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An Approach to High-Performance Affordable Housing Using Point-Source Space Conditioning

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An Approach to High-Performance Affordable Housing Using Point-Source Space Conditioning

Prepared for:
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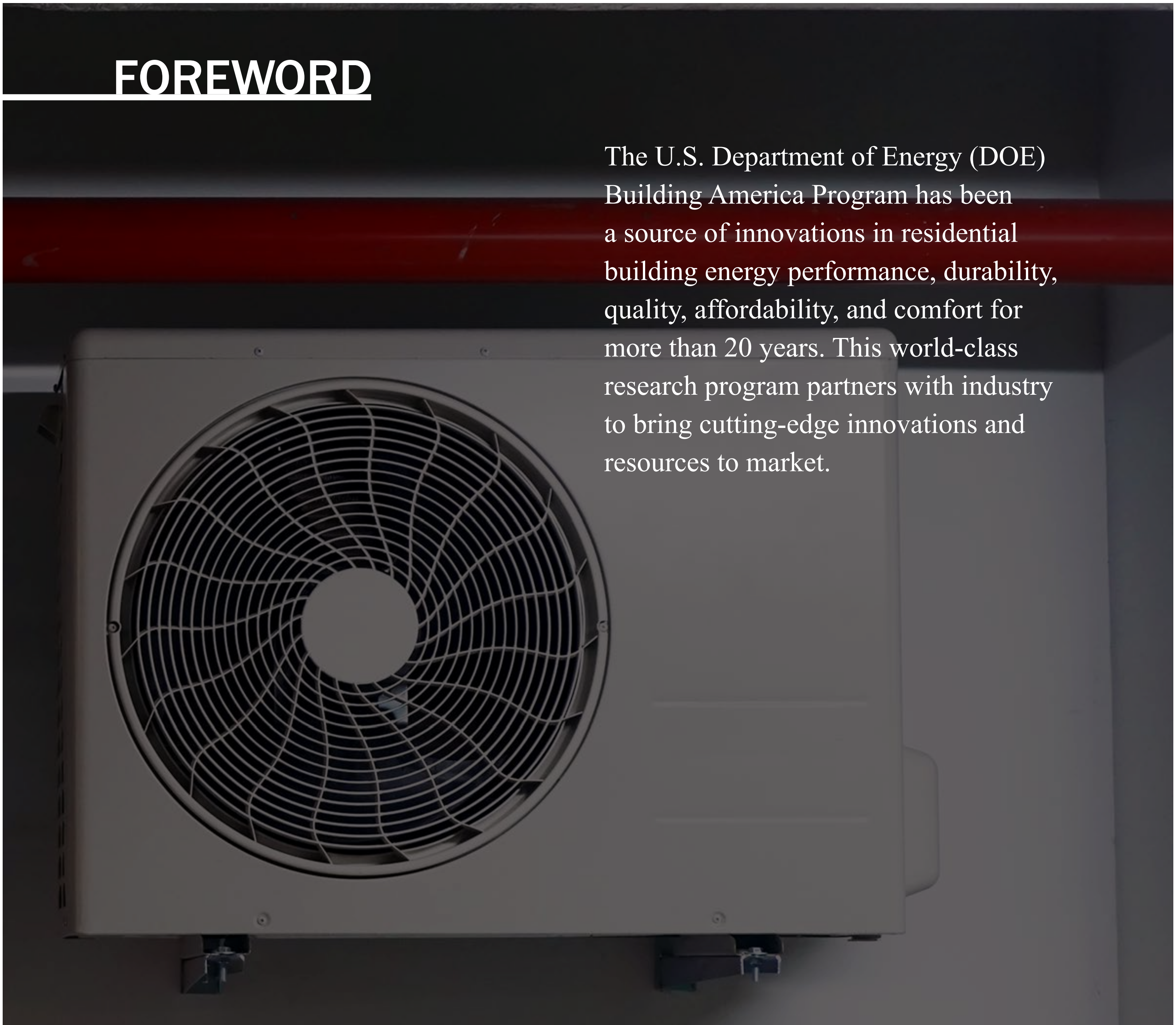
FOREWORD



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

In cooperation with the Building America Program, the Levy Partnership 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, *An Approach to High-Performance Affordable Housing Using Point-Source Space Conditioning*, explores the use of minisplit ductless heat pumps in combination with small through-wall transfer fans to provide space conditioning in small single-story homes, including manufactured homes and homes built by Habitat for Humanity.

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.

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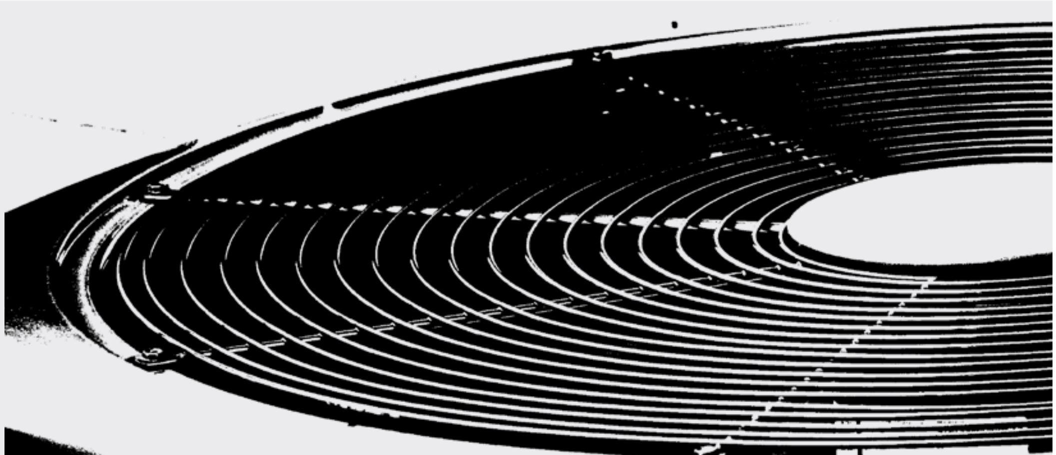


LIST OF ACRONYMS

ACCA	Air Conditioning Contractors of America
ACH_{50}	air changes per hour at 50 pascals
BEopt™	Building Energy Optimization Tool
Btu	British thermal unit
CBE	Center for the Built Environment
cfm	cubic feet per minute
CONTAM	Multi-zone Airflow and Contaminant Transport Analysis software
DOE	U.S. Department of Energy
ERV	energy recovery ventilator
HSPF	heating seasonal performance factor
HUD	U.S. Department of Housing and Urban Development
HVAC	heating, ventilating, and air conditioning
IECC	International Energy Conservation Code
MHCSS	Manufactured Home Construction and Safety Standards
MMBtu	Million British thermal units
MRT	mean radiant temperature
NREL	National Renewable Energy Laboratory
OSB	oriented strand board
PMV	predicted mean vote
R	thermal resistance
RH	relative humidity
RSME	root mean squared error
SEER	seasonal energy efficiency ratio
SHGC	solar heat gain coefficient
TMY	typical meteorological year
TRNSYS	Transient Systems Simulation software
XPS	extruded polystyrene
ZERH	Zero Energy Ready Home

EXECUTIVE SUMMARY

This project explored the use of minisplit ductless heat pumps in combination with small through-wall transfer fans to provide space conditioning in small single-story homes, including manufactured homes and homes built by Habitat for Humanity. The objectives were to reduce space-conditioning energy consumption by 50% compared to the 2009 International Energy Conservation Code (ICC 2009) while maintaining home affordability and thermal comfort. The strategy to achieve this goal also included a highly efficient thermal envelope to reduce space-conditioning loads.

The research process began with the development of a simulation tool using Transient Systems Simulation (TRNSYS) and Multi-zone Airflow and Contaminant Transport Analysis (CONTAM) software calibrated to previously collected lab home data. The tool was used to compare alternative designs and predict performance as a function of variations in building configurations and climates. Testing was then carried out at a lab home in Alabama to evaluate alternative air distribution designs. The information gained from these steps was used to design and build a manufactured test home, which was installed in New Jersey. The test home allowed testing under simulated occupancy conditions for one year, and then the home was monitored for an additional year when occupied. Following this, two additional test homes were built with Habitat for Humanity affiliates (in Massachusetts and Maryland) and monitored when occupied for one year.

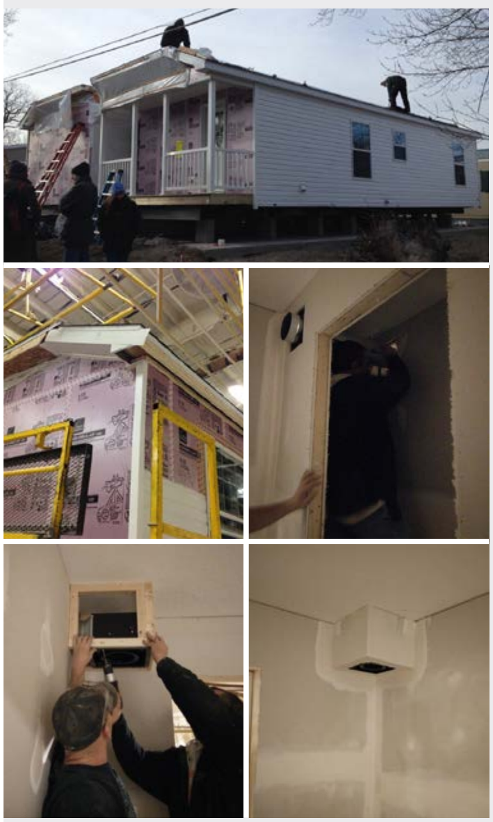
Monitoring at all three homes included space-conditioning energy consumption and thermal comfort metrics. Results were compared to two sets of comfort criteria: the Air Conditioning Contractors of America's (ACCA) Manual RS (Residential Systems) and ASHRAE Standard 55. Energy models were used to assess energy savings compared to baseline. Building cavity moisture was also monitored in the New Jersey home.

The New Jersey test house met some, but not all, defined objectives. Space-conditioning source-energy savings was modeled at greater than 50%. Comfort compliance in heating mode was achieved with the limited use of supplemental electric resistance heating, as planned in the design. Cooling season temperature data showed potential for compliance with some set point adjustment, but did not meet relative humidity criteria, although humidity was not reported as problematic by occupants. Construction costs were estimated to be \$4,600–\$4,800 more at the factory level and \$3,025–\$3,175 less for site work. Residents were satisfied with the home comfort and energy bills. No moisture concerns were observed in the New Jersey home.

The Massachusetts home was less successful than the New Jersey home because of a more challenging layout, small but significant thermal

defects in the envelope, missing fan controls, and occupant preferences that were at odds with some basic strategies of the space-conditioning system. Space-conditioning source energy savings were 27%–45% compared to a gas-heated home and 66%–74% compared to electric resistance, depending on whether it is compared to a best case or a supplemental heating case model.

The Maryland home had a similar layout as the Massachusetts home; however, two indoor heat pump air handlers were installed instead of one: in the living room and a bedroom. It had similar energy savings as the New Jersey home, exceeding project goals when gas and electric baselines are averaged. Residents were satisfied with comfort, although their use of the heat pumps and fans (and lack of use) impacted the comparison to the comfort metrics.



The three test homes were built to comply with the DOE Zero Energy Ready Home program; these specifications were generally adequate for the New Jersey and Maryland homes. The Massachusetts home's insulation levels were satisfactory; however, thermal bridging at the slab edge was reported to be a comfort issue despite compliance with ENERGY STAR® slab edge insulation specification. Humidity levels were difficult to control in all houses. The heat pumps did not dehumidify the homes to meet comfort criteria. Better humidity control equipment is needed for low-load homes such as these. Heat recovery ventilation was investigated as an option for all homes; however, this is a significant additional first cost and would have increased overall annualized energy-related expenses according to BEopt™ modeling. A low-cost energy recovery ventilator (ERV) was integrated into the Massachusetts home.

Lessons gleaned from the test homes include: (1) Supplemental heaters can be controlled to minimize their use by offsetting set points from the heat pump set point. (2) Transfer fans and ventilators should be occupant-controllable with clearly labeled on-off switches. (3) Short circuiting of the transfer fan airflow should be avoided; transfer fan capacity should be in the 90–150 cfm range to move the needed amount of energy into or out of a room, and adjustable fans can permit tweaking the flow rate by season or based on occupant preference to improve comfort.

In terms of home configuration, features such as large windows, sliding glass doors, and other weak links in the thermal envelope were avoided in the test homes because they would have had a negative impact on this design approach. Compact home layouts that organize remote spaces around and directly adjacent to the main living space are better than elongated or L-shaped plans or plans with the main living space at one end of the home.

No code issues were encountered on the Habitat homes. The only code issue encountered on the manufactured home was the U.S. Department of Housing and Urban Development (HUD) Code provision that the fresh air ventilation system deliver air to all bedroom and main living areas, which could force the exhaust fan to be in a hallway rather than a bathroom.

Guideline documents were prepared for the manufactured home industry and for Habitat (and builders of similar homes) to help them apply the lessons from this work if they wish to adopt this approach for high-performance buildings.

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1 Introduction

1.1 Objectives

This research addressed the need to make major reductions in space-conditioning energy use (≥50%) while maintaining home affordability. It focused on manufactured homes placed over unconditioned crawlspaces as well as small, one-story, generally affordable and modest sized single-family detached homes with slab foundations, such as those commonly built by affordable housing organizations like Habitat for Humanity. Both of these industries/organizations serve affordable housing markets and place a high priority on keeping costs as low as possible.

Habitat for Humanity homes are site-built by local Habitat affiliates. Volunteer and self-help labor are a cornerstone of the building process, although professional contractors are a part of the total labor force and a major contributor to cost. For Habitat, the approach depicted herein offers the following advantages:

- **Affordability.** Drastically reducing energy costs with a modest increase in home cost can help financially struggling families make ends meet.
- **Standardization.** Moving to a standard method of construction for envelope, HVAC system, and ventilation design enables aggregate buying across affiliates, capturing volume purchasing power. Standardization enables affiliates to share experiences with a common technology, leading to rapid and continuous improvements in design and accelerating market uptake. It also provides an outlet for Habitat's internal training program.
- **Lower labor costs.** Shifting more of the value-add tasks to volunteer and self-help labor reduces the most expensive labor component—professional contractors. With no duct work to install and the ability of volunteer labor to install many of the constituent parts, overall costs will go down, adding to the affordability case.

Factory builders are quite uniform in their building practices; the Manufactured Home Construction and Safety Standards (MHCSS) (U.S. Code of Federal Regulations N.D.) have a single performance target (Uo value) for each of three climate regions. Virtually all manufactured homes today have plant-installed furnaces (electric or gas) and site-installed cooling systems with ducted distribution. System efficiencies are typically code minimum. The advantages for factory builders to adopt this approach include:

- **Affordability.** Drastically reducing energy costs with a modest increase in home cost can help financially struggling families make ends meet.
- **Address systemic flaws.** Ducts, including external crossovers in multisection homes, tend to be leaky. Because the crossovers are completed at the site, they are outside the manufacturers' control. Point-source space conditioning—where heated or cooled air is delivered to the space at a few points directly where it is generated, rather than being ducted

through a distribution system—would eliminate this distribution method and the associated efficiency losses.

- **Improved equipment efficiency.** Other than the furnace, the interior and exterior HVAC components (AC and heat pump systems) are site-installed (i.e., beyond the control of the manufacturer) and of minimum efficiency. Ductless heat pumps are more efficient and reduce the chances for some of the deficiencies that can result from third-party equipment installed outside of the factory control. In addition, traditional off-the-shelf cooling equipment capacity is limited to two tons or greater—a poor match for small homes with highly efficient thermal envelopes, leading to oversizing. These limitations and deficiencies would be eliminated with a complete factory-installed HVAC system. Installation of all equipment would be moved into the plant, assuring proper sizing, quality installation (including consistent charging), and standard use of heat pumps rather than electric resistance furnaces.

- **Standard footprint.** Factory-built structures have a limited range of sizes and footprints and are overwhelmingly one story. Therefore, a few design variations will serve homes across a broad range of markets. Such standardization and the size of the industry generally will enable manufacturers to purchase components in large volume, driving down system costs.

1.2 Strategy

This project investigated an approach combining a highly efficient thermal envelope, a very low-capacity and efficient mechanical system, an innovative distribution system, and affordable heat recovery ventilation, focusing on modestly sized single-story affordable homes.

The goal was to develop a performance- and cost-optimized package of measures that provides a complete comfort solution while striking a balance between high system efficiency and home cost. The strategy was to improve the thermal envelope to a degree that allows significant downsizing of the space-conditioning equipment and reduce investment in a distribution system. The system must be able to cope with a range of thermal and moisture loads and be replicable at production scale. The resulting approach has the potential to lower barriers to building homes that use less than half the energy of a comparable home built to the 2009 International Energy Conservation Code (ICC 2009) and to do so at a modest increase in cost.

The design approach included the following components:

- **Very high-thermal-integrity building envelope.** The thermal envelope is designed to achieve low loads through the use of insulation and high-performance glazing, minimizing air leakage, enabling the use of low-capacity equipment. Construction features include dense-packed ceiling insulation, high-density batt with foam-sheathed walls, high-performance windows, and full thickness high-density floor insulation.

- **A one- or two-point (head), ductless, minisplit heat pump** placed in a central location in the home. The ability to use only one or two heads, made possible by the thermal envelope and distribution design, is important to minimize system cost. Ductless heat pumps offer advantages such as ease of installation, ability to quickly and accurately charge the system, high heating and cooling efficiencies, operating temperatures below 0°F, and quiet operation.
- **A through-wall transfer fan distribution system** that moves conditioned air between the living spaces, obviating the need for ducts. Each fan serves one living space (e.g., bedroom) and is controlled independently.
- **A whole-house ventilation system** that meets the ASHRAE 62.2 ventilation standard.

1.3 Background and Literature Review

The Levy team started working on this space-conditioning problem for factory-built homes with an earlier Building America lab home project (Levy et al. 2016). The earlier lab home was built and instrumented in Alabama in April 2014. That work was instrumental in identifying gaps that were addressed by this work, and that lab home was reused for the initial tasks conducted for this project.

The literature on point-source space conditioning in low-load homes in heating-dominated climates establishes that such designs can be successful with careful attention to envelope efficiency, home layout, and system configuration. Adapting it to production-level affordable housing is a primary goal of this work. The following is a synopsis of key points extracted from related studies (see the Bibliography for a literature list):

- Several studies indicate that single-point space conditioning in combination with a highly insulated airtight home can satisfy comfort needs in a range of home sizes and configurations.
- Room temperature variations are largely driven by occupant behavior (in particular, interior door opening/closing and consistency of heating unit operation).
- Simplified space-conditioning systems can provide significant energy cost savings, meeting or exceeding the 50% goal.
- Thermostat setbacks increase room-to-room temperature variances. Setbacks save little energy in homes with ductless heat pumps and can result in long recovery times unless heating/cooling equipment is substantially oversized, undercutting efficiency.
- Transfer fans can reduce room-to-room temperature variations, but only to a limited extent.
- Bedrooms may need electric resistance heat to maintain comfort in very cold conditions, but controls can ensure resistance heat is used only when needed.

- Maintaining a set point is challenging in multistory homes because thermal buoyancy leads to temperature differences from floor to floor.
- Importantly, traditional fully ducted distribution systems are often subject to comfort performance failings, especially in low-load houses where conventional equipment is often oversized, resulting in frequent cycling.

2 Experimental Method

2.1 Technical Approach

The project consisted of the following steps:

1. Develop a simulation tool using Transient Systems Simulation (TRNSYS) and Multi-zone Airflow and Contaminant Transport Analysis (CONTAM) software and calibrate it to the Alabama lab home using previously collected data. The tool allowed the team to compare alternative designs and predict performance as a function of variations in building configurations and climates.
2. Use the simulation tool to design and evaluate alternative air distribution designs.
3. Test the new distribution system designs in the field by retrofitting them into the Alabama lab home and collecting additional data.
4. Use the information gained from the above steps to design and build a manufactured test home and install it in New Jersey. Test the New Jersey home under simulated occupancy conditions for one year. Develop a TRNSYS model of the New Jersey home.[1] Continue to monitor the New Jersey home for an additional year when occupied.
5. Design and build two additional test homes with Habitat for Humanity affiliates (in Massachusetts and Maryland). Monitor the occupied homes for one year (see Table 1 for list of test homes).
6. Write guidelines for Habitat and the manufactured housing industry to use in employing this building approach.

Table 1. Summary of Test Homes

Site Name	Test Periods
Alabama lab home	Simulated occupancy
New Jersey home	Simulated occupancy and occupied
Massachusetts home	Occupied
Maryland home	Occupied

Each of these steps and their results are described in the sections that follow. They are geared toward answering the following set of research questions.

1. What level of envelope efficiency and types of features are required to allow for homes to operate with acceptable comfort and energy outcomes?

[1] The calibrated TRNSYS model was going to be compared to a TRNSYS baseline to develop energy savings predictions, but this task was abandoned due to budgetary constraints. Instead, the energy savings predictions were developed based on BEopt™ models.

2. Do envelope provisions increase moisture/condensation risks, particularly at foam sheathing and in attics?
3. Can humidity levels throughout the home be adequately controlled?
4. Are there acoustic and odor issues to be addressed with the transfer fan approach?
5. Can heat recovery ventilation be affordably integrated into homes to optimize ventilation performance and indoor air quality?
6. What type of control system would be needed for system integration, how would the controls be connected to the main unit, and what are recommended thermostat locations?
7. What are the overall energy savings compared to baseline specifications?
8. What is the strategy for locating the heating/cooling source and transfer fans to achieve desired airflow and temperature distribution? How should return air pathways to the indoor unit be designed? What is the required transfer fan capacity to meet the needs of the spaces served without causing drafts?
9. What airflow or throw pattern is acceptable to the occupant?
10. What home design features exacerbate temperature differentials? What home layout strategies are best suited for this approach?
11. What design and production approaches can minimize first costs for each respective product type (site and factory built)?
12. What additions/changes are required to building codes (IECC and MHCSS) to address this design approach?

3 Simulation Tool Development

TRNSYS and CONTAM software programs were used to develop a model of the Alabama lab home and the New Jersey test home. The Alabama model was calibrated based on data collected from the previous Building America project. The focus was on simulating operation of distribution and ventilation systems. It was intended that the New Jersey model would be used for all comparisons to measurements when appropriate, as well as for prediction of thermal and airflow performance for other building configurations and in other climates.

Computer models are mathematical representations of physical processes. All models must contain simplifying assumptions. The more assumptions, the easier it is for the end user to implement the model but the less flexibility the end user has. The fewer assumptions built into the model, the more flexibility there is for the end user but the more data they are asked to provide. All energy modeling software tools exist on a spectrum; on one end are tools that allow end users to get results with a minimum of information, and on the other end are highly flexible tools that require greater model implementation time and expertise. The ideal balance between flexibility and ease of implementation depends on the target audience and use that the tool will be put to. In order to be useful to the commercial sector, a tool must be relatively quick to implement; therefore, many assumptions are built in. Tools that are designed for the research sector must have more flexibility. TRNSYS is designed to have more flexibility than EnergyPlus® (the software engine for BEopt™), for example. It allows modification of assumptions in order to calibrate modeled and measured data for the purposes of thermal and comfort evaluations.

Additionally, many building simulation tools are not capable of simulating room-to-room air movement from natural convection. For buildings with ducted space-conditioning systems, this is less important to ensure room-to-room thermal comfort in a design. For homes with point-source space conditioning, air movement is potentially a significant factor affecting the room-to-room thermal comfort. TRNSYS's capability to model multiple zones in detail, in combination with the algorithms found in CONTAM—a software tool for predicting interzonal airflow (Klein 2013; NIST N.D.)—were a good fit for the project needs. CONTAM can be run from within TRNSYS, making this software combination ideal for the research. With TRNSYS, it is relatively easy to adjust algorithms and model assumptions to calibrate models with respect to measurements. The following process was used to calibrate the TRNSYS model.

- A baseline computer simulation of the Alabama lab home was prepared using TRNSYS with a CONTAM module. TRNSYS modeled energy flows, and CONTAM added the capability of modeling airflows within the home due to convection, infiltration, and mechanical systems. CONTAM considers room temperature stratification, which was observed to be on the order of 3.5°–5.5°F in the lab house. Because the calibrated model was used to investigate alternative transfer fan design scenarios, and given the small temperature difference between living and bedrooms (1.8°–3.5°F), accurately predicting the transfer fan inlet temperature was important.

- The baseline model was run using actual driving inputs (weather data) from a variety of summer and winter periods.
- Simulation results were compared to field-collected data at the whole-house and subcomponent levels (zone temperatures, solar gain, etc.). Various data collection periods were used to compare the model to periods with interior doors open vs. closed, with transfer fans running and not running, and with various heating configurations.
- The model was adjusted to correct for differences between modeled and field-recorded measurements. Appendix A provides more details on the calibration procedure.

4 Design and Evaluation of Alternative Distribution and Ventilation Systems

As part of the design development process for the New Jersey test home, modeling and field testing were used to evaluate space-conditioning distribution and ventilation system designs. A primary goal was to provide temperature control throughout the home given the challenge of a single point source for space conditioning. One strategy was selected based on the simulation results: a ductless minisplit heat pump with high-capacity and efficient transfer fans in a high-wall position along with improved thermal enclosure and backup electric resistance heat seemed to be a logical combination of strategies for improved performance of the new design.

This revised design was first examined via modeling, and then the mechanical equipment aspects were tested in the Alabama lab home with short-term data collection to gauge performance. The testing consisted of room-by-room temperature logging under a variety of conditions (doors open and closed) to compare to modeled predictions.

4.1 Distribution System Design

The original space-conditioning and distribution system in the Alabama lab home consisted of a ductless heat pump in the living room and four transfer fans to move air from the living room to the bedrooms and the master bathroom. The transfer fans were rated at 75 cfm in the small bedrooms and 125 cfm in the master bedroom and master bathroom. The fan inlet was located near floor level in the living room and at the outlet at the ceiling level of the remote rooms. The rationale for this original configuration was that in a cooling-dominated climate, air pulled from near floor level would be cooler, benefitting the bedrooms.

To improve air distribution effectiveness, the modifications described in Table 2 were tested in the TRNSYS simulation. The retrofit design utilized Panasonic FV-11-15VK1 fans installed high on the living room walls and delivering air straight through the walls to a supply high on the wall of the bedrooms/master bathroom. Additionally, the whole-house exhaust fan was moved from the dining room to the master bathroom in order to draw more air from the main space into the master suite.

Table 2. Distribution Design Options

	Option	Heating*	Cooling*	Advantages	Disadvantages
1	Larger transfer fans	✓	✓	Transfer higher volumes of air and energy	Uncertain if cooling improvement; higher cost
2	More efficient transfer fans	✓	✓	Lower electric usage	Marginal impact in cooling; no improvement in heating
3	Reposition transfer fans so inlet and exhaust are high on wall	✓		Simpler, less expensive installation; higher airflow due to less duct loss	May be less suitable for cooling dominated climate
4	Move exhaust fan to master bath	✓	✓	No added cost; helps master bedroom and master bath in current floor plan	No impact on bedrooms 2 and 3; only adds 45 cfm (small impact).

*Anticipated to address home performance in the heating season, cooling season or both.

An assessment of the enhanced distribution system was simulated in the calibrated TRNSYS/CONTAM model with Typical Meteorological Year (TMY) weather data. Thermal comfort of individual spaces was analyzed in both heating and cooling seasons with simulated sensible and latent gains with interior doors closed.

The results indicated that the high-wall position of the larger and more efficient transfer fans provided superior thermal comfort during the heating season compared to the prior design. In the model with local Alabama weather data, the number of hours within the ACCA Manual RS (ACCA 2015) compliant thermal range in the bedrooms increased by an average of 30%. The number of ACCA compliant hours in cooling declined by nearly 15% in the bedrooms. When modeled with Albany, New York, climate data, thermal comfort improved by 25% during the heating season and by 6% during the cooling season in the bedrooms. Both bathrooms saw marginal improvements in thermal comfort during heating and cooling seasons with Albany data. These simulations did not include any improvements to the envelope and did not include any backup electric resistance heating. Therefore, the comfort improvement is solely attributable to the distribution system. Results of these simulations are shown in Figure 1 and Figure 2.

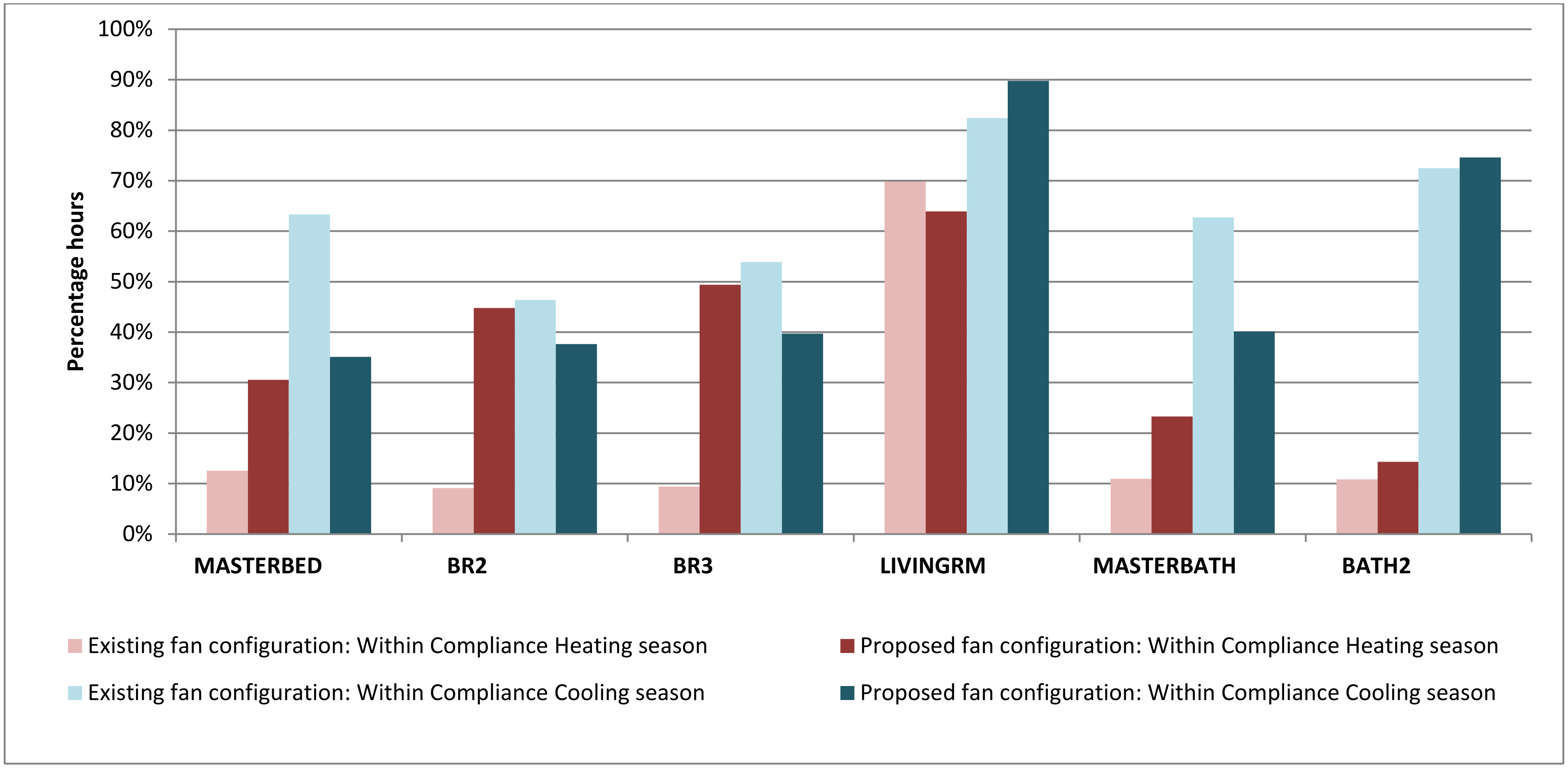


Figure 1. Existing vs. proposed fans: Russellville, Alabama

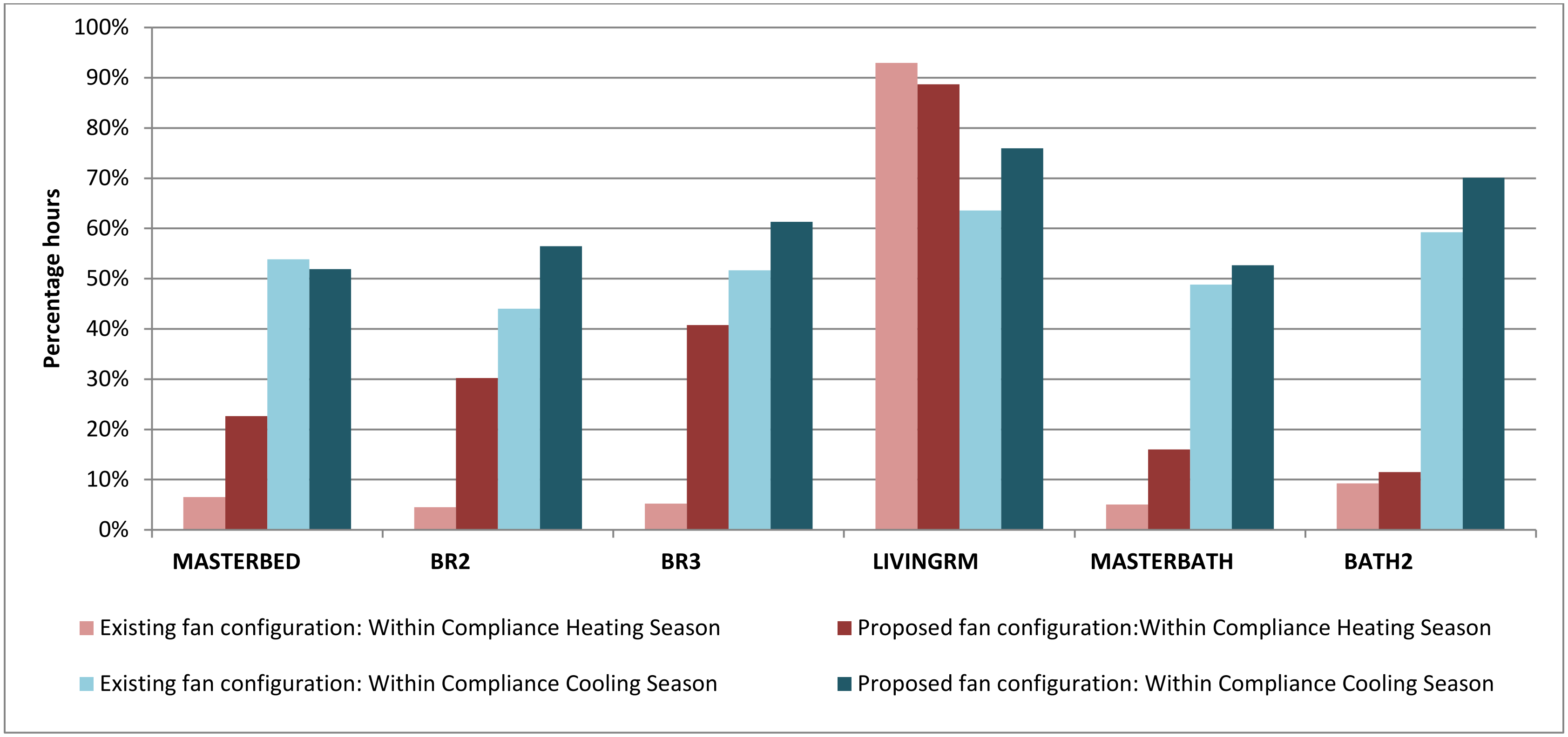


Figure 2. Existing vs. proposed fans: Albany, New York

4.2 Ventilation System Design

The original Alabama lab home utilized a high-efficiency ENERGY STAR® exhaust fan to provide an ASHRAE 62.2 2010 compliant ventilation rate using an exhaust-only strategy with fresh air being provided via infiltration. One of the objectives of the next generation design was to incorporate heat recovery ventilation. Six ventilation system options, including three options with heat recovery were evaluated in BEopt. BEopt was used for this analysis because of the relatively simple nature of the comparison. BEopt also is ideal to estimate the relative impact on annual source energy consumption and annualized energy related costs. This was done for four climate locations in the Northeast and Midwest. The annualized energy related cost accounts for the materials and labor cost as well as utility rates. Table 3 lists the systems, estimated costs and summarizes some advantages and disadvantages of each option.

Table 3. Ventilation System Design Options

	Option	Fan Flow and Power	Product Cost ($)	Labor Cost ($)	Advantages	Disadvantages
1	Exhaust only (Broan XB50-House C)	45 cfm; 4.41 W	150	20	Low cost, low fan power, reliable	Unbalanced
2	Ducted HRV (Broan HRV70SE)	45 cfm; 22.5 W	1,078	200	Up to 77 cfm	Requires ducting, high cost
3	Panasonic Whisper Comfort 40 CFM ERV	40 cfm; 23.2 W	636	50	Simple installation; minimal ductwork	40 cfm max
4	Panasonic Whisper Comfort 100 CFM ERV	45 cfm; 26.1 W	1,309	50	Simple installation; minimal ductwork; up to 100 cfm	High cost
5	Exhaust only (Air King ES80)	45 cfm; 13.8 W	95	20	Low cost, low fan power, reliable	Unbalanced
6	Non-heat recovery balanced	45 cfm; 31.5 W	330	40	Simpler than HRV; basically 2 fans; balanced	No heat recovery; costs more than exhaust with uncertain benefits

The simulated results show that the Panasonic Whisper Comfort ERV 40 CFM had the lowest source energy consumption in all four locations. However, its maximum ventilation rate is below the required continuous 45 cfm as per ASHRAE 62.2 2010 and below the 50 cfm minimum required by the MHCSS (U.S. Code of Federal Regulations N.D.), which contributes to lower energy consumption. Source energy consumption for all options except the non-heat recovery system are within 4–6 MMBtu/yr of each other, and the corresponding costs savings from energy recovery amounts to roughly $65–$80 per year. The lower first cost of the exhaust fans compared to the other options makes them more attractive to home manufactures. The exhaust fan strategy with the lowest whole-house annualized energy-related cost was selected for the next home design. Note that identical utility rates were used in the BEopt analyses in all four climate locations. Detailed results are in Appendix B.

4.3 Field Testing

The new transfer fan distribution system configuration was then tested to validate the modeled results. The measured room temperatures showed that the new configuration helped maintain temperatures close to the living room temperature with interior doors closed and fans on.

The panoramic photo in Figure 3 shows the living room (center) and dining room (right) with the four transfer fans. They are also shown on the floor plan in Figure 4. Because this was an unoccupied test home, the fans were installed exposed on the wall. Three fans delivered air directly into the rooms (Figure 5 through Figure 7). The fan to the master bedroom was connected to a short round duct that extended through a closet (Figure 8). Additionally, return air grilles were installed at the bottom of the doors to bedrooms 2 and 3. Bathrooms already had door undercuts. Transfer fan supply flows are shown in Table 4.

Figure 3. Panoramic view showing all four transfer fans

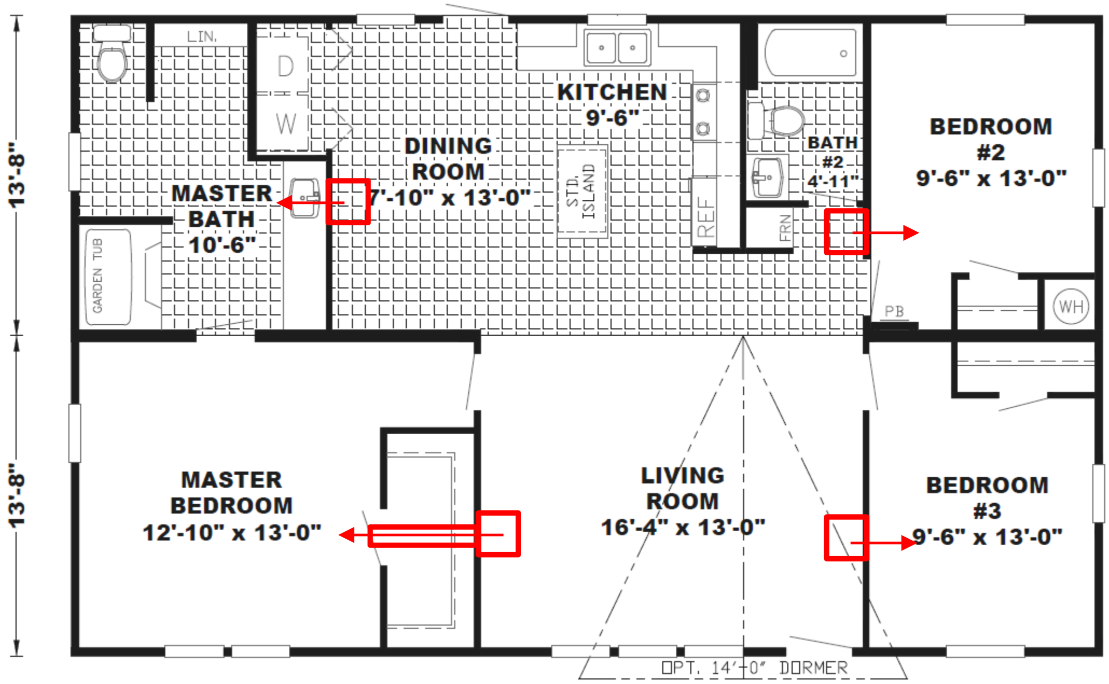


Figure 4. Floor plan showing transfer fan location

Figure 5. Transfer fan serving the master bathroom shown from the dining room side (air inlet)

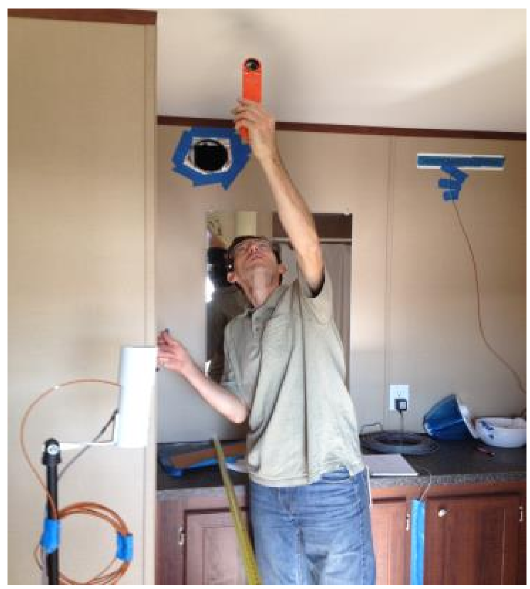


Figure 6. Measuring air speed at the master bathroom transfer fan (air outlet)

Figure 7. Master bedroom transfer fan at the living room side (air inlet)

Figure 8. Master bedroom transfer fan bedroom side showing duct through closet (air outlet)

Table 4. New Transfer Fans Supply Flows (cfm)

Room	Doors Open	Doors Closed
Master Bedroom	188	172
Master Bathroom	190	160
NW Bedroom (#2)	167	157
NE Bedroom (#3)	171	165

Notes: All fans on; exhaust fan off; closet doors closed; heat pump off; interior and exterior temperatures ~70°F; using TEC Flowblaster.

The house was monitored for one month with data collected for the four test conditions listed in Table 5. All configurations were tested for model calibration purposes (to estimate airflow between rooms with open doors) and to compare thermal comfort during best and the worst cases. Due to a power failure on-site, some hours were eliminated from the first test condition. Six hours were disregarded between test periods to allow the house to transition between conditions. The living/dining room was heated with electric resistance at a set point of 73.4°F. In order to understand the impact of transfer fans and door positions, none of the bedrooms were heated.

Table 5. Test Conditions

Test Conditions	Test Period
Doors open; fans on	November 14–22
Doors open; fans off	November 24–29
Doors closed; fans off	November 30–December 3
Doors closed; fans on	December 4–9

Indoor temperatures vs. outdoor air temperatures were plotted for each of the four rooms for these periods (Figure 9 to Figure 12). The temperatures in all four rooms were well below the set point when the doors were closed, fans off, and the outdoor air temperature was below 55°F. With doors open, the indoor temperature was 70°–75°F for most hours. The fans improved space-conditioning distribution when doors were closed; they had a similar effect as opening interior doors.

