and diagnosing problems. Part of his career was spent in the Houston area where he developed innovative approaches to solving indoor humidity problems in commercial buildings. His role in the project was to work with DEG on designing, testing, and modifying the prototype refrigeration system.

Steven Winter Associates (SWA)

Established in 1972, Steven Winter Associates is an internationally recognized research and consulting firm with specialized expertise in technologies and procedures that improve the performance and cost effectiveness of buildings. SWA has offices in Washington, DC, Connecticut, and New York City with a combined staff of over fifty architects, engineers, and scientists. SWA's primary role in the project was to identify a field site, secure monitoring access, and install and maintain monitoring equipment.

RCS/ZTECH

Headquartered in Rancho Cordova, CA, RCS/ZTECH is a leading manufacturer of communicating thermostats, temperature and zone controls, energy management, and home automation products for residential and light commercial markets. RCS/ZTECH was a major contributor in developing the SmartVent and NightBreeze ventilation cooling controls, which have been installed in thousands of homes, primarily in California. RCS/ZTECH's role in the project was to design and develop prototype control hardware and thermostats for the I-HVCD system.

Harlan Strickland

Mr. Strickland is an independent controls programmer with over 30 years engineering experience in firmware design, and digital, analog, and RF hardware design. Mr. Strickland worked closely with DEG to develop, implement, and test the I-HVCD control firmware.

Appendix B:

Site Photos, Monitoring Plans, Floor Plan

Figure B-1: Isometric View of House Front & Side

Figure B-2: Front View of House

Figure B-3: Two-Stage Condensing Unit

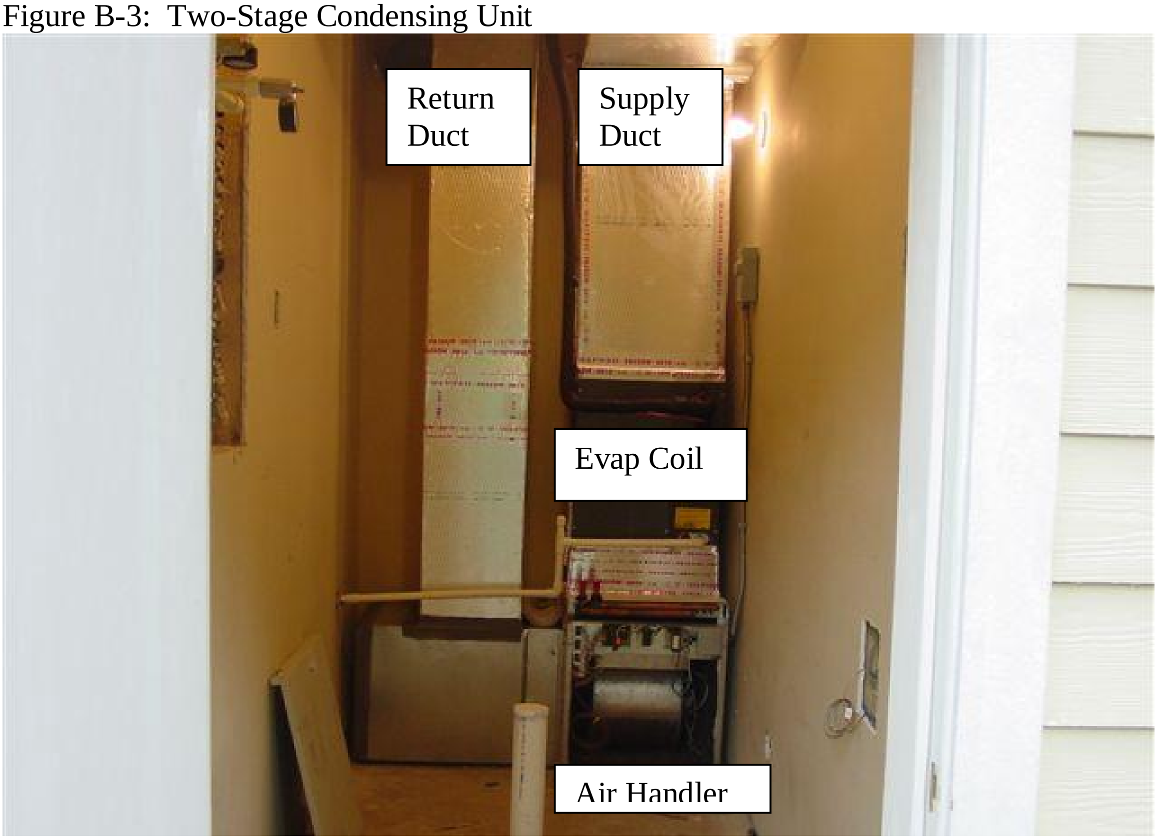


Figure B-4: Mechanical Closet During Construction

1 BACKGROUND & OBJECTIVES

Poor indoor humidity control and inadequate fresh air ventilation are two of the biggest problems in residential buildings located in humid regions of the United States. New homes are being built with better materials (low SHGC glazing, lower internal gain lighting and appliances) and construction practices are improving (tighter building envelopes and low-leakage HVAC duct systems), resulting in decreased sensible cooling loads. Lower sensible loads translate to reduced air conditioning compressor operation and a resulting increase in indoor relative humidity due to reduced moisture removal. In addition, many builders are combating indoor air quality problems by adding systems that provide scheduled delivery of outdoor air. Although these systems improve indoor air quality, they can also introduce significant amounts of moisture to conditioned space.

The following excerpts from three recent publications reinforce the problem of moisture control in new energy-efficient homes.

