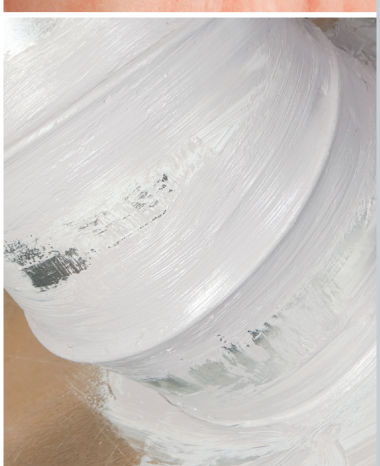
U.S. DEPARTMENT OF
ENERGY

Office of
ENERGY EFFICIENCY & RENEWABLE ENERGY



Evaluating Moisture Control of Variable-Capacity Heat Pumps in Mechanically Ventilated, Low-Load Homes in Climate Zone 2A

February 2018

Building America
U.S. Department of Energy

Available electronically at Office of Scientific and Technical Information website (www.osti.gov)

Available for a processing fee to U.S. Department of Energy and its contractors, in paper, from:

U.S. Department of Energy
Office of Scientific and Technical Information
P.O. Box 62
Oak Ridge, TN 37831-0062

OSTI www.osti.gov
Phone: 865.576.8401
Fax: 865.576.5728
Email: reports@osti.gov

Available for sale to the public, in paper, from:

U.S. Department of Commerce
National Technical Information Service
5301 Shawnee Road
Alexandria, VA 22312

NTIS www.ntis.gov
Phone: 800.553.6847 or 703.605.6000
Fax: 703.605.6900
Email: orders@ntis.gov

Evaluating Moisture Control of Variable-Capacity Heat Pumps in Mechanically Ventilated, Low-Load Homes in Climate Zone 2A

Prepared for:
U.S. Department of Energy Building America Program
Office of Energy Efficiency and Renewable Energy

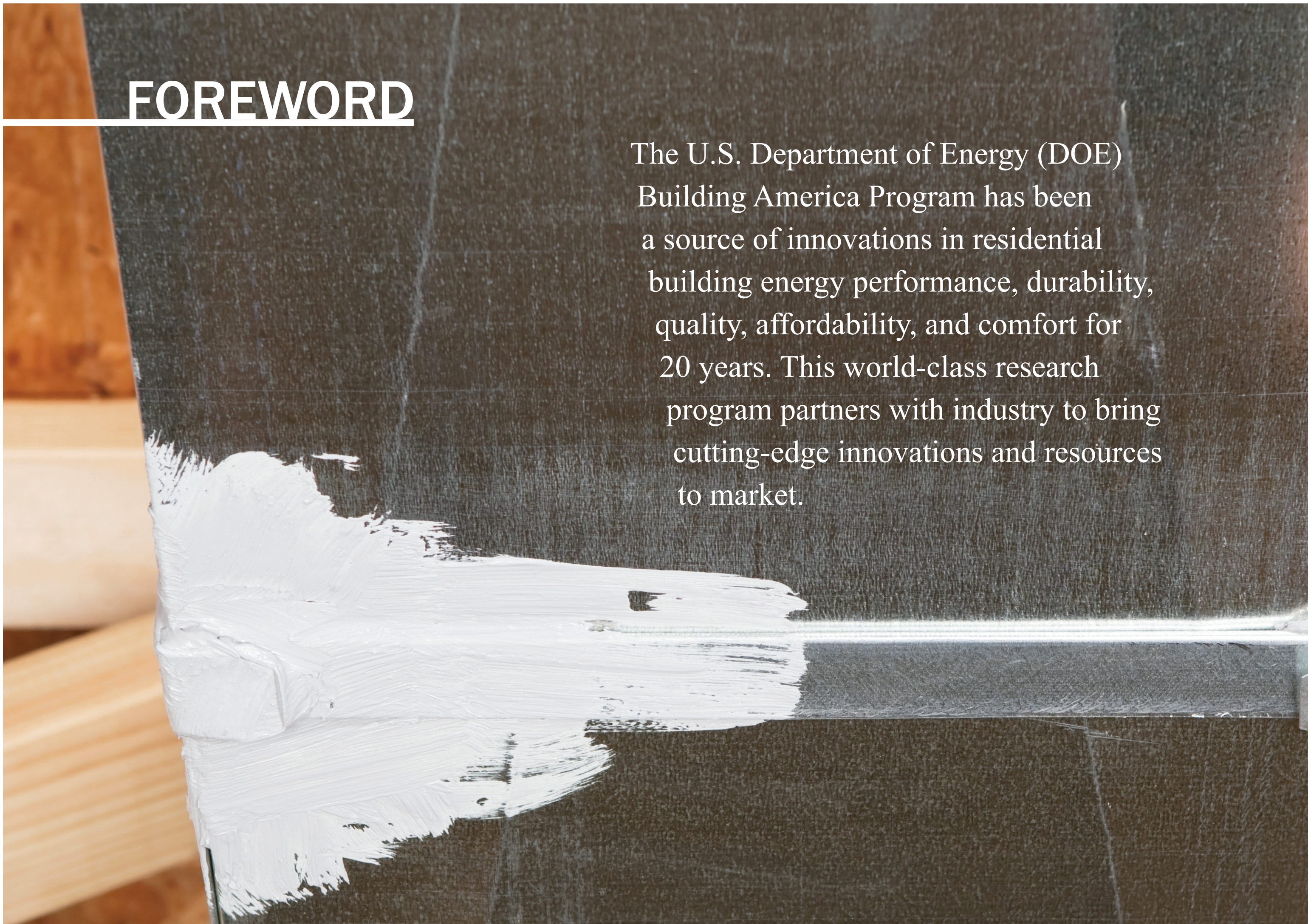
Prepared by:
Eric Martin, Chuck Withers, Janet McIlvaine, Dave Chasar, and David Beal
Florida Solar Energy Center
1679 Clearlake Rd.
Cocoa, FL 32922

February 2018

Suggested Citation
Martin, E.; Withers, C.; McIlvaine, J.; Chasar, D.; Beal, D. 2018. *Evaluating Moisture Control of Variable Capacity Heat Pumps in Mechanically Ventilated, Low-Load Homes in Climate Zone 2A*. Cocoa, FL; Florida Solar Energy Center (FSEC). DOE/EE-1702. https://www.osti.gov/biblio/1421385-evaluating-moisture-control-variable-capacity-heat-pumps-mechanically-ventilated-low-load-homes-climate-zone

This material is based upon work supported by the Department of Energy's Office of Energy Efficiency and Renewable Energy (EERE) under the Building Technologies Office under Award Number(s) EE0007056.

FOREWORD



The U.S. Department of Energy (DOE) Building America Program has been a source of innovations in residential building energy performance, durability, quality, affordability, and comfort for 20 years. This world-class research program partners with industry to bring cutting-edge innovations and resources to market.

The Building America Program supports the DOE Building Technologies Office Residential Building Integration Program goals to:

1. By 2020, develop and demonstrate cost-effective technologies and practices that can reduce the energy use intensity (EUI) of new single-family homes by 60% and existing single-family homes by 40%, relative to the 2010 average home EUI in each climate zone, with a focus on reducing heating, cooling, and water heating loads.
2. By 2025, reduce the energy used for space conditioning and water heating in single-family homes by 40% from 2010 levels.

In cooperation with the Building America Program, the Building America Partnership for Improved Residential Construction is one of many Building America teams working to drive innovations that address the challenges identified in the Program's Research-to-Market Plan.

This report, "Evaluating Moisture Control of Variable-Capacity Heat Pumps in Mechanically Ventilated, Low-Load Homes in Climate Zone 2A," evaluates the performance of variable-capacity comfort systems, with a focus on inverter-driven, variable-capacity systems, as well as proposed system enhancements.

As the technical monitor of the Building America research, the National Renewable Energy Laboratory encourages feedback and dialogue on the research findings in this report as well as others. Send any comments and questions to building.america@ee.doe.gov.

ACKNOWLEDGMENTS

The authors gratefully acknowledge the support and encouragement of U.S. Department of Energy (DOE) Program Managers Eric Werling, Sam Rashkin, and David Lee as well as National Renewable Energy Laboratory Technical Monitor Stacey Rothgeb. This support does not constitute DOE endorsement of the views expressed in this report.

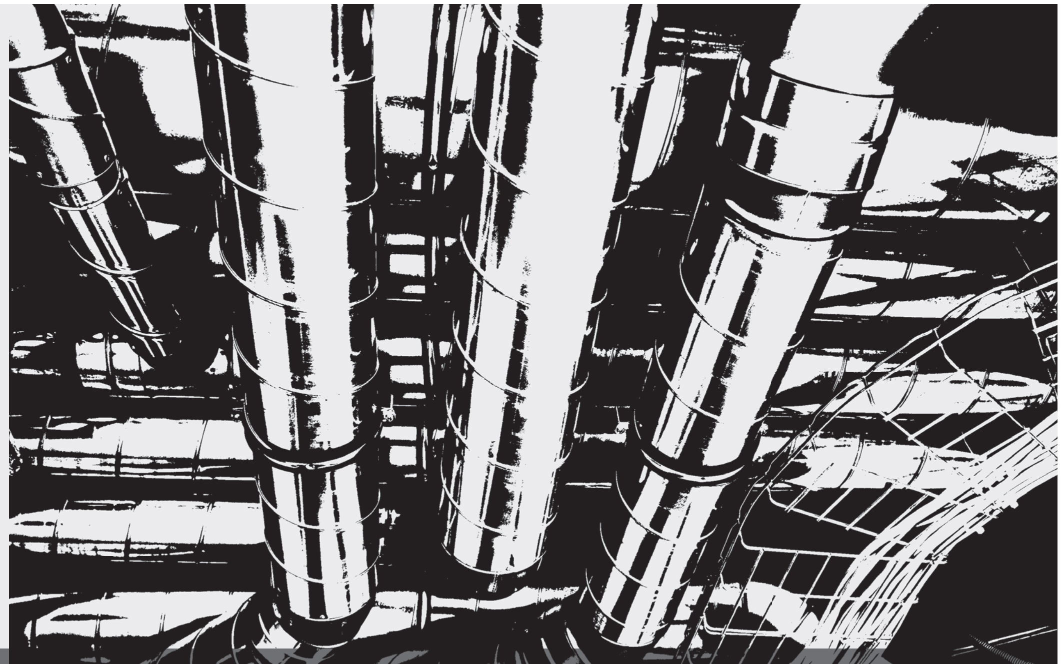
We thank the Unico team for their in-kind cost share, encouragement, and assistance in setting up and testing the iSeries small-duct high-velocity system. We are grateful to Shawn Intagliata for his industry support and guidance. And we extend special thanks to the Unico engineering team for their support, including Craig Messmer, Robert Berry, Jerry Henrich, and Marco Di Giulio.

We are also deeply grateful for the opportunity to conduct the collaborative, applied field research with team members from the affordable housing providers Southeast Volusia County Habitat for Humanity (SEVHFH) in Edgewater, Florida, and South Sarasota Habitat for Humanity (SSHFH) in Venice, Florida:

- SEVHFH Executive Director Rosemary Walker, Construction Director Ray Allnutt, mechanical contractor Servair Heating and Air Conditioning, and owners of the two homes
- SSHFH Executive Director Dee Danmeyer, Construction Director Michael Sollitto, board member Dennis Stroer, mechanical contractor Envirotec Air Inc., and owners of the three homes.

The leadership of both organizations sets a high standard for responsible stewardship in affordable housing construction.

The authors thank heating, ventilating, and air conditioning system manufacturers Panasonic, Mitsubishi, and AirCycler who provided input on product and operating characteristics. Panasonic discounted the units purchased by SEVHFH and Mitsubishi contributed an in-kind cost share toward the research.



LIST OF ACRONYMS

ACCA	Air Conditioning Contractors Association
AHRI	Air-Conditioning, Heating, and Refrigeration Institute
AHU	air handling unit
cfm	cubic feet per minute
DM	dry mode
DOE	U.S. Department of Energy
DR	dining room
dT	difference in temperature
ECM	electronically commutated motor
ERV	energy-recovery ventilator
FS	femtosiemens
FSEC	Florida Solar Energy Center
HVAC	heating, ventilating, and air conditioning
kWh	kilowatt-hour
LR	living room
MHL	manufactured housing laboratory
Pa	Pascal
RH	relative humidity
SAT	supply air temperature
SDHV	small-duct high-velocity
SEER	Seasonal Energy Efficiency Ratio
SEV (1,2)	Southeast Volusia (Unit 1, Unit 2)
SEVHFH	Southeast Volusia Habitat for Humanity
SHR	sensible heat ratio
SS (1,2,3)	South Sarasota (house 1, house 2, house 3)
SSHFH	South Sarasota Habitat for Humanity
WC	inches water column

EXECUTIVE SUMMARY

The well-sealed, highly insulated building enclosures constructed by today's home building industry coupled with efficient lighting and appliances are achieving significantly reduced heating and cooling loads.

These low-load homes can present a challenge when selecting appropriate space-conditioning equipment. Conventional, fixed-capacity heating and cooling equipment is often oversized for small homes, causing increased first costs and operating costs. Even if fixed-capacity equipment can be properly specified for peak loads, it remains oversized for use during much of the year. During these part-load cooling hours, oversized equipment meets the target dry-bulb temperatures very quickly, often without sufficient opportunity for moisture control. The problem becomes more acute for high-performance houses in humid climates when meeting ASHRAE Standard 62.2 recommendations for whole-house mechanical ventilation. This additional latent load coupled with the diminished sensible load of a high-performance thermal envelope makes the moisture-removal capacity of space-conditioning equipment more critical than ever.

One potential solution, beyond the use of supplemental dehumidification, is variable-capacity comfort systems that can adjust capacity in response to varying load. Although such systems primarily operate in response to sensible load, questions remain about whether they can also manage corresponding latent loads. This project evaluates the performance of three emerging strategies using variable-capacity systems to maintain whole-house comfort in low-load, mechanically vented homes:

- A centrally ducted, small-duct high-velocity (SDHV), variable-capacity heat pump
- A centrally ducted, minisplit heat pump with cassette air handling unit
- A ductless multisplit system using transfer fans to control temperatures in bedrooms

The goal is to inform the marketplace about the benefits and limitations of currently available systems and develop recommendations for manufacturers to improve the latent performance of certain equipment by modifying hardware design and control algorithms.

The project consisted of laboratory and field studies. The lab study was performed with a Unico SDHV variable-capacity heat pump in a lab home mechanically ventilated to ASHRAE 62.2-2016. The 1,600-ft2 double-wide manufactured home lab was provided with internal latent and sensible loads to simulate occupancy. The SDHV variable-capacity system had good thermal distribution based on maintaining temperature differences among the bedrooms and living room to within the Air Conditioning Contractors Association (ACCA) Manual RS guideline of 3°F. Results also showed that the SDHV system controlled indoor humidity with warm outdoor conditions in standard cooling mode, with only an incidental need for supplemental dehumidification to maintain indoor relative humidity (RH) less than 60%. The dry mode provided better latent control during low-load periods, but the benefit was limited by overcooling the space. We worked with the manufacturer to optimize their standard operating algorithms in dry mode, targeting a lower sensible heat ratio (SHR) at the lowest cooling capacities to increase moisture removal and limit the amount of sensible overcooling. A series of firmware updates were tested, and although targeted very low airflow rates were not achieved, indications are that they can be achieved, and additional experimentation could produce the desired result.

Variable-Capacity Heat Pumps:

- Inverter-driven systems that vary compressor speed, refrigerant flow rate, and airflow enabling output ranging from 10%–120% of nominal rated capacity
- Operate with extended run time (65%–70% of a typical day compared to 30%–35% for a fixed-capacity system)
- Ability to vary sensible heat ratio creates potential for better indoor relative humidity control during low load.

For the field studies, two types of houses built by Habitat for Humanity were monitored in two Florida locations. Each affiliate employed different space-conditioning and ventilation strategies in highly efficient homes that met the DOE Zero Energy Ready Home program requirements. The Southeast Volusia Habitat for Humanity (SEVHFH) affiliate used a 1.5-ton, ductless multisplit heat pump with outdoor air supplied by an energy-recovery ventilator, and the South Sarasota Habitat for Humanity affiliate used a centrally ducted 1.5-ton minisplit heat pump with an integrated AirCycler g2-k hybrid ventilation system. The system involved a combination of central fan integrated supply with an efficient exhaust fan to supply ventilation when heating and cooling was not called for. Although the technology was designed for a fixed-capacity system, results indicated that with full knowledge of minisplit system operation and monitored data, the g2-k could be set up to deliver the design mechanical ventilation air exchange.

Although field studies show that temperature could be maintained most of the time, managing RH levels is a challenge. Field-study results from the fully ducted systems in South Sarasota (SS1, SS2, and SS3) showed that the ACCA Manual RS guidelines for temperature distribution could largely be met. Results from SEVHFH (SEV1 and SEV2) indicate that, with only minor excursions, the ductless heat pump can also maintain adequate temperature distribution in all rooms when coupled with continuously operated transfer fans; however, with the exception of one home, which maintained exceedingly warm indoor temperatures, indoor RH was greater than 60% much of the time in all homes, with considerable amounts of time when the RH was greater than 65%. Increased indoor RH was found to occur during swing season months, and even summer evenings presented low load comfort challenges for the cooling systems. Many of the hours with higher indoor RH were found to be related to periods when the heat pumps would cycle rather than consistently deliver low-capacity operation, or they would vary operation between high and low capacity and deliver higher SHR. Hours with lower indoor RH corresponded to more consistent heat pump run time, lower coil airflows, and colder supply air temperatures. Overall, although target comfort metrics were not achieved, none of the homeowners reported discomfort.

To address high latent loads in low-load homes, extended run time of the cooling system is required during low-load hours. Although variable-capacity systems have the capability to do so, we frequently find situations where they do not run consistently at the very low end of their operational capacity range. Instead, during low-load conditions occurring during overnight summer hours, many systems exhibit cycling behavior, which inhibits moisture removal. Varied operation is also evident at times during higher load conditions, as units often frequently change speeds in response to loads rather than exhibit steady operation.

In addition to steady operation, the delivery of low SHR—and hence cold supply airflows during operation—is also required to control indoor RH. Although

Variable-capacity heat pumps have great potential to help control indoor relative humidity.

- Need to maintain a colder coil during low load and decrease sensible heat ratios as much as design allows.
- Need to use the lowest capacity consistently for longer periods to avoid (1) cycling and (2) overcooling.
- Coil airflow needs to be able to operate at the low end of the operational range to achieve these objectives.
- Need to use an RH sensor to recognize low-load conditions and enter into/exit from RH control mode.

variable-capacity systems have the capability to vary SHR, many systems tested will often opt for higher SHR in efforts to efficiently control indoor temperature. This is reasonable in a standard cooling mode; however, a dry or RH control mode should also be available that runs at lower SHR as needed for improved humidity control. To deliver low SHR and not result in excessive overcooling during low-load periods, the coil airflow must be low. Data show that manufacturers should even consider *extending* their standard operating airflow ranges and developing special low-airflow modes that are essential for improving dehumidification and increasing overall system efficiency in low-load homes.

As is the case with many variable-capacity systems, an operational "dry" mode exists that attempts such an operational configuration; however, results using the dry mode to control indoor RH while limiting overcooling are mixed. Improving the low-flow accuracy and control algorithms will improve dehumidification in variable-capacity systems by lowering the coil airflow to the point at which low capacity and long run times can be achieved without overcooling. The next improvement would be a smarter dry mode not solely controlled by sensible temperature. This would require a humidity sensor as a control algorithm feedback. A smart dry mode would also be able to move out of the dry mode into the standard high-efficiency cooling mode when RH levels are low enough.

Although the research conducted and associated discussion focuses on inverter-driven, variable-capacity systems, many of the proposed system enhancements could be applied to more conventional two-speed systems with variable-speed electronically commutated motor air handling unit motors to improve their latent performance at part load.

Table of Contents

List of Figures

List of Tables

1 Introduction

The home building industry continues to strive to reduce the heating and cooling costs of new and existing housing. Common techniques include implementing well-sealed, highly insulated building enclosures coupled with efficient lighting and appliances. As a result, peak heating and cooling loads have been reduced. In addition, the seasonal duration for which air conditioning and heating is called for are reduced (Henderson and Rudd 2014; Rudd and Henderson 2007). A challenge has emerged with regard to the selection of appropriate space-conditioning equipment for modestly sized homes that have very low heating and cooling loads—i.e., low-load homes. Conventional, fixed-capacity heating and cooling equipment capacities render readily available equipment oversized for many new, efficient homes. Even if fixed-capacity equipment can be properly specified for peak loads, it will remain oversized for use during most of the year. During these *part-load* cooling hours, oversized cooling equipment meets the target dry-bulb *temperature* very quickly without circulating the air over the cooling coils very often. This is a disadvantage for moisture control (Shirey 2004).

This shortfall in moisture removal with oversized equipment can be especially problematic for high-performance houses in humid climates when meeting ASHRAE Standard 62.2 recommendations for whole-house mechanical ventilation. Mechanical ventilation in humid climates introduces significant moisture from outside air into the conditioned space (Parker et al. 2016; Fairey et al. 2014; Widder et al. 2017; Harriman et al. 1997). The additional latent load coupled with the diminished sensible load of a high-performance thermal envelope makes the need for enhanced moisture removal capacity of space-conditioning equipment more critical than ever.

Although this dynamic is present in homes of all sizes, it is more challenging in small homes that have the same internal moisture gains as larger houses but mixed with a smaller volume of indoor air. In affordable housing, this challenge is sometimes compounded by high occupancy density. If a solution to control moisture in high-performance, low-load affordable housing in the hot-humid climate can be identified, it will likely work throughout the climate zone in larger homes, and it could also benefit homes in other climate zones where low-load conditions also exist, even if such conditions only exist for a shorter duration. Note that the percentage of new and existing homes that need a low-load comfort solution is increasing. In addition to above-code homes, such as those built to comply with the standards for ENERGY STAR® for New Homes and the Zero Energy Ready Home, advancing energy codes for new construction bring urgency to identifying a solution that addresses this nexus of conditioning challenges: low sensible heat gain, steady internal moisture generation, and high outside air moisture gain.

Current solutions to control indoor moisture during low-load conditions typically involve using supplemental dehumidification, which adds to first costs, maintenance costs, and space-conditioning energy use (Withers and Sonne 2014; Rudd, Lstiburek, and Ueno 2005) An integrated space-conditioning system capable of maintaining reasonable indoor relative humidity (RH) during low-load periods could possibly eliminate the need for a supplemental dehumidifier, or at least significantly minimize the hours of operation, and potentially offer a more economical solution. Although most conventional systems primarily respond to dry-bulb temperature, some systems can also be controlled to respond to indoor RH, and they can act to reduce RH by overcooling to less than the desired dry-bulb set point temperature. However, systems are typically limited to only a few degrees of over cooling beyond the dry bulb set point temperature, which can often make this feature ineffective. One potential solution of interest is using inverter-driven comfort systems that have the ability to automatically adjust their capacity in response to varying load. Although this often results in exceptional energy savings, questions remain about whether the way variable-capacity systems respond to sensible load inherently also manages the corresponding latent load and if system operation can be adjusted by homeowners, contractors, or manufacturers to optimize comfort.

One class of variable-capacity space-conditioning systems currently on the market integrates with homes in a manner similar to traditional, centrally ducted split systems, with a single air handling unit (AHU) delivering conditioned air to all spaces of a zone through a conventional ducted distribution system. Recently, Unico, Inc.

designed one such system to operate using their small-duct high-velocity (SDHV) distribution approach. The smaller ducts of the SDHV system make them easier to fit into smaller spaces and more likely to fit into interior duct designs. SDHV systems are also promoted as offering very effective mixing of conditioned air that helps improve thermal distribution and occupant comfort. The SDHV approach enables lower coil airflow per unit of cooling capacity, which is conducive to moisture removal, and air velocity is later boosted and distributed through small-diameter duct work to achieve mixing and evenness of comfort throughout the home. Another readily available type of variable-capacity space-conditioning system is a minisplit heat pump, which uses a single, typically ductless indoor fan coil connected to a single, typically small-capacity outdoor unit. Such systems typically rely on multiple, small-capacity indoor fan coils to supply conditioned air to multiple spaces, rather than relying on ducted distribution. When multiple fan coils are connected to single outdoor unit, the system is typically referred to as a multisplit heat pump. To reduce complexity and the cost of installing multiple fan coils, two strategies are emerging to distribute conditioned air throughout the home: (1) fan coils are available that allow for the use of a duct system with low static pressure to serve all or part of the home, and (2) transfer fans are sometimes installed to force air movement from spaces that have indoor fan coils to spaces that do not.

This project's goal is to evaluate the performance of these new and emerging systems and strategies to maintain whole-house comfort in low-load, mechanically vented homes. The results achieved will build on the body of knowledge that already exists for the performance of variable-capacity comfort systems, as described in the next section. Ultimately, the work seeks to (1) inform the marketplace on the performance benefits and limitations of currently available variable-capacity systems toward providing the moisture control necessary to maintain indoor comfort, and (2) develop recommendations for manufacturers of variable-capacity and conventional equipment to improve the latent performance of equipment through the modification to hardware design and control algorithms.

1.1 Background

A few inverter-driven, centrally ducted heat pumps available from major equipment manufacturers have been tested in laboratory homes to investigate factors affecting efficiency and moisture removal (Cummings 2011; Cummings, Withers, and Kono 2015; Withers, Cummings, and Nigusse 2016a; Munk 2012). Although the potential for advanced indoor humidity control has been documented through the ability to achieve long run times, only moderate improvements compared to fixed-capacity systems have been achieved. Some research on SDHV systems has been conducted by Building America (Poerschke 2017), but this has been primarily on the improvement of air mixing and not specific to improvement in latent control. No published research on the performance of SDHV mated with variable-capacity equipment is available.

Building America has amassed a body of work on the performance of minisplit heat pumps, much of which is described in Ueno and Loomis (2015). Much of the minisplit work has investigated comfort distribution related to single-point or a few discrete points of distribution. The Air Conditioning Contractors Association (ACCA) Manual RS (ACCA 1997) is often used to define minimum/maximum recommended values for comfort distribution, defined as plus or minus 3°F temperature difference between the set point temperature and any room for a single-zone system for the cooling season. Some findings of the research show that although single-point distribution cooling can have trouble achieving these metrics in a two-story home, it can work in a single-story home as long as doors remain open, and mechanically moving air from one room to another might offset the negative effects of door closure. Many comfort distribution failures of single-point distribution cooling have been found to occur during equipment cycling, and the use of inverter-driven systems with higher run time fractions might alleviate such problems.

The bulk of the minisplit research has been done for heating applications, and little has been done for cooling applications with an emphasis on latent control (Brown, Thornton, and Widder 2013). A few select studies with varying results have been conducted in occupied homes in hot-humid climates (Roth, Sehgal, and Akers 2013). Withers (2016b) specifically investigated the ability of inverter-driven heat pumps to energy-efficiently control moisture in a simulated occupancy lab home as part of an integrated approach to comfort, and the study found limitations related to inconsistent delivery of a low sensible heat ratio (SHR). Although not much supplemental dehumidification was needed, indoor RH tended to run close to 60% often during overnight hours between approximately 3 a.m.–8 a.m. It was found that part of the problem was that some variable-capacity systems were not maintaining a cold enough coil during low-load periods during cooling, and one system did not operate even close to the manufacturer's stated lower capacity.

Differing metrics have been used in the literature to define comfort. ASHRAE Standard 55 is one such metric, and it identifies ranges of acceptable combinations of temperature and RH (ASHRAE 2010); however, some consider that its comfort metrics are more closely representative of comfort in a commercial building environment. A recent Building America Expert Meeting report (Rudd 2013) documents an RH limit of 60% as a reasonable residential comfort metric, which is what is used in this study. The report indicates that exceeding 60% RH does not necessarily indicate failure, but that "a 60% RH limit provides the best practice coverage for providing comfort and durability over a reasonable range of varying factors, such as internal moisture generation rate, and occupant comfort perception and susceptibility to illness stemming from elevated indoor humidity." The report also indicates that annual hours greater than 60% RH is the single most appropriate humidity control performance metric to use to compare system performance and to compare required supplemental dehumidification energy. That metric does give generally the same result as looking at 4-hour and 8-hour events greater than 60% RH.

1.2 Relevance to Building America Goals

In pursuit of goals including reducing the energy use intensity of new homes by at least 60% and existing homes by at least 40%, relative to the 2010 average for homes in each major U.S. climate, Building America has identified space-conditioning challenges unique to high-performance homes that impact both energy use and comfort. These include the need to provide more effective part-load temperature and humidity control, which comprise a greater portion of overall conditioning in high-performance homes, and the need to ensure air distribution and temperature control throughout the occupied spaces, even under very low-load conditions, which often means a very low flow of conditioned air.

Elements of the Building America Research to Market Plan (Building America Program 2015) include efforts to ensure that the equipment needs of low-load homes can be met with off-the-shelf products by collaborating with manufacturers on field and laboratory research. To contribute to those efforts, this research was conducted in conjunction with three equipment manufacturers striving to understand the conditioning profiles of low-load homes.

Research described in this report seeks to validate system approaches for the energy-efficient management of temperature and RH in low-load homes located in hot and humid climates. This is particularly challenging in homes with whole-house mechanical ventilation provided at ASHRAE 62.2 levels, such as the test homes in this research.

The research questions to be answered are:

1) How does the total space-conditioning energy consumption (cooling plus dehumidification) of a Unico SDHV variable-capacity system compare to a centrally ducted fixed-capacity Seasonal Energy Efficiency Ratio (SEER) 13 system and a variable-capacity SEER 22 system when indoor RH is maintained less than 60%.

2) How well is indoor temperature controlled with transfer fans compared to fully ducted systems?

3) Can the achievement of design mechanical ventilation rates be ensured when integrating supply ventilation with a variable-capacity minisplit system?

4) What variable-capacity cooling system operational characteristics and patterns are observed in the collected data that might assist manufacturers with improved indoor RH control as they refine existing equipment and develop new products?

2 Laboratory Investigation of Small-Duct High-Velocity iSeries System

The performance of an SDHV variable-capacity heat pump was tested in a laboratory home mechanically ventilated to ASHRAE 62.2-2016 standard. The work involved a partnership with the manufacturer, Unico, which manufactures and sells various heating and cooling product lines. The iSeries product tested here used an inverter-based outdoor heat pump made outside the United States and that relies on engineering from abroad to address firmware changes. This is common in variable-capacity equipment used in the United States. In this case, a clear channel of communications had been established between Unico engineering and overseas engineering, which resulted in quick turnaround of straightforward firmware changes. Remaining challenges regarding firmware changes will be addressed later in this report.

2.1 Laboratory Description

The manufactured housing laboratory (MHL) is a 1,600-ft^2 double-wide manufactured home with an unvented crawl space, R-19 floor insulation, a vented attic with at least R-30 insulation on the ceiling, wood frame wall construction with R-19 insulation, three bedrooms, two bathrooms, and a large open central area. An exterior view is shown in Figure 2-1.

Figure 2-1. Florida Solar Energy Center manufactured house laboratory. The north side is shown.

The test lab floor plan and system layout are shown in Figure 2-2. The AHU was located in the utility room. The blue rectangular boxes show the round 8-inch-diameter and 7-inch-diameter main supply ducts. The smaller blue lines show the individual 2-inch-diameter supply ducts that terminate in each primary room. Supply mechanical ventilation was delivered to the utility room near the AHU return. A stand-alone dehumidifier was placed in the living room area.

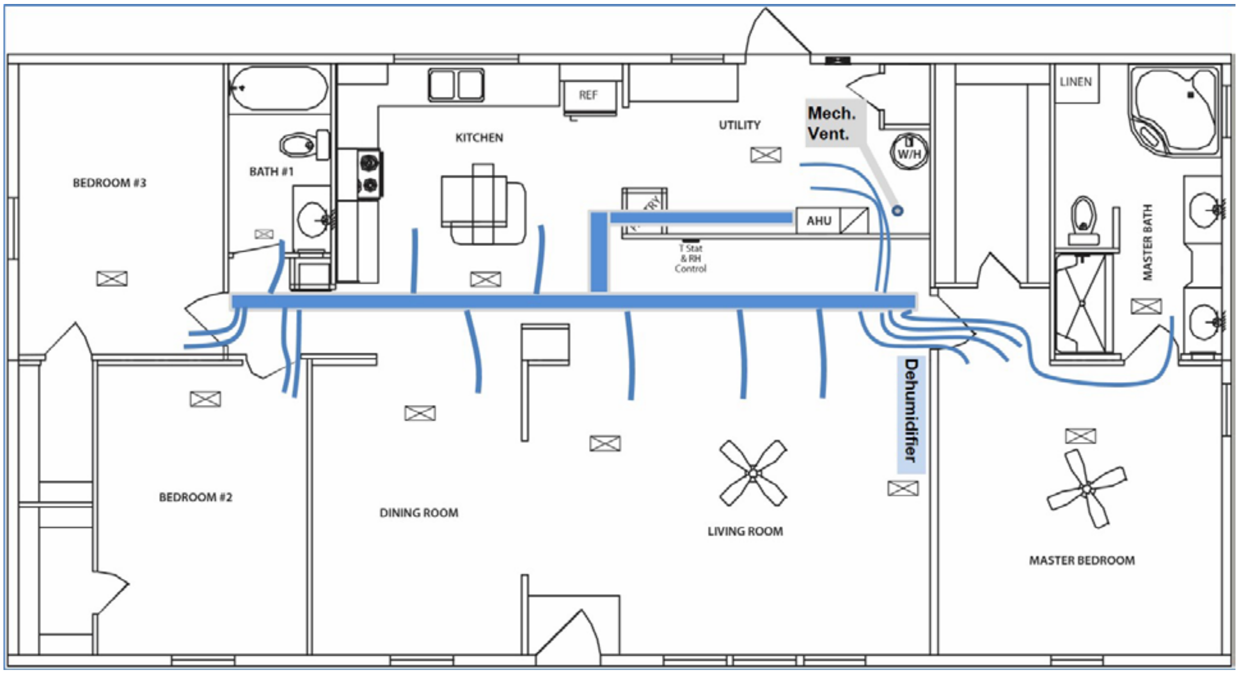


Figure 2-2. Manufactured home laboratory floor plan and equipment layout

Because testing under low cooling loads was important, the system was tested with the ducts in the conditioned space. Figure 2-3 and Figure 2-4 show portions of the shiny insulated ducts of the installed SDHV system within the test lab. Return air came from the utility room through a central plenum on the end of the AHU.

Figure 2-3. System air handling unit shown on the left side of the utility room area, with the main duct and branch in the kitchen

Figure 2-4. Main 8-inch duct with 2-inch ducts shown in living room area. A dehumidifier is against the wall below the ductless minisplit.

2.2 Mechanical Equipment Description

Continuous mechanical ventilation was supplied to the MHL during all testing. It was supplied in the utility room area close to the centrally ducted return intakes. The mechanical outside air was ducted close to the return but not directly into the return plenum. This allowed dry indoor air to mix with the outside air, which helped decrease the RH before the air contacted any indoor surfaces. The outside air was delivered at a velocity low enough that it could be effectively captured by the return air when the cooling system was on. Figure 2-5 shows an in-line fan (top left of image) that was used to bring the outside air into the lab continuously through a 6-inch-diameter rigid metal duct. An adjustable IRIS damper (below the fan) was calibrated against a TSI Wind Tunnel model 8390. The IRIS damper and digital micromanometer were used to monitor the flow of the outside air at 10-second intervals during every test configuration. The iSeries return intake is shown in the low right corner of Figure 2-5. The AHU above the iSeries AHU is from a previously tested SEER 13 heat pump. The indoor and outdoor units of the iSeries system are shown in Figure 2-6.

Figure 2-5. Mechanical ventilation supplied to the utility room

The outside air intake was located at the shaded north side of the building (just out of view, near the stairs, in Figure 2-1). The outside air was pulled in through an air filter at the intake box outdoors. From the intake filter, a 6-inch-diameter metal pipe with R-6 insulation was run under the home through the crawl space, then up through the utility room floor.

The space heating and cooling system tested was the new Unico SDHV variable-capacity heat pump. The Unico variable-capacity system is offered under the iSeries brand. The outdoor inverter unit can be connected to up to four different refrigeration circuits. This could be done with a combination of ducted or ductless units. The SEER can be as high as 18 in some cases. Only a single ducted indoor unit was operated with this system during testing, and testing this feature is covered in this report.

The design cooling load of the MHL was 18,200 kBtu/h with indoor central ducts. Internal sensible loads had a daily average approximately 3,400 Btu/h with approximately 10 pounds of daily internal latent moisture generation. During summer conditions with an outdoor dew point of 72°F, the mechanical ventilation system delivered approximately 48 pounds of latent load. The SDHV system's rated capacity and efficiency are shown in Table 2-1. The tested system had a rated cooling capacity of 29,200 Btu/h and heating capacity of 35,200 Btu/h. The cooling capacity could drop to as low as approximately 2,900 Btu/h, making it well-suited for very low-load periods.

