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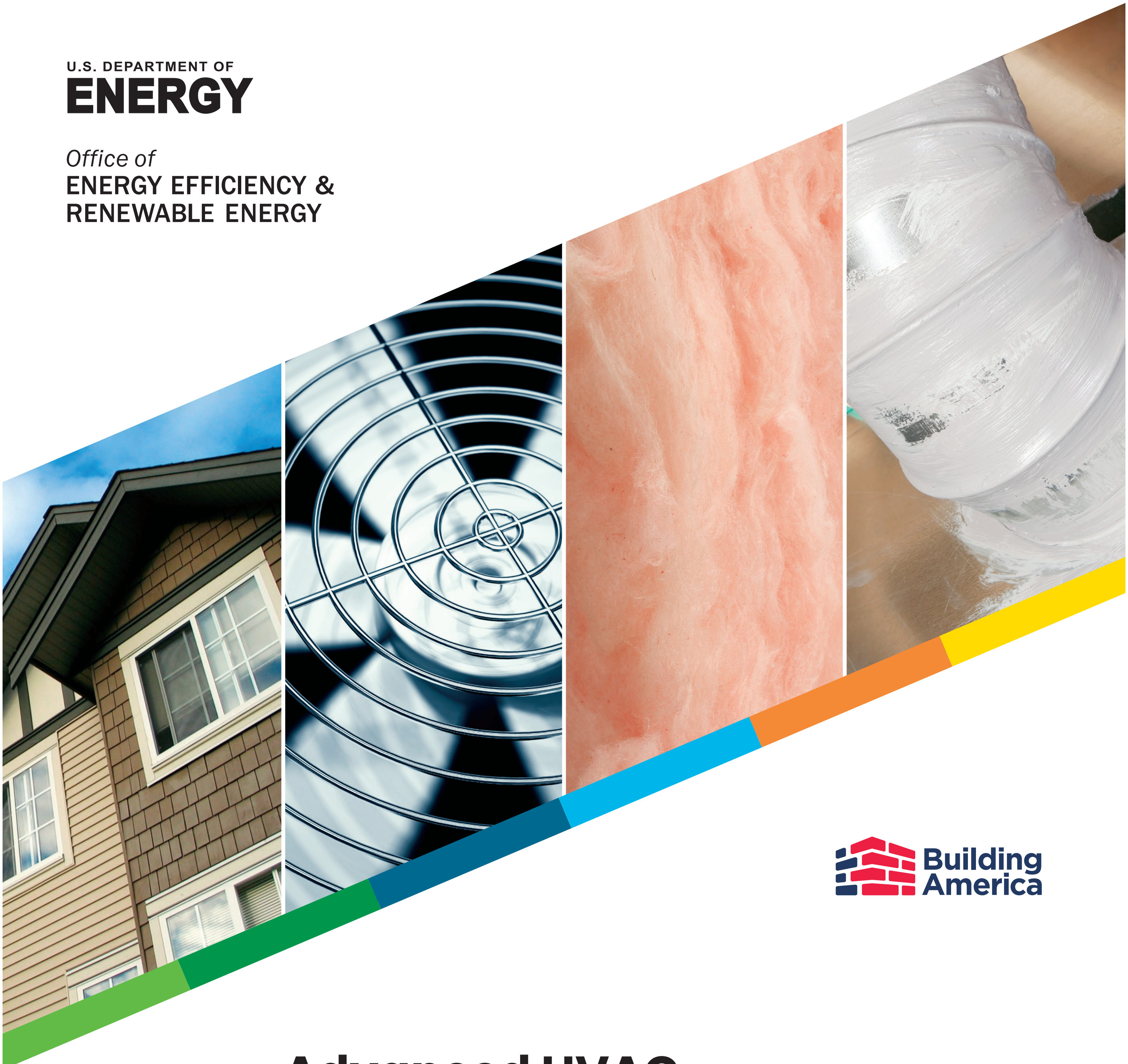

Building America

Advanced HVAC Humidity Control for Hot-Humid Climates

April 2024

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Advanced HVAC Humidity Control for Hot-Humid Climates

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Foreword

The U.S. Department of Energy (DOE) Building America Program has spurred innovations in building efficiency, durability, and affordability for more than 25 years. Elevating a clean energy economy and skilled workforce, this world-class research program partners with industry to leverage cutting-edge science and deployment opportunities to reduce home energy use and help mitigate climate change.

In cooperation with the Building America Program, Home Innovation Research Labs 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, *Advanced Humidity Control for Hot-Humid Climates*, explores a cost-effective solution to improve humidity control and comfort for homes in hot-humid climates using conventional equipment with modified control settings.

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 work presented in this report was funded by the U.S. Department of Energy (DOE), Office of Energy Efficiency and Renewable Energy, Building Technologies Office.

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List of Acronyms

AC	air conditioning
ACCA	Air Conditioning Contractors of America
AFUE	annual fuel utilization efficiency
AHRI	Air-Conditioning, Heating, and Refrigeration Institute
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
Btuh	British thermal units per hour
CFM	cubic feet per minute
CFM/ton	CFM per ton of air conditioning (12,000 Btuh)
DH	dehumidification
DOE	U.S. Department of Energy
ECM	electronically commutated motor
EIA	U.S. Energy Information Administration
ERV	energy recovery ventilator
HRV	heat recovery ventilator
HVAC	heating, ventilating, and air conditioning
kBtuh	thousand British thermal units per hour
kW	kilowatt
kWh	kilowatt-hours
RH	relative humidity
SEER	seasonal energy efficiency ratio
SHR	sensible heat ratio (same as S/T)
S/T	sensible-to-total heat ratio (same as SHR)
TXV	thermostatic expansion valve

EXECUTIVE SUMMARY

Problem Statement

Managing indoor humidity is challenging and more important than ever for new, energy-efficient homes equipped with whole-house mechanical ventilation, particularly during part-load and shoulder season conditions.

Newer homes with higher levels of insulation, more efficient windows, and tighter building envelopes have less heat gain during the cooling season, but the moisture/humidity related loads tend to remain about the same, despite tighter construction practices that reduce infiltration of humid outdoor air (ACCA Manual S). The moisture loads (referred to as latent loads, distinguished from temperature-related heat gains that are referred to as sensible loads) may even be higher due to whole-house mechanical ventilation requirements. The result is that latent loads are proportionately greater relative to total loads (sensible plus latent loads) for more efficient homes, particularly during part-load conditions—periods with lower outdoor temperatures than peak-load design temperatures. To illustrate this challenge, Figure ES-1 shows latent loads as a percentage of total air-conditioning loads for an energy-efficient home compared to a standard, less efficient home, for peak-load and part-load conditions. For efficient homes, typical air-conditioning systems will satisfy the thermostat before running long enough to properly dehumidify the house, particularly during part-load periods.

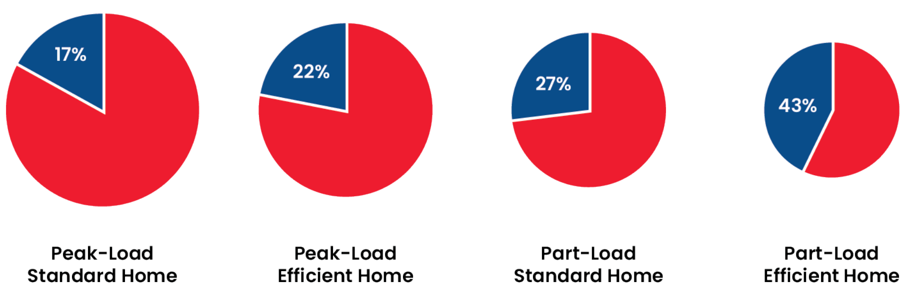

Figure ES-1. Example latent loads as a percentage of total air-conditioning loads: total load equals latent load (blue) plus sensible load (red); the diameter of each pie chart indicates the magnitude of total load for each home.

Image by Home Innovation Research Labs

Compounding this challenge, modern air-conditioning systems typically have less latent capacity and do not dehumidify as well as older, less efficient equipment (ACCA Manual S).

In an attempt to dehumidify, homeowners may simply lower the thermostat setting to run the air conditioning longer. This can result in occupants feeling "too cold," and if indoor humidity still remains high, the result can be occupants feeling "cold and clammy." This would also tend to be an energy penalty. Homeowners may also consider installing a supplemental dehumidifier, but this can be expensive to purchase and operate, and inconvenient to maintain.

Objectives

The purpose of this project is to develop and validate a cost-effective, integrated control solution to improve humidity control and comfort for energy-efficient homes in hot-humid climates. A key emphasis of the study is on systems that are effective, field tested, and practical for builders to install with minimal disruption to standard practices. Successful results would provide a solution to simplify the transition to high-performance humidity control and be the basis for design and installation guidance. By relying on the central system as a starting point, the solution strategy would minimize system complexity and cost for builders and improve comfort and operating cost for homeowners.

The solution strategy was to coordinate the cooling, dehumidification, and ventilation functions of central, ducted HVAC systems to better control indoor humidity, improve occupant thermal comfort, and capture energy savings. The primary strategic goals were to:

- Optimize dehumidification by the central air-conditioning system, particularly during part-load conditions, using conventional equipment with modified control settings and lower system airflows.
- Maximize ventilation during heating/cooling on-cycles, to "bank" and condition outdoor ventilation air, and minimize ventilation during off-cycles, when the HVAC system is not actively heating or cooling but the air handling fan is operating at a low airflow setting for distribution, to minimize energy use and comfort issues due to the circulation of cold or warm-humid outdoor air during off-cycle ventilation.
- Quantify the effectiveness and energy impact of the dehumidification and ventilation strategies. A secondary task was to identify a metric that would be useful to evaluate latent effectiveness.

Example commissioning measurements. *Photo from Home Innovation Research Labs*

For this study, "conventional equipment" refers to HVAC equipment that is centrally ducted, commercially available, and commonly installed by builders. The focus was on single-stage air conditioners or heat pumps, furnaces or air handlers with variable-speed electronically commutated motor (ECM) air drives, and standard control settings for this type of equipment. Some control settings were modified by equipment manufacturers specifically for this project. Home Innovation Research Labs collaborated with builders, HVAC equipment manufacturers, and HVAC designers and contractors to develop HVAC design strategies and identify a series of test houses in Richmond Hill, Georgia; Houston, Texas; and Monroe, Louisiana. The conditioned floor area of the test houses ranged from 2,238 to 3,000 sq. ft., with an average of 2,578 sq. ft. Each test house was monitored for at least one year to verify that performance targets were being met.

The dehumidification strategy, referred to as "enhanced dehumidification," relied on control settings and lower cooling system airflows to increase the latent capacity of the equipment, reduce the time required for the system to achieve peak latent capacity, and increase the duration of the cooling on-cycle to further improve dehumidification.

Initially, the dehumidification strategy was developed based on analysis of manufacturer product data that was expanded, by Home Innovation, to include system performance data at lower airflows. Based on this analysis, the equipment manufacturer reprogrammed some of the "shared data" controls just for this study so that the systems could operate longer at lower airflows.

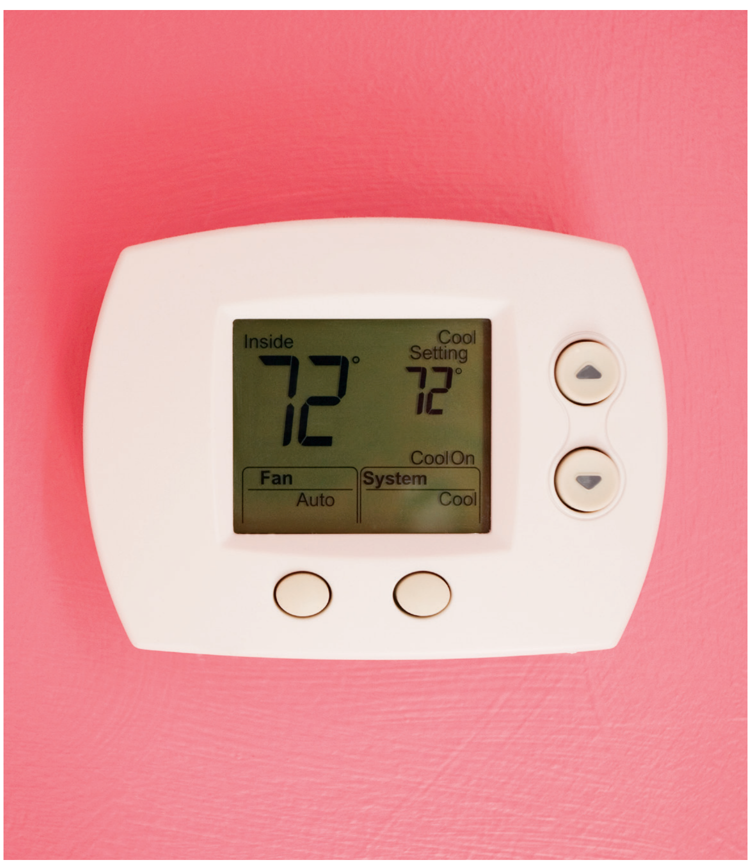

Photo from Getty 1487040894

HVAC equipment manufacturers typically offer a choice of three airflow settings for normal air-conditioning operation: 450, 400, and 350 cfm/ton. For this study, 350 cfm/ton was selected as a starting point for normal cooling airflow in humid climates versus the common choice of 400 cfm/ton.

In addition to a lower airflow during normal cooling, the strategy relied on even lower airflows at the beginning of the cooling cycle, known as the "ramping period," and at any point during the cooling cycle when indoor relative humidity (RH) exceeded a predetermined setpoint, e.g., 55% RH, known as "dehumidification mode."

The results of the expanded product data analysis were used as inputs for a modeling effort. The conclusions for this study were based on monitored field data, for indoor conditions, and modeling results that were used to quantify effectiveness and energy impact.

The ventilation control strategy for this study is referred to as "prioritized ventilation" to differentiate it from "smart ventilation" that is typically delta-T based where ventilation operates more often during off-cycle periods. For this study, the focus is on supply-type ventilation where outdoor air is ducted to the return plenum of the central duct system using a fan-powered ventilator to control the ventilation rate.

Research Questions

This study was designed to answer the following research questions:

1. What is the optimum control strategy to improve dehumidification by the central air-conditioning system in hot-humid climates?

2. What is the optimum control strategy to integrate prioritized ventilation in hot-humid climates?

3. For these humidity control strategies, what are the metrics, including latent efficiency, to quantify the system performance including energy savings and cost-effectiveness?
4. How effective is indoor humidity control for air-conditioning systems with optimized dehumidification compared to typical baseline operation?
5. What are the annual cooling and dehumidification energy profiles for enhanced dehumidification versus baseline air-conditioning operation?
6. How effective is prioritized ventilation with respect to ASHRAE 62.2-2010?
7. What are the potential annual HVAC energy savings using prioritized ventilation compared to typical supply-type continuous ventilation?
8. What is the potential to improve occupant comfort using prioritized ventilation and enhanced dehumidification compared to baseline air-conditioning systems?

Key Results

Key results of this study are presented here for test houses in Richmond Hill, Georgia; Houston, Texas; and Monroe, Louisiana:

- Indoor humidity at the test houses did not exceed 60% RH during the monitored cooling season for 99% of the time in Richmond Hill, 96% of the time in Houston, and 90% of the time in Monroe.
- The dehumidification strategy improved the steady-state latent capacity of the HVAC system at design conditions by 16% to 49% at the Houston test house and by 28% to 71% at the Monroe test house, depending on which mode the system was operating in: ramping, ramping in dehumidification, cooling, or cooling in dehumidification. The greater improvement at the Monroe test house represents an important trade-off to consider: A different indoor evaporator coil (still an Air-Conditioning, Heating, and Refrigeration Institute [AHRI] rated match) was selected to further improve latent capacity, but this reduced the seasonal energy efficiency ratio (SEER) from 16 to 15.
- Indoor humidity was consistently 4%–8% lower at the Houston test house compared to a baseline control house next door.
- The good results at the test houses were primarily due to the amount of time the air-conditioning system operated in ramping or dehumidification modes, or both, particularly during the early cooling season.
- The enhanced dehumidification strategy at the Houston test house increased the estimated annual cooling energy by 212 kWh (4.8%) compared to the same system with standard control settings. This translated to $28/year based on national average electricity prices. Compared to a system with a multispeed ECM air drive (now the required minimum efficiency), the estimated annual energy increased by only 2% or $12/year.
- The control strategy for prioritized ventilation has good potential for saving energy and improving occupant comfort. At the Houston test house, the total amount of ventilation on an annual basis fell short (by about 15%) of the total annual ventilation that would have been provided by using the continuous ventilation required by building code. The control logic could be modified to provide 100% of the code required annual ventilation (ASHRAE 62.2-2010); this would still capture energy savings and significantly reduce the amount off-cycle ventilation time and the risk of associated comfort issues. The estimated annual energy cost savings using prioritized ventilation compared to continuous ventilation, based on providing 100% of the total annual ventilation, was $121.
- The airflow and control settings for ramping and dehumidification modes are critical to control indoor humidity in hot-humid climates, particularly during part-load and shoulder season conditions.

- The dehumidification strategy did not jeopardize the mechanical reliability of the cooling equipment.

Conclusions

- Successful results using an air conditioner or heat pump with a single-stage compressor show that a two-stage or variable-stage compressor system is not required to control indoor humidity to acceptable levels.
- The modeling results and field data showed the dehumidification strategy developed for this study was an effective approach to improve the latent capacity of HVAC systems and practical to install. The strategy controlled indoor humidity to well below the target criteria for this project and performed better than the baseline cases as applicable. The strategy relied on conventional equipment that was straightforward to set up in the field. Measured field data correlated well with expanded manufacturer product data that was used for analysis and modeling.
- Optimized dehumidification provided lower indoor humidity, which improves comfort, so occupants would be less likely to lower the thermostat setting to control humidity and may even raise the thermostat setting during the cooling season to be just as comfortable at a higher temperature (which the occupants did in Richmond Hill).
- The strategies used in this study are applicable across various equipment brands, models, and efficiency levels, and also applicable to a broad range of homes in hot-humid climates. Results will vary by specific equipment, location, and house configuration and construction.
- Some of the control settings for optimized dehumidification are currently available for conventional equipment. It would be straightforward for manufacturers to modify their control settings to optimize systems and achieve the successful results of this study. As of this writing, there is conversation with manufacturers and builders for a next-phase, community-scale project, and on incorporating the control settings for this project as a standard option.