"Because of the airtight envelope, well-shaded low solar heat gain windows and continuous mechanical ventilation the thermostat would not call for sensible cooling until after the RH rose above what would be considered acceptable in some situations." (Christian, 2005)
Note: Data collected at the Zero Energy test house in Tennessee indicated that for approximately 15% of the year, indoor relative humidity exceeded 60%.

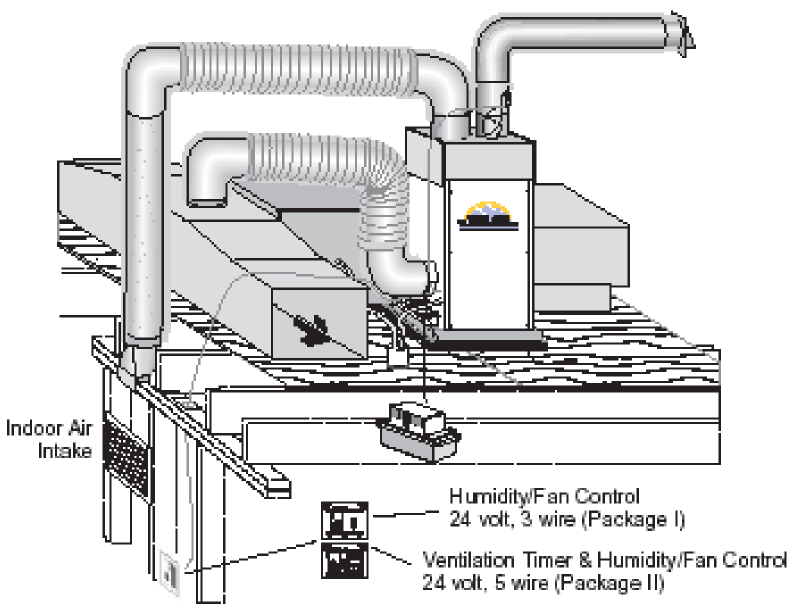
"While both houses were similar in size, total energy consumed for the Energy-efficient Reference house was less than half that of the Standard Reference house. However, because of the reduced sensible heat gain, and the resultant reduction in cooling system operation, humidity control performance in the energy-efficient house was inferior" (Rudd et al, 2005)

"However, it has been noted that some houses built under this program (*Building America*) in the hot and humid climate and equipped with a dedicated ventilation system were reported to have longer periods of elevated interior relative humidity (RH>60%) relative to conventional houses without dedicated ventilation systems (Rudd 2003)" (Moyer et al, 2004)

Conventional air conditioning systems are often not capable of effectively controlling indoor moisture, since the basic ability of the system to provide sensible (temperature reduction) and latent cooling (moisture removal) cannot be varied significantly. To efficiently provide comfort in these humid-climate energy-efficient homes, an integrated design approach capable of responding to varying indoor and outdoor conditions is needed.

Under the U.S. Department of Energy's SBIR program, Davis Energy Group is designing, developing, and demonstrating an advanced integrated residential HVAC system designed to provide temperature and humidity control under a range of indoor and outdoor conditions. In conjunction with Steven Winter Associates (SWA), Davis Energy Group will install an advanced prototype HVAC system at a Building America site in Gainesville, Florida (see floor plan in Appendix A). The prototype Integrated Heating, Ventilation, Cooling, and Dehumidification (I-HVCD) system will be installed and monitored during the 2006 cooling season. In 2005, SWA will monitor best available practice to define a base case performance level. The base case HVAC system will feature a two-speed Amana condensing unit and a Thermastor UltraAire dehumidification and fresh air ventilation system.

The UltraAire system (shown in the figure below) operates independently to the cooling system and is controlled in response to a humidistat. The unit pulls house air from the intake grille, dehumidifies the air, and dumps warmer and dryer air into the supply duct system. In addition, the unit can be configured to provide scheduled fresh air ventilation. The UltraAire system at the Madera site will be configured to provide 75 ft^3 of outside air on an hourly basis.



The objectives of the 2005 monitoring are not only to develop a performance benchmark to which the I-HVCD unit will be compared, but also to better understand how the balance between latent and sensible cooling loads changes under different climate and occupancy patterns.

2 MONITORING STRATEGY

2.1 Key Monitoring Parameters

The principal goal of this comparative monitoring strategy is to document the energy consumed and indoor conditions of the Year 1 and 2 space conditioning strategies. The monitoring approach must be detailed enough to generate a year-to-year comparison of:

- cooling energy delivered (latent, sensible, and total)
- cooling energy consumed (air conditioning and dehumidification)
- overall efficiency (effective EER)
- indoor conditions (temperature and RH in each operating mode)
- outdoor conditions (normalize results for weather differences)

Key parameters required for evaluating impacts and verifying operation include:

- First and second floor indoor air temperature and RH
- Outdoor temperature and RH
- Condensing unit and NightBreeze air handler electrical energy use
- UltraAire energy use (Year 1)
- I-HVCD indoor unit energy use (Year 2)
- Evaporator and UltraAire condensate water flow

Sensible cooling capacity will be calculated on 15-second intervals as shown in Equation 1.

Equation 1: $Q_{clg,sens} = 1.08 \times (T_{sup} - T_{ret}) \times CFM$

Where: Tsup = supply air temperature
Tret = return air temperature
CFM = airflow determined by compressor stage

Latent cooling capacity will be calculated based on the volume of condensate collected[36]. Since condensate flow does not necessarily begin coincident with sensible cooling, total cooling (latent plus sensible) and overall EER[37] will be calculated on a daily basis. Enthalpy calculations, as shown in Appendix B, can be used to calculate total cooling capacity based on system airflow and supply/return air temperature and relative humidity. One shortcoming of enthalpy calculations is that it does not accurately account for reevaporation of condensate during the system off-cycle.