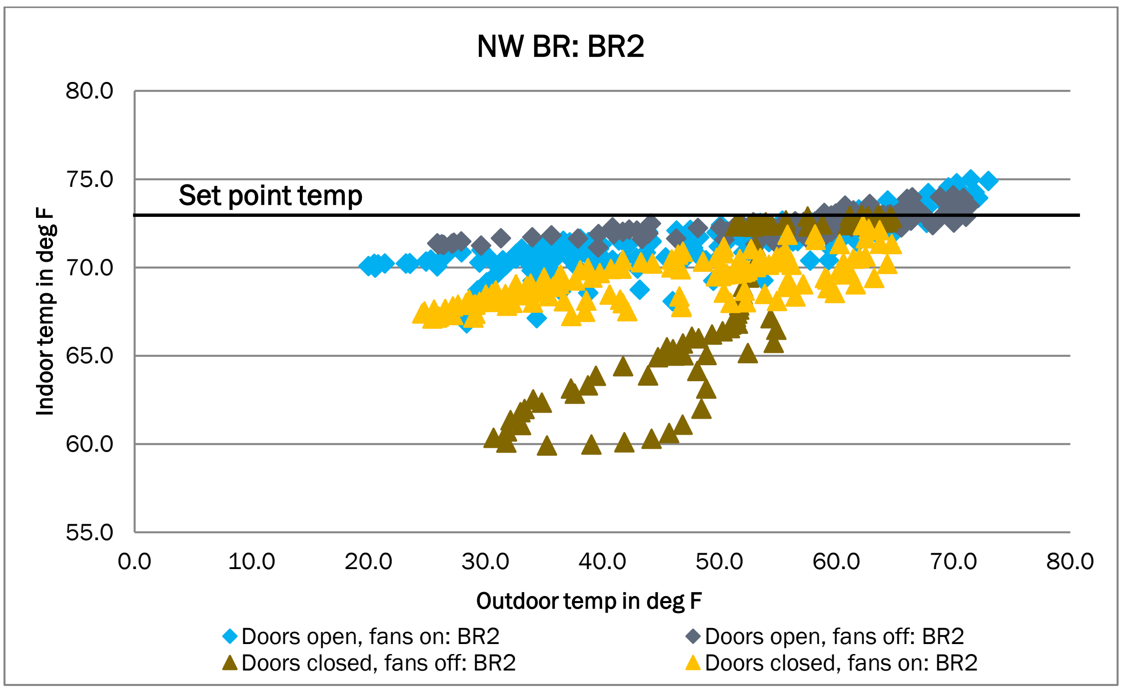


Figure 9. Indoor vs. outdoor temperature in bedroom (BR) #2

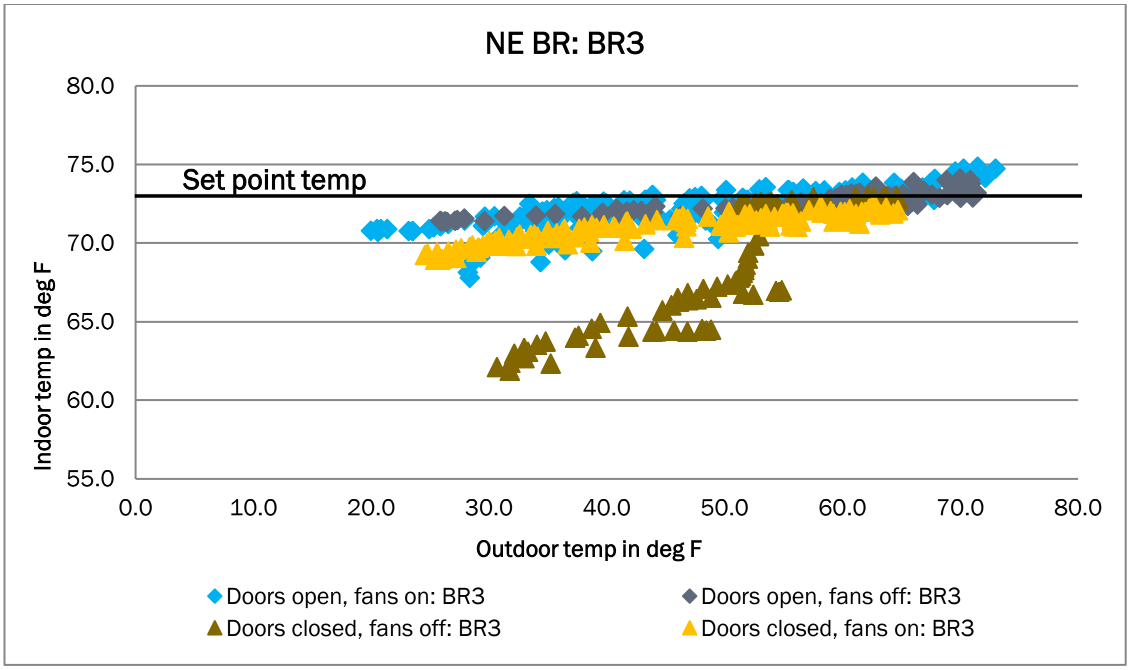


Figure 10. Indoor vs. outdoor temperature in bedroom #3

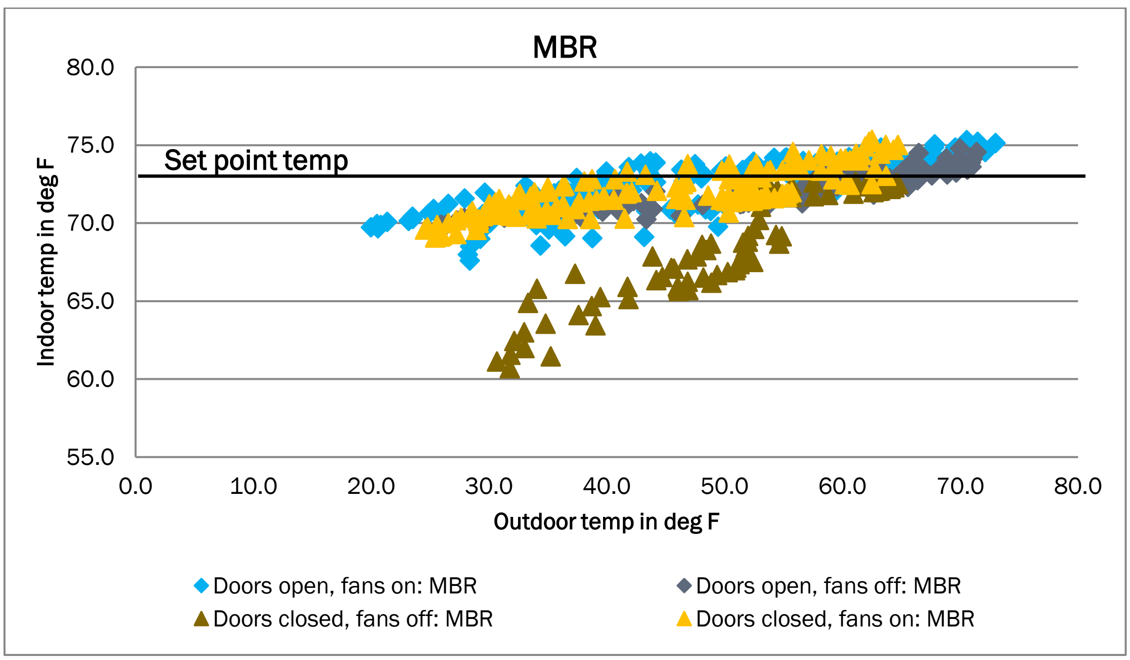


Figure 11. Indoor vs. outdoor temperature in master bedroom

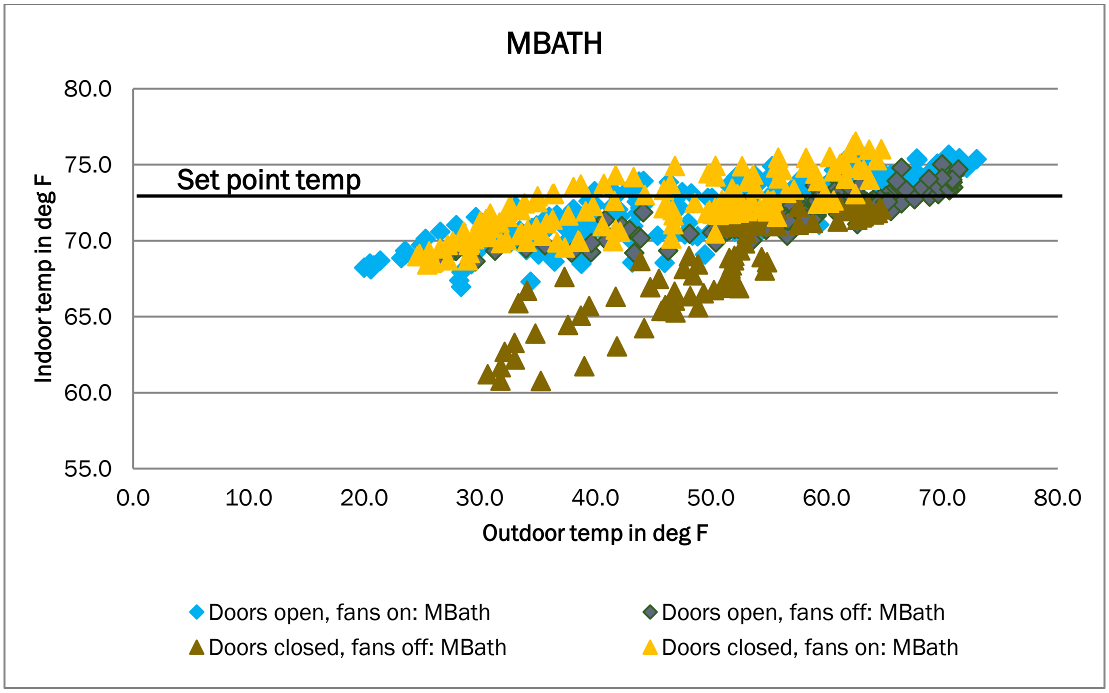


Figure 12. Indoor vs. outdoor temperature in master bath

4.4 Model Calibration of Interzonal Airflows

The commissioning test data and the 1-minute data collected during the monitoring period described previously were used to recalibrate the Alabama lab home TRNSYS/CONTAM model for transfer fan and door operation. The model was fine-tuned to estimate the mixing of air when the doors were open and closed, accounting for door sizes, undercuts, and return air grilles. The model also was revised to locate the continuous exhaust fan in the master bathroom. The blinds were left open at 50%, and solar reflective blind properties along with the estimated insulative properties of the blinds were adjusted as part of the calibration.

The calibrated model most closely matched the field data during the "doors closed and fans on" period. The worst-case condition would be with the doors closed and no transfer fans running, but this is not a proposed normal operating condition as it would almost certainly lead to poorly heated or poorly cooled peripheral rooms. Occupants cannot be expected to keep all of their doors open in order to be comfortable in their rooms, so the condition of having all doors closed with transfer fans running was considered the condition of greatest interest. The simulated data from the calibrated model were more in line with the measured data during periods of low temperature differences between the living room and the other rooms. At higher temperature differences, the simulated model was less accurate in predicting thermal comfort. This trend was predictable, as the periods of smaller temperature differences among rooms correspond to periods of small temperature differences between the set point and outdoor air temperature. In the extreme case of zero temperature difference between set point and outdoors, the model would be expected to predict that all rooms would have temperatures equal to the set point temperature if there were no solar gains.

In general, the model is predictive of the changes in the operation schedule; however, it is conservative in predicting thermal comfort. Comfort performance is better than predicted by the model, especially during periods of colder ambient temperatures. The following figures (Figure 13 through Figure 16) show the model-to-data correlation accuracy before and after calibration for temperature differences between rooms under various conditions.

The lessons learned from these exercises with the lab house led to a new configuration of fans, which proved to be more successful in maintaining temperatures close to the set point in heating, which would reduce the need for electric resistance backup heat. This configuration was then incorporated into the next set of test homes.

A detailed discussion of the TRNSYS building model calibration for both heating loads and interzonal airflow can be found in Appendix A.

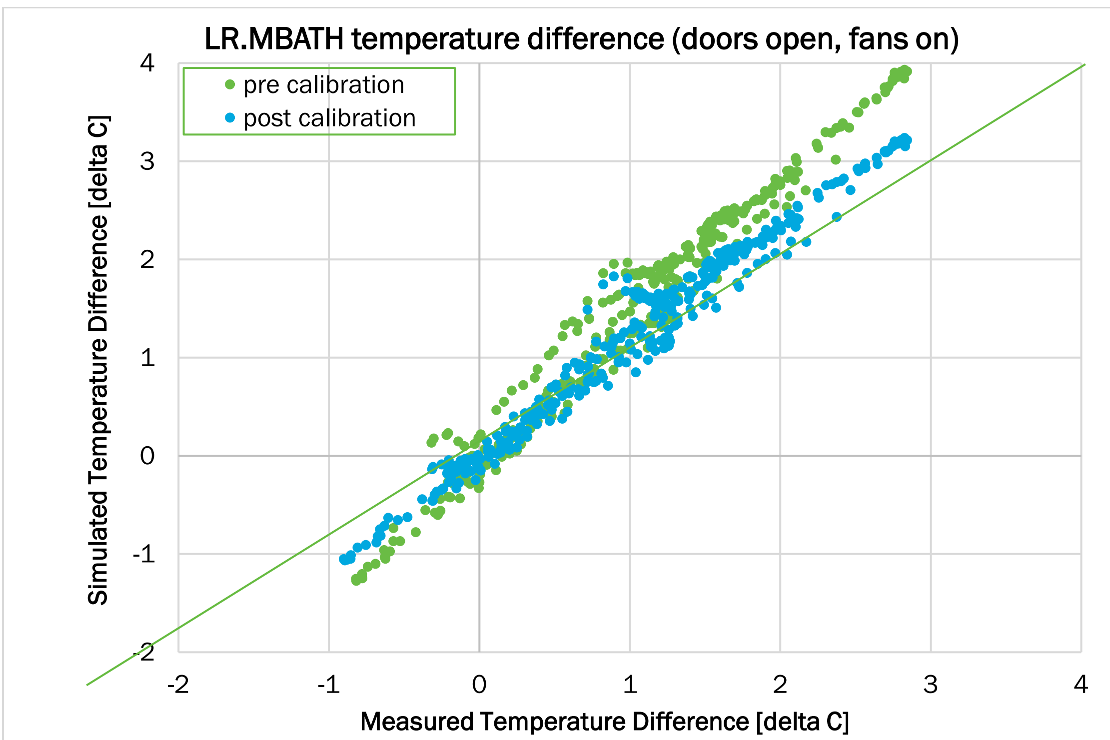


Figure 13. Calibrated model: living room to master bath temperature difference (doors open, fans on)

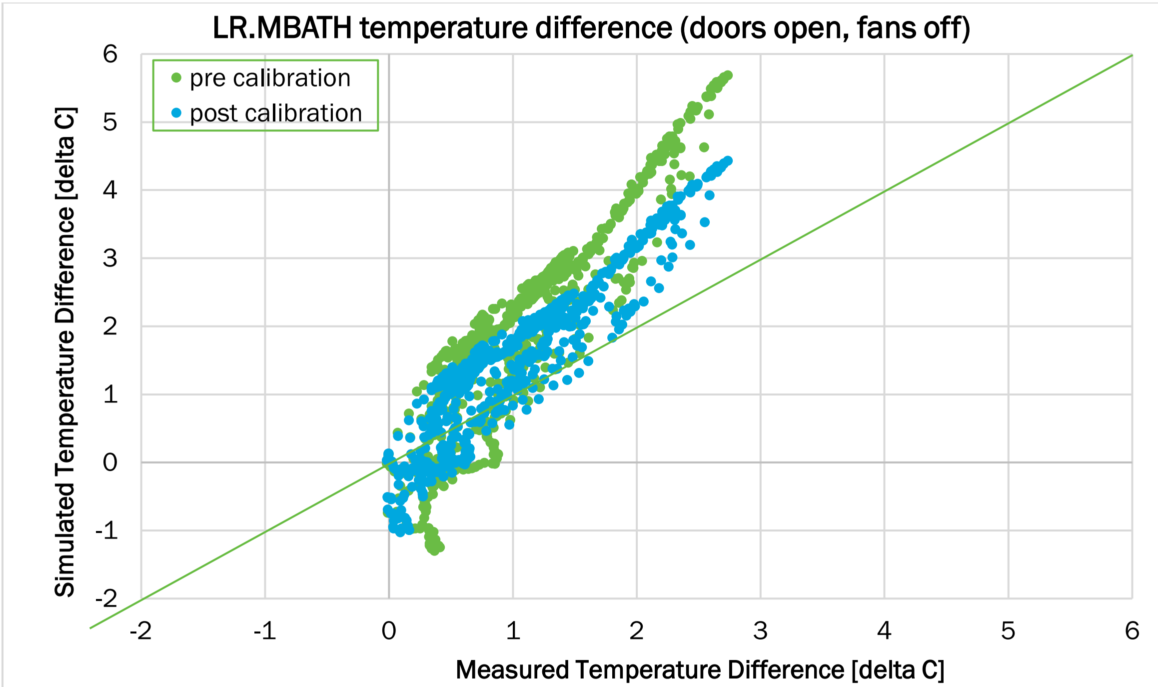


Figure 14. Calibrated model: living room to master bath temp difference (doors open, fans off)

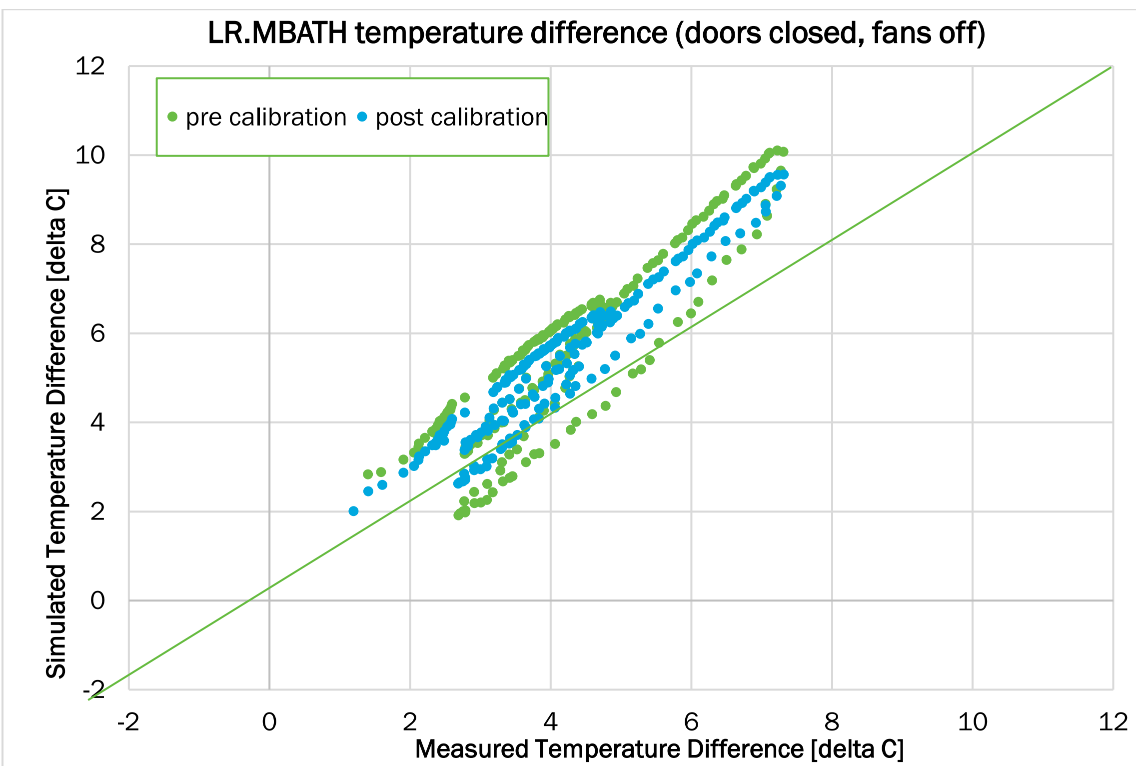


Figure 15. Calibrated model: living room to master bath temp difference (doors closed, fans off)

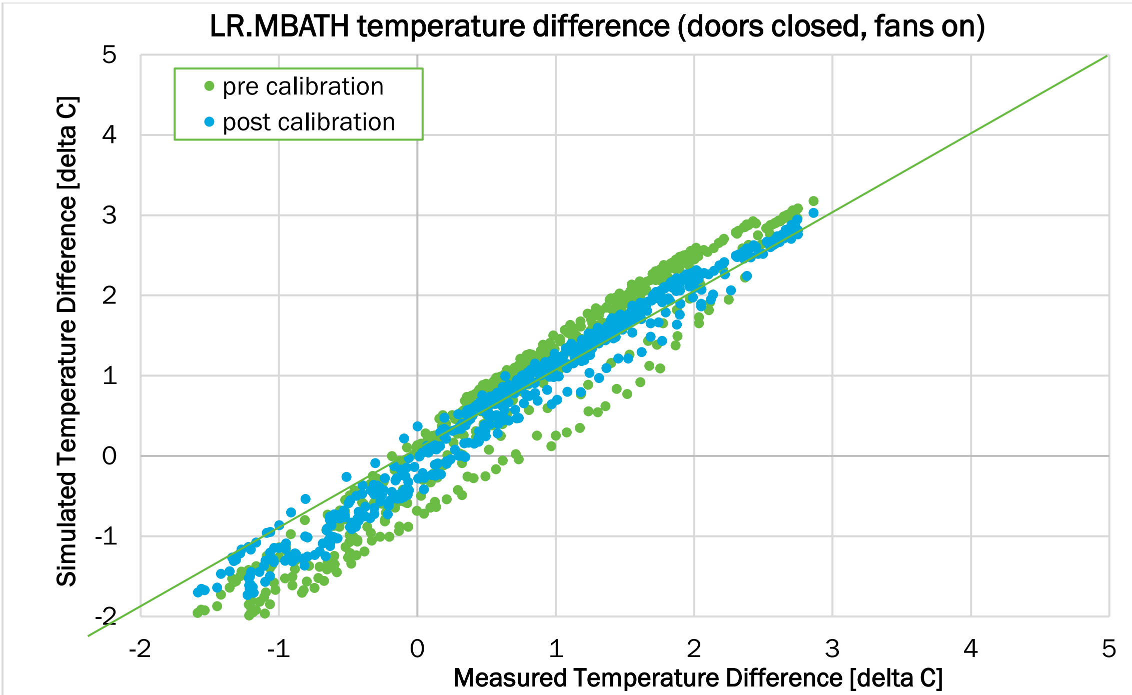


Figure 16. Calibrated model: living room to master bath temp difference (doors closed, fans on)

5 New Jersey Manufactured Home—Unoccupied and Occupied Home Testing

The 1,244-ft^2 New Jersey test home was designed based on lessons learned from the Alabama lab home. This section covers the design, production, and testing of this home. The home was produced at a plant in Pennsylvania and installed in February 2016. It was monitored with simulated occupancy for 18 months, occupied in October 2017 by a family of two, and monitored while occupied for another year until October 2018.

5.1 Design and Specifications

The TRNSYS model helped develop the design of the New Jersey home. The model was revised per the New Jersey plan (Figure 17), which is similar to that of Alabama, with a central living/kitchen area, flanked by a master suite on one side and two smaller bedrooms on the other. The following elements of the calibrated Alabama model were used to inform the New Jersey model, including:

- Thermal stratification observed within the rooms
- Thermal properties of the skirting material in the crawlspace
- Air leakage of the crawlspace skirt
- Ground temperature
- Insulation parameters for the attic blown in insulation, walls, and underfloor batts
- Solar absorptance of the roof
- House infiltration rate (exterior wall leakiness)
- Interior opening characteristics (for airflow between zones).

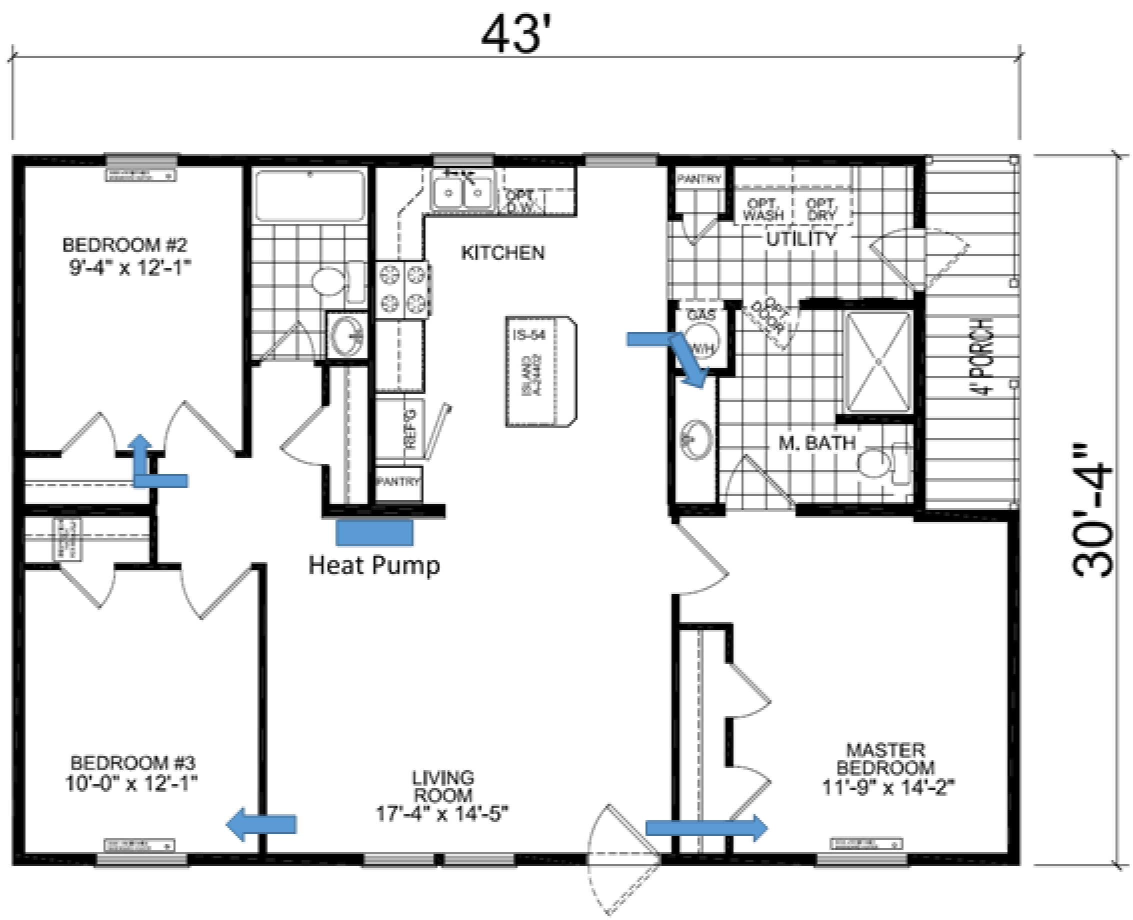


Figure 17. New Jersey floor plan, showing heat pump (blue rectangle) and transfer fans (blue arrows)

A series of alternative specifications were modeled with the TRNSYS New Jersey model, focusing on airtightness and insulation levels. In the model, space conditioning was modeled to occur anytime that the living room temperature went out of the 70°–76°F range throughout the entire year. The comfort range for heating season was 68°–72°F and for cooling season 72°–78°F. A description of each of the cases is given in Table 6.

Table 6. Simulated Case Description

Case	Transfer Fans Strategy	Doors	Exhaust Fan	Infiltration (ACH_{50})	Floor Insulation	Wall Insulation
1	On when room temperature below 70°F	Open	Hall bath	3.8	Added batt	R21+5
2	On when room temperature below 70°F	Closed	Hall bath	3.8	Added batt	R21+5
3	On	Closed	Hall bath	3.8	Added batt	R21+5
4	On	Closed	Master bath	3.8	Added batt	R21+5
5	On	Closed	Master bath	2.5	Added batt	R21+5
6	On	Closed	Master bath	3.8	No added batt	R21+5
7	On	Closed	Master bath	3.8	Added batt	R24+5

Model results are shown in Figure 18 and Figure 19. Following is a summary of the simulation results:

- Case 1 (interior doors open) had the maximum number of hours in compliance.
- In Case 2 the doors were closed, and percent compliance fell (except for the central rooms).
- In Case 3 a rise in cooling compliance was observed. However, when heating was needed at night and cooling during the day, the daytime fan operation cooled the bedrooms, which depressed nighttime heating compliance hours compared to Case 2.
- In Case 4 there was a significant drop in compliance for the hall bathroom.
- In Case 5 there was an approximately 5% improvement in heating compliance and little change in cooling.
- Case 6 resulted in slightly poorer performance than Case 4.
- Case 7 showed a slight improvement in heating compared to Case 4, which was identical except for wall cavity insulation.

Note that in all of these cases, no backup electric resistance heaters were used; therefore, the results show only the impact of the envelope and operational changes on comfort compliance.

The noncompliant heating hours could be brought into comfort compliance with the help of resistance heating in bedrooms and bathrooms.

Earlier tests (see Section 4.3) indicated that the transfer fans as modeled for New Jersey (150 cfm for bedrooms, high on wall, running continuously) are roughly equivalent to an open bedroom door in heating season.

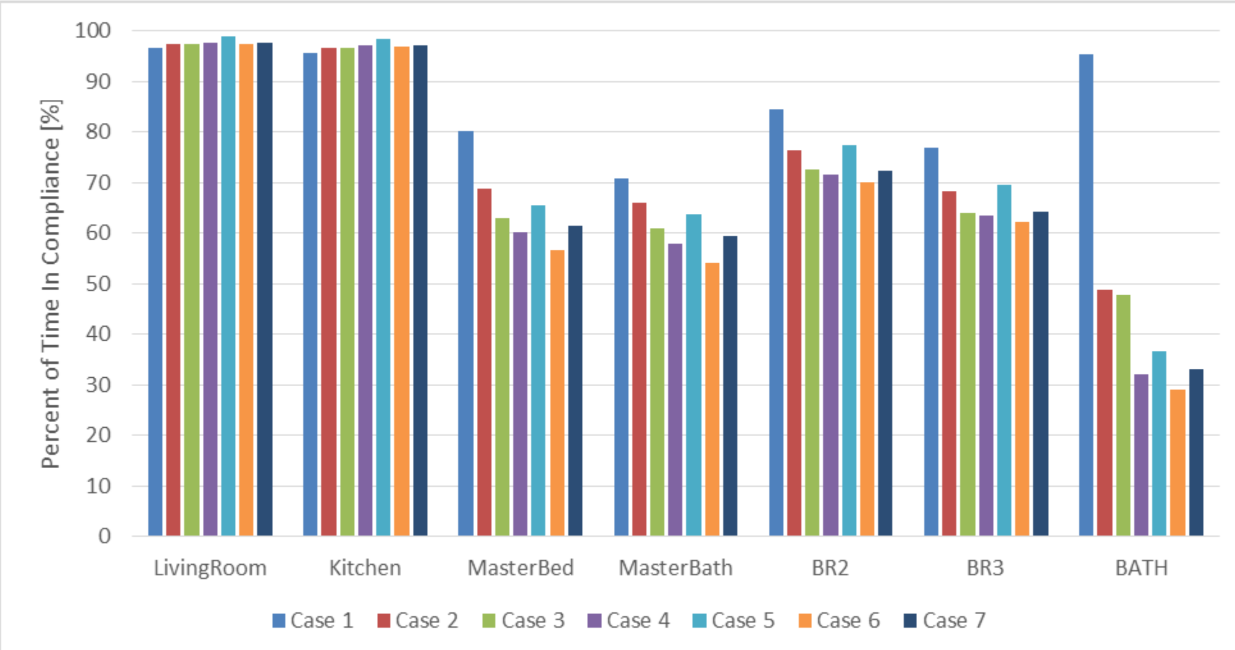


Figure 18. New Jersey simulation options and ACCA compliance in the heating season

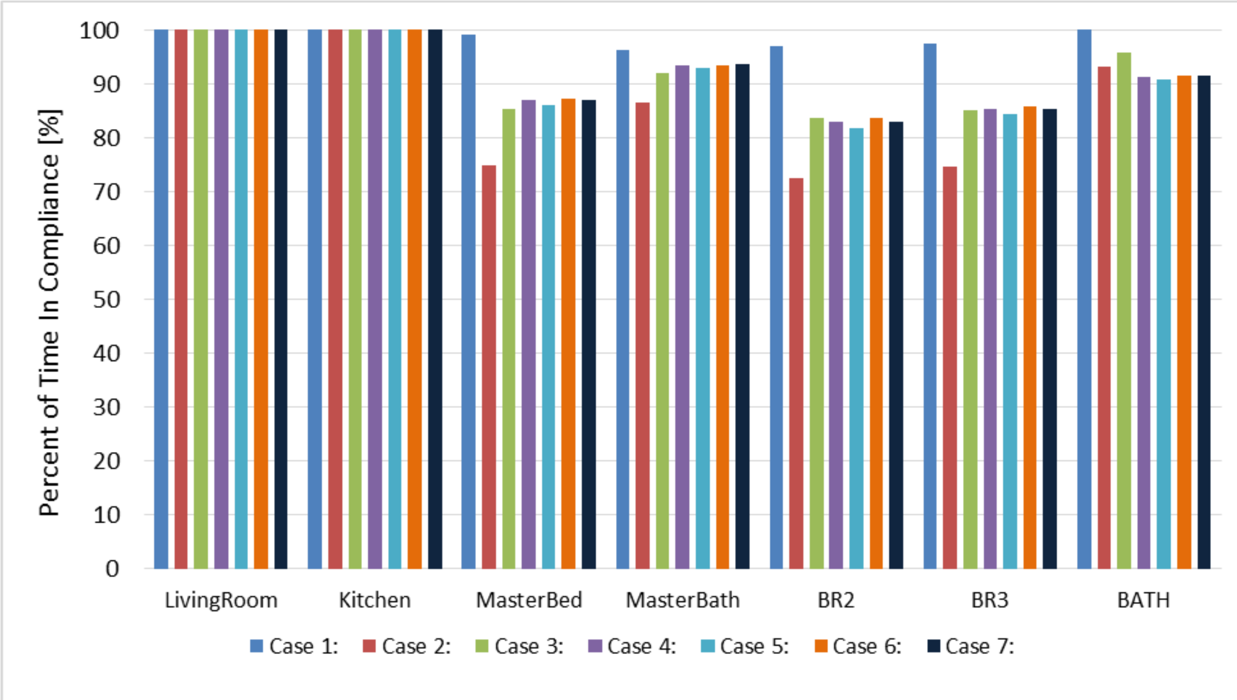


Figure 19. New Jersey simulation options and ACCA compliance in the cooling season

As a result of the above analysis, and taking into consideration the capability of the manufacturing plant, the New Jersey home was designed with the specification shown in Table 7. The 2x6 walls contain R-21 fiberglass batts with 1-in. thick (R-5) extruded polystyrene (XPS) sheathing. The roof is insulated with R-60 blown fiberglass with dense packed eaves. The floor cavity has 2x8 joists with R-19 fiberglass batts in the cavities and double R-11 fiberglass blankets under the floor. Low-emissivity, vinyl, dual-pane, argon-filled windows with a U-value of 0.25 were used. The final New Jersey design corresponds approximately to Case 3 in Table 6 but with lower infiltration.

Additional air sealing measures included sealing the backs of all electrical boxes, more attention to air sealing at ceiling and floor penetrations, and use of gaskets at the top and bottom of all exterior walls and around rough openings.

The home was heated and cooled with a ductless minisplit heat pump mounted on a 2x4 marriage wall in the living room. Through-wall transfer fans were installed between the living room and bedrooms and between the dining area and master bathroom. All distribution system components were sealed with mastic. Supplemental heat was provided by 500-W convector baseboard heaters centered under each bedroom window (one per bedroom). The master bathroom had an under-floor electric heat mat activated by the light switch and thermostat. Water heating was by a natural-gas-fired condensing tankless water heater. A ventilation fan located in the hall bathroom was wired for continuous operation to meet the ASHRAE 62.2 standard. All bedroom and master bathroom doors were undercut by 2 in. The bedroom doors also had return air grilles to ensure return air circulation.

Table 7. New Jersey Home Specifications

Item	Specifications
Space Conditioning and Air Distribution	Heat pump: HSPF 10, SEER 19.5 (Panasonic E18RKUA) Supplemental heat bedrooms: 500-W electric baseboard linear convector (Dimplex) Supplemental heat master bath: Electric heat matt (LaminaHeat)
	Bedroom transfer fans: Panasonic FV-11-15VK1
	Master bathroom transfer fan: Panasonic FV-05-11VK1
Ventilation Fans	Hall bath (continuous exhaust tested at 48 cfm): Panasonic FV-0811VFL5 Master bath: Switch-operated fan
Water Heater	0.95 EF, tankless gas condensing (State GTS-340-NIH)
Floor	2x8 joists 16-in. o.c.; R-11 blanket (2, ~3-in. layers) + R-19 Knauf Ecobatt unfaced batts for joist cavities (5.5-in. thick). Insulation compressed in outrigger areas to approximately 8-in. thick.
Ceiling	R-60 fiberglass blown insulation (Knauf Supercube HD) with ventilation baffles and channels (Accuvent), dense packed at eaves.
Walls	2x6 studs 16-in. o.c., R-21 Knauf Ecobatt unfaced batt (5.5-in. thick) + R-5 Exterior sheathing (Owens Corning FOAMULAR F-150 XPS)
Airtightness	Spray foam, HomeSealR, FlashSealR, RimSealR tapes; Floor penetrations, electrical boxes, extra work on ceiling penetrations, R.O.s; Envelope leakage test result 3.04 ACH_{50}
Windows	U-factor 0.25; SHGC 0.27 (Kinro Enhanced Dual Low-E, Argon; single hung series 9750)
Photovoltaics	450 ft^2, 6,000 watt peak power, oriented south, tilt 22.6 deg, inverter efficiency 96.5%

5.2 Production

The test house was produced on a manufactured home production line alongside standard product. Plant workers were trained on the job in any unusual tasks or techniques. Some delays were encountered on the second and third days of production due to the electric heat mat installation and wiring (day 2) and the additional electrical work in the ceiling for the transfer fans and for wiring of dedicated circuits to facilitate monitoring (day 3). Other tasks that required extra labor—such as additional batt, roll and blown-in insulation, installation of the rigid foam, and air sealing measures such as gaskets—were completed within the normal production timeframes and labor levels. Some eliminated tasks/components such as ductwork offset some of the labor required for these tasks.

Many of the tasks that caused delays on days 2 and 3 would not be necessary if the design were in volume production: the heat mat would be pre-wired and pre-cut and the dedicated circuits for monitoring would be unnecessary. Special framing for securing the transfer fans would be pre-designed and fabricated rather than figured out during production. Dense packing of the ceiling, despite being an unusual technique and the first time for this plant, did not cause any difficulty or delays.

There is no evidence that any of the production methods or materials used in the test could not be employed by any factory builder in the nation. Production at the plant was running at three floors per day. At higher production rates, additional steps such as adding workers or more pre-planning/pre-fabrication may be necessary to ensure the techniques described above do not create delays.

See Appendix C for description and images of the production and installation process.

5.3 Costs

Minimizing incremental costs is critical for maintaining affordability in this housing sector. The incremental costs by item for all components contributing toward reduced space-conditioning energy and for maintaining comfort (including distribution and ventilation) are provided in Table 8 for plant costs and Table 9 for site-related costs. Material costs were from information provided by vendors and otherwise based on the lowest retail prices with an estimated 25% discount to the home manufacturer. Labor costs were based on time observations made during production and estimates of labor cost of $23.32 per hour. The costs are compared to a 2009 IECC baseline home because that is the Building America energy comparison benchmark.

Table 8. Incremental Costs at Plant

Item	New Jersey Home	IECC 2009 Home	Incremental Costs ($)
Primary Space Conditioning	Heat pump (Panasonic E18RKUA)	Electric furnace	1,349
		Gas furnace	1,168
	Heat pump line set & installation	Furnace installation	57
Interior Framing	Upcharge for 2x4 framing for heat pump wall	n/a	2
Backup Heating Equipment	Baseboard heaters (3) + wireless controller	n/a	380
Distribution	Transfer fans for bedrooms (3) (Panasonic FV-11-15VK1)	Interior ducts material	283
	Transfer fans for master bath (Panasonic FV-05-11VK1)	Interior ducts plant labor	53
	Duct components, registers, bedroom door return air grilles	Crossover duct material	47
	Installation		58
Ventilation	Exhaust fan (Panasonic FV-05-11VKSL1)		170
Insulation–Floor	R-11 blanket (2 layers) + R-19 unfaced batts in joist cavities	R-19 fiberglass blanket insulation, 2x8, 16" o.c.	304
Insulation–Ceiling	R-60 blown fiberglass, dense pack eaves	R-38, fiberglass blown insulation	666
	Dense packing labor	n/a	35
Insulation–Walls	R-21 unfaced batt	R-19, fiberglass batt, 2x4 wall, 16" o.c.	83
	Insulated sheathing (FOAMULAR F-150 XPS, R-5)	n/a	615
Airtightening Measures	Spray foam, HomeSealR, FlashSealR, RimSealR tapes	n/a	345
	Floor penetrations, electrical boxes, extra work on ceiling penetrations, R.O.s	n/a	163
Windows	Vinyl, dual low-E, argon, stainless steel spacer, single hung, U-factor 0.25 (KINRO)	Vinyl, low-E dual glazing, U-factor 0.35	159
Incremental Plant Cost Compared to Electric Baseline			4,768
Incremental Plant Cost Compared to Gas Baseline			4,587

Table 9. Incremental Costs at Site

Item	New Jersey Home	IECC 2009 Home	Incremental Cost ($)
Air-Conditioning System (Cost to Homebuyer)	n/a	AC 13 SEER 2 ton installed	(3,000)
Distribution (Duct System)	n/a	Crossover duct installation	(25)
Utility Connection	n/a	Gas hook-up	(150)
Incremental Site Cost Compared to Electric Baseline			(3,025)
Incremental Site Cost Compared to Gas Baseline			(3,175)

5.4 Instrumentation

After installation in New Jersey, the home was instrumented and configured for simulated occupancy testing as follows:

- **Thermostat set points**: The heating set point was 71°F, and the cooling set point was 76°F.
- **Interior doors**: Because the impact of door opening on comfort is a key research question, data were collected both with interior doors open and closed. Short-term tests (2–4 days) were conducted to compare temperature distribution across the home with doors closed and open. Comfort criteria were applied to periods with doors closed.
- **Window blinds**: Window blinds were set at 50% closed.
- **Internal gains**: Sensible internal heat gain was simulated according to the *2014 Building America House Simulation Protocols* (Wilson et al. 2014) through the use of electric resistance heaters located in each bedroom and the main living spaces and controlled by the data loggers. Figure 20 shows the schedule of loads. Latent loads were simulated through the use of ultrasonic humidifiers using methods previously employed by NREL (Fang et al. 2011).

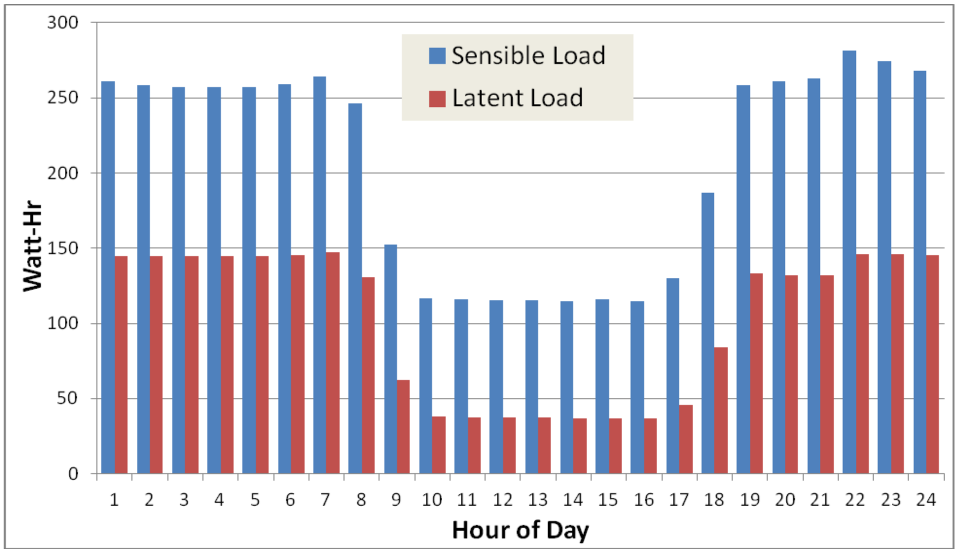


Figure 20. Sensible and latent load profiles

- **Install monitoring system**: The home was instrumented to gather performance and comfort data, including room temperatures, relative humidity (RH), buffer space (attic and crawlspace) temperatures, outdoor conditions, mechanical equipment runtime, and energy use (Table 10). Data were sampled every 20 seconds and were stored as 1-min, 15-min, and 60-min averages using a Campbell Scientific data logger (CR1000) in conjunction with a multiplexer (Campbell Scientific AM16/32B). Data were collected remotely twice daily.