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1 Introduction

1.1 Problem Statement

Managing indoor humidity is challenging and more important than ever for new, energy-efficient homes equipped with whole-house mechanical ventilation, particularly during part-load and shoulder season conditions.

Newer homes with higher levels of insulation, more efficient windows, and tighter building envelopes have less heat gain during the cooling season, but the moisture/humidity related loads tend to remain about the same despite tighter construction practices that reduce infiltration of humid outdoor air (ACCA Manual S). The moisture loads (referred to as latent loads, distinguished from temperature related heat gains that are referred to as sensible loads) may even be higher due to whole-house mechanical ventilation requirements. The result is that latent loads are proportionately greater relative to total loads (sensible plus latent loads) for more efficient homes, particularly during part-load conditions, periods with lower outdoor temperatures than peak-load design temperatures.

To illustrate this challenge, Figure 1 shows latent loads as a percentage of total air-conditioning loads for an example energy-efficient home compared to a standard, less efficient home, for peak-load and part-load conditions. For efficient homes, typical air-conditioning systems will satisfy the thermostat before running long enough to properly dehumidify the house, particularly during part-load periods.

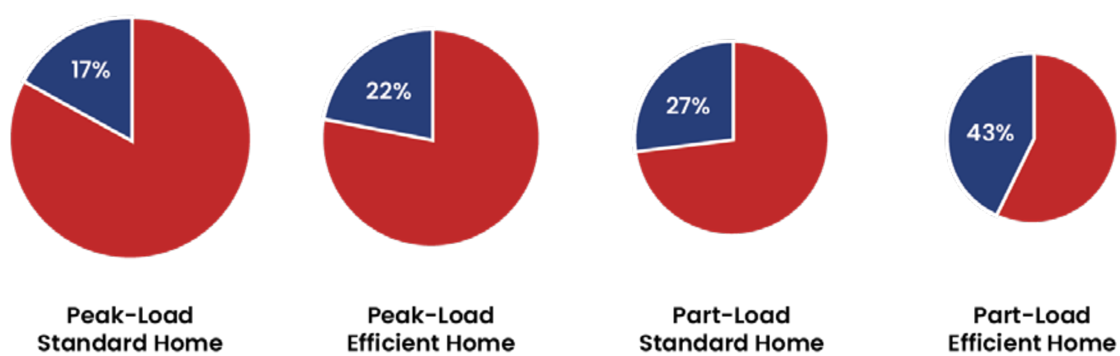

Figure 1. Example latent loads as a percentage of total air-conditioning loads: total load equals latent load (blue) plus sensible load (red); the diameter of the pie chart indicates the magnitude of total load for each home.

Image developed by Home Innovation Research Labs

Compounding this challenge, modern air-conditioning systems typically have less latent capacity and do not dehumidify as well as older, less efficient equipment (ACCA Manual S). In an attempt to dehumidify, homeowners may simply lower the thermostat setting to run the air conditioning longer. This can result in occupants feeling “too cold,” and if indoor humidity still remains high, the result can be occupants feeling “cold and clammy.” This would also tend to be an energy penalty. Homeowners may also consider installing a supplemental dehumidifier, but this can be expensive to purchase and operate and inconvenient to maintain.

1.2 Objectives of the Study

The purpose of this project was to develop and validate a cost-effective, integrated control solution to improve humidity control and comfort for energy-efficient homes in hot-humid climates.

A key emphasis of the study was on systems that were effective, field tested, and practical for builders to install with minimal disruption to standard practices. Successful results would provide a solution to simplify the transition to high-performance humidity control and be the basis for design and installation guidance. By relying on the central system as a starting point, the solution strategy would minimize system complexity and cost for builders and improve comfort and operating cost for homeowners.

The solution strategy was to coordinate the cooling, dehumidification, and ventilation functions of the HVAC system to better control indoor humidity, improve occupant comfort, and capture energy savings. The primary goals of the solution strategy were to:

- Optimize <u>dehumidification</u> by the air-conditioning system, particularly during part-load conditions, using conventional equipment with modified control settings and lower system airflows.
- Maximize <u>ventilation</u> during heating/cooling on-cycles, to “bank,” condition, and distribute outdoor ventilation air; minimize ventilation during off-cycles when the HVAC system is not actively heating or cooling but the air handling fan is operating at a low airflow setting for distribution, to minimize energy use and comfort issues due to the circulation of cold or warm-humid outdoor air during off-cycles.
- Quantify the effectiveness and energy impact of the dehumidification and ventilation strategies. A secondary task was to identify a metric to evaluate latent effectiveness that would be useful for modeling and helpful to provide a measure of the trade-off between rated efficiency (seasonal energy efficiency ratio—SEER) and latent capacity for the different settings or different equipment combinations.

For this study, “conventional equipment” refers to HVAC equipment that is centrally ducted, commercially available, and commonly installed by builders. The focus was on single-stage air conditioners or heat pumps, furnaces or air handling units with variable-speed electronically commutated motor (ECM) air drives, and standard control settings for this type of equipment. Some control settings were modified by equipment manufacturers specifically for this project. Additional details are provided in Section 2.1.6 for equipment and in Section 2.1.7 for control settings.

The dehumidification strategy, referred to as “enhanced dehumidification,” relied on control settings and lower cooling system airflows to increase the latent capacity of the equipment, reduce the time required for the system to achieve peak latent capacity, and increase the duration of the cooling on-cycle to further improve dehumidification. Additional details on humidity control are provided in Section 2.1.2. The HVAC design strategies were evaluated in test houses in hot-humid climates. Details on the test houses are provided in Section 2.1.4.

The solution strategy is applicable across various equipment brands, models, and efficiency levels. It is offered as an alternative to approaches that are optimized around multistage cooling systems, supplemental dehumidifiers, or energy recovery ventilator (ERV)/heat recovery ventilator (HRV) balanced ventilation, although the strategy can complement those approaches if required or desired. The dehumidification strategy can reduce or eliminate the need for supplemental dehumidification; where a dehumidifier is used, the dehumidifier would run less. The focus is on single-stage cooling equipment, but the strategy could improve the dehumidification capacity of two-stage and multispeed compressor systems as well (Rudd 2013).

It was deemed important to consider whole-house mechanical ventilation concurrently with dehumidification because both affect indoor humidity and comfort, and both are considered high-priority issues by builders.[1] The ventilation control strategy for this study is referred to as “prioritized ventilation” to differentiate it from “smart ventilation” that is typically delta-T based where ventilation operates more often during off-cycle periods. For this study, the focus is on supply-type ventilation where outdoor air is ducted to the return plenum of the central duct system using a fan-powered ventilator to control the ventilation rate. The supply fan could be replaced by an ERV or HRV that is integrated with the central heating/cooling duct system—the control strategy would remain the same. Balanced ventilation using heat recovery with a dedicated duct system is beyond the scope of this study.

This research effort supports the U.S. Department of Energy (DOE) Building America objectives to develop and validate optimized comfort systems for low-load homes and high-performance ventilation systems and indoor air quality strategies. The integrated HVAC solution will be applicable to a broad range of energy efficiency targets, from code-minimum to net zero energy ready homes and including high-performing homes that require smaller capacity systems as that market increases. The solution strategy will also be applicable across various equipment brands, models, and efficiency levels. It is expected that the equipment manufacturers will embrace this approach because it

[1] Home Innovation Research Labs, 2020 Annual Builder Practices Survey

focuses on advanced configurations and controls of well-understood equipment and does not require a restructuring of their product lines. Simple solutions with a least-cost impact can have a higher acceptance rate by builders. The developed solution will require only minimum changes to mainstream commercial equipment.

1.3 Research Questions

This study was designed to answer the following research questions:

1. What is the optimum control strategy to improve dehumidification by the central air-conditioning system in hot-humid climates?
2. What is the optimum control strategy to integrate prioritized ventilation in hot-humid climates?
3. For these humidity control strategies, what are the metrics, including latent efficiency, to quantify the system performance including energy savings and cost-effectiveness?
4. What is the energy impact of using the optimized dehumidification strategy compared to typical control settings? How effective is indoor humidity control for air-conditioning systems with optimized dehumidification compared to typical baseline operation?
5. What is the energy impact of using the optimized dehumidification strategy compared to typical control settings?
6. How effective is prioritized ventilation with respect to ASHRAE 62.2-2010?
7. What are the potential annual HVAC energy savings using prioritized ventilation compared to typical supply-type continuous ventilation?
8. What is the potential to improve occupant comfort using prioritized ventilation and enhanced dehumidification compared to baseline AC systems?

1.4 Literature Review/Previous Work

This research benefited from previous Building America research on humidity control in humid climates. Supplemental dehumidification using dehumidifiers (stand-alone, ducted, or integrated with ventilation) has been shown to be effective (Rudd et al. 2002; Williamson and Puttagunta 2013; Kerrigan and Norton 2014; Moyer et al. 2004). The work that has been performed by Building America partners provided a solid basis for developing integrated and optimized design strategies and solutions proposed for evaluation in this project.

Previous field studies have focused on supplemental dehumidification as the central element of the system (Kerrigan and Norton 2014). Simulation-based studies investigated supplemental dehumidification separately from enhanced dehumidification

(Rudd 2013). Simulations show that “enhanced cooling” (lower evaporator airflow and overcooling, versus supplemental dehumidification using dehumidifiers or other means to control humidity) can be effective in some hot-humid locations to control relative humidity (RH) to near 60% if enhanced cooling was activated at a 50% RH setpoint (Rudd 2014). Simply adding a dehumidifier to a house with a conventional cooling system (properly sized) is not nearly as effective without first optimizing the cooling system for dehumidification; optimizing dehumidification first will improve the efficiency and performance of both the cooling system and the dehumidifier (Gehring 2017). Excessive overcooling can exacerbate humidity and condensation concerns (Fang, Winkler, and Christensen 2011). Lower humidity can provide the same comfort at higher temperatures (Research Products 2009).

The limitations of current strategies for humidity control are summarized in Table 1. It is recognized that all options are legitimate alternatives that will always have a market share and will be preferred by some builders and designers. However, the actual market penetration of these solutions remains low. Many builders have concerns with the ability to comply with code required ventilation rates and maintain occupant comfort.[2]

Table 1. Current Strategies to Control Humidity Without Modification to Airflow

Key System	Limitations
HRV/ERV	High first cost; redundant ductwork for independent systems; additional installation and commissioning practices to ensure adequate performance; additional maintenance; potential comfort issues (e.g., air is tempered during winter but is still colder than house air); mainstream builders have not embraced it; ERVs have little to no high humidity control benefit in a hot-humid climate (Rudd 2013).
Smart ventilation	Ventilation more often during heating/cooling off-cycles results in an energy penalty and aggravates humidity and comfort issues due to circulation of warm-humid air during the cooling season, and cold air that feels like a draft during the heating season.
Stand-alone dehumidifier	Most effective when cooling system is first optimized for dehumidification; noise; lack of distribution; maintenance requirements; location; drain piping.
Independently ducted or central duct integrated dehumidifier	High first cost; most effective when cooling system is first optimized for dehumidification.
Overcooling setting at the thermostat	Can create a comfort issue and an energy penalty; more effective when cooling system is first optimized for dehumidification to reduce the use of overcooling.
Two-stage or variable-stage AC	High first cost; does not necessarily dehumidify better than single-stage because the compressor capacity is reduced proportionally to fan speed; variable capacity systems (alone) do not improve RH control (Rudd 2013).

[2] Home Innovation Research Labs, 2020 Annual Builder Practices Survey

The interactions between the various ventilation methods and the rest of the mechanical system in terms of their impact on the indoor humidity, comfort, effectiveness, installed costs, and operating costs are not always considered. Current ventilation methods have mixed results across these criteria. Exhaust ventilation using bath fans meets current ventilation rate requirements but is not considered effective at providing and distributing “fresh” outdoor air. Balanced ventilation using heat recovery and dedicated distribution ducts provides effective ventilation but is considered cost-prohibitive by many builders. Further, energy recovery ventilators have little to no high humidity control benefit in a hot-humid climate (Rudd 2013). Supply ventilation using outdoor air ducted to the return plenum of an air distribution system can provide effective ventilation and is affordable to install, but the actual ventilation rate typically depends on the static pressure in the return plenum and the ventilation duct layout—it is commonly less than expected if not accurately designed for. Further, supply ventilation can be expensive to operate (relies on air handling fan power for distribution) and may create comfort issues, including cold outdoor air or hot, humid outdoor air distributed around the house during periods without heating or cooling.

“Smart” ventilation controls typically delay ventilation until outdoor conditions are more favorable. The primary limitation is that ventilation is more likely to occur during low-load periods when the cooling and heating systems are operating less. This may increase comfort complaints and energy use when the control initiates the air handling fan for distribution because the ventilation air is not conditioned. Further, these controls must default to allow ventilation anyway during extended stretches of unfavorable outdoor conditions, and outdoor specific humidity is less likely to change much over the course of a day.

2 Methodology

2.1 Research Design

2.1.1 Overview

Home Innovation Research Labs collaborated with Builders, HVAC equipment manufacturers, and HVAC designers and contractors to address the project's objectives and research questions. The team developed a test plan, identified project design parameters and performance criteria, developed the "enhanced dehumidification" strategy and "prioritized ventilation" strategy, specified the HVAC design (equipment selection and control settings), and identified test houses to be used to monitor and evaluate the HVAC design during the two phases of this project. Details on the test houses are provided in Section 2.1.4.

To develop the dehumidification strategy, the project team evaluated the performance and trade-offs of various methods to improve humidity control using conventional equipment. The primary methods investigated are discussed below in Section 2.1.2. Again, for this study, conventional equipment refers to HVAC equipment that is centrally ducted, commercially available, and commonly installed by builders. The focus was on single-stage air conditioners or heat pumps, furnaces or air handlers with variable-speed ECM air drives, and standard selections for control settings for this type of equipment. Some control settings were modified by equipment manufacturers specifically for this project. Details are provided for equipment in Section 2.1.6 and for control settings in Section 2.1.7.

Home Innovation analyzed manufacturer product data and expanded this data to include system performance data at lower airflows. Based on this analysis, the equipment manufacturer reprogrammed the "shared data" to adjust some of the control settings for this study, so the systems could operate at the lower airflows.

The results of the expanded product data analysis (Section 2.3) were used as inputs for the modeling effort. The conclusions for this study were based on monitored field data, for indoor conditions, and modeling results that were used to quantify effectiveness and energy impact (Section 2.4).

2.1.2 Humidity Control

When an air-conditioning system first cycles on, it takes several minutes before the indoor evaporator coil gets cold enough (drops to the dew point temperature of the airstream) to start condensing the water vapor from the airstream. It takes several additional minutes to get to its coldest steady-state temperature (below the dew point) for full latent capacity. For modern systems, it can take approximately 12–28 minutes to reach full latent capacity after startup; this compares to 6–8 minutes for older 10 SEER equipment, because the older equipment had a lower final coil sensible heat ratio of about 0.70–0.75 (30% to 25% latent capacity) compared to about 0.80 (20% latent capacity) for newer high-SEER equipment (ACCA Manual S, illustrated in Figure 1-7 Cooling Pull-Down Transient After Startup).

As water vapor in the air condenses to liquid water on the evaporator coil, the water (condensate) flows down the coil to the drain pan and out via the condensate piping. But after a cooling cycle, water remaining on the coil and in the drain pan does not drain immediately or completely, and some will tend to evaporate back into the airstream and house. How much water evaporates back into the airstream depends on the time between cooling cycles and how long the air handling circulation fan keeps running after the cycle.

Given the time it takes to reach full latent capacity, a longer cooling on-cycle is important to improve humidity control. During part-load cooling conditions, a typical system runs for relatively shorter periods to satisfy the thermostat and may provide very little dehumidification.

A lower cooling system airflow across the evaporator coil makes for a colder coil (further below the dew point) which increases the latent capacity of the system and reduces the time required for the system to achieve full latent capacity. The lower airflow also reduces sensible capacity (and total capacity), which results in a longer run time that further improves dehumidification.

Sensible Heat Ratio (SHR, also referred to as Sensible-to-Total Heat Ratio or S/T) varies by equipment selection, entering wet-bulb and dry-bulb temperatures, outdoor condensing temperature, and system airflow across the evaporator coil. The lower the SHR, the greater the latent capacity and humidity control. Equipment can be selected with an eye toward SHR in addition to capacity, although a lower SHR can also result in a lower efficiency rating (SEER). After equipment selection, system airflow becomes an important variable to provide a lower SHR for better dehumidification. The type of whole-house mechanical ventilation can affect return air temperatures and subsequently SHR where outdoor air is introduced to the return plenum.

In addition to a lower supply air temperature and coil temperature, a lower cooling system airflow results in lower refrigerant vapor (suction) pressure, which increases the compression ratio and compressor power. Note that the reduction of air handling fan power at lower airflow is greater than the increase of compressor power, so there is a net decrease in overall system power at lower airflows. Despite this, because the system must run longer at lower airflows to satisfy the thermostat, the total overall annual energy use will be somewhat higher (detailed in Section 2.4 Energy Impact Analysis). The higher energy use makes sense given that the lower airflow system must still satisfy the same thermostat setpoint while removing more moisture from the air.

There are risks associated with excessively low airflows across the evaporator coil during cooling. Lower airflow, at some point, has the potential to cause evaporator coil icing. A lower airflow also has the potential to reduce the refrigerant superheat, which could lead to flooding the compressor with liquid refrigerant and damaging the compressor. The thermostatic expansion valve (TXV) works to prevent this by maintaining target superheat, at least to a point, by throttling refrigerant through the coil (note that a TXV is a requirement for this study). Further, most modern systems have a low-pressure switch to protect the compressor from damage. There is also some risk of condensation at supply registers at lower airflows due to the colder supply air temperature. Another potential risk of lower airflow is that at some point the distribution system may not provide sufficient air mixing within a room, i.e., not enough “throw” from a supply register (this was not measured at the test houses). None of these potential risks were observed at the test houses; see Section 2.5 for additional details on technical risks associated with lower airflows.