[36] The latent heat of condensation is equal to 1080.8 Btu/lb at a temperature of 45°F.
[37] Overall EER equals total cooling in Btu's divided by total energy consumed in Watt-hours.

2.2 Datalogger and Sensor Specifications

A Campbell Scientific CR10X datalogger with multiplexer will be used to sample and log monitoring data. Table 1 lists the types of sensors to be used for the various monitoring points and their performance specifications. Sensor selection was based on functionality, accuracy, cost, reliability, and durability. Specific model numbers are listed as examples; similar models by other manufacturers may be substituted. Appendix C contains specifications on the proposed monitoring equipment.

Table 1: Sensor Specifications

Abbrev.	*Description*	*Location*	*Sensor*	*Part Number*
ATRH1	Ambient (OA intake) air temperature / RH	outdoor	Humirel 2500	HTM2500-ND
			+ radiation shield	41303
ATRH2	Indoor air temperature / RH	1st floor	Vaisala temp/RH	HMW40Y
ATRH3	Indoor air temperature / RH	2nd floor	Vaisala temp/RH	HMW40Y
ATRH4	Return air plenum temperature / RH	return plenum	Vaisala temp/RH	HMD40Y
ATRH5	Supply air plenum temperature / RH	supply plenum	Vaisala temp/RH	HMD60Y
ATRH6	Supply air plenum temperature / RH	supply plenum	Vaisala temp/RH	HMD60Y
ATRH7	Dehumidifier outlet air temperature / RH	mechanical room	Vaisala temp/RH	HMD40Y
P1	Compressor power	electrical panel	CCS Wattnode	WNA-1P-240-P
			CCS current transformer	CTT-0300-030
			CCS current transformer	CTT-0300-030
P2	Air handler unit power	electrical panel	CCS Wattnode	WNA-1P-240-P
			CCS current transformer	CTT-0300-005
P3	Dehumidifier power	electrical panel	CCS Wattnode	WNA-1P-240-P
			CCS current transformer	CTT-0300-015
FR1	Return plenum flow rate	return plenum	AMC Fan-e	custom order
			Setra pressure transducer	2641-0R1WD-11-T1-C
C1	Condensate (rain gauge)	air conditioner	Texas Electronics	TR-525USW-R3
C2	Condensate (rain gauge)	dehumidifier	Texas Electronics	TR-525USW-R3

All sensors will be scanned every 15 seconds, and data will be summed or averaged (as appropriate) and stored in datalogger memory every 15 minutes. Although datalogger memory will be sufficient to store approximately one week of data, the loggers will be downloaded daily using telephone modems. Dataloggers powered by low voltage power supplies with battery backup will protect against data loss during power outages.
Data from the dataloggers will be downloaded to a central computer and screened using automated software to verify that the collected data is within range. Out-of-range data will be reported and investigated to determine whether a sensor or monitoring error exists or equipment has failed.

2.3 Monitoring Period

The goal is to collect sufficient data in each mode of operation (base case and I-HVCD). The project monitoring will conclude at the end of the cooling season in 2006. The earliest we anticipate the start of Year 1 monitoring is August 2005. If the house is initially unoccupied during Year 1, we will operate the cooling and dehumidification systems to gather preliminary data.

2.4 Equipment Panel

The CR10X datalogger with battery backup, terminal strips, and electrical power strips will be mounted in locking metal enclosures located in the mechanical room. The enclosures will also contain pre-wired terminals for connection to powered sensors and for 4-20 mA signal inputs.

2.5 Wiring

Wiring shall be Belden 22 gauge shielded communications cable or equal, #8761 single pair, #8771 3-conductor, and #8723 two pair. Buildings will be pre-wired prior to drywall so that wiring is concealed as much as possible. Recessed boxes will be used to terminate wiring near the equipment panels.

3. MONITORING SYSTEM COMMISSIONING AND CALIBRATION

A commissioning log will be completed to record sensor calibrations, one-time measurements, and other data. On completion of equipment installation, a laptop computer will be connected to the datalogger for reading real time data, and the following calibrations and verifications will be completed:

Air Temperature. Using calibrated temperature sensor, record monitored and calibrated temperatures for each sensor.

Relative Humidity. Using calibrated relative humidity sensor (± 2%), record monitored and calibrated RH readings for each sensor.

Power. Activate power-monitored component and verify power measurement. Reverse polarity of CT, voltage input, and datalogger connections as needed to correct for lack of readings. Record reading and compare to handheld power meter.

HVAC Airflow. An Air Monitor Corporation multi-point, self-averaging pitot traverse station will be used to measure HVAC supply airflow. These airflow levels will be used in cooling (and heating) Btu calculations.

Condensate Flow. Compare tilting bucket rain gauge condensate flow to graduate cylinder measurement.

Communications. Dial the datalogger modem from a remote computer and verify communications to both dataloggers.

Permanent Programming. Enter offsets and other program variables determined during commissioning into the onsite datalogger program. After one day of operation, download and verify all readings. Adjust program, if necessary.