Table 10. Sensors

Data Point	Sensor	Location and Details
Air Temperature	T-type thermocouples in aspirated shields	Near the center of each room mounted on tripods 5 ft above the floor, and at least 5 ft away from heating/cooling/air distribution systems and exterior walls. Additionally, at two locations in the main living space, high/low air temperatures were measured 12-in. and 84-in. above the floor to gauge stratification. Transfer fan inlet and outlet air temperature was measured at one transfer fan location (master bedroom).
Air Temperature	PointSix wireless temperature sensors	In the attic and crawlspace
Relative Humidity	Campbell Scientific CS215 T/RH probe	In a central location in the home; mounted in a passive radiation shield outdoors, and in the north-facing exterior wall cavity
Condensation	OmniSense S900-1	Measured moisture content of the sheathing at two locations in the north-facing wall. One additional sensor was placed in the attic on the underside of the roof sheathing near the ridge.
Power Consumption	Continental Control System ACT-0750 current transformers	In conjunction with Wattnode energy-to-pulse transducers, measured power of HVAC equipment and total house power
Current	Veris H721LC current transducers	Compressor and fan current
Status	Veris H300 current switches	Status of reversing valves of the heat pump, the transfer fans and bath fan
Solar Radiation	Campbell LI200X-L Silicon pyranometer measured	Pyranometer measured horizontal global solar radiation at the site
Wind	RM Young Wind Sentry	Wind speed and direction were measured from a weather station on the roof of the home

When converting to occupied testing, the aspirated thermocouples in the centers of the rooms were replaced with wireless PointSix temperature/relative humidity (T/RH) sensors mounted to the wall near the entry of each room, and the other thermocouples measuring stratification and transfer fan air temperature were removed. Sensible and latent load simulation equipment was removed.

5.5 Results and Analysis

This section of the report summarizes the data and analysis from the New Jersey home and compares it to the success criteria: compliance with ACCA Manual RS, ASHRAE 55-2013 (ASHRAE 2013b), and ASHRAE 62.2-2010 (ASHRAE 2010), and achieving source space-conditioning and ventilation energy savings of approximately 50% compared to baseline code.

5.5.1 Space Conditioning and Ventilation Energy

Model Comparison

BEopt models were used to predict annual energy consumption of baseline houses of the same design but with minimum 2009 IECC energy-related specifications, as well as two versions of the as-built New Jersey house. Table 11 lists the modeling inputs for both as-built and baseline code houses. Two heating equipment options are shown for the baseline code house: one with a natural gas furnace and one with an electric resistance furnace. Both are common for manufactured homes. The two versions of the as-built house represent (1) a best-case scenario with no supplemental resistance heating, and (2) a case with supplemental resistance heating similar to that measured in the simulated occupancy home.

Supplemental space-conditioning energy data were collected for representative periods during heating season when the home was operating under simulated occupancy conditions. These data were then used to generate a regression model (see Appendix E) for the supplemental heating energy as a function of ambient temperature. However, because of poor results of the regression model, an alternative, more conservative approach was used to estimate supplemental heating needs by way of calculations (Appendix F).

Because BEopt is a single-zone model, the supplemental heat in the supplemental heating case model was accounted for by the addition of the load calculated in Appendix F, which was added as an internal plug load. Measured data were used for transfer fan and ventilation fan energy for both as-built models. Transfer and ventilation fans ran continuously. The resulting annual model-runs in the New Jersey climate predict that the as-built test house will consume between 44% and 53% less source space-conditioning energy than the gas-heated code house and between 72% and 76% less than the electric-resistance-heated code house (Figure 21). The range depends on the amount of supplemental electric heat required in the as-built home.

Table 11. Model Inputs New Jersey

Feature	As-Built	Baseline IECC 2009
Primary Space Conditioning	Ductless heat pump: 19.5 SEER, 10 HSPF	Electric furnace OR gas furnace, 78% annual fuel utilization efficiency 13 SEER AC
Backup Heating	500-W linear convector baseboard in three bedrooms; Electric matt in master bathroom	None
Distribution	Transfer fan (bedrooms): 13-W continuous 150 cfm	Ducts: 15% total duct leakage, R-8 duct insulation
	Transfer fan (master bath): 11.5-W continuous 110 cfm	
Ventilation	Hall bath, 0.08 W/cfm, 50 cfm	Standard exhaust fan, 0.3 W/cfm, 50 cfm
Floor Insulation	R-11 blanket x 2 layers + R-19 unfaced batts in 2x8 joist cavities, 16-in. o.c.	R-19 fiberglass blanket insulation, 2x8, 16-in. o.c.
Attic Insulation	R-68 fiberglass blown insulation with dense packed eaves	R-38 fiberglass blown insulation
Wall Insulation	R-21 unfaced batt, 2x6 wall, 16-in. o.c. Exterior sheathing: XPS, 1 in. R-5	R-13, fiberglass batt, 2x4 wall, 16-in. o.c.
Windows	U: 0.25, SHGC: 0.27	Vinyl, low-e dual glazing U: 0.35, SHGC: 0.44
Doors	U: 0.32, SHGC: 0.18 + storm doors	U: 0.32, SHGC: 0.18 + storm doors

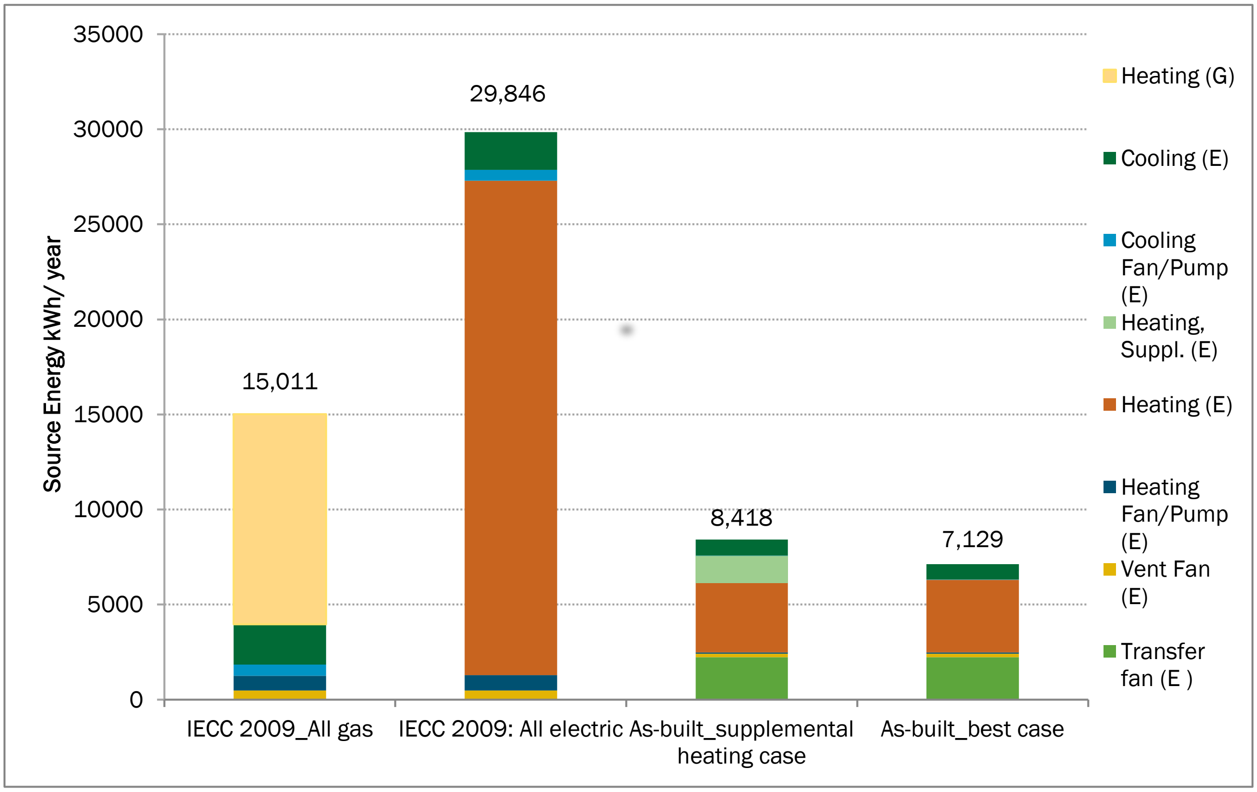


Figure 21. New Jersey BEopt energy model comparison

Occupied Energy Consumption

For the occupied period, energy consumption data were collected during the year-long period spanning October 26, 2017, through October 25, 2018. The energy consumption was then scaled by a factor of 2.80 (U.S. Environmental Protection Agency) to estimate source energy use (for electricity). Results, broken out by category, are presented in Table 12. The occupied period energy consumption reflects the residents' behavior, weather, and thermostat set points and so is not directly comparable to the simulated occupancy period that uses standard occupancy assumptions.

Table 12. Monitored Energy Results New Jersey (Occupied)

Source	Occupied Annual Energy in kWh
Heating (heat pump)	5,080
Supplemental heating	570
Cooling	371
Ventilation fan energy	411
Transfer fan energy	718
Total space conditioning and ventilation energy	7,149
All other energy, except water heating	5,005
Total home electric energy	12,154
Photovoltaic production	-18,079

5.5.2 Comfort—ACCA Manual RS

Comfort was evaluated under two sets of criteria: ACCA Manual RS and ASHRAE Standard 55-2010. The thermal comfort targets in the ACCA Manual RS 2015 are summarized in Table 13.

Table 13. Thermal Comfort Metrics (ACCA Manual RS)

Comfort Item	Heating	Cooling
Thermostat Set Point (Design)	70°F	75°F
Relative Humidity (RH)*	30% RH maximum (20%–30% RH is desirable)	55% RH maximum (25%–50% RH is desirable)
Dry-Bulb Temperature at the Thermostat	Set point temperature ±2°F	Set point temperature ±3°F (single zone) Set point temperature ±2°F (multizone)
Dry-Bulb Temperature in Any Conditioned Room	Set point temperature ±2°F	Set point temperature ±3°F (single zone) Set point temperature ±2°F (multizone)
Room-to-Room Temperature Differences	4°F maximum	6°F maximum (single zone) 4°F maximum (multizone)
Floor Temperature (Slab Floors or Floors Over Unconditioned Space)	65°F minimum at 4-in. above the floor for 70°F thermostat setting (not applicable near outside walls)	N/A

* Humidification is optional, but desirable in many situations. The potential for visible or concealed condensation determines maximum RH for a specific dwelling in a specific location.

For the purposes of this study, these criteria are interpreted as follows:

- Criteria apply only when space conditioning system is operating[2]
- RH compliance are considered successful if achieved 95% of cooling season
- RH compliance are considered successful if achieved 95% of heating season
- Temperature variations are allowable spatially and temporally
- Cooling comfort are considered successful if achieved 95% of cooling season
- Heating comfort are considered successful if achieved 100% of heating season
- Single zone.

The ACCA criteria are separated into heating and cooling, and so are presented separately in the following subsections, with cooling following heating.

5.5.2.1 Simulated Occupancy Heating Period

Relative Humidity

Figure 22 shows the RH in the living room compared to ambient temperature and heat pump set point for the simulated occupancy heating period when the heat pump was operating. The living room RH stayed within a 25%–50% range, most of the time under 40%. The heat pump was not operated in dehumidification mode.

[2] This criterion was interpreted to mean when the heat pump compressor is running.

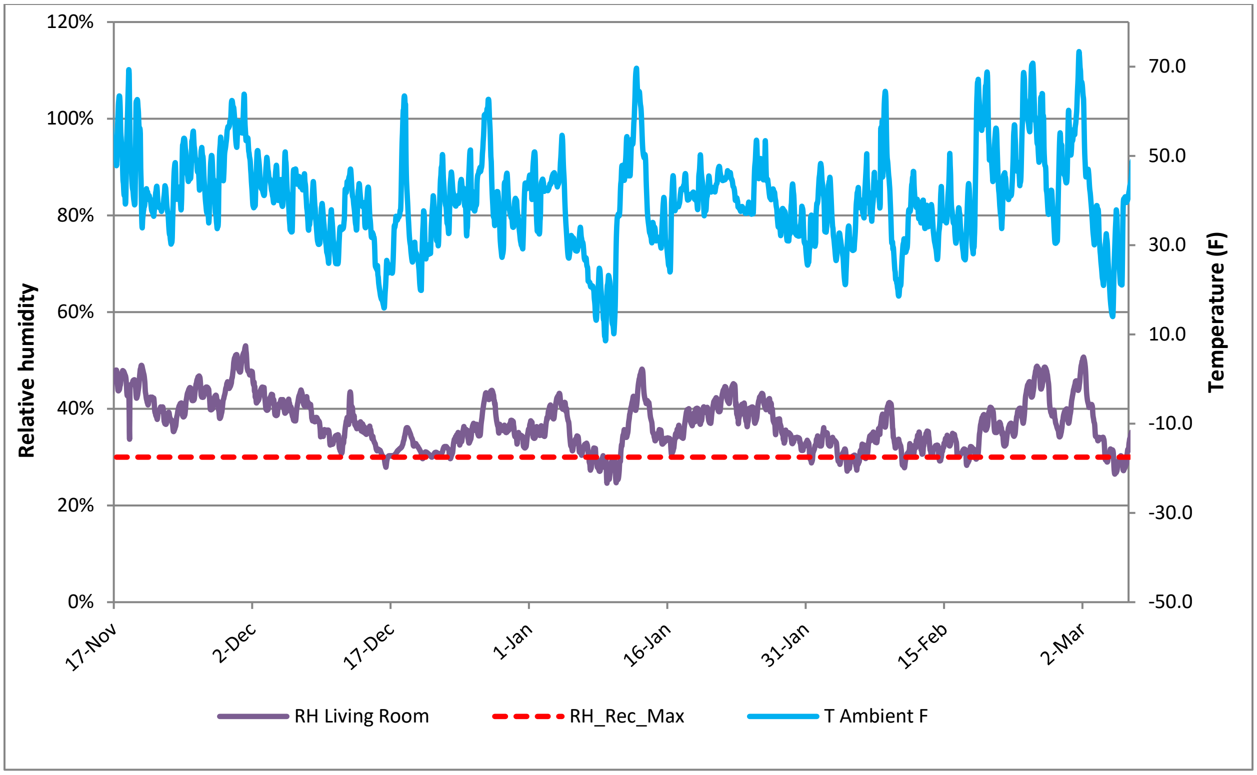


Figure 22. Heating season RH data—living room

Dry-Bulb Temperature at the Thermostat and Dry-Bulb Temperature in Any Conditioned Room

Room temperatures and outdoor temperatures during the simulated occupancy heating season are shown in Figure 23 and Figure 24. The heat pump was set at 74°F in April and 70°F in the second period. The following observations were made:

- When the transfer fans were activated on the afternoon of April 5 (blue arrow in Figure 23), the temperatures in the bedrooms increased by approximately 7°F over the course of 6–12 hours (note that ambient temperatures were also increasing during much of this time).
- When the ambient temperature was below about 50°F, there was a noticeable increase in temperature spread among the bedrooms from about 4°F to 6°F.
- When the ambient temperature was at its lowest, about 10°F, the house could not maintain set point and room temperatures dropped to 66°F. Note that the heating season design temperature for the New Jersey home location is 14°F.
- Despite the set point being 70°F, the temperature in the living room, where the heat pump was located, was consistently higher (70°–76°F) when the heat was operating and ambient temperature was above 50°F. According to the manufacturer (Panasonic), the room temperature is measured at the return-air inlet, which may differ from the average room

temperature. It is possible that the unit's logic assumes a higher discrepancy between room temperature and return-air temperature than the average of 2°F measured here.

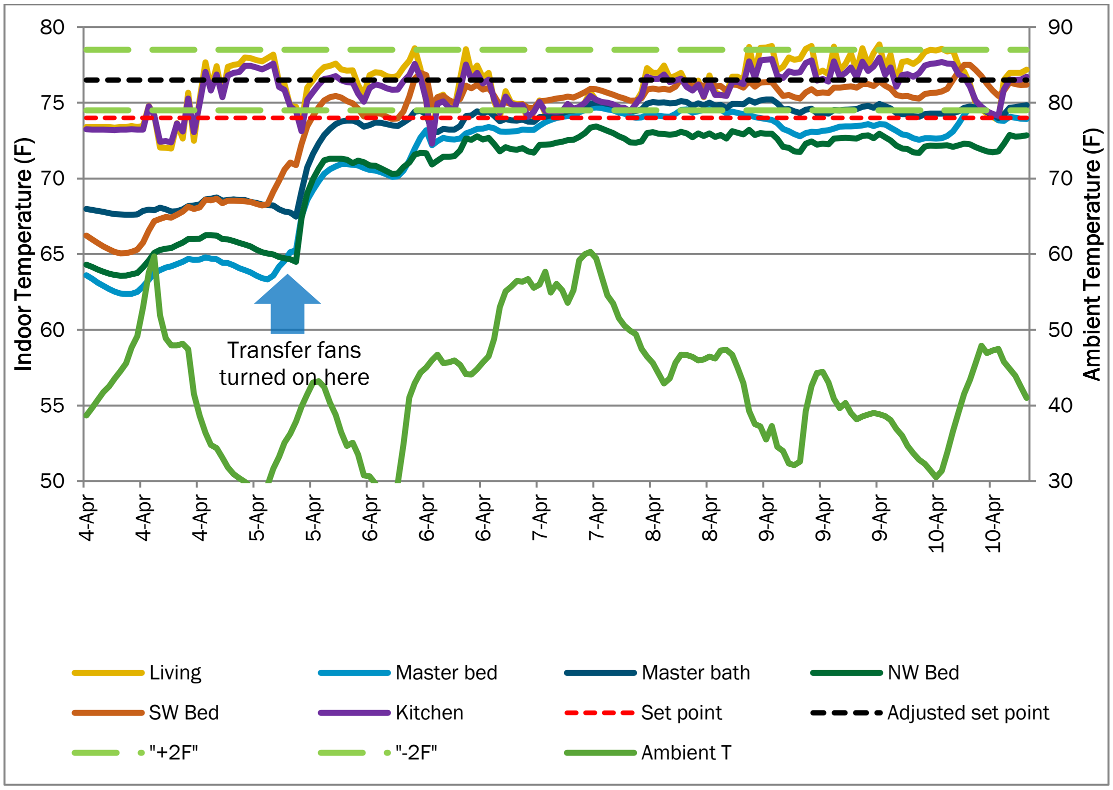


Figure 23. Heating season room temperature data—2016 April

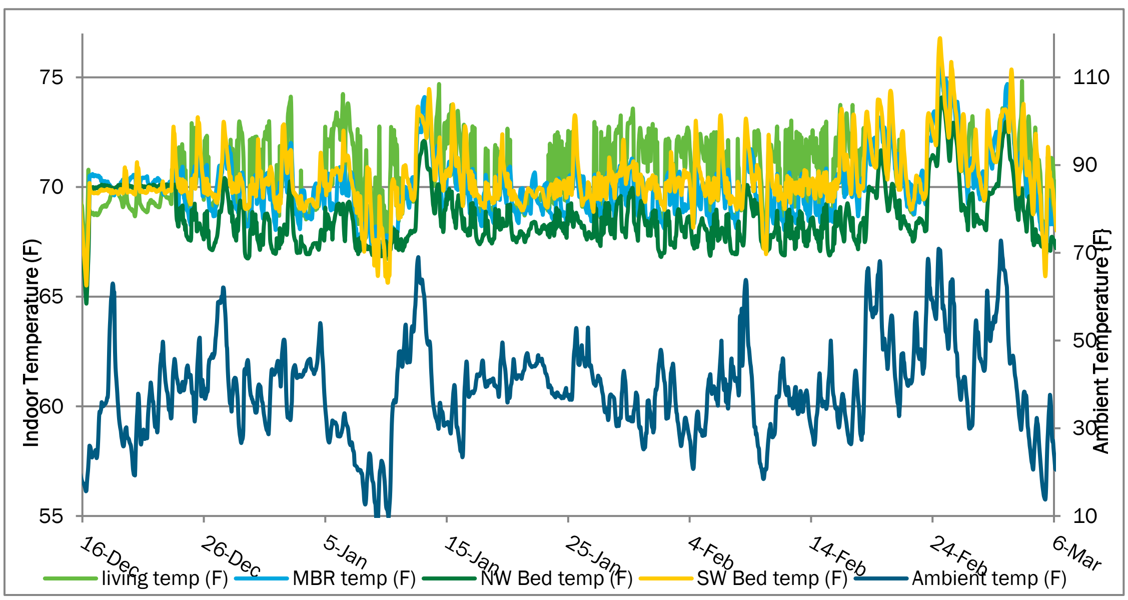


Figure 24. Heating season room temperature data—2016 Winter

Room-to-Room Temperature Differences

Room-to-room temperature differences were set to be within 4°F in heating mode. Both the master bedroom and northwest bedroom had periods when temperature differences were outside the 4°F range. During most of these periods, ambient temperature was below 50°F. During the coldest periods of the data shown in Figure 24, when ambient temperatures were 10°–20°F, however, room-to-room temperature differences were within 4°F. This was due to the operation of backup resistance heaters.

Floor Temperature

The near-floor air temperature in the northwest bedroom was reviewed for compliance with the floor-temperature requirement. Figure 25 shows the temperature near the floor for the duration of the monitoring period with relevant data from December 2016 to March 2017. The northwest bedroom is the coolest room (during heating season) with only a north-facing window and the least favorable transfer fan placement. (The northwest bedroom's transfer fan was located near the door, subject to possible short circuiting of airflows, because this was the only possible placement on the wall between the living room and northwest bedroom without using extra ductwork.) The heating set point was 70°F, so the compliance threshold was adjusted to 5°F lower than the set point, or 65°F. The near-floor temperature was well above the compliance threshold for the duration of the period when heating was operating.

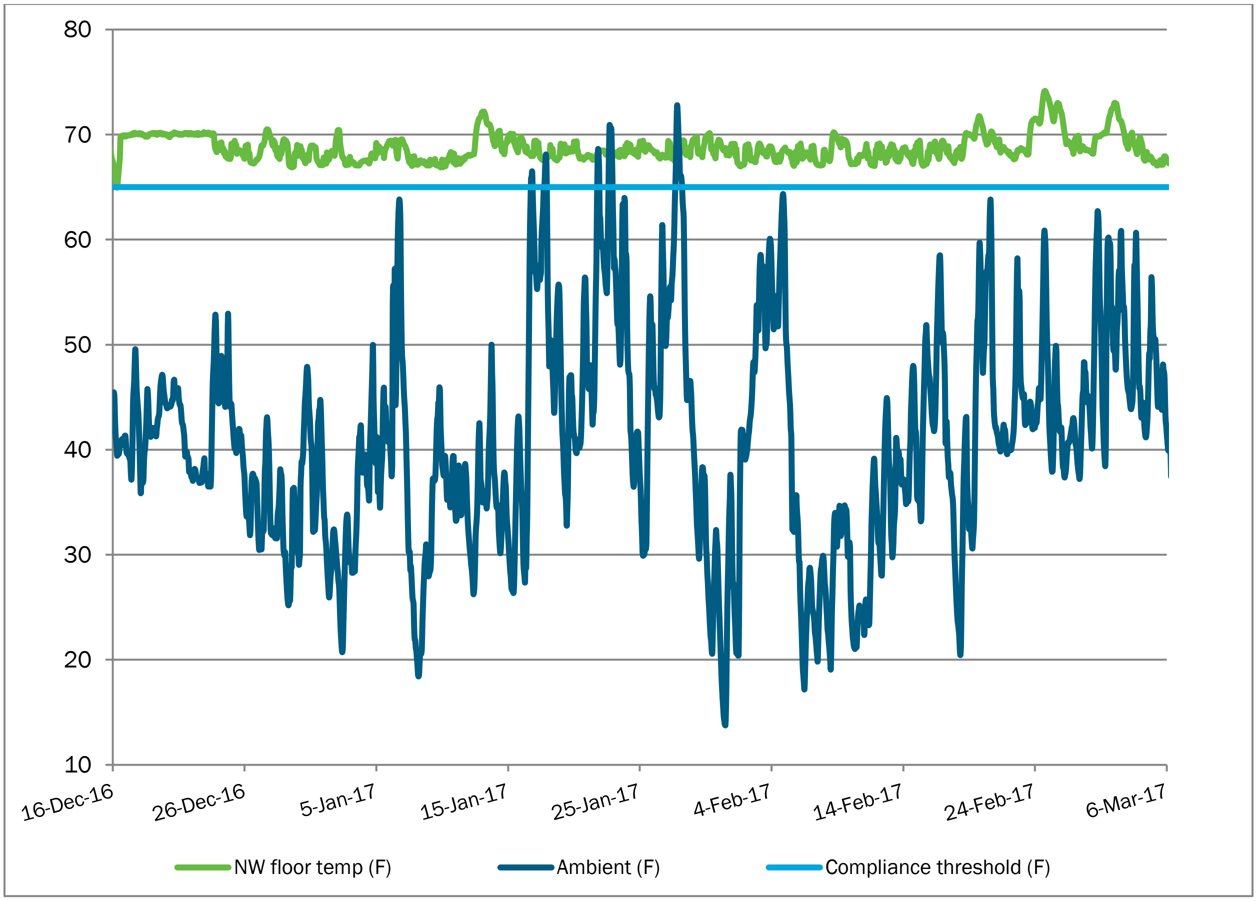


Figure 25. Near-floor air temperatures—northwest bedroom

5.5.2.2 Occupied Heating Period

Relative Humidity During Heating Season

Figure 26 shows the RH in the living room compared to ambient temperature for the main portion of the occupied heating period when the heat pump was operating. The living room RH stayed within a 20%–50% range, and most of the time was under 40%. There were periods above 40% when outdoor temperature was higher than 60°F. For ACCA Manual RS compliance, room RH is higher than 30% for about 50% of the time.

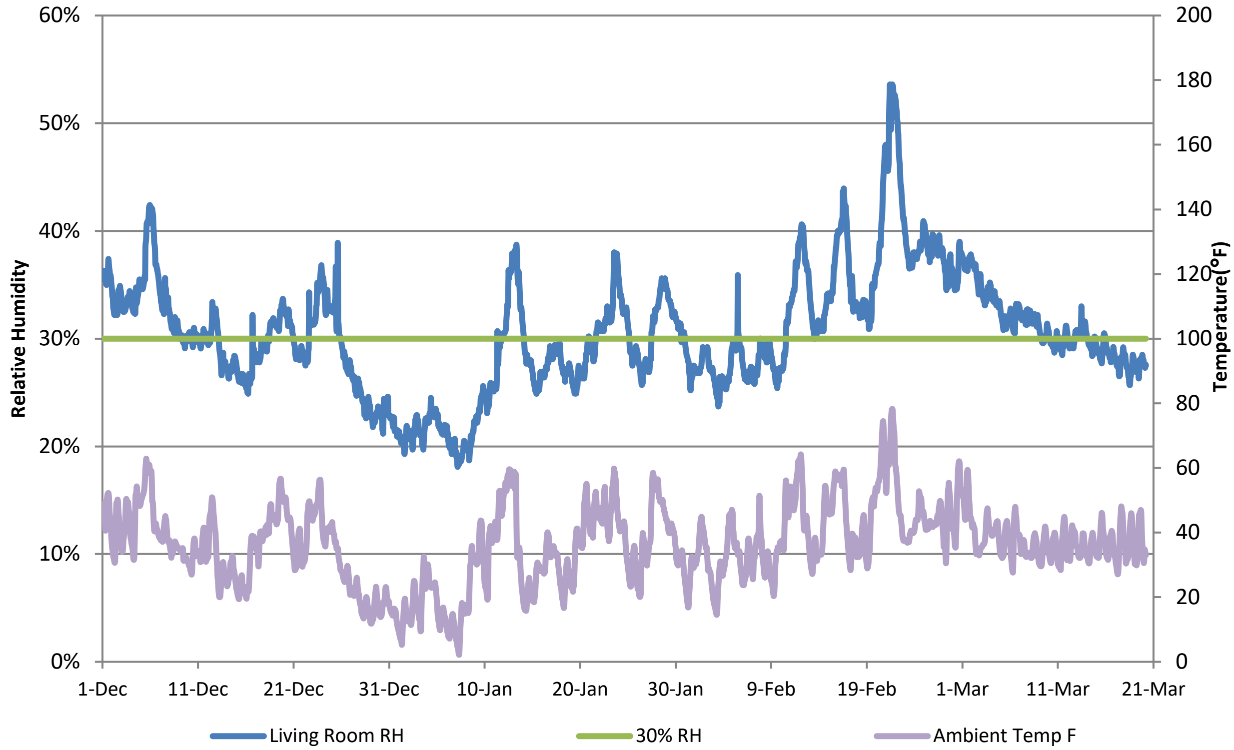


Figure 26. Heating season RH data in living room—occupied period

Dry-Bulb Temperature at the Thermostat and in Any Conditioned Room

Room temperatures and outdoor temperatures during the winter (December 1, 2017–March 20, 2018) are shown on the graph in Figure 27. The heat pump was set to heat at a set point of about 74°F in December 2017 and around 71°F in 2018 (estimated from living room temperature, as set point was not tracked directly). Room temperatures were within ±2°F of the assumed set point for 80% of the time.

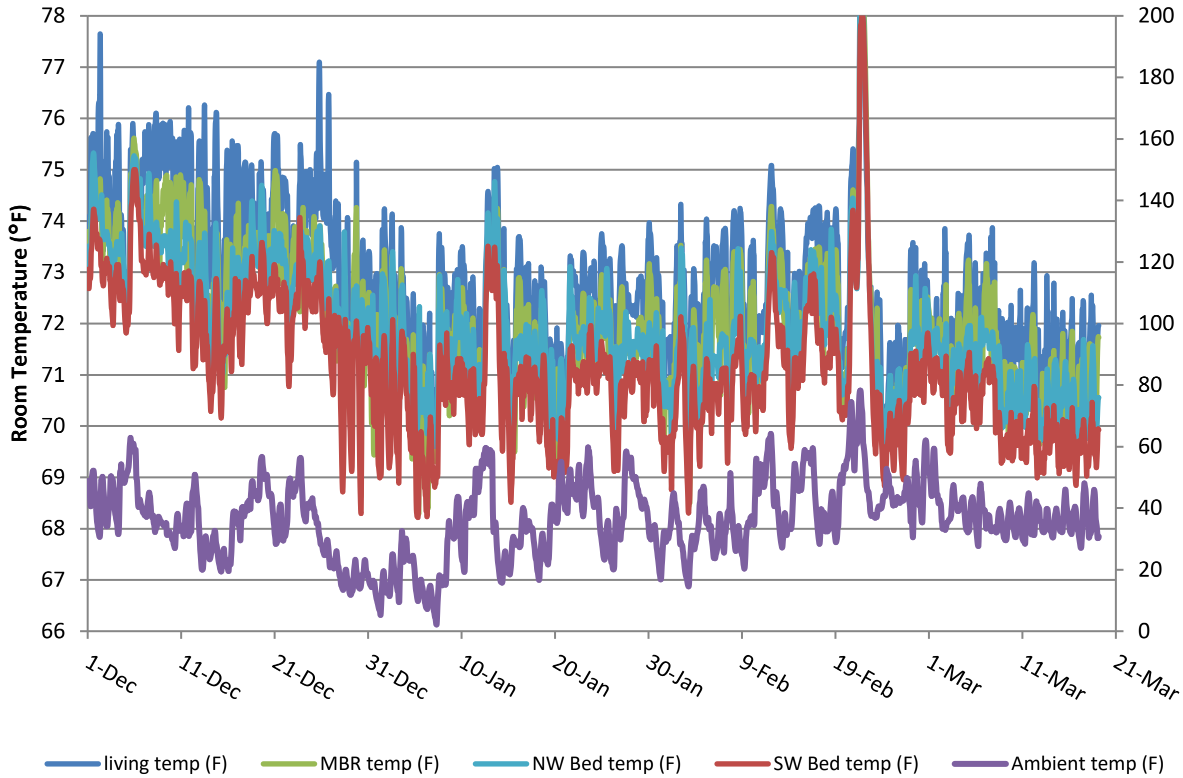


Figure 27. Heating season room temperature data

Room-to-Room Temperature Difference

Throughout the heating season, room-to-room temperature differences were within 4°F for 99.2% of the time. When ambient temperature was below 20°F, the room-to-room temperature difference became smaller, due to the operation of the backup resistance heaters.

5.5.2.3 Simulated Occupancy Cooling Period

During the simulated occupancy cooling analysis period (late July to early September 2016), the house was fully instrumented, latent gains were operative, and the crawlspace skirting was complete.

Relative Humidity

ACCA defines the acceptable RH upper limit as 55% during the cooling season (absolute maximum) with a “desirable” range of 25%–50%. However, the heat pump alone was unable to achieve low enough humidity levels to meet these criteria (Figure 28). The northwest bedroom had the highest RH levels, and the living room had the lowest RH levels of all rooms, but all rooms fell roughly between 55% and 65% RH for the majority of the time. Temperatures never neared dew point.

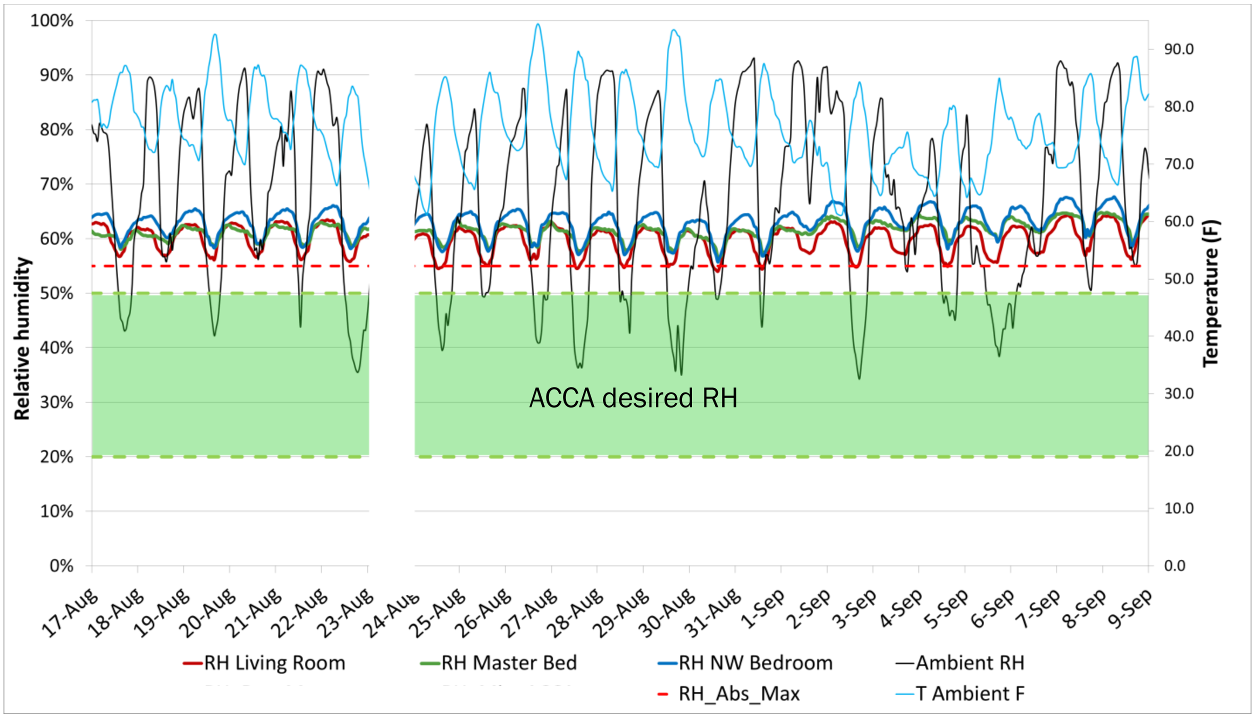
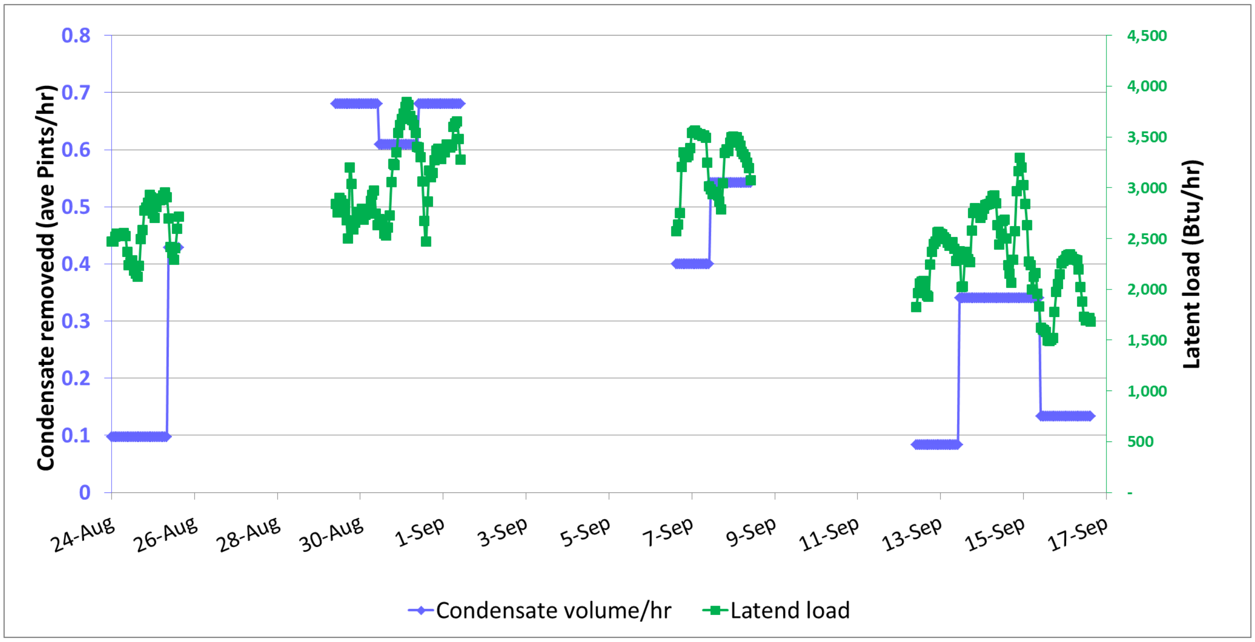


Figure 28. Simulated occupancy cooling season RH data

Temperature was held within allowable limits. The heat pump was unable to meet ACCA criteria for RH, in part because it was oversized for cooling needs. Energy-efficient homes with low cooling loads in climates that experience high humidity commonly face this challenge (Brown, 2013). Because the heat pump achieved the temperature set point quickly, it did not need to run as often or at high speeds, which would have removed more moisture. Figure 29 plots heat pump condensate measurements against outdoor temperature and latent load. Total latent load was calculated hourly based on ambient temperatures, ventilation rate, and internal gains simulated as per the *2014 Building America House Simulation Protocols* (Wilson et al. 2014). When ambient temperatures were higher, and presumably the heat pump was operating more often, more moisture was removed.



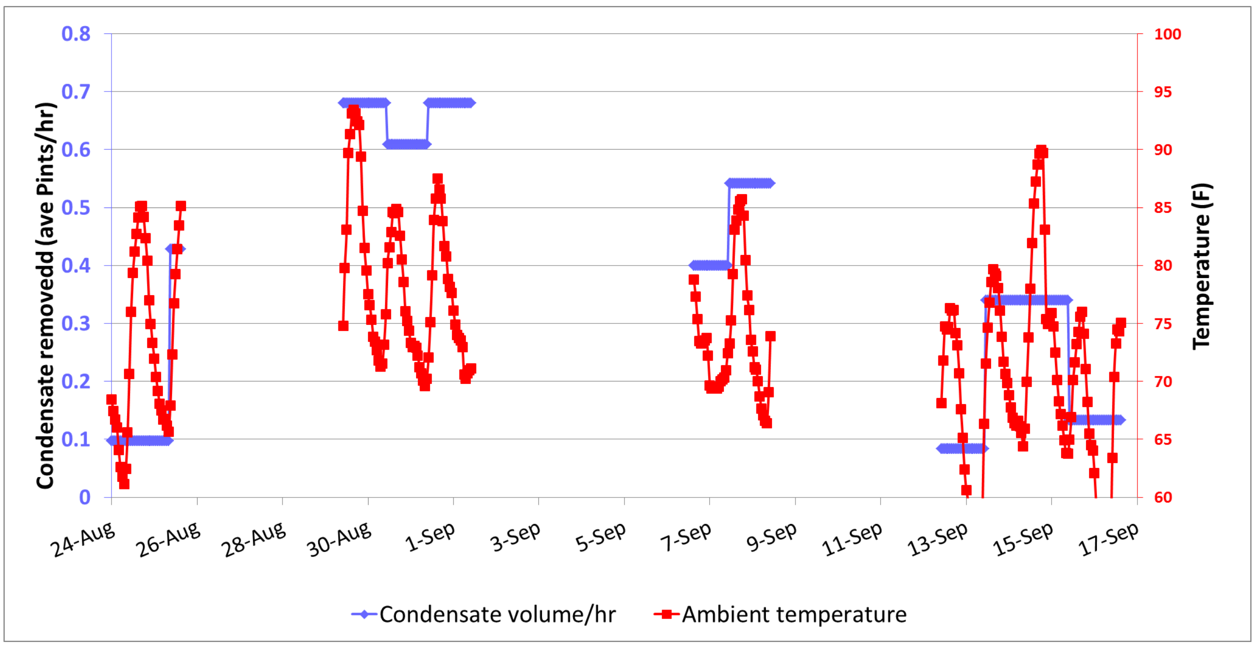


Figure 29. Moisture removal by heat pump vs. site conditions

In an attempt to reduce RH levels in the home, the heat pump was set to "dry" mode on July 28, with a set point of 73°F. "Dry" mode alters the heat pump operation so that it runs more frequently but at higher fan-coil temperatures so that room temperatures do not drop far below set point. This made some progress in reducing RH (from between 68% and 77% to between 55% and 70% RH), and the impact on master bedroom temperature was evident (dropped to the mid-60s) (Figure 30). Next, the system was reverted to "cool" mode and set point was reduced from 73°F to 71°F. This was also effective at reducing humidity (Figure 30).

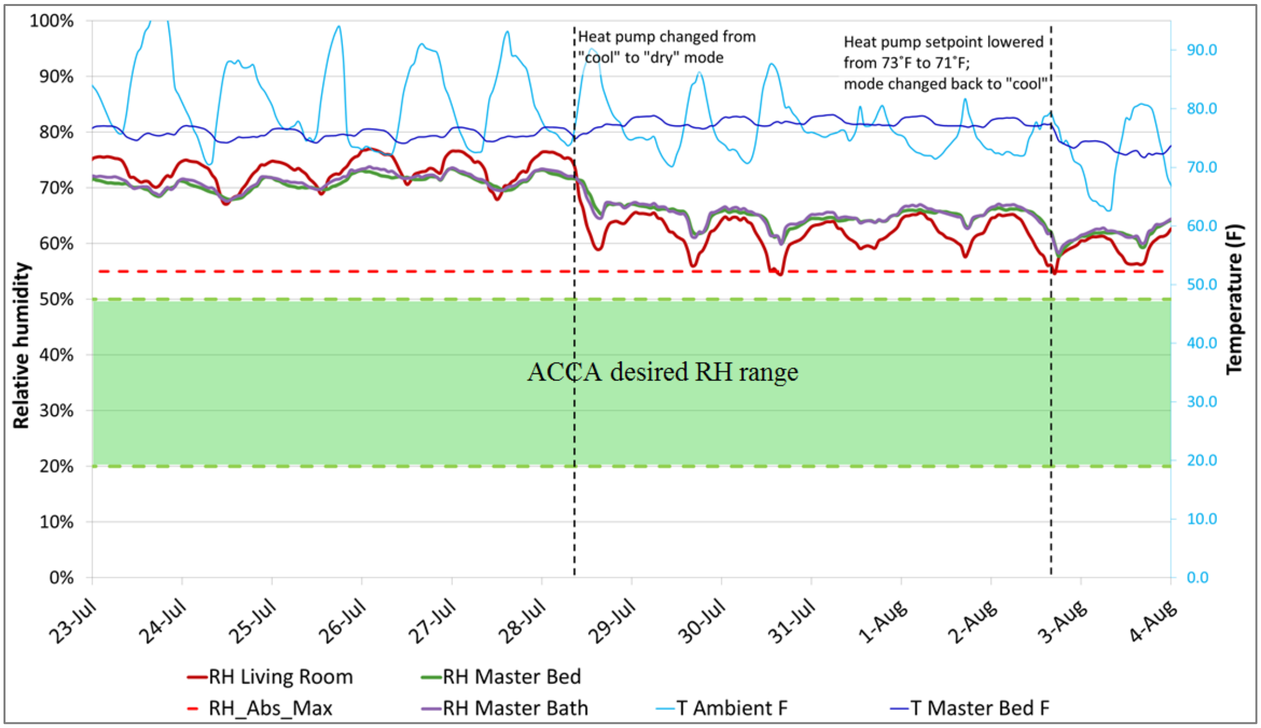


Figure 30. RH in different heat pump modes

An alternative method, the use of a dehumidifier, was tested to reduce RH between August 4 and August 16. The 500-W dehumidifier with an RH set point of 50% was placed in the master bathroom in order to drain into the bathtub. The dehumidifier successfully reduced RH in the master bath to 50% and lowered it in the balance of the home to the high 50s. It also increased air temperature in the master bathroom by about 5°F. However, air distribution was not sufficient, even with 110 cfm from the transfer fan, to effectively dehumidify the entire home (Figure 31). The dehumidifier used about 600 W consistently, which, if employed as the solution to humidity, would unacceptably increase energy consumption.