Initially, the dehumidification strategy was developed based on analysis of manufacturer product data that was expanded, by Home Innovation, to include system performance data at lower airflows. Based on this analysis, the equipment manufacturer reprogrammed some of the “shared data” controls just for this study so that the systems could operate longer at lower airflows.

During commissioning for Phase 1, cooling system airflow was reduced incrementally, down to as low as 175 cfm/ton, while refrigerant operating conditions were closely monitored by the HVAC technician and factory technical representative to observe system operation at lower airflows. The system maintained proper operating conditions down to 200 cfm/ton; below that, at the Houston test house, the refrigerant pressures appeared to become destabilized, i.e., “hunt” to try to maintain a steady superheat. Based on this observation, it was decided to eliminate operation below 250 cfm/ton except during very brief periods (30 seconds) at the beginning of the cooling cycle.

HVAC equipment manufacturers typically offer a choice of three airflow settings for normal air-conditioning operation: 450, 400, and 350 cfm/ton. For this study, 350 cfm/ton was selected as a starting point for normal cooling airflow in humid climates versus the common choice of 400 cfm/ton.

In addition to a lower airflow during normal cooling, the strategy relied on even lower airflows at the beginning of the cooling cycle, known as the ramping period, and at any point during the cooling cycle when indoor RH exceeded a predetermined setpoint, known as dehumidification mode. Ramping and dehumidification mode are detailed in Section 2.1.5.

The results of the expanded product data analysis (Section 2.3) were used as inputs for the modeling effort. The conclusions for this study were based on monitored field data, for indoor conditions, and modeling results that were used to quantify effectiveness and energy impact (Section 2.4 Energy Impact Analysis).

During the design stage, the team investigated using an alternative evaporator coil and TXV to improve humidity control. A TXV (thermostatic expansion valve) is required for this study to efficiently maintain proper refrigerant conditions at lower airflows, and it must be a listed component of the Air-Conditioning, Heating, and Refrigeration Institute (AHRI)-rated system, so the focus turned to the coil. Coil selection can significantly affect latent capacity and can represent a trade-off between greater latent capacity and lower SEER rating. For one of the test houses in this study, the furnace and outdoor condensing unit were identical to those at another test house, but a different evaporator coil was selected to increase latent capacity at the expense of a lower SEER rating (from 16 SEER to 15 SEER). This different coil improved latent capacity by about 10%–15% depending on the operating mode.

The team also considered adjusting the thermostat on-off band setting, also referred to as the “swing” temperature setting or the “cooling differential” setting, to increase cycle length and reduce cycles per hour. The on-off band refers to the number of degrees between when the system cycles on and when it cycles off. The higher the on-off band, the longer the system will run, and the better it will dehumidify. This setting is fixed for some thermostats, while other thermostats may have separate, adjustable on-off band settings for heating and cooling. The original thermostat for this project during Phase 1 had an adjustable on-off cooling band that could be set to 0.5°F or 1.0°F, so if the system turned on at 75.0°F, it turned off at 74.5°F or 74.0°F. But it turned out that thermostat did not have the capability to provide a signal for dehumidification mode (described in Section 2.1.5), so it was replaced. The on-off band for the replacement thermostat was fixed (not adjustable) and tight (approximately 0.3°F), so the results of this study did not rely on this measure. Despite this, increasing the on-off band is recommended to improve humidity control where available. Generally, doubling the on-off band will roughly double the cooling on-cycle period, but this would more than double the amount of dehumidification due to the length of time it takes the system to reach full latent capacity after startup.

In essence, increasing the on-off band provides a measure of overcooling. Overcooling is a separate, optional setting on some thermostats that allows the cooling system to operate below the temperature setpoint if indoor humidity rises above the humidity setpoint. For example, with a cooling setpoint of 75°F and humidity setpoint of 60% RH, if the indoor humidity is 63% during a cooling cycle, the system will continue to “overcool” below 75°F until 60% RH is met or until the temperature drops to a predetermined overcooling limit, normally 1–3°F below the setpoint. Further, overcooling can be set up to activate cooling when indoor humidity is above the setpoint, even if there is not a call for cooling based on the temperature setpoint, within limits.

The replacement thermostat for this study had an overcooling logic that simply lowered the setpoint and did not increase run time, e.g., if the cooling setpoint was 75°F and 1°F overcooling was selected, the system turned on at 74°F and, since it had a tight, non-adjustable on-off band, off at about 73.7°F. This was not considered as improving run time (longer on-cycle) or humidity control in any meaningful way, so overcooling was not used for this study.

For other thermostats where the overcooling setting increases run time, overcooling can be a viable approach to improve humidity control, but excessive overcooling can also create comfort issues and even raise indoor RH. Overcooling by up to 1°F could be beneficial and likely would not pose a comfort issue.[3]

Initially there was a proposed approach to identify a target evaporator coil temperature and then vary the airflow to maintain that target—a lower than normal coil temperature—to improve dehumidification. The technology required for this approach was considered too complex for conventional residential equipment and not practical for this project.

There is a common perception that outdoor units with two-stage or variable-speed compressors provide better dehumidification, but generally this is not the case. A two-stage system operating in low-stage will typically operate the compressor at 2/3 capacity and, proportionately, the air handler at 2/3 airflow. Manufacturer product data shows that the SHR is commonly higher during low-stage operation. This is because lower system capacity with the same evaporator coil results in a warmer coil, i.e., the coil is oversized for low-stage operation, and the latent capacity decreases more than the sensible capacity. One advantage of a two-stage system is it can run longer compared to a single-stage system, but it still typically operates at a higher SHR in low-stage, and in practice, the duration of low-stage operation is limited by system controls (e.g., 10 minutes at the beginning of a cooling cycle), which largely limits any advantage. Applying the enhanced dehumidification strategy would improve the latent effectiveness of multistage cooling systems.

2.1.3 Design Parameters and Performance Criteria

The team identified target design parameters and performance criteria:

- Target test house: 2,200–3,000 sq. ft. single-family detached house
- Location: hot-humid (International Energy Conservation Code climate zone 2A) or warm-humid (climate zone 3A) below the warm-humid line
- HVAC equipment: air conditioner or heat pump with single-stage compressor; furnace or air handler with a variable-speed ECM air drive

[3] Team's opinion, assuming the overcooling feature of the selected thermostat increases the duration of the cooling cycle.

- Indoor design conditions: 75°F and 50% RH cooling; 70°F heating
- Overcooling by up to 1°F was considered acceptable if necessary
- Acceptable upper limit for indoor humidity: 60% RH
- Additional benchmark for indoor humidity: 55% RH
- Whole-house mechanical ventilation rate target: ASHRAE 62.2-2010.

2.1.4 Test Houses

During Phase 1, the HVAC design was installed at two test houses in hot-humid climates: one in Richmond Hill, Georgia (near Savannah) and the other in Houston, Texas. There was also a baseline house in Houston, where the HVAC system was set up for comparison purposes; there was not a baseline house in Richmond Hill. Both test houses were model homes that were sold and occupied during the study. The houses and systems were monitored for more than one year to verify that performance targets were being met.

During Phase 2, the HVAC design was installed at two new test houses, one in Houston, and the other in Monroe, Louisiana. There was also a Phase 2 baseline house in Houston but not in Monroe. Both test houses were model homes; the Monroe test house was sold and occupied during the study.

Additional information on the project timelines for the test houses is provided in Section 3.1 Sample Field Data.

The test houses and associated HVAC systems are described below. All HVAC systems included a variable-speed ECM air drive (except as noted for one baseline house) and evaporator coil with TXV. All HVAC systems used standard R-410A refrigerant. For all houses, ducts were located in vented attics. The team conducted a design review of ACCA Manual J, Manual S, and Manual D procedures for each house. Builders were responsible for testing house tightness and duct leakage (test results were not available for this report).

- Phase 1 Richmond Hill, Georgia, test house: 2-story, slab-on-grade, 2,536 sq. ft., 4-bedroom; 2.5-ton 15 SEER heat pump system, air handler in a second-floor closet.
- Phase 1 Houston, Texas, test house: 2-story, slab-on-grade, 3,000 sq. ft., 4-bedroom, 4-ton 16 SEER AC, 80% AFUE gas furnace in attic.
    - Phase 1 Houston, Texas, baseline house: same house design and equipment as the test house except for a 2-stage condenser.
- Phase 2 Houston, Texas, test house: 1-story, slab-on-grade, 2,538 sq. ft., 4-bedroom, 3-ton 16 SEER AC, 80% AFUE gas furnace in attic

  - Phase 2 Houston, Texas, baseline house next door to the test house: 1-story, slab-on-grade, 3,058 sq. ft., 4-bedroom, 3.5-ton 14 SEER AC, 80% AFUE gas furnace; note that the furnace has a multispeed ECM (not a variable-speed ECM), and supply ventilation with the same control but a motorized damper (not a fan-powered ventilator).

- Phase 2 Monroe, Louisiana, test house: 1-story, slab-on-grade, 2,238 sq. ft., 4-bedroom, 3-ton 15 SEER AC, 80% AFUE gas furnace in a mechanical closet.

2.1.5 Operating Modes

The dehumidification strategy relied on lower system airflows during different modes of operation. Again, a lower cooling system airflow across the evaporator coil makes for a colder evaporator coil, which increases the latent capacity of the system, reduces the time required for the system to achieve peak latent capacity, and results in a longer run time that further improves dehumidification. For this study, the modes of operation were defined as follows:

- Ramping: the period at the beginning of each cooling cycle where the system operates at a lower than normal cooling airflow. Manufacturers commonly provide optional settings with various ramping “profiles” for variable-speed ECM air handlers. These profiles include the initial ramping period, the normal cooling period after ramping, and a brief period at the end of cooling cycle known as the “OFF delay,” where the compressor cycles off while the air handling fan continues to run. An OFF delay can improve the efficiency rating somewhat, but in humid climates, the OFF delay can evaporate water left on the coil back into the airstream. Commonly, the default setting for OFF delay is one minute at normal cooling airflow. Ramping profiles typically include an option that uses an OFF delay with reduced airflow and duration—ideally this would be none for optimum humidity control. For this study, the ramping profile operates the air handling fan as follows: 50% of normal cooling airflow for the first 30 seconds, 85% airflow for the next 7.5 minutes, 100% airflow for the duration of the cooling cycle, and 50% airflow for a 30-second OFF delay. This profile was modified by the manufacturer to allow dehumidification mode (described below) to operate during ramping in addition to during normal cooling. The profile was also supposed to have been modified to eliminate the 30-second OFF-delay, but this was not accomplished.

- Cooling: the normal cooling period after the ramping period where the system operates at 100% normal cooling airflow for the duration of the cooling demand. Manufacturers typically offer a choice of three nominal airflow settings for cooling: 450, 400, or 350 cfm/ton. The system will operate at full rated capacity and efficiency except during ramping, unless dehumidification mode (described below) is activated. For this study, cooling airflow was set to 350 cfm/ton, versus the typical setting of 400 cfm/ton.

- Dehumidification (DH): periods where the system airflow is reduced when indoor humidity rises to a predetermined setpoint, e.g., 55% RH. DH mode is activated by a thermostat with an integral dehumidistat that sends a signal to a furnace or air handler with DH capability. Ideally, DH mode should operate as needed during both ramping periods and normal cooling periods, although some manufacturers limit DH mode to normal cooling periods. For this study, DH mode was set at the thermostat to activate at 55% indoor RH; DH mode, if activated, reduced airflow by 15% during both ramping and cooling periods, hence this study distinguishes between DH cooling (85% of cooling airflow) and DH ramping (about 72% of cooling airflow) to distinguish the different airflows and operating characteristics.
- Fan-only: periods when there is no active heating or cooling, but the air handling fan operates for periodic circulation, including off-cycle ventilation. Lower fan-only airflow will minimize comfort issues and energy usage during these periods. Note that continuous fan-only circulation (referred to as fan-on, which can be selected at the thermostat) is not recommended because this will increase indoor humidity due to evaporation of water from the wet coil back into the airstream. Periodic fan-only operation can also increase indoor humidity during the cooling season, so this is another reason to minimize ventilation during off-cycles. For this study, fan-only airflow was set to 25% of normal cooling airflow (about 88 cfm/ton), which was the lowest fan-only airflow selection provided by the manufacturer.

2.1.6 HVAC Equipment Selection

The steps used to select the HVAC equipment for this study are:

- Perform an accurate heating and cooling load calculation (ACCA Manual J); select air-conditioning equipment that is not oversized (ACCA Manual S).
- Select an HVAC system that is an AHRI-rated match for all components; the system must include a thermostatic expansion valve (TXV) installed at the evaporator coil.
- The furnace or air handler must have a variable-speed (constant airflow) ECM air drive, i.e., not a multispeed (constant torque) ECM; the furnace or air handler must have capability to operate in ramping mode and dehumidification (DH) mode.
- Select the outdoor unit: for this study, an air conditioner or heat pump with a single-stage compressor. A two-stage compressor is not required, but the

enhanced dehumidification strategy would improve dehumidification for a two-stage system as well (see Section 2.1.2).

- Select the indoor evaporator coil, or air handler with integral coil, to balance the trade-off between desired SEER efficiency rating and improved system latent capacity (still must be an AHRI-rated match).
- Select a thermostat with integral dehumidistat and capability to send a control signal to the furnace or air handler to activate DH mode (described above) as needed.

2.1.7 Dehumidification Control Settings

The steps used to set up the dehumidification control settings for this study are:

- Ensure the HVAC system is commissioned in accordance with the manufacturer's installation instructions.
- Set normal cooling airflow to 350 cfm/ton for optimum latent capacity, i.e., not the typical 400 cfm/ton; the three settings commonly offered by manufacturers are 450, 400, and 350 cfm/ton.
- Select the most aggressive ramping profile, i.e., the one with the lowest airflows for the longest periods. Like cooling airflow, this is normally set at the furnace/air handling control board using dipswitches. For this study, the ramping profile was 50% airflow for the first 30 seconds, 85% airflow for 7.5 minutes (298 cfm/ton), 100% airflow after ramping for the duration of the cooling cycling, and 50% airflow for 30 seconds after the end of the cooling cycle (OFF-delay).
- Set the thermostat to activate DH mode at 55% indoor RH (recommended). For this study, DH mode operated the system at 85% airflow, about 300 cfm/ton during cooling (DH cooling mode), and about 250 cfm/ton (85% of 350 cfm/ton) during ramping (DH ramping mode). These DH airflows aligned reasonably well with the 50 cfm/ton increments for the three typical cooling airflow setting choices commonly offered by manufacturers (450, 400, and 350 cfm/ton); the expanded manufacturer product data were extrapolated for analysis using 300, 250, and 200 cfm/ton values (Section 2.3).
- If the thermostat has the capability (it did not for this study), it is recommended to increase the thermostat on-off band setting (also referred to as the "swing" temperature setting) by up to 1°F to increase cycle length and reduce cycles per hour for better dehumidification. Alternatively, it is recommended to select overcooling at the thermostat by up to 1°F if doing so increases the length of the cooling cycle (it did not for this study). Details for both settings are provided in Section 2.1.2.

- Set the fan-only mode to relatively low airflow, e.g., 25% of normal cooling airflow (about 88 cfm/ton, for this study), selected at the furnace or air handling control board, to minimize comfort issues and energy use during periodic circulation or ventilation during heating/cooling off-cycles. Do not select continuous fan-on circulation, as this will increase indoor humidity during the cooling season.

Figure 2 illustrates the ramping profile and its effect on indoor humidity. This sample data shows air handling power (blue) and indoor relative humidity (% RH) measured at the thermostat (gold) during one day in May at the Phase 2 Houston test house. Air handling power is roughly 75 W during DH ramping, 110 W during DH cooling, 100 W during ramping (without DH), and 160 W during normal cooling (without DH). Figure 3 zooms in to show the one-minute data points.