4. HVAC EQUIPMENT SETUP

The Amana two-stage cooling system will be set up to operate in typical fashion. First and second stage supply airflow levels will be determined in cooperation with the installing HVAC contractor. We will suggest an indoor thermostat comfort setting of 75°F to the occupants of the home. The UltraAire system will be configured to provide ventilation air consistent with the requirements of the ASHRAE 62.2 residential ventilation standard. The UltraAire unit will be initially set to maintain an indoor humidity level of 50%. The temperature and humidity settings are representative of typical desired indoor comfort conditions. If the occupants are not comfortable with the initial temperature and humidity settings, we will work with them to determine appropriate settings. For the remainder of the monitoring, the goal will be to keep these settings fixed.

During year 2 monitoring, the same indoor comfort targets will be used. The NightBreeze system will provide outdoor ventilation air to meet ASHRAE 62.2 standards.

5. DATA ACQUISITION

Data will be downloaded to Steven Winter Associates offices on a daily basis. Software will be developed to read in the "raw" data and verify that all readings are within expected values (e.g. indoor air temperature is between 40 and 90°F). An automated screening program scans the data and reports measurements that are out of range. If data are out of range, the suspect data will be visually examined to determine whether a sensor is defective. If the review indicates sensor error a service call will be scheduled to repair or replace sensors. On a weekly basis, data will be graphed in time-series format to further insure the data are physically consistent. For example, if the system is operating in heating mode, supply air temperature and pump energy consumption should both reflect heating operation.

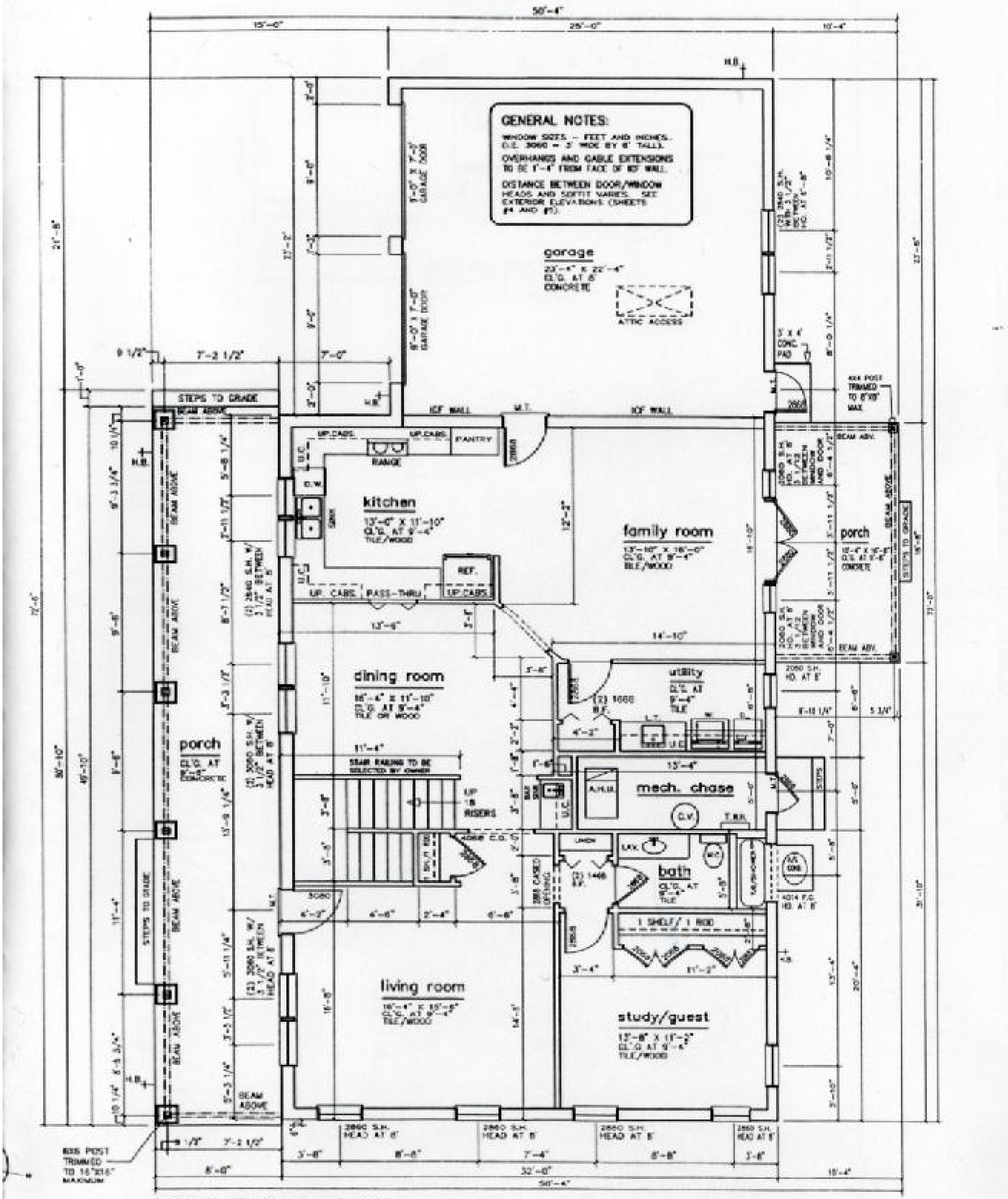
6. ANALYSIS AND REPORTING

Interim and final field monitoring reports will be provided to the project manager. The interim report will document Year 1 operation. The interim report will be valuable in demonstrating the operating efficiency of the base case system, the amount of supplemental dehumidification needed, and the indoor conditions. Homeowner comfort assessments will be obtained as part of the interim report. Operational issues will also be documented.

The final field monitoring report will document the overall field monitoring effort including I-HVCD installation issues, system reliability and maintenance issues, performance data, comparative efficiency and energy use relative to the base case, and homeowner and contractor assessments.