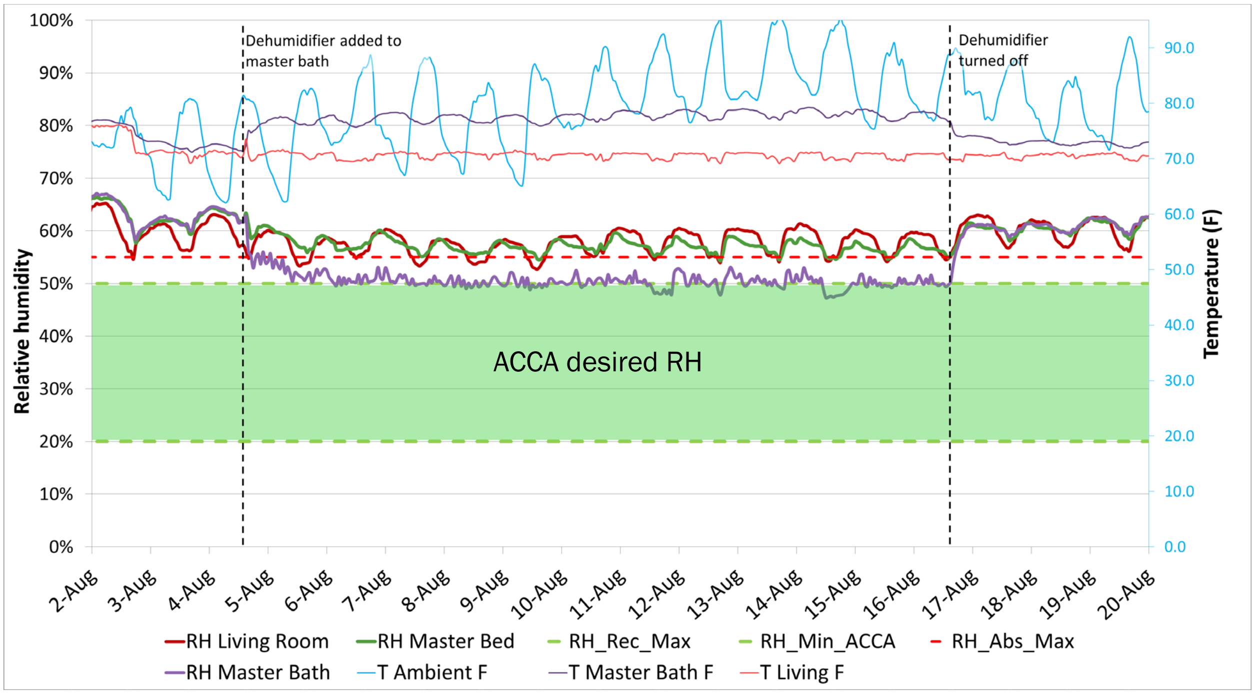


Figure 31. RH levels with and without dehumidifier

This led to brainstorming other solutions pertaining to the heat pump itself, including a split-coil heat pump, multizone heat pumps, and simply a heat pump with lower cooling capacity (although the lower latent capacity of a lower-cooling-capacity unit may offset the longer heat pump runtime). A heat pump with higher latent load removal capability was sought, which could directly address the present heat pump's shortcomings in this context. Panasonic's split-coil unit, currently used in Japan, boasts proportionally higher latent capacity, but with a slight energy penalty (SEER of 19 to 21 instead of 22). The device works by operating two coils that simultaneously cool (and dehumidify) and heat; however, this device is not yet available or approved for the U.S. market. Daikin's Quaternity product—a similar technology available in the United States—may be a viable option, albeit with a cost premium.

Although humidity levels exceeded ACCA standards during most of the cooling period, other standards and anecdotal evidence suggest comfort was satisfactory. ASHRAE 62.1-2010 and 2013 state that RH should be at or below 65% in the presence of mechanical systems with dehumidification capability (ASHRAE 2010; ASHRAE 2013a). This reinforces the importance of multiple comfort benchmarks—another of which will be discussed in Section 5.5.3.

Dry-Bulb Temperature at the Thermostat and Dry-Bulb Temperature in Any Conditioned Room

The ACCA design set point temperature was the goal for the home as a whole during the cooling period. Because a transfer-fan air distribution system was employed in the test home instead of ducted distribution, the thermostat set point temperature was lowered from 75°F to 73°F in a strategy to achieve temperatures within ACCA's acceptable range in the remote rooms. Another reason for using a lower set point temperature was that the master bedroom and southwest bedroom (both conditioned through transfer fans from the living area) experience solar gains. Further, the set point was dropped to 71°F on August 2 to increase heat pump runtime to combat humidity. However, this set point decision prevented some rooms from achieving complete compliance with ACCA targets, given that the acceptable temperature range is 72°F to 78°F. Rooms with direct connection to the heat pump (living and dining) experienced lower temperatures.

During the simulated occupancy cooling season, the temperature readings at the heat pump return and in the living room were very close (within 0.1°F), so an adjusted set point was used when heating was not required. Temperature distribution across rooms was mostly within +/- 3°F from the set point of 71°F, but often was below the acceptable range given the ACCA-defined design temperature (Figure 32). Specifically, the living room, kitchen, and northwest bedroom fell below 72°F, which is expected given their proximity to the heat pump, and in the case of the northwest bedroom, the lack of solar gains.

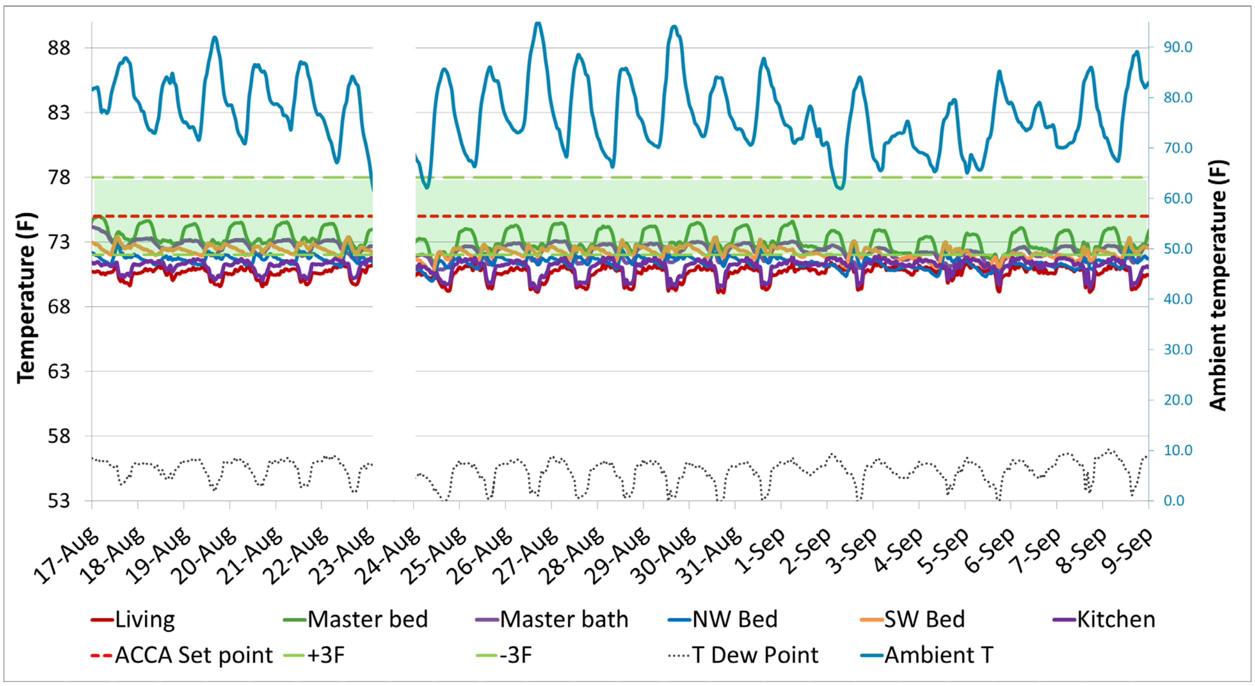


Figure 32. Warm-weather room temperature data

Room-to-Room Temperature Differences

Although indoor temperatures as a whole were relatively low during the simulated occupancy measurement period, all rooms were within 6°F of one another for 100% of the time measured, as required by ACCA (Figure 32). This result suggests that the transfer fans distributed the conditioned air sufficiently throughout the home in this configuration, and the heat pump set point could be raised, as long as humidity is addressed through other means (if it proved to be uncomfortable to occupants).

5.5.2.4 Occupied Cooling Period

The occupied cooling analysis period covers the entire cooling season of 2018: May 26, 2018–September 24, 2018.

Relative Humidity

ACCA defines acceptable RH limits to be between 25% and 55% during the cooling season. Still, the heat pump was unable to achieve low enough humidity levels to meet ACCA criteria in the occupied period. The southwest bedroom had the highest RH levels, and the northwest room had the lowest RH levels, but all rooms fell roughly between 45% and 75% RH for the majority of the time.

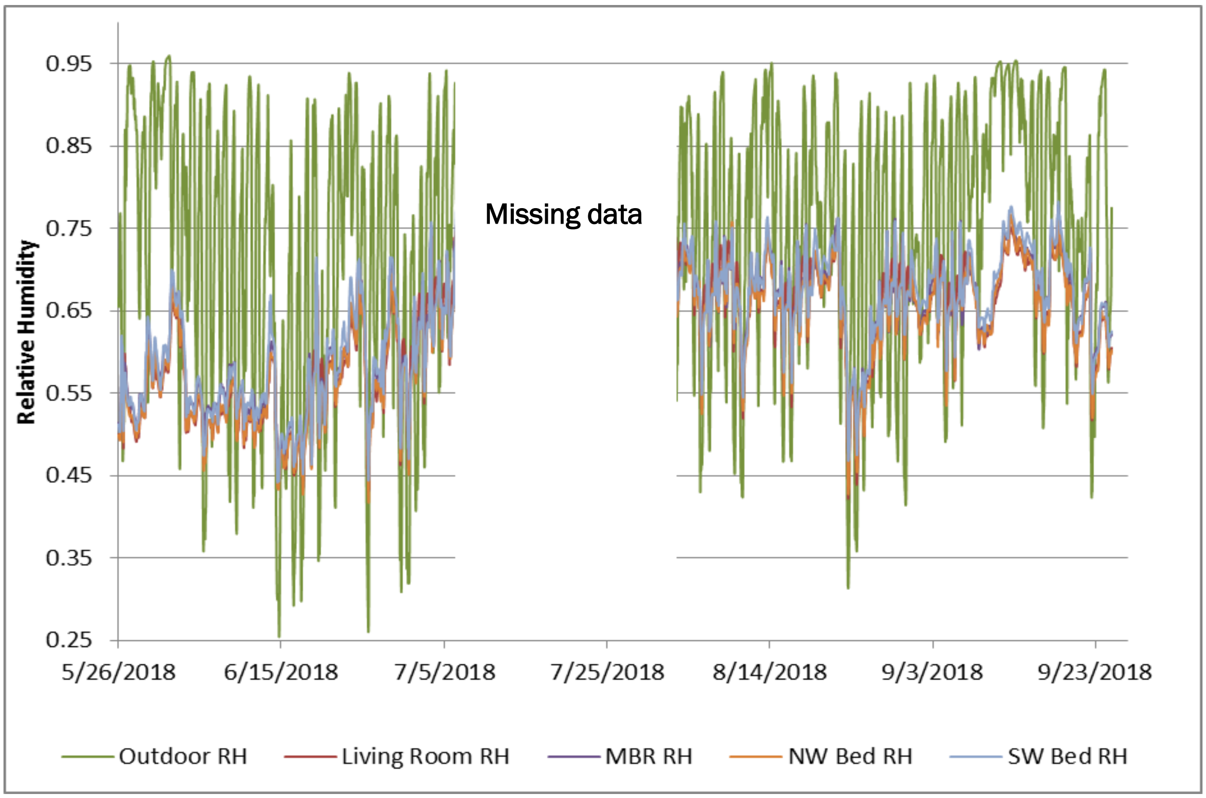


Figure 33. Occupied cooling season RH data

As shown in the following histograms, throughout the occupied cooling season (Figure 34), the RH was between 25% and 50% for 12% of time, and between 50% and 75% for 75% of time. When the heat pump was operating (Figure 35), 99% of time RH was under 75% but only 4% of time it was between 25% and 50%, which is the acceptable range defined by ACCA.

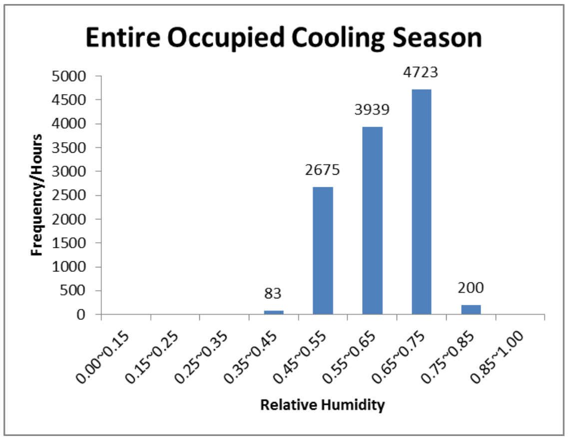


Figure 34. RH distribution of entire occupied cooling season

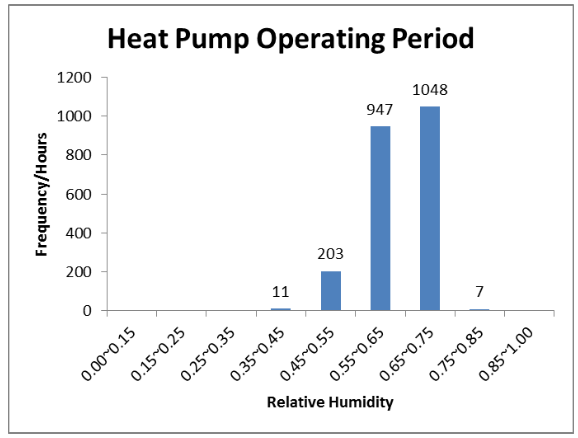


Figure 35. RH distribution of occupied cooling season with heat pump operating

Dry-Bulb Temperature at the Thermostat and Dry-Bulb Temperature in Any Conditioned Room

Set point temperature was not recorded. Because the heat pump unit was located at the living room, in this analysis bedroom temperatures are compared to the living room temperature instead of the set point temperature.

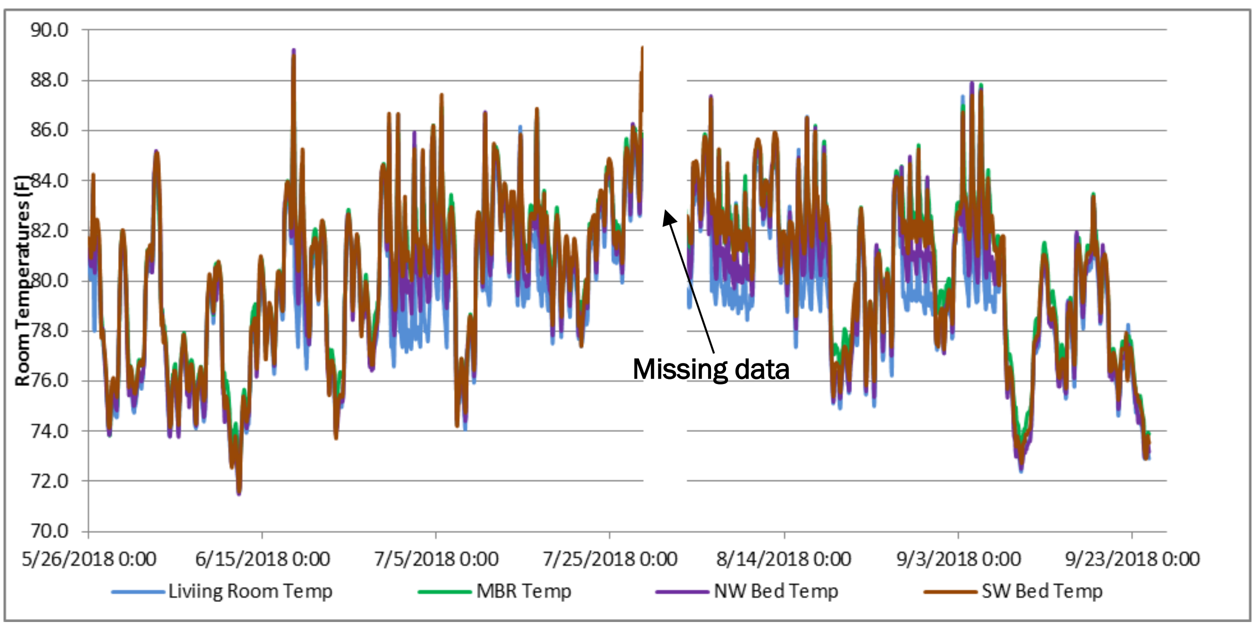


Figure 36. Occupied cooling season room temperatures

The master bedroom temperature ranged from 1.4°F below to 5°F above the living room temperature; the northwest bedroom temperature ranged from 1.0°F below to 3.4°F above the living room temperature; the southwest bedroom temperature ranged from 1.4°F below to 4.1°F above the living room temperature.

Distributions of room temperature differences are shown in the histograms in Figure 37, Figure 38, and Figure 39. For 72% of the time, the master bedroom temperature was within 3°F of the living room temperature; for 99% of the time, the northwest bedroom temperature was within 3°F of the living room temperature; and 66% of the time, the southwest bedroom temperature was within 3°F of the living room temperature. The northwest bedroom had the best performance because it is closest to the heat pump and because of the lack of solar heat gain in that room.

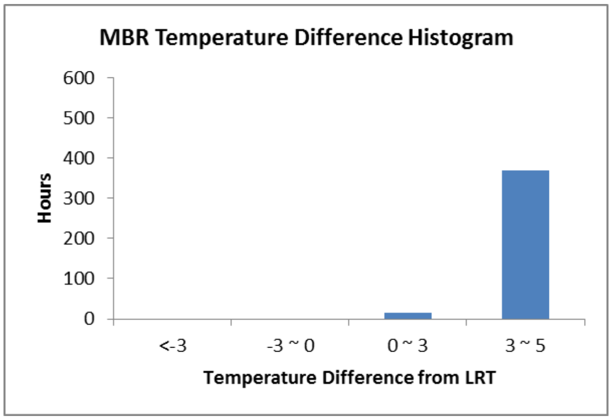


Figure 37. Master bedroom temperature difference

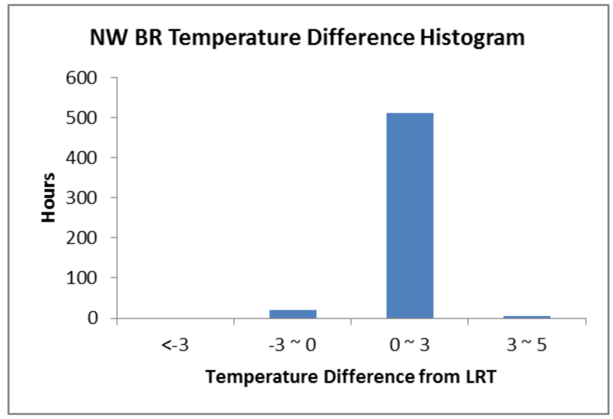


Figure 38. Northwest bedroom temperature difference

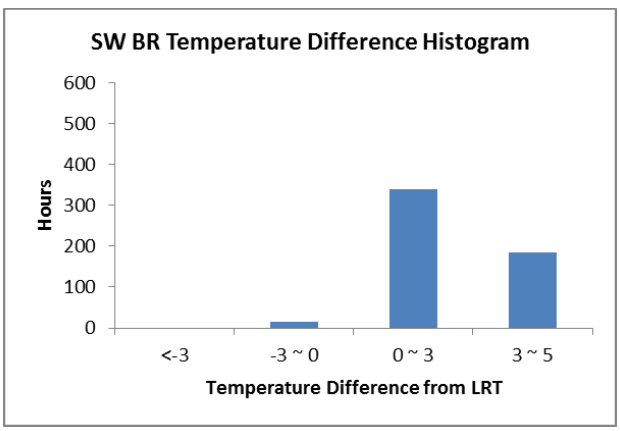


Figure 39. Southwest bedroom temperature difference

Room-to-Room Temperature Differences

When the heat pump was operating, the room-to-room temperature difference was less than 6°F 100% of the time. The temperature difference between the maximum room temperature and the minimum room temperature was consistently less than 4°F for 97% of the time, and the greatest difference was 5°F.

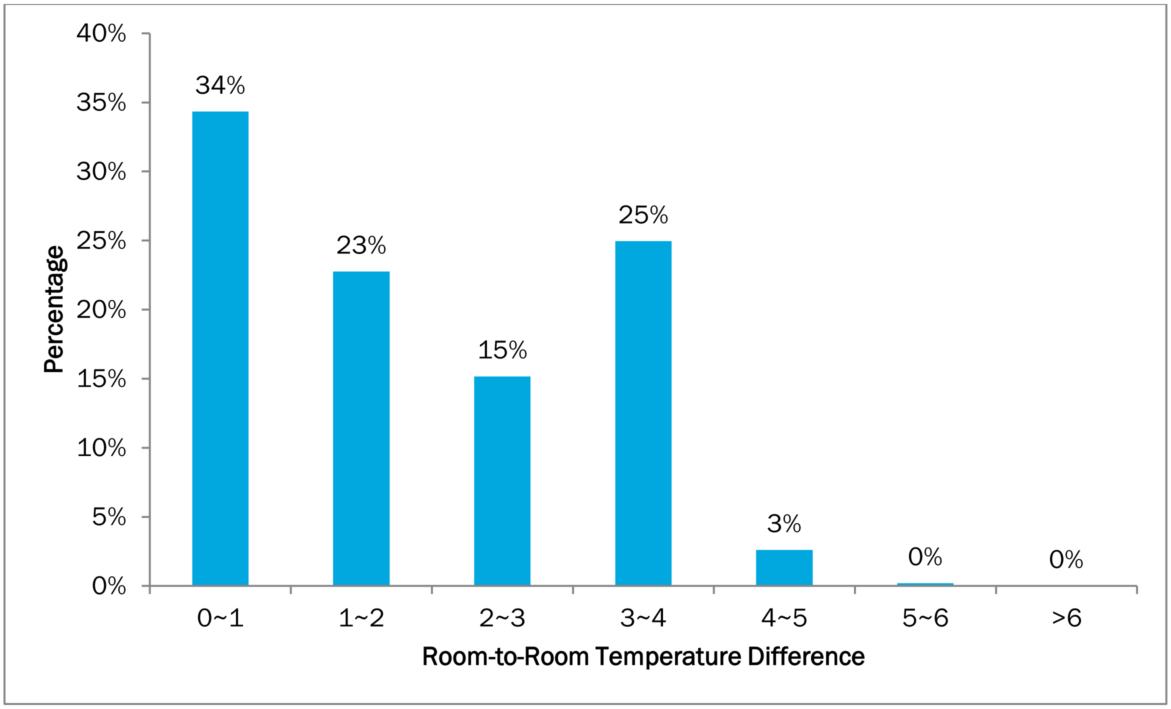


Figure 40. Room-to-room temperature difference histogram

5.5.2.5 ACCA Manual RS Comfort Compliance Summary

A summary of comfort compliance with respect to ACCA Manual RS during the simulated occupancy period is presented in Table 14, and the occupied period is presented in Table 15.

Table 14. ACCA Manual RS Comfort Compliance—Simulated Occupancy Period

Comfort Item	Season	Requirement	Summary
Relative Humidity	Heating	30% RH maximum	RH exceeded 30% about 11% of the time. The heat pump dehumidification mode was not active.
	Cooling	55% RH maximum	RH consistently ranged between 55% and 65%.
Dry-Bulb Temperature at the Thermostat	Heating	Set point temperature ±2°F	The temperature in the living room was within 2°F of the adjusted set point (re-calibrated to the realized room temperature)
	Cooling	Set point temperature ±3°F	The temperature in the living room was within 1°F of the heat pump set point, which meant it was 3°F –5°F below the ACCA value of 75°F. Adjusting the set point could bring it within compliance.
Dry-Bulb Temperature in Any Conditioned Room	Heating	Set point temperature ±2°F	74% compliance
	Cooling	Set point temperature ±3°F	If heat pump set point is increased to meet ACCA criteria, temperatures will likely remain within 3°F of this set point.
Room-to-Room Temperature Differences	Heating	4°F maximum	90% compliance
	Cooling	6°F maximum	Temperatures in all rooms were within 6°F of one another.
Floor Temperature	Heating	65°F minimum at 4-in. above the floor for 70°F thermostat setting	Met criteria 100% of time heating was operational.
	Cooling	N/A	

Table 15. ACCA Manual RS Comfort Compliance—Occupied Period

Comfort Item	Season	Requirement	Summary
Relative Humidity	Heating	30% RH maximum	RH exceeded 30% about half of the time.
	Cooling	55% RH maximum	RH consistently ranged between 55% and 75%. Occupants were satisfied with this.
Dry-Bulb Temperature at the Thermostat	Heating	Set point temperature ±2°F	Set point was not measured during the occupied period.
	Cooling	Set point temperature ±3°F	The temperature in the living room was within 2°F of the assumed 71/74°F set point for 86% of time (actual set point is unknown).
Dry-Bulb Temperature in Any Conditioned Room	Heating	Set point temperature ±2°F	Set point was not measured or recorded during the occupied period. The temperature in any conditioned room was within 2°F of the assumed 71/74°F set point for 80% of time (actual set point is unknown).
	Cooling	Set point temperature ±3°F	The temperature in any conditioned room was within 2°F of the assumed 71/74°F set point for 80% of time (actual set point is unknown).
Room-to-Room Temperature Differences	Heating	4°F maximum	Room-to-room temperature differences were within 4°F for 99.2% of time.
	Cooling	6°F maximum	Temperatures in all rooms were within 6°F of one another 100% of the time.
Floor Temperature	Heating	65°F minimum at 4-in. above the floor for 70°F thermostat setting	Floor temperatures were not measured during occupancy.
	Cooling	N/A	

5.5.3 Comfort—ASHRAE Standard 55

Project staff completed an analysis to assess comfort relative to ASHRAE Standard 55-2010 during the heating period and the cooling period. Table 15 summarizes the criteria and results of this analysis. Predicted mean vote (PMV) is used as the indicator for thermal comfort in this section. Equations from Fanger are used to calculate PMV of a group of subjects for a particular combination of air temperature, mean radiant temperature, RH, air speed, metabolic rate, and clothing insulation (Fanger 1970). PMV equal to zero represents thermal neutrality, and the comfort zone is defined by the combinations of these six parameters for which the PMV is within the recommended limits (-0.5<PMV<+0.5).

Table 16. ASHRAE 55 Thermal Comfort Compliance—Simulated Occupancy Period

	Allowable Limits	Heating Season Results	Cooling Season Results
Operative Temperature	Within the comfort zone defined by ASHRAE 55, -0.5<PMV<+0.5	94% compliance	100% compliance
Humidity Ratio	Below 0.012 as per humidity limits Section 5.2.2 of ASHRAE 55.	100% compliance	100% compliance
ASHRAE 55 Section 5.2.4.1: Local Discomfort Due to Radiant Temperature Asymmetry	Ceiling not allowed to be more than 5°C (9°F) warmer than other surfaces. Wall may be no more than 23°C (41°F) warmer than other surfaces. Ceiling not allowed to be more than 14°C (25.2°F) cooler than other surfaces. Wall may be no more than 10°C (18°F) cooler than other surfaces.	Surface temperatures not measured	
ASHRAE 55 Section 5.2.4.2: Local Discomfort Due to Draft	Generally, air speeds should be below 30 ft/min surrounding the body.	Transfer fan air speeds exceed 30 ft/min only within a few inches of ceilings and walls	
ASHRAE 55 Section 5.2.4.3: Local Discomfort Due to Vertical Air Temperature Difference	Not greater than 3°C (5.4°F) from ankle height to head height.	Maximum vertical difference measured was 3.3°F, or 1.8°C	Maximum vertical difference measured was 4°F, or 2.3°C
ASHRAE 55 Section 5.2.4.4: Local Discomfort Due to Floor Surface Temperature	Floor temperatures stay in the range of 19°–29°C (66°–84°F).	Surface temperatures not measured	

	Allowable Limits	Heating Season Results	Cooling Season Results
Cyclic Variations: Positive or Negative (Drifts/ Ramps)	In any 15-min period, up to 2°F change In any 30-min period, up to 3°F change In any 60-min period, up to 4°F change In any 120-min period, up to 5°F change In any 240-min period, up to 6°F change	Nearly 100% compliance, with exception discussed in Section 5.5.3.4	100% compliance

Table 17. ASHRAE 55 Thermal Comfort Compliance—Occupancy Period

	Allowable Limits	Heating Season Results	Cooling Season Results
Operative Temperature	Within the comfort zone defined by ASHRAE 55, -0.5<PMV<+0.5	95% compliance	57% compliance
Humidity Ratio	Below 0.012 as per Humidity limits Section 5.2.2 of ASHRAE 55.	100% compliance	100% compliance
ASHRAE 55 Section 5.2.4.1: Local Discomfort Due to Radiant Temperature Asymmetry	Ceiling not allowed to be more than 5°C (9°F) warmer than other surfaces. Wall may be no more than 23°C (41°F) warmer than other surfaces. Ceiling not allowed to be more than 14°C (25.2°F) cooler than other surfaces. Wall may be no more than 10°C (18°F) cooler than other surfaces.	Surface temperatures not measured	
ASHRAE 55 Section 5.2.4.2: Local Discomfort Due to Draft	Generally, air speeds should be below 30 ft/min surrounding the body.	Transfer fan air speeds not measured during occupancy	
ASHRAE 55 Section 5.2.4.3: Local Discomfort Due to Vertical Air Temperature Difference	Not greater than 3°C (5.4°F) from ankle height to head height.	Occupied measurements not made due to reductions in sensor points for occupancy	

	Allowable Limits	Heating Season Results	Cooling Season Results
ASHRAE 55 Section 5.2.4.4: Local Discomfort Due to Floor Surface Temperature	Floor temperatures stay in the range of 19°–29°C (66°–84°F).	Surface temperatures not measured	
Cyclic Variations: Positive or Negative (Drifts/ Ramps)	In any 15-min period, up to 2°F change In any 30-min period, up to 3°F change In any 60-min period, up to 4°F change In any 120-min period, up to 5°F change In any 240-min period, up to 6°F change	100% compliance	100% compliance

5.5.3.1 Operative Temperature and Humidity Ratio

The factors affecting operative temperature and humidity comfort under ASHRAE 55 are dry-bulb temperature, mean radiant temperature (MRT), RH, air speed, metabolic rate (met), and clothing value (clo). Average room air temperature data across the house and ambient temperature data were used to calculate surface temperatures throughout the house, which were then used to calculated MRT.

Simulated Occupancy Heating Period

For the heating season, a winter clothing value of 1.0 clo was used. A default metabolic rate of 1.1 met was used, as was an air speed of 0.15 meter per second. The MRT was approximated as a weighted average of interior surface temperatures, which were calculated based on indoor temperatures, outdoor temperatures, and conductance of the assembly. Conditions were examined on an hourly basis and were plotted on a psychrometric chart using the CBE Thermal Comfort tool (Hoyt 2013).

$$\mathrm{MRT} = (T_{wall}A_{wall} + T_{ceiling}A_{ceiling} + T_{floor}A_{floor} + T_{window\ glazing}A_{window\ glazing} + T_{window\ frame}A_{window\ frame} + T_{door}A_{door})/A_{total}$$

Equation 1. Mean radiant temperature

In which: MRT is the mean radiant temperature

T_{wall} is the interior temperature of the wall surface

T_{floor} is the interior temperature of the floor surface

$T_{window\ glazing}$ is the interior temperature of the window glazing surface

$T_{window\ frame}$ is the interior temperature of the window frame surface

T_{door} is the interior temperature of the door surface

A_{wall} is the area of the wall surface

A_{floor} is the area of the floor surface

$A_{window\ glazing}$ is the area of the window glazing surface

$A_{window\ frame}$ is the area of the window frame surface

A_{door} is the area of the door surface

A_{total} is the total area of interior surfaces.

Figure 41 overlays the range of conditions observed in the lab house during the monitoring period on the psychrometric chart along with the ASHRAE Standard 55-2010 compliance region. 94% of the hours were within compliance during the period. As per the CBE tool (Figure 42), as long as the interior room temperatures are within the range of 65.8°F–88.1°F and the average RH is above 30%, all occupants will experience thermal comfort. When MRT is higher, a lower interior dry-bulb temperature is required to maintain comfort.

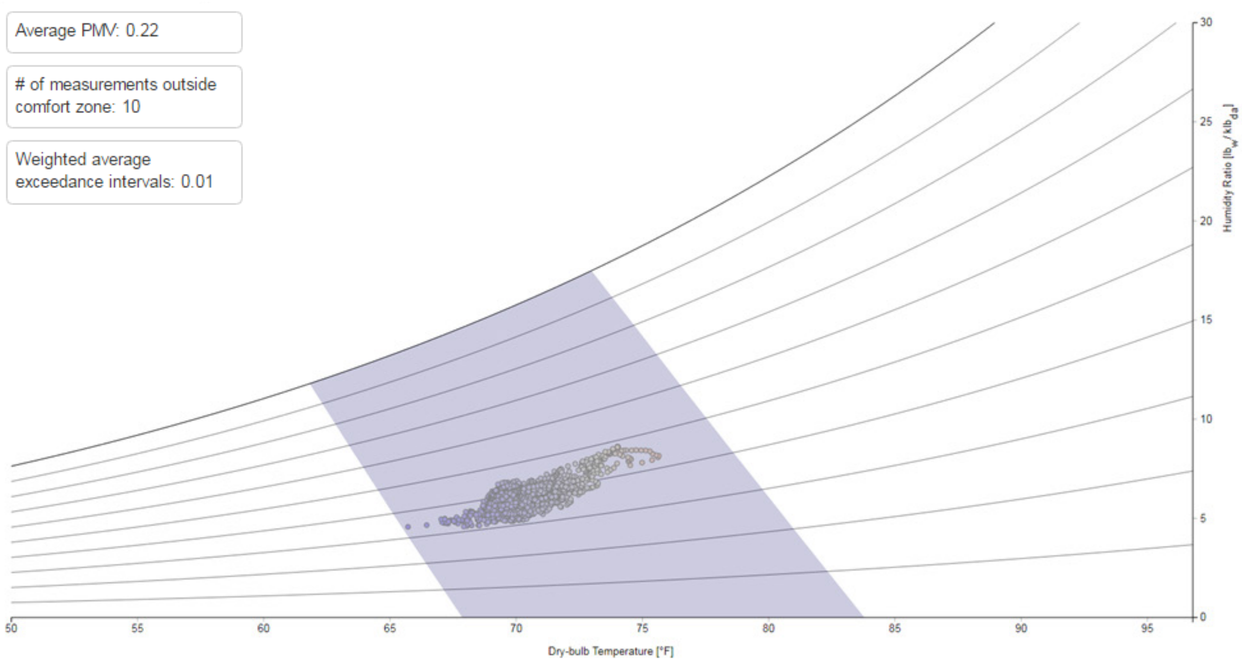


Figure 41. ASHRAE Standard 55 compliance—CBE thermal comfort tool (simulated occupancy heating period)

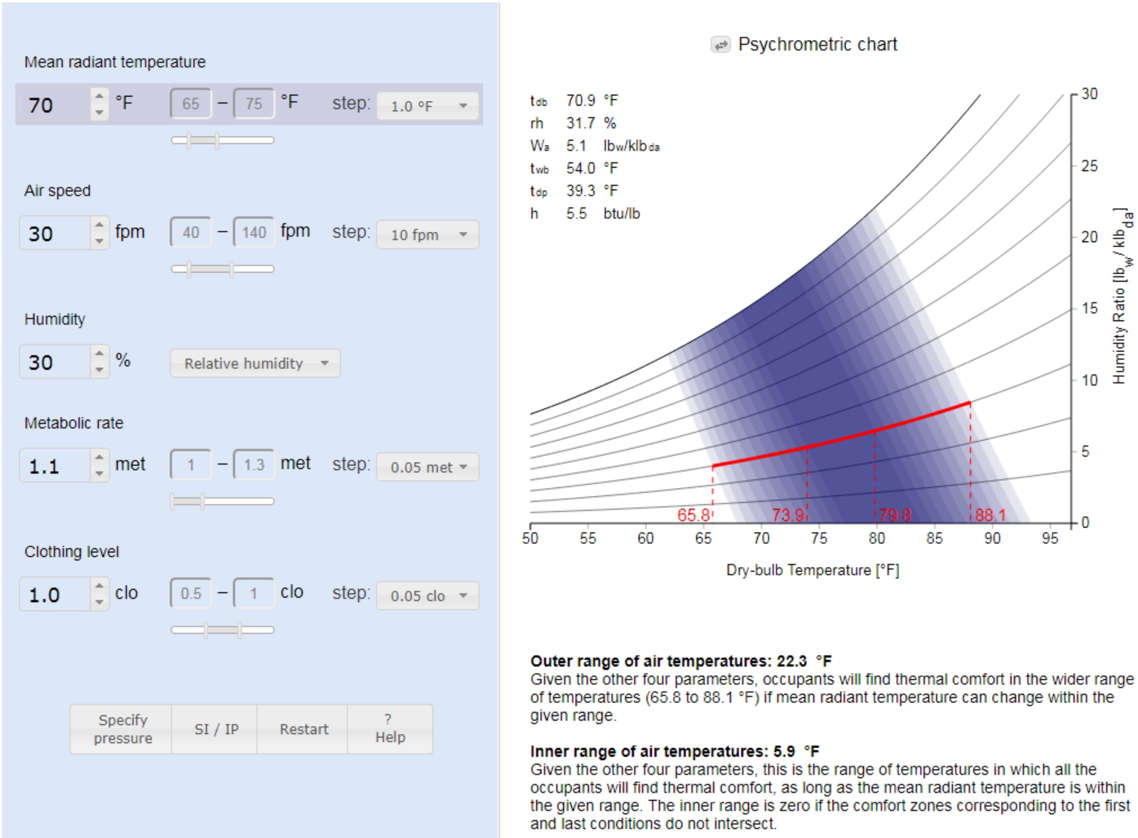


Figure 42. Range of dry-bulb temperatures that achieve ASHRAE 55-2010 thermal comfort (simulated occupancy heating period)

Occupied Heating Period

Figure 43 overlays the range of conditions observed in the house during the occupied monitoring period on the psychrometric chart along with the ASHRAE Standard 55-2010 compliance region. 95% of the hours were within compliance during the period. As per the CBE tool (Figure 44), as long as the interior room temperatures are within the range of 63.7°F–85.7°F and the average RH is above 30%, all occupants will experience thermal comfort.

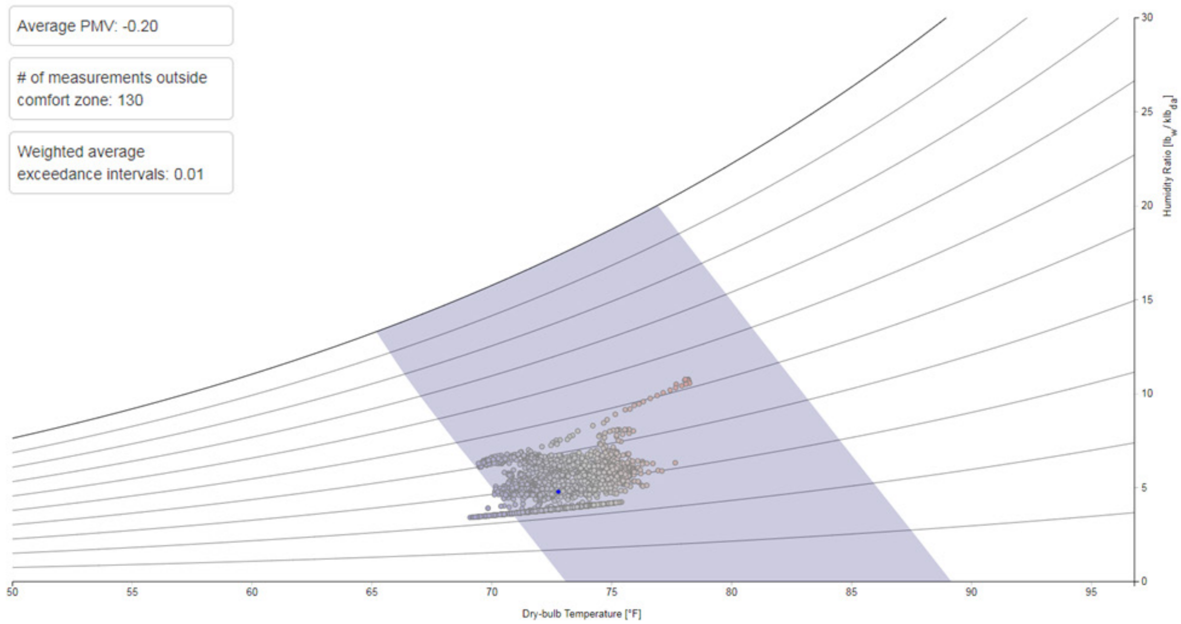


Figure 43. ASHRAE Standard 55 compliance—CBE thermal comfort tool (occupied heating period)

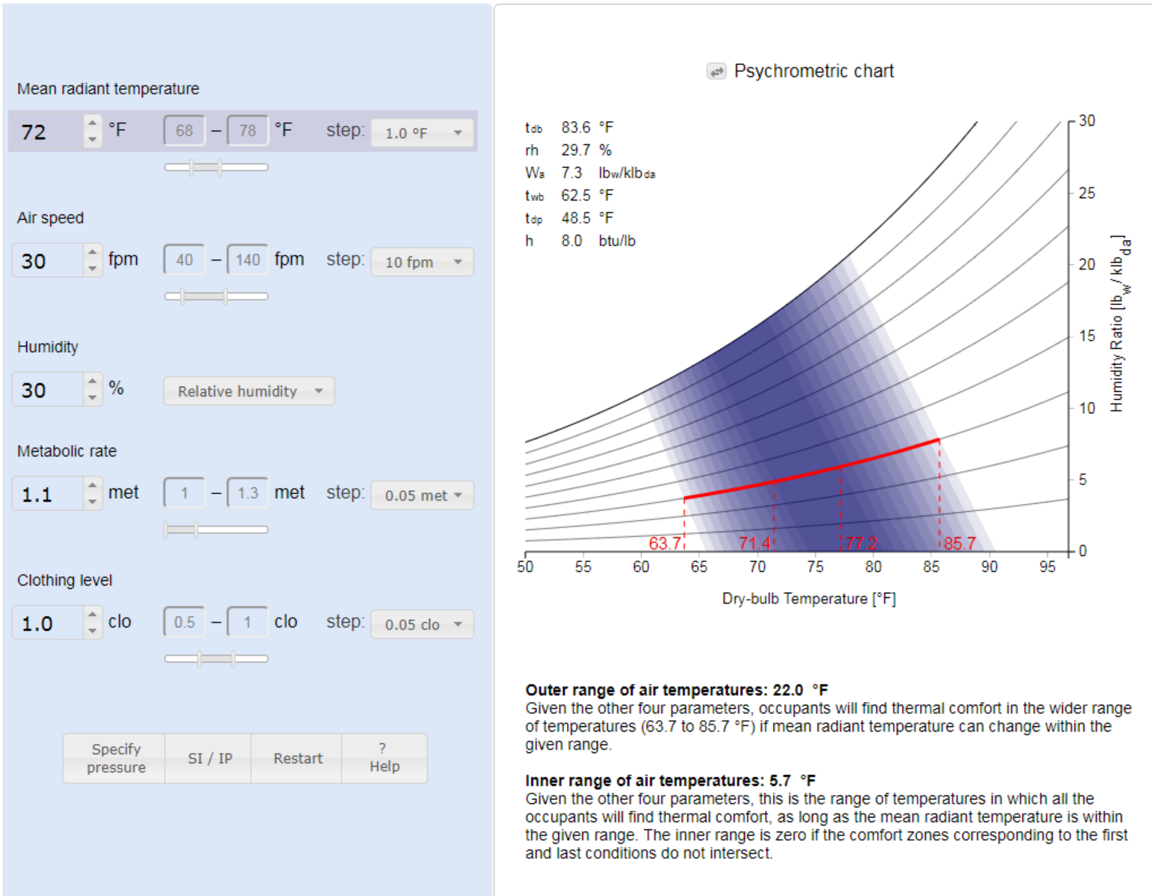


Figure 44. Range of dry-bulb temperatures that achieve ASHRAE 55-2010 thermal comfort (occupied heating period)

Simulated Occupancy Cooling Period

For the cooling season, a summer clothing value of 0.5 clo was used. The default metabolic rate of 1.1 met and air speed of 0.15 meters per second remained the same for this season.