Table 2-1. Unico iSeries Small-Duct High-Velocity System Air-Conditioning, Heating, and Refrigeration Institute (AHRI) Data

AHRI Reference Number	Outdoor Unit	Indoor Unit	Cooling					Heating				
			95°F		82°F		SEER	47°F		17°F		HSPF[c]
			Btu/h	EER[a]	Btu/h	EER		Btu/h	COP[b]	Btu/h	COP	
7849805	IS36G110	M3036CL1-A + M3036BL1-EA2	29200	7.45	31600	8.80	14.00	35200	2.40	23200	2.05	8.35

[a] Energy Efficiency Ratio
[b] Coefficient of Performance
[c] Heating Seasonal Performance Factor

The Unico system is different from other centrally ducted systems. The indoor coil design is one difference. It was designed for relatively lower airflow (cubic feet per minute [cfm]/ton of cooling) and has four rows of tubes instead of the typical three-row coil. This coil design is intended to meet capacity at lower flow rates by supplying much colder supply air than a more conventional ducted system. A benefit of lower supply air temperature (SAT) is that it improves dehumidification. Another difference is the SDHV air distribution. The terminal ducts are only 2 inches in diameter and result in higher supply air velocity than standard ducts. This reportedly allows more even air temperatures in each room by mixing the airstream rather than relying on throw (Baskins and Vineyard 2003). A special terminal duct design helps attenuate noise associated with higher velocity.

Figure 2-6. Indoor and outdoor iSeries units

A Sunpentown SPT model SD-71E supplemental dehumidifier was placed in the living room main space. This dehumidifier had a rated moisture removal rate of 70 pints/day and a rated energy factor of 1.85 L/kWh. The listed power draw was 720 W, but measured average power was only 580 W. On-board dehumidistats are limited to sensing at the appliance's location, which might not be the best for sensor control. The dehumidifier operation was controlled by a separate dehumidistat controller that was placed on the interior wall in the large central living room area next to the space-cooling thermostat controls. A Green Products dehumidistat that was designed for placement at remote locations had a built-in power relay control that turned power on and off to the dehumidifier as needed. The dehumidifier on-board RH control was set to the lowest (driest) setting so it would not turn the unit off before the remote control called for it to do so. The location of a dehumidistat is very important. It has been found to be preferable to locate it in a central conditioned area without direct impact from supply air or mechanical ventilation air (Withers, Cummings, and Nigusse 2016b).

Other equipment in the lab was used to generate internal latent and sensible loads. An oven, heat lamps, dishwasher, and bathroom shower were controlled by an automation system. More detail on this can be found in Withers, Cummings, and Nigusse 2016b).

2.3 Instrumentation and Equipment Descriptions

More than 100 channels of data were collected for meteorological parameters; the envelope; the heating, ventilating, and air-conditioning (HVAC) systems; and interior space conditions. The instrumentation package consisted of multiple data loggers and associated peripheral devices. Sensors were read every 10 seconds and stored at 15-minute intervals. Therefore, each 15-minute data interval was represented by 90 samples.

A data acquisition system recorded a variety of information about the HVAC system operation, energy consumption of various items within the house (including internally generated sensible and latent loads), and indoor and outdoor conditions. Temperature and RH of air flowing into and out of the HVAC systems were recorded only when the systems were operating (conditionally). A list of test equipment and monitoring sensors used is presented in Table 2-2.

- Temperatures were recorded conditionally at the entrance to the system returns (which were in the conditioned space and less than 2 feet long), at the discharge from the systems, and at five supply registers for the ducted systems. Temperatures entering the condenser coil (outdoor units) were also recorded.
- Temperatures were recorded unconditionally (continuously) at various indoor locations, in the attic, in the crawl space, and at various locations on the roof system.
- RH was recorded conditionally at the entrance to the return and the discharge from each system.
- RH was also recorded at various indoor locations, in the attic, in the crawl space, and outdoors, all unconditionally (continuously).
- The airflow rate of cooling systems was recorded by airflow stations located at the entrance to the returns.
- Power meters recorded energy use for the house, HVAC indoor and outdoor units, the refrigerator, the domestic water heater, the oven, air circulation fans, the dishwasher, and heat lamps that simulated internal loads.
- Condensate draining from the HVAC systems and the dehumidifier was measured by a pair of tipping buckets that provided redundant measurements of moisture removed by the coils.
- Weather conditions of air temperature, RH, rainfall, wind speed/direction, and solar radiation (on the horizontal) were measured.

Table 2-2. Lab Testing and Monitoring Equipment Used in the Experiments

Measurement	Equipment	Accuracy
Data collection	Campbell Scientific CR10 with (2) AM416 multiplexers and (1) SW8A pulse expansion module	0.06%
Pressure differentials (airflow sensors, air distribution pressures)	DG700 and DG-2 digital pressure gauge with analog output	1%
Central ducted system airflow	Supply flow station; DG-2 digital pressure gauge	In situ calibration
Ventilation airflow	Continental Fan Manufacturing IRIS Damper; DG-2	In situ calibration
Airflow calibration	TSI Model 8390 Bench Top Wind Tunnel	2%
Temperature	Type T thermocouple	0.2°F
RH (return, supply, outside air, indoor, outdoor, attic)	Vaisala HMP50 and HMP60	3% RH
Condensate	Texas Electronic TR-4 and TR-525I tipping buckets	3%, 1%
Energy (whole-house, AHU, condenser unit, domestic hot water, oven, refrigerator, dishwasher, heat lamp circuit)	Continental Wattnode and Ohio Semitronics, Inc., energy transducers with current transformers from 5–200 amps	1%
Building envelope air leakage	Minneapolis Blower Door System with DG-700 digital gauge	3%
Duct system air leakage	Minneapolis DuctBlaster System with DG-700	3%

Central heating and cooling system airflow were measured based on a correlation of airflow versus supply static pressure. An Energy Conservatory TrueFlow flow plate was used at the return intake and supply static pressure measured with a DG700 manometer and static pressure probe. A total of 18 different airflow measurements were taken. The data were used to develop a flow curve for the installation, which is shown in Figure 2-7.

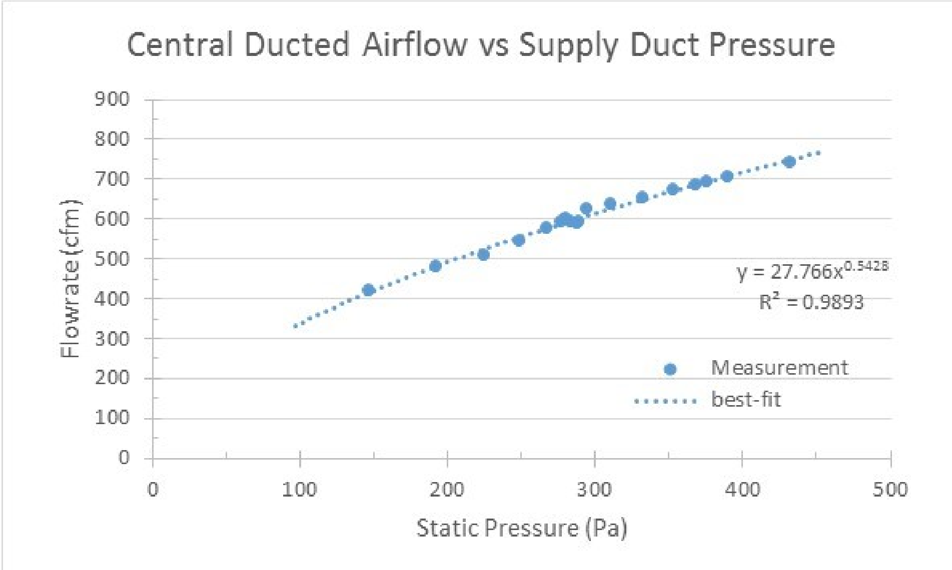


Figure 2-7. Flow calibration for small-duct high-velocity central system

2.4 Test Sequence Description

After installation, the system was operated, and the performance was evaluated by the manufacturer. The Unico iSeries system has a unique built-in proprietary data logging system that can allow a laptop to be connected to it to collect several performance-related data points designed to allow the manufacturer to review and identify any issues. It also allowed the manufacturer to confirm that the system was performing as expected.

The Unico iSeries SDHV heat pump was operated and tested under its heating, standard cooling, and dry cooling modes. Test configurations were changed throughout time to test it under a variety of weather conditions. Heating weather was very mild, and there was very limited opportunity to test heating. Upon initial testing, it was found that the existing firmware shipped with the unit resulted in the central indoor fan staying on after cooling cycles had ended. We asked the manufacturer to develop a version of firmware that disabled the continuous ventilation feature, and the manufacturer quickly sent an updated version that ended the continuous AHU fan operation. Firmware was uploaded using a laptop with USB cable connection to the iSeries communication terminal.

The iSeries standard cooling mode operated very similarly to other variable-capacity standard cooling modes. Delivered cooling capacity is adjusted based on the difference between the thermostat set point and the room temperature sensor. As the cooling load increased, the delivered cooling output increased. During very low cooling load periods, the system would cycle off and on.

The iSeries also had a dry cooling mode designed to improve indoor moisture control during very low-load periods. This mode must be selected to enable it. The iSeries thermostat control did not have an RH sensor, therefore it relied on a proprietary temperature-based control. This control would lower cooling capacity based on how much cooler the room was compared to the set point. A total of three levels of dry control operation were included; one of these includes cycle off, and another is a transition into standard cooling. If the room was approximately 3.6°F warmer than the set point, the system was intended to run in the standard cooling mode. The second level operated at a nearly fixed low-cooling output, and it was intended to modulate the AHU fan between the very low and low fan speeds. The third level was designed to provide the lowest cooling capacity possible when the room was approximately 1.8°F cooler than the set point. This level was also allowed to continue until the room temperature reached 50°F. This allowed too much overcooling, and the manufacturer improved this later.

After several months of testing, recommendations were made to the manufacturer to further improve the dehumidification performance in the dry mode. This test mode will be referred to as the dry II mode. Manufacturer engineering staff were able to implement some of the recommendations in the time available. One improvement was to eliminate the very low 50°F cycle-off set point and establish a control based on the temperature difference between the room and the set point. An eight-degree difference was set; however, this can allow significant overcooling depending on the set point. Another improvement was to try to further reduce airflow because we did not observe the lowest flow rates expected. There were technical challenges in overcoming this during the research project; these are discussed in more detail in Section 2.6. Although we were not successful in getting the AHU to operate at the lowest flows intended, the manufacturer found the cause and determined that the issue could be overcome.

The Unico iSeries variable-capacity heat pump integrated with the SDHV central system was ordered in November 2015. Installation was completed at the end of December 2015. Testing began at the beginning of January 2016 and continued into July 2017.

2.5 Results

Test results reported here cover energy use of the iSeries heating and cooling central system as well as the supplemental dehumidifier. Dehumidification performance is discussed, and the resulting indoor comfort metrics of temperature and RH are also summarized.

2.5.1 Space-Conditioning Energy

Daily total space-conditioning energy was plotted against the daily average temperature difference between the outside and inside, as shown in Figure 2-8. A least-squares regression analysis was completed with available data and best-fit lines.

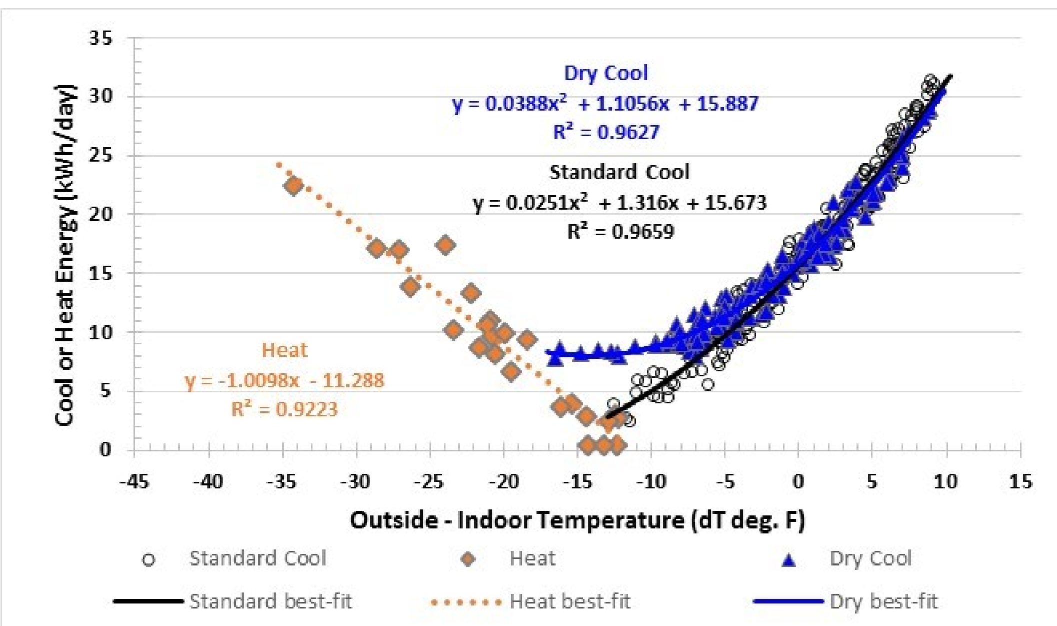


Figure 2-8. Space cooling and heating energy compared to dT. Two different cooling modes are shown.

The most obvious result is how the iSeries dry mode daily energy is nearly the same as the iSeries standard mode at daily average dT more than approximately -1°F. This is expected because dry mode was designed to be the same as the standard mode during high cooling loads. The significant increase in iSeries dry mode energy at low dT is also shown here. This can be explained by the way the dry mode control algorithm was set, as previously discussed. The iSeries unit did not have an RH sensor, and it operated based on only interior temperature and the temperature set point. When the room air temperature increased to more than the set point, the system began to operate in the normal cooling mode, thus the dry and standard energy converges at higher dT (warmer weather). At very cool temperatures, the dry mode energy becomes constant because of a timer-based control at the dry cool level of the lowest cooling output.

The dry mode was not observed to cycle off until a very low indoor temperature of approximately 58°F was reached. The cycle-off set point was reported to us to be 50°F, so we are not sure why it cycled off sooner than expected. A cold front came in quickly during this particular time, and outdoor temperatures approached 45°F, whereby an outdoor unit control might have interceded to shut down for a few hours.

This very low shutoff point was pointed out to the manufacturer. Significant overcooling no doubt wasted energy not needed to dehumidify and poses potential building degradation from condensation on overcooled building surfaces under certain conditions. Unico responded and provided a firmware upgrade for the dry mode that increased the timer-based cooling cycle time compared to the previous version and also created a system based on a minimum room temperature limit associated with the set point temperature instead of setting an absolute minimum of 50°F. Dry-mode operation and opportunities for improvement will be covered more in Section 2.6.

The iSeries system controlled indoor humidity very well. There was no need for the supplemental dehumidifier when the system was run in dry mode. There was very little need for supplemental dehumidification when solely run in the standard cooling mode. Figure 2-9 shows the cooling energy in standard mode on the left axis and the dehumidification energy on the right axis. In the 188 standard cool test days, the dehumidifier operated only 12.7 hours out of 4,512 hours (0.28%). Dehumidifier operation occurred 27 days out of 188 days, or 14.4% of test days.

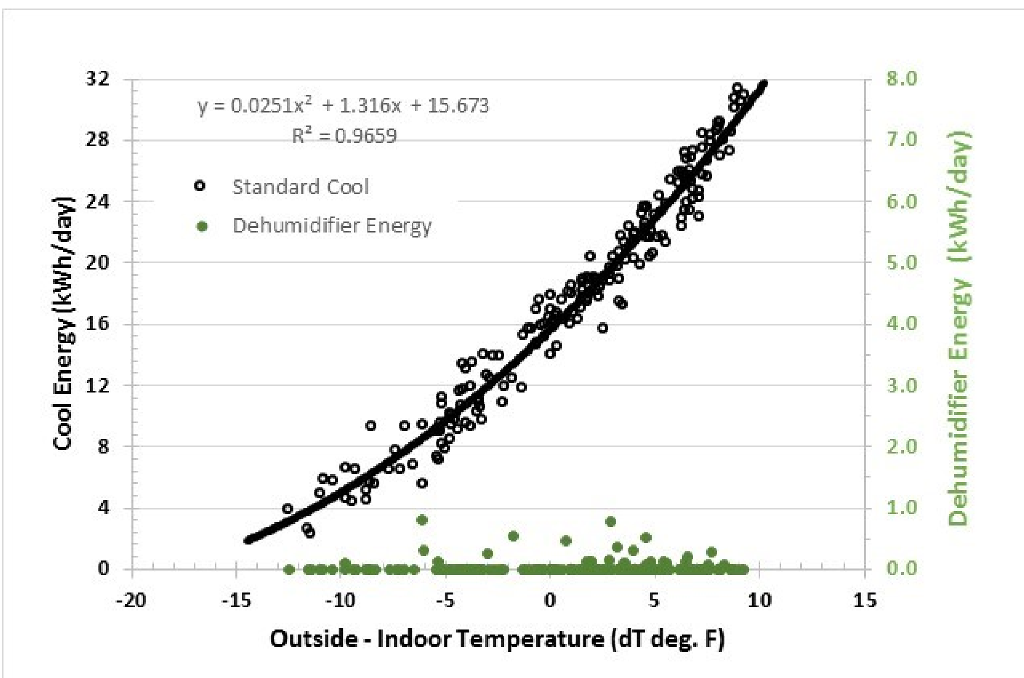


Figure 2-9. Cooling and dehumidifier energy compared to dT shown for the small-duct high-velocity standard cooling test

Figure 2-10 shows the iSeries SDHV test data along with test results from other selected systems previously tested during 2014–2015 and reported in Withers (2016b). All lab conditions were the same in these tests. Previous testing occurred with the same interior loads as the iSeries testing, but the data to compare to iSeries were more limited. The only change was weather and the space-conditioning system used. The data shown in

Figure 2-10 include any dehumidifier energy as well as the air-conditioning energy. As shown for the Unico SDHV data, the other systems in Figure 2-10 used very little dehumidifier energy at the 60% RH set point.

The tests shown from Withers (2016b) are for a fixed-capacity, centrally ducted SEER 13 heat pump (SEER 13), SEER 22 variable-capacity heat pump operated in humidity-control mode (SEER 22 Dry), and a ductless minisplit SEER 21.5. The minisplit test used the minisplit for primary cooling along with the ducted SEER 13 system for secondary cooling and air cycling (minisplit and SEER 13). The centrally ducted systems shown here used supply ducts located in a vented attic.

Adequate data were not available for significant heating analysis, so no heating data are compared here. The SDHV standard cool SEER14 rated energy use (black line in Figure 2-10) falls between the SEER 13 (red line) and previously tested variable-capacity systems. This is approximately what would be expected given the SEER ratings. The Unico SEER 14 rating is for standard cooling. Dry-cool mode does not require a SEER rating.

Previous testing of other cooling systems under test conditions (Withers 2016b) similar to those of the iSeries testing also indicate low dehumidifier use in less than 1% of the hours tested during summer conditions. Although this is low, the dehumidifier use of the other systems was more, and average indoor RH was higher than the iSeries system.

All systems shown in Figure 2-10 were able to maintain indoor cooling temperatures between 76°F–77°F during summer conditions; however, the SDHV system maintained daily average indoor humidity at an average of 10% RH lower than the SEER 22 ducted system and approximately 7% RH lower than the SEER 13 ducted system during days when the average outdoor temperatures were between 80°F–82°F and outdoor dew point temperatures were higher than 70°F. The drier air resulting from better latent performance from the SEER 14 SDHV system is important to consider when comparing cooling energy based on sensible dry-bulb temperature control. The implications of space cooling plus supplemental dehumidifier energy become even more important at lower RH set points, as is discussed in Section 2.5.2.

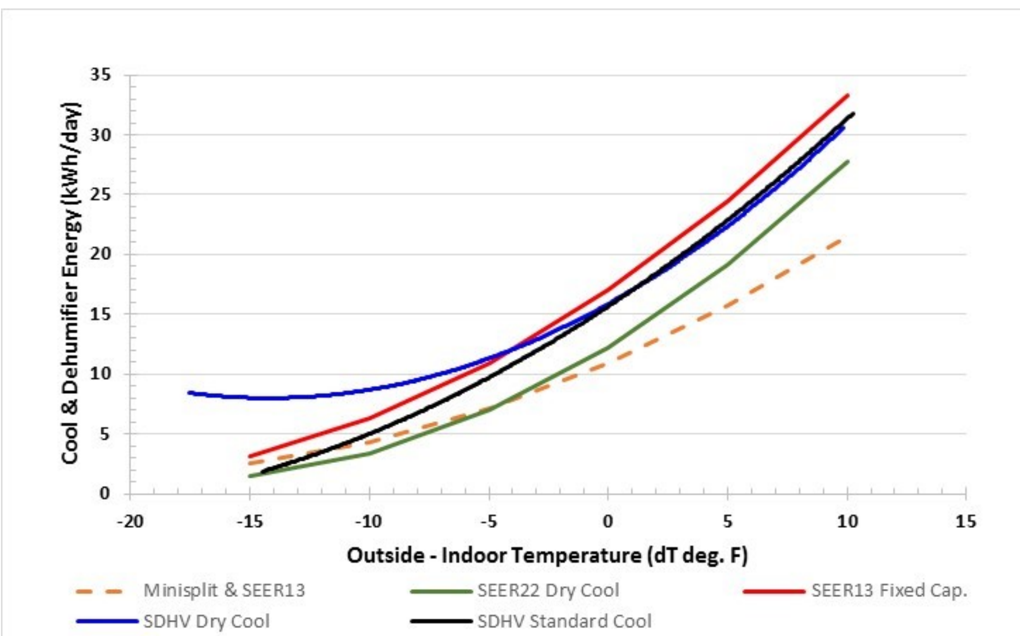


Figure 2-10. Least-squares regression best-fit lines shown for four different systems and five different tests

Table 2-3 summarizes the daily cooling energy for a typical summer day with a dT = 5°F (outdoor 81°F – indoor 76°F) for three centrally ducted systems with different rated efficiencies. The SEER 13 and SEER 22 results are from previous testing (Withers 2016b).

Table 2-3. Daily Space-Conditioning Energy for a Typical Summer Day with dT = 5°F. Includes Dehumidifier Energy Set at 60% RH Set Point

	Test Configuration		
	Fixed-Capacity Central Duct SEER 13 (kWh/day)	Variable-Capacity SDHV SEER 14 Standard Cool (kWh/day)	Variable-Capacity Central Duct SEER 22 (kWh/day)
	24.4	22.4	19.2
Savings relative to SEER 13	---	2.0 (8.2%)	5.2 (21.3%)
Savings relative to SEER 14	---	---	3.2 (14.3%)

2.5.2 Simulation Results

An annual simulation was completed to explore space cooling and heating energy use as well as dehumidifier energy at different RH set points. The challenge is that without details about actual air conditioner and dehumidifier dehumidification performance, the dehumidifier results can differ significantly from reality. The purpose of this simulation work was not to simulate using actual performance data but to see if a lower rated efficiency heat pump with good dehumidification performance might use less total annual space-conditioning energy than a higher efficiency heat pump with perhaps typical dehumidification performance. We know from testing, that the iSeries system did not need a dehumidifier to maintain at least 60% RH if dry mode was used as needed. The iSeries system also resulted in much lower humidity levels than a previously tested SEER 22 system and might be able to maintain RH at 50% without supplemental dehumidification with some minor control modification. If a dry mode can be turned on only as needed and SHR dropped in the lowest stage of the variable-capacity SDHV system, it is expected that no dehumidifier would be needed to maintain approximately 50% RH all hours of the year.

The simulations used EnergyGauge USA v5.1.01 with the MHL attributes. Seven simulations were run for the MHL with a SEER 22 heat pump and supplemental dehumidifier. Each simulation was run by changing only the dehumidifier RH set point. Figure 2-11 shows an example of space-conditioning energy use simulated for the test lab. A separate simulation was run for a SEER14 without supplemental dehumidification to represent a system like the SDHV iSeries system. The SEER 14 SDHV simulation was completed with the assumption that supplemental dehumidification would not be needed for an optimized SEER 14 SDHV system capable of maintaining indoor humidity of no more than 50% RH all hours of the year.

The simulations do not account for specific performance data such as variability in total cooling output and SHR under different indoor entering coil conditions. An average SHR of 0.76 was used for each cooling system simulation. Dehumidifier energy would be more than indicated if actual SHR is higher. This is possible because very high SEER equipment tends to have higher SHR under specific test conditions, as indicated in manufacturer performance data. Likewise, dehumidifier energy might be less than indicated in Figure 2-11 if SHR operates less than the simulation assumption.

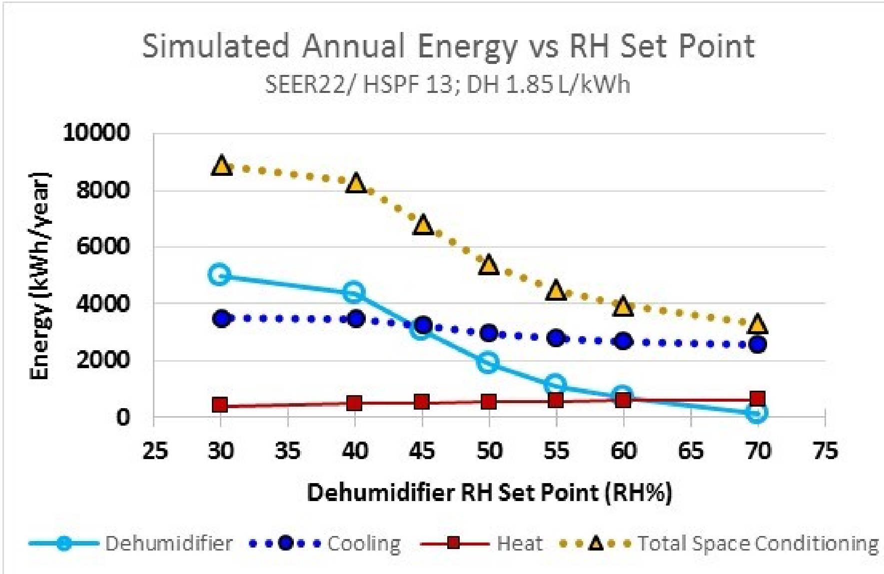


Figure 2-11. Simulated space-conditioning energy in the manufactured housing laboratory with SEER 22 heat pump and supplemental dehumidification

Note that simulation work shows that dehumidifier energy increases rapidly as the dehumidifier RH set point decreases. As shown in Figure 2-11, the dehumidifier energy increases noticeably from 60% RH to 40% RH. The dehumidifier uses more energy than the cooling at a low RH set point of 40% and represents 44% of total annual space-conditioning energy use. A more reasonable RH set point of 50% shows that the dehumidifier uses approximately 35% of the total annual space-conditioning energy. Two other key points are that heat from the dehumidifier increases air-conditioning load and that annual cooling energy increases as RH set point decreases.

Dehumidifier energy might increase enough at lower RH set points such that air-conditioning systems with lower SEER ratings that provide much better dehumidification might be capable of using the same or even less total house space-conditioning energy (heat plus cool plus dehumidifier) than air conditioners with higher SEER ratings. To demonstrate this, simulation results with a supplemental dehumidifier RH set at 50% are compared in Table 2-4. The comparison is made between a SEER 22 heat pump in standard cooling mode and a SEER 14 variable-capacity SDHV heat pump with some improvements beyond the iSeries system tested. More assumptions about each simulated system are provided next.

Table 2-4. Simulated Annual Space-Conditioning Energy in the Manufactured Housing Laboratory for SEER 22 with a Supplemental Dehumidifier and an iSeries Heat Pump without Dehumidifier

	Heating kWh	Cooling kWh	Dehumidifier kWh	Total kWh
Variable-capacity central SEER 22 Dehumidifier @ 50% RH	549	2,978	1,875	5,402
Variable-capacity central SDHV SEER 14 with excellent RH control	818	3,556	0	4,374

Specific cooling performance map data were not input into the models, so they do not account for how the SHR might vary from one minute to another. It was assumed that the variable-capacity SEER 22 system with a typical centrally ducted system would need supplemental dehumidification. An assumption was made that a SEER 14 variable-capacity SDHV would not need supplemental dehumidification to maintain 50% RH all hours of the year if the dry mode could be enhanced and enabled only during periods when RH reached 50%. Although indoor RH levels discussed in Section 2.5.5 show test results of hourly periods more than 50% RH, performance data indicate the feasibility of variable-capacity equipment being able to maintain indoor RH at approximately 50% under the lab test conditions.

When the simulation was performed for a SEER14 system and no dehumidifier, the total annual space-conditioning energy use was only 4,374 kWh. If this can produce at least 50% RH all hours of the year without supplemental dehumidification, it indicates that the lab house with SEER 22 heat pump and a supplemental dehumidifier set at 50% RH might use approximately 5,401 kWh/year. This is 1,027 kWh/year (23%) more annual energy than the assumed SEER 14 simulation.

Even if the SDHV system as tested required 1 hour of supplemental dehumidification for every hour observed that was more than 50% RH, the total space-conditioning energy would still be approximately 3% less than the simulated SEER 22 with dehumidifier at 50%. Dehumidification energy in this case was estimated based on a measured 17% frequency of SDHV indoor RH more than 50% and measured dehumidifier power of 580 W. Power at 580 W x 1,489 hours/year (0.17 x 8,760 hours) = 864 kWh/year estimated supplemental dehumidification with SDHV SEER 14 iSeries system. This 864 kWh added to the 4,374 kWh from Table 2-4 equates to 5,238 kWh/year.

Considering the measured indoor humidity measurements and the energy simulations shown in Figure 2-11, the energy use shown in Figure 2-10 should be interpreted in light of the fact that the energy consumption of the SDHV SEER 14 achieves an indoor humidity typically 10% less than the other units in the graph. When the additional supplemental dehumidification energy required for SEER 22 systems to achieve 50% RH is considered, the energy use between the iSeries 14 SEER SDHV unit and the 22 SEER ducted heat pump is negligible.

This simulation result shows merit in completing a more rigorous simulation and test work. Controlled lab experiment is needed to verify this potential. Previous MHL testing has been at only 60% RH and needs to be completed at lower RH levels to observe measured impacts because it appears that the energy use potential can greatly change.

2.5.3 Air Handling Unit Power

SDHV distribution systems operate under high static pressure by design. Under typical applications, power use increases as static pressure increases with electronically commutated motors (ECMs) so that the intended airflow is delivered. This means that under rated cooling capacity and flow rates, an SDHV design using an ECM would have a lower fan efficiency than an ECM used in a traditional lower static centrally ducted system. But does this mean that an SDHV system with an ECM blower will use more energy because it operates variably and is almost never at peak flow?

Data from the iSeries SDHV ECM were compared to an ECM used in a SEER 22 centrally ducted system previously studied by the Florida Solar Energy Center (FSEC) (Withers 2016b). Data were selected to compare periods with similar cooling loads during a typical summer day. Table 2-5 compares outdoor and indoor conditions as well as the average SAT. The average outdoor conditions are the same, and indoor temperatures are very similar. The resulting indoor RH was approximately 13% less with the iSeries under these conditions. This is largely because the SAT was approximately 9°F colder.

Table 2-6 compares measured average fan power and daily energy use along with supply plenum pressure, airflow rates, and run time hours. The average fan efficiency is also shown in Table 2-6. The highest measured fan power during the period represents the highest average for a 15-minute period. The column showing average, minimum, and maximum fan watts represent values occurring during 15-minute intervals. It is not surprising that the SDHV system would show the higher maximum. There was a surprising difference between the standby power. The indoor unit standby power (called standby) is power consumed by the indoor unit when the indoor and outdoor system is cycled off.

The measured average maximum flow rate was lower than the total maximum flow rate possible for both units shown. The SDHV maximum possible flow was 750 cfm, but the maximum flow needed was only 550 cfm, and most of the time it was only approximately 340 cfm. The maximum flow would be more likely to occur during the coldest weather in heating mode. Likewise, the SEER 22 system was capable of an airflow up to 1,088 cfm, but only 726 cfm maximum was observed. Variable-capacity systems are well suited to deliver the right amount of capacity to meet load and need to run only at lower flow most of the time. Maximum flow rates and capacities are available typically to overcome built-up load caused by a change in set point or for design load. Both systems were sized according to the closest available rated capacity; however, the top range of capacity and flow were more than enough for a design day.

Table 2-5. Environmental Conditions Shown with Air Handling Unit Power, Run Time, and Duct Pressure During Average Summer Conditions

Test Configuration	Dates	Avg. Out Temp. (°F)	Avg. SAT (°F)	In Temp. at Tstat (°F)	Daily Avg. RH
Variable-capacity SDHV SEER 14	June 2, 2016–June 4, 2016	82.1	49.7	77.7	41.2%
Variable-capacity central duct SEER 22	July 29, 2013–Aug.1, 2013	82.1	58.6	77.1	54.3%

Table 2-6. Measured Air Handling Unit Power, Energy, Supply Static Pressure, and Fan Efficiency During Average Summer Conditions

Test Configuration	Highest Meas. Watts	Standby Watts	Avg. Fan Watts (min-max)	Avg. Fan Efficiency cfm/W run time (min-max)	Whr/day Incl. Standby	Duct dP in WC (Pa)	Avg. Airflow cfm (min-max)	Run Time (h/d)
Variable-capacity SDHV SEER 14	184	8	57 (12–184)	5.51 (2.7–25.5)	1,365	0.41 (103)	340 (300–550)	18.2
Variable-capacity central duct SEER 22	160	25	73 (56–160)	9.7 (6.2–28.1)	1,760	0.08 (21)	726 (590–990)	19.6

Fan power summary:

- The daily average measured SDHV fan efficiency as cfm/W was 43% lower than the SEER 22 fan.
- However, SDHV indoor unit fan plus standby energy was 22% lower than the SEER 22.
- The lower daily total indoor unit energy consumption of the SDHV is caused by lower standby power, lower average fan wattage because of operation at much lower flow, and less daily run time.
- A very significant 68% decrease occurs in standby power for the SDHV system compared to the SEER 22 unit.
- The SDHV unit can meet load and maintain good RH control because of the very cold SAT, which was almost 9°F colder than the SEER 22.

2.5.4 Small-Duct High-Velocity Temperature and Relative Humidity Control

The SDHV system controlled indoor temperature well. Thermal distribution was evaluated based on hourly room temperature averages. The bedroom temperatures were compared to the temperature at the thermostat. Cooling set point was 77°F to maintain a house average at approximately 76°F, similar to past testing of other systems. The heating set point was approximately 72°F. The difference between each hourly bedroom and thermostat temperature was calculated and plotted for heating, standard cooling, dry mode cooling, and dry II mode cooling. The results are shown in Figure 2-12 through 2-15. ACCA Manual RS was referenced as a means to evaluate thermal distribution. This document calls for room-to-thermostat temperature differences to be no more than 3°F. It also establishes a limit of no more than a 6°F difference from one room to another.

Figure 2-12 shows that the heating temperature distribution was good. The master bedroom ran cooler than other spaces. The total period average temperature differential for three bedrooms was -0.5°F.

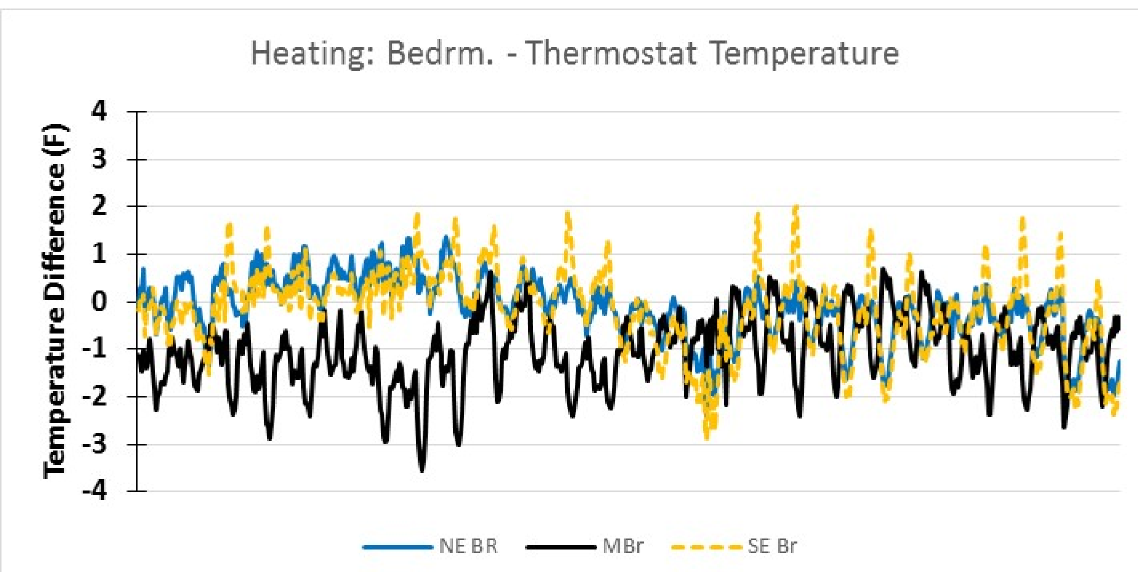


Figure 2-12. Hourly average room-to-thermostat temperature differentials during heating

In standard cooling, Figure 2-13 shows that the master bedroom ran warmer, and the other two bedrooms ran cooler than the central area. The northeast bedroom exceeded -3°F (overcooled) 2% of the test hours, and the southeast bedroom exceeded the threshold only 0.4% of the time. The events correspond to very hot periods when the master bedroom also runs warmer. The distribution could easily be adjusted by adding a damper to at least one of the smaller bedroom supplies and by adding an additional supply branch with damper to the master bedroom.