7. DECOMMISSIONING

Upon the conclusion of monitoring, all equipment will be removed. Wiring will be left in place. Wiring at indoor sensors will be clipped and any in drywall holes will be patched and painted.



FIRST FLOOR PLAN

SCALE: 1/4" = 1'-0"

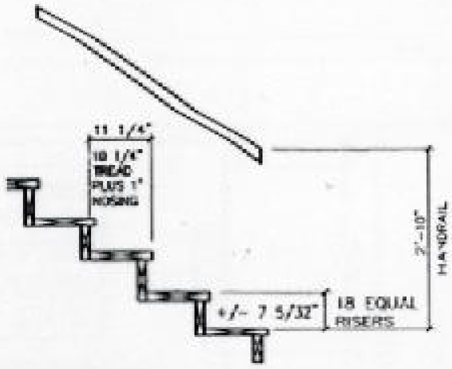
LISA W. WOLFE
DESIGN AND DRAFTING
MAYO, FLORIDA

UNIVERSITY OF FLORIDA
INSTITUTE OF FOOD AND AGRICULTURAL SERVICES
FLORIDA ENERGY EXTENSION SERVICE
PIERCE JONES, DIRECTOR
P.O. BOX 110940 GAINESVILLE, FL 32611-0940

MADERA
MODEL #3038

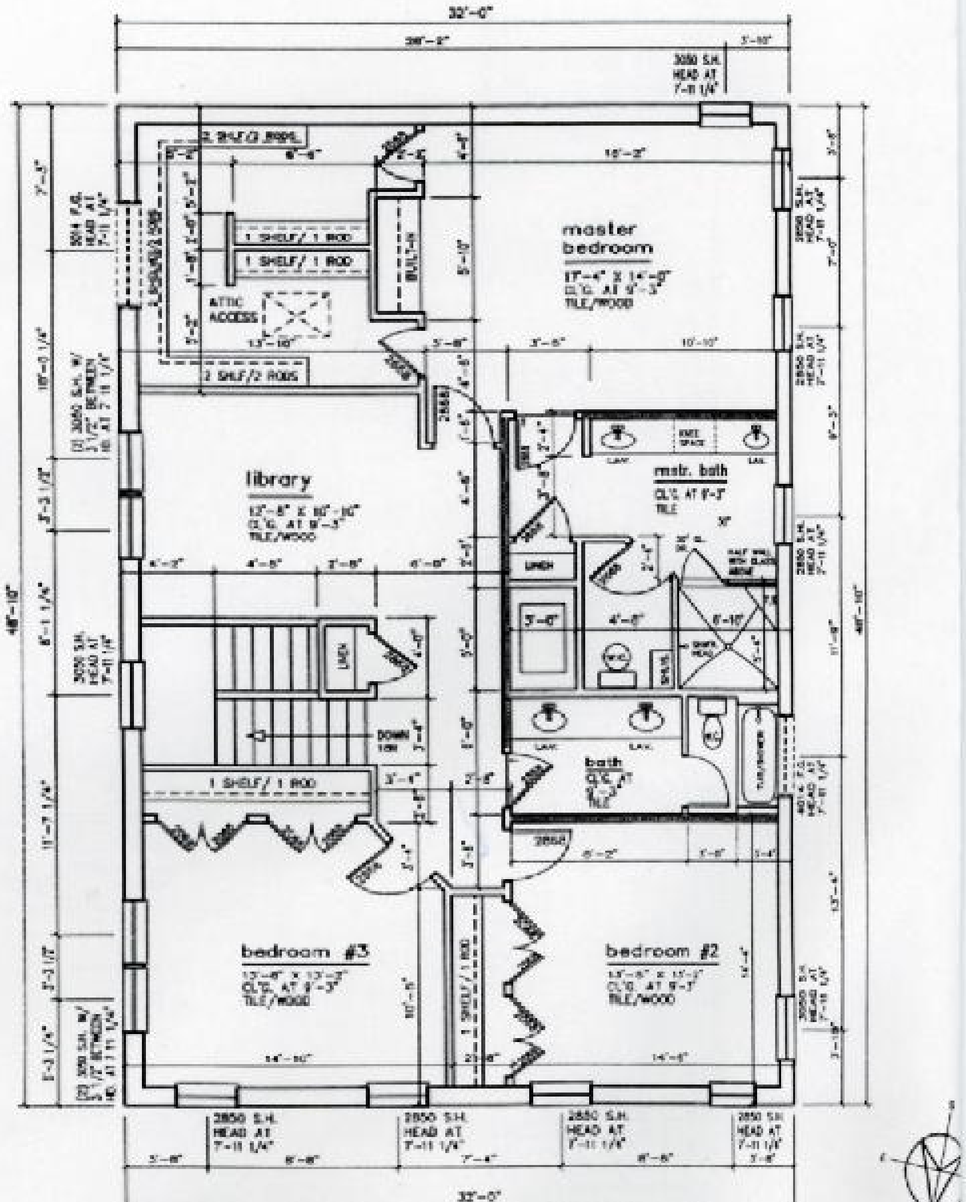
FLOOR PLANS
SCALE: VARIES
JOB #0420

3



STAIR DETAIL

SCALE: 3/4" = 1'-0"



SECOND FLOOR PLAN

SCALE: 1/4" = 1'-0"

Appendix C:

Product Literature

Heading & Air Conditioning

Amana

LASTS AND LASTS AND LASTS.™

Specification Sheet

16 SEER

Cooling Capacity:
34,000 - 56,000 BTUH

LIMITED WARRANTY

- Lifetime Limited Replacement of Entire Unit if Compressor Fails for Original Owner
- 10-Year—Parts

Asure[SM]
EXTENDED SERVICE PLAN

ENERGY STAR

RSG Ultron®

Split System Air Conditioner

The Amana® brand RSG Ultron® Air Conditioner uses the environmentally friendly refrigerant R-410A and features energy efficiencies and operating levels that are among the best in the heating and cooling industry. R-410A is chlorine-free to help prevent damage to the ozone layer. The RSG features a new technology—the 2-stage, high-efficiency Copeland scroll compressor—that better matches the cooling requirements of the conditioned space, providing improved temperature and humidity control.