Figure 45 shows that because the heat pump set point was so low, MRT fell below what is considered comfortable by ASHRAE for 100% of the time. PMV was almost -1, meaning that occupants would find the home “slightly cool” under defined conditions. Acceptable dry-bulb temperatures for the defined conditions range from about 74.5°F to 88.3°F, as shown in Figure 46. If humidity can be better controlled by the heat pump without reducing the set point, more measurements will fall within the ASHRAE-defined comfort range during the cooling months.

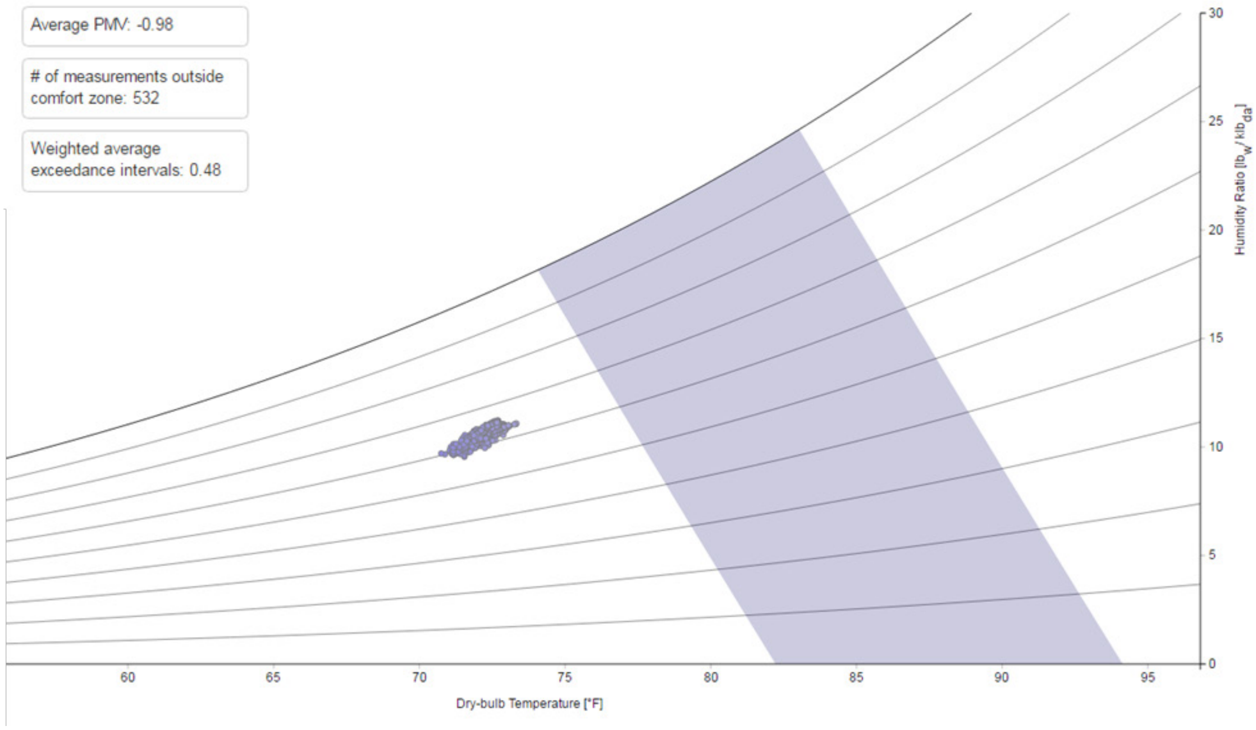


Figure 45. ASHRAE Standard 55 compliance—CBE thermal comfort tool (simulated occupancy cooling period)

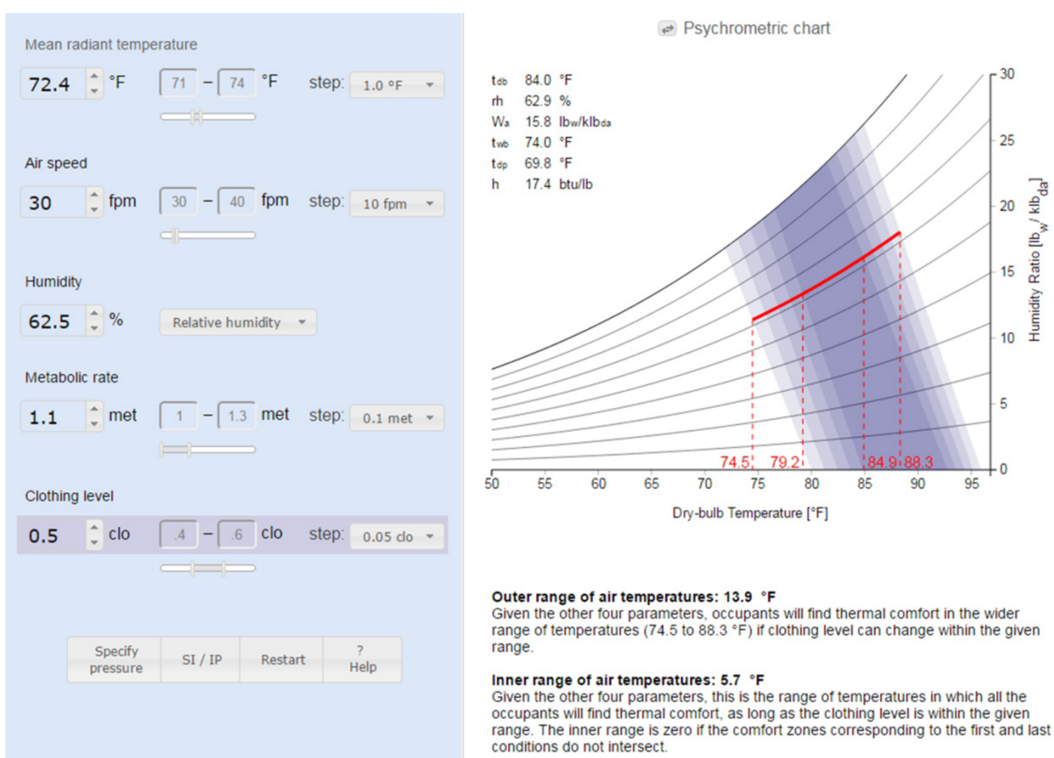


Figure 46. Range of dry-bulb temperatures that achieve ASHRAE 55-2010 thermal comfort (simulated occupancy cooling period)

Occupied Cooling Period

After 2017, the Center for the Built Environment Thermal Comfort Tool no longer provided the function to analyze thermal comfort using uploaded data. Therefore, R, a programming language and software environment for statistical computing, was used in this section, and no graphs are shown for the occupied cooling period. "Comf" is a package of functions for thermal comfort research using R (Schweiker et al. 2019). With air temperature, mean radiant temperature, air velocity, RH, clothing insulation level, and metabolic rate, it can calculate the PMV index and the percentage of compliance.

Based on the PMV calculation results using Comf, only 57% of the hours were within ASHRAE 55-2010 compliance during the period. The occupants kept the cooling set point higher than the thermal comfort zone defined by ASHRAE 55-2010. The occupants expressed satisfaction with cooling performance.

5.5.3.2 Draft

Air speeds due to transfer fans were measured with a hand-help anemometer in the bedrooms and master bathroom. The placement of the fans near the ceiling resulted in the high-speed air flowing within about 6 in. of the ceiling to the far wall. At the far wall, the air tended to draft down the wall, staying within a few inches of the wall. As a result, the high-speed air was generally outside of the habitable zone. Figure 17 shows the transfer fan and heat pump locations on the floor plan.

5.5.3.3 Vertical Air Temperature Difference Cooling—Simulated Occupancy

Dry-bulb temperatures were measured at about 6–10 in. from the floor and at about 5 ft from the floor in all rooms. The average and maximum temperature differences between the two measurements in each of six rooms are shown in Table 18.

Table 18. Vertical Temperature Differences (°F)

		Living	Kitchen	Master Bed	NW Bed	SW Bed	Master Bath
Heating Season	Average	0.2	1.8	0.1	0.2	0.2	0.1
	Maximum	1.0	3.3	0.1	0.5	0.5	0.2
Cooling Season	Average	1.2	0.7	0.1	0.0	0.0	-0.1
	Maximum	2.3	1.4	0.3	0.1	0.1	0.2

Because the transfer fans are located near the ceiling, it was especially important to monitor temperature stratification from ceiling to floor in order to assess both the drive of convection current between spaces and the effectiveness of the air distribution method to condition each space. The cooling season especially threatened the transfer fans' ability to maintain cooler temperatures in the remote rooms, because warmer temperatures at the ceiling could mean a reduced ability to distribute cool air.

Figure 48 shows that the rooms directly served by the heat pump (kitchen and living room) had higher levels of stratification, especially when ambient temperatures were high. This is almost certainly due to the heat pump supplying cool air directly into these rooms, which sinks to the floor. In contrast, the air introduced to the remote rooms via transfer fans at the ceiling is warmer and closer to the average remote room temperature after having mixed with the living room air.

Additionally, during cooling, the air coming from the living room and kitchen is warmer than the air leaving the heat pump, so the air passing through the transfer fans does not cause as much temperature stratification in these remote rooms. Recall that temperatures remained within acceptable limits during the entire cooling season in most rooms, so even though the transfer fans do not deliver air as cold as seen by the living room and kitchen, the air distribution was sufficient in the bedrooms to be considered comfortable.

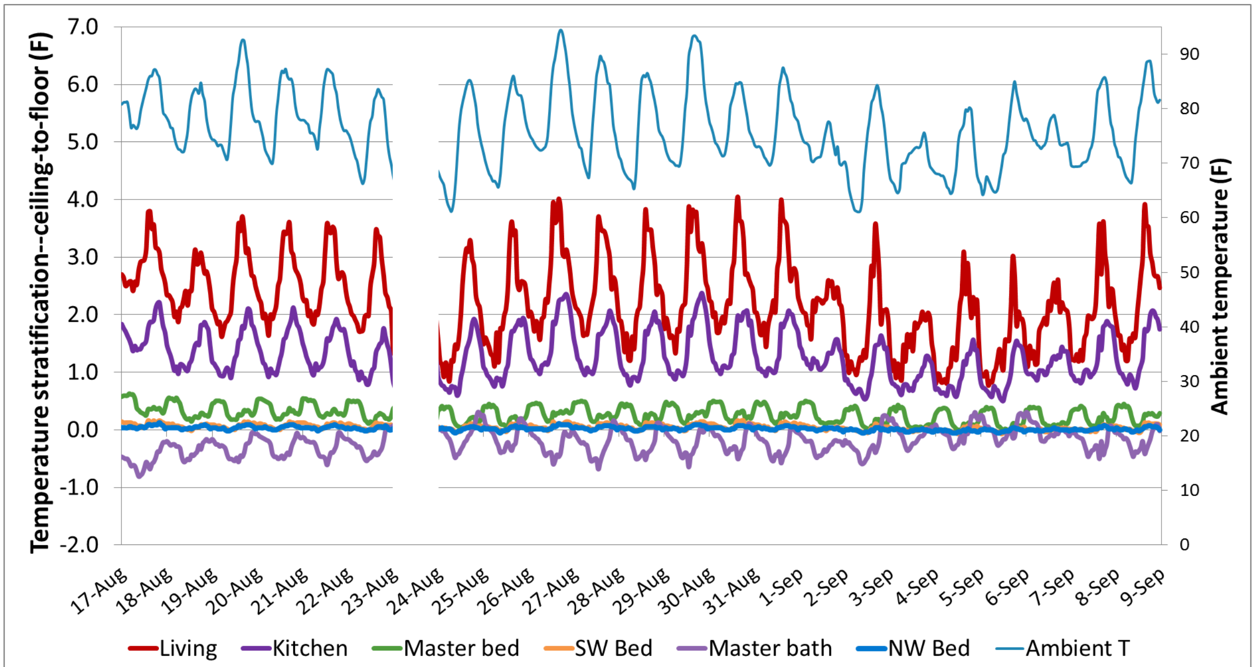


Figure 47. Cooling season temperature stratification

5.5.3.4 Cyclic Variations: Positive or Negative (Drifts/Ramps)

Simulated Occupancy Heating Season

The percent of time that each room passed the cyclic variations criteria during the heating period is shown in Table 19. Most rooms complied fully for all criteria. The only significant exception was the living room, which had only 81% compliance for the 15-minute criteria. Because this is the room with the heat pump, the cause is most likely the deadband on the heat pump controls and/or the “smart” control feature that implements 2°–4°F setbacks when the infrared motion sensor fails to detect activity within certain periods of time.

Table 19. Cyclic Variations Passing Rate—Simulated Occupancy Heating Season

Criteria	Living	Kitchen	Master Bed	NW Bed	SW Bed	Master Bath
15-min, max 2°F change	81%	96%	100%	100%	100%	100%
30-min, max 3°F change	97%	100%	100%	100%	100%	100%
60-min, max 4°F change	100%	100%	100%	100%	100%	100%
120-min, max 5°F change	100%	100%	100%	100%	100%	100%
240-min, max 6°F change	100%	100%	100%	100%	100%	100%

Occupied Heating Season

The percentage of time that each room passed the cyclic variations criteria during the heating period was 100%.

Simulated Occupancy and Occupied Cooling Season

Temperatures in all rooms in the home complied with ASHRAE 55 cyclic-variation requirements 100% of the time. In the cooling season, air distribution appeared more effective, likely due to a tighter deadband from deactivating the smart-control feature.

5.5.4 Building Cavity Moisture Measurements

Two OmniSense S900-1 moisture sensors were installed in north-facing exterior wall cavities. One was mounted to the inside of the oriented strand board (OSB) sheathing at the northwest bedroom; the other was mounted to the bottom of a wall stud behind the pantry closet. A third sensor was mounted to the underside of the roof sheathing near the center of the home at the ridge. The primary question of interest was whether the wood moisture content became elevated due to the lower permeability of the insulating foam sheathing. Also of interest was the RH in the cavities and whether the air temperature in the cavities reached dew point.

Two methods were used to determine whether prolonged mold growth could occur. The first method is percent moisture content (% MC) of the wood. Literature on mold growth risk indicates that below 20% MC, wood (including OSB) will not support mold growth, and even higher levels are required for decay to set in (Morris).

The second method used to determine conditions of mold growth was ASHRAE Standard 160 (2016). This standard describes the air temperature and RH thresholds under which mold growth can occur. ASHRAE 160 describes that in hygrothermal conditions with RH above 80% and temperature between 41°F and 104°F, mold growth can be supported; however, this standard has been criticized as overly conservative (Lstiburek et al. 2016).

Table 20 summarizes the data over the entire monitoring period (simulated and actual occupancy) from each of the sensors as well as the relevant danger thresholds. None of the locations exceeded the 20% threshold for OSB moisture content. Also, temperature at the surface of each of the assemblies did not reach its corresponding dew point temperature for any hour. The bedroom sheathing and the roof sheathing had some hours where the RH at the OSB surface exceeded 80%, which could promote microbial growth. The attic sheathing time was minimal, but the northwest bedroom sheathing exceeded 80% for 17.5% of the time. On a 30-day running average basis and limiting it to periods where the wall cavity air temperature was between 41°F and 104°F, the occurrence was 12.5% of the time. However, because OSB moisture content and dew point were not a concern, no assembly can be considered a failure in this test.

Table 20. Building Cavity Moisture Measurements Summary

	Bottom Stud, Wall Behind Pantry	Northwest Bedroom Sheathing	Roof Sheathing Near Ridge	Danger Threshold
Max wood moisture content %	10.3	12.9	10.3	20%
% time relative humidity exceed 80%	0	17.5	1.5	80%
Air temperature – dew point temperature (minimum (F) and % time less than 2°F)	7.3, 0%	1.0, 0.02%	1.8, 0.04%	0 (temperature meets dew point)

5.6 Occupant Response

At the end of the monitoring period, an occupant interview was conducted, focusing on thermal comfort, indoor air quality, energy costs, maintenance of energy, and ventilation-related equipment. Responses are discussed in this section.

- Compared to their previous home, the occupant was impressed with the overall thermal comfort of the house during winter, summer, and swing seasons.
- The occupant described the evenness of temperature across the various rooms in the house as good and mentioned the north room was a little colder, which is consistent with the monitored data.
- According to the occupant, they never experienced high humidity inside the home.
- The occupant was most pleased with the way the house was put together and least pleased with the fact that the washroom was small.
- The occupant thought the utility bills for the home were better than he expected.

- The occupant was 80% satisfied with the heating system because it was cooler in the bedrooms and mentioned hot water would sometimes get cold but overall was good. The occupant was aware of and very satisfied with all other features.
- The occupant was satisfied with the durability and maintenance of the house and had no issue with street noise and transmission of odors between rooms.
- According to the occupant, the set point was 76°F in the summer and 72°F in the winter. He also set back the thermostat when the home was unoccupied, and backup heaters were never needed. These are consistent with observations and measurements of the monitoring periods.

The list of interview questions is in Appendix D.

5.7 New Jersey Home Conclusions

Based on these results, the test house met some, but not all, defined objectives. Space-conditioning source-energy savings was greater than 50% compared to the 2009 IECC, based on field data and modeling when a weighted average of electric and natural gas heated homes is considered for the baseline (new manufactured homes are roughly split between electric and fossil fuel heating). Furthermore, comfort compliance (per ACCA Manual RS and ASHRAE 55) in heating mode can be achieved with the limited use of supplemental electric resistance heating as planned in the design. The amount of resistance heating has been factored into the modeled energy savings.

Cooling season data show potential for the home to comply with ACCA Manual RS in terms of maintaining set point temperature, temperature variation across rooms, and temperature stratification, but meeting the RH criteria is more difficult. Using a different heat pump setup (with lower total cooling capacity, multiple indoor units, or heat pumps with humidity control technology such as dual/split coils) or otherwise dehumidifying would better address RH levels and thus allow for an increase in thermostat set point, albeit at an increase in cost and some energy penalty. The impacts of this issue are highly climate dependent.

At the set point used, the home fell outside of ASHRAE 55 comfort bounds at 0.5 clo in the summer and would be considered slightly cool. Adjusting the set point as discussed would likely bring it within compliance. Nevertheless, temperatures were relatively stable, and conditioned air was effectively distributed to all rooms, meaning that the design approach shows promise in meeting comfort criteria with relatively minor adjustments.

6 Massachusetts Habitat Home—Occupied Testing

6.1 Design and Construction

The 1,044-ft^2 site-built Massachusetts home had an end-loaded living area with bedrooms and bathrooms accessed from a short hallway (Figure 49). This is a more challenging design to serve with a point-source heat pump because conditioned air has to be transported along the length of the house through longer transfer ducts. Other major differences between the Massachusetts home and New Jersey home include the use of an ERV, raised slab-on-grade foundation, heat pump water heater, and somewhat higher envelope specification to account for the colder climate (Table 20).

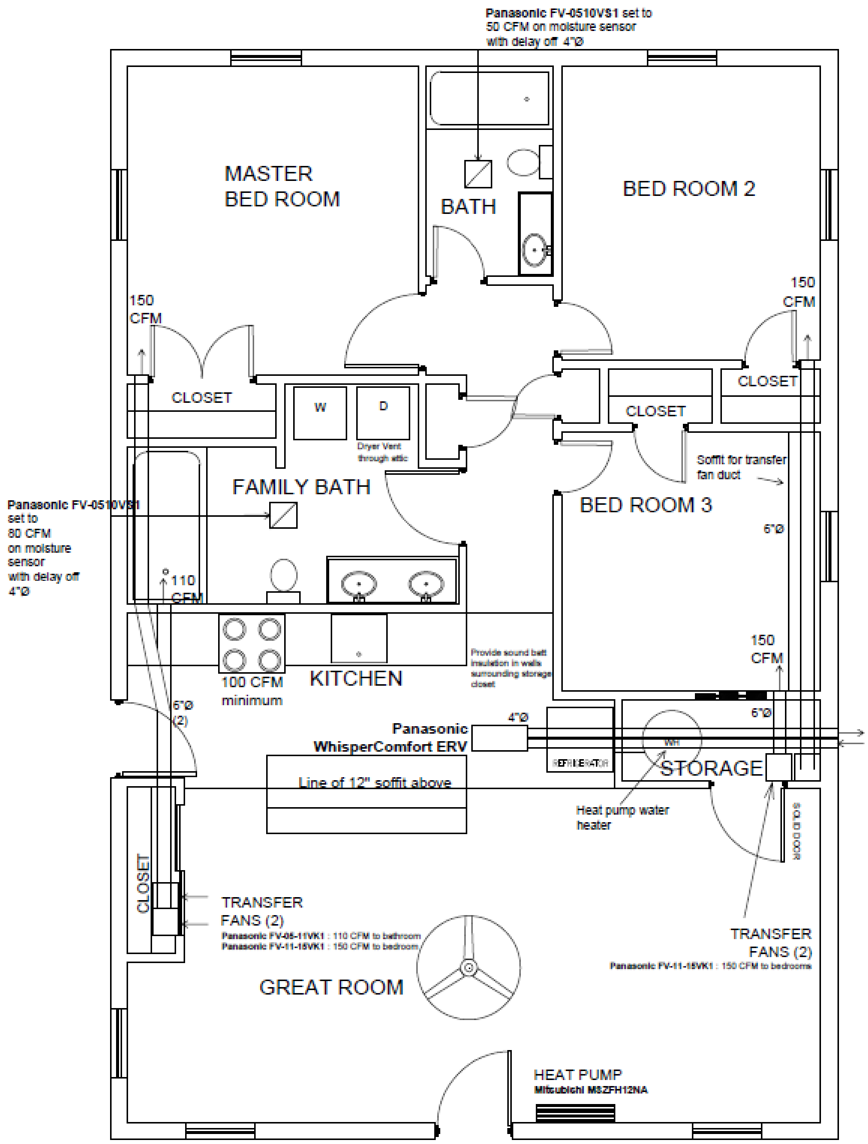


Figure 48. Massachusetts home floor plan (28 ft x 41 ft overall)

Table 21. Massachusetts Habitat Specifications

	Item	Details
Space Conditioning, Ventilation	Heat pump	Mitsubishi MSZFH12NA; fan coil unit above front door in living room
	Backup heating	Original design called for wiring for electric resistance supplemental heat in bedrooms and baths but not installing them. If installed, the backup heaters would be connected to a separate electrical subpanel that would be disabled by a relay when an outdoor temperature sensor measured above an outdoor cutoff set point. Furthermore, each heater was to be controlled by an occupancy-sensing thermostat to prevent unneeded use and fixed to a upper limit set point. These features were not installed by the builder; instead, baseboard heaters were installed in bedrooms and living room, and a wall heater with timer was installed in the master bath.
	Transfer fans	3 – Panasonic FV-11-15VK1 – 150 cfm (bedrooms) 1 – Panasonic FV-05-11VK1 – 110 cfm (family bath) Original intent was for fans to be connected to individual switches located in mechanical closet; however, the builder hardwired them without switches.
	Whole-house ventilation	Panasonic Whisper Comfort ERV (tested at 27 cfm supply); supplemented by continuous bath fan. Installed in soffit near kitchen. Original intent was to connect to switch in mechanical closet, but the builder hardwired the ERV without a switch.
	Bathroom ventilation	Family bath: FV-0510VS1 WhisperValue DC set to 80 cfm on moisture sensor with delay-off (tested at 80 cfm) Hall bath: FV-0510VS1 WhisperValue DC set to 50 cfm on moisture sensor with delay-off (tested at 59 cfm)
	Kitchen ventilation	Range hood exhausted to outside
	Ceiling fan	1 – ENERGY STAR fan in living room
Water Heating	Water heater	Heat pump water heater; Rheem Performance Platinum. Original intent was to use duct kit to direct exhaust through wall behind/above refrigerator, but duct kit was not used. Because of comfort complaints (cold air flowing from mechanical closet door into living area), exhaust/supply ducts were installed through ceiling into the attic.
Envelope	Floor	4-in. concrete slab with 2-in. XPS (R-10) under-slab and 2-in. XPS (R-10) slab edge insulation
	Ceiling	19.4" cellulose insulation in the attic (R-70); trusses
	Walls	R-21 cellulose (5.5" thick) in 2x6 wall; exterior sheathing: 2" XPS, R-10
	Windows	U: 0.18, SHGC: 0.31; triple pane, low-E
	Exterior doors	U: 0.20 fiberglass
	Attic ventilation	Yes
	Envelope leakage	Prescriptive measures per ENERGY STAR requirements; test result 2.53 ACH_{50}

Figure 49. Framing outside (left), inside (right) showing OSB ceiling air barrier sealed with tape

Figure 50. Blown-in cellulose insulation at Grade I level quality

Figure 51. Ventilation and transfer fan ductwork

Figure 52. Transfer fan duct; an air barrier enclosed the cellulose within the wall cavity and drywall covered the soffit

Figure 53. Insulated attic; heat pump outdoor unit

Figure 54. Completed Massachusetts home

6.2 Results and Analysis

The Massachusetts home was instrumented for data collection in December 2017 and occupied by a family of five shortly thereafter. Monitoring continued through May of 2019. Problems with the initial heat pump installation rendered data from the 2017–2018 winter unusable, so 2018–2019 winter data were used for analysis.

6.2.1 Space Conditioning and Ventilation Energy

6.2.1.1 Energy Model Comparison—Massachusetts

BEopt models were prepared for two baseline code (ICC 2009) homes: all electric and gas furnace. The specifications for the as-built house are shown in Table 20, and the baseline code house specifications are the same as those in Table 10 (except for wall insulation which in IECC climate zone 5 is R-20). The two versions of the as-built house represent a best-case scenario with no supplemental resistance heating and a case with supplemental resistance heating.

BEopt has no built-in function capable of modeling the impact of the transfer fans on the conditioned space and cannot simulate energy usage of supplemental resistance heaters when used in combination with a primary space-conditioning system such as a ductless heat pump. The Massachusetts home had no unoccupied monitoring data available from which to generate projections. Therefore, a series of room-by-room heating load calculations was conducted to estimate the annual supplemental resistance heat needed by room for input into the model (see Appendix F for calculation methodology).

Results are shown in Figure 55. The models predict the as-built house will use 66%–74% less source space-conditioning energy than the baseline all-electric home and 27%–45% less than the baseline gas-heated home, depending on whether compared to the best case or supplemental heating case model.

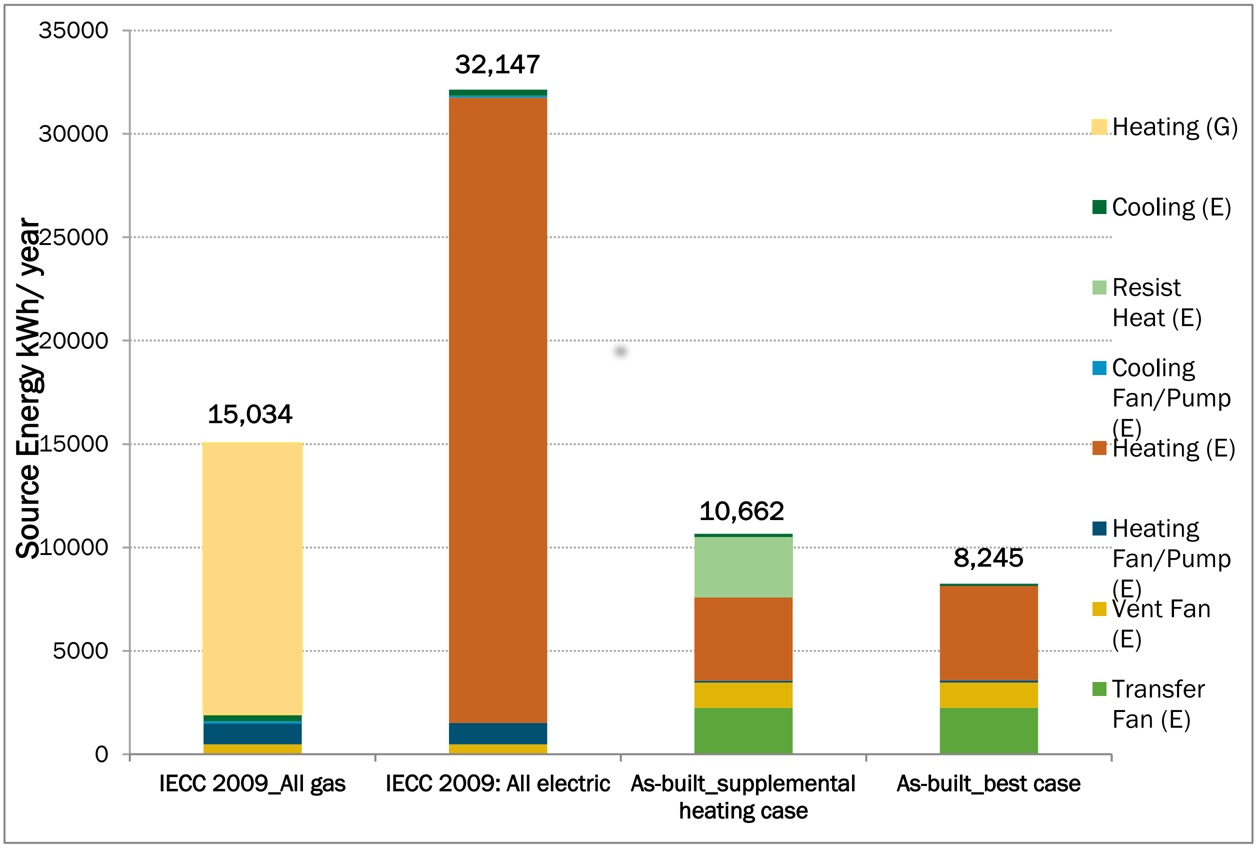


Figure 55. Massachusetts home modeled energy comparison

6.2.1.2 Occupied Energy Consumption—Massachusetts

Energy consumption data were collected from the occupied home during the entire one-year-long period of May 31, 2018, through May 30, 2019. The electric energy consumption was then scaled by a 2.80 source-to-site energy use factor and is shown in Table 21.

Table 22. Monitored Energy Results—Massachusetts

Source	Occupied Annual Source Energy in kWh
Heating (heat pump)	4,769
Supplemental heating	2,727
Cooling	2,783
Ventilation fan energy	471
Transfer fan energy	2,165
Total space conditioning and ventilation energy	12,916
Domestic water heating energy	7,014
Total house energy	42,786

6.2.2 Comfort—ACCA Manual RS

The ACCA Manual RS criteria were separated into heating and cooling and are presented separately in this section, with cooling following heating. Because the thermostat set point was controlled by the occupants and unknown, dry-bulb temperature at the thermostat was not evaluated.

6.2.2.1 Heating Season

6.2.2.1.1 Relative Humidity

In the heating season (October 1, 2018–May 30, 2019), ACCA defines acceptable RH limits to be between 20% and 30%. However, relative humidity of all conditioned rooms was higher than the desired level. It was only within the 20%–30% range between 0.1% and 4.7% of the time. RH in all rooms was frequently between 30% and 50% (Figure 57).

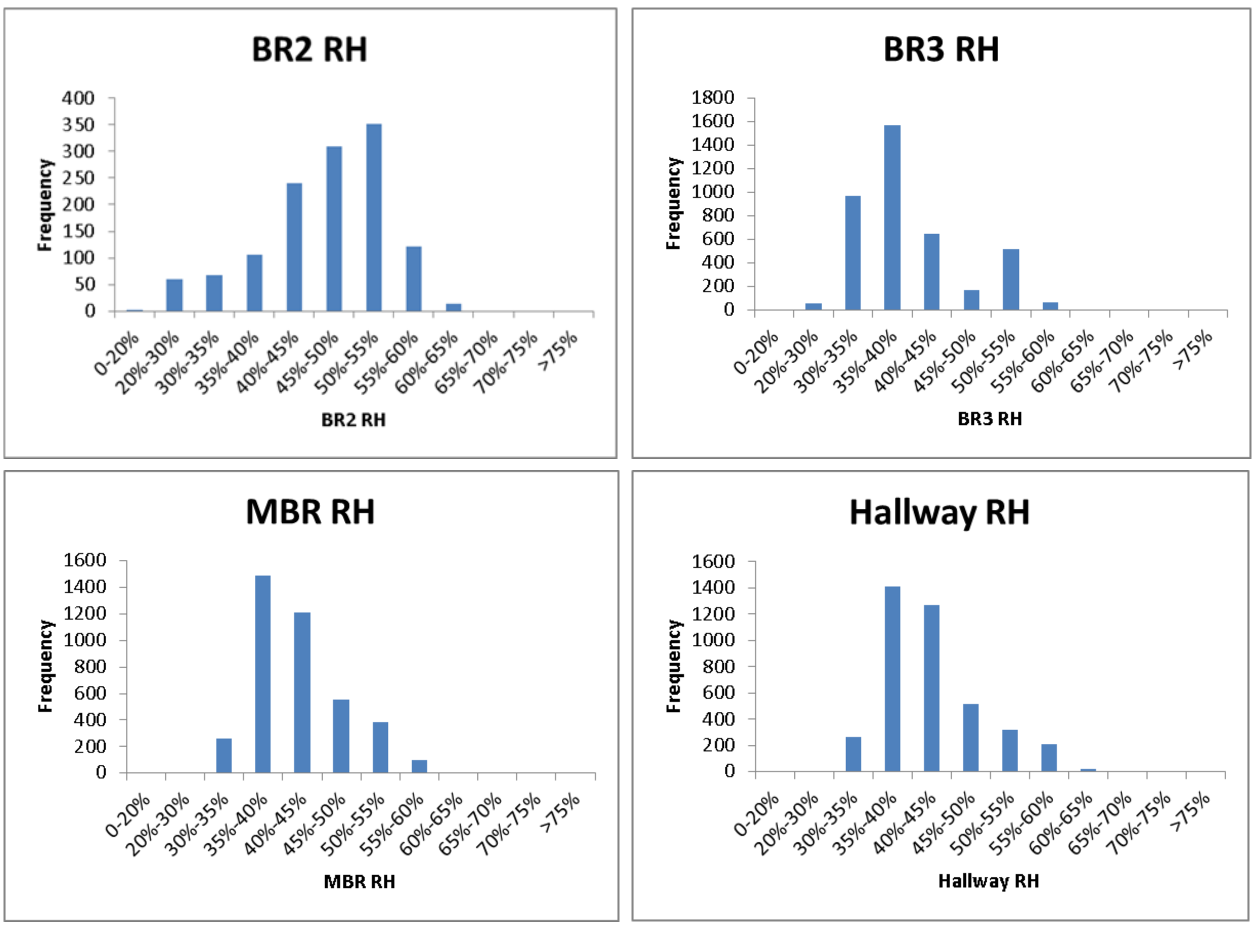


Figure 56. Massachusetts home heating season room relative humidity during heating season when heat pump was operating

6.2.2.1.2 Dry-Bulb Temperature at the Thermostat and Dry-Bulb Temperature in Any Conditioned Room

The dry-bulb temperature of all the conditioned rooms was constantly between 72°F and 78°F. For the purpose of this study, temperatures in conditioned rooms were compared with the dry-bulb temperature of the hallway, where the thermostat was located (the heat pump used a wall-mounted thermostat rather than a hand-held remote). For 99% of the time, the master bedroom temperature was within 2°F of the hall temperature, while Bedroom 2 and Bedroom 3 only achieved 57% and 87% compliance, respectively (Figure 58).

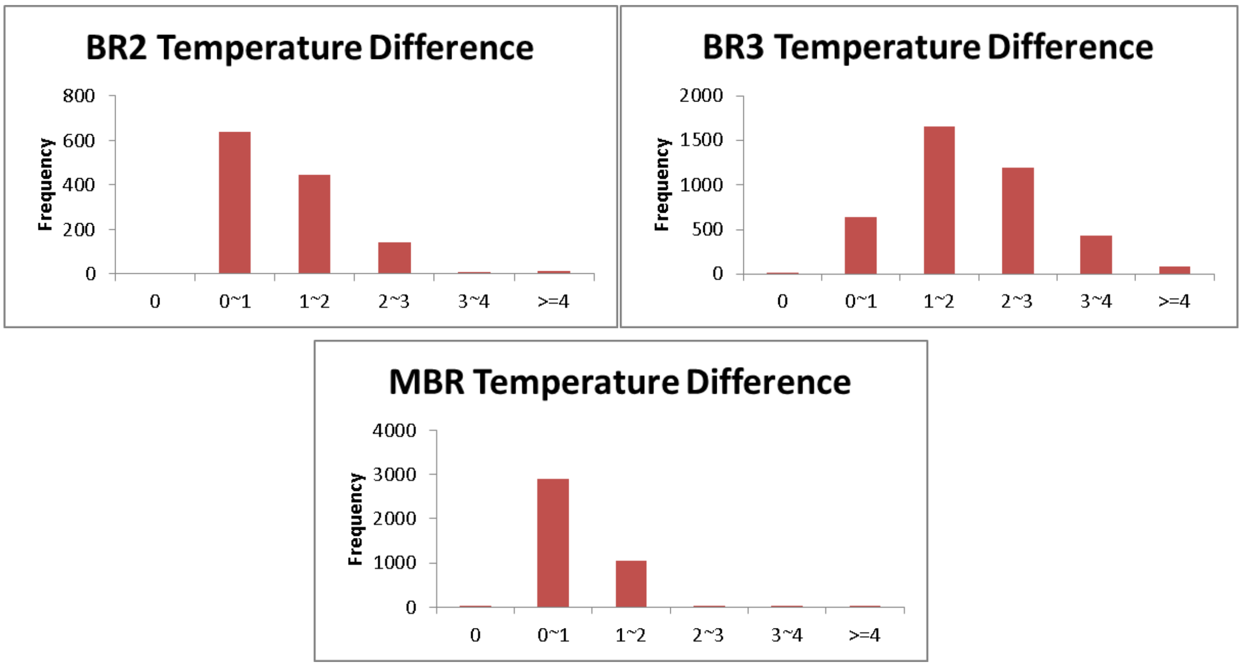


Figure 57. Massachusetts home heating season room temperature relative to the thermostat location when heat pump was operating

6.2.2.1.3 Room-to-Room Temperature Differences

The maximum room-to-room temperature difference is the difference between the highest temperature of any room and the lowest temperature of any room at any given point in time. In the heating season, room-to-room temperature difference was less than 4°F 85% of the time (Figure 58).

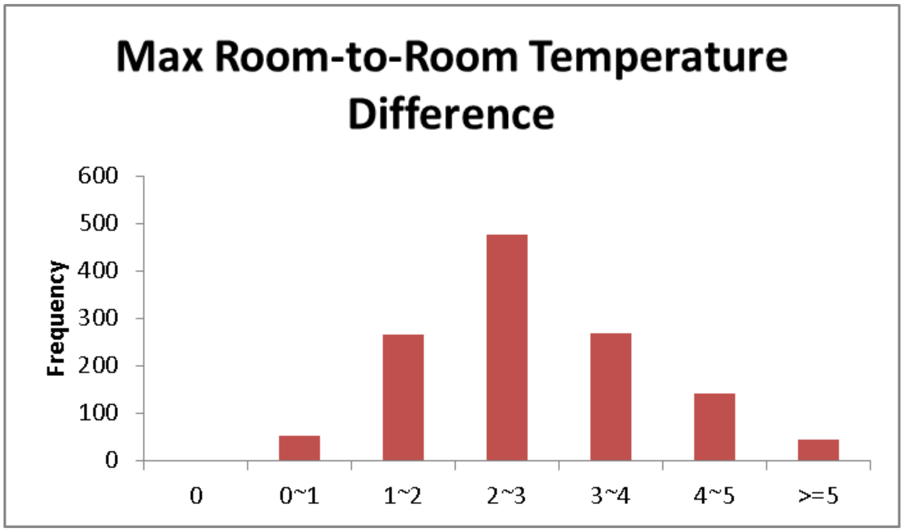


Figure 58. Massachusetts home heating season maximum room-to-room temperature difference when heat pump was operating

6.2.2.2 Cooling Season

6.2.2.2.1 Relative Humidity

Cooling season data from May 31, 2018, to September 30, 2018 was analyzed. ACCA defines acceptable RH limits to be between 25% and 55% during the cooling season. Both bedroom 2 and the master bedroom had RH below 55% for 93% of time when the heat pump was operating. The hallway and BR3 had higher RH levels and were compliant with the ACCA requirement 60% of time when the heat pump was on (Figure 59).

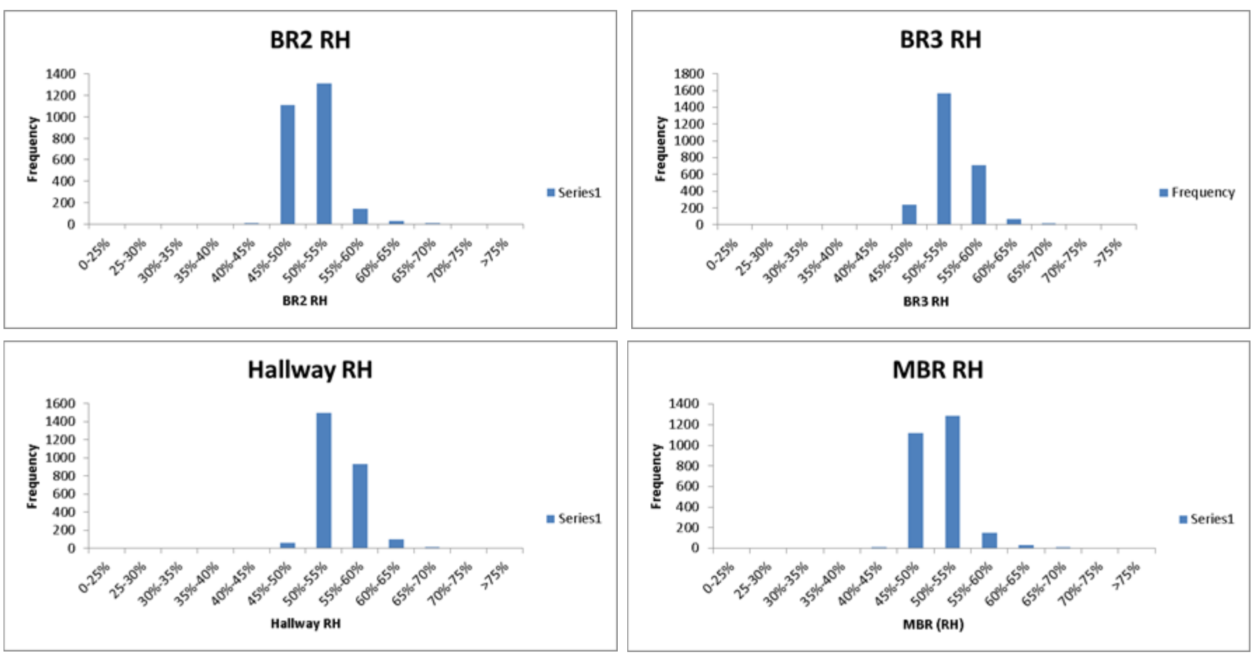


Figure 59. Massachusetts home cooling season relative humidity when heat pump was operating

6.2.2.2.2 Dry-Bulb Temperature at the Thermostat and Dry-Bulb Temperature in Any Conditioned Room

The dry-bulb temperature of all the conditioned rooms was generally between 68° and 75°F, with occasional excursions to 76° or 77°F (Figure 60). The thermostat was located in the hallway; therefore, for the purpose of this study temperatures in conditioned rooms are compared to the hallway temperature. Bedroom 2 and 3 temperatures were within ±3°F of the hallway temperature 100% of the time (Figure 61). The temperature difference between the master bedroom and the hallway was larger and within compliance only 74% of the time.