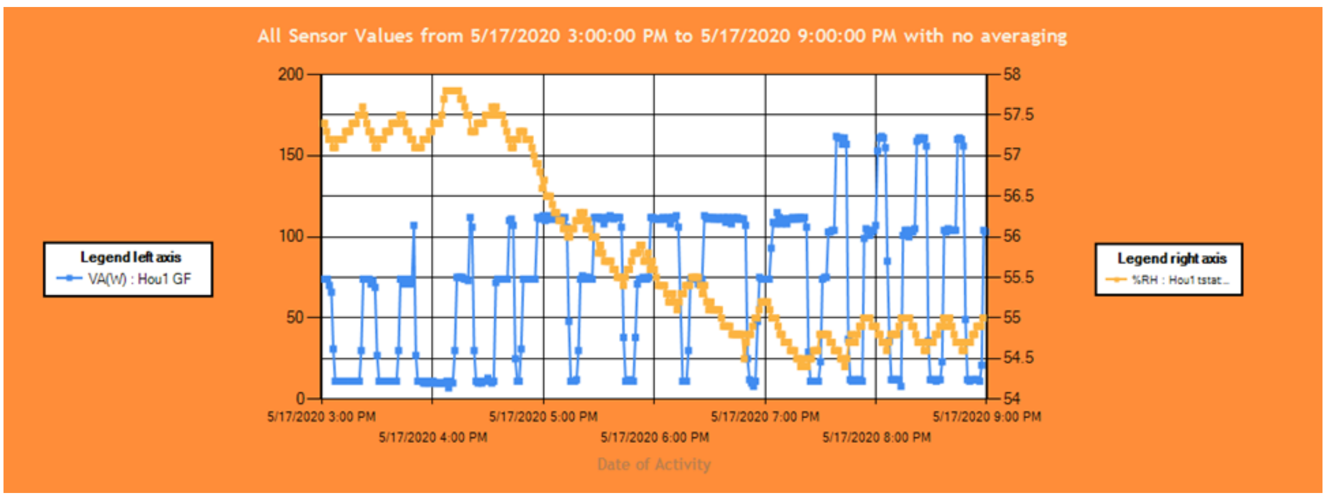

Figure 2. Houston ramping profile illustrated by air handling power (watts, blue) and its effect on indoor humidity (% RH, gold)

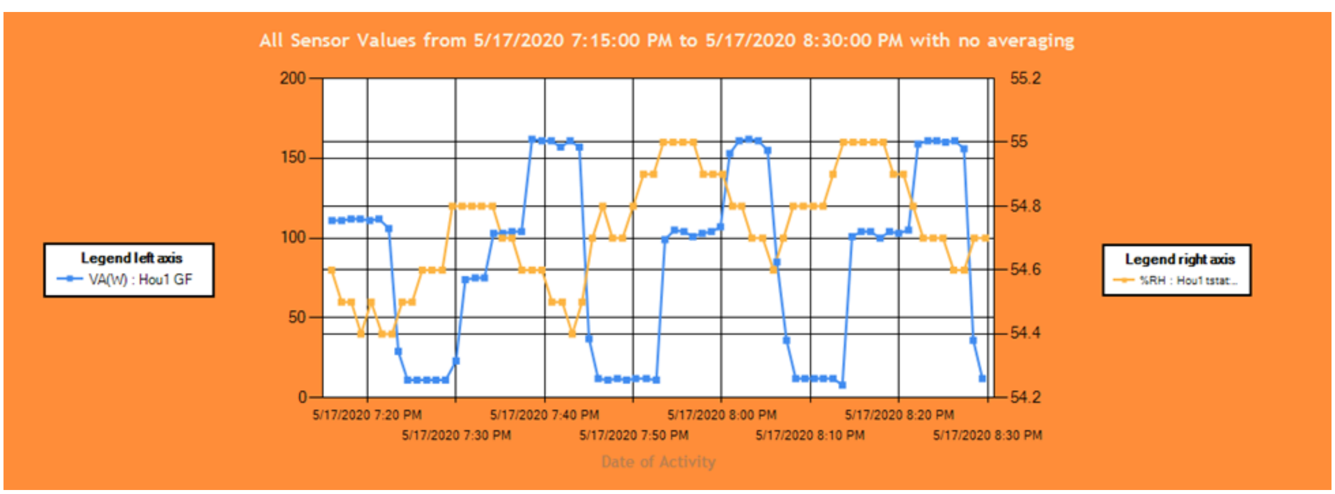

Figure 3. Zoom-in on Houston ramping profile operation to show one-minute data points for air handling power (watts, blue) and indoor humidity (% RH, gold)

2.1.8 Ventilation Strategy

The whole-house mechanical ventilation strategy was based on supply-type ventilation, with outdoor air ducted to the return plenum of the HVAC system, using a fan-powered inline ventilator to control the airflow rate. During heating/cooling on-cycles, outdoor air was filtered, conditioned, and distributed throughout the house. During heating/cooling off-cycles, outdoor air was filtered and distributed while the air handler operated at much lower airflow to minimize energy use and comfort issues. Settings were designed to maximize ventilation during on-cycles and minimize ventilation during off-cycles. The ventilation control settings were:

- Ventilation airflow target: up to twice the 62.2-2010 continuous rate but not to exceed 10% of the air-conditioning system airflow, e.g., for a 3-ton system operating at 1,050 cfm (350 cfm/ton), the target ventilation rate would be 105 cfm.
- Control limits to deactivate ventilation: 95°F high outdoor air temperature limit; 30°F low outdoor air temperature limit; 65% high indoor RH limit.
- Off-cycle ventilation airflow: 25% of normal cooling airflow, set at the furnace or air handling control board.
- Control logic, modified by the manufacturer for this project (Note: The control logic was modified for “comfort” mode; the control could be set to “code” mode if standard operation is desired.):
    - Ventilation time ≡ (62.2-2010 continuous airflow rate/measured airflow rate)*60. For example, if the 62.2 continuous rate is 60 CFM, and the measured ventilation airflow is 100 CFM, then the required ventilation time is 36 minutes per hour ((60cfm/100cfm)*60 min/hr).
    - During a four-hour cycle period, the control will attempt to ventilate at least twice the ventilation time to provide minimum ventilation for occupants.
    - Ventilation will occur during entire heating or cooling calls, even if the ventilation time has been met, unless restricted by ventilation air temperature or indoor RH limits.
    - If heating or cooling calls do not occur for twice the ventilation time during the four-hour cycle period, then the control will ventilate at the end of the four-hour cycle period to ensure that ventilation occurs for twice the ventilation time, unless restricted by short-cycle protection or by ventilation air temperature or indoor RH limits.

  - Example: If ventilation time is 30 minutes and the heat was on for 20 minutes, the control will turn on the ventilator (and air handler) for the last 40 minutes ((2x30)-20) of the four-hour cycle period.

During the design stage, the approach was to over-ventilate during on-cycles with the intent to meet the overall annual amount of ventilation, on average, required by code (based on ASHRAE Standard 62.2-2010) but without excessively loading the HVAC system with hot, humid air during cooling cycles or cold air during heating cycles. The upper limit for ventilation airflow of 10% of system airflow was selected based on the return air conditions, which at this mix would be within normal operating parameters for Air Conditioning Contractors of America (ACCA) design assumptions and within ACCA recommendations. Thus, the minimum continuous ventilation rate required by code would be used for load calculations, and therefore the ventilation strategy would not drive the selection of higher capacity equipment, which would adversely affect humidity control.

2.2 Field Data Monitoring

Home Innovation oversaw the installation of data monitoring hardware at each house. The wireless sensors communicated to a cellular gateway with a data collection interval of one minute. The sensors are described here:

- Temperature and relative humidity (T&RH) sensors, locations: thermostat; master bedroom; attic; outdoor; within supply and return plenums of the HVAC system; selected closets (Phase 2 only); within selected supply register boots (Phase 2 only).
- Energy sensors were installed within the main electrical panel to measure power and operating hours for the outdoor unit (air conditioner or heat pump), indoor unit (furnace or air handler), and whole-house mechanical ventilation fan.
- Refrigerant temperature and pressure sensors were installed at the outdoor unit; the sensor manufacturer's website provided subcooling and superheat values in real time although data could not be downloaded (Phase 2 only).
- Thermostat, Wi-Fi enabled, allowed for temperature and dehumidification settings to be checked and adjusted remotely.

Monitored T&RH data were compared to target indoor conditions, and results indicated if comfort criteria were being met. Results identified periods where indoor humidity exceeded 60% RH to indicate when supplemental dehumidification might be desired. Monitored data for equipment operating points, including supply and return plenum air conditions and refrigerant line temperatures and pressures, confirmed that equipment was operating within manufacturer specifications (Figure 4 and Figure 5).

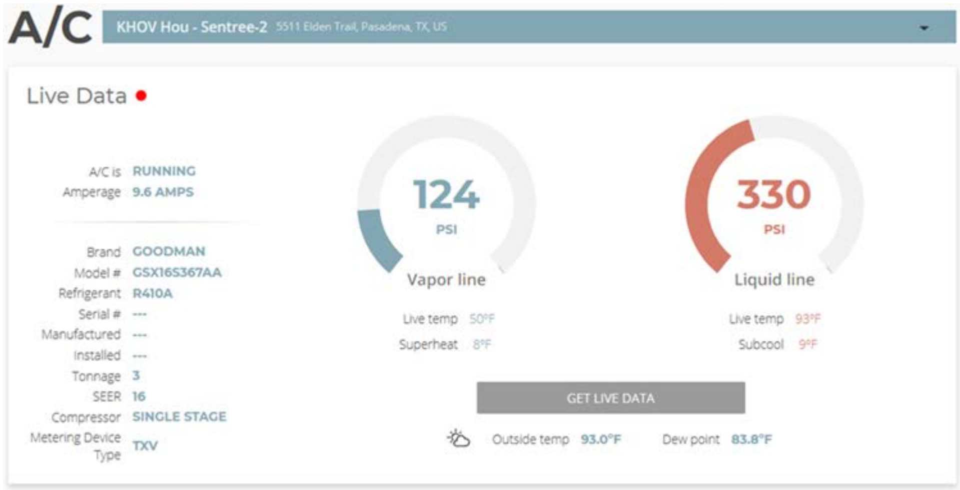

Figure 4. Remote monitoring of refrigerant system

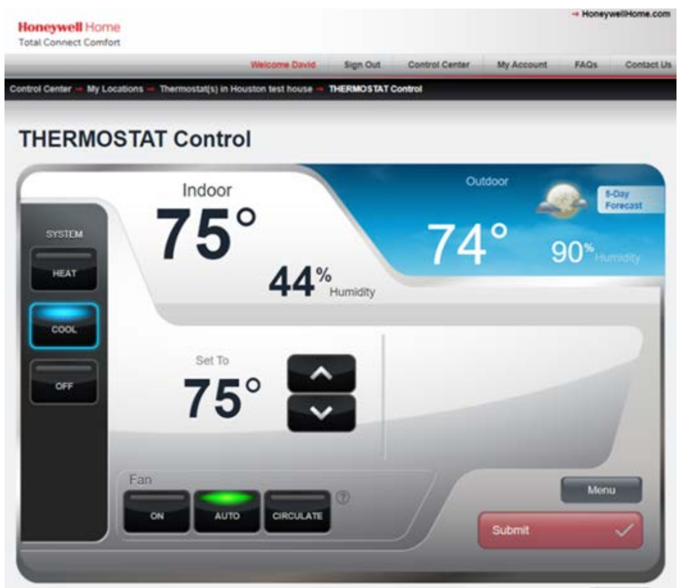

Figure 5. Remote monitoring of thermostat

Monitored ventilation rates were assessed for compliance with ASHRAE Standard 62.2-2010 requirements. Ventilation data included duration by on-cycle and off-cycle periods.

The conditions at the test houses were monitored using the following equipment:

- OmniSense G-3-C cellular gateway, data collection interval set to one minute
- OmniSense S-2-3 wireless temperature and relative humidity (T&RH) sensor (for indoor, outdoor, and attic locations): -40 – 85°C, 0%–100% RH, ± 0.3°C, ± 2.0% RH
- OmniSense S-2-2 wireless T&RH sensor (for supply and return plenum location) with A-1-200 T&RH probes: ± 0.3°C, ± 2.0% RH

- OmniSense S-60 wireless energy meter with split core current transducer (for outdoor air conditioner or heat pump, indoor furnace or air handler, and whole-house mechanical ventilation fan): 45-65 Hz, Class 1.0 ± 1%
- Sentree by Alert Labs monitoring system with cellular gateway: live temperature sensor data for ambient air and vapor and liquid refrigerant lines; live pressure sensor data for vapor and liquid refrigerant lines; installed at the outdoor unit; website provided calculated subcooling and superheat values in real time, although data could not be downloaded (Phase 2 only).
- Honeywell TH8321WF1001 Wi-Fi VisionPro smart thermostat: communicated via a Wi-Fi hotspot; thermostat temperature and dehumidification setpoints could be checked or adjusted online, although data could not be downloaded.

2.3 Expanded Product Data Analysis

Latent capacity varies by equipment selection, outdoor dry bulb temperature, indoor dry bulb and wet bulb temperatures, and airflow. Home Innovation developed a methodology to estimate the performance characteristics of an air-conditioning system while operating at lower than normal system airflows by extrapolating manufacturer product data to include system performance data for the lower airflows. The HVAC manufacturer's engineering department acknowledged this approach was accurate as performance varies linearly with airflow, at least down to about 200 cfm/ton. The expanded product data were used to inform the development of the dehumidification strategy, and the equipment manufacturer agreed to reprogram some of the “shared data” control settings just for this study. During the monitoring period, the measured field data (e.g., power) correlated well with the extrapolated values. The expanded product data were also used as modeling inputs to quantify the effectiveness and energy impact of the dehumidification strategy relative to a standard system with normal/typical operation (Section 2.4).

Table 2 shows sample expanded manufacturer product data and the calculated (not measured) greater latent capacities at lower cooling system airflows for the HVAC equipment at the Phase 2 Houston test house. The Goodman manufacturer product data in this table is shaded blue. This product data is unique to the specific AHRI matched furnace, condenser, evaporator coil, and TXV. This sample data is for the 95°F outdoor temperature bin (outdoor dry bulb) and includes total capacity (kBtuh), the sensible-to-total heat ratio (S/T), and total system power (kW) for the three common system airflows (CFM). Note that the full cooling analysis includes all temperature bins, in 5-degree increments, down to 65°F (provided in Appendix A for the Houston Phase 2 test house). Also note that this Goodman product data is available online and is not proprietary.

Home Innovation extrapolated manufacturer data for three lower airflows, and then calculated for all airflows the corresponding temperature drop across the evaporator coil (Delta Temp), the sensible and latent capacities (kBtuh), and the percentage change for the sensible and latent capacities relative to 400 cfm/ton values. The 400 cfm/ton baseline was used so that the latent capacity improvement for 350 cfm/ton could be compared to this most common setting. In Table 2, the percentage increase in latent capacity is shown in blue text, and the corresponding percentage decrease in sensible capacity is shown in red text.

Note that this analysis used steady-state performance data to calculate improvements to latent capacity, and these estimates do not account for momentary transient performance due to equipment cycling, given the time it takes the system to reach full latent capacity (final coil sensible heat ratio) after startup. Still, the results indicate that the increase in latent capacity will be significant during all modes of operation to improve humidity control.

For this data set, the steady-state SHR decreases from 0.83 (400 cfm/ton) to 0.80 during normal cooling (350 cfm/ton), 0.76 during ramping or DH cooling (300 cfm/ton), and 0.73 during DH ramping (250 cfm/ton). Correspondingly, latent capacity increases by 16% during normal cooling (350 cfm/ton), 35.5% during ramping or DH cooling (300 cfm/ton), and 49.1% during DH ramping (250 cfm/ton). The 200 cfm/ton airflow row is included for reference (this airflow was considered early in the project).

The corresponding decreases in sensible capacity indicate the system will run longer to meet the thermostat setpoint temperature. This is a benefit for humidity control—longer cooling cycles at lower airflows. Note that system power is lower at lower airflows; this is because the reduction in air handling power is greater than the increase in compressor power. The analysis in the next section shows that longer run times, despite lower system power, result in some additional annual energy use as expected.

The expanded product data also shows that the increasingly large temperature drop across the evaporator coil presents a potential condensation concern at lower airflows and lower supply air temperatures (none was observed at the test houses).

Table 3 shows the expanded product data for the HVAC equipment at the Phase 2 Monroe, Louisiana, test house. The furnace and condenser are the same as in Houston, but the evaporator coil was selected to increase latent capacity at the expense of a lower SEER. Note that the system was still an AHRI-rated match. The SHR at 350 cfm improved from 0.80 for Houston to 0.74 for Monroe, but the efficiency was reduced from 16 SEER in Houston to 15 SEER in Monroe. This change is significant: latent capacity (steady-state) increases by 28.0% during normal cooling (versus 16.3% for the Houston system), 50.3% during ramping or DH cooling (versus 35.5%), and 71.8% during DH ramping (versus 61.9%). In other words, the coil at the Monroe test house further improved steady-state latent capacity by about 10%–15% compared to the coil at the

Houston test house, depending on the operating mode. At the Monroe test house, SHR decreased from 0.80 (400 cfm/ton) to 0.74 during normal cooling, 0.69 during ramping or DH cooling, and 0.64 during DH ramping.

Table 2. Houston Expanded Product Data

Home Innovation Description		Goodman System Configurator, 6/24/2021 Houston Expanded Cooling Data, 75 IDB, 63 IWB					Home Innovation Calculations				
Mode (% of 350 cfm)	Airflow CFM/ton	ODB	CFM	kBtuh	S/T	kW	Delta Temp	Capacity, kBtuh Sensible	Latent	Change, base 400 Sensible	Latent
	450	95	1350	34.7	0.85	2.81	20.2	29.5	5.2		
Cooling, typical	400		1200	33.9	0.83	2.74	21.7	28.1	5.8		
Cooling, this project (100%)	350		1100	33.5	0.80	2.70	22.6	26.8	6.7	-4.8%	16.3%
Ramping; DH Cooling (~85%)	300		900	32.5	0.76	2.61	25.4	24.7	7.8	-12.1%	35.5%
DH Ramping (~70%)	250		750	31.8	0.73	2.55	28.7	23.2	8.6	-17.4%	49.1%
Reference point (~60%)	200		600	31.1	0.70	2.48	33.6	21.8	9.3	-22.6%	61.9%

Table 3. Monroe Expanded Product Data

Home Innovation Description		Goodman System Configurator, 6/24/21 Monroe Expanded Cooling Data, 75 IDB, 63 IWB					Home Innovation Calculations				
Mode (% of 350 cfm)	Airflow CFM/ton	ODB	CFM	kBtuh	S/T	kW	Delta Temp	Capacity, kBtuh Sensible	Latent	Change, base 400 Sensible	Latent
	450	95	1350	33.5	0.84	2.82	19.3	28.1	5.4		
Cooling, typical	400		1200	33.0	0.80	2.76	20.4	26.4	6.6		
Cooling, this project (100%)	350		1050	32.5	0.74	2.69	21.2	24.1	8.5	-8.9%	28.0%
Ramping; DH Cooling (~85%)	300		900	32.0	0.69	2.63	22.7	22.1	9.9	-16.4%	50.3%
DH Ramping (~70%)	250		750	31.5	0.64	2.56	24.9	20.2	11.3	-23.6%	71.8%
Reference point (~60%)	200		600	31.0	0.59	2.50	28.2	18.3	12.7	-30.7%	92.6%

As discussed in Section 2.1.2, when an air-conditioning system first cycles on, it takes several minutes before the indoor evaporator coil gets cold enough to start condensing the water vapor from the airstream, and several additional minutes to get to its coldest steady-state temperature for full latent capacity. For modern systems, this can take approximately 12–28 minutes, compared to 6–8 minutes for older 10 SEER equipment, because older equipment had a lower final coil SHR of about 0.70–0.75 compared to about 0.80 for newer high-SEER equipment (ACCA Manual S).

Given that the SHR at 250 and 300 cfm airflows ranges from 0.64 to 0.76, it was considered reasonable to expect the cooling system to achieve full latent capacity within the 8-minute ramping period.