High-efficiency Copeland Ultra-tech Scroll Compressor with Factory-installed Sound Blanket. The 2-stage, high-efficiency Copeland scroll compressor provides consistent energy conservation. This energy-saving design reduces internal resistance and friction and increases operating efficiency. Internal parts are spring isolated, and the entire compressor is mounted on rubber cushions to help absorb vibrations. A high-density foam compressor sound blanket reduces noise. Tubing access to the compressor is located on the top or sides to provide plenty of internal room for installation and removal of parts.

Economy and Performance. The balance of economy and efficiency makes RSG units ideal for replacement or new construction. The RSG Ultron series delivers a 16 SEER to help reduce your energy bills. Refrigerant line connections and service valves are easy to reach. A 2-speed fan motor and heavy-gauge, coated wire condenser fan grilles protect the fan, motor and coil, as well as result in low fan operating sound. The refrigerant (R-410A) system is protected by manual-reset high- and low-pressure switches. Embossments in the bottom allow drainage and air flow under the unit to reduce corrosion. Another standard feature is the 5-minute anti-short cycle protection.

Efficient Cubed Coil. This space-saving design provides more active square feet of cooling surface for increased cooling efficiency. The compact cubed coil forms the body of the unit. The unit's compact footprint allows for versatile applications.

Copper and Aluminum Coils. Our condenser and evaporator coils are made from seamless copper tubing and enhanced aluminum fins. Tubing life is extended and fewer leaks experienced because we use only refrigeration-grade copper tubing.

Factory-installed Liquid Line Filter Drier. Factory-installed filter drier keeps moisture and dirt out of the system for a long and reliable system life. This standard protection adds reliability by helping keep the R-410A refrigerant clean and dry, ensuring preservation of the compressor and expansion devices.

Back-seating Service Valves and Gauge Ports. To speed installation and service, controls and service ports are accessible during operation and can be serviced without interruption. Fully front- and back-seating service valves with sweat connections and gauge ports provide years of reliable, leak-free use. Valves and ports angled at 45° allow enough clearance to service the unit quickly and easily. The service panel swings open at the corner for effective service from two directions. Inside, a pre-wired control panel with single-panel access speeds installation. Minimal quantity of numbered and color-coded wires to assure fast field wiring.

Cabinet and Construction. Durable, long-lasting painted steel cabinet maintains its attractive finish for many years. Cabinet and screws resist rust and fading due to ultraviolet rays. Tightly spaced, heavy-gauge, coated wire grille protects the coil from inadvertent damage, vandalism and inclement weather, while allowing excellent access for cleaning. When properly anchored, meets the 2001 Florida Building Code unit integrity requirements for hurricane-type winds.

THE ULTRA-AIRE APD
AIR PURIFYING DEHUMIDIFIER

Ultra-Aire APD UA-100V
Fresh Air Ventilation, Filtration and High Capacity Dehumidification

Fresh Air Ventilation
Outdoor air is ducted to the unit via 6” round duct. This provides necessary air changes to dilute pollutants and maintain high oxygen content in the air. The amount of fresh air ventilation can be regulated by a variety of controls.

Air Filtration
A measured amount of outdoor air is combined with indoor air and passes through the filter chamber. A 30 percent efficient (ASHRAE Dust Spot Test) media filter is standard. An optional 95 percent efficient (ASHRAE Dust Spot Test) pleated media filter is available. If the optional filter is selected, the standard filter operates as a pre-filter.

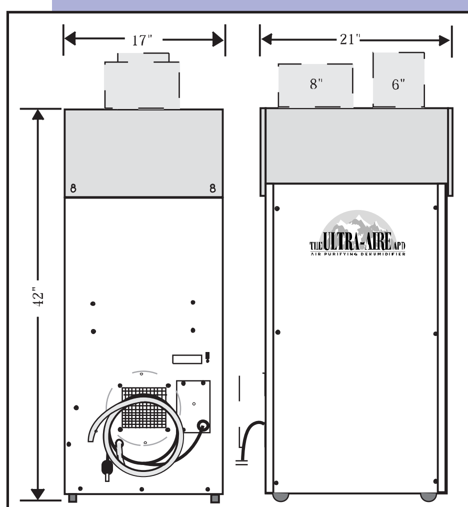
Dehumidification
High efficiency dehumidifier that utilizes refrigeration to cool the incoming air stream below its dew point as it passes through the dehumidification (evaporator) coil. This cooling results in the the removal of moisture (latent heat) and reduction in temperature (sensible heat). The cooled and dried air is used to pre-cool the incoming air stream resulting in up to a 200 percent increase in overall efficiency. After the pre-cooling stage the processed air is reheated by passing through the condenser coil. The latent heat removed by the evaporator coil is returned to the air stream as sensible heat, which results in an overall temperature increase from the incoming air.