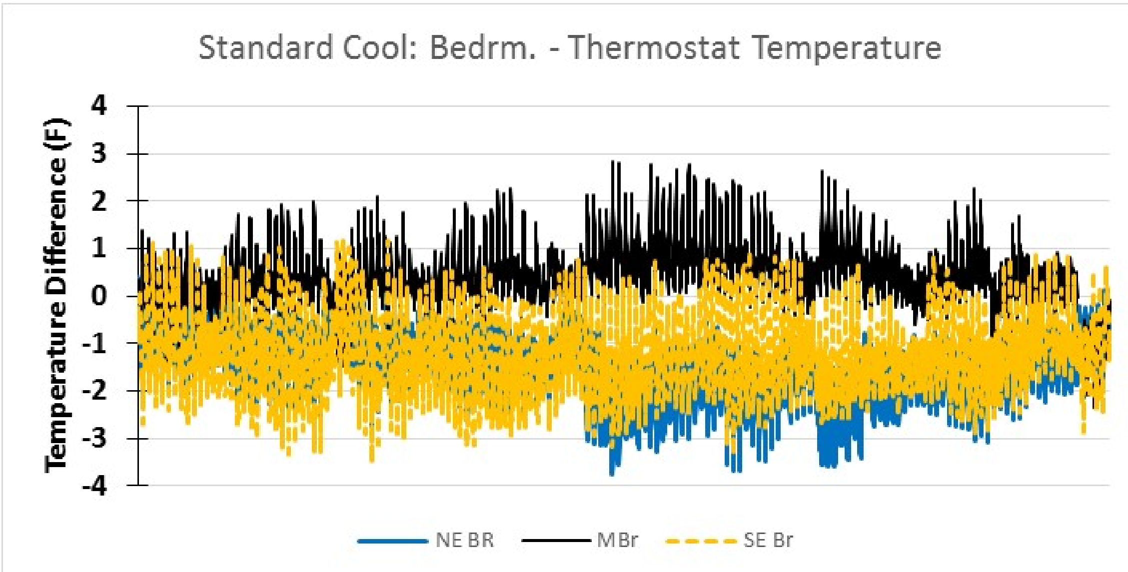


Figure 2-13. Hourly average room-to-thermostat temperature differentials during standard cooling

Dry mode cooling also resulted in good temperature control relative to the central zone. Figure 2-14 data show only that the northeast bedroom exceeded the 3°F room-to-thermostat limit 0.1% and the southeast bedroom only 0.2% of the time. Although distribution is good, the reader is reminded that this mode allowed severe overcooling of the central zone during cool weather without an occupant to turn off dry cooling. This is shown in Figure 2-15, which shows the central living room temperature. The most severe drop in temperature is at approximately 62°F, when cold weather occurred. Heating mode testing was enabled within a day.

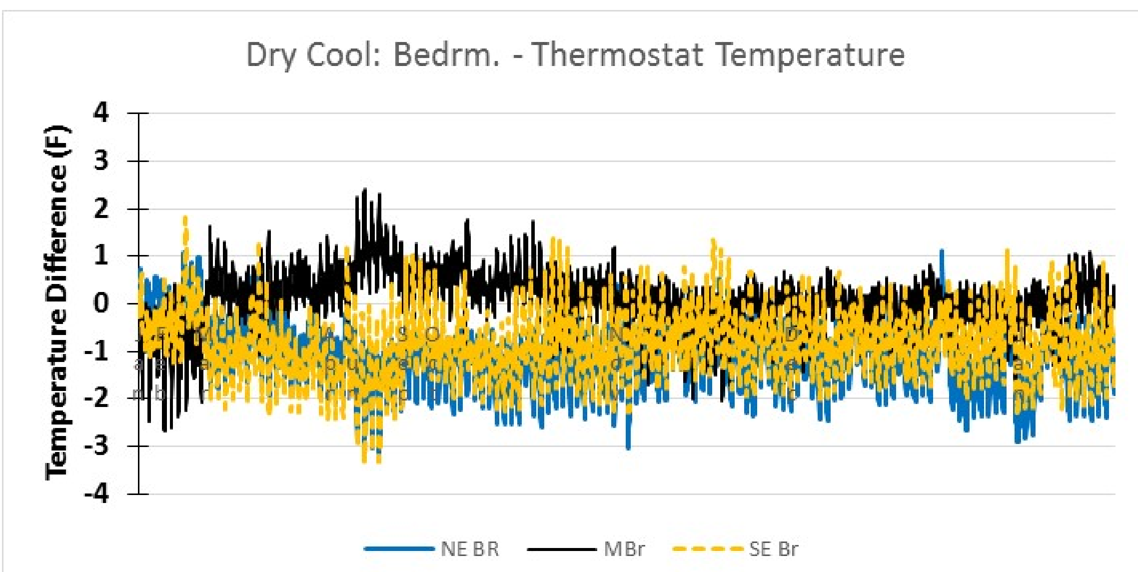


Figure 2-14. Hourly average room-to-thermostat temperature differentials during dry mode cooling

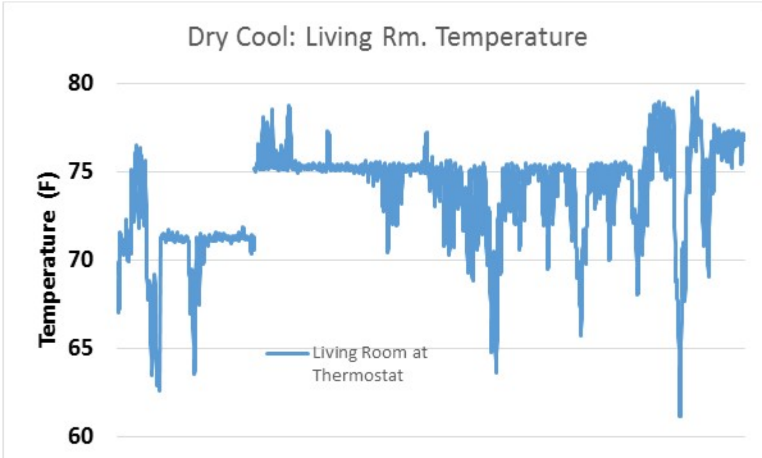


Figure 2-15. Hourly average indoor temperature during dry mode testing

In May 2017, a firmware change was made to modify the dry mode from a shutdown temperature of 50°F to a shutdown proportional to set point temperature with a fixed differential of 7.2°F (4°C). This firmware is referred to as dry II mode. This differential allowed overcooling the space by 7.2°F when in dry mode. This change was intended to stop the unnecessary energy use at colder outdoor temperatures when this mode is not likely to be needed. The authors feel that this differential is still too much to guard against potential surface moisture issues in some buildings (especially if unoccupied for extended periods) if typical cooling set points are used. Although the interior humidity is under control, poor construction practice allowing high vapor transmission into a very cold wall would result in high envelope surface humidity and possibly mold and moisture problems. Despite these concerns, no moisture issues were observed on building surfaces within the confines of the house lab test conditions.

The dry II mode also maintained good thermal distribution, as shown in Figure 2-16. The northeast bedroom exceeded the 3°F threshold 0.4% and southeast bedroom only 0.3% of the time. Figure 2-17 shows that the dry II modification eliminated the serious overcooling potential shown in the dry mode in Figure 2-15.

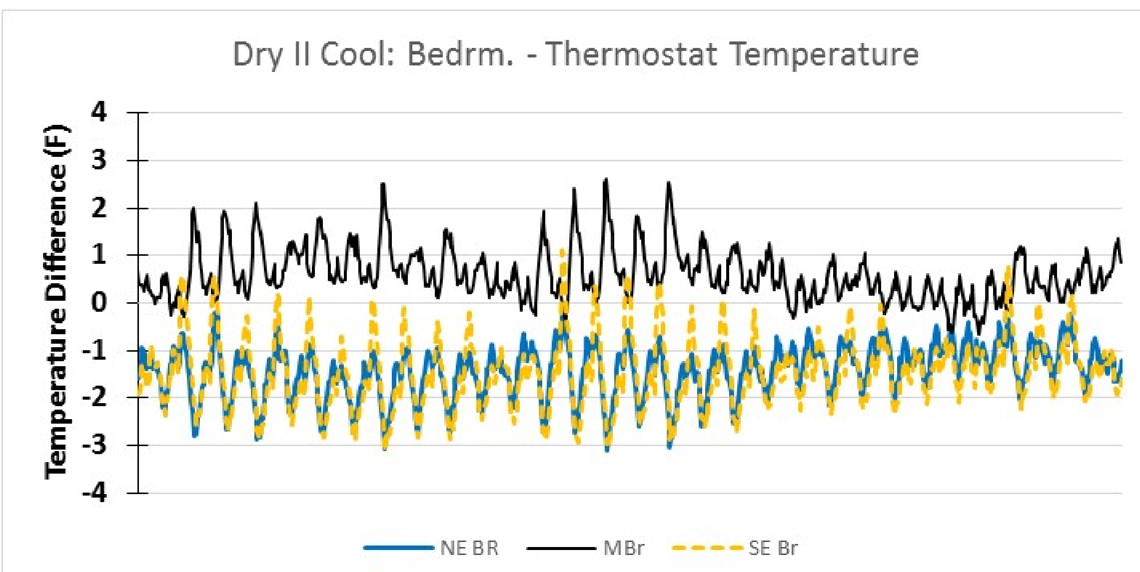


Figure 2-16. Hourly average room temperature differentials during dry II mode cooling

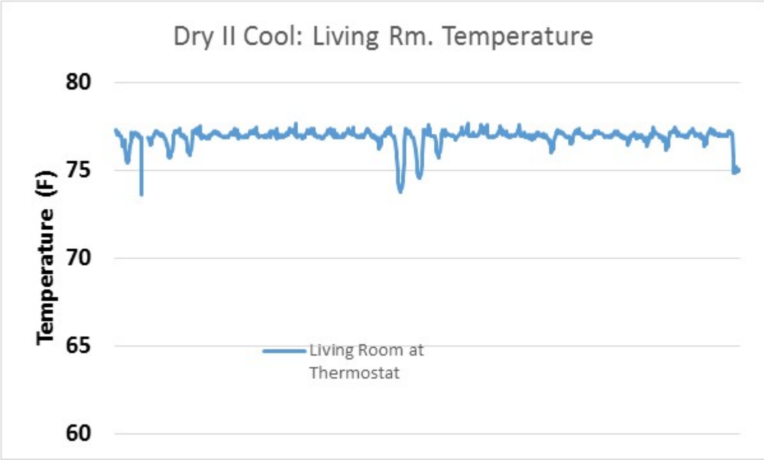


Figure 2-17. Hourly average indoor temperature during dry II mode testing

2.5.5 Small-Duct High-Velocity Relative Humidity Control

RH control was very good with the iSeries SDHV system. A small amount of supplemental dehumidification operation occurred during standard cooling mode, but none was needed during the dry cooling control modes. A summary of the indoor average humidity for four locations is shown in Table 2-7. The house average is a straight average of the four values shown, and the house area-weighted average shows a proportional adjustment for the representative zones. The only difference in these averages was in the heating mode. This is because the utility room RH was much lower during very dry outdoor conditions. The utility room showed more influence from outdoor conditions because mechanical ventilation is delivered here. The heating period represented very dry outdoor air and very limited run time with less air mixing.

Table 2-7. Average Indoor RH (%) During Three Test Periods

iSeries Test	Living Room	Master Bedroom	Hall	Utility	House Average	House RH Area Weighted	# days
Heating	41.8	44.3	42.7	31.3	41.7	40.0	26
Standard cool	44.7	44.2	46.3	44.1	44.8	44.8	153
Dry cool	48.1	48.7	49.7	47.4	48.5	48.5	133

Indoor RH was evaluated based on the frequency that the room hourly averages exceeded 60%, 70%, and 80%. No hourly average ever exceeded 85%, and only the utility room exceeded 60%. Table 2-8 shows the frequency that the utility room exceeded 60% RH during different conditioning tests.

Table 2-8. Utility Room RH Frequency at Elevated Humidity Levels During Space-Conditioning Tests

iSeries Test	Utility Avg. RH %	Frequency of Hourly Avg. RH >60%	Frequency of Hourly Avg. RH >70%	Frequency of Hourly Avg. RH >80%	# Days Evaluated
Heating	31.3	1.0 %	0.0 %	0.0 %	26
Standard cool	44.1	9.7 %	1.8 %	0.0 %	153
Dry cool	47.4	5.6 %	0.2 %	0.0 %	133

The utility room RH had higher RH levels because this is where the outdoor mechanical ventilation air was delivered. Mechanical ventilation supplied continuously at approximately 57 cfm brought in approximately 2 pounds of water every hour in the moisture-laden air from outdoors during summer conditions with an outdoor dew point temperature of 72°F. Without much moisture capacity of indoor materials and during limited cooling operation, the RH increased. The RH could drop from 65% to 55% 15 minutes after the iSeries cooling cycle began. The iSeries latent removal rate was typically between 3–5 pounds of water per hour, depending on the cooling load during summer conditions.

Although elevated humidity in the utility room might not be as big a concern as in primary habitable rooms, it does demonstrate the potential for other spaces to have elevated humidity if unconditioned supply mechanical ventilation is delivered elsewhere.

The utility room RH sensor was placed approximately 5 feet from the mechanical ventilation supply and out of the direct path of the ventilation discharge. The sensor location was in near proximity of the AHU, which would be the coldest condensing surface within the room. There was never any condensation on the AHU cabinet, supply duct, or other moisture issues on building surfaces.

Figures 2-18 and 2-19 provide a visual context for the range of indoor RH levels measured during the course of testing. These figures also include some brief interruptions to testing because of test reconfigurations and severe storm events. Figure 2-18 shows that the space humidity in the central living and master bedroom is well under 60%. Figure 2-19 shows frequent periods of elevated RH in the utility room.

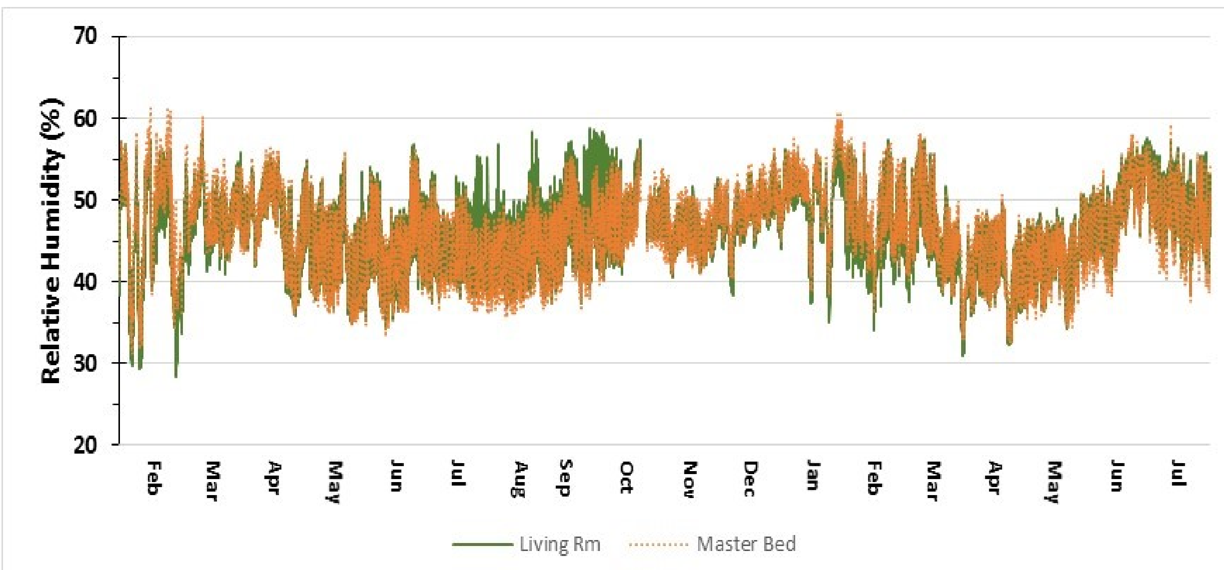


Figure 2-18. Living room and master bedroom RH percentage during all testing periods

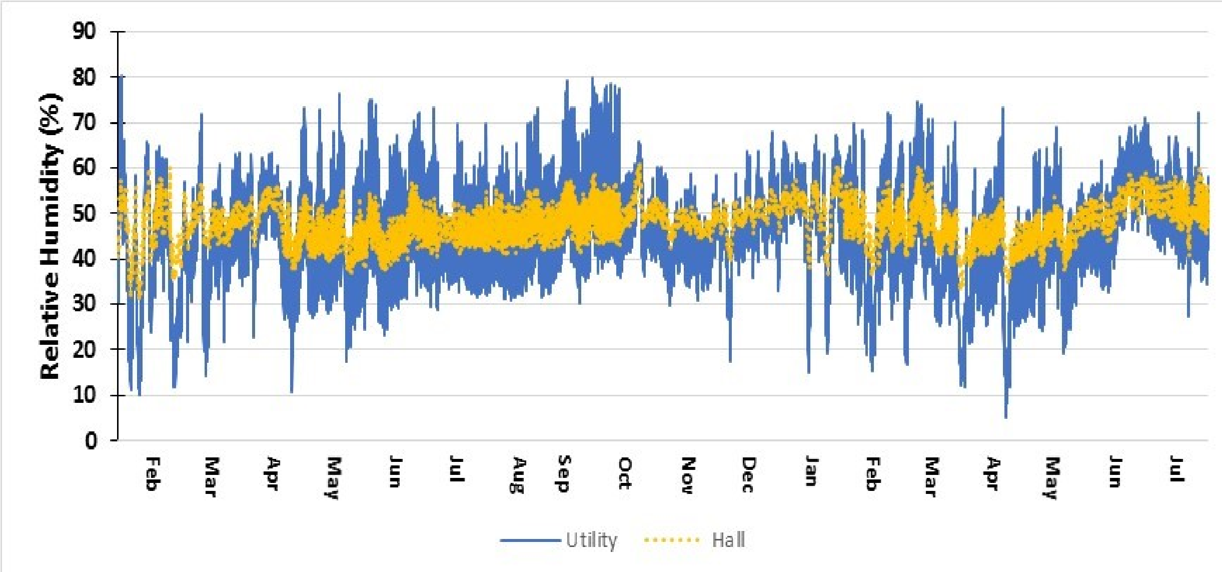


Figure 2-19. Utility room and hallway RH percentage during all testing periods

Figures 2-20 through 2-22 show a series of indoor RH plots during *standard* cooling. Figure 2-20 shows a large set of test data, Figure 2-21 shows a series of a few hot-humid days, and Figure 2-22 shows a daily profile for one hot-humid day.

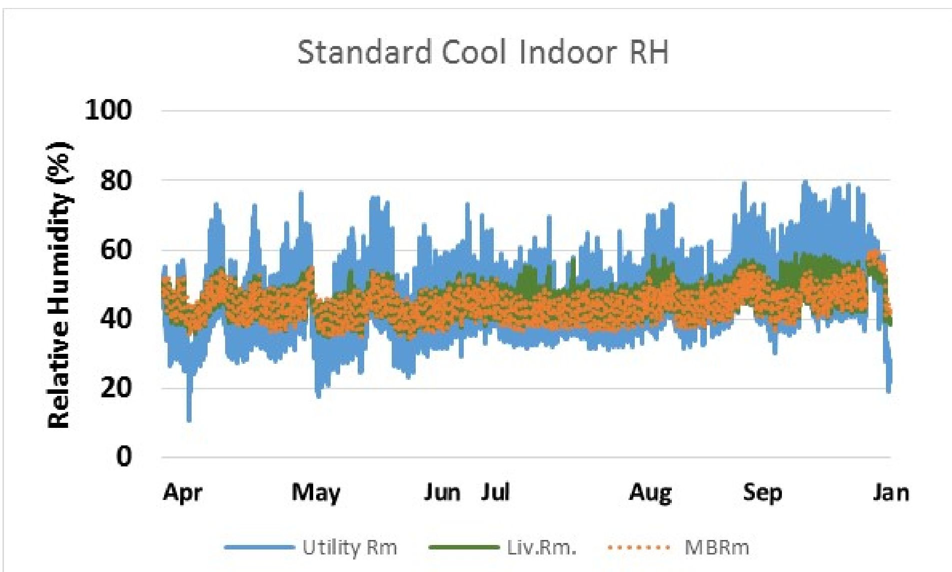


Figure 2-20. Hourly average indoor humidity during entire standard cooling testing period

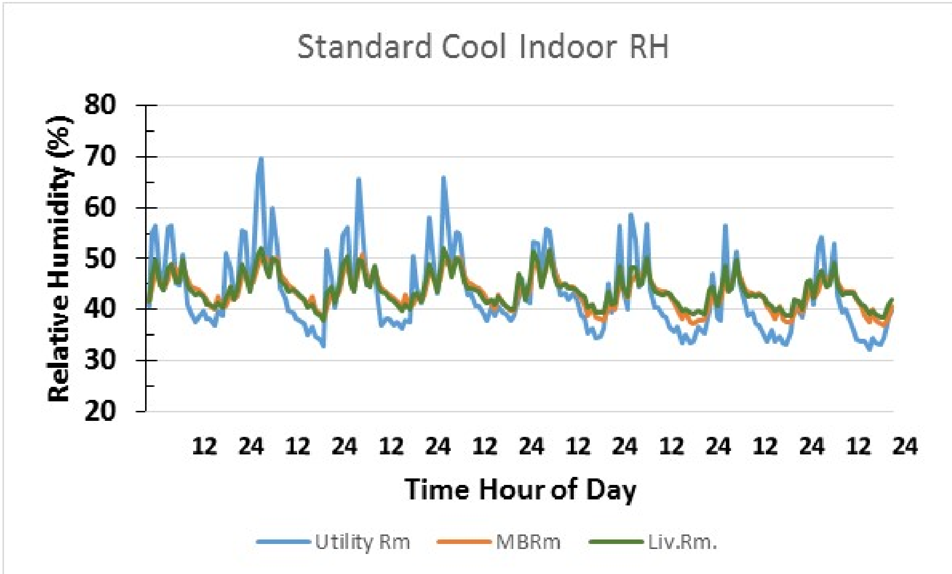


Figure 2-21. Hourly average indoor humidity during standard cooling testing period (June 30–July 7, 2016)

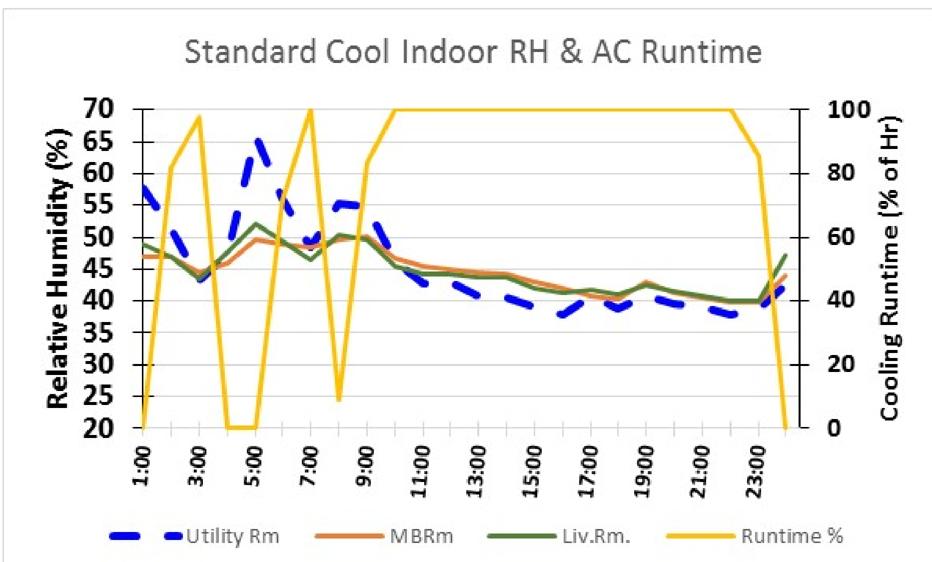


Figure 2-22. Hourly average indoor humidity and run time during a 24-hour period on July 3, 2016

Figures 2-23 through 2-25 show a series of indoor RH plots during *dry* cooling. Figure 2-23 shows a large set of test data, Figure 2-24 shows a series of a few hot-humid days, and Figure 2-25 shows a daily profile for one hot-humid day.

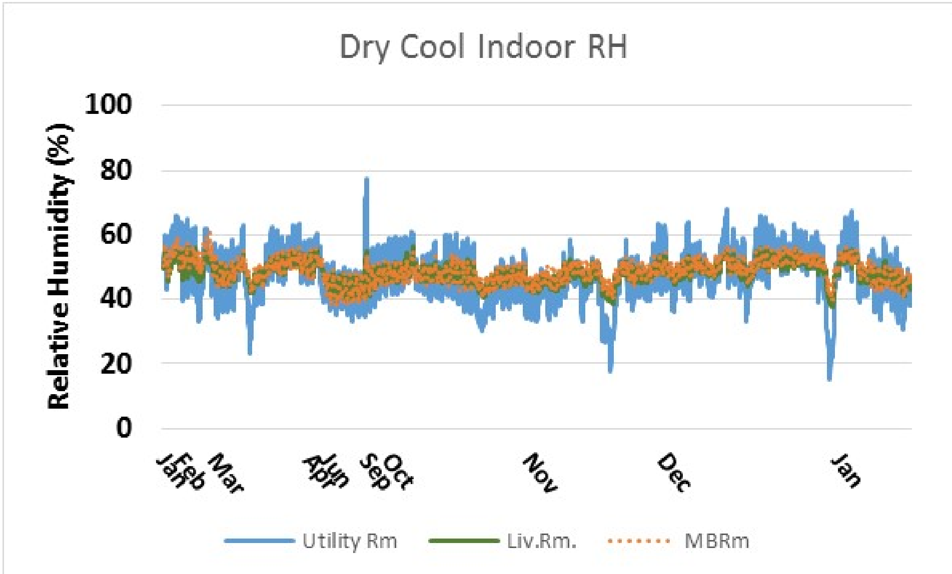


Figure 2-23. Hourly average indoor humidity during entire dry cooling testing period

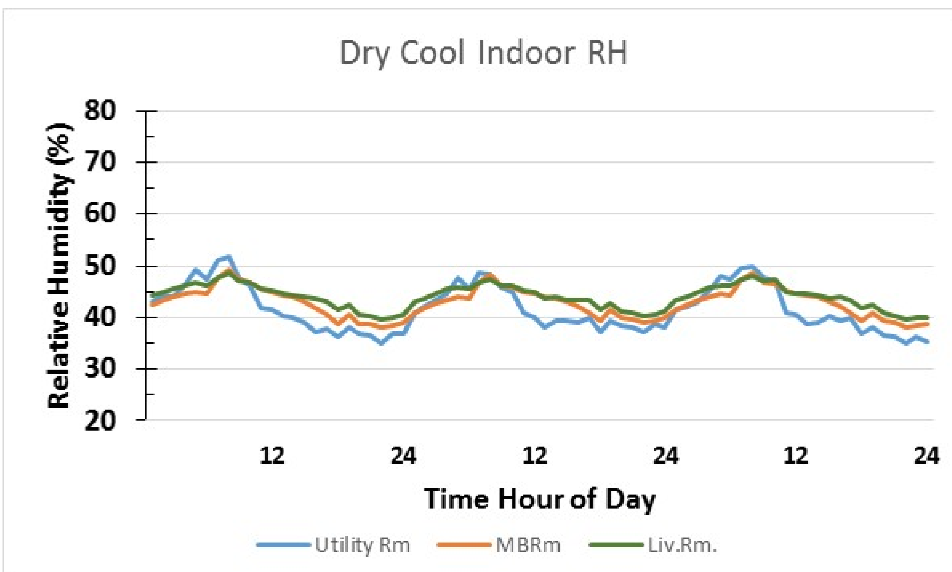


Figure 2-24. Hourly average indoor humidity during dry cooling testing period from June 23–25, 2016

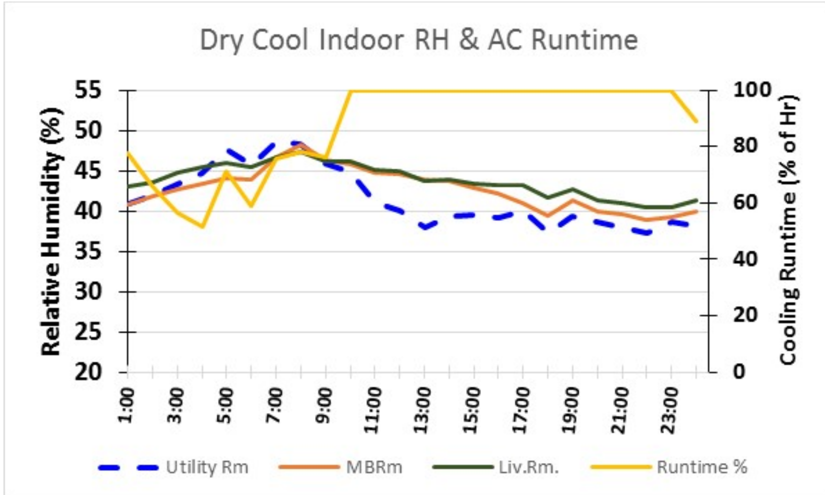


Figure 2-25. Hourly average indoor humidity and run time during a 24-hour period on June 24, 2016

Notable findings from Figures 2-18 through 2-25 show that:

- RH is controlled well with some exceptions for the utility room.
- Utility RH increases during the off-cycle of the air conditioner because mechanical ventilation steadily supplies moist outdoor air to the space (Figure 2-22). Utility room RH during the highest cooling load periods show that it becomes lower than other parts of the house lab. This is because of an increase in temperature in the utility room caused by hot mechanical ventilation air delivery.
- Utility room RH control improved significantly with dry mode operation.
- Elevated RH levels are most likely to occur during overnight hours with continuous mechanical ventilation.

Shifting to a smart ventilation or even intermittent control might help reduce indoor RH in the utility room. A short experiment was run in the MHL in which the whole-house mechanical ventilation flow rate was doubled, but the run time was cut in half. Figure 2-26 shows a 24-hour period on July13, 2017, during which the mechanical ventilation was changed from approximately 57 cfm 24 hours per day to approximately 100 cfm delivered 13 hours per day. The ventilation was provided from approximately 8:45 a.m.–9:45 p.m. when the greatest sensible loads typically occur.

These results in Figure 2-26 show that scheduling to better match mechanical ventilation to higher periods of sensible cooling load might improve moisture control; however, this alone is not enough for ideal humidity control during the most challenging low-load periods in remote spaces of the home with higher latent loads. A mechanical means of moisture removal will be needed through either improved latent air-conditioning performance or supplemental dehumidification.

Utility RH remained less than 55% RH overnight during cycled-off cooling periods with mechanical ventilation off, unlike in Figure 2-22 when it shot up to approximately 65% RH during cycled-off cooling with mechanical ventilation on. It is not guaranteed at this point to control all periods, as shown during mid-morning and later in the day when the cooling cycled off and the mechanical ventilation system was on at 100 cfm. The utility RH reached 65% in the morning and up to 74% following a heavy rainstorm. The utility room RH quickly recovered to lower levels after one cooling cycle after the mechanical ventilation was scheduled off.

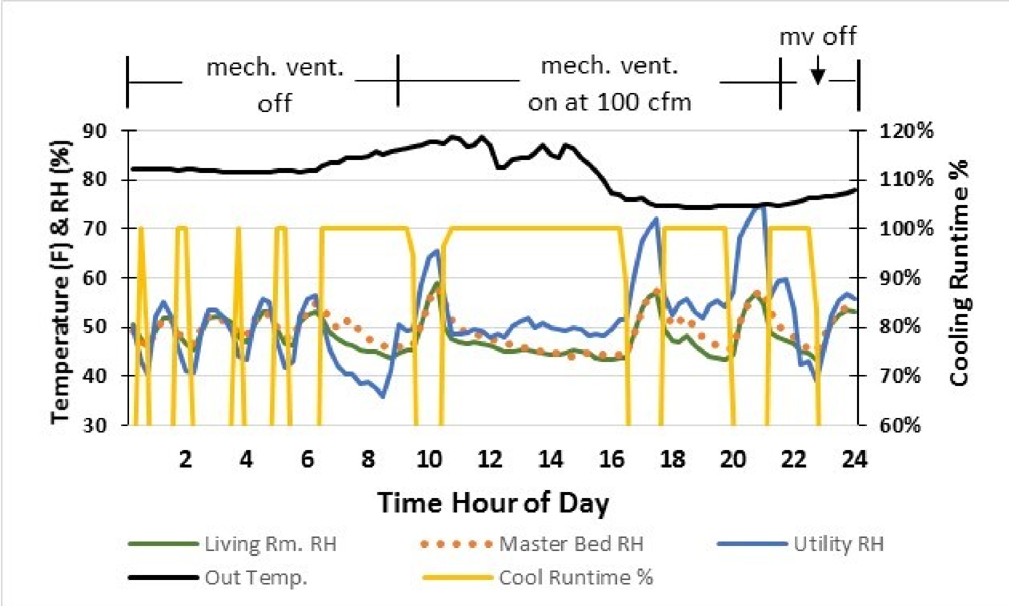


Figure 2-26. Utility room RH overnight during off cooling periods stayed less than 60% RH, but it still spiked upward later when the mechanical vent was on and cooling was off

2.5.6 Small-Duct High-Velocity Low-Load Cooling Performance

Because controlling moisture during low-load periods is the biggest challenge, cooling performance was examined during periods of low load. Periods of greatest challenge are when outdoor dew points are high (70°F or higher), but sensible load is minimized by either little or no solar load.

Three cooling modes are compared here:

- Unico iSeries SDHV standard cool during low-load periods: June 30, July 3–6, 2016
- Unico iSeries SDHV dry cool mode: June 23–25, 2016, average from 4 a.m.–8 a.m.
- Unico iSeries SDHV dry cool II June 23–25, 2017, average from 4 a.m.–8 a.m.

The outdoor dry-bulb and dew point temperatures from each period were used to select cooling data with similar outdoor conditions.

Table 2-9 shows average indoor and outdoor conditions. It includes indoor temperature at the thermostat (In T stat), indoor RH near thermostat (In RH), indoor dew point temperature (In T dp), indoor wet-bulb temperature at the return (Ret Twb), outdoor temperature (Out T), and outdoor dew point temperature (out T dp). Table 2-10 shows cooling performance data. These data include SAT, distribution airflow rate (cfm), delivered cooling from indoor coil (BTUH), distribution airflow rate per cooling ton delivered (cfm/ton), average cooling power (W), and an average dehumidification efficiency in liters of moisture removed per kWh of energy consumed. The dehumidification efficiency cannot be compared directly to actual dehumidifier efficiencies because this has not been tested under the controlled rating conditions of 80°F and 60% RH. The conditions are much cooler and drier, and less moisture is removed at these conditions compared to dehumidifier rated conditions.

Table 2-9. Environmental Conditions During Low Cooling Load Periods

	In T stat	In RH	In T dp	Ret Twb	Out T	Out T dp
Standard	76.9	44.4	55.0	63.3	78.5	72.9
Dry	75.2	45.6	54.0	62.4	78.4	73.4
Dry II	74.8	52.0	56.4	63.6	78.8	72.6

Table 2-10. Cooling Performance Measures During Low Cooling Load Periods

	SAT	cfm	BTUH	cfm/ton	Watts	Dehumidifier Efficiency[a] l/kWh
=Standard	50.4	322	13718	284	1050	1.75
Dry	51.8	381	13560	338	846	1.53
Dry II	53.7	394	13549	352	752	2.00

[a] *Does not represent a rated dehumidifier efficiency under rated conditions*

Figures 2-27 through 2-29 show cooling performance characteristics for three cooling test modes. These data are the same as those reflected in Table 2-10. The standard mode (Figure 2-27) shows periods of no cooling during overnight periods. This is when interior RH begins to increase, especially in the utility room, where mechanical ventilation was delivered. It also shows that other than at start-up or cycle-down periods, there was not much variation in delivered cooling rate. During low-load overnight periods, the dry and dry II modes (Figure 2-28 and Figure 2-29) show more variation in cfm/ton and warmer SAT than the standard mode.