Notably, SHR becomes increasingly lower for the lower temperature bins, which makes the enhanced dehumidification strategy even more effective during shoulder seasons.

2.4 Energy Impact Analysis

Modeling was conducted for selected systems using Wrightsoft HVAC software.[4] This software was selected because it provided actual manufacturer operating data by outdoor temperature bin for specific HVAC systems. This was important because these operating characteristics were used to calculate adjusted run times and power by

[4] MiTek Wrightsoft HVAC software: https://www.wrightsoft.com/

temperature bin. Modeling also relied on field data that provided operating hours by mode (ramping, DH ramping, cooling, DH cooling) for each temperature bin.

The expanded product data were used to estimate the energy impact of the dehumidification strategy. The specific AHRI-rated HVAC system was modeled as a standard 400 cfm/ton cooling system. The software provided bin temperature data, broken down by month, and run hours by temperature bin; this data were combined to tabulate normal run hours by month (Table 4 and Table 5).

Home Innovation then calculated the adjusted run hours, by airflow: Adjusted run hours = (run hours) x (actual % of time operated by airflow that month, from field data) x (additional run time factor). The additional run time factor was based on the percentage change in sensible capacity (from the expanded product data) for a given temperature bin and airflow. For this analysis, it was assumed that the additional run time is the same percentage as the change in sensible capacity. For example, if the sensible capacity decreased by 4.8%, then the run time was assumed to be 4.8% longer, and a factor of 1.048 was applied. This assumption was based on the equipment run time fraction being roughly equal to the momentary sensible load divided by the momentary sensible capacity (ACCA Manual S). Again, the steady-state performance data estimates do not entirely capture momentary transient performance due to equipment cycling for both latent capacity and sensible capacity. Startup transients can cause run times to be somewhat longer than indicated by a steady-state model (ACCA Manual S). That said, the estimated longer run times are considered reasonable because these are relative to the steady-state values even where the actual momentary values may be different, and the additional run time was added at the end of the cooling cycle.

The run hours were totaled for each airflow, and then multiplied by system power for each airflow (from the expanded product data), to calculate annual energy. For example, for the 95°F bin temperature in Table 4, the energy for a standard 400 cfm/ton system is 114.61 kWh (red text), and the energy for the system with optimized dehumidification is 119.85 kWh (also in red text), which is 4.6% more than the standard system. Note that this calculation approach is the same approach used by the modeling software; the difference is we are substituting the adjusted power and estimated run times for the lower airflows.

This analysis was done for all temperature bins, and the average increase in energy for the optimized dehumidification system over one complete cooling season was 213 kWh or 4.8%. Again, the higher energy use makes sense given that the lower airflow system must still satisfy the same thermostat setpoint while removing more moisture from the air. Based on the 2020 residential national average electricity cost of $0.1315/kWh (EIA 2022), this translates to an annual cost of $28 to operate this system using the optimized dehumidification strategy.

Table 4. Houston Energy Impact for 95°F Bin Data: Adjusted Annual Operating Hours and Energy Use

Wrightsoft data	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Total Hrs	kWh	KWh
95F bin hours	0	0	1	5	15	21	5	0	0			Change
Run hours at 0.89 run fraction			0.89	4.45	13.35	18.69	4.45	0	0	41.83	114.61	
Adjusted run hours, 350 cfm/ton			0.19	2.38	8.25	9.99	0.93	0.00	0.00	21.7	58.70	
Adjusted run hours, 300 cfm/ton			0.66	2.34	5.54	9.85	3.29	0.00	0.00	21.7	56.63	
Adjusted run hours, 250 cfm/ton			0.15	0.05	0.63	0.22	0.73	0.00	0.00	1.8	4.52	
											119.85	4.6%

Table 5. Houston Energy Impact for 85°F Bin Data: Adjusted Annual Operating Hours and Energy Use

Wrightsoft data	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Total Hrs	kWh	KWh
85F bin hours	0	12	77	140	140	136	108	37	2			Change
Run hours at 0.60 run fraction		7.20	46.20	84.00	84.00	81.60	64.80	22.20	1.20	391.20	970.18	
Adjusted run hours, 350 cfm/ton		0.38	9.68	44.90	51.94	43.61	13.58	1.16	0.00	165.25	403.22	
Adjusted run hours, 300 cfm/ton		3.38	34.12	44.18	34.78	42.92	47.86	10.43	0.26	217.92	514.30	
Adjusted run hours, 250 cfm/ton		4.47	7.57	0.98	3.93	0.96	10.62	13.78	1.14	43.46	99.95	
											1017.47	4.9%

Based on results of energy modeling, the overall annual ventilation provided by the modified ventilation control was calculated for the Houston test house to provide 85% of the overall annual ventilation that would have been provided by the code required continuous rate. This less than 100% result is attributed to design-stage estimates that the annual duration of heating and cooling on-cycles would total 3,000 hours (modeling results showed 2,687 hours based on typical meteorological year data) and that the design ventilation rate was lower than originally estimated; the rate was originally estimated to be twice the continuous rate, but this was reduced to 165% of the continuous rate after deciding to limit the rate to 10% of air handling airflow. To achieve 100% overall annual ventilation, the ventilator control would have needed to be adjusted to operate for up to 2.84 hours of each 4-hour off-cycle period compared to 2 hours originally (per the ventilation control strategy described in Section 2.1.8).

The estimated energy savings to meet code-required ventilation using prioritized ventilation was calculated based on the following: ventilation fan power 40 watts; gas furnace fan power 30 watts at 25% airflow and 200 watts at 100% airflow; 2,687 heating/cooling on-cycle hours and 2,611 off-cycle ventilation hours to meet code requirement (both modeling results). The incremental annual power required for prioritized ventilation was calculated as 291 kWh [(2,611*(40+30) + 2,687*40)/1,000]. The incremental annual power required for standard continuous ventilation was calculated as 1,215 kWh [(8760-2687)*200/1,000]. This represents an annual savings of 924 kWh or $122 at $0.1315/kWh.

2.5 Risk

Using conventional equipment for this project reduced the risk of the unknown for builders. Regarding technical risks, the humidity control strategy did not jeopardize the mechanical reliability of the cooling equipment during the study. The HVAC systems and test houses were closely monitored during and after commissioning. The system airflow settings could be adjusted in the field, so the team had the option to modify those if needed. Further, analysis was conducted in collaboration with engineers from the equipment manufacturer, and commissioning was overseen on-site by factory technical representatives. The technical risks are described below.

A lower airflow results in lower refrigerant vapor pressure and temperature that, at some point, has potential to cause evaporator coil icing. A lower airflow also has potential to reduce superheat that could lead to flooding the compressor with liquid refrigerant and damaging the compressor. The TXV works to prevent this by maintaining target superheat, at least to a point, and most modern systems have a low-pressure switch to prevent damage to the compressor and coil icing. A lower airflow will reduce the supply air temperature, so there is also some increased risk of condensation at supply registers, but this was not observed at any of the test houses.

The duration of the ramping period was limited to eight minutes, and the ramping profile could have been changed or turned off if necessary. After the ramping period, the air-conditioning system would operate in normal cooling mode at full capacity and efficiency (for the selected airflow) so the systems would operate normally and control sensible heat gain during hotter days. In DH mode, it was considered unlikely that the lower sensible capacity would prevent the system from satisfying the thermostat on hot days: Where outdoor temperatures approached design values, it was expected that DH mode would operate significantly less or not at all, i.e., the system would operate at full capacity. DH mode was expected to operate more often during mild outdoor conditions, e.g., 83°–85°F. In the event of a DH mode control malfunction, the system would continue to operate conventionally in normal cooling mode.

Another potential risk of lower airflow is that at some point the distribution system may not provide sufficient air mixing within a room, i.e., not enough “throw” from a supply register. The ratio of supply air to floor area (cfm/sq. ft.) is referred to as the air loading factor and must be sufficient for the supply air outlets that serve the space (ACCA Manual LLH). This was not measured at the test houses. The expectation was that air mixing would not be affected in any meaningful way when the system operated more often in normal cooling mode, i.e., relatively less in ramping or DH modes. There is more potential for insufficient air mixing during shoulder seasons where the system runs more often in ramping and DH modes, but this was not observed at the test houses.

For ventilation, the amount of outdoor air could be adjusted during the monitoring period if necessary. Also, the ventilation control could be switched from the modified “comfort” setting to the standard “code” setting, if necessary. Further, where houses became occupied, it was expected that the occupants would be less likely to simply turn off ventilation systems due to comfort issues.

3 Results

3.1 Sample Field Data

3.1.1 Phase 1 Richmond Hill, Georgia, test house

The HVAC system at this test house was commissioned during 2018, but the control settings were not fully functioning until late May 2019. The cooling season data below is from June 1 through October 1, 2019. There was limited cooling and heating during October, and no cooling after November 1. This model home was sold and occupied during the last week of June 2019, so most of these results are for an occupied home. The data collection hardware was removed in February 2020. Again, the data collection interval for the sensors in this study is one minute; some of the graphs show hourly or daily averages for clarity.

During this cooling period, indoor humidity was below 60% RH for 99% of the time; indoor humidity was below 55% RH for 91% of the time on the first floor and 84% of the time on the second floor (Table 6). The outdoor RH during this period is provided for reference in Figure 8.

Table 6. Richmond Hill Cooling Season RH Data

Richmond Hill, GA 2019 Cooling Season Humidity Data (Jun 1 – Oct 1)		
Indoor Relative Humidity	**1st floor**	**2nd floor**
≤ 60% RH, % of time	99.8%	99.6%
≤ 55% RH, % of time	91.1%	84.3%
> 60% RH, hours	4.5 hours	9.5 hours

The system operated in ramping mode (normal ramping and DH ramping) for nearly 70% of the total air-conditioning operating hours (Table 7). Breaking this down by month, the percentage of time the system operated in ramping period was 75% in June, 68% in July, 67% in August, and 71% in September. The takeaway from this data is that system settings during the ramping period are critical to control indoor humidity.

Table 7. Richmond Hill Cooling Season Operating Data

Richmond Hill, GA 2019 Cooling Season Operating Data (Jun 1 – Oct 1)	
Operating Mode	**Percentage of Total AC Operating Hours**
Ramping: normal ramping and DH ramping	69.9%
Cooling: normal cooling and DH cooling	30.1%
DH cooling only	6.1%

Humidity levels were well within the target limit of 60% RH except for brief, occasional excursions above 60%, as shown in Figure 6 (daily average data) and Figure 7 (hourly average data). Note that the system controlled humidity well, even with continuous on-cycle ventilation and intermittent off-cycle ventilation operating as designed. Figure 8 shows outdoor humidity during this period for reference.

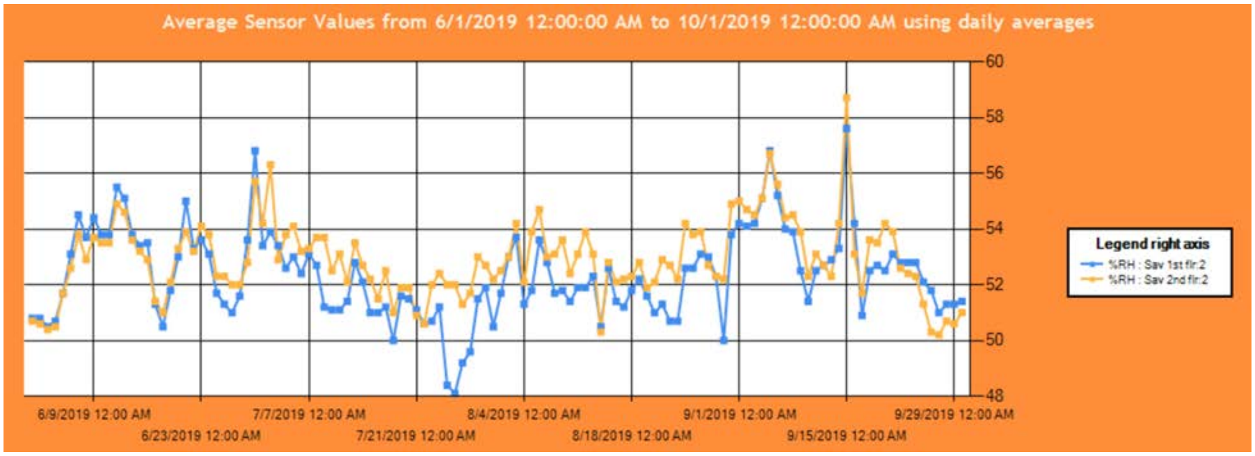

Figure 6. Richmond Hill indoor RH first floor (blue) & second floor (gold), daily average, Jun 1–Oct 1

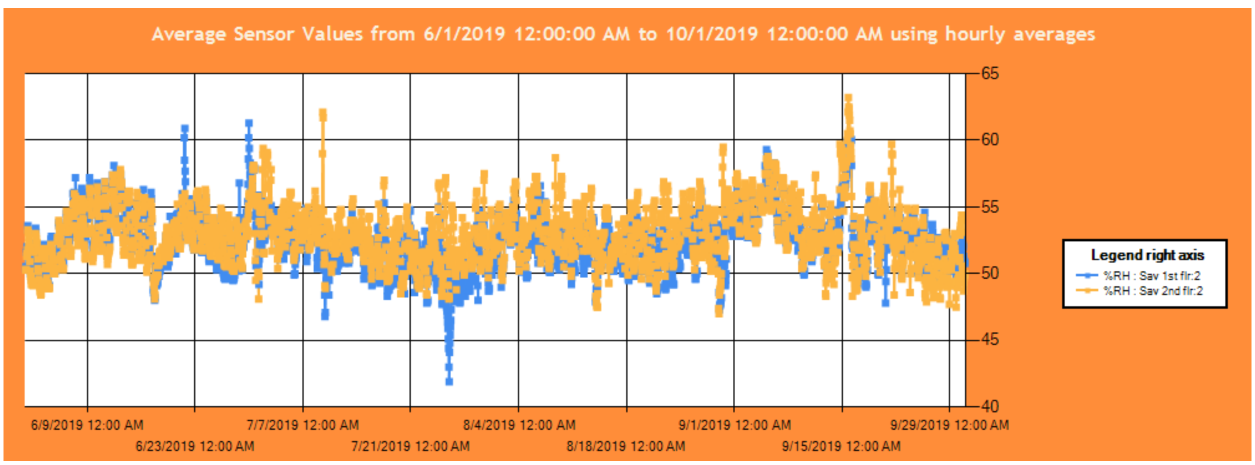

Figure 7. Richmond Hill indoor RH first floor (blue) & second floor (gold), hourly average data, Jun 1–Oct 1

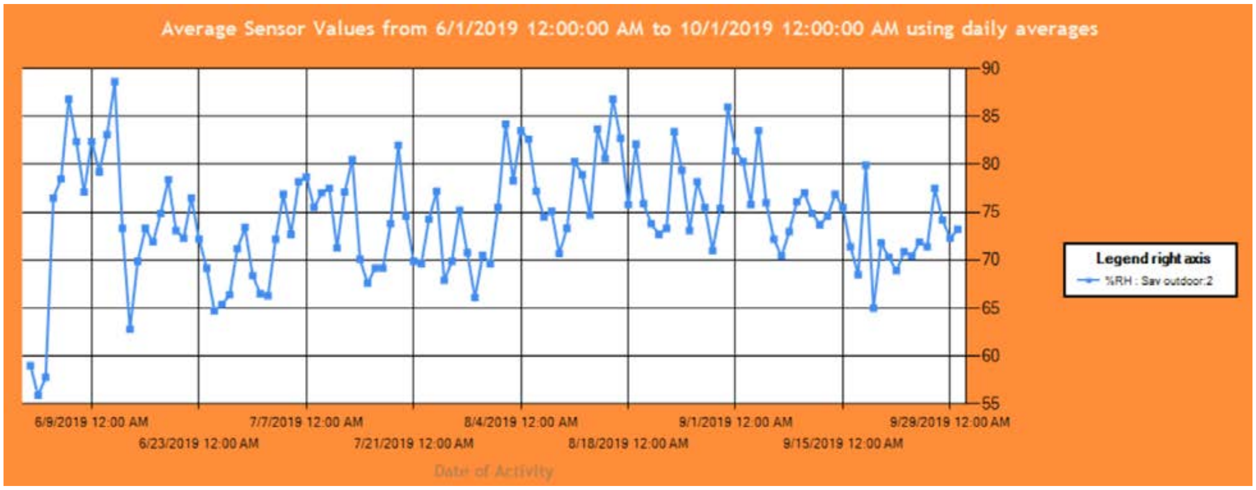

Figure 8. Richmond Hill outdoor humidity, % RH, hourly average data, Jun 1–Oct 1

During the heating season beginning in November, indoor humidity averaged well below 50% RH (Figure 9 and Figure 10). For the heating and cooling period June 1 through February 1, most of the hours where indoor humidity exceeded 60% RH occurred in the fall shoulder season months of October (115 hours, equivalent to less than five days) and early November (23 hours, equivalent to one day) during periods when the HVAC system was not heating or cooling.