The unit is controlled by a 24 volt remote control panel. The standard control contains a fan/filter switch that allows continuous air circulation independent of dehumidification. The panel also contains a dehumidistat with settings from 20 to 80 percent and positive “on” and “off” settings. The optional Ultra-Aire control also contains a timer that activates a motorized damper to provide programmed fresh air ventilation periods.

Additional Features

- The unit is provided with 6 inch (fresh air) and 8 inch (indoor air) inlet collars. The processed air exits the unit via an 8 inch round duct.
- The condensed moisture is gravity drained from the unit by a 6 foot vinyl hose.
- The unit is powered through a factory installed ten foot power cord; 115 volt with ground.

The Ultra-Aire UA-100V eliminates mold, mildew and dust mites.



DIMENSIONS:		SHIPPING SPECS
Width:	21 inches	30 inches
Height:	42 inches	47 inches
Depth:	17 inches	25 inches
Weight:	119 lbs.	134 lbs.

Ultra-Aire APD UA-100V Capacities and Performance*:

Blower:	220 CFM @ .1 IWG
Supply Voltage:	115 volt - 1 Phase - 60 Hz.
Amps:	6.8
Capacity:	100 Pints/Day (80°F, 60% RH)
Filter Efficiency:	30% std., 95% opt., (ASHRAE Dust Spot)

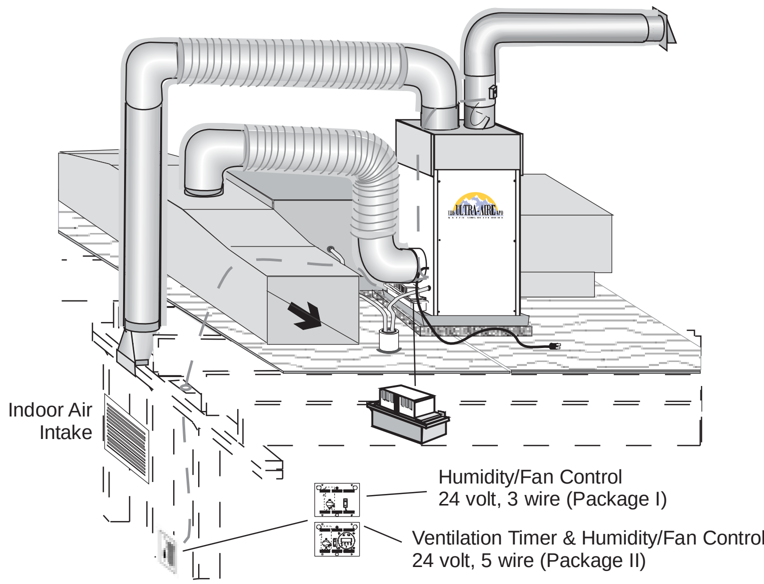
Therma-Stor
1-800-533-7533 WWW.THERMASTOR.COM

** Specifications subject to change without notice*

Installation Recommendations and Options

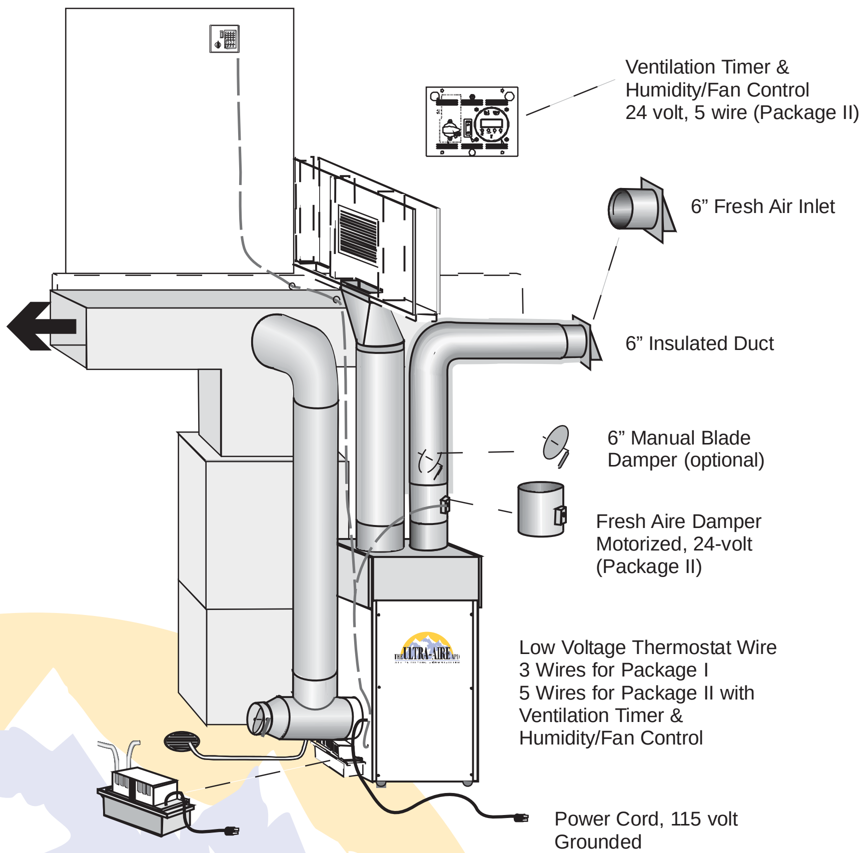
Ultra-Aire UA-100V Attic Installation:

1. Indoor air return should come from an open area of the first (preferred) or second floor.
2. The Ultra-Aire supply should be ducted into the forced air system past the air conditioning coil. The duct connection should be perpendicular to the air flow.
3. The Ultra-Aire should be installed in a drip pan placed on a vibration absorbing foam pad.
4. The six inch fresh air intake should be located at least 6 feet away from any exhaust ports.
5. In high humidity climates, an optional insulation kit is available to prevent condensation from occurring on the Ultra-Aire cabinet.