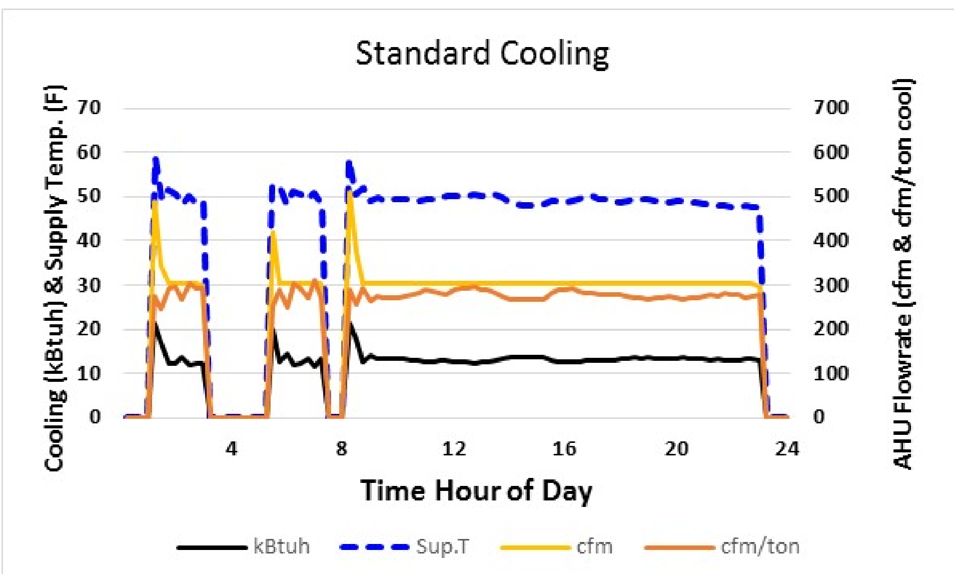


Figure 2-27. Daily cooling, supply temperature, airflow, and flow rate per delivered cooling ton

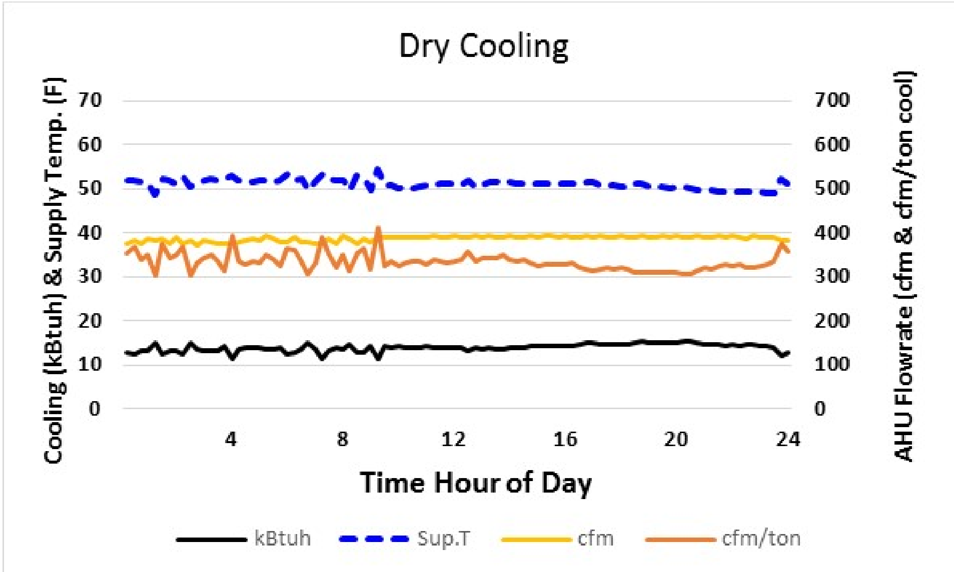


Figure 2-28. Daily cooling, supply temperature, airflow, and flow rate per delivered cooling ton

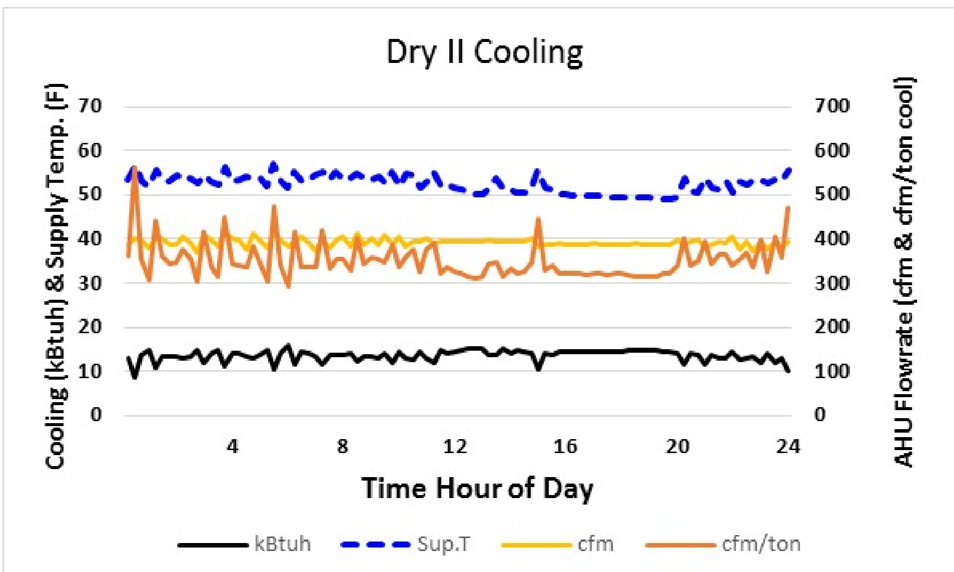


Figure 2-29. Daily cooling, supply temperature, airflow, and flow rate per delivered cooling ton

The iSeries unit was promoted as being able to deliver at as little as 10% of its rated capacity. This would be approximately 2,900 Btuh for our tested unit and was considered to be well suited for the low-load period. The actual measured rates of cooling output during 15-minute periods turned out to be approximately 12,000 Btuh, or three times the lowest expected rate. It was learned that the lowest capacity is derived by running at a fixed low capacity for a fraction of the hour. The fractional run time is reserved for the dry cooling mode previously described when there are 3 minutes of cooling followed by 9 minutes off. This results in 15 minutes of run time per hour, or 25% of run time. The 25% run time at a cooling rate of 12,000 Btuh results in an average cooling rate of 3,000 Btuh, the stated 10% lowest capacity.

During higher cooling loads, air-conditioning runs at a higher capacity for longer periods and removes more moisture from the air. On a very hot and humid day, 105 pounds (12.6 gallons) of condensate was measured. Condensate removal correlated very well with outdoor temperature, as shown in Figure 2-30.

What happens if there are high latent loads when it is cooler outside? The significance is that a high-performance home with low sensible loads and high internal latent loads has much less potential to remove moisture using air conditioning. This makes it more important to be able to lower the SHR, thereby shifting more cooling from sensible to latent. Fortunately, variable-capacity systems have the potential to do this.

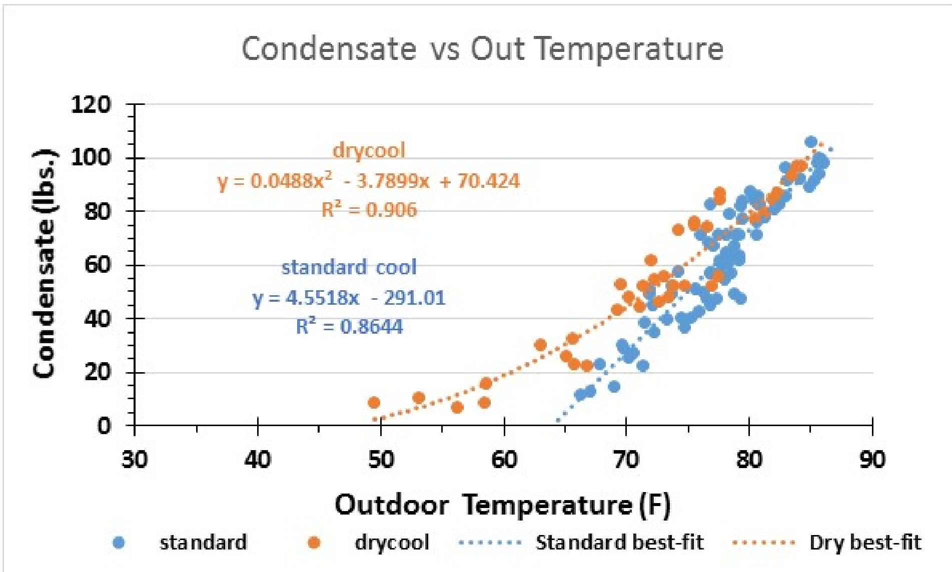


Figure 2-30. Measured daily condensate removed from indoors compared to the daily average outdoor temperature

Based on an analysis of cooling dehumidification performance during low-load periods, the following can be said:

- Standard cooling dehumidification performance was better than the dry mode during warm conditions. This was caused by a lower flow rate per ton of cooling (cfm/ton), which resulted in colder coil and colder SAT.
- The dry cooling mode enabled more dehumidification during very low-load periods.
- The firmware modification from dry to dry II was not able to reduce the cfm/ton and did not result in better dehumidification.
- Dry II resulted in 15% less run time and a 6% RH increase in humidity from 46% to 52% compared to the dry mode.
- The dehumidification performance of dry II worsened because of an unintended 30-second fan run-on period when the fan flow rate ramped up to high speed each time after the compressor cycled off. This increased energy use and drove some moisture off the coil back into the air. The issue of decreased dehumidification performance related to airflow over a wet, warm coil driving moisture off the coil has been understood for years. This feature was accidental in the firmware update, and it should be able to be corrected.

2.6 The Low-Airflow Target Challenge

We worked with the manufacturer to optimize their standard operating algorithms in dry mode (DM), targeting an SHR of 0.55 to increase moisture removal and limit the amount of sensible overcooling. Our approach was to reduce the standard airflow in dry mode, thereby limiting sensible cooling, increasing system run time, and increasing dehumidification. Unfortunately, we were unable to achieve the low target airflows required to realize these improvements during our testing.

The system under test offered both normal cooling mode operation and a dry/dehumidify mode with three progressive stages of dehumidification. In the first stage (DM0), the system is designed to operate identically to normal cooling mode. In the second stage (DM1), the cooling capacity of the system is fixed, and airflow is cycled between low (400 cfm for this system) and very low (160 cfm) every 30 seconds. In Stage 3 (DM2), the airflow is set to very low, and the system operates in a 15-minute cycle, with 3 minutes on, 9 minutes off, and it is allowed to overcool until the ambient temperature is 7.2°F (4°C) lower than the thermostat set point.

Our results showed that neither DM2 nor DM3 achieved their airflow targets. For DM2, our expected 15-minute average airflow was 280 cfm; however, our measured average airflow was approximately 385 cfm. We estimate the uncertainty in our flow measurement to be approximately +/-15% of flow. Our low-flow reading was in reasonable agreement with the expected 400 cfm; however, our very low-flow reading was twice as high as expected. We learned that the typical ramping time of the motor between the present value and set point was 60 seconds, so it was likely that the higher average measured airflow resulted from the inability of the motor to cycle between the two airflows during the targeted 30-second period.

Similarly, in DM3, the system did not ramp down to the low-flow target as expected. Even after the very low flow was decreased from 160 cfm to 100 cfm in the dry II firmware change, the target was not reached. After discussing this with the manufacturer, we discovered that the standard operating range for the indoor unit in our study was 400–900 cfm and that the algorithms that control airflow accuracy deteriorate as the commanded airflow drifts from this range. This is likely the source of the discrepancy between the commanded and measured airflows. Without this very low airflow control, we were unable to achieve the lowest target SHR.

3 Field Investigation of Minisplit and Multisplit Systems

For this study, FSEC partnered with two affiliates (local chapters) of Habitat for Humanity International that had recently switched from fixed-capacity, centrally ducted heat pumps to minisplit and multisplit heat pumps. A total of five newly constructed and newly occupied homes were monitored during 2016–2017 to collect data on heat pump energy use, run time characteristics, and indoor environmental conditions. The research questions posed in the introduction of this report that apply to the field investigation are:

1) How well is indoor temperature controlled with transfer fans compared to fully ducted systems?

2) Can the achievement of design mechanical ventilation rates be ensured when integrating supply ventilation with a variable-capacity minisplit system?

3) What variable-capacity cooling system operational characteristics and patterns are observed in the collected data that might assist manufacturers with improved indoor RH control as they refine existing equipment and develop new products?

In the late 1990s, Habitat for Humanity International established ENERGY STAR standards for homes as a best practice for the more than 2,600 affiliates in the United States. Building to high-performance standards supports Habitat for Humanity's mission to build homes that are affordable to operate and easy to maintain, last for generations, and provide a healthy indoor environment for Habitat homebuyers. This advancement in standard practice significantly increases the number of domestic Habitat for Humanity affiliates facing the challenges of conditioning low-load homes.

For this field study, FSEC partnered with two Florida affiliates: Southeast Volusia Habitat for Humanity (SEVHFH) in the Daytona Beach area on Florida's central-east coast and South Sarasota Habitat for Humanity (SSHFH) between Tampa and Ft. Myers on the central-west coast of the Florida peninsula (Figure 3-1). Both affiliates have constructed homes to the U.S. Department of Energy (DOE) Zero Energy Ready Home standard, and both have won Housing Innovation Awards for their efforts.

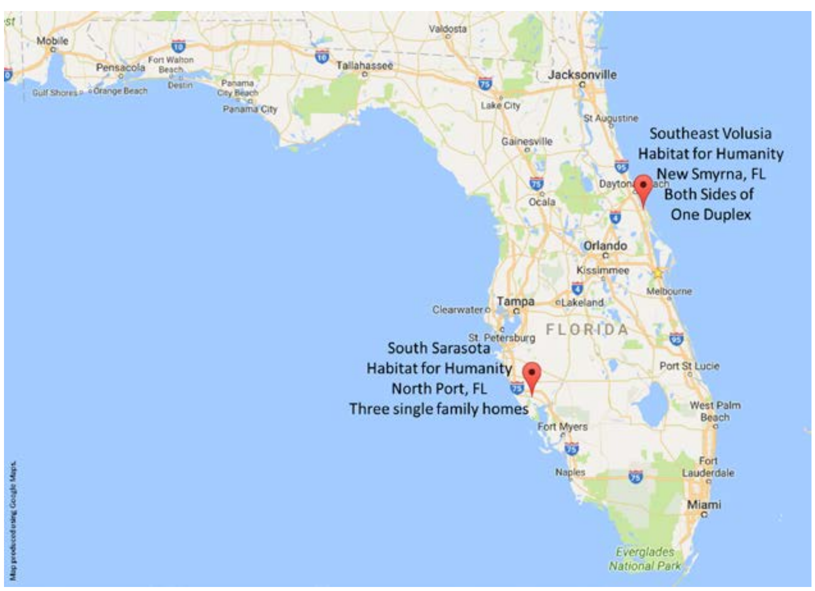


Figure 3-1. Location of low-load field study homes

SEVHFH provided both sides of a new high-performance, single-story duplex (SEV1 and SEV2) for the study with a Home Energy Rating System Index rating of 48 located in New Smyrna, Florida, on the east-central coast. SSHFH provided three single-family detached houses of the same design and construction for the study with a Home Energy Rating System Index of 51 located in North Port, Florida, on the west-central coast. The characteristics of the two sets of homes are described in Table 3-1.

Table 3-1. Characteristics of Low-Load Field Study Homes

	SEVHFH New Smyrna, Florida (SEV1 and SEV2)	SSHFH North Port, Florida (SS1, SS2, and SS3)
Home Energy Rating System Index	48	51 Certified DOE Zero Energy Ready; DOE Housing Innovation Award
Type	Both units of a single-story duplex	3 single-story, single-family detached homes
Bedrooms	2 bedrooms, 2 baths	3 bedrooms, 2 baths
Area	1,075 ft^2 per unit	1,290 ft^2
Walls	2 x 4 frame	Concrete block
Roof/ceiling	Conventional vented	Unvented, spray foam
ACH50	2.68 SEV2	3.0 average
Qn,out	NA	0.013 average
Heating and cooling system	Ductless, variable-capacity 2-head, multisplit heat pump Both heads installed in the main body	Centrally ducted, variable-capacity, minisplit heat pump
Cooling efficiency	SEER 22	15.5
Heating efficiency	HSPF 10.5	10
Air distribution	2 Panasonic Whisper Green exhaust fans installed to move air from main body to bedrooms	Fully ducted supply and return
Heating peak load	12,497 Btuh (approximately 1 ton)	17,887 Btuh (approximately 1.5 ton)
Cooling peak load	11,363 Btuh (approximately 1 ton)	12,242 Btuh (approximately 1 ton)
Whole House Mechanical Ventilation	1 Panasonic energy recovery ventilator in the main body	Run time ventilation through ducted, filtered, dampered outside air duct, exhaust fan operation by occupant and, if needed, AirCycler g2-k controller
Target ASHRAE 62.2-2010 Continuous Ventilation Rate	33	43

3.1 Southeast Volusia Habitat for Humanity Test Home Design and Mechanical Systems

The two test homes provided by the Southeast Volusia Habitat for Humanity (SEVHFH) are the right and left unit of a single duplex, referred to for the remainder of the report as SEV1 and SEV2, respectively (Figure 3-2). The two-bedroom, one-bath, one-car garage units are 1,020-ft^2. The floor plan is mirrored in the two units of the duplex.

Figure 3-2. Southeast Volusia Habitat for Humanity test houses, left and right side of duplex

In each unit, the builder installed Panasonic's Total Home Comfort Solution, which combines three Panasonic products: ductless, inverter-driven multisplit heat pump, energy recovery ventilator, and two exhaust fans installed to transfer air from the main body to each bedroom. Panasonic provided input on the design and selection of the installed equipment, as detailed in Table 3-2.

In each of the SEVHFH units, there are two 9-kBtu Panasonic variable-speed indoor AHUs (also referred to as heads and fan coils) connected to a single 19-kBtu/h variable-capacity compressor. Both air handlers are located in the main body, one facing the dining room and kitchen and the other facing the living room and bedrooms (Figure 3-3). Florida code requires builders to install passive return-air pathways from bedrooms not served by a ducted return, and the Florida energy code for new, low-rise residential construction provides sizing guidelines. In this case, the builder increased the size (register area) of the return-air path to ensure adequate return airflow in anticipation of lower pressure differences when the air handlers are running at lower speeds. Figure 3-4 shows the dining room AHU and the through-the-wall passive return-air pathway from Bedroom 1. Closing the front and back bedroom doors during heat pump operation produced a negligible change in whole-house pressure with respect to outside. Transfer fan flow was measured at approximately 80% of rated flow with the bedroom doors closed, and discussed in more detail in the next section.

Table 3-2. Southeast Volusia Habitat for Humanity Mechanical Equipment

Component	Manufacturer	Model
Compressor	Panasonic	CU-3E19RBU
AHU (2) with built in thermostat	Panasonic	CSE9RKUAW
Transfer Fans (Whisper Green exhaust fans)	Panasonic	FV-05-11VK1
Energy Recovery Ventilator	Panasonic	FV-04VE1

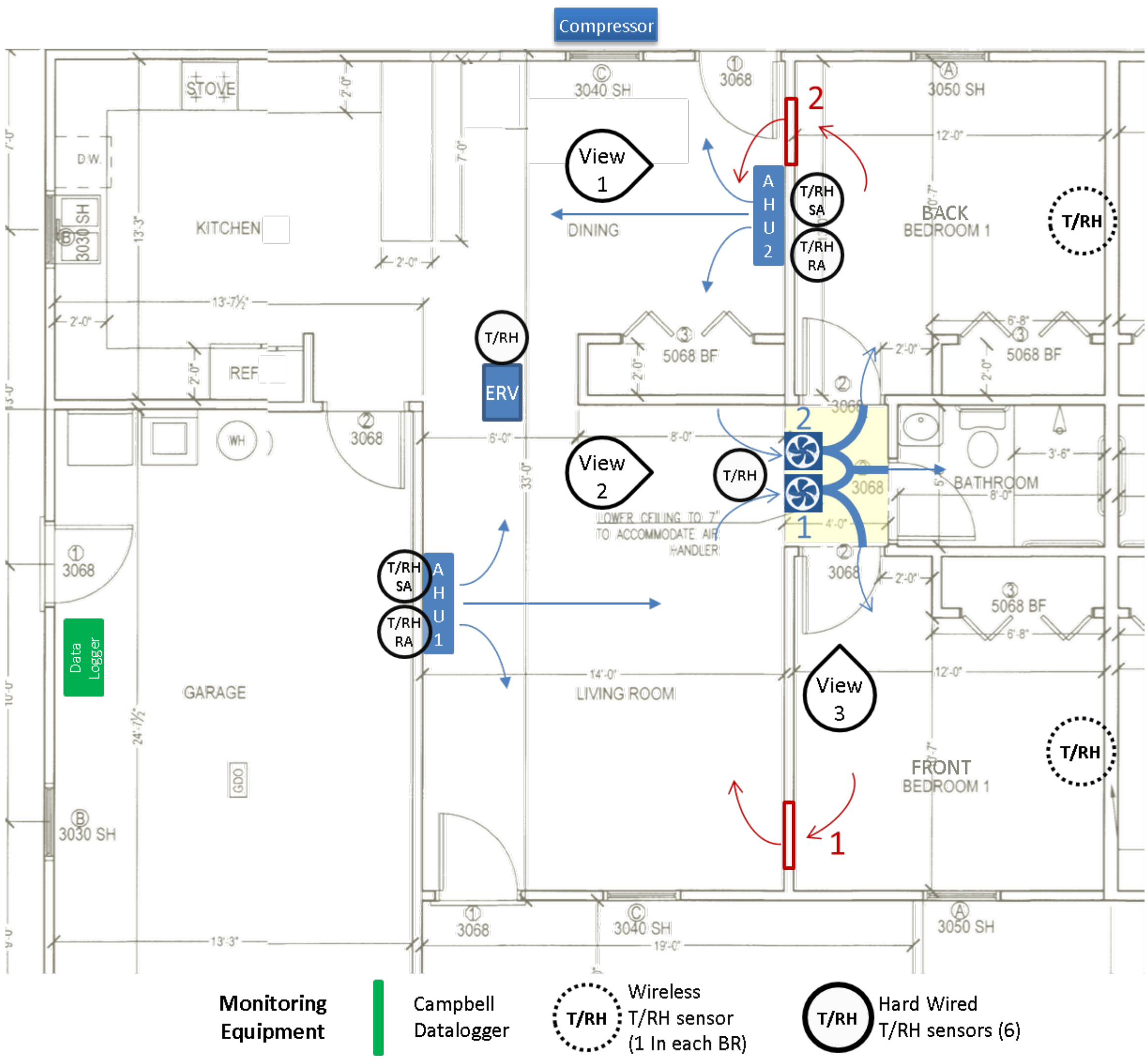


Figure 3-3. Southeast Volusia Habitat for Humanity test house floor plan, left side of the duplex. Elements of the HVAC system are shown in blue (supply) and red (return).

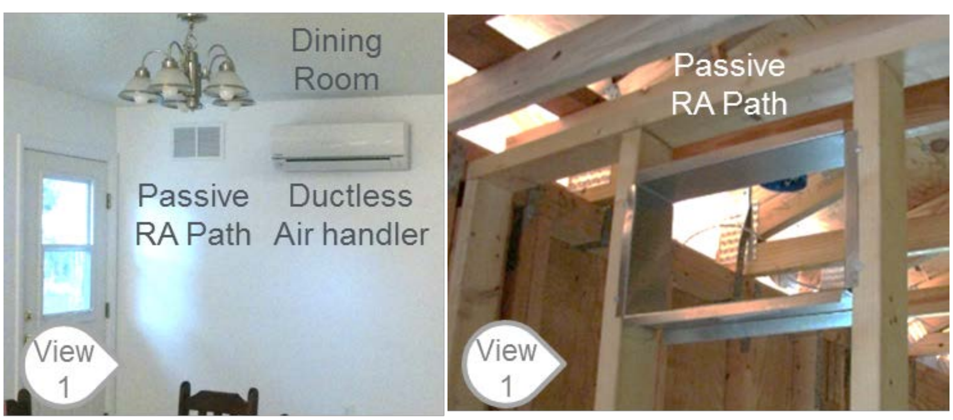


Figure 3-4. Southeast Volusia Habitat for Humanity home: dining room air handling unit and Bedroom 1 passive return-air pathway (left), passive return during construction (right)

To force air exchange between the main body of the home and the bedrooms, the Panasonic Total Home Comfort Solution includes Panasonic 110-cfm Whisper Green exhaust fans as circulation or transfer fans. In this application they are used to move air from spaces with indoor fan coils to those without. In the Southeast Volusia homes, the fans move air from the main body, through a shared filter-back, return-air grille and a compact interior duct system, into each bedroom and the bathroom (see plan in Figure 3-3 and Figure 3-5, left). Moving air into the bathroom was not a Panasonic recommendation; rather, it was a detail the builder added to improve conditioning in the bathroom. There is no intentional return-air transfer path from the bathroom when the door is closed.

Panasonic Whisper Green fans are rated for constant duty, and in this design they are intended to operate continuously. Fan speed can be adjusted manually using an internal switch. At the time of installation, the fans were set to operate at 110 cfm. The builder constructed an interior duct cavity in the hallway between the bedrooms (Figure 3-5, right). Air pulled from the main body by the transfer fan flows through standard supply registers into each bedroom (Figure 3-6).

Figure 3-5. Southeast Volusia Habitat for Humanity home: transfer fans installed in the hall ceiling behind a shared filter-back return-air grille (left) in a small interior duct chase (right)

Figure 3-6. Southeast Volusia Habitat for Humanity home: transfer fan interior duct work construction

A ducted Panasonic energy-recovery ventilator (ERV) in the living room (Figures 3-7 A and B) of each side of the SEVHFH duplex provides whole-house mechanical ventilation. The insulated flex duct work for the ERV is located in the attic (Figure 3-7 B and C) and terminates in registers mounted in the rear soffits (Figure 3-7 C and D). The Panasonic ERV has off, low, and high operating modes controlled by a wall switch. On low, the ERV rated flow is 20 cfm for both the exhaust and supply streams; on high speed, the rated flow is 40 cfm exhaust and 30 cfm supply.

Figure 3-7. Ceiling-mounted energy recovery ventilator (A,B) installed in the main body with attic-mounted supply and exhaust ducts (B,C) terminating in soffit-mounted registers (D)

SEVHFH's standard SEER 16 split-system typically costs $4,700. The installed ductless Panasonic multisplit was an additional $313 including $240 for transfer fans. The HVAC contractor provided the cost breakdown in Table 3-3. For comparison, Greater Nashville Habitat for Humanity installed this system with only a slight difference in the AHU specifications (one 12-KBtu unit and one 9-kBtu unit) including the transfer fans. The AHUs were installed in exterior walls with brick detailing around each opening that added to the rough-in cost. Also, the transfer fan ducting was installed in the attic. The Nashville cost was in line with the SEVHFH first cost, as shown in Table 3-3.

Table 3-3. First-Cost Comparison

	SEVHFH SEER 16.5 Centrally Ducted, Fixed-Capacity, Lennox Split-System Heat Pump	SEVHFH SEER 22 Unducted, Variable-Speed, Panasonic Multisplit Heat Pump with Transfer Fans	Nashville HFH SEER 16 Centrally Ducted, Fixed-Capacity, Lennox Split System	Nashville HFH SEER 18 Unducted Variable-Speed Panasonic Multisplit Heat Pump with Transfer Fans
Rough-in	$1,700[1]	$1,500		Combined $5,223
Equipment and installation	$3,000	$3,273		
Transfer fans		$240		$240
Total for installation	$4,700	$5,013	$7,000[a]	$5,463

[a]Includes ductwork

3.2 Southeast Volusia Habitat for Humanity Monitoring Strategy

The same instrumentation package and measurement protocol is deployed in both units (Table 3-4). Real-time data acquisition is accomplished with a Campbell Scientific CR1000 data logger operating at a 10-second scan rate and storing 15-minute data. Data are transferred via broadband Internet and stored by servers on a daily basis. Energy is summed and recorded each 15-minute period for the ductless multisplit heat pump, transfer fans, and ERV. Total multisplit heat pump energy use is recorded at the electrical breaker panel, which, for each unit, includes the indoor and outdoor units. An AC current sensor is also located at the power supply to each indoor head to qualitatively determine the difference in run time between the two heads.

Ten-second scans of temperature and RH are averaged every 15 minutes at the supply and return airstream of each indoor head as well as at the transfer fan intake and at the outdoor airstream near the entry point to the ERV unit. Bedroom temperature and RH are recorded by Pointsix WiFi sensors every 15 minutes from a location 5 feet above grade on the wall in the corner of each room by the closet.

Table 3-4. Southeast Volusia Habitat for Humanity Monitoring and Test Equipment

Measurement	Equipment	Accuracy
Data acquisition	Campbell Scientific CR1000	0.06%
Pressure differentials (fan flow sensors, air distribution pressures)	DG700 digital pressure gauge	1%
Transfer fan ducted airflow and whole-house ventilation flow	Flowblaster capture hood attachment to duct blaster with DG700	5%
Temperature and RH at AHU and transfer fans	Vaisala HMP60	±0.5°C, ±3% RH
Bedroom temperature and RH	Pointsix 3008-04-V6 WiFi transmitter	±0.4°C, ±3% RH
Energy (minisplit AC, transfer fan, ERV) and indoor fan coil unit operation	Continental Control Systems Wattnode energy meters and current transformers	±1% of rated current
AC current	Acuamp ACTR Series AC current transducer	1% FS
Building envelope air leakage	Minneapolis Blower Door System with DG-700 digital gauge	3%

3.3 Southeast Volusia Habitat for Humanity Results and Discussion

3.3.1 Outdoor Conditions and Monitoring Period

Data collection for SEV1 and SEV2 began shortly after initial occupancy for each unit, in May 2016 and June 2016, respectively, and continued through June 2017. Figure 3-8 shows outdoor temperature and dew point during that period from the closest weather station, Daytona Beach International Airport (KDAB NWS), located 14 miles from the SEV1 and SEV2 project site. Evident in the data, and typical for central Florida, is a peak cooling period with consistently high outdoor temperature and dew point from July 2016–September 2016, followed by a transitional period of steadily decreasing outdoor dew point from September 2016–October 2016. November 2016–April 2017 is characterized by variable conditions, yet still with appreciable cooling load throughout the period. The need for occasional heating is also apparent, but actual heating use varies between the homes because of occupant preferences.

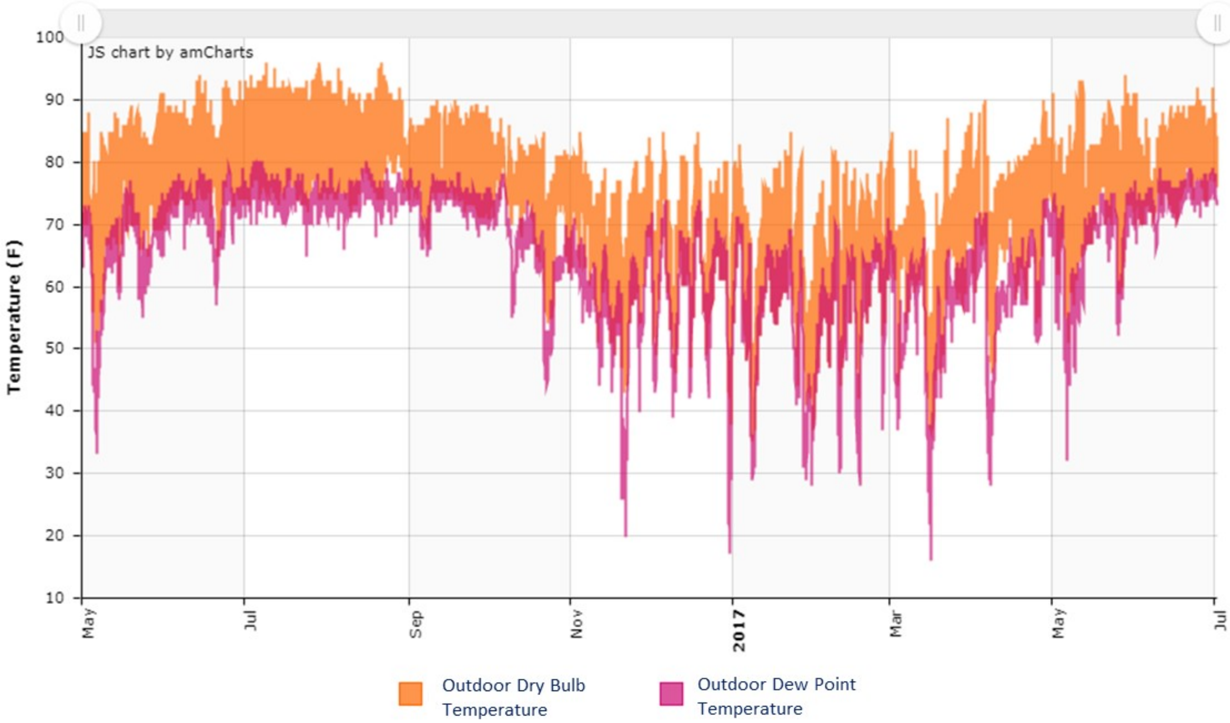


Figure 3-8. Outdoor temperature and dew point from Daytona Beach International Airport National Weather Service Station near the Southeast Volusia Habitat for Humanity Homes

Orientation and layout of the SEVHFH duplex results in very similar solar exposure for both units, with equal southwest- and northeast-facing exterior wall and window areas. Each bedroom in the two units is limited to one 12-ft x 8-ft exterior wall with one 3-ft x 5-ft window. The front bedroom window of each dwelling faces southwest, and the back bedroom of each faces northeast. Both sides are heavily shaded by a mature tree canopy.

3.3.2 Southeast Volusia Habitat for Humanity Mechanical Ventilation Operation

ASHRAE 62.2-2010[1] recommends 33 cfm of continuous ventilation for the homes. With the ERV on “low”, FSEC measured actual flow of 22 cfm exhaust, 27 cfm supply. With the ERV set on high, researchers measured actual flow of 34 cfm exhaust, 37 cfm supply. Although the design intent is continuous operation, the SEVHFH homeowners used the ERV’s differently, and monitored power data are shown in Figure 3-9. SEV1 rarely interacted with the ERV and reported leaving the unit on the low setting. The owner of SEV2 also reported primarily using the low-flow setting, but switched the unit from low to high occasionally and turned the unit off entirely to better control comfort, primarily in the colder months during the period of November 2016–February 2017. Table 3-5 summarizes mechanical ventilation system flow by month, as a percentage of the design target, based on initial flow testing and continuous power monitoring.

[1] ASHRAE 62.2-2010 was used because these homes were designed to align with Energy Star and Zero Energy Ready Home standards.

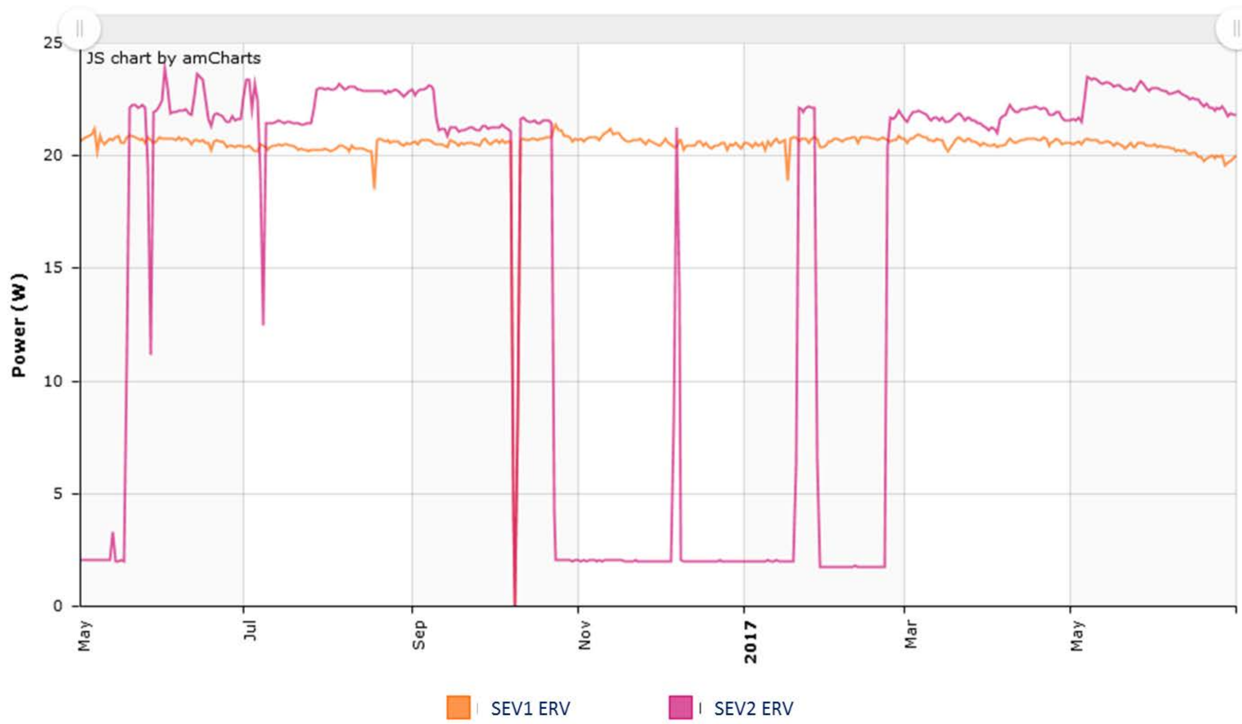


Figure 3-9. Energy recovery ventilator power for SEV1 and SEV2

Table 3-5. Percentage of ASHRAE 62.2-2010 by Month

Month	SEV1	SEV2
May 2016	81.8%	Unoccupied
June 2016	81.8%	89.9%
July 2016	81.8%	90.6%
Aug. 2016	81.8%	112.1%
Sept. 2016	81.8%	89.9%
Oct. 2016	81.8%	47.5%
Nov. 2016	81.8%	0.0%*
Dec. 2016	81.8%	5.3%*
Jan. 2017	81.8%	18.5%*
Feb. 2017	81.8%	20.5%*
March 2017	81.8%	81.8%
April 2017	81.8%	81.8%
May 2017	81.8%	108.2%
June 2017	81.8%	99.0%
*SEV2 homeowner reported turning the ERV off frequently during these months for comfort		

3.3.3 Southeast Volusia Set Point Temperatures, Transfer Fans, and Resulting Room-to-Room Temperature Distribution

Overall, the temperature distribution from one room to another was found to be adequate, although there were some differences between the units. Adequacy of temperature distribution is attributed to the forced mixing created by the transfer fan system, especially during cooling months when run times in Florida are long.