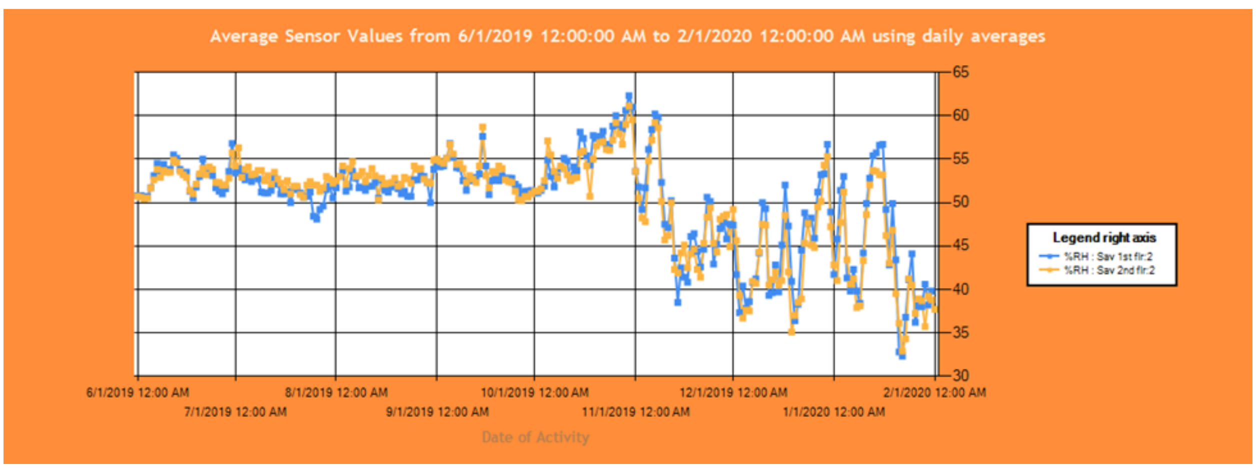

Figure 9. Richmond Hill indoor RH first floor (blue) & second floor (gold), daily average data, Jun 1–Feb 1

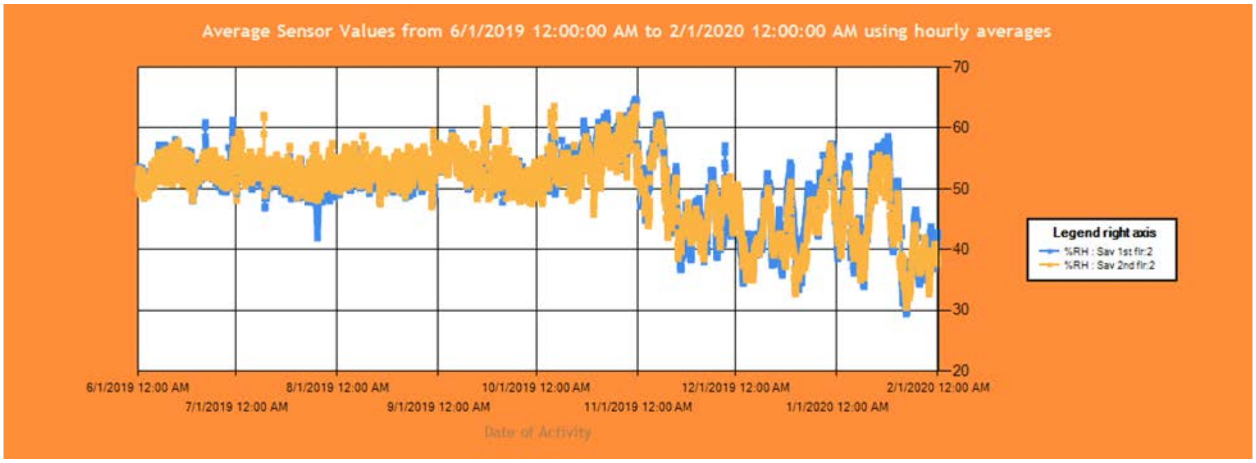

Figure 10. Richmond Hill indoor RH first floor (blue) & second floor (gold), hourly average data, Jun 1–Feb 1

The average outdoor dew point temperature in nearby Savannah from June 1 to October 1 is 71°F, so ventilation during this period would tend to increase indoor humidity although it was well-controlled during cooling on-cycles.[5] Ventilation during the heating season at times increased indoor humidity, but more often decreased indoor humidity as the average outdoor dew point from December 1 to March 1 is 39°F.

Figure 10 shows the effect of off-cycle ventilation during an example two-day period in June. The air handling power data (shown in blue; actual power is double) shows when off-cycle ventilation occurs (low power periods, about every four hours) and its corresponding effect on indoor humidity (first floor RH shown in gold, second floor RH shown in red). During this period, indoor humidity initially increases by up to about 2%–3%, but the magnitude of this effect will vary by outdoor conditions.

Figure 11. Richmond Hill: effect of off-cycle ventilation on indoor humidity, first floor (% RH, gold) & second floor (% RH, red); air handling power (blue)

[5] Dew point data source: https://www.redcalc.com/weather-station-data/

3.1.2 Phase 2, Houston, Texas, Test House

The HVAC system at the test house and baseline house next door were commissioned in January 2020. The systems began operating in cooling mode during March 2020. Data were collected through 2021 and into 2022. Both houses were model homes and not occupied during this period.

Indoor humidity at the test house was well below the project's 60% RH upper limit target and below 55% most of the time; compared to the baseline house, the test house performed better, particularly for the 55% RH metric (Table 8). Table 9 shows indoor humidity data at the test house broken down by month.

Table 8. Houston Cooling Season Comparison Data

Houston Cooling Season Data: 2020 (Mar 12–Dec 1) and 2021 (Mar 23–Aug 5)				
	Test House		Baseline House	
Metric	2020	2021	2020	2021
≤ 60% RH, % of time	95.8%	99.8%	85.1%	75.6%
≤ 55% RH, % of time	75.8%	87.8%	9.0%	4.0%

Table 9. Houston Cooling Season Humidity Data

Houston Test House 2020 Cooling Season Humidity Data (Mar 12–Dec 1)										
	Percentage of Time Indoor Relative Humidity is Below Target RH									
Target RH	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Total
≤ 60% RH	64%	86%	100%	100%	100%	100%	100%	100%	100%	96%
≤ 55% RH	0%	28%	72%	81%	81%	99%	100%	100%	92%	76%

Table 10 shows the percentage of the total air-conditioning operating hours by operating mode: (ramping, DH ramping, cooling, and DH cooling) for six months of the cooling season.

During March, the cooling system operated 83% of the time in the ramping period (81% in DH ramping) and 17% in cooling (all 17% in DH cooling), so 98% in DH mode. By July, the system operated 40% in ramping (4% in DH ramping) and 60% in cooling (1% in DH cooling).

The operating data shows that the ramping period is important for the entire cooling season but critical during shoulder seasons, and that dehumidification mode is important during shoulder seasons but becomes less important during the heat of the summer.

Table 10. Houston Cooling Season Operating Data

Houston Test House 2020 Cooling Season Operating Data (Mar 12 – Sep 1)						
	Percentage of Total AC Operating Hours					
Operating Mode	Mar	Apr	May	Jun	Jul	Aug
Ramping: normal ramping and DH ramping	83%	85%	65%	45%	40%	46%
DH ramping only	81%	53%	14%	1%	4%	1%
Cooling: normal cooling and DH cooling	17%	15%	35%	54%	60%	53%
DH cooling only	17%	10%	15%	3%	1%	2%

Indoor RH is consistently 4%–8% lower at the test house compared to the baseline house (Figure 11). Note that the systems at both houses were commissioned with the oversight of factory representatives; the system at the baseline house is working normally and is considered representative of typical systems that are not optimized for dehumidification.

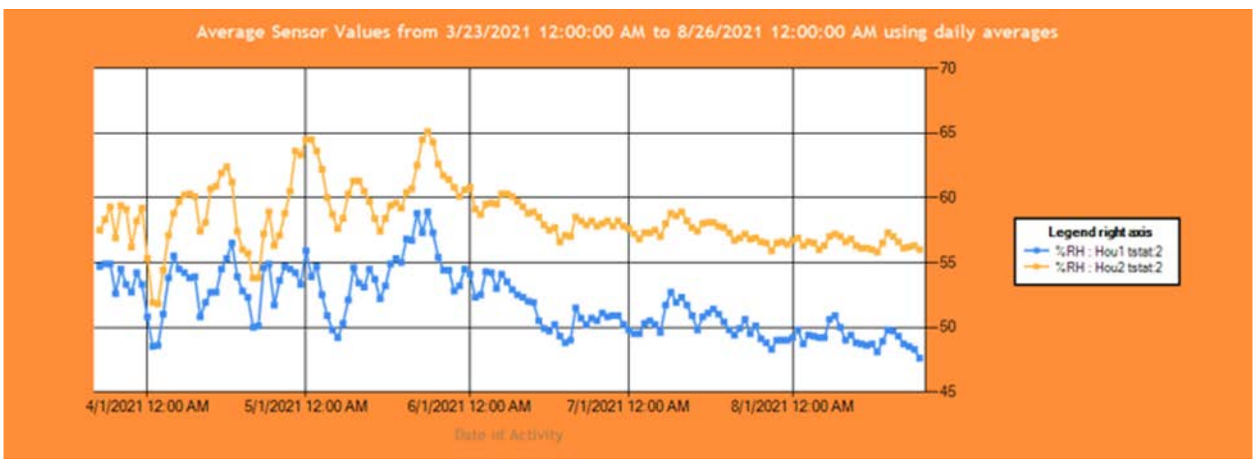

Figure 12. Houston indoor RH: test house (blue); baseline house (gold); daily averages Mar 23–Aug 26, 2021

The indoor humidity at the test house was notably lower than at the baseline house for the rest of the cooling season, although the difference is less during the shoulder season month of November (Figure 12).

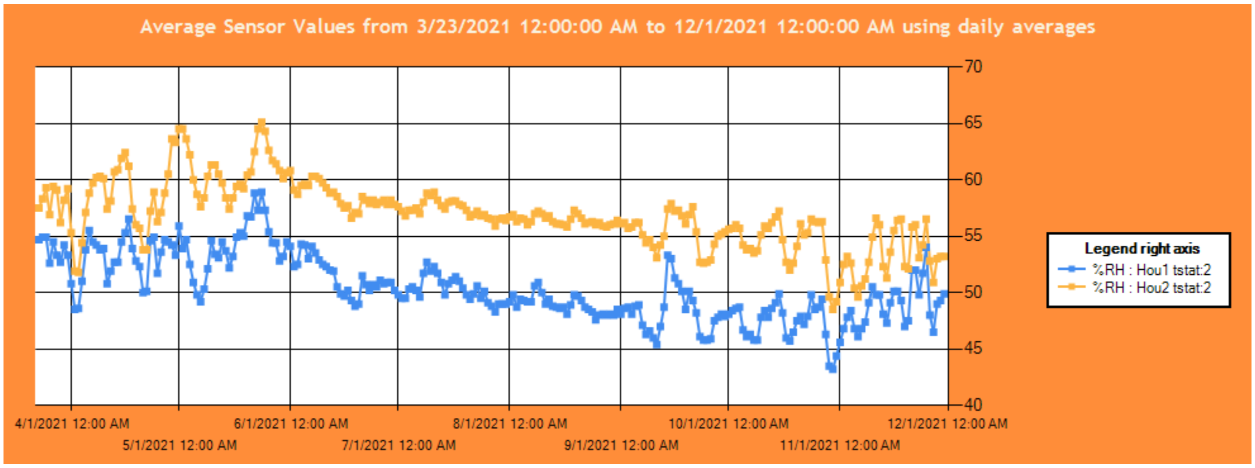

Figure 13. Houston indoor RH: test house (blue); baseline house (gold); daily averages Mar 23–Dec 1, 2021

The results from the 2021 cooling season were consistent with the 2021 results (Figure 13).

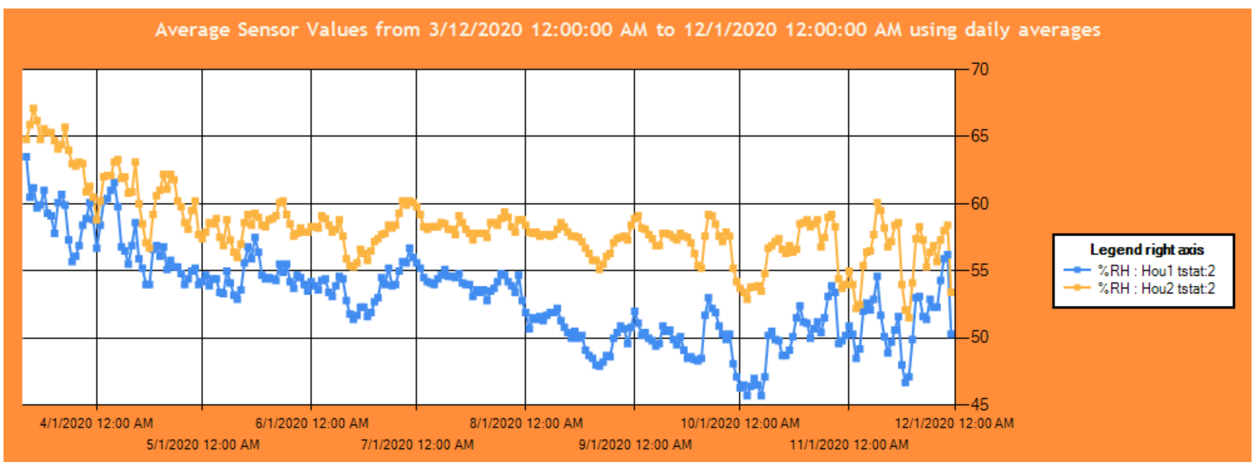

Figure 14. Houston indoor RH: test house (blue); baseline house (gold); daily averages Mar 12–Dec 1, 2020

To illustrate the ventilation strategy, Figure 14 shows ventilation during heating on-cycles and during heating off-cycles for an example period in March. The fan-powered ventilator (power shown in gold) operates during furnace operation in heating (power shown in blue) and during off-cycle ventilation. During off-cycles, the furnace operates at very low airflow and power. The indoor dew point temperature (shown in red) correlates with the average 50.7°F outdoor dew point temperature in Houston from December 1 through March 1. During this period, the indoor dew point temperature at the baseline house is the same as the test house.

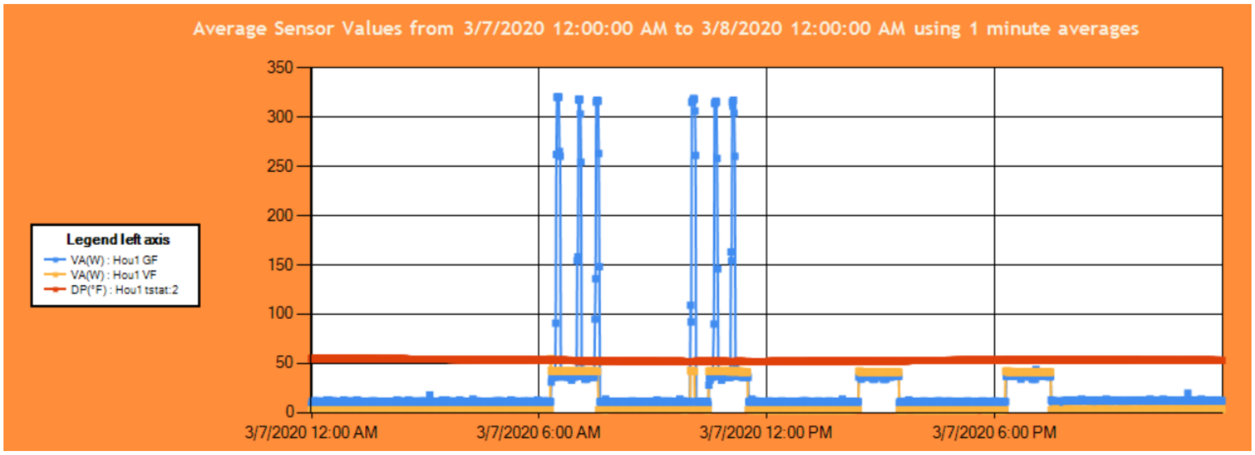

Figure 15. Houston example heating on-cycle and off-cycle ventilation periods to illustrate the ventilation strategy: air handling power (blue), ventilation fan power (gold), indoor dew point temperature (red)

3.1.3 Phase 2, Monroe, Louisiana, Test House

The HVAC system at this test house was commissioned in Oct 2019, but cooling was not started until late May 2020. The sensor data indicated the system was not ramping or operating in dehumidification mode despite high indoor humidity. The HVAC partner conducted a system check on-site and confirmed settings were correct but ramping and dehumidification still were not operating as intended. Further troubleshooting was a challenge due to the pandemic. In late May 2021, the HVAC system was checked again and a furnace control wiring issue was discovered and corrected. After “drying out” the house during the first two weeks in June, the indoor RH was below 60% RH for 99% of the time through October 1 (Table 11). The house was sold and occupied on June 14, 2021, so most of this data is for an occupied house. The last day of cooling in 2021 was October 21.

It is important to note that the factory reprogrammed shared data were not fully implemented here. The system operated as intended during cooling: 100% airflow (350 cfm/ton) during normal cooling, and 85% airflow (300 cfm/ton) during DH cooling. During the ramping period, the system operated as intended during normal ramping at 85% (300 cfm/ton, after the first 30 seconds at 175 cfm/ton), but when there was a signal for dehumidification during ramping, the airflow did not decrease to 70% (250 cfm/ton). Despite this, the cooling system performed well for the rest of the cooling season in 2021 (Figure 15). Compared to the same cooling period in 2020 (Figure 16), ramping and dehumidification mode improved indoor humidity by at least 2%–3% RH. If DH ramping had been operational, the results would have been even better.