Ultra-Aire UA-100V Basement or Crawl Space Installation

1. Indoor air return should come from an open area of the first or second (preferred) floor.
2. The Ultra-Aire supply should be ducted into the forced air system supply above the “A” or air conditioning coil. The duct connection should be perpendicular to the air flow.
3. An eight inch tee fitting with an adjustable blade damper on the straight run should be attached at the Ultra-Aire supply duct. This allows for increased air flow to the basement during the summer months.
4. The 6 inch fresh air intake should be located at least 6 feet away from any exhaust ports, such as, dryer, range hood, or combustion device exhaust.



Therma-Stor
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Madison, Wisconsin 53708-8050
WWW.THERMASTOR.COM

THE ULTRA-AIRE APD
AIR PURIFYING DEHUMIDIFIER

Call for More Information
1-800-533-7533
ph. 608-222-5301 fx. 608-222-1447

TL-119 10/03

Appendix D:

I-HVCD 2007 Data Summary

Date	Ambient Temp	Ambient RH	1st Floor Temp	1st Floor RH	1st Floor RH Max	2nd Floor Temp	2nd Floor RH	2nd Floor Max RH	Avg Air handler [cfm]	Cond Unit (kWh)	Air Handler (kWh)	Condensate (pints)	Ttl Daily Energy (kWh)
13-Sep	85.3	76.4	77.7	42.2	48.4	78.2	44.3	48.3	806	28.2	4.9	34.4	33.2
14-Sep	81.4	86.3	76.2	41.5	42.2	76.2	44.5	46.2	397	22.5	2.9	23.8	25.4
15-Sep	83.0	85.0	75.8	41.9	42.9	76.2	45.6	48.2	404	23.6	3.1	22.7	26.7
16-Sep	82.9	83.4	75.7	42.5	43.2	76.2	46.2	47.6	356	22.4	2.6	22.0	25.0
17-Sep	77.3	91.2	76.0	43.1	45.6	76.2	46.1	47.9	152	10.2	1.0	14.5	11.2
18-Sep	76.5	85.7	76.1	44.5	46.8	76.1	47.2	49.5	162	11.5	1.1	15.6	12.7
19-Sep	73.3	90.9	76.1	45.1	46.2	75.9	48.2	49.2	66	6.1	0.6	10.2	6.7
20-Sep	78.2	92.7	75.8	44.2	46.4	75.8	47.6	49.0	141	12.8	1.0	19.7	13.8
21-Sep	80.1	89.5	75.5	43.9	46.0	76.0	47.2	49.1	214	17.0	1.5	24.2	18.6
22-Sep	79.6	93.3	75.4	43.6	46.5	76.0	47.1	48.4	202	16.2	1.4	22.6	17.6
23-Sep	79.2	96.7	75.4	43.5	45.1	76.1	47.3	49.0	162	13.0	1.2	17.3	14.1
24-Sep	80.8	87.6	75.5	43.9	48.8	76.0	47.1	48.8	192	16.1	1.4	22.0	17.5
25-Sep	79.4	88.9	75.5	43.7	45.1	76.1	46.8	48.3	170	13.8	1.1	18.9	14.9
26-Sep	79.3	91.5	75.4	43.5	45.6	76.0	47.2	48.8	182	14.2	1.2	18.2	15.4
27-Sep	79.7	89.3	75.3	43.2	45.6	76.0	47.2	48.7	183	14.8	1.3	20.2	16.1
28-Sep	80.6	86.0	75.1	43.3	45.7	76.0	47.3	48.9	223	17.4	1.6	21.7	19.0
29-Sep	79.9	81.2	75.4	43.6	49.4	76.0	46.6	48.5	164	13.6	1.1	16.6	14.7
30-Sep	74.9	83.3	75.6	44.6	46.1	75.9	47.1	48.5	96	8.7	0.7	12.0	9.4
1-Oct	75.7	90.3	75.5	44.8	46.0	75.6	47.6	48.5	97	10.2	0.7	16.7	11.0
2-Oct	75.7	99.3	75.3	44.7	46.2	75.3	48.0	49.0	78	9.3	0.7	17.3	10.0
10-Oct	78.9	87.4	73.7	46.4	48.3	73.3	51.5	53.8	345	19.8	2.6	25.9	22.4
11-Oct	76.5	76.6	73.9	46.6	48.3	73.5	51.0	52.7	203	11.8	1.5	14.9	13.3
12-Oct	70.7	67.9	73.6	46.9	48.8	72.8	51.0	52.7	316	6.0	1.5	1.3	7.5
13-Oct	73.8	80.2	73.7	48.3	49.6	73.1	52.3	54.0	207	7.5	1.1	5.6	8.6
14-Oct	73.2	83.1	73.6	47.4	50.6	72.9	51.3	53.6	239	11.3	1.3	10.0	12.6
15-Oct	75.4	81.1	73.9	48.4	50.3	73.5	52.2	54.2	149	8.1	0.9	8.5	9.0
16-Oct	77.4	86.0	73.8	47.0	48.7	73.7	50.8	52.8	180	12.5	1.2	16.5	13.7
17-Oct	77.2	88.2	73.6	47.8	50.3	73.5	51.8	53.2	170	11.8	1.2	13.8	13.0