Fan forced supply and passive return airflows generated by the transfer fan system and pressure mapping with interior door closure were measured in SEV1 around the time of initial occupancy. Results are shown in Table 3-6.

Table 3-6. Pressure Mapping Test Results for SEV1

Room configuration	Supply Flow (cfm)	Return Flow (cfm)	Pressure Difference, Bedroom with Respect to Main Body (Pa)
Front bedroom, door open	94		0
Front bedroom, door closed	88	44	0
Back bedroom, door open	108		0
Back bedroom, door closed		49	0
Bathroom, door open	36	NA	
Bathroom, door closed	35	NA	

The operational characteristics of SEV1 and SEV2 are very different and important to understanding the temperature distribution. The owner of SEV1 reported leaving both air handlers set to 72°F, yet indoor temperature varied from 69°F–79°F. The SEV1 homeowner also reported never interacting with the transfer fans except to change the single filter that serves both transfer fans. In contrast to SEV1, the owner of SEV2 preferred much warmer indoor temperatures (76°F–80°F); and the owner reported frequently adjusting set points, turning transfer fans (one or both) off and on, and not changing the filter serving both transfer fans.

Power measurements of the transfer fans (Figure 3-10) document their use by the homeowners. It should be noted that a transfer fan filter was installed by the builder to protect the transfer fans; however, this is not part of normal installation. As shown, SEV2 (pink) turned the transfer fan system off for much of the period between November 2016 and April 2017. The homeowner reported that this improved his comfort in the bedrooms during this time. The power measurements also show the influence of static pressure on power use. The owner of SEV1 (orange in the figure) reported changing the transfer fan filter regularly, yet a progressively increasing power signature was recorded beginning halfway through the year. Researchers ensured that the transfer fan filters were changed in both SEV1 and SEV2 in June 2017, and a dramatic drop in power use is apparent in the data. The owner of SEV2 reports rarely changing transfer fan filters. The Panasonic Whisper Green transfer fans automatically respond to increasing static pressure and attempt to maintain the set flow rate, although data representing how airflow changes with static pressure is not available. Some decrease in airflow is expected. Despite the absence of continuous flow monitoring, the transfer fan system in SEV2 is expected to be less effective at maintaining even temperatures throughout the home for much of the period because of higher static pressure and less overall run time.

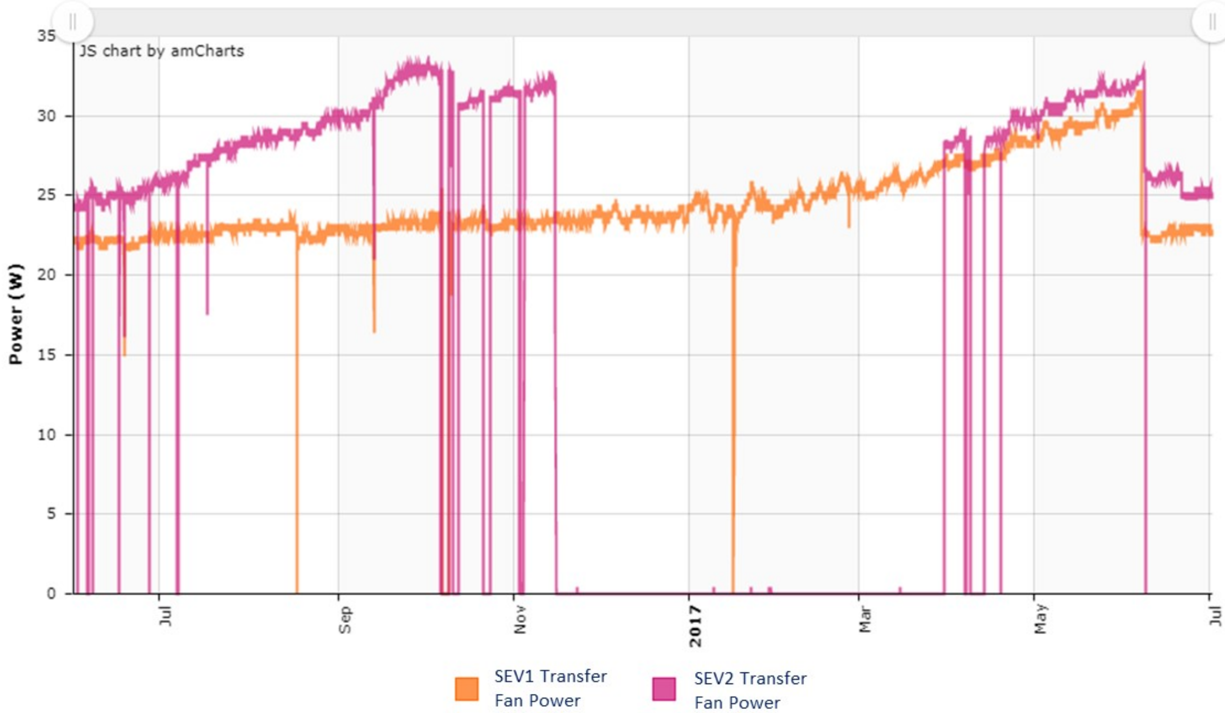


Figure 3-10. Transfer fan power consumption comparison of SEV1 (orange) to SEV2 (Pink)

Without a duct system to distribute conditioned air, there is concern that rooms without a fan coil unit will not be comfortable. To assess this in the SEVHFH homes, the research team measured temperature in each bedroom for comparison to the main body. The ACCA Standard Manual RS (Residential Systems Overview, ACCA 1997) proffers that a ±3° temperature difference from the thermostat set point for a single-zone system maintains comfortable indoor temperature variation within a dwelling during the cooling season. Although there are two indoor units, both serve the same single zone, which is not always the case in multisplit installations.

Bedroom temperatures in SEV1 were consistently ±3°F of the main body. Table 3-7 shows the number of hours each bedroom in SEV1 exceeded the target.

Table 3-7. Hours When the SEV1 Front and Back Bedrooms Were Outside the ±3°F Target Range

Temperature Difference	Front Bedroom		Back Bedroom	
	Hours	%	Hours	%
Total hours (n)	9767.0		9153.0	
Bedroom > 3°F Warmer	202	2.1%	46	0.5%
Bedroom <-3°F Cooler	0	0%	0	0%

In Figure 3-11, SEV1 bedroom temperature differences are shown as positive when the bedroom was warmer than the main body and negative when the bedroom was cooler. The two red lines indicate the upper and lower limit of the ±3°F target comfort range. The color of each month is a qualitative indicator of season. Pink and red indicate summer; green and orange indicate spring and fall, respectively; and blue indicates winter. The front bedroom was out of the target range for 202 hours, or 2.1% of 9,767 total hours. The back bedroom experienced approximately half as many hours outside the target range. Neither bedroom in SEV1 dropped to less than the target of 3°F cooler.

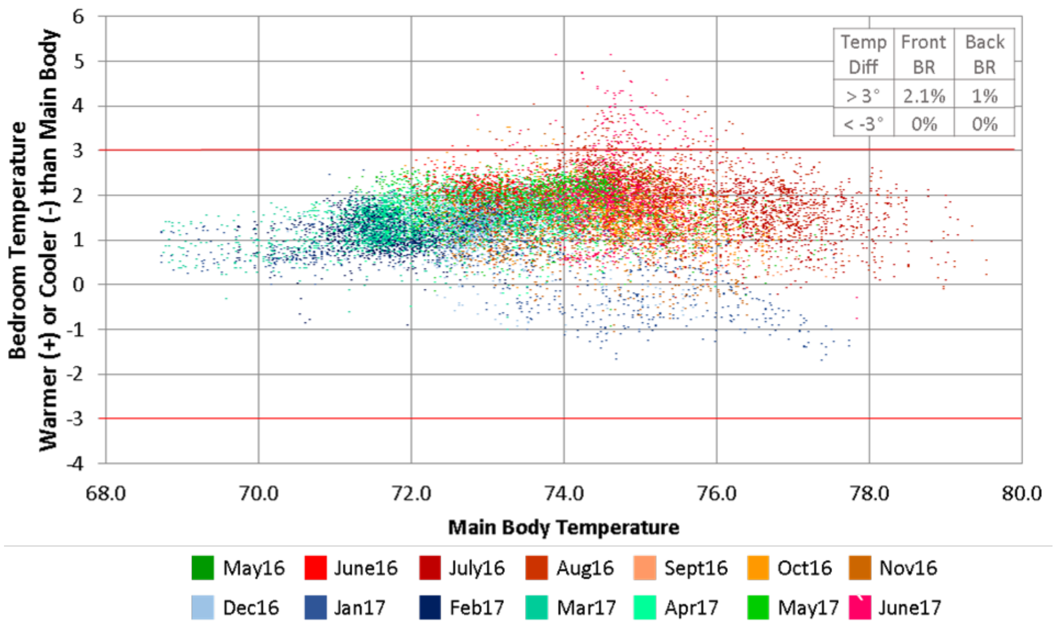


Figure 3-11. Average hourly temperature difference between the SEV1 bedrooms and the main body

Table 3-8 refines the data, showing that 86% of the hours exceeding 3°F warmer occurred in August 2016 (40%) and June 2017 (46%). Researchers found no ready explanation for these excursions. Main body and ambient temperatures were consistent with the data set in general. As evident in Figure 3-12, only 2 hours in the back bedroom (outlined markers) and 26 hours in the front bedroom exceeded 4°F warmer than the main body, suggesting an overall adequate provision of conditioned air to the bedrooms in SEV1.

Table 3-8. Hours the SEV1 Front and Back Bedrooms Were More Than 3°F Warmer than Main Body (by Month)[a]

	June 2016	July 2016	**Aug. 2016**	Sept. 2016	Oct. 2016	Nov. 2016	May 2017	**June 2017**	Total
Front bedroom		3	**70**	2	10	5	1	**111**	202
Back bedroom	2	10	**30**					**4**	46
Total	2	13	**100**	2	10	5	1	**115**	248
Percentage	1%	5%	**40%**	1%	4%	2%	0%	**46%**	100%

[a] Months with no hours that exceeded 3°F are omitted.

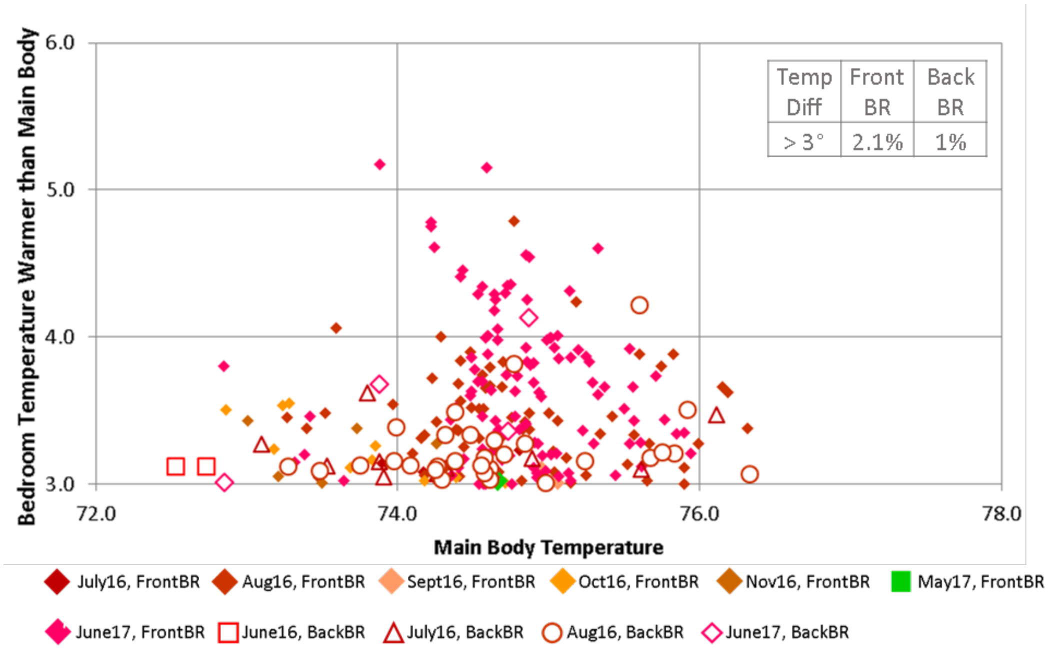


Figure 3-12. Periods when the SEV1 front (solid) and back (outline) bedroom temperatures were more than 3°F warmer than the main body

In contrast to the "set it and forget it" thermostat management strategy and continuous transfer fan operation in SEV1, the owner of SEV2 regularly interacted with both the AHUs, the ERV, and the transfer fans to customize comfort. The SEV2 owner also preferred warmer temperatures than the owner of SEV1, in the range of 78°F to 80°F in the main body. As previously discussed, the owner of SEV2 reported manually turning the transfer fans "off" to control comfort in the bedrooms, primarily during heating. Transfer fans were off for much of the period spanning from November 2016 to March 2017, which is shown in Figure 3-10. Additionally, the owner reported not changing the filter that serves both transfer fans, which in turn increased static pressure and diminished fan flow. These factors combined suggest that bedroom temperatures will fall outside the target comfort range of ±3°F more frequently than experienced in SEV1, which the data confirm; however, it is important to acknowledge that rather than being an indication of discomfort, these conditions met the comfort objectives of this homeowner.

As shown in Table 3-9, the SEV2 front bedroom temperature exceeded the ACCA RS upper limit target of 3° F warmer than the main body for 456 hours (4.9%) of the monitoring period; however, the back bedroom exceeded the target 3,052 hours, or 32.7% of the monitoring period.

Figure 3-13 shows that the ACCA target of ±3°F (between the red lines) was exceeded primarily during the cooling months, represented by red and pink markers in the graph. Taking a closer look at the SEV2 hours when the bedroom temperature exceeded 3°F warmer than the main body reveals that approximately 90% of those occurred during the cooling months from May through September; with fewer hours during the shoulder months of April, October, and November; and minimal hours during the winter months of December, January, and February (Table 3-10 and Figure 3-14).

Table 3-9. Average Hourly Bedroom Temperature Compared to Main Body Temperature

Temperature Difference	Front Bedroom		Back Bedroom	
	Hours	%	Hours	%
Total hours (n)	9335		9335	
Bedroom > 3°F Warmer	456	4.9%	3052	32.7%
Bedroom <-3°F Cooler	106	1%	0	0%

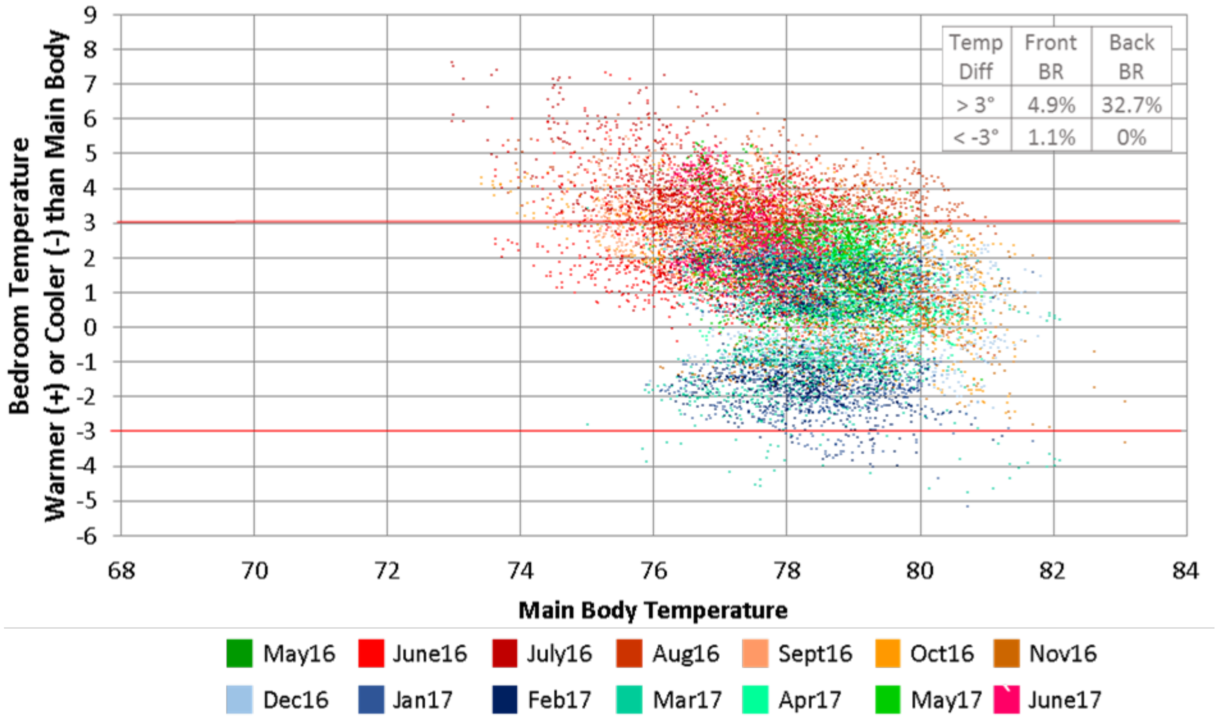


Figure 3-13. Average hourly temperature difference between the SEV2 bedrooms and the main body

Table 3-10. Hours When the SEV2 Front and Back Bedrooms Were More Than 3°F Warmer than the Main Body

	June 2016	July 2016	Aug. 2016	Sept. 2016	Oct. 2016	Nov. 2016	Jan. 2017	March 2017	April 2017	May 2017	June 2017	Total	% of Monitoring Period
Front bed-room	40	257	72	67	18	0	0	0	0	2	0	456	4.9%
Back bed-room	508	608	593	461	186	64	3	3	46	258	322	3,052	32.7%
Total	548	865	665	528	204	64	3	3	46	260	322	3,508	
	15.6%	24.7%	19.0%	15.1%	5.8%	1.8%	0.1%	0.1%	1.3%	7.4%	9.2%	100.0 %	

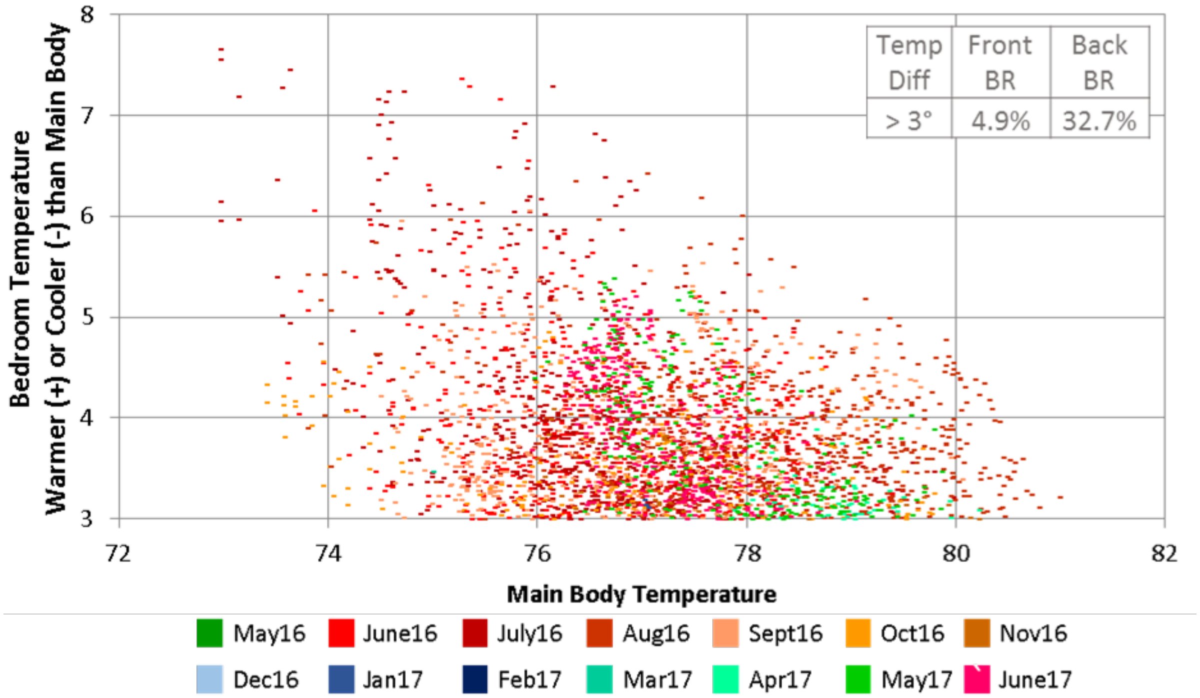


Figure 3-14. Periods when the SEV1 front and back bedroom temperatures were more than 3°F warmer than the main body

In summary, when operated as designed with transfer fans running continuously, fan filter changed regularly, and a “set it and forget” thermostat management strategy, the thermal comfort distribution is within guidelines set forth in ACCA Manual RS and is comparable to that achieved with the fully ducted systems in the test homes built by SSHFH discussed in Section 3.4. Conversely, the owner of SEV2 illustrated that the system can be operated to achieve specific thermal goals in different rooms of the house, such as preferring warmer indoor temperatures. The owner expressed satisfaction with larger temperature differences in bedrooms despite being out of the range identified in ACCA Manual RS. That flexibility might be an attractive feature for occupants who struggle with agreeing on a set point or who have seldom-used rooms they would rather not condition.

3.3.4 Indoor Relative Humidity

Indoor RH was monitored during the experimental period, and the tabulation of average hourly RH is binned into RH ranges and shown in Figure 3-15 for SEV1 and Figure 3-16 for SEV2. Sensors in the main body (at the entrance to the transfer fans) and in each bedroom were averaged to determine a single RH value for each hour in each home. Differing total hours for each month reflect missing data or data that were removed for other reasons, such as power outages and hurricanes. Average monthly indoor temperature is also shown in the figures, by a black dot in each bar, highlighting different comfort preferences of occupants in the two units.

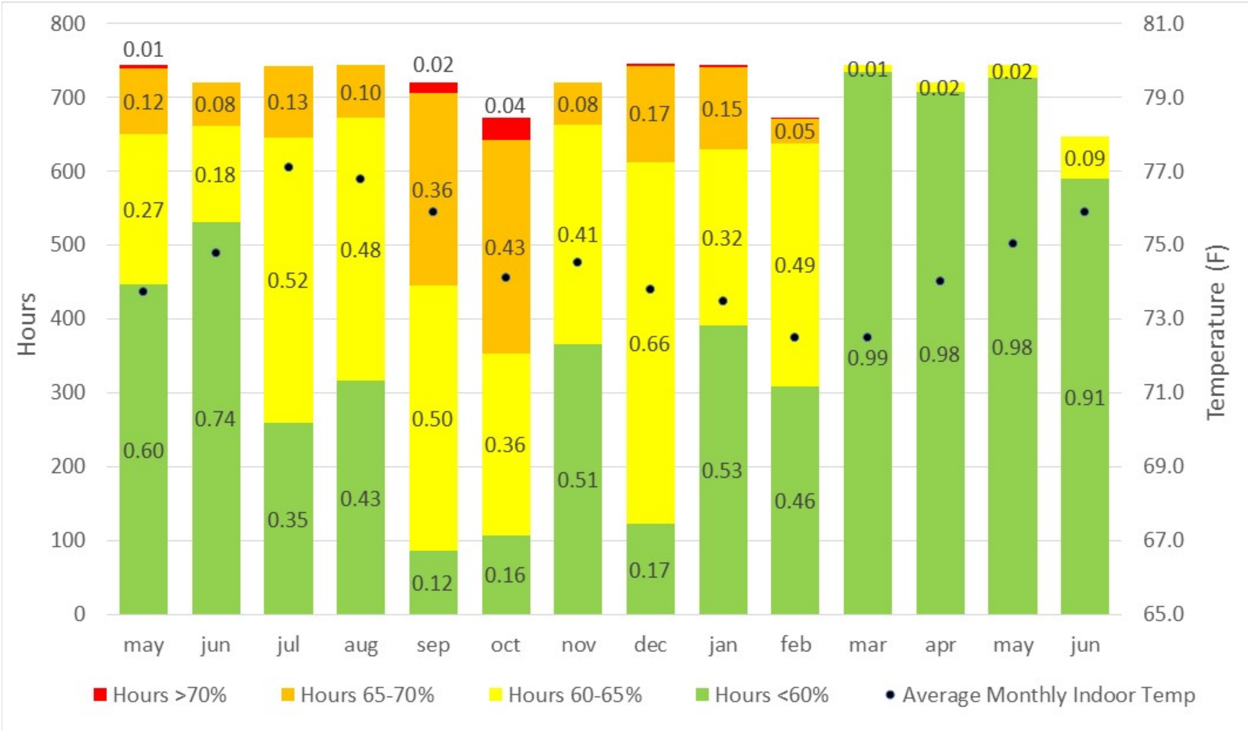


Figure 3-15. SEV1 average hourly RH and monthly average temperature

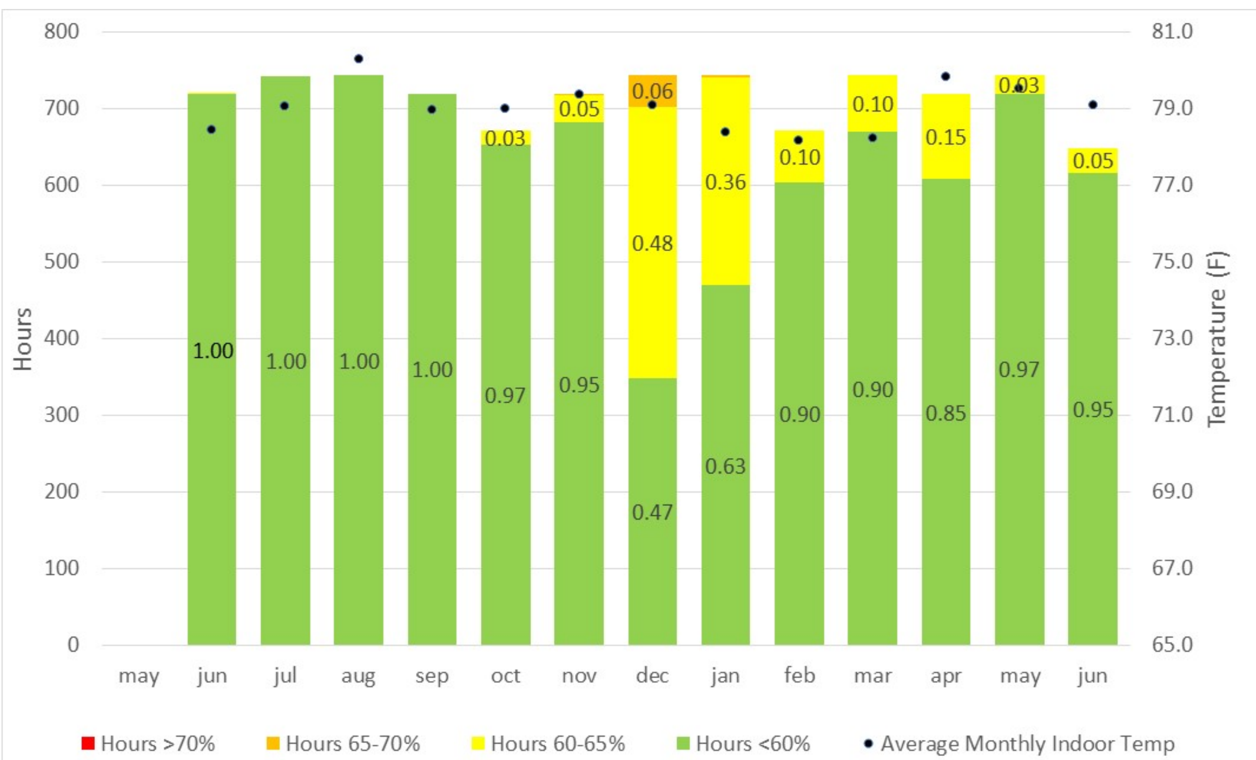


Figure 3-16. SEV2 average hourly RH and monthly average temperature

In SEV1, we observed that indoor RH was more than 60% for 43% of the monitoring period. Until March 2017, indoor RH was typically more than 60% for 60% of the time. For a significant fraction of hours, RH was more than 65% during the transitional months of September and October as outdoor temperature fell yet outdoor dew point remained high (refer to Figure 3-8). Appreciable cooling operation continued during the variable weather period of November 2016–April 2017, with only sporadic heating identified for a few days at a time in November, December, and January. Specific circumstances resulting in the transition to improved overall indoor RH beginning in March 2017 are unknown, but differences in indoor RH profiles before and after this point in time appear to be related to heat pump operational characteristics, which are discussed in Section 3.3.5.

In SEV2, we observed that indoor RH was more than 60% for only 11% of the time. A significantly increased number of hours at more than 60% was observed from December 2016–January 2017 when the ERV was turned off. The preference for warmer indoor temperatures required appreciable heating energy use from the end of October 2016 through March 2017 and again for a few days in April. Although SEV2 indoor RH data show the achievement of *comfort* metrics by being less than 60% most of the time, note that the preference for higher indoor temperatures in SEV2 skewed the comparison between the units when using RH as a metric of *moisture* because of psychometrics. Figure 3-17 shows that the indoor dew point, a better indicator of absolute moisture, in SEV2 was actually higher than in SEV1 for part of the year.

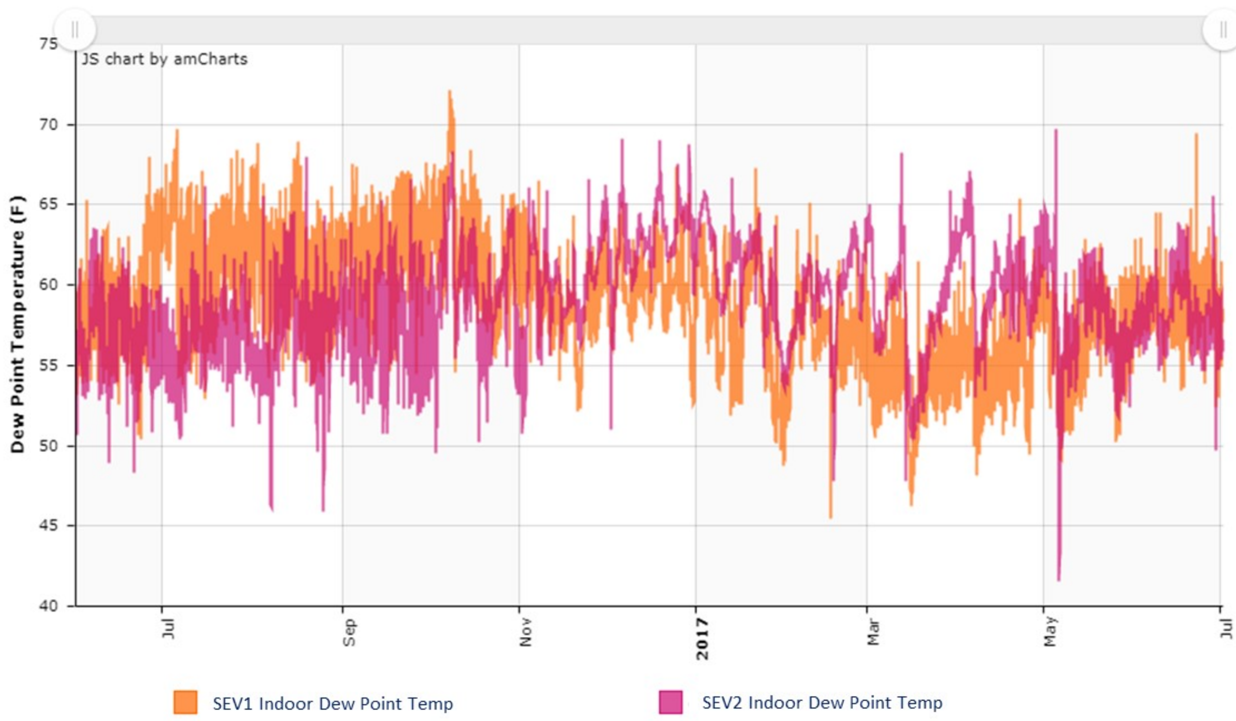


Figure 3-17. SEV1 and SEV2 average hourly indoor dew point temperatures

3.3.5 Variable Heat Pump System Operation

Although differences in indoor RH between SEV1 and SEV2 are affected by differing indoor temperature, they also appear to be largely related to how the multisplit system operates. Run time of the multisplit is the primary mechanism to remove indoor moisture, and the variable-capacity nature of the multisplit enables it to provide total (sensible plus latent) cooling throughout a wide range. In addition to responding to building load, multisplit run time is affected by how the occupants interact with the controls on the indoor fan coils. The homeowner in SEV1 reported a “set it and forget it” approach, whereas the homeowner in SEV2 interacted more frequently with the two indoor fan coils to customize comfort. Specific preferences for and changes to the variety of settings on indoor fan coils could not be tracked, but a variety of operational characteristics were continuously monitored.

Figures 3-18 and 3-19 summarize how the indoor RH in each home responded to multisplit operation, which in these plots are represented as 15-minute averages of total system power, for a few representative days during the peak cooling month of August 2016. During overnight hours, the systems in both homes operated at a lower capacity, evidenced by the lower power draw (purple dashed line), and cycling behavior is evident from variable supply air temperatures (SATs) (orange and pink lines), with longer overnight "off" periods evident at SEV2. During these overnight hours, indoor RH (light blue line) steadily climbed until higher power system operation resumed the next morning (likely caused by the solar load in SEV1 and occupant interaction in SEV2 because power ramps up prior to sunrise). A multisplit system that enabled the indoor units to cycle off ("thermal off") was specifically specified to minimize re-entrainment of moisture into the supply airstream. Many minisplit and multisplit systems either run the indoor fan constantly or at regular intervals to sample indoor air temperature. Withers (Withers 2016b) monitored one minisplit unit that exhibited cycles of fan operation for 15 seconds, followed by a 45-second off period. This pattern was continuous until interrupted by a cooling cycle.

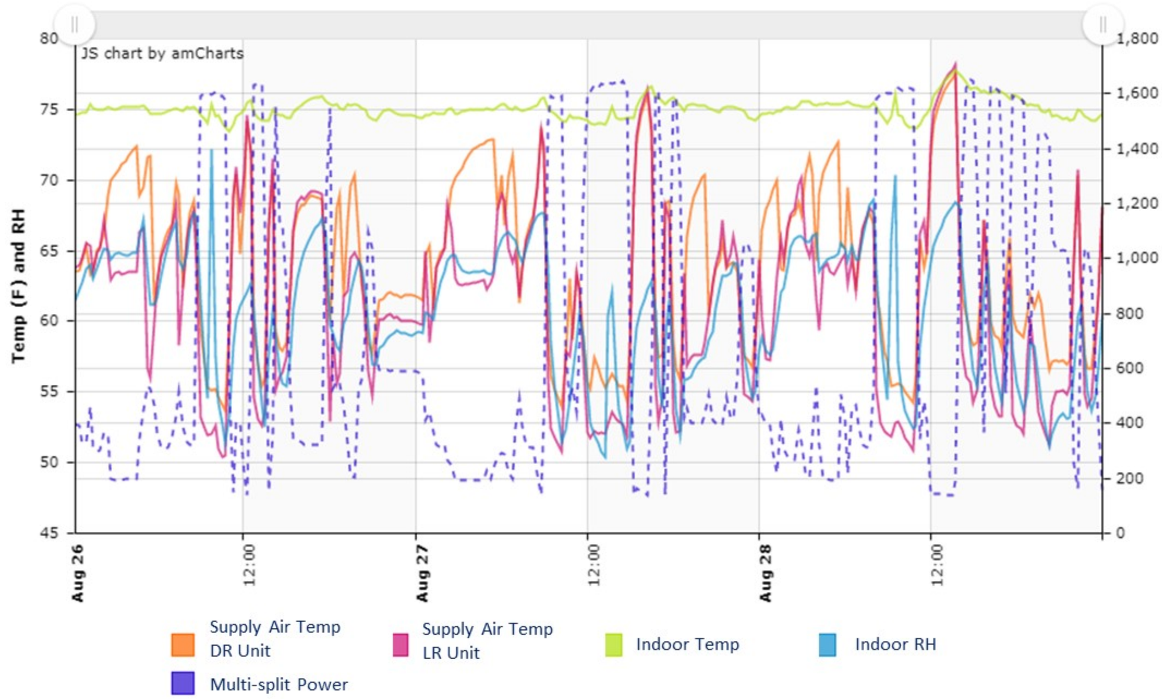


Figure 3-18. SEV1 indoor temperature and RH, along with SATs of the dining room (DR) and living room (LR) indoor fan coil units. The dashed line shows the total power of the multisplit system.