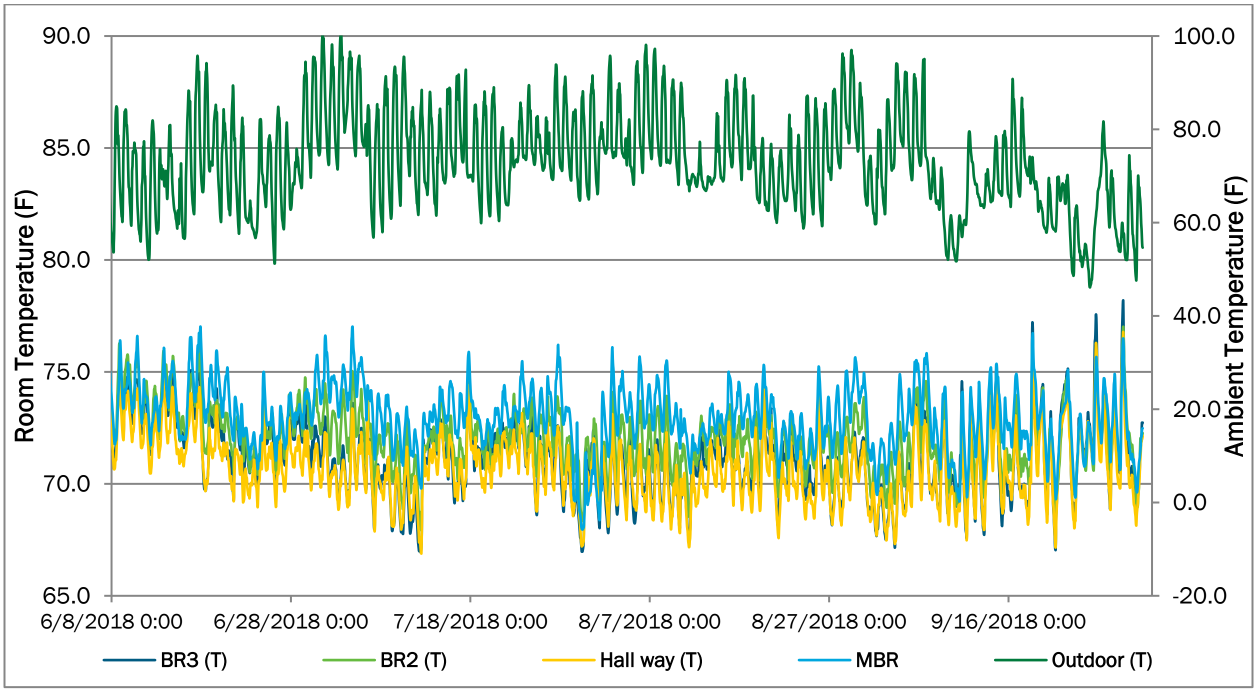


Figure 60. Massachusetts home cooling season room temperatures when heat pump was operating

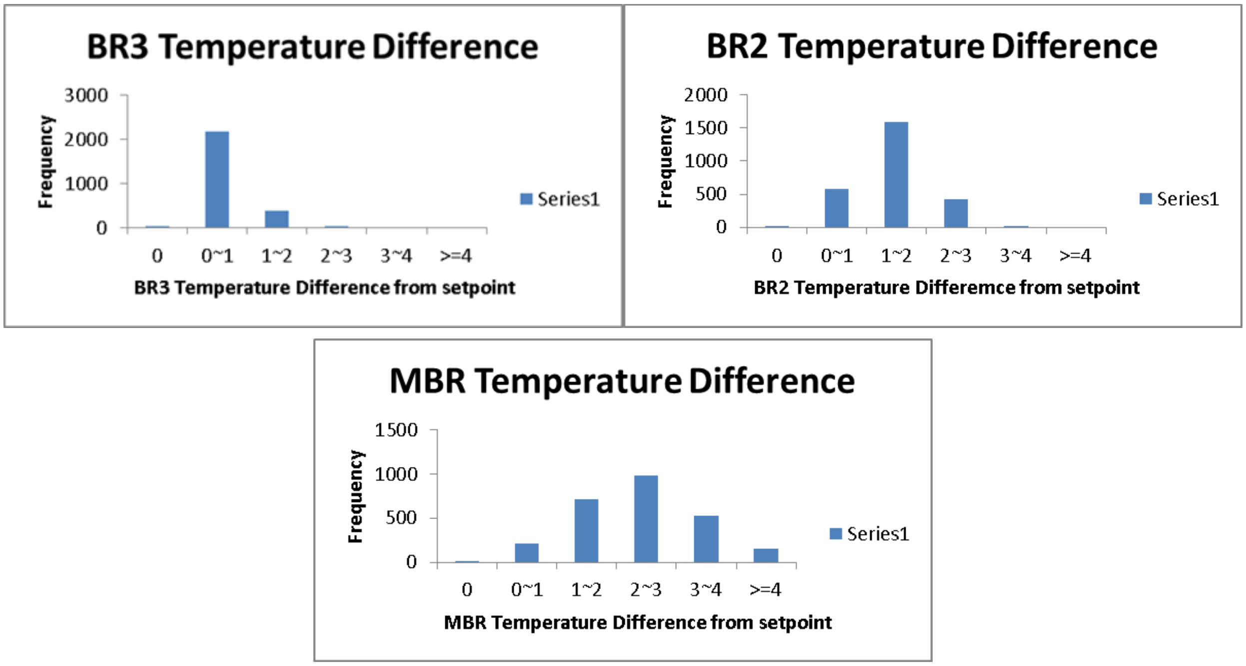


Figure 61. Massachusetts home cooling season room temperature relative to the thermostat location when heat pump was operating

6.2.2.2.3 Room-to-Room Temperature Differences

As shown in the following histogram, room-to-room temperature differences in cooling were always below 6°F (Figure 62).

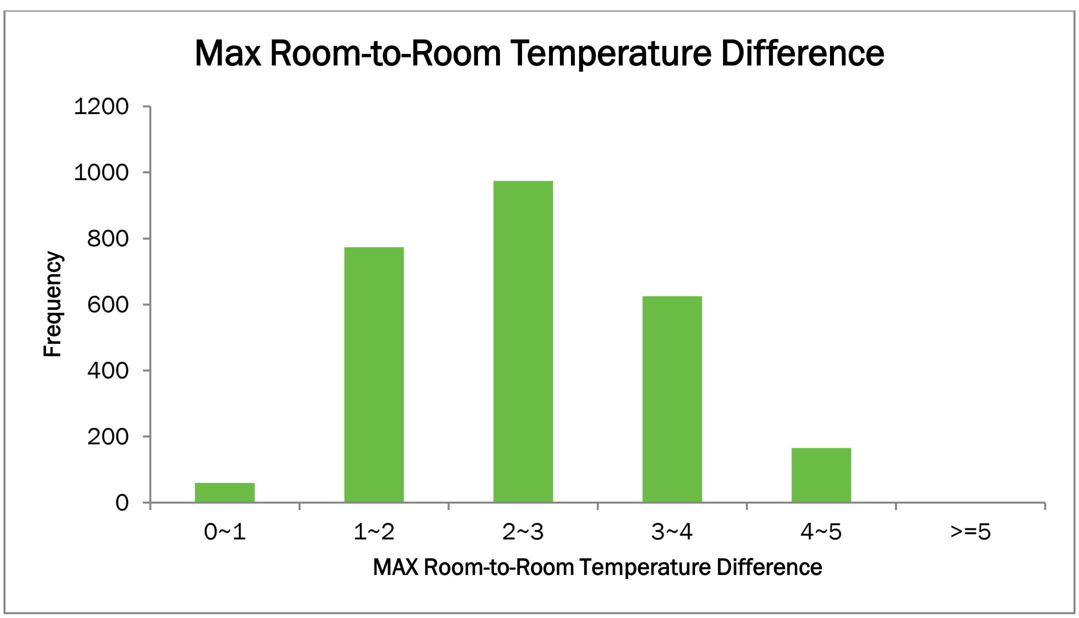


Figure 62. Massachusetts home cooling season maximum room-to-room temperature difference when heat pump was operating

6.2.2.3 ACCA Manual RS Comfort Compliance Summary

A summary of comfort compliance with respect to ACCA Manual RS during the occupied period is presented in Table 22.

Table 23. Massachusetts Home ACCA Manual RS Compliance Summary

Comfort Item	Season	Requirement	Summary
Relative humidity	Heating	30% RH maximum	RHs of all conditioned rooms were higher than 30% for significant periods of time
	Cooling	55% RH maximum	BR2 and MBR: Compliance 93% of time BR3 and hallway: RH within desired range 60% of time
Dry-bulb temperature at the thermostat	Heating	Set point temperature ±2°F	Set point was not recorded during the occupied period
	Cooling	Set point temperature ±3°F	Set point was not recorded during the occupied period
Dry-bulb temperature in any conditioned room	Heating	Set point temperature ±2°F	MBR: 99% compliance BR3 and BR2: 87% and 57% compliance, respectively
	Cooling	Set point temperature ±3°F	BR3 and BR2 temperatures were always within ±3°F of room temperature where thermostat was located. MBR met the criteria 74% of time.
Room-to-room temperature differences	Heating	4°F maximum	Room-to-room temperature difference was less than 4°F 85% of time
	Cooling	6°F maximum	Temperatures in all rooms were within 5°F of one another; 100% compliance
Floor temperature	Heating	65°F minimum at 4-in. above the floor for 70°F thermostat setting	Floor temperatures were not measured
	Cooling	N/A	Floor temperatures were not measured

6.2.3 Comfort—ASHRAE Standard 55

An analysis was completed to assess comfort relative to ASHRAE Standard 55-2010 during the heating period and the cooling period. Table 23 summarizes the criteria and results of this analysis for relevant criteria (surface temperatures and stratification not measured).

Table 24. Massachusetts Home ASHRAE Standard 55 Compliance Summary

	Allowable Limits	Heating Season Results	Cooling Season Results
Operative Temperature	Within the comfort zone defined by ASHRAE 55, -0.5<PMV<+0.5	94% compliance	2% compliance
Humidity Ratio	Below 0.012 as per humidity limits Section 5.2.2 of ASHRAE 55	99% compliance	99.5% compliance
Cyclic Variations: Positive or Negative (Drifts/ Ramps)	In any 15-min period, up to 2°F change In any 30-min period, up to 3°F change In any 60-min period, up to 4°F change In any 120-min period, up to 5°F change In any 240-min period, up to 6°F change	100% compliance	100% compliance

6.2.3.1 Operative Temperature and Humidity Ratio

The factors affecting operative temperature and humidity comfort under ASHRAE 55 are dry-bulb temperature, mean radiant temperature (MRT), RH, air speed, metabolic rate (met), and clothing value (clo). Average room-air-temperature data across the house and ambient temperature data were used to calculate surface temperatures throughout the house, which were then used to calculated MRT. Different assumptions of clo, met and air speed were made for heating and cooling season. As done for the New Jersey home occupied period, the thermal comfort package of R statistics was used to calculate the PMV index for this analysis.

6.2.3.1.1 Heating Period

For the heating season, a winter clothing value of 1.0 clo was used. A default metabolic rate of 1.1 met was used, as was an air speed of 0.15 meters per second. MRT (see Equation 1) was estimated as a weighted average of interior surface temperatures, which were calculated based on indoor temperatures, outdoor temperatures, and conductance of the assembly.

Based on the PMV calculation results using the thermal comfort package of R statistics, the home was within compliance of ASHRAE 55-2010 94% of the time during the monitoring period. Average PMV value during this period was -0.16. During this period, the humidity ratio of all rooms was below 0.012 g/g for 99% of the time.

6.2.3.1.2 Cooling Period

For the cooling season, a summer clothing value of 0.5 clo was used. The default metabolic rate of 1.1 met and air speed of 0.15 meter per second remained the same for this season. MRT was estimated as described above. The home was within ASHRAE 55-2010 compliance only 2% of the time during the period (PMV of -0.5 to 0.5). For 44% of the time, PMV value was within -1.0 and +1.0. The average PMV value during this period was -1.04, indicating that the indoor environment was consistently under a slightly cool condition. In the cooling season, room humidity ratio was lower than 0.012 g/g for 99.5% of time.

6.2.3.2 Cyclic Variations

6.2.3.2.1 Heating and Cooling Season

Temperatures in all rooms in the home complied with ASHRAE 55 cyclic-variation requirements 100% of the time.

6.3 Occupant Response

The occupants were dissatisfied with the comfort to a greater extent than the comparison to the two comfort standards would indicate. A number of issues were isolated, their likely cause identified and solutions proposed. The solutions were implemented at the end of the monitoring period so no data are available on their effect; however, feedback will be solicited from occupants after the next heating season. Table 25 lists the issues, likely causes, and remediation actions.

Table 25. Massachusetts Comfort Issues and Resolution

Issue	Likely Cause	Recommended Action
"Chill" in living room in winter	A technician visited the home and reported cold floor temperatures along street-facing edge of living room floor (47°F at edge and 63°F at 18-in. from edge; low 30s°F outside and living room air temperature at 74°F. He also reported feeling air movement near the base of the living room wall. He observed children sitting on the laminate floor. Chill may be caused by a cool floor and airflow from heat pump above, despite the warm air temperature. Occupants were in the habit of activating the living room electric baseboard heater to combat the chill.	The first approach was to reduce the chill from the floor by (1) installing a rug/carpet + carpet pad in living room; (2) air sealing along the bottom of the wall behind the base molding in living room; and (3) insulating from the outside the exposed foundation wall around the living room. Note that the slab is insulated at the edge and underneath; however, the insulation has a beveled top and there is a large exposed foundation wall.
Warm bedroom in winter	The technician measured 0.5°F lower in the bedrooms compared to the living room. Because the residents maintained high temperatures in the living room (using resistance heat), warm air from near the ceiling was drawn by the transfer fans into the bedrooms where there were no similar local cold spots.	Eliminating the living room chill should allow the occupants to avoid using the baseboard heater, thereby reducing the living room air temperature. As a result, air from transfer fans will be cooler. Also, putting transfer fans on switches as originally intended would have allowed them to be turned off individually when desired.
Warm bedroom in summer	Residents prefer cool bedroom temperatures. This is unlikely to be achieved with the current configuration; i.e., the bedrooms are warmer than the living room in summer.	Install a new heat pump (Mitsubishi FH06) in the master bedroom; on the wall opposite and facing the doorway. Deactivate the transfer fan to this bedroom and the baseboard heater in this bedroom. Install ceiling fans in bedrooms.
Overcooled living room in summer	Living room heat pump set point very low to minimize temperature of air that is delivered to bedrooms through transfer fans.	Addition of new heat pump will reduce need to overcool living room, as primary problem bedroom (master bedroom) will have dedicated cooling.
Master bathroom heater use	Draft from transfer fan created chilly conditions that were routinely combatted by use of resistance heater.	Putting transfer fan on switch will allow it to be turned off to improve comfort when desired.
High energy bills	Overworking living room heat pump because of low cooling set point and high heating set point forces unit to work in less efficient range of operation.	Actions recommended above will allow heat pump set point to be moderated. Note that space heating was 35% of energy use with the other major factors being plug/lights/appliance (35%), ventilation (1%), hot water (20%), and clothes dryer (9%).

6.4 Massachusetts Home Conclusions

The Massachusetts home was less successful than the New Jersey home because of a combination of factors, including a more challenging layout (end-loaded living area), small but significant thermal defects in the envelope, missing fan controls, and occupant preferences that were at odds with some basic strategies of the home (i.e., it was not possible to cool the bedroom below the living room temperature in summer). Higher cooling and supplemental heating use contributed to higher than expected energy bills. Nevertheless, important lessons were learned, and the planned remediation will be followed to assess impact on subjective comfort. Homes with this plan layout may be best served by a two-heat-pump configuration.

7 Maryland Habitat Home—Occupied Testing

7.1 Design and Construction

Like the Massachusetts home, the 1,056 ft^2 site-built Maryland home had an end-loaded living area with bedrooms and bathrooms accessed from a short hallway (Figure 63). This is a challenging design to serve with a point-source heat pump, so in this case a second heat pump indoor unit was placed in bedroom 2 (the owners preferred it be located in bedroom 2 rather than the master bedroom). Other major differences between this house and the Massachusetts home were the use of exhaust ventilation, ceiling fans throughout, and an electric resistance tank water heater (Table 25). Photos of the home under construction are shown in Figure 64 through Figure 67.

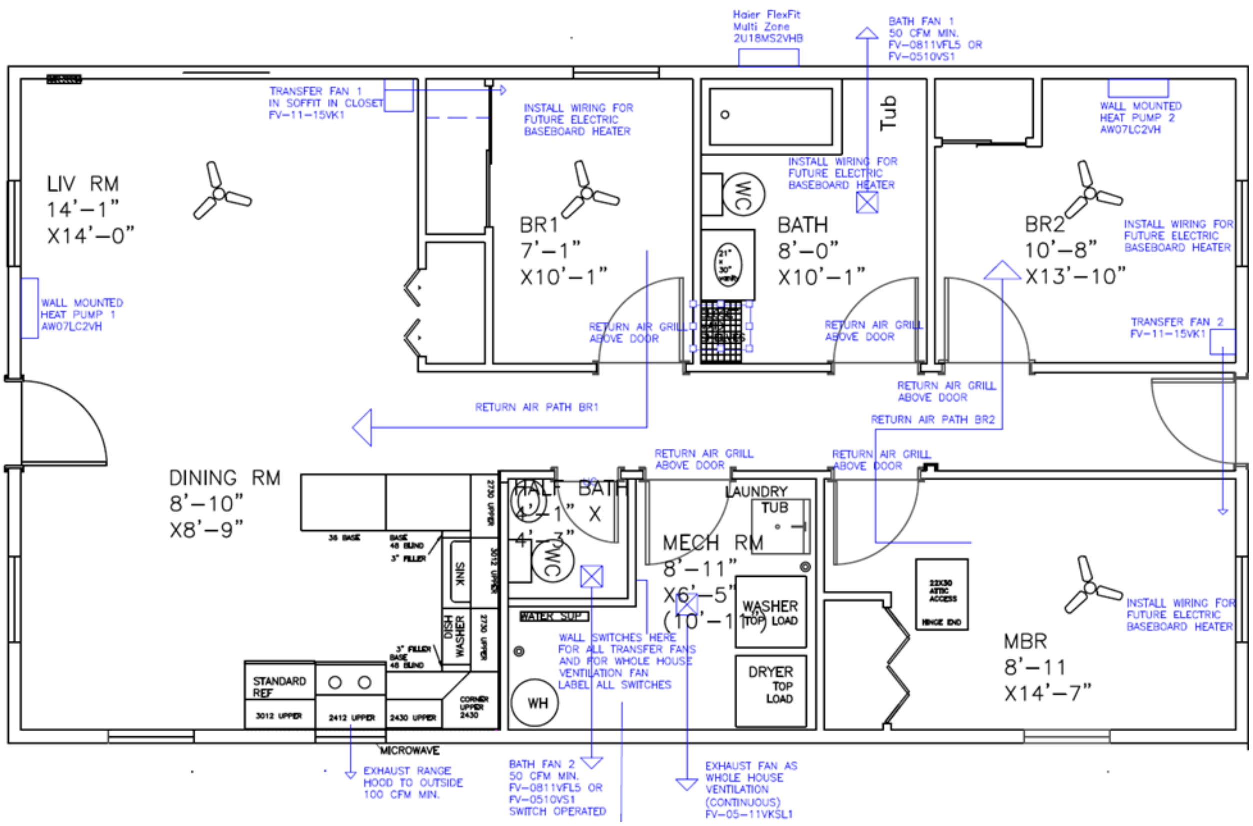


Figure 63. Maryland home plan

Table 26. Maryland Habitat Specifications

	Item	Maryland Habitat Home
Space Conditioning, Ventilation	Heat pump	Wall-mounted ductless heat pump: Haier FlexFit Multi Zone 2U18MS2VHB + (2) AW07LC2VH
	Backup heating	Electric resistance supplemental heat in bathrooms only; on wall timers
	Fans	Transfer fans: Panasonic FV-11-15VK1 – 150 cfm in 2 bedrooms Full bath: Panasonic FV-0811VFL5, set to 80 cfm; Whisper switch with condensation sensor Half bath: Panasonic FV-0811VFL5, set to 50 cfm, switch activated
	Whole-house ventilation	FV-05-11VKSL1; 41 cfm continuous (tested at 52 cfm)
	Ceiling fan	4 ENERGY STAR qualified in bedrooms and living room
Water Heating	Water heater	Conventional electric storage tank
Envelope	Floor	4-in. concrete slab with 2-in. XPS (R-10) under slab and 1-in. XPS (R-5) slab edge insulation
	Ceiling	~16.5-in. cellulose insulation in attic (R-60); 5.5-in. heel truss providing about R-22.5 over sidewall top plates 128 ft^2 storage attic: .5-in. gypsum wallboard, R-21 fiberglass batt, 2-in. XPS foam with joints taped on both sides and floor decking
	Walls	R-21 cavity insulation in 2x6 wall; plus 2-in. XPS (R-10) exterior sheathing
	Windows	U-value: 0.22, SHGC: 0.18; triple pane, low-e; window area: 123 ft^2
	Exterior doors	U-value: 0.20
	Attic	Ventilated
	Infiltration	2.51 ACH_{50} depressurization test; ENERGY STAR prescriptive measures

Figure 64. Maryland home foundation with vertical slab edge insulation (left); continuous exterior insulation (right)

Figure 65. Maryland home interior rough framing (left); sealed ceiling plane and insulated walls (right)


Figure 66. Maryland home street elevation (left); heat pump (right)

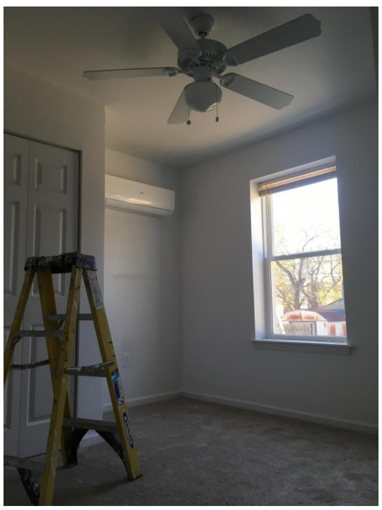
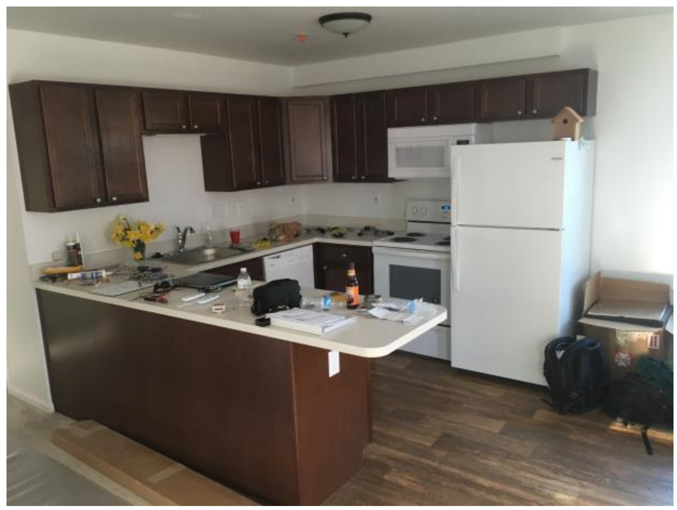



Figure 67. Maryland home bedroom with ceiling fan and heat pump fan coil (left); kitchen (right)

7.2 Results and Analysis

7.2.1 Space Conditioning and Ventilation Energy

7.2.1.1 Energy Model Comparison—Maryland

BEopt models were prepared for two baseline code (ICC 2009) homes: all electric and gas furnace. The specifications for the as-built house are shown in Table 25, and the baseline code house specifications are the same as those in Table 10. Results are shown in Figure 68. The as-built house uses 69% less source-space-conditioning energy than the baseline all-electric home and 41% less than the baseline gas-heated home. There was no backup resistance heating except for a bathroom heater on a timer. The energy use of the bathroom heater (52 kWh) was estimated based on occupied period usage and added into the model as a plug load.

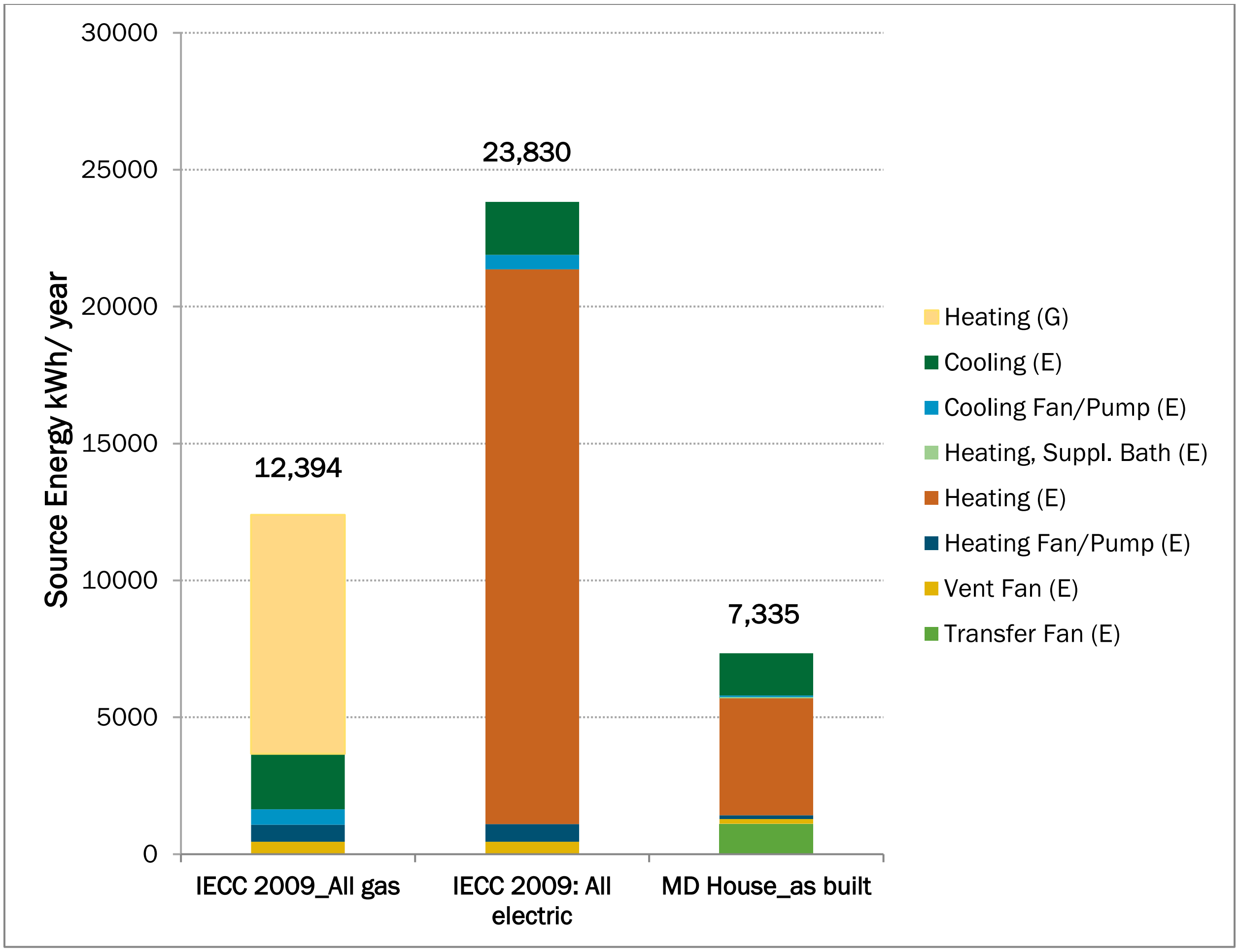


Figure 68. Weather-normalized model results vs. monitored data

7.2.1.2 Occupied Energy Consumption—Maryland

Energy consumption data were collected from the occupied home from August 2, 2018, to August 1, 2019. The electric energy consumption was then scaled by the 2.80 source-to-site energy use factor and is shown in Table 26.

Table 27. Monitored Energy Results—Maryland

Source	Occupied Annual Energy in kWh
Heating (heat pump)	4,858
Supplemental heating bathroom	183
Cooling	1,782
Ventilation fan energy	652
Transfer fan energy	1,117
Total space conditioning and ventilation energy	8,592
Domestic water heating	6,799
Total house energy	28,800

7.2.2 Comfort—ACCA Manual RS

The ACCA Manual RS criteria were separated into heating and cooling and are presented separately in this section, with cooling following heating. Because the thermostat set point was controlled by the occupants and unknown, dry-bulb temperature at the thermostat was not evaluated.

7.2.2.1 Heating Season

7.2.2.1.1 Relative Humidity

Figure 69 through Figure 72 show the distribution of the RH when the heat pump was operating. RH was nearly always higher than 20% to 30%, the desired range of ACCA. As shown in the histograms, the living room was less humid than bedrooms, probably because one of the indoor heads of the heat pump was located there and was used frequently.

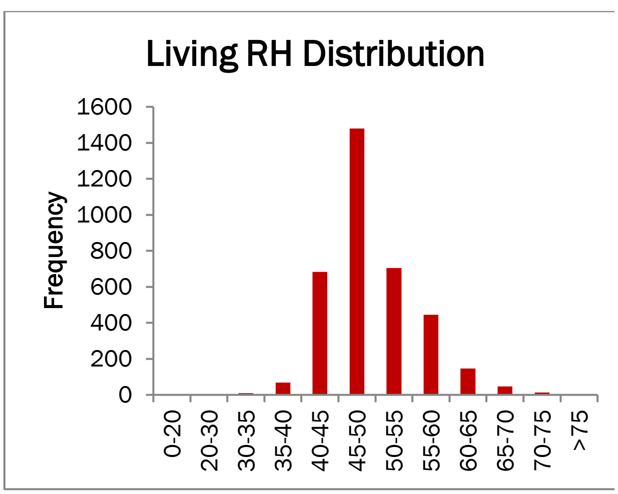


Figure 69. Heating season living room RH when heat pump compressor was operating

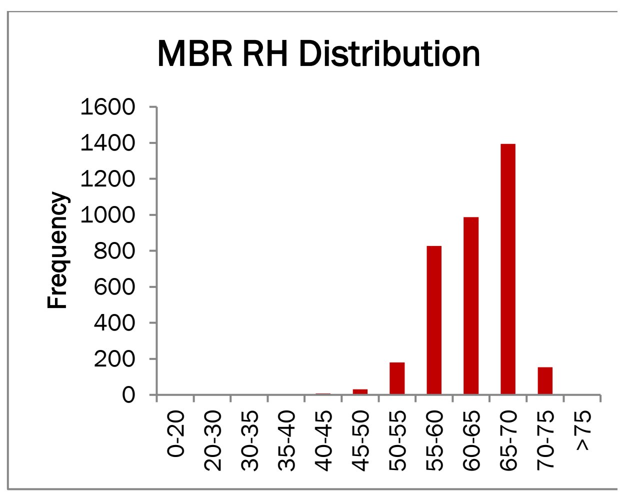


Figure 70. Heating season master bedroom RH when heat pump compressor was operating

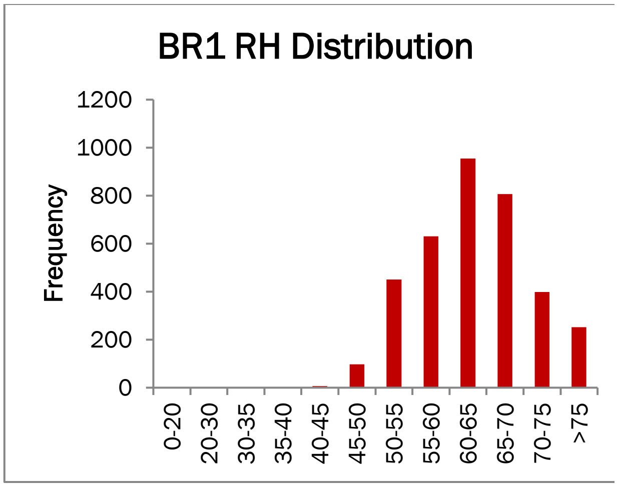


Figure 71. Heating season bedroom 1 RH when heat pump compressor was operating

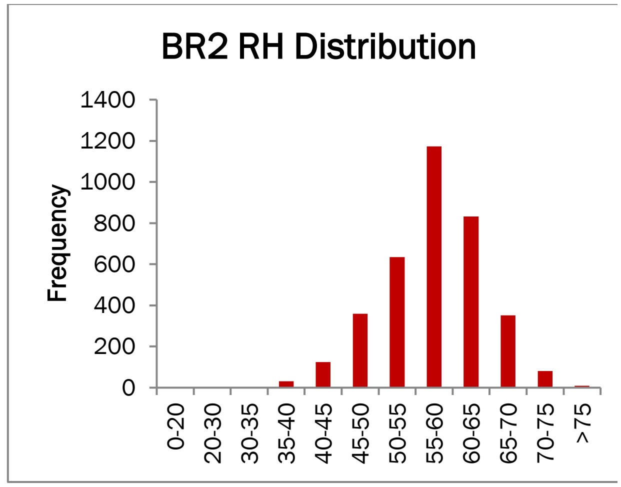


Figure 72. Heating season bedroom 2 RH when heat pump compressor was operating

7.2.2.1.2 Dry-Bulb Temperature at the Thermostat, Dry-Bulb Temperature in Any Conditioned Room, and Room-to-Room Temperature Difference

As the set point temperature was not recorded in this occupied home, dry-bulb temperature at the thermostat was not compared with the set point. Instead, dry-bulb temperatures of the rooms where thermostats are located were taken to be at set point and then compared to dry-bulb temperatures in other conditioned rooms. The temperature differences between rooms connected by transfer fans were compared during heat pump operation and are shown in Figure 73 and Figure 74.

For 44% of time, the room temperature difference between bedroom 2 and the master bedroom was less than 2°F. However, the room temperature difference between bedroom 1 and the living room was compliant only 5% of time, which was probably caused by the fact that bedroom 1 was rarely occupied and the transfer fan connecting bedroom 1 and the living room was switched off by the occupants during the heating season.

Room-to-room temperature differences are to be within 4°F in heating mode. Using the same data as the previous paragraph, the temperature difference between the master bedroom and bedroom 2 was less than 4°F for 70% of the time; whereas the temperature difference between bedroom 1 and living room was less than 4°F for only 11% of time, because its transfer fan was kept off often during heating season.

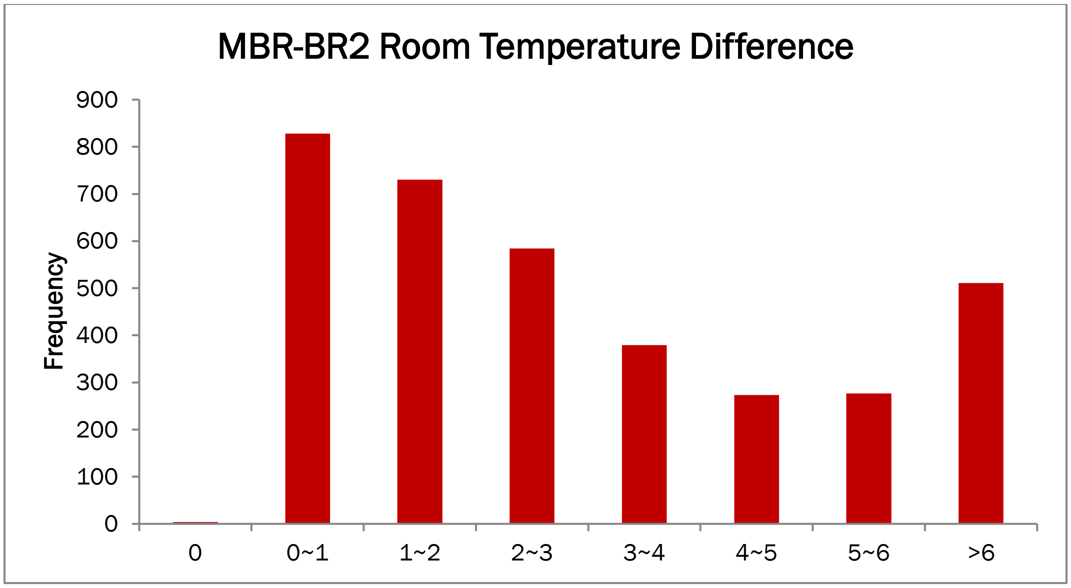


Figure 73. Room temperature difference between master bedroom and bedroom 2—heating

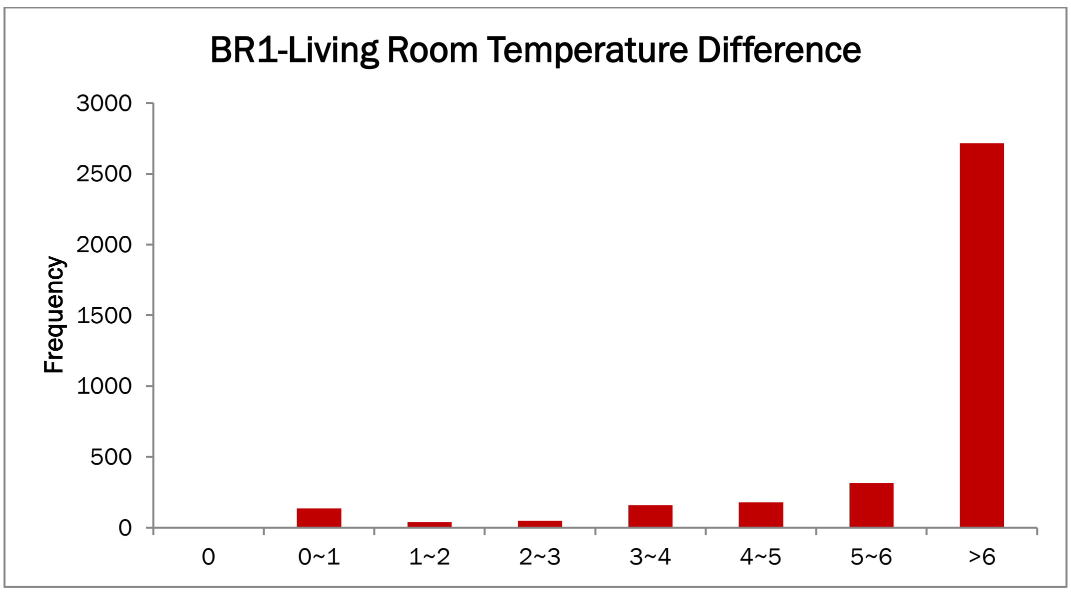


Figure 74. Room temperature difference between bedroom 1 and living room—heating

7.2.2.2 Cooling Season

7.2.2.2.1 Relative Humidity

RH distributions of rooms during cooling season are shown in Figure 75 through Figure 78. The living room is the most humid among the rooms; its RH is less than 55% only 35% of the time. Master bedroom, bedroom 1, and bedroom 2 have RH less than 55% for 57%, 68%, and 56% of the time during active cooling, respectively. ACCA Manual RS defines the desired range of RH during cooling to be between 25% and 50%, with a maximum of 55%. As the least humid room, bedroom 1 has RH within the desired range 13% of time. The RH of the rest of the house was within this range for less than 10% of time.

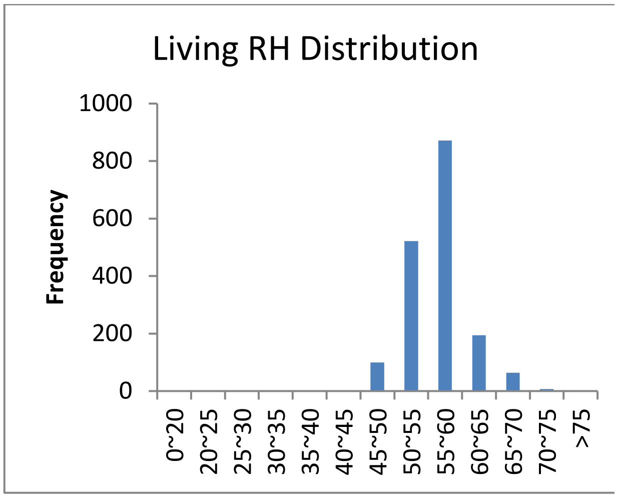


Figure 75. Living room RH distribution—cooling

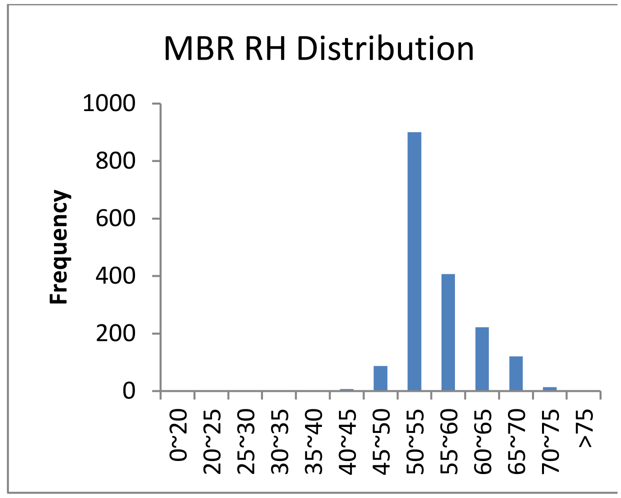


Figure 76. Master bedroom RH distribution—cooling

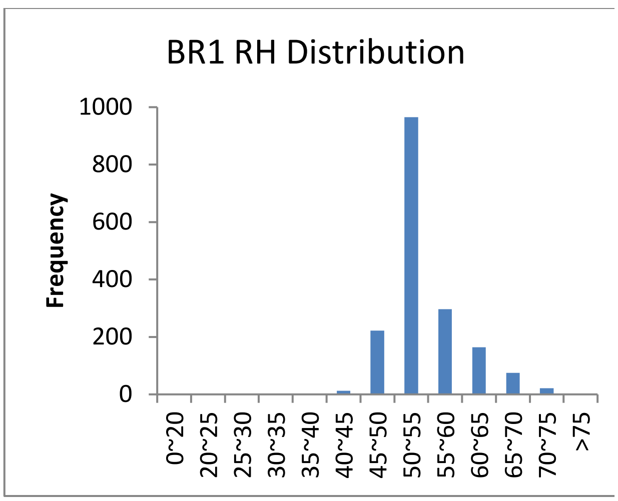


Figure 77. Bedroom 1 RH distribution—cooling

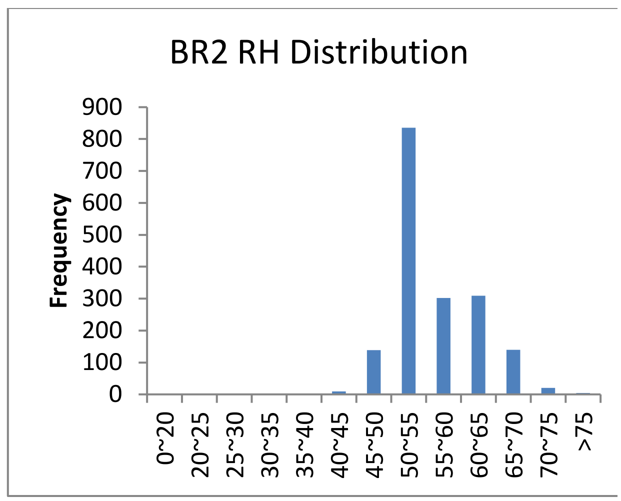


Figure 78. Bedroom 2 RH distribution—cooling

7.2.2.2.2 Dry-Bulb Temperature at the Thermostat, Dry-Bulb Temperature in Any Conditioned Room, and Room-to-Room Temperature Difference

As discussed in the heating section, dry-bulb temperature at the thermostat was not compared with the set point in this study. Instead, dry-bulb temperatures of the rooms where thermostats were located were taken as the set point and then compared to dry-bulb temperatures in other conditioned rooms. Rooms connected by transfer fans were grouped together, and their temperature differences during heat pump operation are shown in Figure 79 and Figure 80.

The master bedroom dry-bulb temperature was within 3°F of bedroom 2 temperature, which is taken as the set point, 86% of the time; bedroom 1's dry-bulb temperature was within 3°F of the living room temperature, which is taken as the set point, 96% of the time.

Room-to-room temperature differences are to be within 6°F in cooling mode. The temperature difference between the master bedroom and bedroom 2 was less than 6°F 99% of time; the temperature difference between bedroom 1 and the living room was less than 4°F 100% of time.