Table 11. Monroe Cooling Season Humidity Data

Monroe, LA 2021 Cooling Season Humidity Data						
	Percentage of Time					
Indoor Humidity	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Average**
≤ 60% RH	54.7%	99.6%	99.9%	99.9%	97.5%	90.0%
≤ 55% RH	0.0%	13.2%	50.1%	35.5%	9.7%	22.6%
Above 60% RH	45.3%	0.4%	0.1%	0.1%	2.5%	10.0%

Figure 16. Monroe indoor humidity (% RH), daily averages Jun 1–Oct 21, 2021

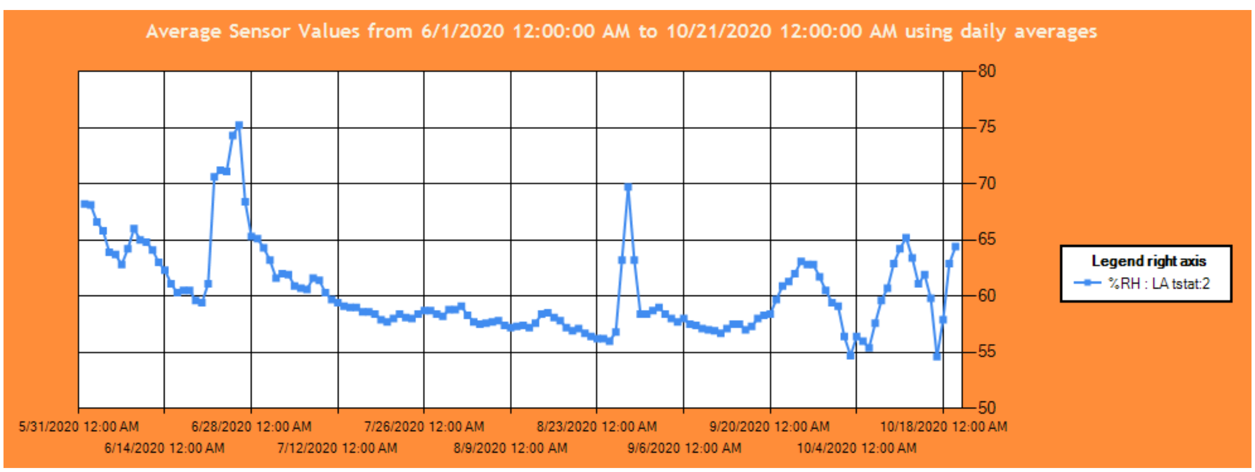

Figure 17. Monroe indoor humidity (% RH), daily averages Jun 1–Oct 21, 2020

Table 12 shows the system operated in ramping or DH cooling more than 98% of the total AC operating hours in June, 73% in July, 65% in August, 76% in September, and 94% in October. Recall there was no ramping in DH mode, and because the power signatures were the same for ramping and DH cooling, those could not be reported separately.

Table 12. Monroe Cooling Season Operating Data

Monroe, LA 2021 Cooling Season Operating Data					
	Percentage of Total Air-Conditioning Operating Hours				
Operating Mode	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**
Ramping or DH cooling	98.7%	73.4%	65.5%	76.7%	94.2%
Normal Cooling after ramping	1.3%	26.6%	34.5%	23.3%	5.8%

The 2019–2020 heating season ran from late October through mid-March; there was no heating after March 15.

Figure 18 shows ventilation during heating on-cycles and during off-cycles for an example period in December. The fan-powered ventilator (power shown in gold) operates during furnace operation in heating (power shown in blue) and during off-cycle ventilation. During off-cycles, the furnace operates at very low airflow and power. The indoor dew point temperature (shown in red) correlates with the average 37.7°F outdoor dew point temperature for Monroe in December.

Figure 19 shows indoor dew point temperature (blue) during a sample period November 1 through March 1. The average dew point temperature for this period is 41.8°F). The highest readings occur during shoulder season periods (November in this data set) with no heating (furnace operation is shown in red) and high outdoor humidity (outdoor dew point is shown in gold; note that this sensor failed in late December).

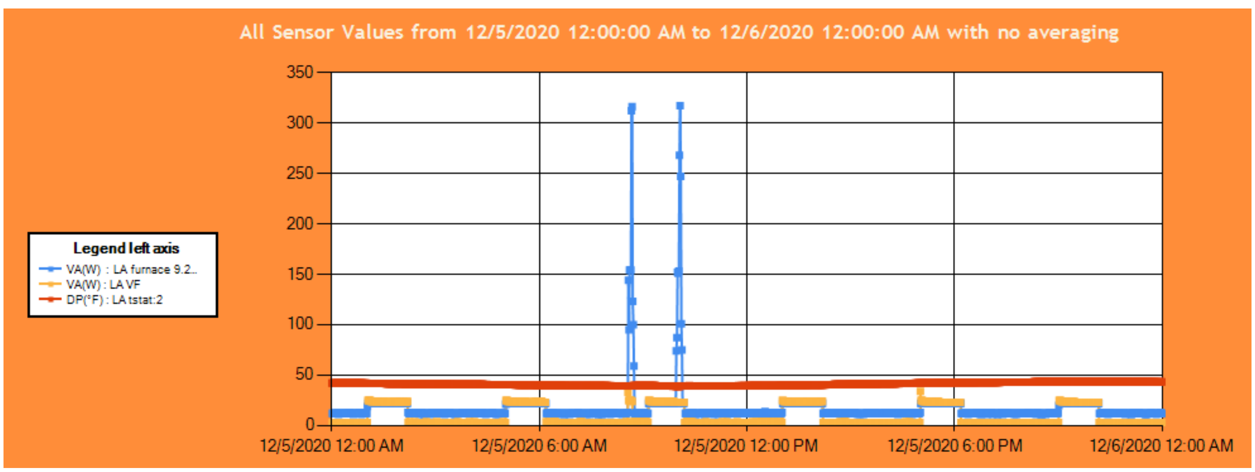

Figure 18. Monroe example on-cycle and off-cycle ventilation periods: air handling power (blue), ventilation fan power (gold), indoor dew point temperature (red)

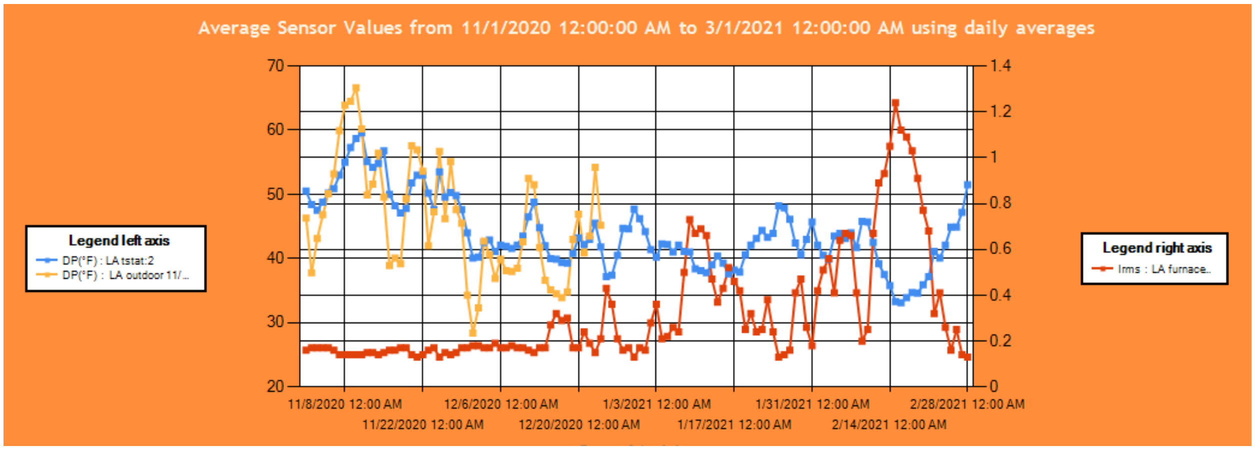

Figure 19. Monroe winter data: indoor dew point temperature (red), outdoor dew point temperature (gold), furnace operation (amps, red)

3.2 Key Takeaways

Key takeaways from the results of this study are presented here for the test houses in Richmond Hill, Georgia (Phase 1), Houston, Texas (Phase 2), and Monroe, Louisiana (Phase 2). Results could vary for different locations, HVAC systems, house configuration, and house construction. Indoor humidity at the test houses did not exceed 60% RH during the monitored cooling season for 99% of the time in Richmond Hill, 96% of the time in Houston, and 90% of the time in Monroe.

- The dehumidification strategy improved the steady-state latent capacity of the HVAC system at design conditions by 16% to 49% at the Houston test house and by 28% to 71% at the Monroe test house depending on which mode the system was operating in: ramping, DH ramping, cooling, or DH cooling. Note that these steady-state estimates do not account for the transient performance due to equipment cycling. That said, the results indicate that humidity control will be improved significantly.
- The SHR at design conditions improved in Houston from 0.83 to 0.80 during normal cooling, 0.76 during ramping or DH cooling, and 0.73 during DH ramping; in Monroe, SHR improvement was greater, from 0.80 to 0.74, 0.69, and 0.64 for the same operating modes, respectively. Notably, SHR becomes increasingly lower for the lower temperature bins, which makes the enhanced dehumidification strategy even more effective during shoulder seasons.
- The greater improvement in latent capacity at the Monroe test house represents a case where substituting a different evaporator coil can further improve latent capacity if the builder is willing to trade some rated efficiency (SEER) for better dehumidification (see RQ5 in Section 3.3 for additional details).

- The dehumidification strategy did not jeopardize the mechanical reliability of the cooling equipment.
- Indoor humidity during the cooling season was consistently 4%–8% lower at the Houston test house compared to the baseline house next door.
- Successful results using an air conditioner or heat pump with a single-stage compressor show that a two-stage or variable-stage compressor system is not required to control indoor humidity to acceptable levels. There is a common perception that outdoor units with two-stage or variable-speed compressors provide better dehumidification, but generally this is not the case. That said, enhanced dehumidification could improve dehumidification for multistage systems as well.
- The control settings for the ramping period (at the beginning of each cooling cycle) and the dehumidification mode (during ramping and during normal cooling after ramping) are critical to control indoor humidity, particularly during shoulder seasons, in hot-humid climates. For Richmond Hill, the amount of time where indoor humidity rose above 60% RH was less than 10 hours during the cooling season (6/1–10/1) and 138 hours (equivalent to about 6 days) during October and November. For Houston, with a longer cooling season (3/12–12/1 in 2020), the total time above 60% RH was 237 hours (equivalent to about 10 days over nearly 9 months) compared to 823 hours at the baseline house. The difference was even more pronounced for time below 55% RH during 2020: 4,784 hours at the test house versus 568 hours at the baseline house. While it remains true that the system does not dehumidify when there is not a call for cooling during the shoulder seasons, the amount of time that this results in indoor humidity above 60% was relatively short, in part because the house was dryer to begin with and overall, the amount of time during the year when a supplemental dehumidifier may be desired by some is greatly reduced using enhanced dehumidification.
- The optimized dehumidification system at the Houston test house increased the estimated annual energy by 212 kWh (4.8%) compared to the same system with standard control settings. This translated to $28/year based on national average electricity prices. Compared to a system with a multispeed ECM air drive (now the required minimum), the estimated annual energy increased by only 2% ($12/yr).
- The control strategy for prioritized ventilation has good potential for saving energy and improving occupant comfort. At the Houston test house, the total amount of ventilation on an annual basis fell short (by about 15%) of the total annual ventilation that would have been provided by using the continuous

ventilation required by building code. The control logic could be modified to provide 100% of the required annual ventilation; this would still capture energy savings and significantly reduce the amount off-cycle ventilation time and the risk of associated comfort issues. The estimated annual energy cost savings using prioritized ventilation compared to continuous ventilation, based on providing 100% of the total annual ventilation, was $121.

- The strategies used in this study are applicable across various equipment brands, models, and efficiency levels, as well as to a broad range of homes in hot-humid climates. Results will vary by specific equipment, location, and house configuration and construction.
- Some of the control settings for optimized dehumidification are currently available for conventional equipment. It would be straightforward for manufacturers to modify their control settings to optimize systems and achieve the successful results of this study.

3.3 Key Research Findings

This study was designed to answer the research questions presented in Section 1.3. The results of this study are explained below relative to how they answer each research question (RQ). The research questions are repeated here for reference.

RQ1. What is the optimum control strategy to improve dehumidification by the central air-conditioning system in hot-humid climates?

The optimum control strategy for this project relied on single-stage air conditioners or heat pumps, furnaces or air handlers with variable-speed ECM air drives, and standard selections for control settings for this type of equipment. Some control settings were modified by equipment manufacturers specifically for this project. The control settings were:

- Normal cooling airflow: 350 cfm/ton.
- Ramping profile: 50% airflow for the first 30 seconds, 85% airflow for 7.5 minutes, 100% airflow for the duration of the cooling cycle, and 50% airflow for 30 seconds after the compressor cycles off.
- Dehumidification mode: set to activate at 55% indoor RH (85% airflow during normal cooling and about 72% airflow during ramping).
- Fan-only airflow: 25% of normal cooling airflow; used for heating/cooling off-cycle ventilation; continuous fan-on circulation was not used.

The modeling results and field data showed the dehumidification strategy developed for this study was an effective approach to improve the latent capacity of HVAC systems and practical to install. The strategy controlled indoor humidity to well below the target criteria for this project and performed better than the baseline cases as applicable. The strategy relied on conventional equipment that was straightforward to set up in the field. Measured field data correlated well with expanded manufacturer product data that was used for analysis and modeling. Specific data results are explained in RQ4 for effectiveness and RQ5 for energy impact. The incremental cost of the equipment is addressed in the Discussion section.

The outdoor unit (air conditioner or heat pump) performed as expected, so the additional expense of installing a two-stage or variable-speed compressor system is not required for successful results. The variable-speed ECM air drive (furnace or air handler) was required to accurately provide the range of airflows needed for the dehumidification mode, ramping profile, and off-cycle ventilation periods. It also provides significant energy savings and quiet operation. An air handler with a multispeed ECM typically lacks the required range of airflow settings, ramping profile options, and input terminal for the dehumidification signal.

The thermostatic expansion valve (TXV) at the indoor evaporator coil is required to maintain proper operation of the refrigerant system at lower airflows without jeopardizing equipment durability. A TXV is standard on many coils or available as a low-cost option. This research also quantified a case for further improving latent capacity by substituting a different evaporator coil where the builder is willing to trade some rated efficiency (SEER) for better dehumidification.

The thermostat with integral dehumidistat was required to send the dehumidification signal to the control board within the air handler so that the system could operate in dehumidification mode as needed. This was an important component for the successful results, particularly during shoulder seasons. Notably, successful results did not rely on overcooling (normally set at the thermostat).

The cooling system airflows for normal cooling, ramping, and dehumidification modes were selected to optimize the balance between latent capacity, power, supply air temperature, and protecting against equipment malfunction. The normal cooling airflow (350 cfm/ton) is a standard selection normally provided by HVAC equipment manufacturers, and it provides a significant improvement to latent capacity compared to the typical setting (400 cfm/ton). The lower airflows used during ramping and dehumidification mode did not present any issues with equipment performance. Manufacturers commonly offer ramping profile and dehumidification mode options with selected high-efficiency air handlers, but the optimized settings for this project relied to some extent on reprogrammed controls by the manufacturer.

RQ2. What is the optimum control strategy to integrate prioritized ventilation in hot-humid climates?

The optimum ventilation control strategy for this study relied on supply-type whole-house mechanical ventilation using a fan-powered inline ventilator to control the airflow rate, and control that was modified by the manufacturer. The control settings were:

- Ventilation rate: increased to 10% of HVAC system airflow (165% of the continuous rate required by code for this study)
- Deactivation limits: 95°F outdoor high limit; 30°F outdoor low limit; 65% RH indoor high limit
- Ventilation occurs during entire heating and cooling calls
- If heating or cooling calls do not occur for twice the "ventilation time" [defined as (62.2-2010 continuous airflow rate / measured airflow rate) x 60] during a four-hour cycle period, then the control will ventilate at the end of the four-hour cycle period to ensure that ventilation occurs for twice the ventilation time, unless restricted by short-cycle protection or by ventilation air temperature limits or indoor RH limits.

The modeling results show the ventilation strategy developed for this study provided benefits regarding comfort and energy efficiency. However, the total amount of ventilation on an annual basis fell short (by about 15%) of the total annual ventilation that would have been provided by using the minimum continuous ventilation rate required by ASHRAE Standard 62.2-2010. Results are further explained in RQ6 for effectiveness and RQ7 for energy impact. The incremental cost of the equipment is addressed in the Discussion section.

The higher ventilation airflow rate, relative to the minimum continuous rate required by the International Residential Code, maximized ventilation during heating and cooling on-cycles and minimized ventilation time during heating and cooling off-cycles. The higher ventilation airflow rate did not overwhelm the HVAC system during heating or cooling, i.e., the system was able to maintain the temperature setpoints and acceptable indoor humidity levels during heating and cooling periods. During off-cycles, the ventilation air was mixed with house air, which tempered the circulated air, e.g., for a 3-ton cooling system, 105 cfm ventilation air (10% of system airflow) was mixed with 157 cfm house for a total of 262 cfm circulated air (25% of cooling airflow). Operating the furnace/air handler at 25% cooling airflow during off-cycle ventilation was effective to minimize power; it is expected that this airflow will also reduce comfort issues, as this low rate distributed through the house should be less noticeable, although this will vary by individual sensitivities, outdoor conditions, and the use of ventilation limit settings. Note that off-cycle ventilation during cooling has some potential to evaporate water from the coil into the airstream and increase indoor humidity, but there was typically a long period between the end of a cooling cycle and the beginning of a cooling off-cycle ventilation period that allowed most of the water on the coil to drain. Ultimately, indoor humidity remained well within acceptable limits.

RQ3. For these humidity control strategies, what are the metrics, including latent efficiency, to quantify the system performance including energy savings and cost-effectiveness?

Early in the project, there was concern and reluctance to introduce an additional regulatory metric that could be widely used to quantify and compare the latent efficiency of cooling systems. Instead, the effort evolved to identify a metric for latent effectiveness that could be useful to quantify the performance and energy impact. This study identifies SHR as a useful metric to evaluate the performance of a system. SHR varies by equipment selection, entering wet-bulb and dry-bulb temperatures, outdoor condensing temperature, and airflow across the evaporator coil. This study did not identify a specific target or target range of SHR, but this study showed the significant improvement that a lower SHR can have on humidity, and importantly an approximate lower limit for airflow that can contribute to this lower SHR without jeopardizing the operation of the cooling system.

RQ4. How effective is indoor humidity control for air-conditioning systems with optimized dehumidification compared to typical baseline operation?

The lower the airflow, the greater the increase in latent capacity, plus the system will run longer and dehumidify even better due to the corresponding decrease in sensible capacity. Analysis of the expanded product data for the HVAC system at the Phase 2 Houston test house showed that compared to typical systems, lower airflows can increase steady-state latent capacity by 16% during cooling, 35% during ramping or DH cooling, and 49% during DH ramping, at design conditions. For the HVAC system at the Phase 2 Monroe test house, the results were greater: 28%, 50%, and 71%, respectively.

The greater improvement at the Monroe test house was due to a different indoor evaporator coil (still an AHRI-rated match), which was selected to provide better latent capacity (lower SHR) but reduced the SEER rating from 16 to 15.