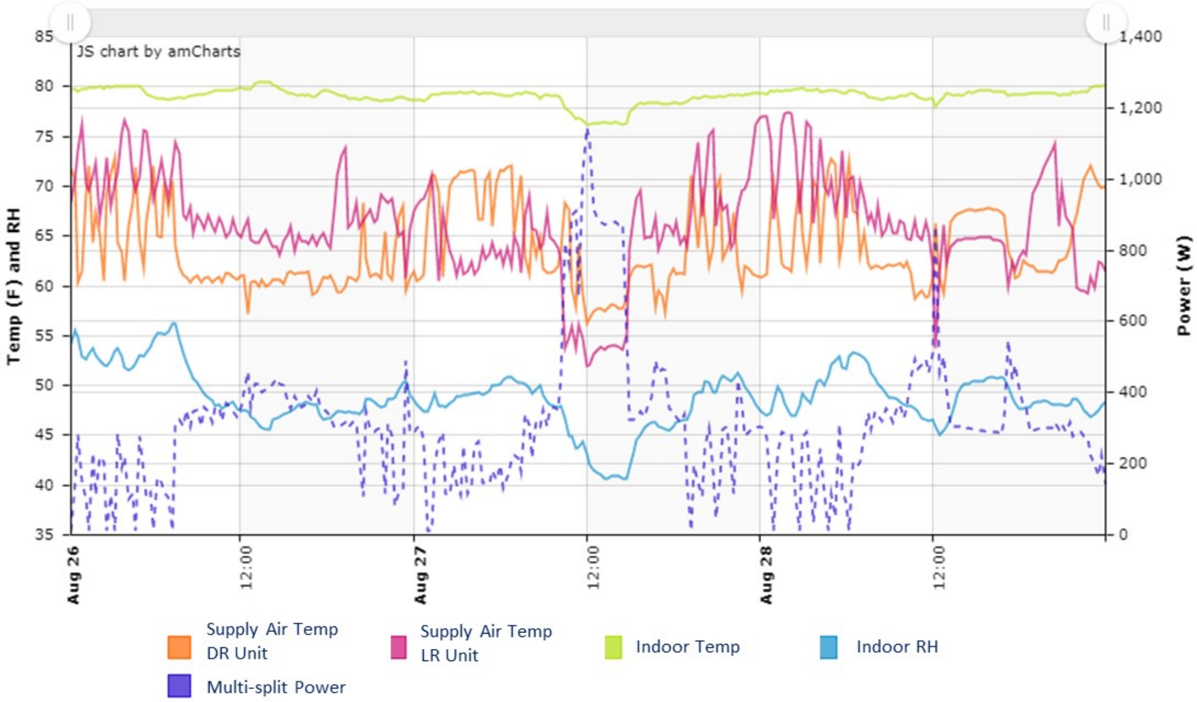


Figure 3-19. SEV2 indoor temperature and RH, along with SATs of the dining room (DR) and living room (LR) indoor fan coil units. The dashed line shows the total power of the multisplit system.

As shown in Figures 3-18 and 3-19, steady, consistent system operation (power draw, purple dashed line) resulted in consistently cold SAT (orange and pink) from at least one of the indoor units. System operation was much more variable in SEV1, resulting in more variation in SAT and more variation in RH (blue line). Even during the day, despite higher power use than overnight, what appears to be cycling behavior is apparent when the system in SEV1 provided short bursts of high-powered cooling with low SAT, followed by brief periods of lower powered cooling with warmer SAT. Conversely, the system in SEV2 exhibited more consistent operation and more consistently cold SAT.

A detailed look at the operation of the indoor fan coils in SEV1 and SEV2 is shown in Figures 3-20 and 3-21. For these plots, the power draw of each indoor fan coil unit was placed into qualitative bins from 1–5, with 1 representing the lowest power operation, and hence the lowest airflow, and 5 representing the highest power operation, and hence highest airflow. This relationship was determined on-site by watching amp draw readings while operating the units under differing control settings. Different colors denote the different bins for each indoor unit. The plots represent the fraction of time each indoor fan coil (living room and dining room) operated at each speed during each 15-minute period for the same monitoring period as the previous two graphs. Although the varied operation shown in Figure 3-20 makes it difficult to visually discern proportional run time at each speed, one can qualitatively see the extent of speed variation in SEV1 (Figure 3-20) compared to that of SEV2 (Figure 3-21). The plots show that the dining room unit in SEV2 had much more consistent, lower speed operation than either of the units in SEV1, which were frequently adjusting speed and engaging higher speeds. This consistent operation in SEV2 partially contributed to lower indoor RH.

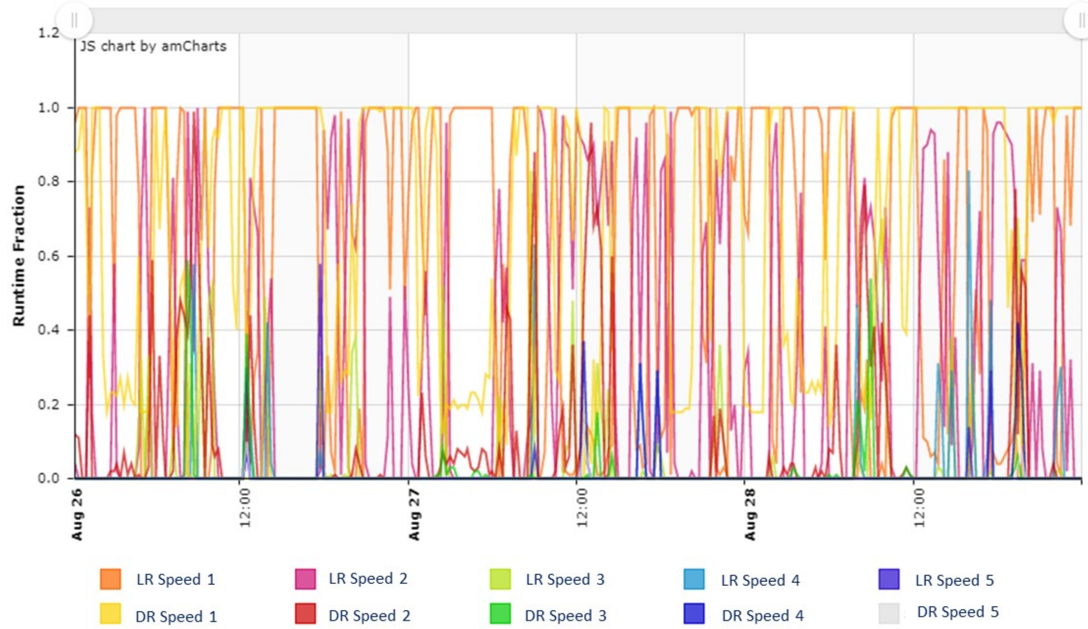


Figure 3-20. SEV1 run time fraction for the living room (LR) and dining room (DR) indoor fan coil units at each speed

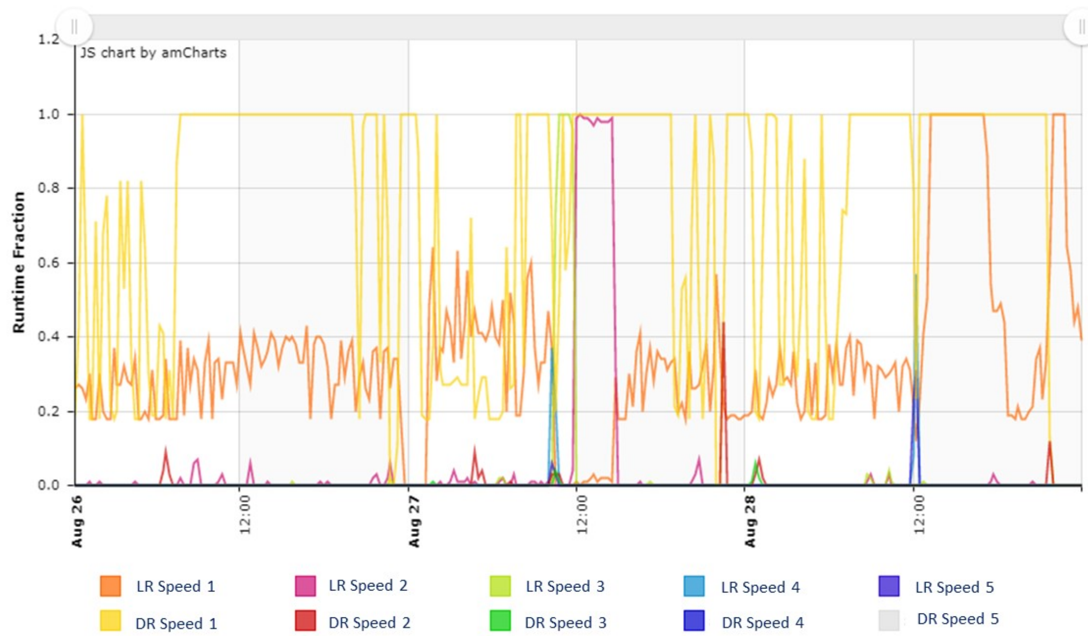


Figure 3-21. SEV2 run time fraction for the living room (LR) and dining room (DR) indoor fan coil units at each speed

Although the exact circumstances leading to the differing system operations between SEV1 and SEV2 are unknown, it is suspected that the difference in set point temperatures between the two homes plays a role. Many minisplit and multisplit systems respond according to degree of disparity between how far the indoor temperature is from the target set point temperature, and because of the desire for warmer indoor temperatures in SEV2, the rate of change in indoor temperature that the heat pump can accomplish might be faster than in SEV1. This quicker response time might result in less variation in compressor speed and indoor fan coil operation. One thing that can create this disparity is a change in set point temperature. Past research has recommended against using thermostat setups/setbacks with these systems for this reason (Ueno and Loomis 2015). In a previous study involving a multisplit heat pump, indoor RH was also not well controlled as a result of significant variation of a multisplit system (Sutherland, Parker, and Martin 2016). In this case, when the occupant would lower the thermostat to achieve comfort, the multisplit would immediately ramp up to high capacity, use a high SHR mode of operation, overshoot the set point, and turn off. The occupant would then raise the thermostat because it was too cold in the home, and the multisplit would remain off and allow the indoor temperature to drift slightly above the new set point (a relatively large deadband seems common with minisplits and multisplits, in part caused by the controls sometimes being configured in increments of 2°C rather than 1°F). Now, with the home too warm, the occupant would reduce the thermostat, and the cycle would continue, with poor indoor RH control throughout. Once the homeowner let the multisplit system find an equilibrium, constant, lower capacity operation and better indoor RH control was achieved. Although in the case of SEV1 we do not suspect that the occupants were changing the set point temperatures, we see that the warmer set point in SEV2 enabled the unit to maintain temperature with consistent, lower speed operation. To maintain the lower set point in SEV1, the system was forced to regularly ramp up to higher speed operation, similar to what would occur in response to a set point change.

Note that indoor RH conditions improved in SEV1 for the second monitored cooling season beginning March 2017. An example of the improvement is shown in the next two figures for a 3-day period in June 2017. The change in indoor RH seems to be largely related to more consistent system operation at low power states (orange and yellow), with the dining room unit operating at lower speed nearly 100% of the time. This might result from the occupant becoming more familiar with optimized multisplit operation and adjusting at least one of the indoor fan coil unit set points to enable better equilibrium to be achieved. The result confirms a relationship among lower speed, consistent operation, and indoor RH control. Ideally, for optimized RH control, equipment would shift into this mode of operation based on internal algorithms that do not require user interaction—for example, when the unit detects escalating RH without attendant temperature change; however, maintaining a cold SAT is also key to indoor RH control, and the next two figures show that the living room unit was also operating at low speed, but run time was much less consistent, resulting in higher average SAT.

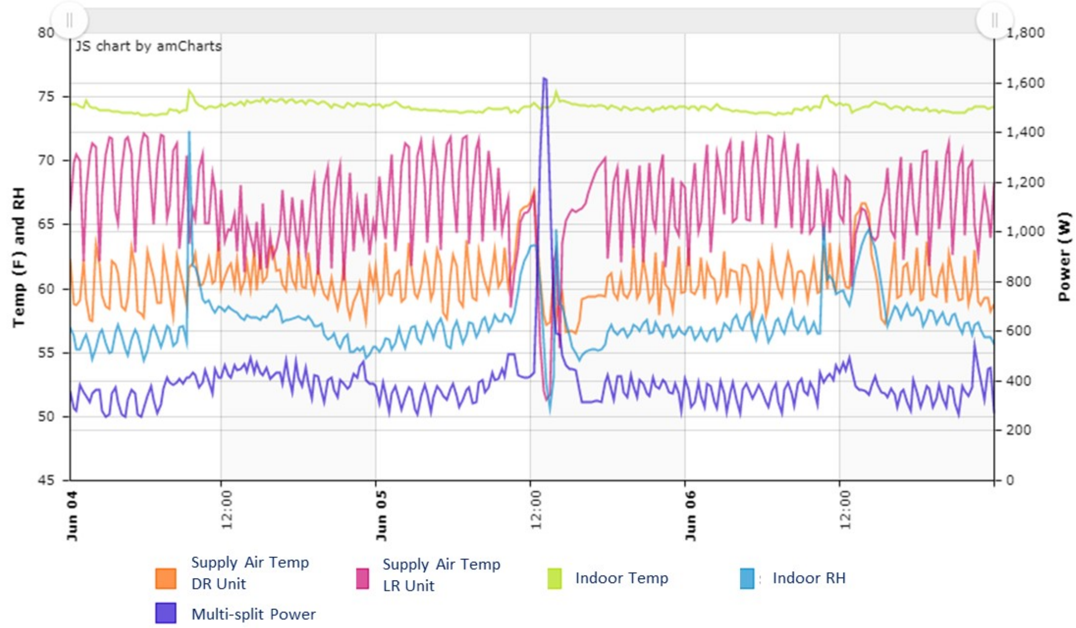


Figure 3-22. SEV1 indoor temperature and RH, along with SATs of the dining room (DR) and living room (LR) indoor fan coil units. The purple line shows the total power of the multisplit system.

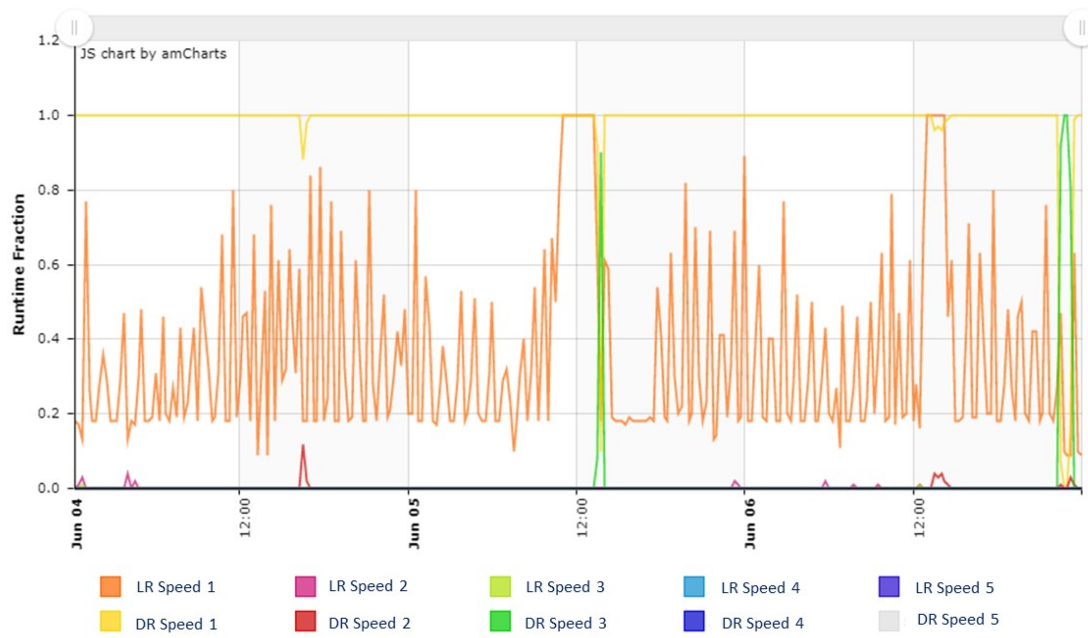


Figure 3-23. SEV1 run time fraction for the living room (LR) and dining room (DR) indoor fan coil units at each speed

3.4 South Sarasota Habitat for Humanity Test Home Design and Mechanical Systems

The test homes provided by SSHFH are identical single-family houses with three bedrooms, two bathrooms, a one-car garage, and 1,290 ft^2 of conditioned space (Figure 3-24). The study began with two houses initially, and one of the original test houses was replaced in 2017 when the homeowners elected to drop out of the study. The homes are referred to as SS1, SS2, and SS3 for the remainder of the report.

Figure 3-24. South Sarasota Habitat for Humanity test house, front and side views

In each house, the builder installed a fully ducted (return and supply) Mitsubishi Mr. Slim inverter-driven minisplit heat pump with one indoor AHU located in the unvented attic and one outdoor unit (Figure 3-25).

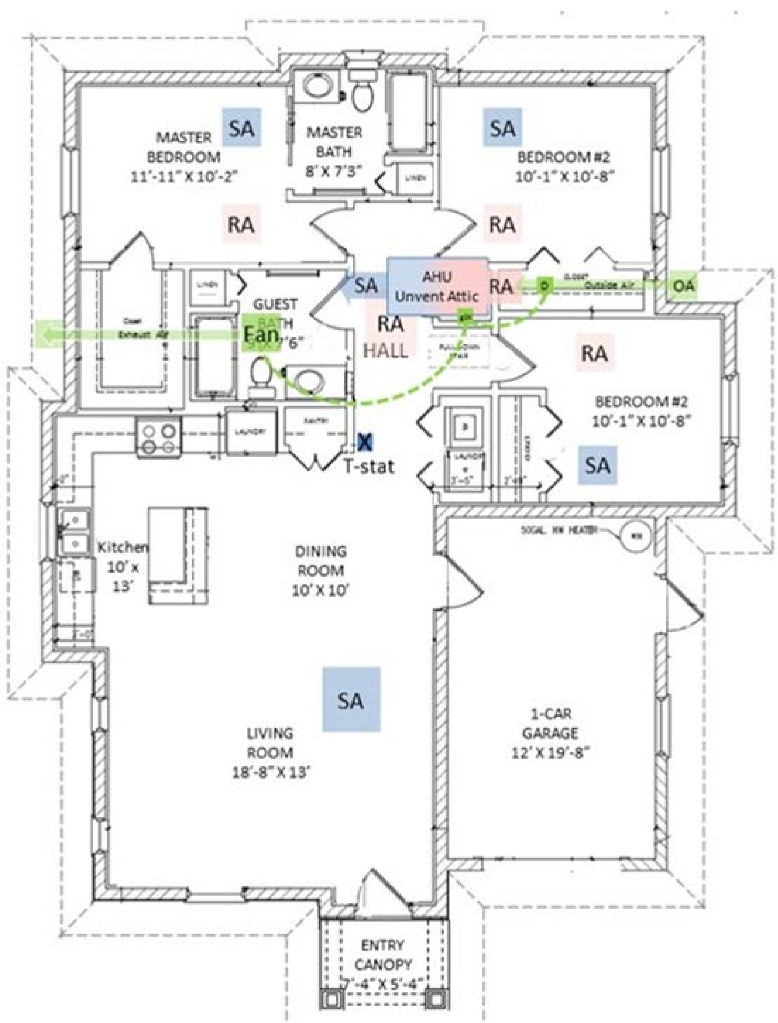


Figure 3-25. Floor plan with location of heating, ventilating, and air-conditioning components

Mechanical ventilation is designed to be supplied primarily by an outdoor air duct that runs from a soffit-mounted, filter-back grille (outside air intake) to the return plenum of the AHU, which supplies ventilation air to the home during heating and cooling system run time. An AirCycler g2-k ventilation control module tracks run time of the AHU fan and a bathroom exhaust fan, and it determines the hourly volume of air exchange based on user-entered values for outside airflow rate. If the flow in any given hour is not sufficient to meet the user-provided ASHRAE Standard 62.2-2010 hourly flow target, the g2-k turns on the Panasonic exhaust fan for an additional period. The g2-k simultaneously opens the electronic damper in the outdoor air duct in an attempt to provide a ducted path for makeup outside air. The system operates in this fashion, with the g2-k tracking airflow, until the target hourly ventilation rate is achieved. The outside air damper is always open when the minisplit heat pump is operating or when the exhaust fan is operating. In some hours, this may result in exceeding the ASHRAE 62.2 target. The g2-k includes an algorithm that allows for some "banking" of excess flow from one hour and to be credited toward the next. Schematics of the ventilation system components and the attic installation are shown in Figure 3-26 and Figure 3-27, respectively.

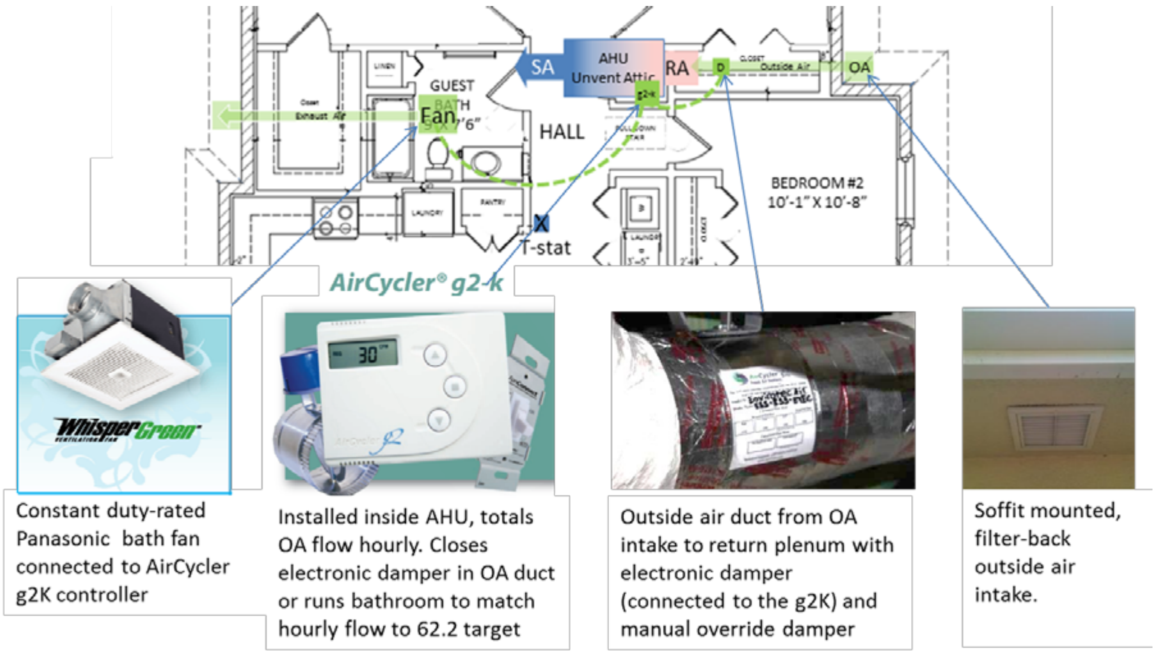


Figure 3-26. South Sarasota Habitat for Humanity outside air ventilation system components

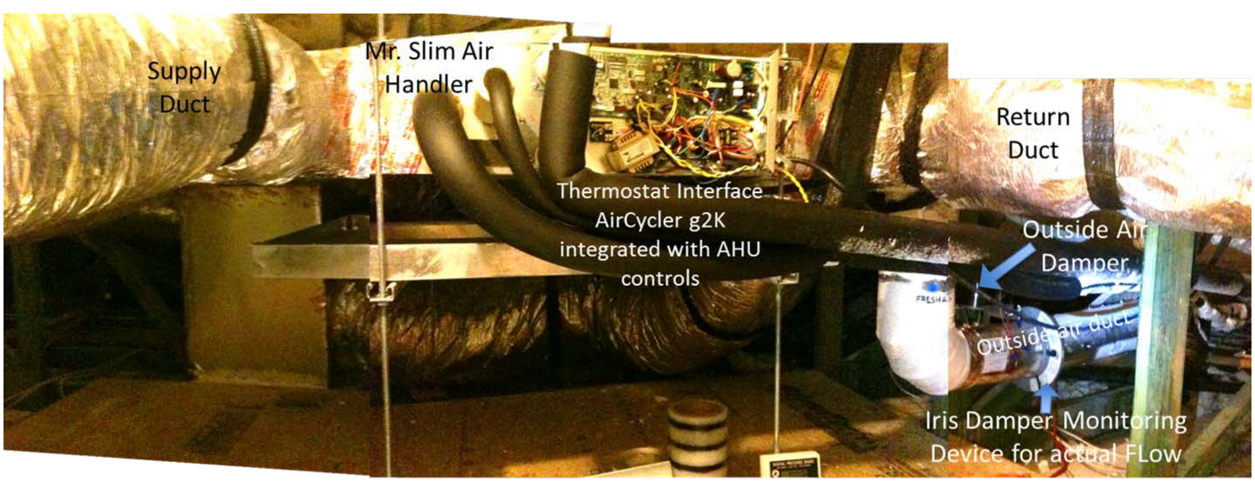


Figure 3-27. South Sarasota Habitat for Humanity thermostat interface and g2-k-integrated Mr. Slim air handler control box, outside air duct and electronic damper, and IRIS damper installed by Florida Solar Energy Center to measure actual outside airflow. Installation is in an unvented attic.

For the g2-k system to work, it needed to be mated with a conventional thermostat, and therefore the standard thermostat integrated in the Mr. Slim AHU, typically set using an infrared remote control, could not be used. A thermostat interface supplied by Mitsubishi was used to enable the Mr. Slim unit to operate with the conventional, wall-mounted thermostat. This control approach slightly limits the ability of the Mr. Slim unit to vary its capacity, and it forces it to operate at a fewer number of discrete speeds; however, the benefits of the variable-capacity nature of the system can still be realized.

The cost of SSHFH's minisplit installation was $7,500, compared to $6,800 for a conventional, fixed-capacity split system. Model numbers for all HVAC components are provided in Table 3-11.

Table 3-11. Heating, Ventilating, and Air-Conditioning Components for Test Homes Built by South Sarasota Habitat for Humanity

Component	Manufacturer	Model
Compressor	Mitsubishi	SUZ-KA15NA
AHU (ducted, unvented attic mounted)	Mitsubishi	SEZ-KD15NA "Mr. Slim"
Thermostat (conventional wall-mounted digital)	Honeywell	PRO6000 2-stage
Thermostat interface	Mitsubishi (third-party manufacturer)	PAC-US444CN-1
Outside air ventilation controller	AirCycler	g2-k
Exhaust ventilation fan	Panasonic	FV-05-11VK1

3.5 South Sarasota Habitat for Humanity Monitoring Strategy

The majority of data are acquired with a SiteSage 14-channel home energy monitor with sensor pod. One-minute measurements are transferred via broadband Internet to SiteSage servers and later retrieved by FSEC servers on a daily basis. Energy measurements include the minisplit heat pump and bathroom exhaust fan. Minisplit heat pump energy is recorded at the electrical breaker panel, which includes both indoor and outdoor units. An AC current sensor is located at the AHU power supply to determine its run time and qualitatively determine the speed at which the fan operates. One-minute measurements of temperature and RH are recorded with Vaisala HMP60 sensors for the minisplit supply air just after the cooling coil.

Temperature and RH measurements within the home are recorded every 2 minutes at four locations using Pointsix WiFi sensors. Locations include one adjacent to the thermostat in the hall and one in each of the three bedrooms, 5 feet above grade on the wall behind the door. Pointsix data are transmitted via broadband Internet direct to FSEC servers several times throughout the day.

Researchers installed a calibrated IRIS damper in-line in the outside air duct to measure real-time outside airflow, which varies with AHU fan speed. Monitoring equipment is summarized in Table 3-12.

Table 3-12. South Sarasota Habitat for Humanity Monitoring and Test Equipment

Measurement	Equipment	Accuracy
Energy and data acquisition	SiteSage Energy Monitor with Sensor Pod	±1% of rated current
Pressure differentials (fan flow sensors, air distribution pressures)	DG700 digital pressure gauge	1%
Whole-house ventilation, supply and return airflows	Flowblaster capture hood attachment to duct blaster with DG700	5%
Temperature and RH at supply air and outdoor airstreams	Vaisala HMP60	±0.5°C, ±3% RH
Room temperature and RH	Pointsix 3008-04-V6 WiFi transmitter	±0.4°C, ±3% RH
Pressure differential (outside airflow)	Energy Conservatory DG2 pressure gauge	±1%
AC current	Acuamp ACTR Series AC current transducer	1% FS
Building envelope air leakage	Minneapolis Blower Door System with DG-700 digital gauge	3%
Airflow	Continental Fans IRIS-06 damper	5%

3.6 South Sarasota Habitat for Humanity Results and Discussion

3.6.1 Outdoor Conditions and Monitoring Period

Data collection for the first test home (SS1) built by SSHFH began in July 2016, shortly after initial occupancy, and continued until the homeowner elected to leave the study in December 2016. A second test home (SS2) was instrumented, and data collection began at the end of July 2016, with initial occupancy in mid-August 2016, and continued through June 2017. To replace the SS1 home, a third test home (SS3) was instrumented before initial occupancy in April 2017.

Figure 3-28 shows outdoor temperature and dew point encompassing the entire monitoring period from the closest weather station, Sarasota/Brad International Airport (KSRQ), located 30 miles from the SSHFH project sites, which are in close proximity to each other. Conditions are similar to those shown for the SEVHFH ambient conditions, and the area is still considered Central Florida, even though the sites are very close to being in the South Florida climate zone, as demarcated for Florida Building Code purposes. In this case, the peak cooling period with consistently high outdoor temperature and dew point extends through September 2016, followed by a transitional period of steadily decreasing outdoor dew point from October through November 2016. December 2016 through approximately mid-March 2017 was characterized by variable conditions, with similar appreciable cooling load throughout the period, as was the case for the same period at the SEVHFH site. The need for occasional heating is still apparent, but less so than in Southeast Volusia. No significant heating was identified for SS2, and data for SS2 and SS3 during the bulk of the heating period are not available.

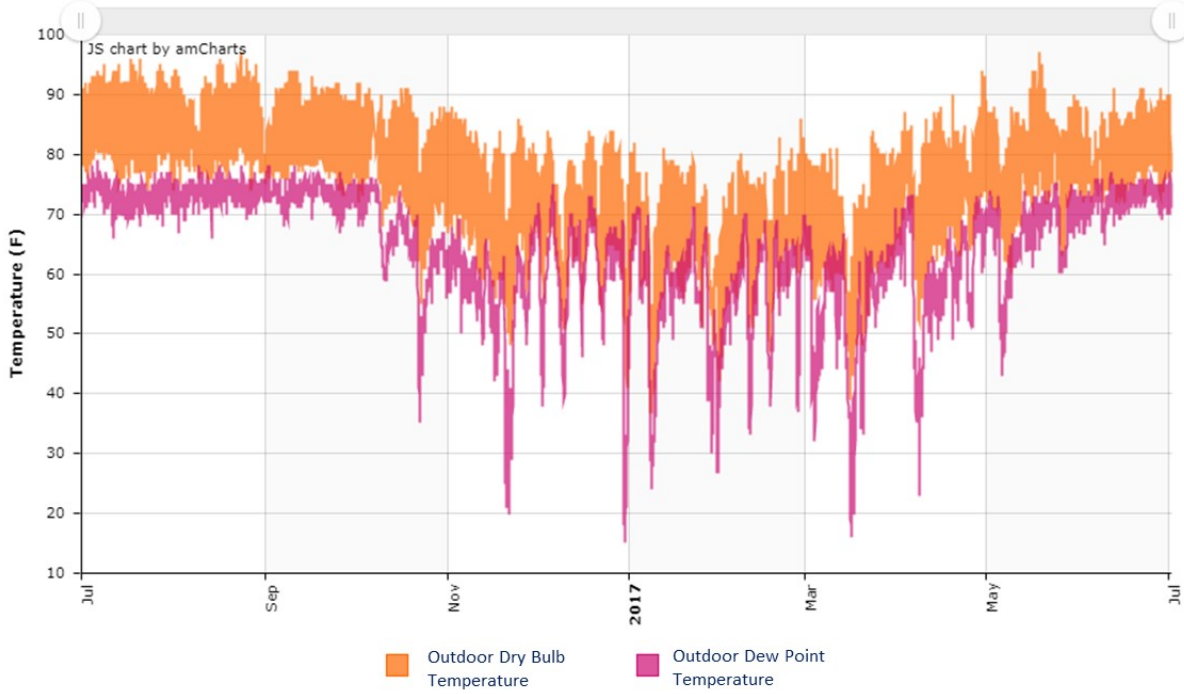


Figure 3-28. Ambient temperature and dew point from nearby Sarasota/Bradenton Airport National Weather Station

3.6.2 South Sarasota Habitat for Humanity Mechanical Ventilation Operation

Note that the g2-k does not measure outside airflow. It calculates outside airflow based on user inputs and logged run time. Operation of the g2-k ventilation controller is based on three fixed user-entered flow parameters: (1) required outdoor airflow, (2) supply ventilation airflow to return plenum, and (3) exhaust fan flow. Item 1 is fixed at the known, calculated ASHRAE 62.2-2010 ventilation rate (43 cfm), and Item 3 is the fixed exhaust fan flow rate when the unit is operating, but Item 2 can be more difficult to determine, especially with the variable nature of inverter-driven systems. With the benefit of measured flow data, the initial rate of ventilation supply air estimated by the contractor and entered into the g2-k was found to be generally higher than required to meet ASHRAE 62.2-2010 and was reduced early in the study. This type of finding and resulting calibration is beyond typical system commissioning because installers do not have access to monitored outside airflow data. Additional adjustments were made during the course of the experiment in an effort to match the design flow with mixed results.

Each of the two components of g2-k ventilation (supply and exhaust) were measured during the study. Supply flow measurements through the ventilation duct were calculated at 1-minute intervals based on manometer pressure measurements at the IRIS damper resulting in real-time flow readings of supply ventilation. Exhaust ventilation was determined indirectly via 1-minute fan power readings. A one-time exhaust flow measurement in each home was used to determine the ongoing flow rate in proportion to measured power.

High-resolution (1-minute) data collection provided greater detail of varying ventilation rates than typical hourly or 15-minute monitoring can provide. Collecting separate flow rates for both supply and exhaust ventilation required an account of each flow when operated individually. When operated simultaneously, ventilation rate analysis was based on the larger of the two flows during each 1-minute interval because balanced flows are not additive.

Mechanical ventilation data collection at SS2 began in late July, just prior to occupancy, and the data during the unoccupied period provides a snapshot of how the system operates (Figure 3-29). Prior to occupancy, the g2-k system was found to perform as intended, delivering outdoor air very close to the ASHRAE 62.2-2010 target of 43 cfm through a combination of supply and exhaust ventilation. As the minisplit run time (orange line) drops off, the exhaust fan (purple) turns on to make up the shortfall to the ventilation target, mainly during the evening. Figure 3-29 also shows that ventilation air (blue) increased after the house was occupied compared to the vacant period. Extended periods of constant exhaust fan use (purple) by the occupant are clearly seen, resulting in a total ventilation increase of 16% on average after occupancy. The August 5 move-in day was excluded from the comparison.