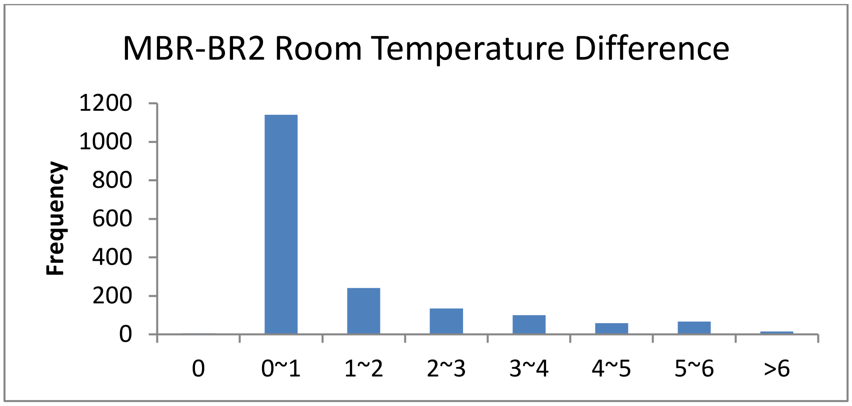


Figure 79. Room temperature difference between master bedroom and bedroom 2—cooling

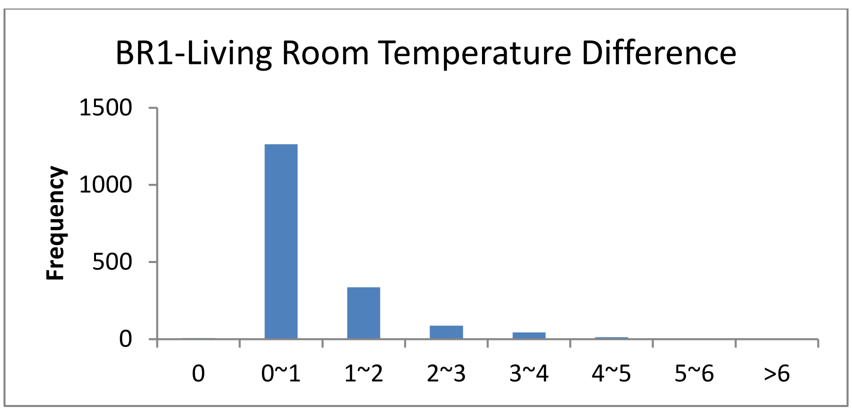


Figure 80. Room temperature difference between bedroom 1 and living room—cooling

7.2.2.3 ACCA Manual RS Comfort Compliance Summary

A summary of comfort compliance with respect to ACCA Manual RS is presented in Table 28.

Table 28. ACCA Manual RS Comfort Compliance

Comfort Item	Season	Requirement	Summary
Relative Humidity	Heating	30% RH maximum	RH exceeded 30% all the time
	Cooling	55% RH maximum	RH was lower than 55% for 35% of the time in the living room; 57% of the time in the master bedroom; 68% of the time in bedroom 1; and 56% of the time in bedroom 2.
Dry-Bulb Temperature at the Thermostat	Heating	Set point temperature ±2°F	Set point was not recorded
	Cooling	Set point temperature ±3°F	Set point was not recorded
Dry-Bulb Temperature in Any Conditioned Room	Heating	Set point temperature ±2°F	MBR-BR2: within 2°F 44% of time BR1-Living: within 2°F 5% of time
	Cooling	Set point temperature ±3°F	MBR-BR2: within 3°F 86% of time BR1-Living: within 3°F 96% of time
Room-to-Room Temperature Differences	Heating	4°F maximum	MBR-BR2: within 4°F 70% of time BR1-Living: within 4°F 11% of time
	Cooling	6°F maximum	MBR-BR2: within 6°F 99% of time BR1-Living: within 6°F 100% of time
Floor Temperature	Heating	65°F minimum at 4-in. above the floor for 70°F thermostat setting	Floor temperatures were not recorded
	Cooling	N/A	Floor temperatures were not recorded

7.2.3 Comfort—ASHRAE Standard 55

An analysis was completed to assess comfort relative to ASHRAE Standard 55-2010 during the heating period and the cooling period. Table 29 summarizes the criteria and results of this analysis (surface temperatures and stratification not measured).

Table 29. ASHRAE 55 Thermal Comfort Compliance

	Allowable Limits	Heating Season Results	Cooling Season Results
Operative Temperature	Within the comfort zone defined by ASHRAE 55, -0.5<PMV<+0.5	22% compliance	67% compliance
Humidity Ratio	Below 0.012 as per humidity limits Section 5.2.2 of ASHRAE 55	Living Room: 72% compliance MBR: 93% compliance BR1: 77% compliance BR2: 84% compliance	Living Room: 46% compliance MBR: 71% compliance BR1: 69% compliance BR2: 71% compliance
Cyclic Variations: Positive or Negative (Drifts/ Ramps)	In any 15-min period, up to 2°F change In any 30-min period, up to 3°F change In any 60-min period, up to 4°F change In any 120-min period, up to 5°F change In any 240-min period, up to 6°F change	100% compliance	100% compliance

7.2.3.1 Operative Temperature and Humidity Ratio

The same assumptions and process were used to analyze the Maryland home as was used for the Massachusetts home.

7.2.3.1.1 Heating Period

Based on the PMV calculation results using the thermal comfort package of R statistics, the home was compliant with ASHRAE 55-2010 for 22% of the time during the period. The average PMV value during this period was 0.71, indicating a slightly warmer indoor condition than what is defined by the ASHRAE thermal comfort zone. Most likely this was due to occupant preference.

During this period, humidity ratio of the living room was below 0.012 g/g for 72% of time; the humidity ratio of the master bedroom was below 0.012 g/g for 93% of time; the humidity ratio of bedroom 1 was below 0.012 g/g for 77% of time; and the humidity ratio of bedroom 2 was below 0.012 g/g for 84% of time.

7.2.3.1.2 Cooling Period

The home was compliant with ASHRAE 55-2010 for 67% of the period tested. The average PMV value during this period was -0.24, indicating indoor environment was consistently under a slightly cool condition.

During this period, the humidity ratio of the living room was below 0.012 g/g for 46% of time; the humidity ratio of the master bedroom was below 0.012 g/g for 71% of time; the humidity ratio of bedroom 1 was below 0.012 g/g for 69% of time; and the humidity ratio of bedroom 2 was below 0.012 g/g for 71% of time.

7.2.3.2 Cyclic Variation

7.2.3.2.1 Heating Season Cyclic Variations: Positive or Negative (Drifts/Ramps)

The percent of time that each room passed the cyclic variations criteria during the heating period is shown in Table 30. Most rooms complied fully for all criteria.

Table 30. Cyclic Variations Passing Rate—Heating Season

Criteria	Living	MBR	BR1	BR2	Family Bath	Second Bath
15-min, max 2°F change	100%	100%	100%	100%	100%	100%
30-min, max 3°F change	100%	100%	100%	100%	100%	100%
60-min, max 4°F change	100%	100%	100%	100%	100%	100%
120-min, max 5°F change	99%	100%	100%	99%	100%	100%
240-min, max 6°F change	100%	100%	100%	100%	100%	100%

7.2.3.2.2 Cooling Season Cyclic Variations: Positive or Negative (Drifts/Ramps)

Table 31 shows that temperatures in all rooms complied with ASHRAE 55 cyclic-variation requirements nearly all the time.

Table 31. Cyclic Variations Passing Rate—Cooling Season

Criteria	Living	MBR	BR1	BR2	Family Bath	Second Bath
15-min, max 2°F change	100%	100%	100%	100%	100%	100%
30-min, max 3°F change	100%	100%	100%	100%	100%	100%
60-min, max 4°F change	100%	100%	100%	100%	100%	100%
120-min, max 5°F change	100%	100%	100%	100%	100%	100%
240-min, max 6°F change	100%	100%	100%	99%	100%	100%

7.3 Occupant Response

Residents rated the comfort of the home as good in all seasons, with even temperatures and no humidity complaints. They were satisfied with the utility bills, saying they "seemed ok." They rated the heating and cooling system as well as controls to be good. Self-reported set points were 75°F for heating and 78°F for cooling. They mostly turned off the heat pump when the home was unoccupied.

An on/off switch was installed for the rear bedroom transfer fan, but the residents would have liked an on/off switch for the living room/center bedroom transfer fan as well. There was dust accumulation on the grille of the continuously operating transfer fan, something that was also observed at the Massachusetts home. A filter on the transfer fan inlet might mitigate both of these issues. Residents also noted that cooking odors were detectable in the rear bedrooms, although there was no direct transfer fan from the living/dining area to those bedrooms.

7.4 Maryland Home Conclusions

The Maryland home had a similar layout as the Massachusetts home; however, two indoor heat pump air handlers were installed instead of one, in the living room and one bedroom. It was also in a milder climate. It had similar modeled energy savings as the New Jersey home, exceeding project goals when gas and electric baselines are averaged. Residents were satisfied with comfort, although their use of the heat pumps and fans (and lack of use) impacted the comparison to the comfort metrics. For example, the temperature in the unused bedroom (bedroom 1) varied significantly from the temperature in the living room during heating season because the transfer fan was intentionally turned off and the door was closed. During cooling season, this temperature difference was small—within 3°F of the living room 96% of the time. Humidity was again over the ACCA thresholds about 90% of the time in cooling and 100% in heating. Humidity was within ASHRAE 55 limits 72%–93% of the time in heating and 46%–71% in cooling.

8 Conclusions

This approach to homebuilding emphasizes a superior thermal enclosure in order to simplify the space-conditioning system. Similar to other high-performance building approaches such as Passive House, the idea is to drive down space-conditioning loads to minimize heating equipment needs and costs; however, it is specifically tailored to small single-story homes in IECC climate zones 4–5. Specifically, this project analyzed manufactured homes over unconditioned crawlspaces and single-story, modest-sized wood-framed homes with slab foundations such as those commonly built by affordable housing organizations like Habitat for Humanity.

One source of cost savings is a simplified space-conditioning distribution method, which is made possible by the lower thermal loads. Some of the cases tested succeeded in reducing space conditioning loads by half compared to the 2009 International Energy Conservation Code. Some but not all comfort criteria were consistently met; two homeowners were satisfied while a third was not.

The key elements of these certified Zero Energy Ready Homes were an efficient building envelope, a one- or two-point (head) ductless heat pump, through-wall transfer fan distribution system, and a whole-house ventilation system.

The potential benefits of this approach compared to traditional code-minimum homes with central forced air space-conditioning include less noise from large air handlers and high volumes of moving air; low energy bills and the ability to make the short jump to net zero energy with the addition of photovoltaics; passive resiliency due to the superior thermal envelope; and homeownership costs similar or lower than standard homes when factoring in the first costs and monthly energy bills. Because the space-conditioning system is atypical, it may take some getting used to and may not be a good fit for all occupants. Those who want to keep very low bedroom temperatures in summer may not be satisfied. Conversely, those who prefer higher bedroom temperatures in winter may use excessive backup heat (if provided) or be unsatisfied. Test homes were more humid than the desirable room condition defined by ACCA Manual RS in both cooling and heating seasons, although occupants did not complain.

8.1 Answers to Research Questions

Responses to the initial research questions are presented next:

1. What level of envelope efficiency and types of features are required to allow homes to operate with acceptable comfort and energy outcomes?

The three test homes were built to comply with the Zero Energy Ready Home (ZERH) program; these specifications were generally adequate for the New Jersey and Maryland homes. The Massachusetts home insulation levels were satisfactory; however, thermal bridging at the slab edge was reported to be a comfort issue despite compliance with ENERGY STAR slab edge insulation specification. Airtightness measures could have been improved, as the homes did not meet ZERH prescriptive levels of airtightness.

2. Do envelope provisions increase moisture/condensation risks, particularly at foam sheathing and unvented attics?

No moisture concerns were observed in the New Jersey home where this was measured.

3. How can humidity levels throughout the home be adequately controlled?

Humidity levels were difficult to control in all three houses. The heat pumps did not dehumidify the homes to ACCA Manual RS comfort criteria, although they were not reported as problematic by occupants and were more successful in meeting ASHRAE 55. A dedicated dehumidifier may address local humidity at a high cost in energy. Heat pump dehumidification mode had a modest impact. Heat pumps with higher latent load removal capability may address this need, but are not widely available in the United States.

4. Are there acoustic and olfactory issues to be addressed with the transfer fan approach?

The Maryland occupants noted that cooking odors were detectable in the rear bedrooms, although there was no direct transfer fan from the living/dining area to those bedrooms; the Massachusetts occupants did remark on the transfer fan sound; the residents of the New Jersey home had no issues.

5. How can heat recovery ventilation be affordably integrated into homes to optimize ventilation performance and indoor air quality?

Heat recovery ventilation is a significant additional first cost and increases overall annualized energy-related expenses according to the modeling done at the outset of the project. A low-cost ERV was successfully integrated into the Massachusetts home, but lower-cost and balanced ventilation products with adequate flow rates to meet ASHRAE standards are needed.

6. What type of control system would be needed for system integration, how would the controls (thermostats) be connected to the main unit, and what are the ideal thermostat locations?

Heat pumps, transfer fans, and backup heaters were all employed in the test homes. A control scheme that would have limited the backup heaters based on outdoor and indoor temperature as well as room occupancy was planned for the Massachusetts home but not installed. Transfer fans and ventilators should be occupant-controllable with clearly labeled on/off switches. Heat pump thermostats are best located so as to measure the main living space temperature rather than return air temperature, which may vary from that of the occupied zone.

7. What are the overall energy savings and usage compared to baseline specifications?

The New Jersey home as-built models predicted 7,129–8,418 kWh source space-conditioning energy, which is 44%–53% less than the baseline all-gas home and 72%–76% less than the baseline all-electric home, depending on whether compared to the best case or supplemental heating case model. When occupied by a family of two, the New Jersey home used 7,149 kWh source space-conditioning energy, based on the data collected during a one-year occupied period.

The Massachusetts home as-built models predicted 8,245–10,662 kWh source space-conditioning energy, which is 29%–45% less than the baseline all-gas home and 67%–74% less than the baseline all-electric home, depending on whether compared to the best case or supplemental heating case model. When occupied by a family of five, the Massachusetts home used 12,916 kWh source space-conditioning energy, based on the data collected during a one-year occupied period.

The Maryland home had no backup resistance heating except for a bathroom heater, so there is only one as-built model, which predicts 7,335 kWh source space-conditioning energy, 41% less than the baseline all-gas home and 69% less than the baseline all-electric home. The Maryland home used 8,592 kWh source space-conditioning energy, based on the data collected during a one-year occupied period.

Note that the models are not calibrated to measured energy consumption and use TMY weather data, whereas space-conditioning energy used in the occupied homes is dependent on occupant behavior and weather conditions during the occupied period. Therefore, the modeled and measured results are not directly comparable.

8. What is the strategy for locating the heating/cooling source and transfer fans to achieve desired airflow and temperature distribution? How should return air pathways to the indoor unit be designed? What is the required transfer fan capacity to meet the needs of the spaces served without causing drafts?

Short circuiting of the transfer fan should be avoided (i.e., air leaving a bedroom should mix with the air in the main body of the house before recirculating back into the bedroom). This can sometimes be hard to achieve when bedrooms share limited wall space with the main living area. Transfer fan capacity must be in the 90–150 cfm range to move the needed amount of energy into or out of a room. Adjustable fans are a good idea to permit changing the flow rate by season or based on occupant preference.

9. What airflow or throw pattern is acceptable to the occupant?

Only the Massachusetts residents complained about air from the transfer fans, and theirs was the only home without control over the fans. High wall delivery with air thrown across the ceiling and down the far wall was adequate and satisfactory for most occupants for most of the time.

10. What home design features exacerbate temperature differentials? What home layout strategies are best suited for this approach?

Wide temperature variations were not observed. Features such as large windows, sliding glass doors, and other weak links in the thermal envelope were avoided in the test homes. Compact home layouts that organize the remote spaces (bedrooms, bathrooms) surrounding and directly adjacent to the main living space are the best layouts for this approach. Elongated or L-shaped plans and plans with the main living space at one end of the home away from some of the remote rooms are more difficult and will require additional sources of space conditioning.

11. What design and production approaches can minimize first cost for each respective product type (site and factory built)?

The largest incremental cost items for the factory-built home were the heat pump, ceiling insulation, and foam insulation wall sheathing. These costs are all dominated by material/equipment supply, so options for reducing them are limited. Increasing proficiency of plant staff in installing heat pumps and increasing volume of heat pump purchases may help. Incremental costs for Habitat were harder to collect, but also are likely to be driven by higher insulation materials costs, because labor is largely volunteer. HVAC labor costs should be similar to baseline because transfer fan installation is similar to ductwork cost, and heat pump equipment costs are similar to baseline equipment.

12. What additions/changes are required to building codes (IECC and MHCSS) to address this design approach?

No code issues were encountered on the Habitat homes. The only code issue encountered on the manufactured home was the MHCSS provision that the fresh air ventilation system deliver air to "all bedroom and main living areas" (U.S. Code of Federal Regulations). This could be interpreted as prohibiting exhaust ventilation from the bathroom serving as whole-house ventilation, even though make-up air would be entering the home from dispersed locations including into the main living space and bedrooms.

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Appendix A. Model Calibration Procedure

Introduction

Usually, a simulation model of a building does not initially predict the actual thermal performance of that building, even when the modeler has attempted to describe every detail of the building correctly to the best of her or his abilities. This is because there are many aspects of a building that are unknown or can only be estimated. In order to more closely match a model to actual performance, it is necessary to understand the details of the model, then use this understanding to design a set of experiments to perform on the actual building. The goal of the experiments is to provide data that can be used to calibrate the simulation model.

A good approach to calibrating a complex model such as a building is to conceptually break up the building into smaller parts so as to be able to adjust fewer model parameters simultaneously when attempting to match model to measurement. Another option is to design experiments that allow fewer simultaneous parameter adjustments. For example, if the building is controlled to a steady temperature, the effects of mass are largely unimportant because no heat is being added or removed from the mass due to temperature changes. This means that parameters relating to the mass of the building can be ignored during the constant-temperature period while adjusting other parameters to match model to measurement.

For the purposes of calibrating the building model, a number of specific experiments were performed on the building:

1. Experiments were performed during periods when the outdoor temperature was low enough to produce a significant heating load.
2. The building was heated using electric heaters and controlled to a steady-state temperature, with all rooms controlled to the same temperature. Total electrical energy entering the building envelope was monitored and considered the total heating load.
3. The infiltration rate of the building was measured directly using the tracer-gas concentration decay technique.

Process

The basic calibration procedure was as follows:

1. Build a thermal model of the building in TRNSYS (University of Wisconsin-Madison Solar Energy Laboratory 2019) and a bulk airflow model of the building in CONTAM (National Institute of Standard and Technology 2019) using as much information as possible from the home builder and filling in with experiential “best” practice values where information is unavailable.
2. Conceptually break the model into pieces in order to reduce the number of parameters that are simultaneously influencing the behavior of the model. Three pieces were chosen: the

attic, the crawlspace, and the conditioned area. The conditioned area was further broken down into a core volume (living room and kitchen) and a perimeter volume (bedrooms).

3. Define the variables that are going to be adjusted, the variables that are going to be monitored as figures of merit, and the metrics by which the success of the calibration are to be judged.
    a. When possible, it is desirable to use heating energy use as the figure of merit because it is, ultimately, the figure of merit for comparing the final building model's predictions to other cases. It also lends a sense of scale; one can directly report the root mean squared error (RMSE) between model and measurement in a set of units that can be understood and correlated with cost. In some cases, where the total energy entering or leaving the space is not known, dry-bulb temperature was used as a figure of merit. Humidity ratio was also used in calibrating the infiltration model. It was important to select these figures of merit before designing the measurement plan for the building to be sure that the appropriate figures of merit were measured.
    b. Parameters to be adjusted were determined in two ways. One criterion was to focus on the parameter values in which we had the least confidence. For example, in the case of an insulated stud wall, the density and specific heat of the combined material (stud + insulation) are relatively easy to calculate based on knowledge of the ratio of stud volume to insulation volume. It is harder to compute the effective thermal conductivity of the combined material. Thermal conductivity, then, would be the preferred tuning variable.

       The other criterion was sensitivity. Preference should be given (where physically appropriate) to tuning parameters to which the figures of merit are more sensitive. For instance, in tuning the wall R-value, not only the conductivity of the wall materials but also the convection coefficient on the inside and outside of the wall impact heat transfer through the wall. Both values can be used to force a simulated interior temperature to match measured results. However, the simulated temperature is more sensitive to wall thermal conductivity, and in order to force calibration by means of the convective heat transfer coefficient, a physically unreasonable value may have to be used.
    c. In almost all cases, the RMSE and coefficient of determination between measured and simulated results (R-squared) were used to determine the success of the calibration.
4. Modify the model so that the behavior of the zones that are not being calibrated is defined (forced) to be equal to the measured values for the calibration period. In almost all cases, the temperature of adjacent spaces is not constant but varies significantly over the calibration period. It is therefore necessary to use the minute-by-minute measured temperature of the adjacent zones as both the heating and cooling set point temperature for those zones. This forces the modeled adjacent zone temperatures to be exactly equal to those measured during the calibration period.

5. Select a data period that is appropriate to the space being calibrated. If necessary and possible, run specific tests such as tracer gas tests for overall infiltration, electric heating overall shell U-value, nighttime test for conduction without solar input, and core conditioning with floating perimeter tests for interior air transfer.
6. Drive the model with measured ambient conditions. In general, the model was driven with ambient dry-bulb temperature, RH, solar radiation measured on a horizontal plane, and wind speed. During the initial calibration exercise (i.e., the Alabama house) the wind direction was not recorded and so was assumed to come steadily from the west based on conversations with people on-site.
7. Run the model and compute RMSE and r^2 in order to assess how well the model compares to measurement. However, it is also important to look at how the shape of a simulated curve compares to the measured curve for a particular figure of merit. In this calibration exercise we had comparatively few "gauges" (e.g., temperature, energy consumption, humidity) and nearly an endless number of parameters that we could adjust. It was vital to make educated decisions about which parameters to adjust to "correct" a particular kind of difference between measurement and simulation. Some notable considerations include:
   a. A phase delay (i.e., peaks occurring at different times) between model and measurement may indicate that the model's capacitance should be adjusted; energy transfer is impacting the space either faster or slower in simulation than in reality.
   b. If the space exhibits a high-frequency temperature oscillation, then infiltration rate may be at issue. Temperature changes due to energy transfer by conduction take longer to manifest in the space than temperature changes resulting from air infiltration.
   c. Daytime peak temperatures were seen to coincide with solar gains through windows. Window shading factor, as well as glazing properties, solar absorptance, and reflectance of windows can all be adjusted.
   d. Nighttime heating energy differences between model and measurement—particularly in the few hours before dawn after several days' control at a steady temperature—are typically an indication of an incorrect building shell R-value because mass effects are near zero and residual solar effects are at a minimum.
   e. Low wind periods (particularly low wind periods at night) can be used to tune wall properties, whereas high wind periods can be useful in tuning infiltration rates.

Key Features

A number of features in TRNSYS (and to a lesser degree CONTAM) contributed significantly to the ability to perform high-fidelity calibration between measured and simulated results.

First, TRNSYS does not rely on a particular form of weather file. Although it can read standard formats such as TMY3 or EPW (EnergyPlus Weather file), it can also read generic weather data such as the data that were measured and recorded minute by minute at both the Alabama and

New Jersey sites. Little manipulation of the recorded data files was needed before being able to feed them into TRNSYS and to drive the TRNSYS model with any combination of measured values. TRNSYS also allows the user to watch the value of any output variable of any component in the simulation as the simulation progresses. The output values are always the mean value over the time step, and as a result can be easily compared to measured values.

Second, TRNSYS contains a number of models for ground heat transfer. The most complex of these allows the user to define a 3D mesh of soil nodes extending away from the building for as far as the user wants, define the starting temperature of the nodes, and then resolve the temperature of each node in the mesh at each time step. The disadvantage of the complex model is in the slow speed at which it solves. However, because numerous ground heat transfer models are available, the detailed model was used to tune parameters of a simpler, faster-solving model for use during the calibration exercise.

Third, the user has the ability to force zone temperatures in a building model to exactly match some known sequence of temperatures. In this case, we were able to force the temperature of certain parts of the residence to match the measured temperatures in those spaces while allowing adjacent space temperatures to float. This proved to be particularly helpful in calibrating the attic and crawlspace zones. The temperature of the adjacent conditioned portion of the building was set equal to the measured values, while the attic and crawlspace temperatures were allowed to float. We were then able to adjust model parameters until the floating temperatures best matched their respective measured values.

Fourth, the user is able to specify and/or scale heat transfer coefficients (especially for radiative and convective transfer) directly. In other tools, it is not uncommon to be able to choose from a library of built-in correlations but the ability to directly set not only the correlation but its coefficients is less common.

Lastly, multivariate optimization can be performed by coupling TRNSYS to a generic optimization tool called GenOpt (https://simulationresearch.lbl.gov/GO/) by means of a dedicated interface called TRNOpt (Thermal Energy System Specialists, LLC, 2019). The user identifies (or develops) an error function (such as the RMSE between simulated and measured results) and then identifies any number of variables to be used in minimizing that error. GenOpt (Wetter 2004) then uses built-in algorithms to choose values of those variables and repeatedly calls TRNSYS simulations until it arrives at a minimized error.

Statistics Reported for Calibration Procedures

For each set of calibrations, a number of statistics are reported. The meanings of the statistics are discussed next.

- **Mean**: the arithmetic mean of all data points.

- **RMSE**: the Root-Mean-Squared Error, or RMSE; this statistic is a measure of the accuracy of the model over the time period of interest. The RMSE is always a positive number, and a value of zero represents a perfectly accurate model. The RMSE, because it has units (e.g., deg C, kW), gives a sense of scale to the reader. For example, an RMSE of 0.25 kW gives the reader a sense that the model has an uncertainty, by one definition, of 0.25 kW.

- **RMSE, Percentage of Mean**: also known as normalized RMSE (NRMSE), this is the RMSE normalized by the mean of the measured data points. This is useful with measures such as power, where the magnitude of the value is intuitively meaningful. For example, if an RMSE of 0.25 kW is reported without knowledge of the mean, it is unclear to the reader whether this is a relatively large uncertainty. If the mean is equal to 1.0 kW, then the normalized value is 0.25/1.0 = 0.25 = 25%, which indicates a large uncertainty in the model. If the mean is 100 kW the normalized value is 0.25%, which represents a very small uncertainty. For a measure such as room temperature, the NRMSE can be misleading. For example, if the RMSE is 1.0°C and the mean temperature of the sample is 10.0°C, then the NRMSE = 1.0/10.0 = 0.1 = 10%. If the mean temperature is 20.0°C then the NRMSE is 5%. On its face, an NRMSE of 10% appears "twice as bad" as an NRMSE of 5%, but for a building's temperature control it may be that predicting the temperature within 1.0°C, regardless of the value of the set point, is the goal and therefore the RMSE is the only relevant statistic. In addition, simply using different temperature units changes the NRMSE (Table A-1):

Table A-1. Different Temperature Units Changing NMRSE

Units	Mean	RMSE	NRMSE
Celsius	20.0	1.0	5.0%
Kelvin	293.15	1.0	0.3%
Fahrenheit	68	1.8	2.6%

For these reasons, we have only reported NRMSE where it provides an intuitively meaningful result.

- **Mean Bias**: this is the difference between the mean of the measured points and the mean of the modeled points. Because RMSE is by definition always a positive number, a set of data whose modeled values are typically greater than the measured values can have the same

RMSE as a set of data whose modeled values are typically less than the measured values. The mean bias shows the difference between these two examples; the former would have a positive mean bias, whereas the latter would have a negative mean bias.

- **Coefficient of Determination (r^2)**: describes how well the modeled points match the measured points. A value of 1.0 would indicate a perfect match of modeled to measured data.

- Measured vs. Modeled: Linear Regression

  1. **Coefficient of Determination (R^2)**: describes how well the modeled points match the points predicted by a linear regression of the modeled and measured data. A value of 1.0 would indicate a perfect match of modeled to measured data.
  2. **Slope**: slope of the line predicted by a linear regression of modeled and measured data (modeled = intercept + slope X measured). This is a useful statistic for understanding if the model has a bias dependent on the magnitude of the measured value. A slope of 1.0 indicates no value-dependent bias, whereas a slope greater than 1.0 indicates that the model overpredicts at high values and underpredicts at low values.
  3. **Intercept**: slope of the line predicted by a linear regression of modeled and measured data (modeled = intercept + slope X measured). This value indicates the bias at a measured value of 0.0.

Infiltration Calibration

Early in the project (December 2015), a tracer gas decay test was performed on the Alabama house over a period of several days. The intention was to use this period to calibrate the steady-state heat-loss parameters of the building. Without a measurement of the actual infiltration rate, it is difficult to compare modeled-to-measured heating energy and know which portion of the difference is due to uncertainty or oversimplification in modeled infiltration and which is due to uncertainty in conduction losses of the building envelope. The idea is that by directly measuring infiltration over a period, one can then set the infiltration rate to the measured value in the model and perform a calibration on the envelope losses. This appendix overall details the process of calibrating the envelope characteristics to match measured heat losses, and this section of the appendix deals with calibrating the infiltration model.

Infiltration rate is a notoriously difficult building characteristic to calibrate. The models available tend to be either very simple (e.g., Sherman Grimsrud (ASHRAE 1997)) or very complex (e.g., CONTAM). A blower door/tracer gas test results in an average infiltration rate but does not provide details of how local building geometry impacts wind speed, which in turn drives infiltration. Blower door/tracer gas tests give a whole-house value of infiltration rate but do not address the individual sources of infiltration. For instance, it is not possible to determine the relative amount of infiltration entering around windows, through the ceiling from the attic, and through the joints between walls and floors. Furthermore, the infiltration measurement addresses infiltration in the conditioned space; other tests would have to be done in order to estimate infiltration into the attic and crawlspace.

It would have been a relatively simple matter to use blower door test results to set coefficients for the Sherman-Grimsrud infiltration model. However, this model does not allow for different rates of infiltration into different parts of the building (attic, crawlspace, conditioned space) and also does not provide an estimate for interzonal air transfer.

CONTAM, the tool selected to simulate air infiltration (and interzonal airflow) takes the leakiness factors for a host of airflow paths and uses them to come up with a whole-house infiltration rate. However, because CONTAM's estimate is assembled from the contributions of dozens of sources, there are essentially an infinite number of combinations of input data that can result in a given whole-house value, such as the one that had been measured by the tracer gas test.

The direct measurement of infiltration rate was used as a guide in setting up a CONTAM infiltration model. The infiltration rate measured using the tracer gas decay test was used to set CONTAM leakiness factors for the walls, doors, windows, ceilings, and floors of the conditioned space. Individual leakiness factors were adjusted during later calibration of the conditioned space as discussed in this appendix.

During the 18-hour test period, the measured infiltration rate was about 0.15 ach (Figure A-1).

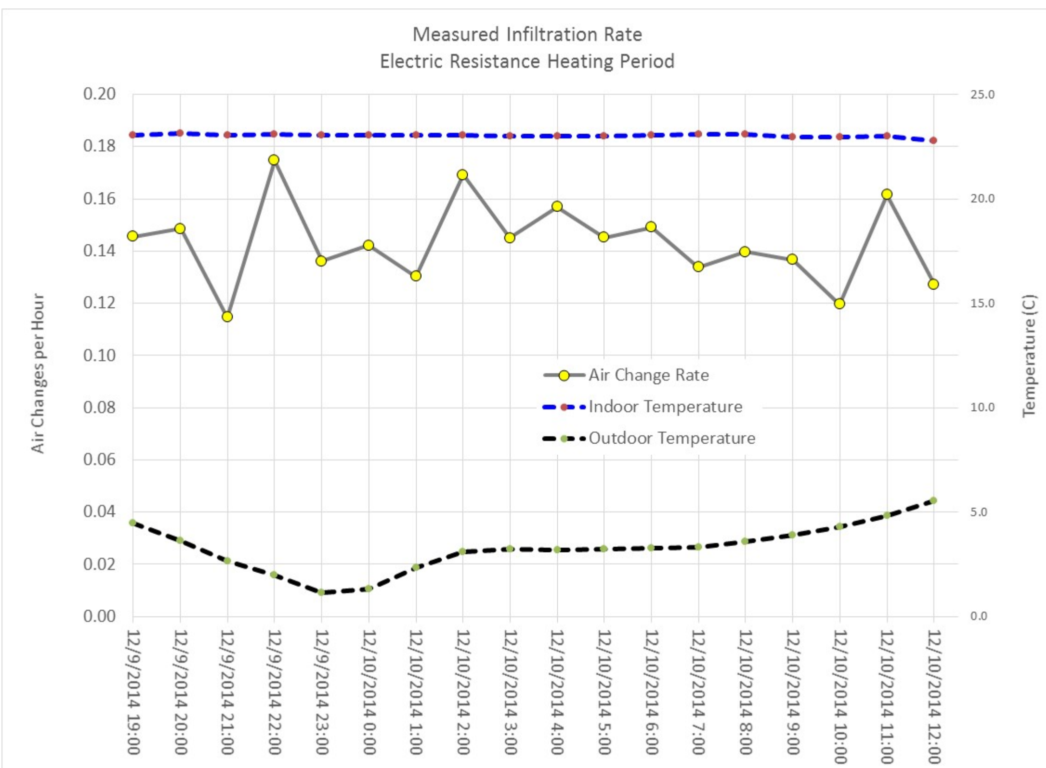


Figure A-1. Measured infiltration rate, using the tracer gas decay method, during an 18-hour period

Attic Calibration

Attic spaces are subject to energy transfer through the insulation to the space below; through the end walls and roof to ambient air; by infiltration of ambient air; by infiltration of conditioned air from below; by short wave solar gains onto the roof; and by long wave radiation exchange between the roof and sky.

In this calibration, the temperature of the conditioned space below was forced to be equal to the measured values, and the model was driven with measured ambient conditions for a 20-day period starting April 1. The interior dry-bulb temperature of the attic was used as a figure of merit to improve the RMSE and coefficient of determination between measured and simulated results. Parameters adjusted to tune the model were the solar absorptance and long wave emissivity of the roof surface, the zone thermal capacitance, the ceiling insulation depth, and the equivalent leakage area (Table A-2).

Table A-2. Attic Space Calibration Parameters

Parameter	Best Guess	Limit (Minimum)	Limit (Maximum)	Calibrated Value
Solar absorptance	0.60	0.50	0.90	0.78
Roof emissivity	0.90	0.50	0.99	0.70
Thermal capacitance multiplier *	5.0	0.5	15.0	5.0
Ceiling insulation depth (cm)	38.0	20.0	40.0	30.5
Equivalent leakage area (cm^2/m^2) **	1.0	0.1	10.0	0.4

* Models must account for the thermal capacitance of items such as furnishings, appliances, etc. (or in this case, the exposed roof trusses) contained in rooms of a building in order for the room's air temperature to react in a realistic way to external changes. Manual calculation of the thermal capacitance would involve making an inventory of all items in each room and determining its effective volume, specific heat, and density. The "thermal capacitance multiplier" is a commonly used simplification in which the thermal capacitance of the air in the room is multiplied by a user-selected value in order to account for the additional thermal capacitance of the items in the zone. It presumes that the zone air and zone contents are at the same temperature.

** The equivalent leakage area is expressed in units of cm^2/m^2. Equivalent leakage area expresses how easy it is for air to pass through a material or (in this case) through part of a building facade. It describes the size of a hole (in cm^2) in a fictitious perfectly impermeable unit area of facade (in m^2) that is equivalent to the permeability of the actual facade.

Table A-3. Attic Space Calibration Statistics

Statistic	Pre-Calibration	Post-Calibration
Measured mean temperature (C)	22.3	
RMSE, temperature (C)	2.24	2.16
Mean bias (C)	0.50	0.06
Coefficient of determination (r^2)	0.949	0.953
Measured vs. Modeled Temperature: Linear Regression		
Coefficient of determination (R^2)	0.952	0.953
Slope	0.943	0.947
Intercept (C)	1.775	1.127

TRNOpt/GenOpt was used to simultaneously adjust the values in Table A-2 in order to minimize the RMSE between measured and modeled attic air temperature. Parameter values were restricted to be between the minima and maxima shown in Table A-2. Statistics before and after calibration are shown in Table A-3. The observed mean temperature over the period was 22.3°C (min. -4.1°C/max. 52.4°C).

The two plots in Figure A-2 show data for the 20-day period over which the attic model was calibrated: temperature as a function of time and simulated vs. measured temperatures from the pre- and post-calibration models.

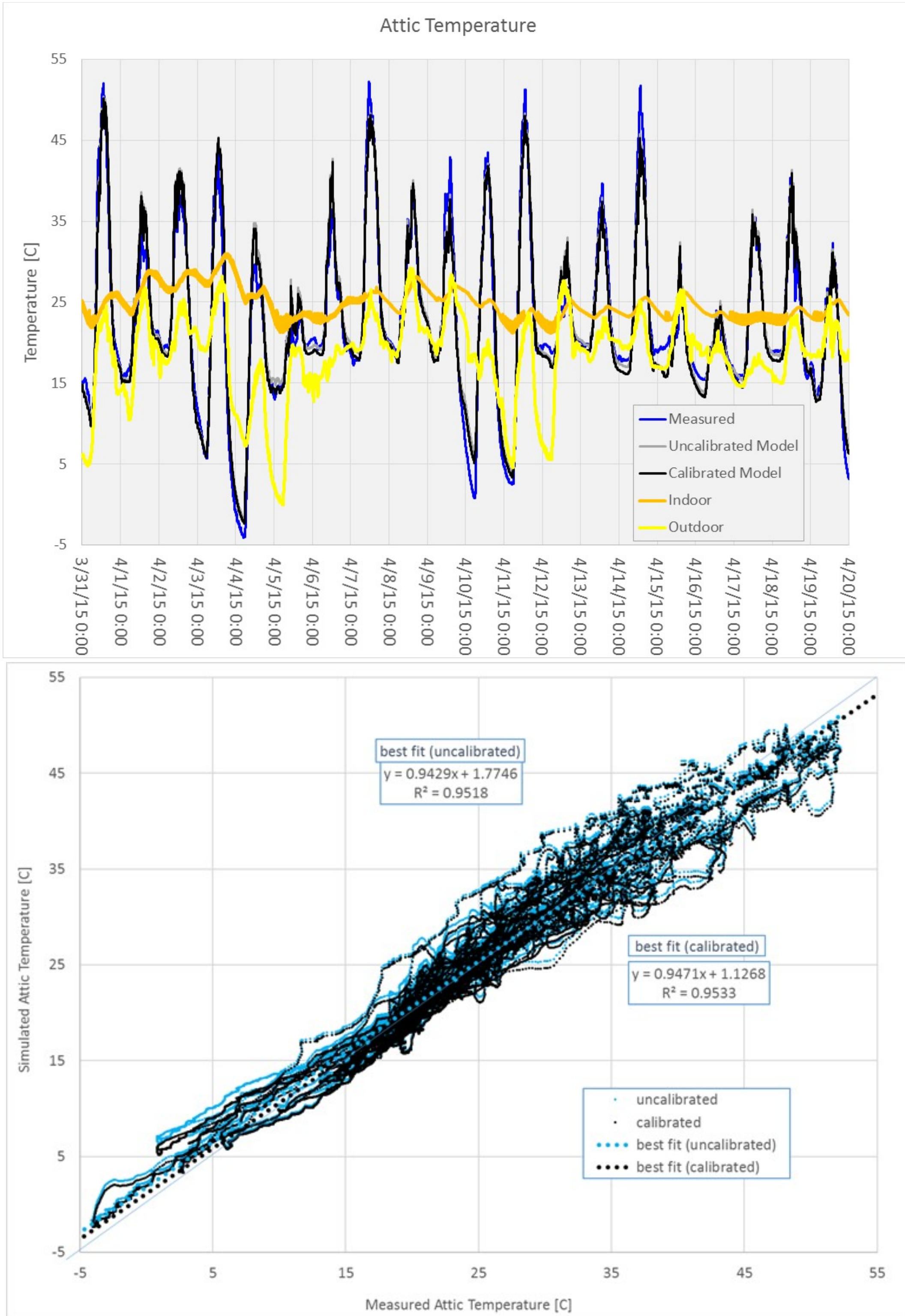


Figure A-2. Simulated and measured attic temperatures for the 20-day attic calibration period

Each point represents the average temperature over a 1-minute period. Even in the calibrated model, there appears to be a slight bias in which lower temperatures are overpredicted while higher temperatures are underpredicted.

It is important to note that the R^2 values shown on the charts (and in Table A-3, no background shading) are not the same r^2 value that is shown in Table A-3 with the light green background shading. The green r^2 value in Table A-3 describes how well the collection of modeled points fit the measured points, whereas the R^2 value shown in Figure A-2 indicates how well the linear regression equation fits the plotted data. Another way of thinking about it is that the green r^2 value in Table A-3 indicates how far away the point cloud lies from the 1:1 correspondence line (x=y), while the R^2 value in Figure A-2 indicates how far away the point cloud lies from the best fit trend line through the point cloud. If the best fit trend line were y = x then one would expect that the green r^2 value in Table A-3 would be the same as that in Figure A-2.

Crawlspace Calibration

The crawlspace underneath the residence is subject to energy transfer through the insulation to the conditioned space above (conduction and infiltration), by conduction to the ground below, and through the vinyl skirting to ambient conduction and infiltration. Short wave radiation gains and long wave radiation exchange are also accounted for in the model but do not contribute significantly to the energy transfer due to localized shading by nearby objects.

For the crawlspace calibration, the temperature of the conditioned space above was forced to be the measured values, and the model was driven with ambient conditions for the same 20-day period in April. The dry-bulb temperature of the crawlspace was used to generate RMSE and R^2 coefficient of determination between measured and simulated results.

The crawlspace proved to be challenging to calibrate for two reasons. First, its primary connection to ambient conditions is through a “ventilated vinyl skirting” product that has a very low thermal resistance and a comparatively high air infiltration rate. Published data, either from the product manufacturer or from literature, were not available to give baseline values from which to start a calibration. Initial values had to be estimated with little more than guessing. While the thermal resistance and infiltration rate were then tuned, it was not easy to cleanly separate the two from one another because the product also has such low thermal mass. Consequently, the time constants of the crawlspace to changes in wind-driven air infiltration and ambient temperature are essentially equal.

Second, the temperature of the ground surface and of the soil directly below the crawlspace has a very strong impact on the air temperature in the crawlspace. TRNSYS includes both simplified and highly detailed ground temperature models. The highly detailed model relies on a finite difference approach to solve soil node temperatures extending away from the building in all three dimensions. Unfortunately, the model is slow to solve and requires a significant amount of run-up computation if initial values are not to dominate the results. One of the simpler ground temperature models (Kasuda) estimates ground temperature only as a function of time of year and depth (i.e., it does not take into account energy transferred to the ground from the building) but solves quickly. Both models require knowledge of the soil thermal properties (density, specific heat, and thermal conductivity).

Both ground models were tested. The Kasuda model proved to be unsuitable unless its predicted temperature was combined with ambient temperature to create a weighted average temperature. The problem was that the weighted average temperature, which was developed for the Alabama house and for the 20-day period in April, did not hold for other times of year or for the calibration of the New Jersey home.

The finite difference model was run using measured ambient conditions as well as measured crawlspace temperatures for a (simulated) period of 6 months so as to obtain an estimated profile of temperatures in the soil beneath the crawlspace during the April calibration period. That temperature profile was then used as the boundary temperature for the “floor” layer of the modeled crawlspace. The floor layer was taken to be a 20-cm-thick layer of soil with a known boundary temperature behind it.

TRNOpt/GenOpt was used to simultaneously adjust the values in Table A-4 in order to minimize the RMSE between measured and modeled crawlspace air temperature. Parameter values were restricted to be between the minima and maxima shown in Table A-4. Statistics before and after calibration are shown in Table A-5.

Table A-4. Crawlspace Calibration Parameters

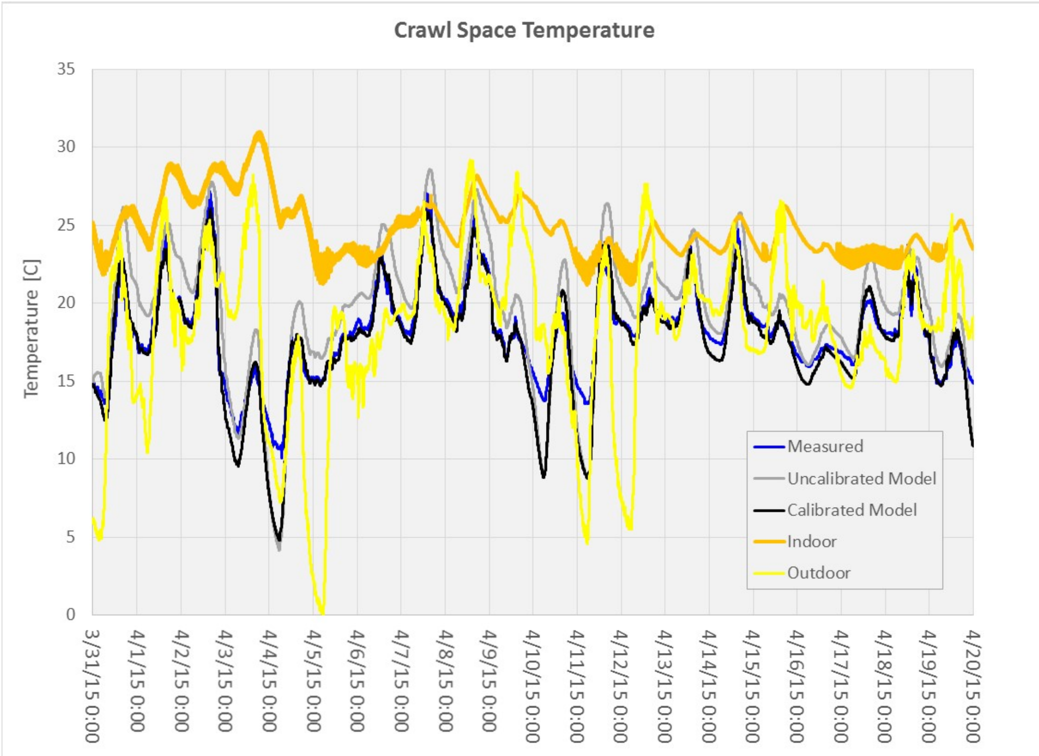
Parameter	Best Guess	Limit (Minimum)	Limit (Maximum)	Calibrated Value
Vinyl skirting R-value (K.m^2/W)	0.106	0.05	2.0	0.881
“Floor slab” thickness (cm)*	0.0	0.0	30.0	10.0
Thermal capacitance multiplier	1.5	0.5	15.00	5.0
Floor insulation depth (cm)	38.0	10.0	40.0	19.0
Infiltration leakage area (cm^2/m^2)	5.0	1.0	500.0	250.0

* The “floor slab” consisted of a layer of soil separating the crawlspace zone from the time-dependent boundary temperature that resulted from running the finite-difference model. It was added to help decouple the crawlspace air temperature from the soil temperature. The “best guess” is listed as 0 because this layer was not part of the pre-calibration model.