The criteria used to evaluate the effectiveness of the dehumidification strategy was how often measured indoor humidity exceeded 55% RH and 60% RH. Indoor conditions at the test houses were monitored continuously during the study; the data collection interval was one minute, and data could be viewed live at any time and the entire data set was downloaded for analysis purposes. For the Phase 1, Richmond Hill, Georgia, test house during the 2019 cooling season, indoor humidity was below 60% RH for 99.8% of the time on the first floor and 99.6% of the time on the second floor. This represents 4.5 hours (first floor) and 9.5 hours (second floor). Indoor humidity was below 55% RH for 91.1% of the time on the first floor and 84.3% of the time on the second floor. The good results were attributed largely to the system operating in ramping mode for 69.9% of the total air-conditioning operating hours, versus 30.1% in cooling mode after ramping (6.1% in DH cooling mode, and 24.0% in normal cooling mode). Unlike the Phase 2 test houses, there was not a local base case for comparison, so the results were compared to performance criteria defined during the design stage of the project: acceptable upper limit for indoor humidity of 60% RH, and additional benchmark for indoor humidity of 55% RH.

The field data at the Houston location shows that indoor RH was consistently 4%–8% lower at the test house compared to the baseline house next door. The systems at both houses were commissioned with the oversight of factory technical representatives, so the system at the baseline house was confirmed to be working well and is considered representative of typical systems. During the 2020 cooling season (3/12/20 – 12/1/20), the indoor humidity was below 60% RH for 95.8% of the time at the test house compared to 85.1% of the time at the baseline house, and below 55% RH for 75.8% of the time at the test house compared to 9.0% of the time at the baseline house.

The good results at the test houses are primarily due to the amount of time the cooling system operates in ramping and DH modes (DH ramping and DH cooling), particularly during the early cooling season. At the Houston test house:

- During March, the cooling system at the Houston test house operated 83% in ramping mode (81% in DH ramping) and 17% in cooling mode (all 17% in DH cooling), so 98% in DH mode; indoor humidity was below 60% RH for 64% of the time.
- During April, the system operated 85% in ramping mode (53% in DH ramping) and 15% in cooling mode (10% in DH cooling), so 63% in DH mode; indoor humidity was below 60% RH for 86% of the time.
- During May, the system operated 65% in ramping mode (14% in DH ramping) and 35% in cooling mode (15% in DH cooling), so 29% in DH mode; indoor humidity was below 60% for 100% of the time.
- During June, July, and August, the system operated 40%–46% in ramping mode (1%–4% in DH ramping) and 54%–60% in cooling mode (1%–3% in DH cooling), so only 3%–5% in DH mode; indoor humidity was below 60% RH for 100% of the time.

The Monroe test house did not have a baseline house for comparison, but there was data from two cooling seasons with different equipment operating parameters. During the first season, ramping and dehumidification were not functional. During the second season, after a wiring issue was diagnosed and corrected, ramping and dehumidification were operational, except dehumidification did not work during ramping periods (because the factory reprogrammed shared data were not fully implemented here). Despite this, the system performed well during the second season where indoor humidity was at least 2%–3% RH lower than the first season (not normalized for weather); a full shared data update would have provided even better results. Note that this house was occupied for most of the second cooling season.

The amount of time where indoor humidity exceeded 60% RH was the criteria to identify if supplemental dehumidification would be required or desired. Indoor humidity exceeded 60% RH during the cooling period for 4.5 hours downstairs and 9.5 hours upstairs at the Richmond Hill test house (June 1 through October 1). At the Houston test house, this was 252 hours (4% of the time) during the 2020 cooling season: 164 hours in March and 88 hours in April (compared to 823 hours total or 13% of the time at the baseline house). At the Monroe test house, this was 326 hours in June (the system had been adjusted in late May and the house was still drying out), but only 3 hours in July, less than 1 hour in August and September, and 18 hours in October. Based on the 60% RH upper limit, one could argue that a dehumidifier might be desired for 11 days at the Houston test house, 15 days at the Monroe test house, and 1 day at the Richmond Hill test house.

The systems at the test houses did not rely on overcooling to effectively control indoor humidity. Early on there was a plan to experiment with overcooling by up to 1°F because beyond that there is more potential for comfort issues and an energy penalty. The original thermostat during Phase 1 had an adjustable cooling operation band (e.g., system on at 75.0°F, and off at 74.5°F or 74.0°F) that would allow overcooling to be dialed in, but it turned out that thermostat did not have the capability to provide a signal for dehumidification, so it was replaced. The replacement thermostat had an overcooling logic that was not compatible with project goals: it simply lowered the setpoint by whichever overcooling amount was selected and did not increase run time. For example, if the cooling setpoint was 75°F, and 1°F overcooling was selected, the cooling system would activate at 74°F and cycle off as soon as the thermostat was satisfied for a 74°F setpoint, i.e., at about 73.7°F. Ideally, a thermostat would activate cooling at a 75°F setpoint and continue operating down to 74°F (overcool by 1°F); this would allow the system to run longer and dehumidify better.

RQ5. What is the energy impact of using the optimized dehumidification strategy compared to typical control settings?

Energy modeling was conducted for the Houston test house based on using inputs from the expanded manufacturer product data and field data. The estimated increase in energy use for optimized dehumidification settings versus standard settings was 4.8% over the course of a typical summer: 4,673 kWh for the optimized settings compared to 4,461 kWh for the standard settings, for a difference of 212 kWh. This translates to $28 annually, based on the national average 2020 residential cost of electricity of

$0.1315/kWhr (EIA 2022). It is expected, but not calculated here, that an auxiliary dehumidifier would cost at least this much to operate.

The HVAC system at the Monroe test house used the same furnace and condenser as the Houston test house but a different evaporator coil that increased the system latent capacity but decreased the SEER rating from 16 to 15. If the Monroe HVAC system with optimized dehumidification settings was substituted in the Houston test house, that system would use an estimated 8% more energy compared to that system with standard settings. This translates to $47 annually, based on national average electricity rates, for even better dehumidification capability.

If a furnace with a multispeed ECM air drive (versus a variable-speed ECM air drive) is substituted at the Houston test house (standard settings, changes SEER from 16 to 15.5), this system would use 2.7% more energy annually compared to the Houston test house system with standard settings. For this basis of comparison, the optimized system at the Houston test house would use 2.0% more energy or about $12/yr.

RQ6. How effective is prioritized ventilation with respect to ASHRAE 62.2-2010?

The total annual ventilation was estimated for the Houston test house based on the ventilation control strategy and the modeled results for annual operating hours (on-cycles and off-cycles). This was compared to the total annual ventilation required by ASHRAE 62.2-2010, based on the minimum continuous ventilation rate. The expected annual ventilation was estimated to provide 84.9% of the required annual ventilation. Originally, the strategy called for setting the ventilator at double the required continuous rate—this would have provided 100% of the required annual ventilation—and the control settings were modified accordingly. However, during the design stage, it was decided that the maximum ventilation airflow should not exceed 10% of the air-conditioning airflow, so a lower ventilation airflow was selected (e.g., for a 3-ton system and a required continuous rate of 63.5 cfm, the 105 cfm setting provided 165% of the continuous rate instead of double).

As is, the quantity of annual operating hours for heating/cooling off-cycle ventilation at the Houston test house is estimated to be 70% less than that of a standard system operating at the minimum continuous ventilation rate. If the control logic was adjusted to provide 100% of the annual required ventilation, the quantify of off-cycle ventilation hours would still be 57% less, and this translates to 3,462 hours less operation annually during heating/cooling off-cycles, which would be effective to significantly reduce the potential for occupant comfort issues.

RQ7. What are the potential annual HVAC energy savings using prioritized ventilation compared to typical supply-type continuous ventilation?

For the prioritized ventilation strategy at the Houston test house, annual energy was estimated based on the ventilator fan (40 W) operating during all on-cycle periods

(2,687 hours), and the ventilator fan and furnace at 25% airflow (30 W) operating during the off-cycle hours that will provide 100% required ventilation on an annual basis (2,622 hours). For the alternative ventilation with minimum continuous ventilation rate, annual energy was estimated based on the furnace operating during all off-cycle periods (6,073 hours) at normal cooling airflow (200 W); it is assumed there is a motorized damper instead of a fan-powered ventilator with a built-in motorized damper, so no additional power is required during on-cycles. Typically, furnaces or air handlers with multispeed ECM air drives, unlike variable-speed ECM air drives, do not have an option to select a very low airflow for continuous fan operation (e.g., 25% of normal cooling airflow). The estimated difference in annual energy between prioritized ventilation (291 kWh) and continuous ventilation (1,214 kWh) is 923 kWh, or $121/yr at the 2020 national average electricity cost.

RQ8. What is the potential to improve occupant comfort using prioritized ventilation and enhanced dehumidification compared to baseline AC systems?

Optimized dehumidification provides lower indoor humidity, which improves comfort, so occupants will be less likely to lower the thermostat setting during shoulder seasons to control humidity and may even raise the thermostat setting during the cooling season to be just as comfortable at a higher temperature.

Prioritized ventilation improves comfort because a greater proportion of outdoor ventilation air is provided and conditioned during on-cycles, and there is much less off-cycle ventilation that can be a comfort issue year-round. Further, off-cycle ventilation is more likely to occur during mild conditions when ventilation is less objectionable.

4 Discussion

4.1 Optimized Dehumidification

Generally, the results of implementing the enhanced dehumidification strategy will tend to vary by location, HVAC system, and house configuration and construction. Yet while actual results will vary by application, the enhanced dehumidification strategy should help to improve dehumidification by a central cooling system in humid climates.

As of this writing, there is conversation with the manufacturer on incorporating the updated settings for this project as a standard option. Home Innovation would like to see a few changes in the future to further improve humidity control. For the ramping profile, the primary ramping period could be increased from 7.5 minutes to 9.5 minutes, after the initial 30-second period with 50% airflow, for a total of 10 minutes. We would also like to eliminate the 30-second OFF-delay at the end of the cooling cycle, to minimize water on the evaporator coil being evaporated back into the airstream.

A thermostat with an adjustable on-off band that could be adjusted to 0.5 or 1.0 degrees, although not required for this project, would have further improved results and may be a benefit in some cases. It is expected that overcooling by up to 1 degree would not present a comfort issue for most occupants.

Lower HVAC system airflow reduces fan power for the furnace or air handler. For this study, compared to a typical cooling airflow of 400 cfm/ton, air handling fan power decreased by 33%–75% depending on the cooling mode, and duct static pressure decreased by 23%–60%.

4.2 Prioritized Ventilation

A ventilator fan with a variable-speed EC motor would provide a more consistently accurate airflow over a range of changing static pressures. It would also be simpler and faster for a technician to dial in the desired airflow setting compared to setting up one with the use of a manometer and fan curve.

The prioritized ventilation strategy could be implemented using a motorized damper instead of the fan-powered ventilator to provide supply-type ventilation, although this arrangement commonly does not provide sufficient airflow. The air handler operating at 25% of cooling airflow would not pull in enough air, so the air handler would need to operate at or near full cooling airflow during off-cycle ventilation, an energy penalty and potential to aggravate comfort issues. Even then, this arrangement commonly does not provide sufficient airflow. Further, this may not be code compliant because the fan efficacy of the air handler for that amount of ventilation air may exceed the code limit.

The prioritized ventilation controls could be modified to provide the total amount of ventilation on an annual basis. This would still minimize comfort issues and capture some energy savings compared to ventilating continuously at the minimum required rate. Alternatively, the 2021 International Residential Code allows a 30% airflow credit for balanced ventilation, so if a bath exhaust fan was interlocked with the ventilator, or an ERV or HRV was substituted in, the system would then provide sufficient ventilation on an annual basis. Additionally, recent developments in the market around demand-based controls for ventilation, with sensors that monitor indoor levels of volatile organic compounds, particulates, humidity, CO, etc., could soon impact ventilation control strategies.

Prioritized ventilation is expected to reduce comfort issues so that the system is less likely to be turned off. If meeting code-prescribed levels of ventilation at the test houses is a priority, then the existing control can simply be switched from “comfort” mode (with the update control logic) to “code” mode.

4.3 Latent Effectiveness

One of the project goals evolved from “develop a secondary equipment rating based on latent load efficiency” to “identify the metrics to quantify the performance and energy savings of the enhanced dehumidification and prioritized ventilation strategies.” A latent efficiency metric to complement or enhance a SEER rating could be a useful design tool. However, DOE, manufacturers, and builders were sensitive to developing or introducing an additional regulatory rating.

This study identified SHR as a useful metric to evaluate the performance of a system. SHR varies considerably by equipment selection and operating conditions, so this study did not identify a specific SHR target. but this study did show the importance of a lower SHR on indoor humidity and a practical approach to achieve lower SHR in various operating modes that did not jeopardize the operation of the cooling system.

4.4 Incremental Costs

The estimated incremental cost of equipment is provided in Table 13. The estimated costs are based on an online search of national equipment supply houses. The estimated costs will vary by builder and region, and no builder or subcontractor markup was applied. It was assumed that supply type ventilation was already being installed, so there was no additional ducting required.

Table 13. Estimated Incremental Cost

Component	Cost
80% AFUE gas furnace with variable-speed ECM air drive	$1,595
80% AFUE gas furnace with multispeed ECM air drive	($1,025)
Incremental furnace cost	$570
Thermostatic expansion valve (TXV)	$80
Programmable thermostat, with integral humidistat	$210
Programmable thermostat, standard	($71)
Incremental thermostat cost	$139
Ventilation system with fan-powered ventilator and control	$472
Ventilation system with motorized damper and control	($230)
Incremental ventilation cost	$242
Total estimated incremental cost of equipment	**$1,031**

4.5 Design and Installation Guidance

This study presented a dehumidification strategy that included some control settings that were modified by the equipment manufacturer just for this study and which could be adopted by manufacturers in the future. As a companion to this report, the Building America Solutions Center[6] includes general design and installation guidance from this project with the intent to show builders and HVAC professionals what they can do now to improve humidity control and comfort by increasing the latent capacity of currently and commonly available HVAC systems.

[6] Available at https://basc.pnnl.gov/

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Appendix A. Sample Expanded Product Data

The expanded product data for all temperature bins for the HVAC cooling system at the Houston test house are provided here. Note that the manufacturer product data (shaded in blue) is available online and is not proprietary.

Expanded Product Data: Houston Test House HVAC System (manufacturer data is shaded in blue)											
Home Innovation Description		Goodman System Configurator, 6/24/2021					Home Innovation Calculations				
Airflow		Houston Expanded Cooling Data, 75 IDB, 63 IWB					Delta	Capacity, kBtuh		Change, base 400	
Mode (% of 350 cfm)	CFM/ton	ODB	CFM	kBtuh	S/T	kW	Temp	Sensible	Latent	Sensible	Latent
	450	95	1350	34.7	0.85	2.81	20.2	29.5	5.2		
Cooling, typical	400		1200	33.9	0.83	2.74	21.7	28.1	5.8		
Cooling, this project (100%)	350		1100	33.5	0.80	2.70	22.6	26.8	6.7	**-4.8%**	**16.3%**
Ramping; DH Cooling (~85%)	300		900	32.5	0.76	2.61	25.4	24.7	7.8	**-12.1%**	**35.5%**
DH Ramping (~70%)	250		750	31.8	0.73	2.55	28.7	23.2	8.6	**-17.4%**	**49.1%**
Reference point (~60%)	200		600	31.1	0.70	2.48	33.6	21.8	9.3	**-22.6%**	**61.9%**
	450	90	1350	35.5	0.84	2.68	20.5	29.8	5.7		
	400		1200	34.8	0.82	2.61	22.0	28.5	6.3		
	350		1100	34.4	0.79	2.57	22.8	27.1	7.2	-4.8%	15.3%
	300		900	33.4	0.75	2.49	25.8	25.1	8.4	-12.0%	33.6%
	250		750	32.7	0.72	2.42	29.1	23.6	9.2	-17.3%	46.6%
	450	85	1350	36.3	0.83	2.54	20.7	30.1	6.2		
	400		1200	35.6	0.81	2.48	22.3	28.8	6.8		
	350		1100	35.2	0.78	2.44	23.1	27.5	7.7	-4.8%	14.5%
	300		900	34.3	0.74	2.36	26.1	25.4	8.9	-11.9%	31.9%
	250		750	33.7	0.71	2.30	29.5	23.9	9.8	-17.1%	44.3%
	450	80	1350	36.8	0.82	2.42	20.5	30.0	6.8		
	400		1200	36.1	0.80	2.36	22.1	28.7	7.4		
	350		1100	35.7	0.77	2.32	23.1	27.5	8.2	-4.2%	11.0%
	300		900	34.8	0.73	2.23	26.3	25.5	9.2	-11.0%	25.1%
	250		750	34.1	0.71	2.17	29.8	24.1	10.0	-15.9%	35.2%
	450	75	1350	37.2	0.80	2.30	20.4	29.8	7.4		
	400		1200	36.5	0.78	2.23	22.0	28.5	8.0		
	350		1100	36.1	0.76	2.19	23.1	27.4	8.7	-3.6%	7.9%
	300		900	35.2	0.73	2.10	26.4	25.6	9.6	-9.9%	19.3%
`	250		750	34.6	0.70	2.04	30.0	24.3	10.2	-14.5%	27.4%
	450	70	1350	37.4	0.80	2.19	20.5	29.9	7.5		
	400		1200	36.7	0.78	2.13	22.1	28.6	8.1		
	350		1100	36.3	0.76	2.08	23.0	27.4	8.9	-4.3%	10.1%
	300		900	35.4	0.72	1.99	26.2	25.4	9.9	-11.0%	23.3%
	250		750	34.7	0.69	1.93	29.7	24.0	10.7	-16.0%	32.6%
	450	65	1350	37.5	0.80	2.08	20.6	30.0	7.5		
	400		1200	36.8	0.78	2.02	22.1	28.7	8.1		
	350		1100	36.4	0.75	1.97	23.0	27.3	9.1	-4.9%	12.4%
	300		900	35.5	0.71	1.88	25.9	25.2	10.3	-12.1%	27.2%
	250		750	34.9	0.68	1.82	29.3	23.7	11.2	-17.4%	37.8%

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