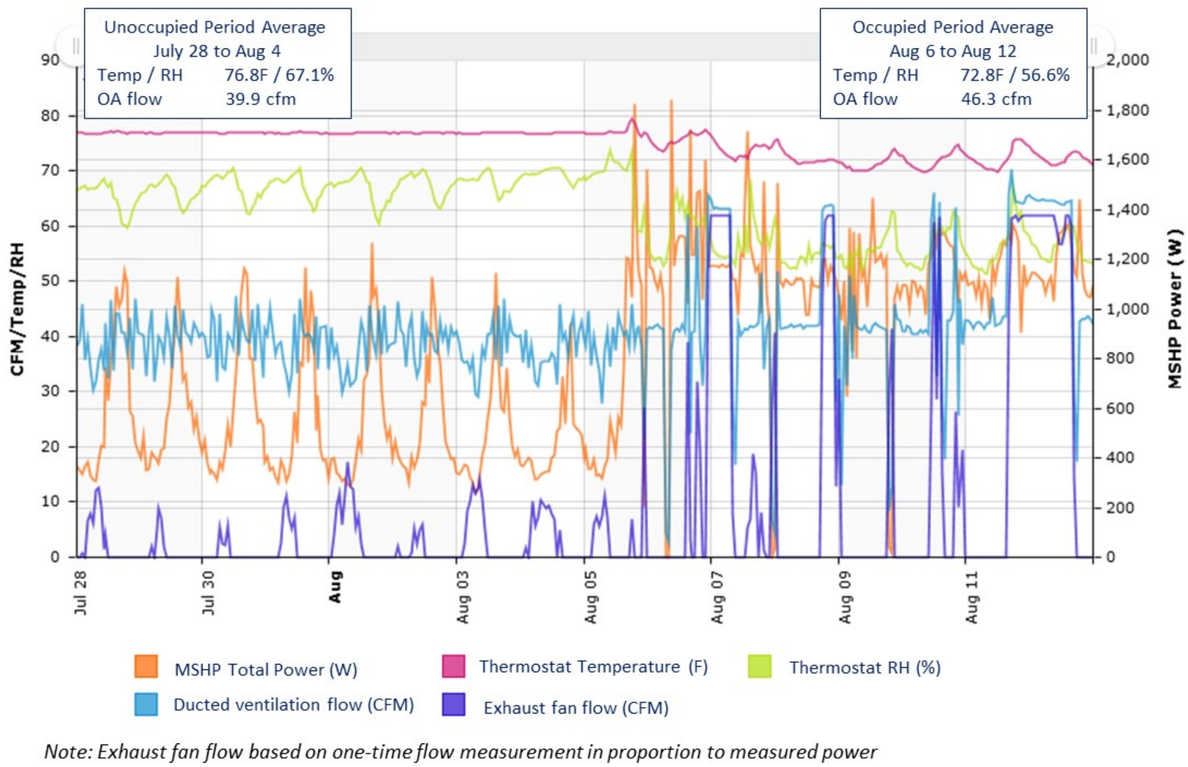


Figure 3-29. SS2 ventilation system performance, average hourly data

Figure 3-30 provides a detailed look at pre-occupancy performance on August 1, 2016, with 1-minute readings of energy, ventilation flow, and indoor temperature and RH. At this level of detail, discrete periods of supply ventilation air introduced by the system (blue) can be discerned as the damper operated to provide the programmed level of outdoor air. Supplemental ventilation via the exhaust (purple) fan occurred only during the morning hours when the run time of the central AHU (orange) was reduced in proportion to the cooling load. Increased AC run time during the afternoon and late evening allowed the central AHU to exclusively provide the full amount of intended ventilation, as evidenced by the lack of exhaust fan run time beginning at approximately noon. Humidity levels (green) rose noticeably during the morning and reduced during the afternoon in proportion to AC run time.

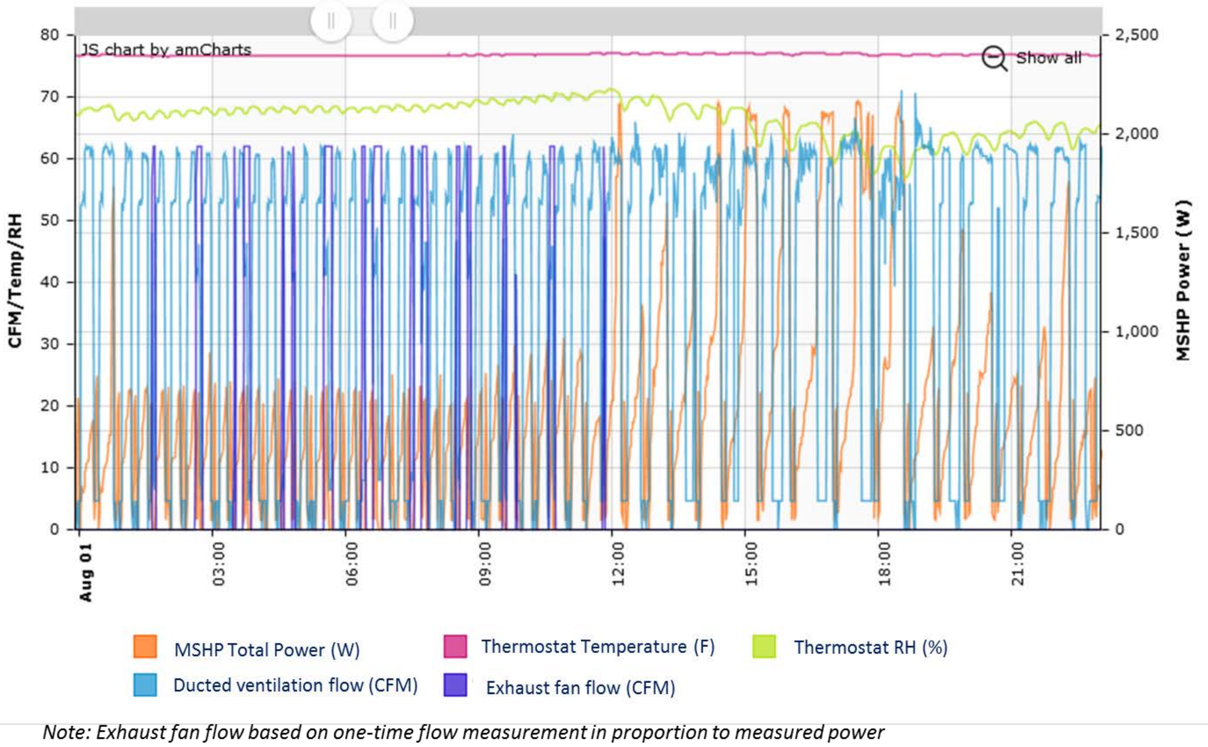


Figure 3-30. SS2 ventilation system performance pre-occupancy, 1-minute data values

By contrast, detailed post-occupancy performance is shown in Figure 3-31 on August 8, 2016. The 1-minute power readings showed nearly constant AC system operation. With nearly constant AHU operation, the outdoor air damper opened each hour according to system programming with no need for supplemental operation of the exhaust fan. There were, however, two periods of constant exhaust fan run time in the evening, which are attributed to use by the occupant rather than the g2-k system and a source of additional ventilation air. The g2-k controller always opened the damper on the supply ventilation air duct while the bath fan was running in an attempt to provide a balanced path. As a result of this extended bath fan run time, the damper "open" condition and constant minisplit run time, ventilation airflow that exceeds ASHRAE 62.2-2010 was achieved for these hours.

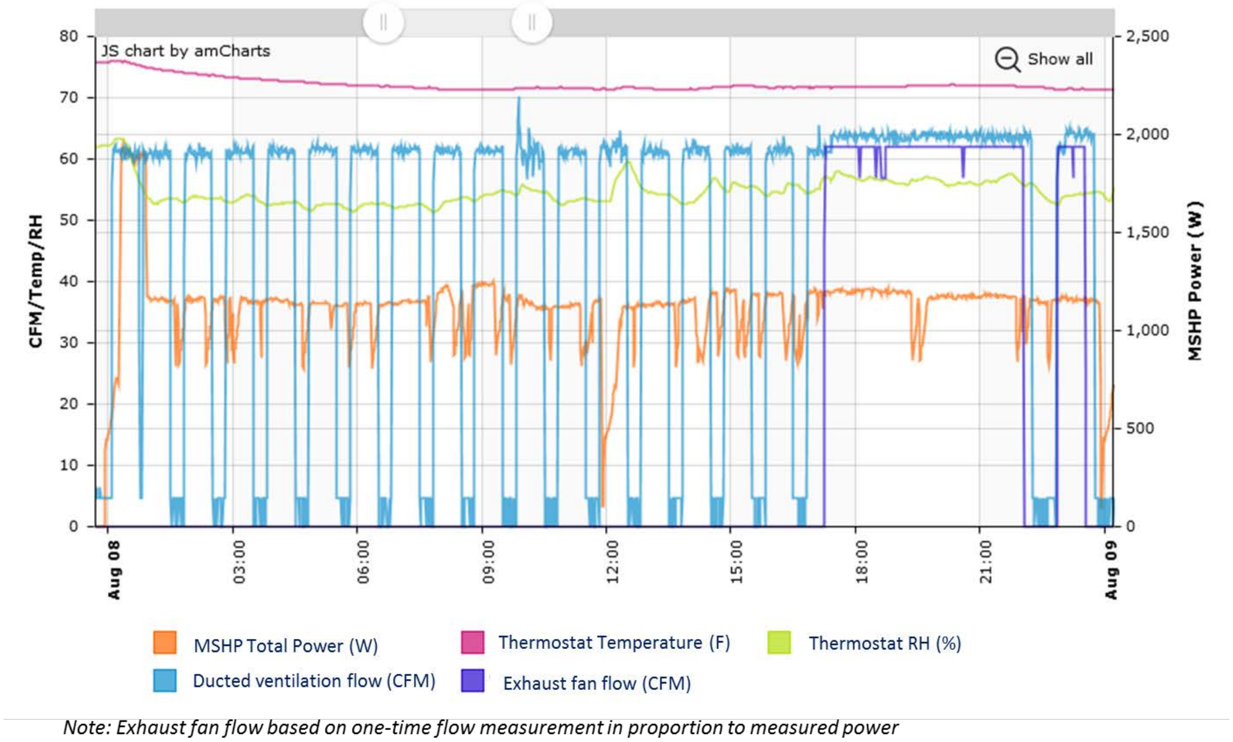


Figure 3-31. SS2 ventilation system performance post-occupancy, 1-minute data values

Table 3-13 shows monthly performance of the ventilation system at SS2 with respect to the ASHRAE 62.2-2010 target. Daily minimums and maximums are shown to indicate the variable nature of the air exchange achieved with this integrated system arrangement of a g2-k controller paired with a variable, inverter-driven system. Deficiencies in the collection of ventilation data caused gaps during some months, especially early in the study, as indicated by the number of days shown in the first column for each month. The last two columns show percentage contributions toward the ventilation rate from the supply and exhaust sources, with total ventilation attributed to the larger of the two sources when operating simultaneously (simultaneous supply and exhaust flows are not additive). As shown in Figure 3-31, manual operation of the exhaust fan will induce additional flow through the supply duct during extended AHU run times because it triggers the damper to remain open. Although *occupant* use of the bath *exhaust* fan contributes to a portion of the overventilation shown in the table, a large part of the surplus results from the *controller* calling for the supply air duct damper to be open longer than necessary for *supply* ventilation purposes. This demonstrates the difficulty of proper g2-k programming with this integrated system. An explanation follows.

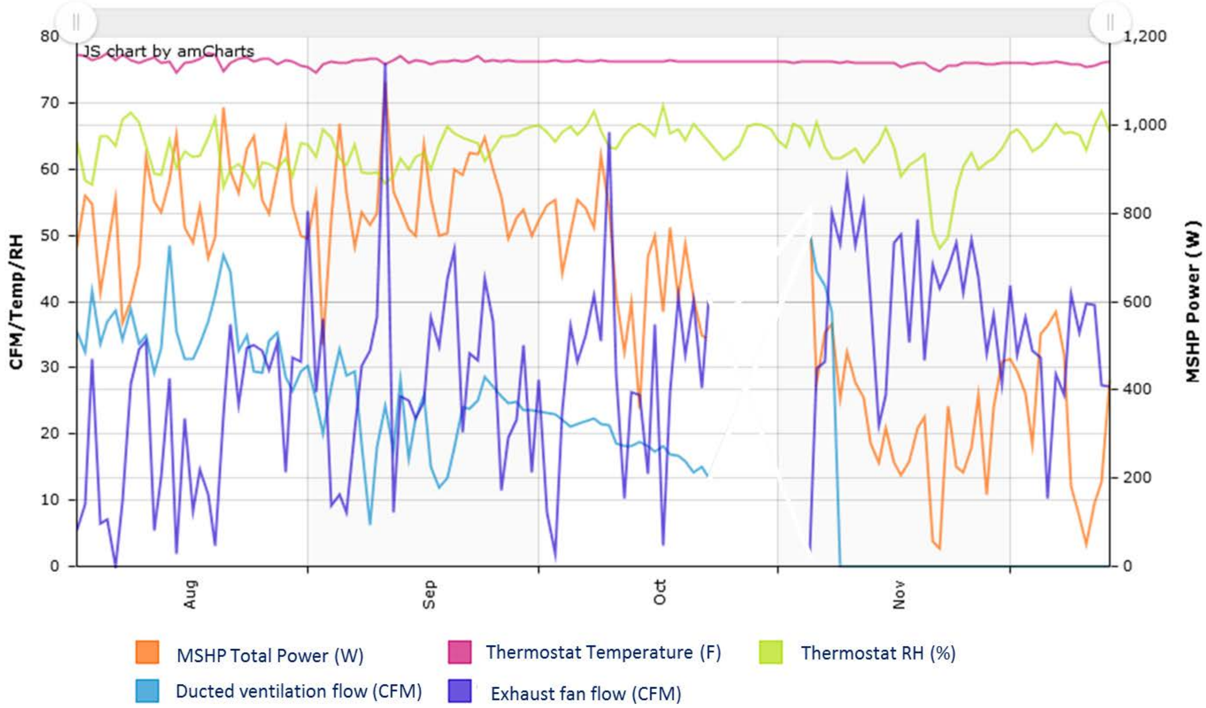
All months prior to April in Table 3-13 show overventilation (except for January). A large part of this might have been caused by an initial system setting upon installation that fixed AHU speed, the driving force for supply ventilation airflow, at a relatively high value (discussed in more detail in Section 3.6.5). The g2-k controller was initially programmed with an estimated supply ventilation airflow that assumed more variation in AHU flow.

January suffered a loss of exhaust fan data because of monitoring problems. The only similarly cooler months with sufficient data (February/March) indicate greater use of the exhaust fan to provide ventilation, as might be expected with reduced run time of the space-conditioning system. A noticeable reduction in ventilation below the design level occurred beginning in April, which coincided with an adjustment made by the contractor to the minisplit heat pump system on April 5. The change was intended primarily to address poor humidity control by reducing AHU airflow rates and thus the SAT, increasing latent capacity (discussed in more detail in a later section). This caused a noticeable reduction in supply ventilation because lower AHU flow effectively reduced the return plenum pressure and flow at the ventilation supply duct. This again illustrates the inherent variability in parameters affecting the g2-k system setup. It is suspected that the occupant replaced dirty return air filters around this time, which further reduced system pressure and also contributed to lower supply outdoor airflow than expected.

Table 3-13. SS2 Monthly Performance of the Ventilation System

Full Days of Data	Month	Percentage of 62.2 Met	Daily Minimum 62.2 Met	Daily Maximum 62.2 Met	Supply Duct Contribution	Exhaust Fan Contribution
21	Aug.	100.6%	91.4%	128.5%	97.3%	2.7%
20	Sept.	122.5%	96.3%	134.8%	99.3%	0.7%
28	Oct.	136.9%	119.0%	160.0%	98.6%	1.4%
23	Nov.	132.3%	65.1%	157.4%	98.1%	1.8%
	Dec.	Insufficient data				
31	Jan.	86.1%	25.9%	116.2%	100.0%	No Fan Data
14	Feb.	120.5%	99.3%	130.1%	88.3%	11.7%
31	March	121.2%	87.5%	138.1%	85.3%	14.7%
30	April	79.2%	64.4%	137.9%	93.1%	6.9%
31	May	83.5%	66.6%	101.3%	95.0%	5.0%
22	June	97.0%	92.1%	102.4%	100.0%	0.0%

Data collection at SS1 began near the end of June 2016 with the residence already occupied by a family of four. Problems with the pressure sensor used in tandem with the IRIS damper to determine real-time outside airflow prevented measurements until late July. In contrast to SS2, this home's thermostat setting was several degrees warmer, resulting in a reduced cooling load, less minisplit run time, and greater exhaust fan run time necessary to achieve the design ventilation flow. Figure 3-32 illustrates ventilation performance during the course of several months of monitoring using daily average values. Shortly after monitoring began, the home experienced a problem with the supply ventilation damper, which appeared to progressively restrict flow through the duct, because there was no apparent problem with the measurement sensors. Toward the end of October and into November, a data collection problem resulted in some missing data. Although data collection resumed in November, no flow through the supply ventilation duct was recorded. It was later confirmed that the supply ventilation airflow damper was stuck closed. All monitoring was suspended in this home in December at the owner's request.



Note: Exhaust fan flow based on one-time flow measurement in proportion to measured power

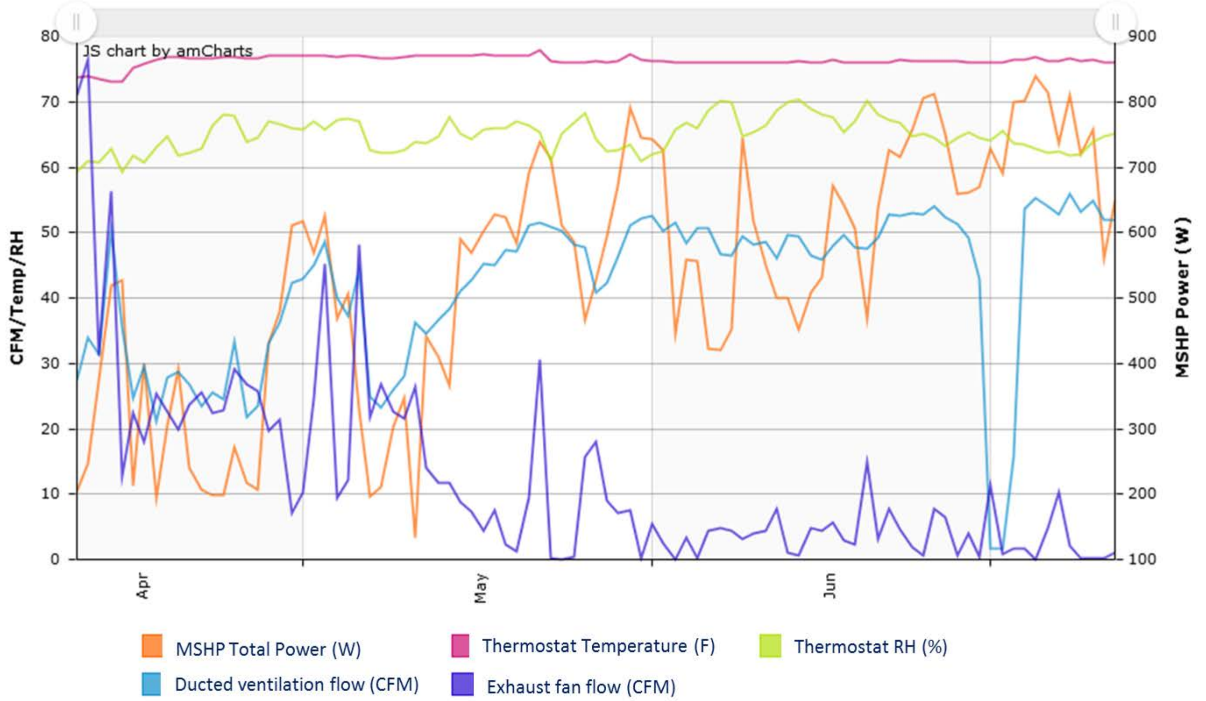
Figure 3-32. SS1 ventilation system performance post-occupancy, daily average values

Table 3-14 shows monthly ventilation performance in SS1. In contrast to SS2, this home was under ventilated on a monthly basis with a much larger contribution of total ventilation air coming from the exhaust fan. Figure 3-32 and Table 3-14 both illustrate the steady reduction in supply duct ventilation air that began in mid-September and continued into October. Starting in early November, flow measurements through the supply duct indicate that the damper was closed, which was confirmed in mid-December when monitoring was suspended. It appears that the g2-k continued to operate during this period as if the damper was operational and using the exhaust fan as if air was flowing through the supply ventilation air duct.

Table 3-14. Site SS1, Monthly Performance of the Ventilation System

Full Days of Data	Month	Percentage of 62.2 Met	Daily Minimum 62.2 Met	Daily Maximum 62.2 Met	Supply Duct Contribution	Exhaust Fan Contribution
31	Aug.	95.9%	70.1%	124.5%	60.0%	40.0%
30	Sept,	87.8%	55.1%	153.5%	35.0%	65.0%
22	Oct,	80.1%	46.1%	135.4%	31.3%	68.7%
26	Nov,	89.9%	43.0%	142.5%	9.8%	90.2%
12	Dec,	63.6%	20.4%	81.2%	0.0%	100.0%

A third home, SS3, which was occupied in April 2017, was monitored in Sarasota near the end of the study. The thermostat setting and minisplit heat pump power use in this home were similar to those of SS1 with a reduced cooling load and less minisplit heat pump run time than SS2. Figure 3-33 illustrates ventilation performance during three months of monitoring using daily average values. High levels of exhaust ventilation (purple) occurred in April and early May coincident with lower minisplit heat pump power and supply ventilation (orange and blue, respectively). As the weather warmed after mid-May and minisplit heat pump power increased, the supply duct contributed more ventilation, whereas exhaust fan use decreased.



Note: Exhaust fan flow based on one-time flow measurement in proportion to measured power

Figure 3-33. SS3 ventilation system performance post-occupancy, daily average values

Table 3-15 shows monthly ventilation performance in SS3. Data show that this home was consistently over-ventilated by 15% to 16% on a monthly basis. A clear trend of exhaust ventilation giving way to a larger proportion of supply ventilation occurred from April through June, when increasing cooling loads and minisplit heat pump use is expected.

Table 3-15. SS3 Monthly Performance of the Ventilation System

Full Days of Data	Month	Percentage of 62.2 Met	Daily Minimum 62.2 Met	Daily Maximum 62.2 Met	Supply Duct Contribution	Exhaust Fan Contribution
30	April	116.2%	97.1%	178.1%	43.4%	56.6%
31	May	114.5%	97.8%	148.1%	73.3%	26.7%
20	June	115.6%	107.3%	119.7%	92.9%	7.1%

3.6.3 South Sarasota Set Point Temperatures and Resulting Room-to-Room Temperature Distribution

Operational characteristics among the SSHFH homes differ, primarily in preference for indoor temperature. SS1, and later the replacement home SS3, kept indoor temperatures in the range from 75° to 78°F, whereas SS2 kept colder indoor temperatures throughout the year, in the range from 68° to 72°F. These numbers are reflective of cooling set points because winter data are not available on these homes.

Given that these homes have fully ducted supply and return systems, researchers expected thermal distribution throughout the house to be adequate, meaning that they were within the ±3° of the main body guideline set forth in ACCA Manual RS previously discussed. Researchers measured airflow at each supply and return register at site SS1 with the system forced into high capacity by the HVAC contractor. Results are shown in Table 3-16.

Table 3-16. SS1 Room-to-Room Supply and Return Airflow

	Supply Flow (cfm)	Return Flow (cfm)
Master bedroom	61	57
Master bathroom	25	No return
Master closet	26	No return
BR1	51	60
BR2	35	78
Hall bathroom	32	No return
Kitchen and living room	117 + 186 = 303	
Central hall		276
Total	533	471

Bedroom temperature in SS1 was within the ±3°F difference (red lines in the next three figures) compared to the main body, except for 12 hours (0.3%) that were scattered during 5 days from July–September, with only 2 days with temperatures higher than 3.5°F, as shown in Figure 3-34. The SS2 bedroom temperatures were out of the ±3°F difference slightly more often, but they did not exceed 3% of hours in any bedroom, as shown in Figure 3-35. Bedroom 2 in SS3 saw the most out-of-range hours of the Sarasota homes, with 86 hours (4.5%) at more than a 3°F temperature difference (Figure 3-36); however, all but three of those hours fell to less than 4°F warmer than the main body, as shown in Figure 3-37. In short, all three of the fully ducted homes built by South Sarasota provided an evenly conditioned home for the vast majority of the monitoring period.

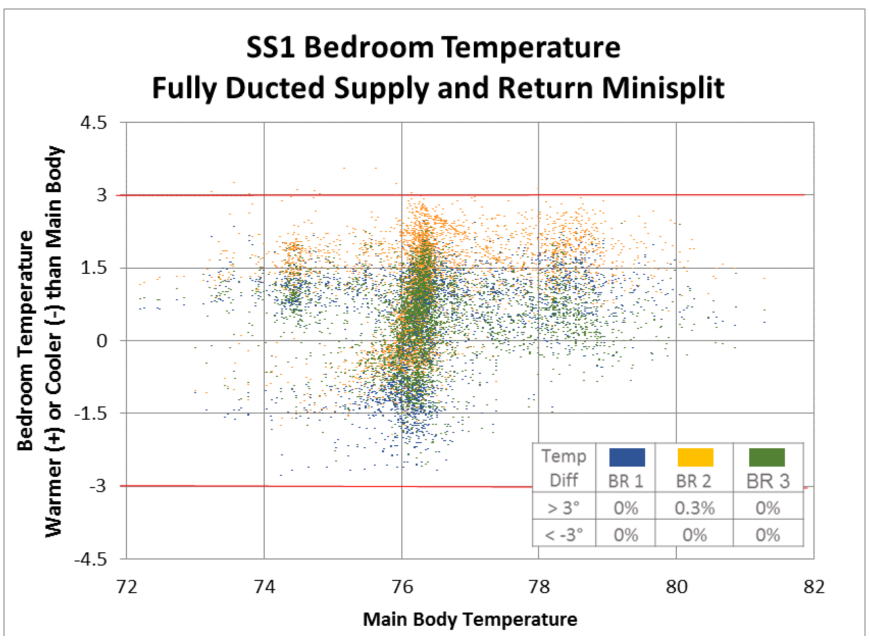


Figure 3-34. SS1 average hourly temperature differences between bedrooms 1 (master), 2, and 3 and the main body of the house

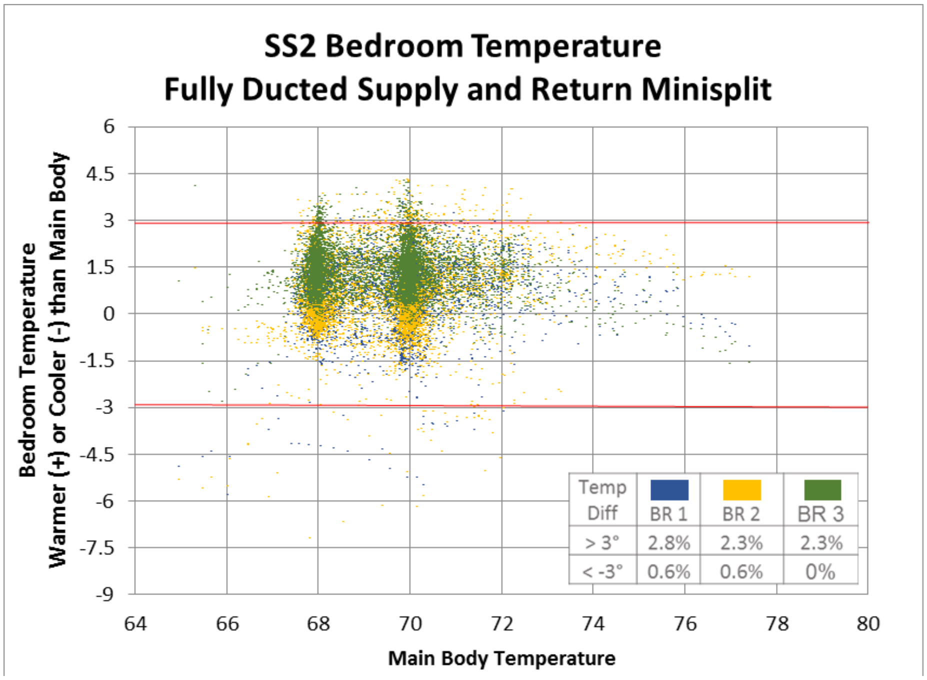


Figure 3-35. SS2 average hourly temperature difference between bedrooms 1 (master), 2, and 3 and the main body of the house

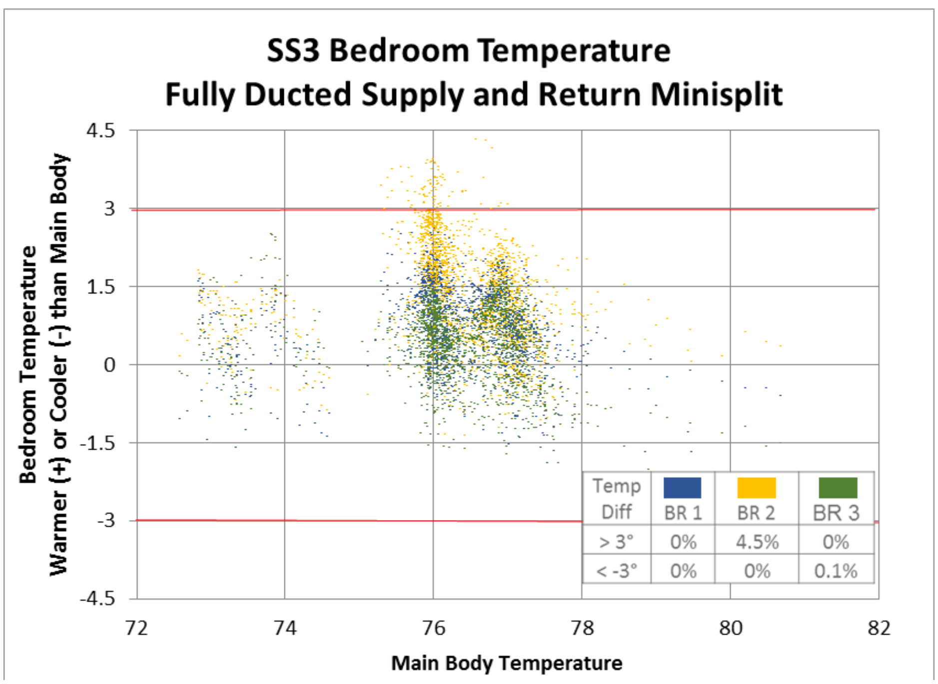


Figure 3-36. SS3 average hourly temperature difference between bedrooms 1 (master), 2, and 3 and the main body of the house

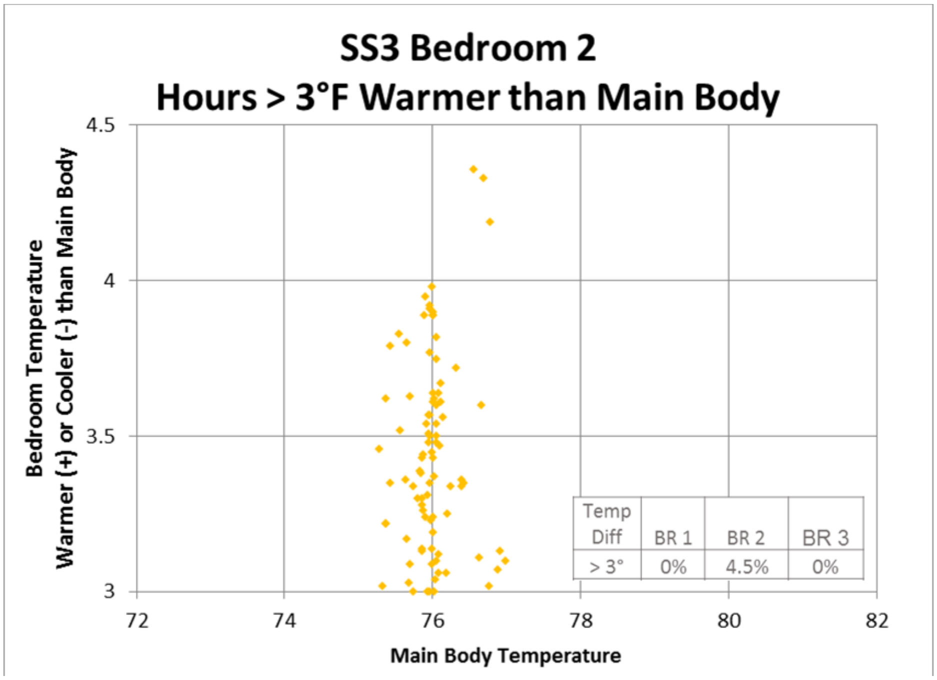


Figure 3-37. SS31 periods when the front and back bedroom temperatures were more than 3°F warmer than the main body

3.6.4 Indoor Relative Humidity

Tabulation of average hourly average indoor RH is binned into RH ranges and shown in Figure 3-38 for SS1 and SS3 and Figure 3-39 for SS2. Data are averaged from sensors located near the thermostat and in each bedroom. Average monthly indoor temperature is indicated in each monthly bar. As shown, indoor RH is more than 60% for a significant fraction of the months for which data are available. We observed 57% of hours when RH exceeded 60% in SS1 for the monitoring period from July–December, 86% of hours when the RH exceeded 60% in SS3 for the monitoring period from April–June, and 47% of hours when the RH exceeded 60% in SS2 for the monitoring period from August–June. Indoor RH was largely less than 60% in SS1 for July–September 2016, months with warmer outdoor temperatures that require greater heat pump run time. This trend extended into October–November 2016 for SS2, where lower indoor set point temperatures (average less than 73°F) generated more heat pump run time during these months than SS1; however, the low-load period of December 2016–March 2017 showed significant hours that exceeded 65% in SS2 as well as for December in SS1. After March 2017, SS2 indoor RH appears to decline, as outdoor temperatures increased and heat pump run times became longer approaching the summer of 2017; however, high indoor RH existed in SS3 during this time, likely because of a warmer set point generating less run time.

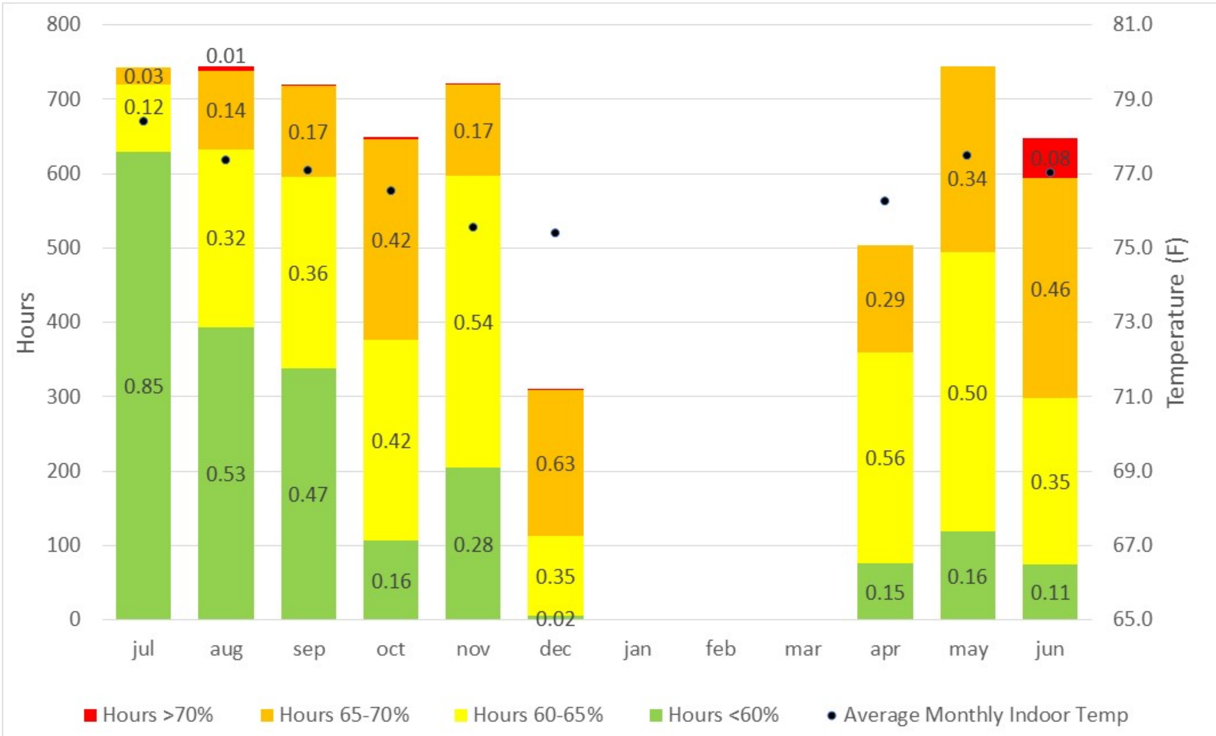


Figure 3-38. Average hourly RH and monthly average temperature in SS1 from July–December and SS3 from April–June

Figure 3-39. SS2 average hourly RH and monthly average temperature

3.6.5 Variable Heat Pump System Operation

Taking a closer look at heat pump system operation in the SSHFH homes shows an operational trend similar to what we saw resulting in higher indoor RH in the SEVHFH homes. Although the variable-capacity nature of the heat pump was expected to enable it to run relatively continuously during a daily cycle varying output with load, the data show that for months other than peak summer months, the heat pump in SS2 exhibited cycling behavior during overnight hours, resulting in highly variable SAT, and progressively increasing indoor RH while indoor temperature remained stable. A sample from April exhibiting this dynamic is shown in Figure 3-40. Note that the instantaneous high SAT shown with each cycle resulted from the integrated supply mechanical ventilation generating an initial blast of warm, outdoor air picked up by the supply air sensor prior to the heat pump coil generating typical supply air conditions. The indoor RH steadily decreased when the heat pump returned to consistent operation during the day, once outdoor conditions created sufficient cooling load on the house.

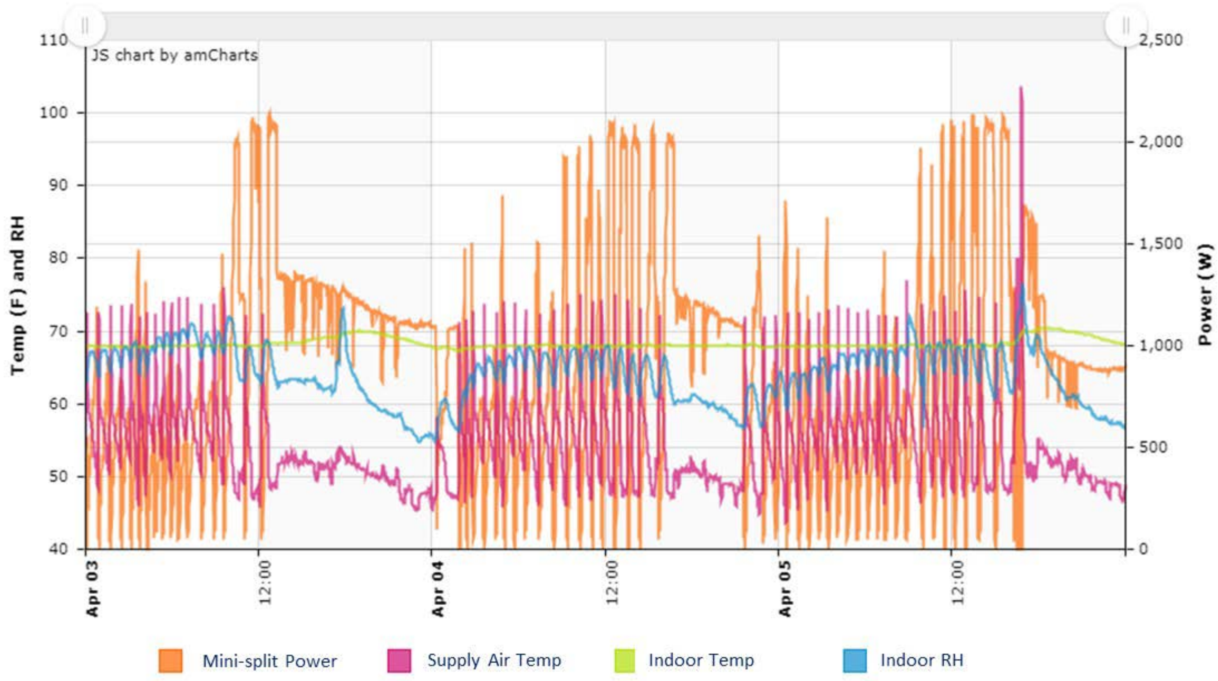


Figure 3-40. SS2 indoor temperature and RH, along with minisplit power and SAT

A number of operational characteristics of the minisplit heat pumps are customizable upon installation, including supply airflow, which can be configured to vary over a narrow range. After consultation with the air-conditioning contractors, it was learned that the systems were initially set up to deliver relatively higher supply airflows, and hence higher sensible cooling capacities, in an attempt to ensure quick response when occupants desired to cool the homes quickly. Essentially, the supply airflow was fixed relatively "high" regardless of the speed of the outdoor unit. It was postulated that this higher fixed sensible capacity might limit the latent performance of the units and might not be necessary except in extreme situations of high occupancy, such as during a party.

Mechanical contractors adjusted the heat pumps to allow supply airflow to vary in closer proportion to the speed of the outdoor unit on April 5, 2017, just before a late season cold front significantly reduced outdoor dew point, as shown in Figure 3-41.

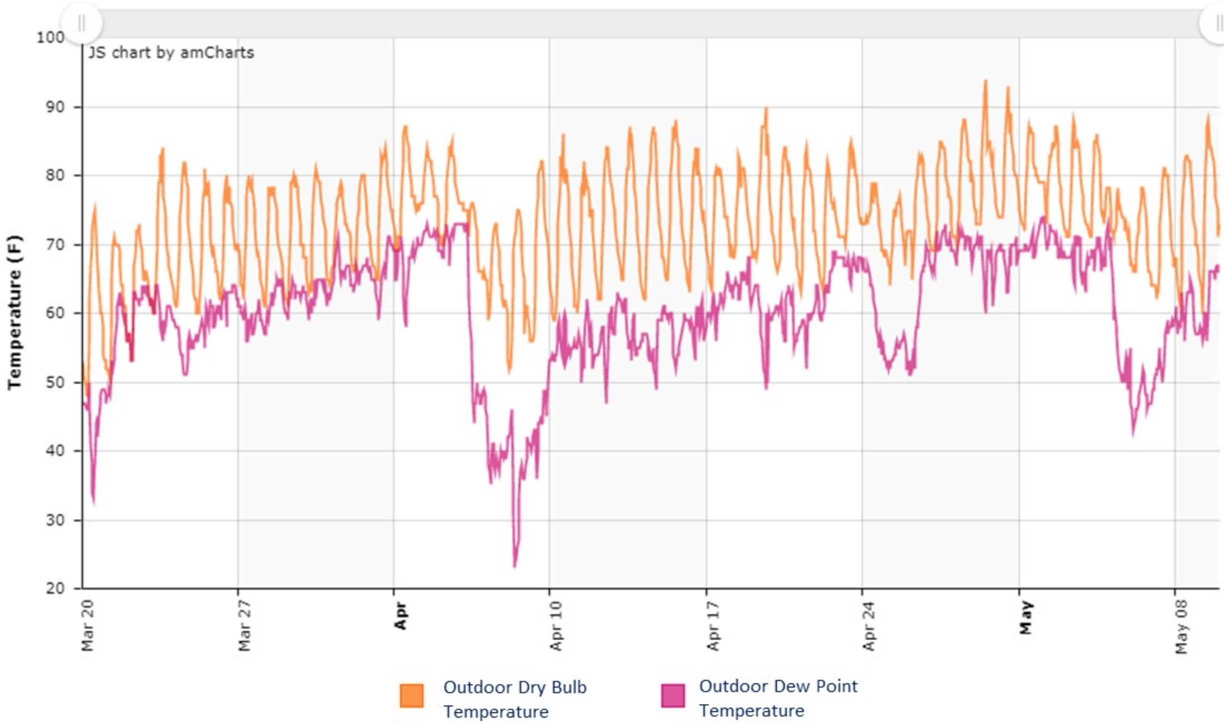


Figure 3-41. Ambient temperature and dew point from nearby Sarasota/Bradenton Airport National Weather Service Station

A minor effect occurred from this adjustment, and data before and after the change are shown in Figure 3-42. A reduction in fan speed (represented in the plot as normalized amp draw of the indoor fan coil, gray line) occurred after the change late in the day on April 5, and a bit of a reduction in SAT (pink) after the change, especially when the weather warmed back up on April 7 and April 8. Total minisplit power (not shown) was also reduced; however, the heat pump continued to exhibit similar overnight cycling behavior, and as a result indoor RH responded to system operation in a fashion similar to that observed before the change. Although the change might have improved the heat pump's latent capacity while it was running, it did not create more consistent operation during the overnight, low-load conditions, and high indoor RH during this time continued. Note that the instantaneous high SAT that occurred with each cycle resulted from the integrated supply mechanical ventilation generating an initial blast of warm, outdoor air picked up by the supply air sensor prior to the heat pump coil generating typical supply air conditions.

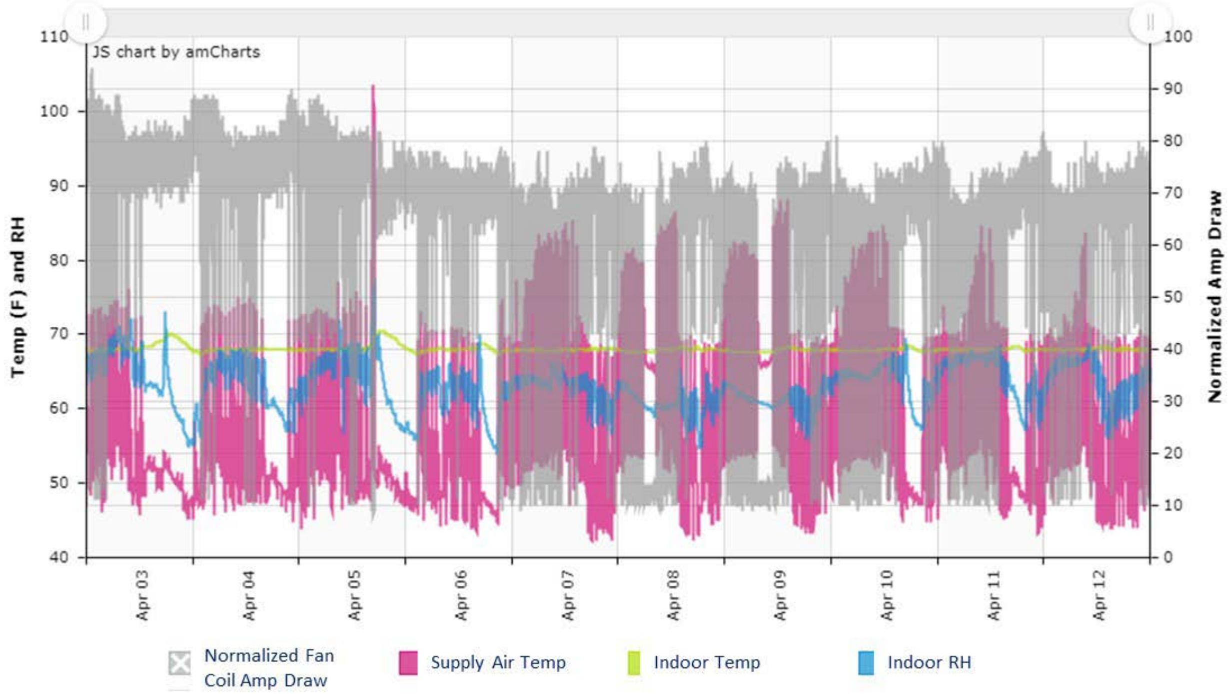


Figure 3-42. SS2 indoor temperature and RH, along with minisplit SAT and normalized amp draw of the indoor fan coil

It is interesting to note conditions that cause the diurnal indoor RH profile to reverse. Referring back to Figure 3-41, beginning April 27 outdoor dew point returned to a consistently high springtime value of 70°F, and outdoor temperatures reached daytime highs in the 90s. As shown in Figure 3-43, these outdoor conditions, and the occupant's desire for 68°F indoor temperature, stressed the capacity limits of the heat pump, and it was not able to consistently maintain set point (green line). As the indoor temperature ran away from the desired set point on April 29, the heat pump entered a mode in which it prioritized sensible cooling with higher coil airflow. As a result, the SAT warmed and indoor RH (blue) increased. Note that another factor increasing indoor RH during these conditions is the integrated supply mechanical ventilation that delivers higher outdoor airflows (in the range of 70 cfm) when the indoor unit fan speed is high. In any event, as cooling load on the building decreased coincident with sundown, the heat pump ran rather continuously overnight in an attempt to cool the building back down to the desired set point temperature, resulting in a *decrease* in indoor RH overnight, the opposite of what occurred during previous data periods when the set point was easily maintained. A drop in outdoor temperature and dew point resulted in the heat pump returning to cycling conditions overnight, with high overnight RH.

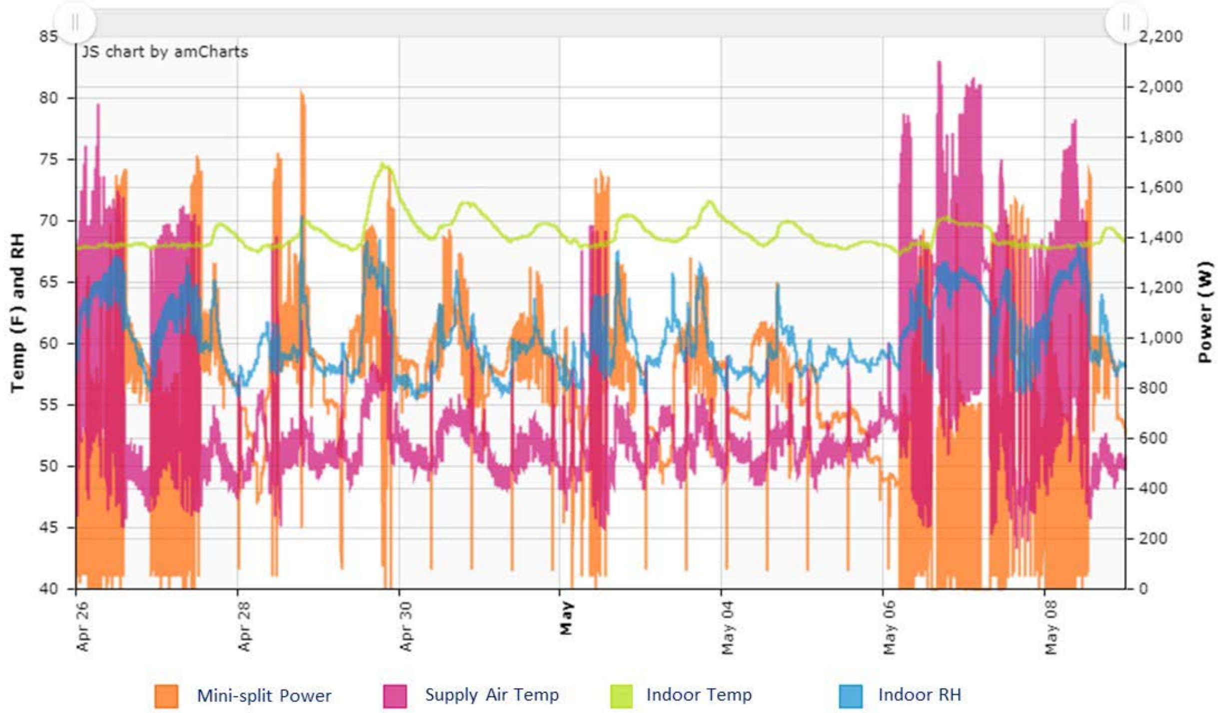


Figure 3-43. SS2 indoor temperature and RH, along with mini-split power and SAT

As shown in Figure 3-44, during this exact same time period, SS3 exhibited the more typical overnight cycling behavior, with indoor RH higher overnight and lower during the day because the more modest desired indoor temperature of 75°F can mostly be achieved.

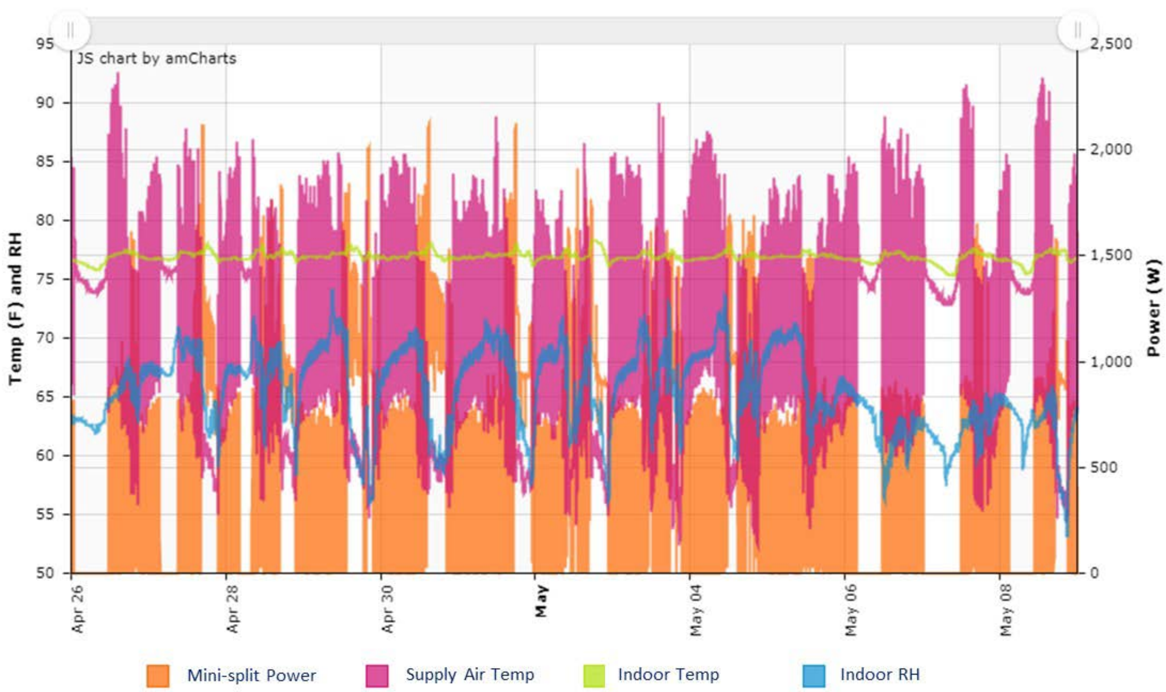


Figure 3-44. SS3 indoor temperature and RH, along with minisplit power and SAT

Around the middle of May, SS2 increased their indoor set point from 68° to 70°F, and the heat pump continued to exhibit the overnight cycling behavior as the summer of 2017 warmed up. With more cooling load generating more heat pump run time, average RH, along with daily peak RH, was reduced. As shown in Figure 3-45, the heat pump continued to struggle to maintain the desired 70°F set point, although the system was able to recover from excursions quickly such that indoor RH could still be reduced during the day. As the heat pump exited its overnight cycling mode, a few higher power blasts (orange) generated a lower SAT that provided an immediate reduction to indoor RH; however, a secondary peak in indoor RH was coincident with an increasing coil airflow and SAT as the heat pump tried to deliver enough sensible cooling to prevent an increase in indoor temperature. As cooling load decreased later in the afternoon and into the evening, lower airflow and hence SAT could be delivered, and indoor RH decreased to the point that indoor temperature returned to a steady-state condition, and overnight cycling of the heat pump returned.

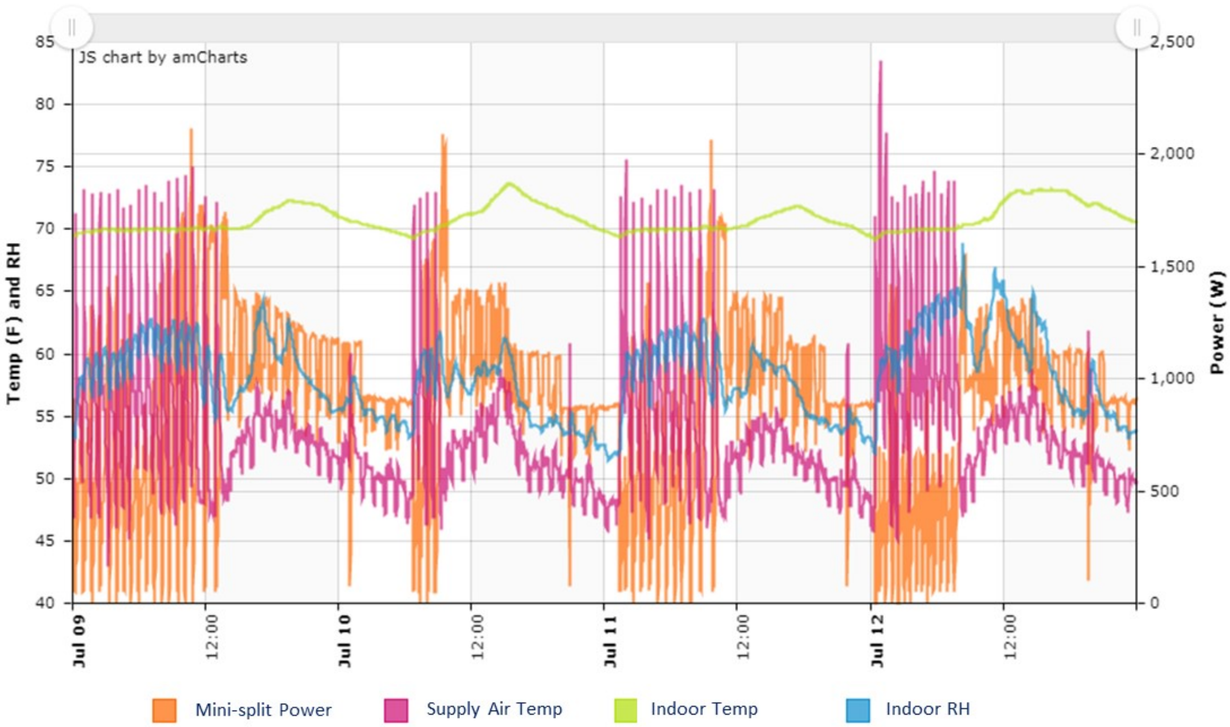


Figure 3-45. SS2 indoor temperature and RH, along with minisplit power and SAT

4 Conclusions and Recommendations for Comfort System Manufacturers

This project collected data in a mix of laboratory home and occupied field-test home sites to evaluate the performance of variable-capacity comfort systems. Although the systems tested represent a small share of the current marketplace, their ability to maintain whole-house comfort in low-load, mechanically vented homes is of interest because integrated solutions for enhanced moisture control are needed for this growing market. Although the research conducted and associated discussion focuses on inverter-driven, variable-capacity systems, much of the same proposed system enhancements could be applied to more conventional two-speed systems with variable-speed ECM AHU motors to improve their latent performance at part load.

The research answered the following research questions:

1) How does the total space-conditioning energy consumption (cooling plus dehumidification) of a Unico SDHV variable-capacity system compare to a centrally ducted fixed-capacity SEER 13 system and a variable-capacity SEER 22 system when indoor RH is maintained at less than 60%.

The iSeries SDHV system and supplemental dehumidification energy was 8.2% lower than a SEER 13 system including supplemental dehumidification, and the SDHV system (standard cool) and supplemental dehumidification energy was 16.7 % more than a centrally ducted SEER 22 system including supplemental dehumidification during typical summer conditions. Inadequate heating data were available from previously tested systems for significant comparison. It seems that a higher efficiency centrally ducted system could be offered with the inverter outdoor unit given the variable-capacity control. SEER 18 can be acquired by using ductless minisplits with the outdoor unit.

Although it was no surprise that a SEER 22 system used less total cooling energy than a SEER 14 system, it was surprising to see that the tested iSeries indoor fan unit daily energy use was less than the daily energy use of a SEER 22 indoor fan unit. The daily average measured fan efficiency (cfm/w) of the iSeries SDHV unit was 43% lower than the SEER 22 fan. Based simply on fan efficiency, one would expect the daily average indoor unit energy use to be higher, but the SDHV daily average AHU energy was 22% lower than the SEER 22 system. This is because of operation at lower average fan wattage caused by an average airflow rate that was approximately 53% less. Two other causes are that the daily AHU energy use included the standby energy use, which was 68% less in the SDHV system, and the SDHV system operated approximately 1 hour less per typical summer day. The SDHV system met cooling load with lower flow rates and less run time because the supply air was 9°F colder than the SEER 22 system.

This testing found that the SDHV system can have very effective dehumidification performance during low cooling load periods. The iSeries system maintained the lowest average indoor humidity compared to other systems in the MHL previously tested under the same test conditions. The iSeries system dry mode was able to maintain indoor humidity less than 60% in all bedroom and central living areas without supplemental dehumidification. Very infrequent dehumidifier operation occurred in the standard cooling mode. Of 4,512 standard cool test hours, the dehumidifier operated only 0.28% of the time. Dehumidifier operation was most likely to be between midnight and 10:00 a.m. The iSeries unit had a house average indoor humidity of 46.7% RH during all standard and dry mode testing (286 days); however, even though the utility room average RH was 45.8%, it exceeded 60% RH at a frequency of 6% of all testing hours in the dry mode and at a frequency of 10% in the standard cooling mode. Continuous whole-house mechanical ventilation was the reason the utility room experienced occasional elevated humidity.

Considering the SDHV system's ability to maintain a relatively low average indoor RH compared to other, higher efficiency variable-capacity systems previously tested, a simulation was conducted to evaluate the combined energy impacts of space cooling and supplemental dehumidification at different indoor RH targets. An indoor RH set point of 40% results in a SEER 22 cooling system using more energy for supplemental dehumidification than for cooling on an annual basis. At an indoor RH set point of 50%, supplemental dehumidification represents 35% of the total annual space-conditioning energy use for the MHL with a SEER 22 system. With some improvement in the dry mode control, the iSeries SDHV SEER14 system should be able to maintain RH at or less than 50% all hours of the year based on test results. An annual simulation of a SEER 22 heat pump with supplemental dehumidifier set to maintain 50% RH used 23% more space-conditioning energy than a SEER 14 heat pump needing no supplemental dehumidifier to maintain 50%. More rigorous simulation work and testing is needed to confirm the simple simulation results presented here, but it points out that dehumidification energy should not be overlooked in household energy use and equipment selection.

Dealing with elevated humidity in a remote area where mechanical ventilation is supplied poses a challenge even for a system with good dehumidification performance. The utility room was less than 60% RH whenever the cooling system was running. This was true even during the lowest load periods (and lowest delivered capacity) as long as the system was providing some cooling. But if the cooling system was off long enough, RH in the room began to increase (from continuous moist outdoor air supply) until the next cooling cycle. This prompted further research effort to lower SHR during the dry mode. Lowering SHR increases the latent removal and reduces sensible removal, thereby increasing run time and limiting overcooling.

Efforts to decrease SHR in the dry mode did not work. We discovered that the “Very Low” airflow rate was approximately two times higher than that called for by the control algorithm target. This resulted in a higher SHR than desired during the lowest cooling load periods and lowest stage of dry cooling. The higher airflow at the very low target was caused by inaccuracies in a control algorithm embedded in the system. The airflow was as expected at low to high levels, but it deteriorated as the commanded airflow drifted down to very low.

2) How well is indoor temperature controlled with transfer fans compared to fully ducted systems?

Data from the homes with multisplit/transfer fan systems show mixed results, but they do show that the approach can perform on par with a fully ducted system. The fully supply and return ducted minisplit homes served as one baseline, and data showed decent but not perfect temperature distribution throughout rooms in the home, as evaluated with ACCA Manual RS. The worst-case bedroom exceeded the ±3°F threshold 4.5% of the time, with other bedrooms exceeding the threshold much less than that. All hours exceeding the threshold were less than ±4°F.

Testing with the iSeries SDHV variable-capacity system served as another baseline, and data showed that that system also provided good but not perfect thermal distribution as evaluated with ACCA Manual RS. The northeast bedroom had the greatest frequency of overcooling, to a slightly more than 3°F difference from the main living room. This occurred in 2% of the test hours during standard cooling mode. All other rooms under all tests had a frequency of exceeding a 3°F limit less than 0.5% of the time. The greatest distribution challenge occurred in the standard mode during the hottest sunny weather. The distribution could have been easily improved by limiting airflow from one of the northeast bedroom supplies by damper control and by adding an additional supply branch into the master bedroom.

The occupied transfer fan home SEV1 exceeded the ACCA Manual RS threshold only 2.1% of the time. The hours were largely confined to two months in particular, with no clear explanation for the departure. Most hours were less than ±4°F, with very few hours exceeding that value. Temperature distribution differences from one room to another were much poorer in one of the bedrooms in the other transfer fan home, SEV2, but researchers suspect that this is largely a result of the homeowner’s desire for warmer temperatures and actions to turn off transfer fans to achieve desired comfort. These results show that (1) perfect achievement of ACCA Manual RS can be difficult even with fully ducted systems because exact occupant behavior and diversity of space-conditioning loads are typically not known during design, (2) minor departures from ACCA Manual RS are acceptable because no comfort complaints were received from any occupied homes, and (3) ductless systems with transfer fans can be configured to provide temperature distribution on par with fully ducted systems.

3) Can the achievement of design mechanical ventilation rates be ensured when integrating supply ventilation with a variable-capacity minisplit system?

Yes, the Air Cycler g2-k controller was able to achieve ASHRAE 62.2-2010 ventilation rates when integrated with the variable-capacity system; however, determining proper inputs to the g2-k system to achieve this result was more complex compared to a fixed-capacity system. Because the Air Cycler g2-k ventilation system required a conventional thermostat to operate, a thermostat interface was used that enabled the minisplit heat pump to also operate with a conventional thermostat. The thermostat interface limits some of the heat pump capacity variation, forcing it to operate at fewer discrete speeds. That in turn does limit some variation in supply mechanical ventilation flow; however, considerable variation in indoor fan speed of the heat pump remains, and estimating an average or typical fan speed, and hence supply ventilation flow, is required as an input to the g2-k system. Although there was some difficulty arriving at appropriate numbers, with full knowledge of heat pump installation parameters such as airflow settings, it is possible to arrive at g2-k settings that will ensure design ventilation flows are achieved; however, ventilation system component failures are always possible, as shown with the supply ventilation air damper at SS1. A consideration for ventilation system manufacturers is to include fault-detection features that alert homeowners or contractors to failures that result in unexpected reductions in airflow.

4) What variable-capacity cooling system operational characteristics and patterns are observed in the collected data that might assist manufacturers with improved indoor RH control as they refine existing equipment and develop new products?

Although the Unico SDHV system had effective dehumidification performance during low-load periods, the minisplit and multisplit systems experienced challenges in the occupied homes. With the exception of multisplit home SEV2, which maintained exceedingly warm indoor temperatures, indoor RH was exceeded 60% much of the time in all homes. Minisplit home SS2, which maintained exceedingly cool indoor temperatures, largely controlled RH during the summer months, but RH exceeded 65% nearly 50% of the hours during the low-load winter months. Although the elevated indoor RH is not a health or durability concern, it does not show the achievement of desired comfort metrics. Note that no comfort complaints were received by the Habitat for Humanity affiliates from any of the homeowners. At the onset of their second summer, SEV1 exhibited significantly improved RH control. Although the exact circumstances are unknown, it does seem related to a change in variable-capacity system operational characteristics, and it points to a correlation between certain characteristics and improved indoor RH control.

To address high latent loads in low-load homes, extended run time of the cooling system is required during low-load hours. Although variable-capacity systems have the capability to do so, frequently they do not run consistently at the very low end of their operational capacity range. Instead, as shown during low-load conditions occurring on overnight summer hours, many systems exhibit cycling behavior, which inhibits moisture removal. In addition, large variations in system operation are evident at times during higher load conditions because units often change speeds in response to loads, rather than exhibiting steady operation. This can result in higher SAT not conducive to good indoor RH control.

In addition to steady operation, delivery of low SHR, and hence cold-supply airflows during operation is required to control indoor RH. Although variable-capacity systems have the capability to vary SHR, many systems tested will often opt for higher SHR in efforts to efficiently control indoor temperature. To deliver low SHR, and not result in excessive overcooling during low-load periods, coil airflow must be as low as possible maintained at the lowest possible cooling capacity. In the occupied homes, we found periods when indoor RH was well controlled corresponding to continuous run time of the heat pumps at low speeds (airflows). In the case of the SDHV system, the design enabled low coil airflow and hence lower SHR during much of its operation across its capacity range, and its ability to control indoor RH is evident.

Data show that manufacturers should even consider *extending* their standard operating airflow ranges and developing special low-airflow modes that are essential for achieving lowest rated capacity, improving dehumidification, and increasing overall system efficiency in low-load homes. As is the case with many variable-capacity systems, an operational dry-mode often exists that attempts such an operational configuration; however, results using the dry mode to control indoor RH while also limiting overcooling are mixed. Some limitations in improving dehumidification might be directly related to distribution airflow accuracy determined by on-board equipment control, particularly at the very lowest flow rates.

Improving the low-flow accuracy and control algorithms will improve dehumidification in variable-capacity systems by lowering coil airflow to the point at which low capacity and long run times can be achieved without overcooling. The next improvement would be a smarter dry mode not solely controlled by sensible temperature. This would require a humidity sensor as a control algorithm feedback. A smart dry mode would also be able to move out of the dry mode into the standard high-efficiency cooling mode when RH levels are low enough. The iSeries dry mode algorithm demonstrates this ability, but it would be improved by an RH-based measurement. In consultation with Mitsubishi, the inclusion of an RH-based measurement to invoke the equivalent of an enhanced dry mode was considered; however, in the SSHFH homes, the g2-k ventilation system and thermostat interface created a physical wiring interference, preventing a humidistat to be configured into the system controls.

References

Air Conditioning Contractors Association of America. 1997. ACCA Manual RS.

ASHRAE. 2010. Standard 55-2010. "Thermal Environmental Conditions for Human Occupancy".

Baskins, Evelyn, and Edward A. Vineyard. January 2003. "Thermal Comfort Assessment of Conventional and High-Velocity Distribution Systems for Cooling Season", *ASHRAE Transactions* (109) Part 1, Chicago, CH-03-7-5.

Building America Program. 2015. "Building America Research to Market Plan". Building America Program, DOE-EE 1285. https://energy.gov/sites/prod/files/2015/11/f27/Building%20America%20Research%20to%20Market%20Plan-111715.pdf

Brown, S., B. Thornton, and S. Widder. 2013. "Review of Residential Low-Load HVAC Systems". Richland, WA. Pacific Northwest National Laboratory. PNNL-23017.

Cummings, J., C. Withers, and J. Kono. 2015. "Cooling and Heating Season Impacts of Right-Sizing of Fixed- and Variable-Capacity Heat Pumps With Attic and Indoor Ductwork." Golden, CO. National Renewable Energy Laboratory.

Cummings, J. and C. Withers. 2011. "Energy Savings and Peak Demand Reduction of a SEER 21 Heat Pump vs. a SEER 13 Heat Pump with Attic and Indoor Duct Systems." Golden, CO. National Renewable Energy Laboratory.

Fairey, Philip, Danny Parker, Robin Vieira, and Eric Martin. 2014. "Vent Right and Then? Mechanical Ventilation, Dehumidification and Energy Use in Humid Climates." Cocoa, FL. Florida Solar Energy Center. FSEC-PF-460-14.

Harriman III, Lewis G., Dean Plager, and Douglas Kosar. November 1997. "Dehumidification and Cooling Loads from Ventilation Air." *ASHRAE Journal*; pp. 37-45.

Henderson H.; Rudd, A. 2014. "Energy Efficiency and Cost Assessment of Humidity Control Options for Residential Buildings." *ASHRAE Transactions*, (120) Part 1. NY-14-013 (RP-1449).

Munk, J., A. Gehl, and R. Jackson. 2012. "Performance of Variable Capacity Heat Pumps in a Mixed Humid Climate." Oak Ridge, TN. Oak Ridge National Laboratory. ORNL/TM-2012/17

Parker, Danny, et al. September 2016. "Flexible Residential Test Facility: Impact of Infiltration and Ventilation on Measured Cooling Season Energy and Moisture Levels". Cocoa, FL. Florida Solar Energy Center. FSEC-CR-2038-16

Poerschke, A. 2017. "Measured performance of a varied Airflow Small-Diameter Duct System." Golden, CO. National Renewable Energy Laboratory.

Rudd, A.; Henderson, H. 2007. "Monitored Indoor Moisture and Temperature Conditions in Hot-Humid US Residences." *ASHRAE Transactions* (113) Part 1.

Rudd, A. 2013. *Expert Meeting: Recommended Approaches to Humidity Control in High Performance Homes.* Somerville, MA: Building Science Corporation.

Rudd, Armin, Joseph Lstiburek, Kohta Ueno. February 2005. "Residential Dehumidification Systems Research for Hot-Humid Climates." Golden, CO. National Renewable Energy Laboratory. NREL/SR-550-36643.

Roth, K., N. Sehgal, and C. Akers. 2013. "Ductless Mini-Split Heat Pump Comfort Evaluation." Golden, CO. National Renewable Energy Laboratory.

Shirey III, Don B., Hugh I Henderson Jr. April 2004. "Dehumidification at Part Load." *ASHRAE Journal*,.

Sutherland, K., D. Parker, and E. Martin. August 2016. "Evaluation of Mini-Split Heat Pumps as Supplemental and Full System Retrofits in a Hot Humid Climate." 2016 ACEEE Summer Study on Energy Efficiency in Buildings.

Ueno, K. and H. Loomis. 2015. "Long-Term Monitoring of Mini-Split Ductless Heat Pumps in the Northeast". Golden, CO. National Renewable Energy Laboratory.

Widder, S., E. Martin, D. Chasar, J. McIlvaine, B. Amos, K. Fonorow, and S. Widder. February 2017. "Comparative Performance of Two Ventilation Strategies in a Hot Humid Climate." Golden, CO. National Renewable Energy Laboratory.

Withers, Charles R. Jr., and Jeff Sonne. June 2014. "Assessment of Energy Efficient Methods of Indoor Humidity Control for Florida Building Commission Research," Florida Solar Energy Center.

Withers, C., J. Cummings, and B. Nigusse. 2016(a). "Cooling Seasonal Energy and Peak Demand Impacts of Improved Duct Insulation on Fixed-Capacity (SEER 13) and Variable-Capacity (SEER 22) Heat Pumps." Golden, CO. National Renewable Energy Laboratory.

Withers, C. 2016(b). "Energy Efficient Management of Mechanical Ventilation and Relative Humidity in Hot-Humid Climates" Golden, CO. National Renewable Energy Laboratory.

U.S. DEPARTMENT OF
ENERGY

Office of
ENERGY EFFICIENCY &
RENEWABLE ENERGY

For more information, visit: buildingamerica.gov

DOE/EE-1702 · February 2018