Table A-5. Crawlspace Calibration Statistics

Statistic	Pre-Calibration	Post-Calibration
Measured mean temperature (C)	18.2	
RMSE, temperature (C)	2.32	1.30
Mean bias (C)	1.35	0.50
Coefficient of determination (r^2)	0.36	0.80
Measured vs. Modeled Temperature: Linear Regression		
Coefficient of determination (R^2)	0.856	0.921
Slope	1.34	1.21
Intercept (C)	-4.93	-4.34

The two plots in Figure A-3 show data for the 20-day period over which the crawlspace model was calibrated: temperature as a function of time and simulated vs. measured temperatures from the pre- and post-calibration models.



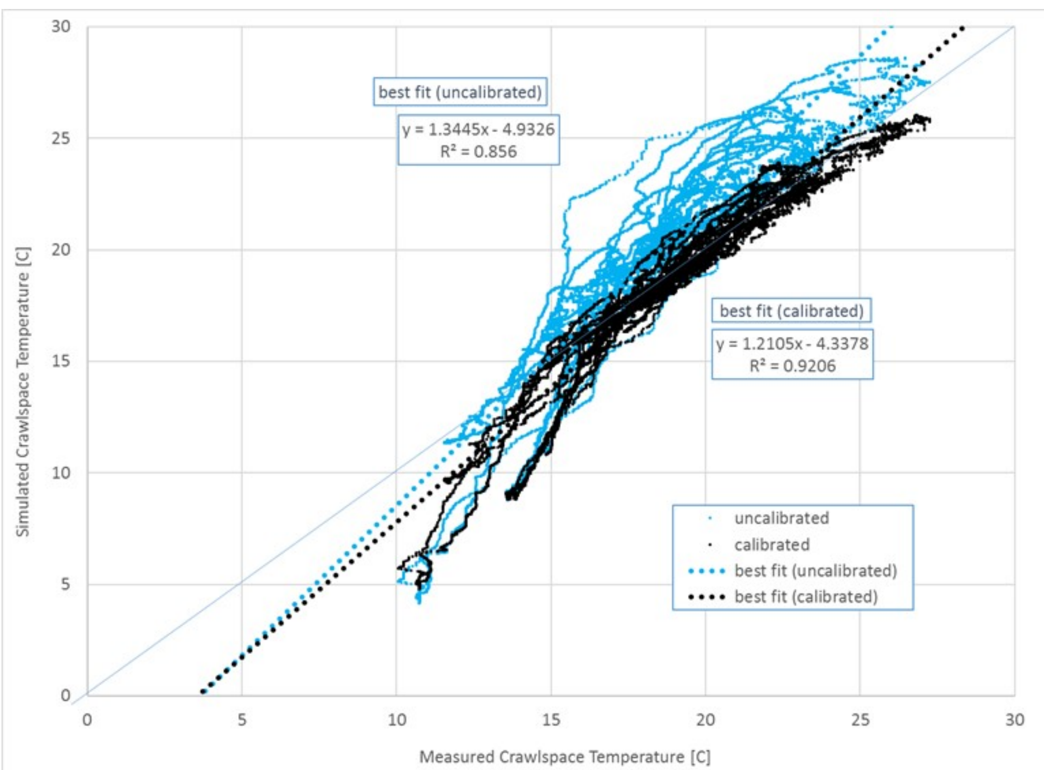


Figure A-3. Simulated and measured crawlspace temperatures for the 20-day crawlspace

Conditioned Space Calibration

For calibration of the conditioned space, a 3-day experiment was conducted in which the house was heated by multiple electric heaters scattered throughout the house, controlled to a fixed temperature set point, in order to maintain a fixed interior temperature in all zones (rooms). During this period, the infiltration rate was measured using the tracer gas decay method. Because we knew (by experiment) the infiltration over that particular period, we could set it in the model and tune only the wall R-value and window properties. Floor and ceiling R-values had already been tuned during the attic and crawlspace calibration processes. The TRNSYS model was run with ambient conditions, crawlspace, and attic temperatures set to the measured values and with the measured conditioned space set point temperatures set to their respective measured temperatures. The simulation generated a curve showing simulated energy consumption by the heaters as a function of the time of day.

Ordinarily, at the end of the 3-day constant-temperature period we would turn off all the heaters at midnight and allow the building to cool. This period would be used to calibrate the modeled heat capacity of the building after the building's heat loss and solar gain parameters had been calibrated. Due to a lack of communication and planning, however, the cool-down test was never performed, and calibration of the building mass was not done.

We had originally intended to adjust window properties and shading factors in order to tune the amplitude of the modeled daytime residence temperature rise on sunny days. We ran early simulations with window properties obtained from the glazing system manufacturer's specification sheet and noted that the amplitude of these rises was quite similar to the measured values. We therefore did not adjust the window properties from their original values. It should be noted that TRNSYS's window model is complex and requires more data than a U-value and solar heat gain coefficient (SHGC). Data such as the number of panes, brand of panes (especially regarding proprietary low-e coatings), gas gap thickness, backfill gas type, and frame material properties are entered into a software tool called WINDOW (Lawrence Berkeley National Laboratory 2019). WINDOW generates a data file that is read by TRNSYS, portions of which is used as the basis of TRNSYS's calculation of dynamic window properties.

Most parameters in a TRNSYS simulation are contained directly in the input file and, because of the way TRNOpt/GenOpt is designed, can therefore be selected as tuning variables in a multivariate minimization of RMSE. Unfortunately, most of the parameters describing the physical properties of the building itself are contained in a separate file and therefore cannot be accessed by GenOpt. As a consequence, the tuning variables used in calibrating heating energy consumption had to be modified manually, and the minimization of RMSE could not be as rigorous as it otherwise might have been. Power consumption of the heaters was used as the figure of merit, and the effective thermal conductivity of the insulated stud cavities was used as the primary tuning variable.

It should be noted that subsequent to having run this manual calibration, we developed an Excel-based tool for performing a multivariate minimization. Due to budget constraints, we have not been able to rerun this part of the calibration using the automated tool.

Calibration parameters and statistics for the conditioned space are shown in Table A-6 and Table A-7.

Table A-6. Conditioned Space Calibration Parameters

Parameter	Best Guess	Limit (Minimum)	Limit (Maximum)	Calibrated Value
Effective R-value of walls (m^2-C/W)	2.51	2.0	4.0	2.47
Window shading factor	1.0	0.0	1.0	1.0
Equivalent leakage area (walls) (cm^2/m^2)	1.0	0.1	10.0	4.0
Wind speed multiplier (0.1)	0.36	0.0	1.0	1.0
Window/wall joint open area (cm^2/m)	2.0	0.5	5.0	2.5

Table A-7. Conditioned Space Calibration Statistics

Statistic	Pre-Calibration	Post-Calibration
Measured mean heating energy (kW)	2.00	
RMSE, heating energy (kW)	0.67	0.25
RMSE, percent of mean	33.7	12.5
Mean bias (kW)	0.63	0.10
Coefficient of determination (r^2)	-0.65	0.77
Measured vs. Modeled Power: Linear Regression		
Coefficient of determination (R^2)	0.864	0.821
Slope	1.064	0.905
Intercept (kW)	0.506	0.089

The two plots in Figure A-4 show data for the 24-hour period over which the conditioned space model was calibrated for heat loss and solar gain: electric heating power as a function of time and simulated vs. measured power consumption from the pre- and post-calibration models.

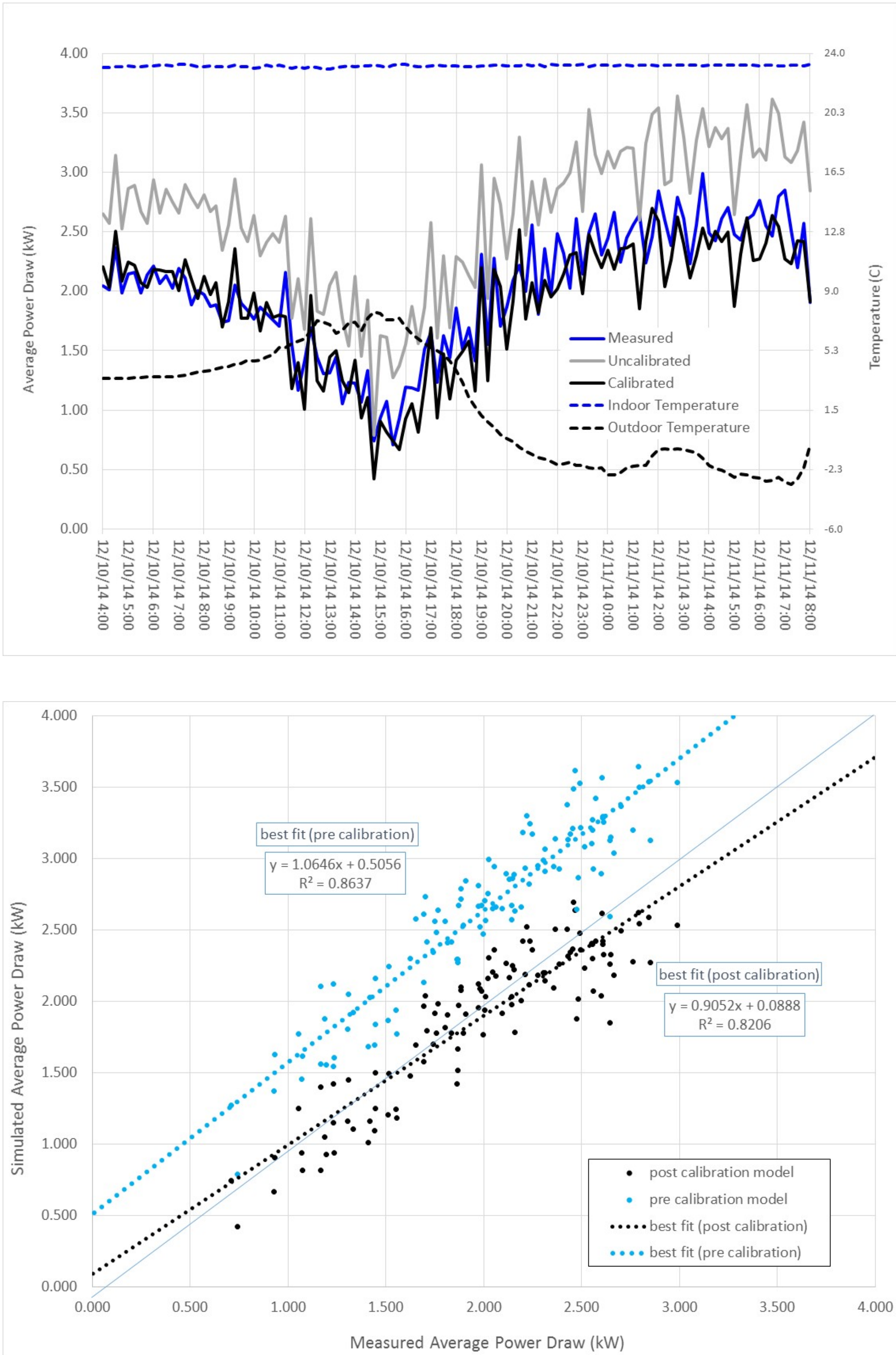


Figure A-4. Simulated and measured heating power for the 24-hour electric resistance heating period

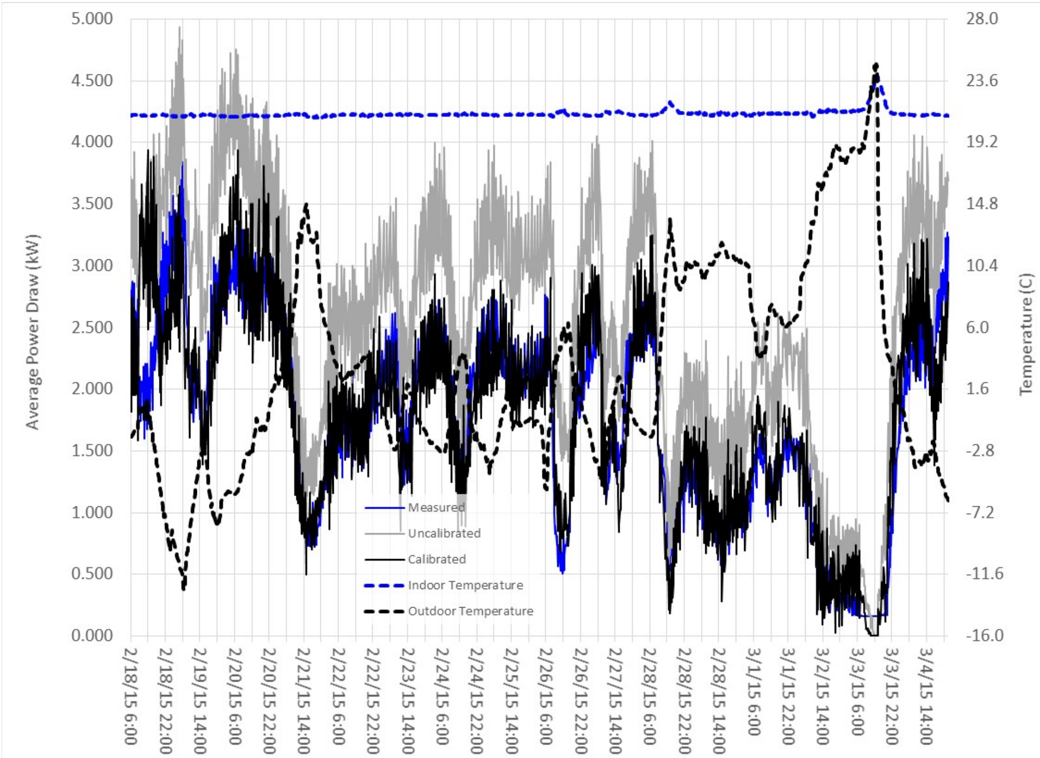
Check of Whole-House Model Calibration

As a way of double-checking the model calibration, a different period of time was selected, and the model was rerun for a 16-day period in February 2016. In this case the attic and crawlspace temperatures were allowed to be calculated by the model instead of being set equal to their respective measured values. The conditioned space temperature, however, was set equal to the measured value so that the model could compute the amount of heating energy required to maintain that temperature. It should be noted that the house was again put into its electric heating mode, and the heat pump was not run during the period. Also, the model was not recalibrated during this check. The results were generated by modifying only the simulation time period and rerunning the pre- and post-calibration models for the attic, crawlspace, and conditioned space.

Table A-8. Whole-House Calibration Check Statistics

Statistic	Pre-Calibration	Post-Calibration
Measured mean heating power (kW)	1.78	
RMSE, heating power (kW)	0.88	0.37
RMSE, percent of mean	50.1	21.2
Mean bias (kW)	0.79	0.03
Coefficient of determination (r^2)	-0.262	0.773
Measured vs. Modeled Power: Linear Regression		
Coefficient of determination (R^2)	0.850	0.787
Slope	1.16	0.922
Intercept (kW)	0.505	0.160

The two plots in Figure A-5 show data for the February “check” period: electric heating power as a function of time and simulated vs. measured power consumption from the pre- and post-calibration models.



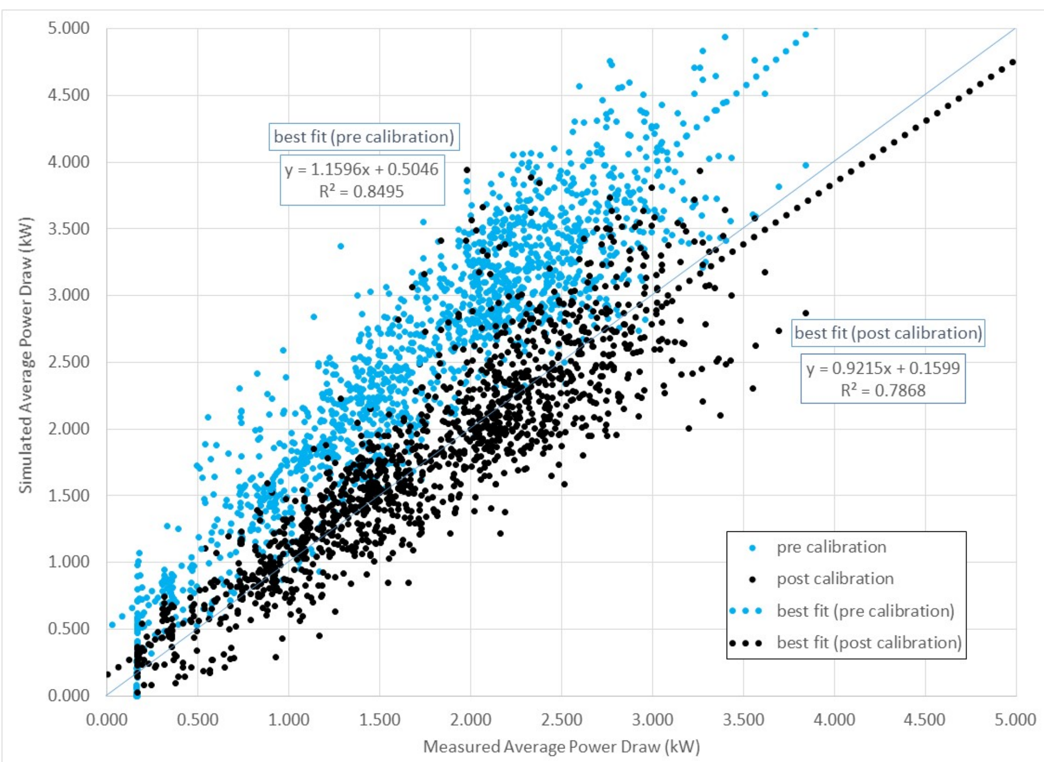


Figure A-5. Simulated and measured heating power for a 16-day period in February

Interzonal Air Transfer Calibration

Perhaps one of the more challenging calibrations carried out was that of the interzonal (room to room) transfer of air. During a 16-day period, December 13–29, the interior doors in the residence were closed and the core (living room/kitchen) temperature was maintained, but the perimeter zones (three bedrooms and the master bath) were allowed to float. The transfer fans that move air from the core to the perimeter zones were on continuously throughout the period.

During times when the bedroom doors were known to be open, the modeled perimeter zone temperatures matched the measured perimeter room temperatures reasonably well. During the periods when the bedroom doors were closed, however, the model predicted appreciably lower perimeter room temperatures than had been measured. The challenge of calibrating the model was to find appropriate adjustment factors that would impact only the times when the doors were closed. Fortunately, during these closed-door times, the temperature difference between the core and perimeter zones tended to be greater than it was when doors were open.

The most significant degree of improvement was initially obtained by adjusting the flow rates of each of the transfer fans. As a starting point, we had used the fans' rated flow rates. These values had been corroborated by tests; however, the fans had been tested with the doors open, and with doors closed (and thus more resistance to airflow) their flow rates were lower. In early calibration work, we found that the fan effective flow rates had to be set to unrealistically low values (on the order of 20%–40% of tested flow) in order to achieve a satisfactory match between modeled and measured results. In examining the measured data more carefully, it was noted that there was a consistent and measurable vertical temperature stratification in the core of the residence. Temperatures near the ceiling were generally 2.2°C higher than they were at the floor. The fans were located near the floor, and when we used the measured fan intake temperature (rather than the average core zone temperature) as the fan inlet temperature, then correlation between measured and simulated results improved dramatically. The addition of the stratification factor allowed the model to be calibrated without modification of the fan flow rates. In Table A-9, the values in parentheses under the "calibrated value" heading are the fan flow rates that calibrated the model without the stratification factor.

We also increased the air leakiness factor of the closed interior doors from 75 cm^2 to 150 cm^2, and the interior wall equivalent leakage area from 8 cm^2/m^2 to 10 cm^2/m^2. As with the whole-house calibration, it was not possible to use TRNOpt/GenOpt to perform an automatic calibration; fan flow rates can be modified by TRNOpt/GenOpt, but equivalent leakage areas cannot.

Interzonal air distribution parameters and statistics are shown in Table A-9 and Table A-10.

Table A-10. Interzonal Air Distribution Calibration Parameters

Parameter	Best Guess	Limit (Minimum)	Limit (Maximum)	Calibrated Value
Flow rate, transfer fan to MBR (L/s)	70.8	5.0	100.0	70.8 (14.2)
Flow rate, transfer fan to BR2 (L/s)	70.8	5.0	100.0	56.4 (28.4)
Flow rate, transfer fan to BR3 (L/s)	70.8	5.0	100.0	70.8 (70.8)
Flow rate, transfer fan to master bath (L/s)	70.8	5.0	100.0	70.8 (70.8)
Air leakiness factor, each interior door (cm^2)	75.8	25.0	200.0	150.0
Air leakiness factor, interzonal walls (cm^2/m^2)	8.0	1.0	20.0	10.0
Core zone temperature stratification (°C)*	0.0	0.0	2.2	2.2

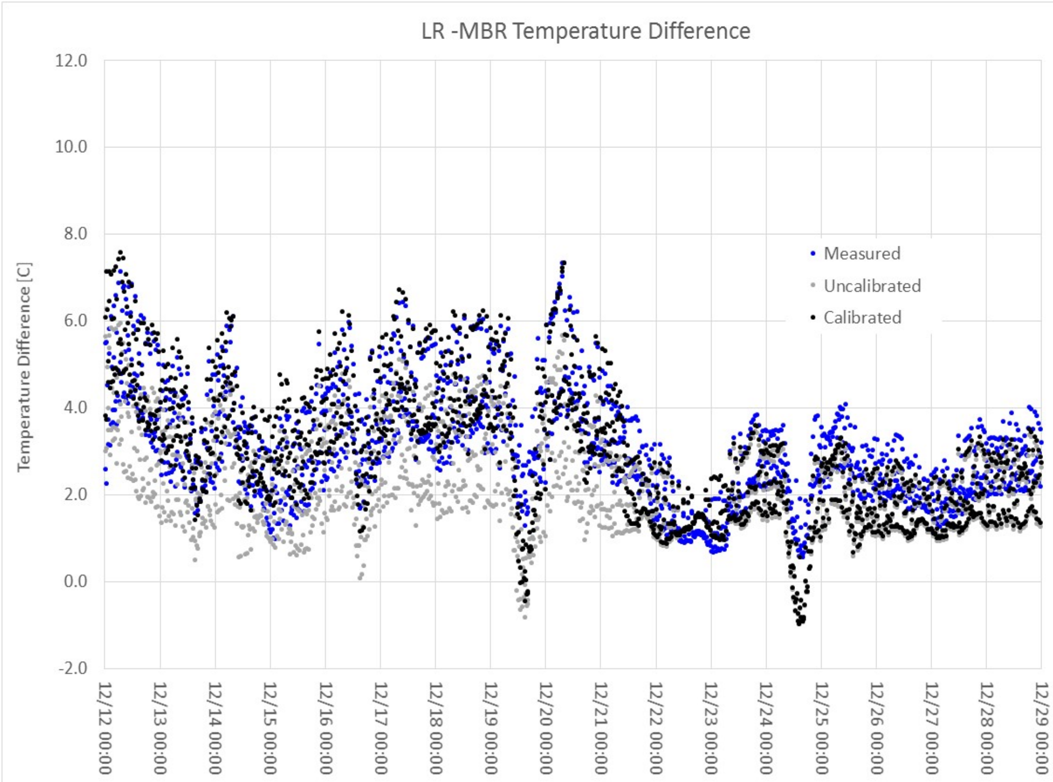
* The temperature stratification was not strictly a calibration parameter in that its value was not varied to achieve better calibration. It was instead a value that came from measurements that was not included in the original (pre-calibration) model, but which was added to the model in the course of the calibration process.

Table A-11. Interzonal Air Distribution Calibration Statistics

Statistic	Pre-Calibration	Post-Calibration
Measured mean temperature difference, LR-MBR (°C)	3.1	
RMSE, temperature difference, LR-MBR (°C)	1.07	0.56
Mean bias, LR-MBR (°C)	0.92	0.25
Coefficient of determination (r^2)	0.31	0.82
Measured mean temperature difference, LR-BR2 (°C)	4.3	
RMSE, temperature difference, LR-BR2 (°C)	1.28	0.71
Mean bias, LR-BR2 (°C)	0.91	0.18
Coefficient of determination (r^2)	0.73	0.92
Measured mean temperature difference, LR-BR3 (°C)	3.4	
RMSE, temperature difference, LR-BR3 (°C)	1.30	1.35
Mean bias, LR-BR3 (°C)	1.00	1.06
Coefficient of determination (r^2)	0.60	0.60
Measured mean temperature difference, living room-master bath (°C)	3.4	
RMSE, temperature difference, living room-master bath (°C)	1.05	0.80

Statistic	Pre-Calibration	Post-Calibration
Mean bias, living room-master bath (°C)	0.92	0.66
Coefficient of determination (r^2)	0.30	0.61
Measured vs. Modeled Temperature Difference, LR-MBR: Linear Regression		
Coefficient of determination (R^2)	0.82	0.90
Slope	0.78	1.09
Intercept (°C)	-0.25	-0.54
Measured vs. Modeled Temperature Difference, LR-BR2: Linear Regression		
Coefficient of determination (R^2)	0.88	0.93
Slope	0.77	1.00
Intercept (°C)	0.05	-0.20
Measured vs. Modeled Temperature Difference, LR-BR3: Linear Regression		
Coefficient of determination (R^2)	0.864	0.932
Slope	0.707	1.00
Intercept (°C)	-0.012	-0.198
Measured vs. Modeled Temperature Difference, Living Room-Master Bath: Linear Regression		
Coefficient of determination (R^2)	0.839	0.878
Slope	0.821	0.905
Intercept (°C)	-0.313	-0.333

Figure A-6 through Figure A-9 show pre- and post-calibration temperature differences between the living room and each of the four other rooms for the December period.



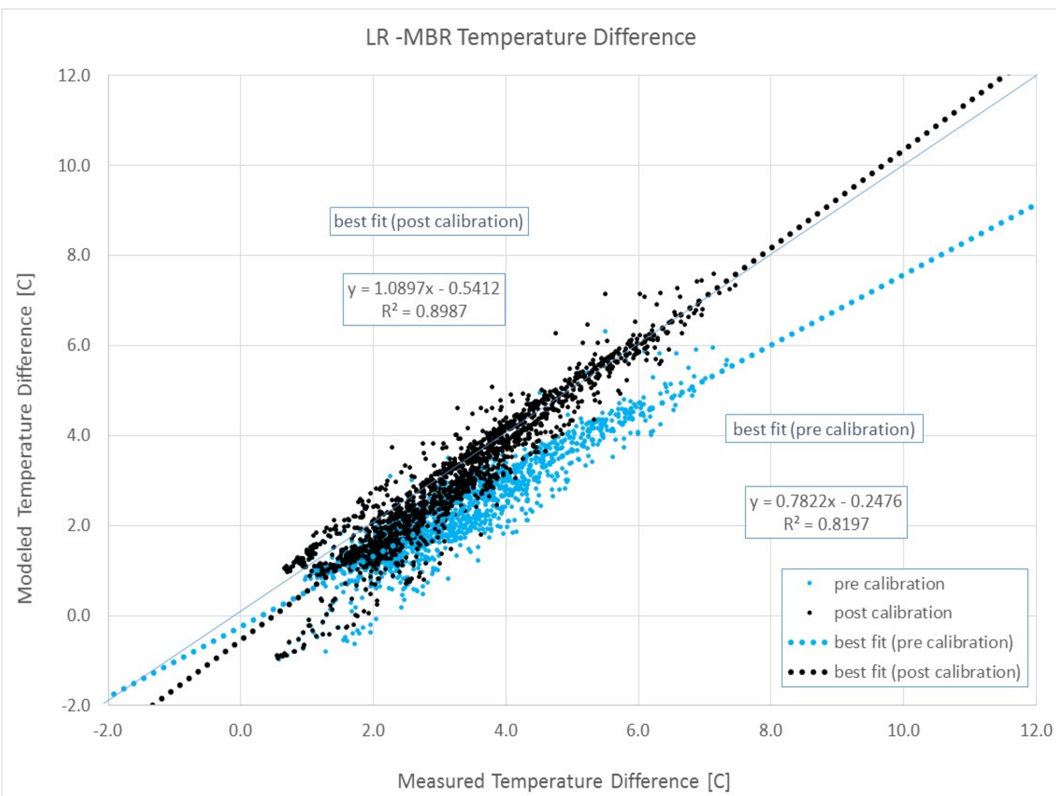


Figure A-6. Pre- and post-calibration living room/master bedroom temperature difference

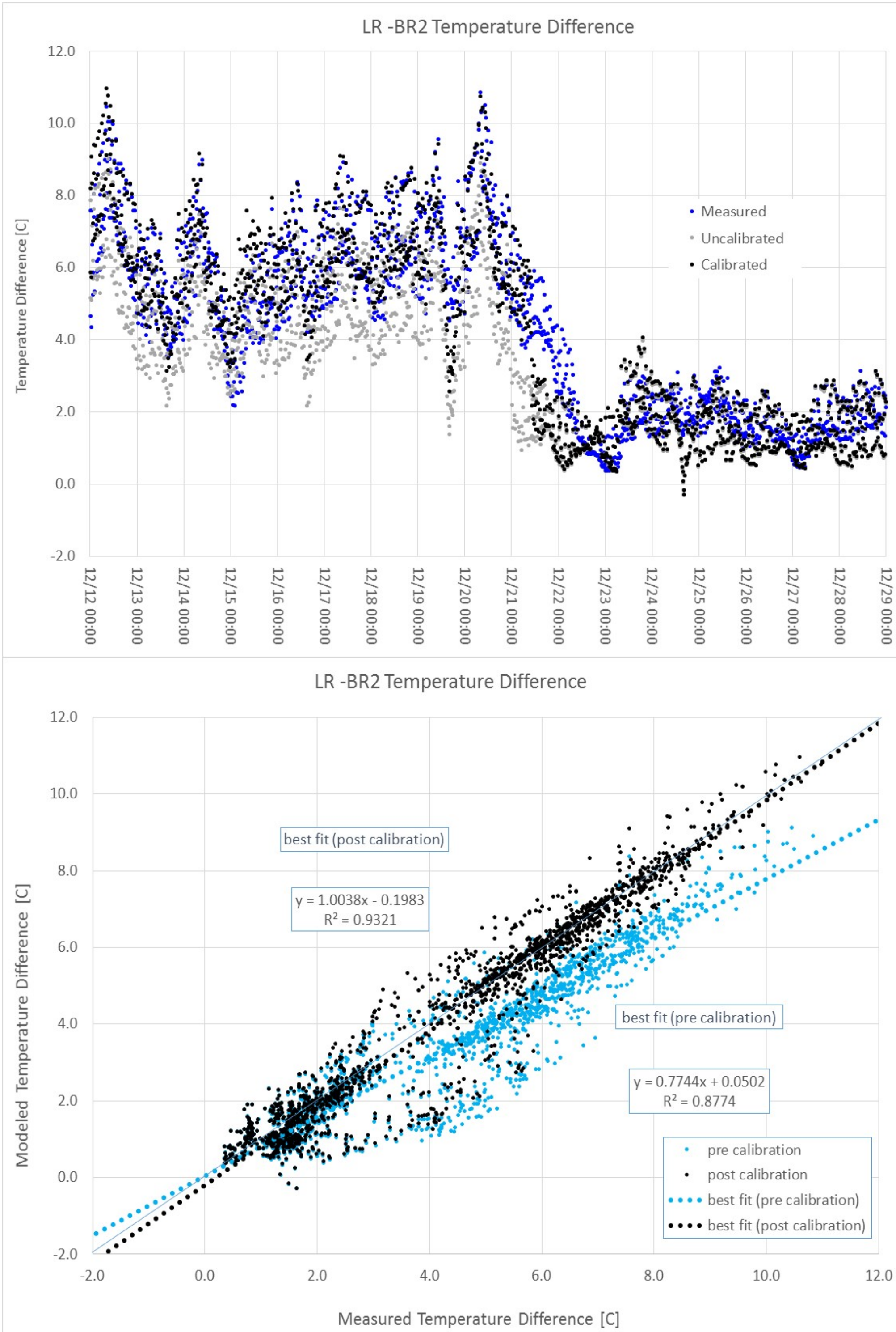
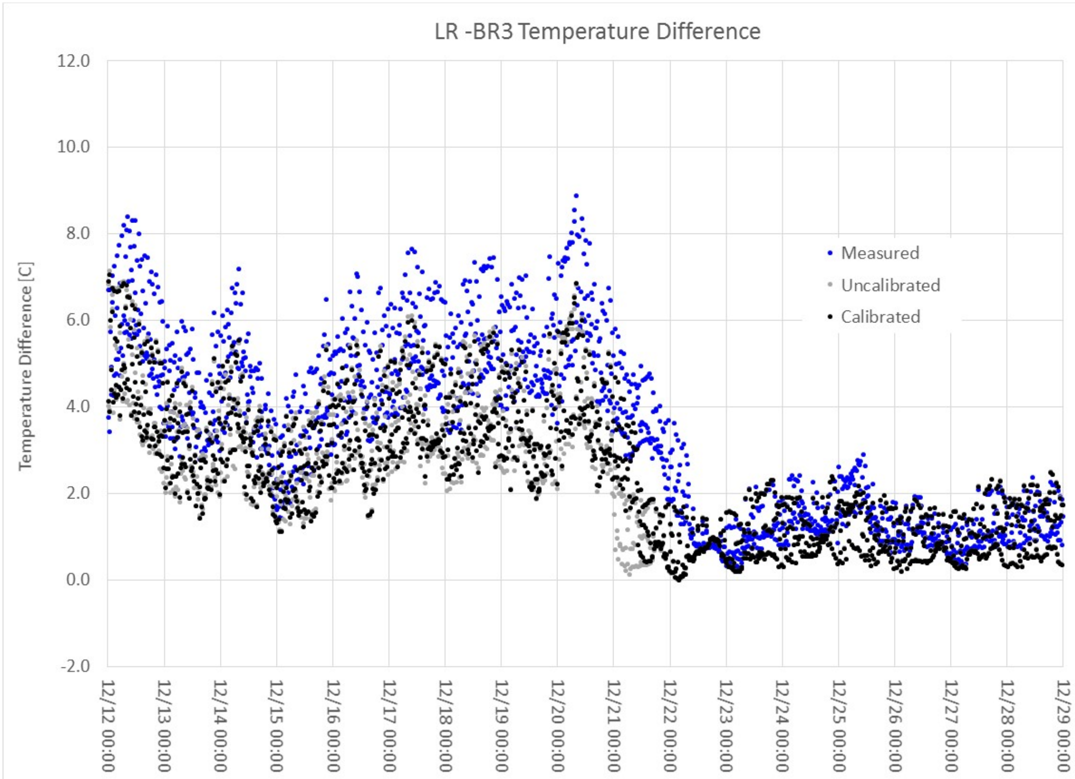


Figure A-8. Pre- and post-calibration living room/bedroom 2 temperature difference



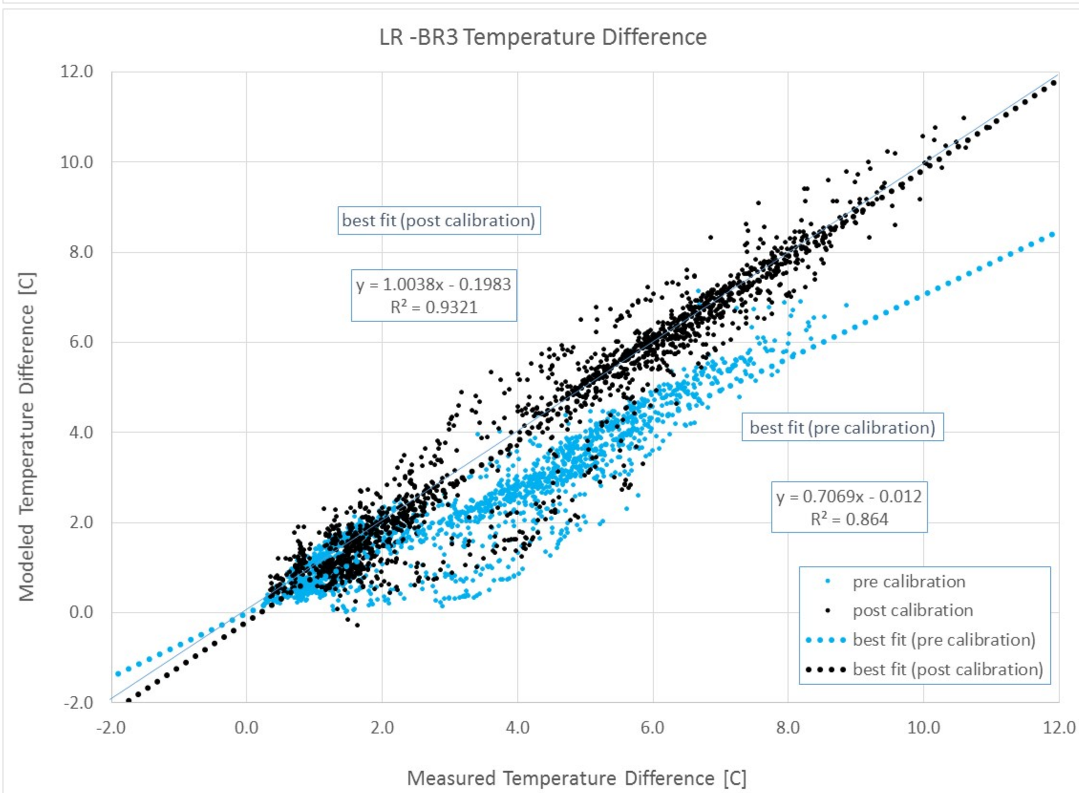


Figure A-9. Pre- and post-calibration living room/bedroom 3 temperature difference

Appendix B. Ventilation System Analysis

This appendix provides the BEopt modeling results for ventilation system options modeled in four climate locations using the Alabama lab home.

Table B-1. Ventilation System Design Simulation Results: Binghamton, NY

Binghamton, NY	Source Energy Consumption (MMBtu/yr)	Whole-House Energy Related Costs, Annualized ($/yr)
Broan XB50	154	1,618
Broan HRV	152	1,677
Whisper Comfort 40 CFM	151	1,638
Whisper Comfort 100 CFM	152	1,690
Air King ES80	156	1,625
Non recovery type: balanced	169	1,784

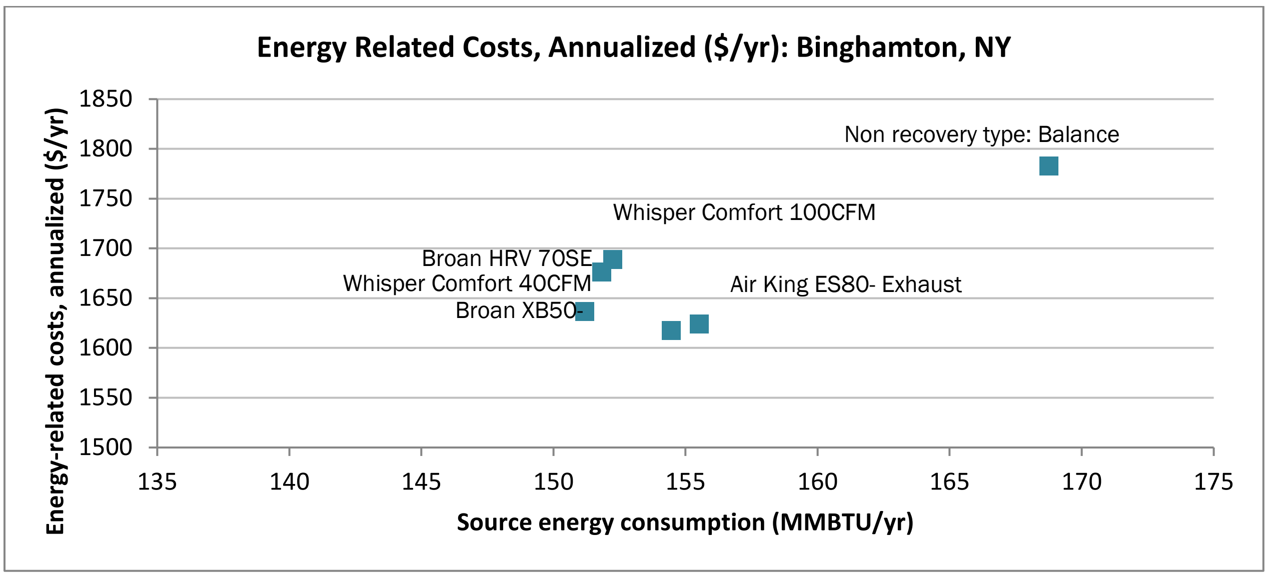


Figure B-1. Ventilation system design simulation results: Binghamton, NY

Table B-2. Ventilation System Design Simulation Results: Mansfield, OH

Mansfield, OH	Source Energy Consumption (MMBtu/yr)	Whole-House Energy Related Costs, Annualized ($/yr)
Broan XB50	151	1,587
Broan HRV 70SE	149	1,652
Whisper Comfort 40 CFM	149	1,613
Whisper Comfort 100 CFM	150	1,664
Air King ES80	152	1,594
Non recovery type: balanced	165	1,747

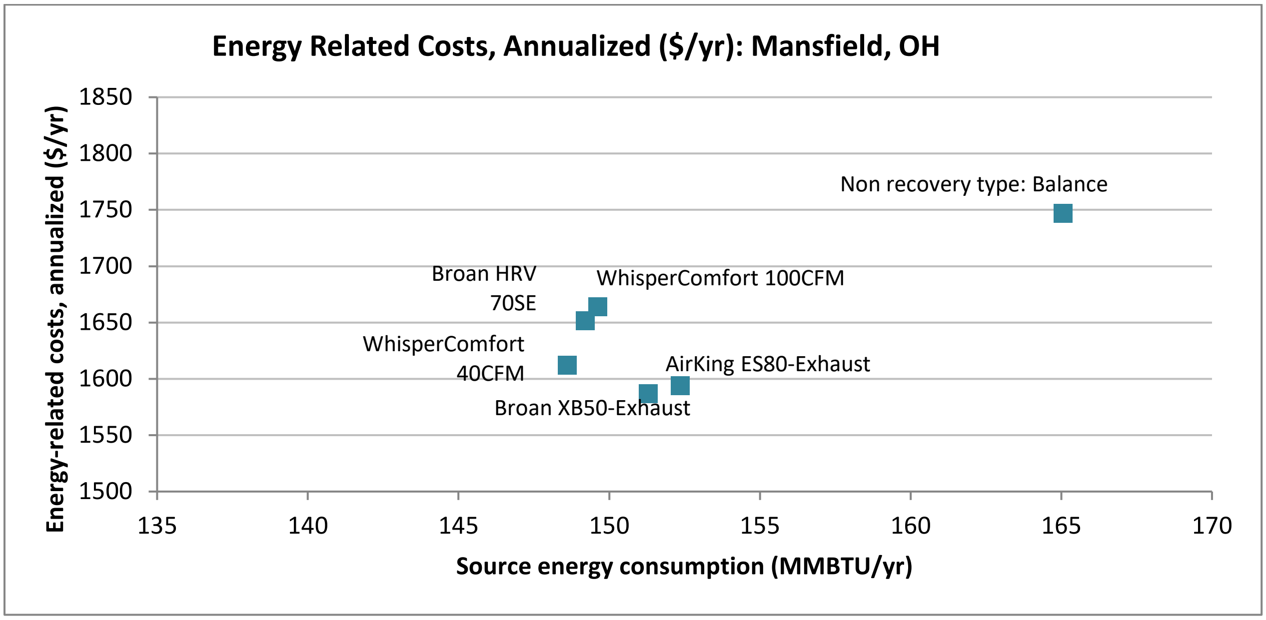


Figure B-2. Ventilation system design simulation results: Mansfield, OH

Table B-3. Ventilation System Design Simulation Results: Harrisburg, PA

Harrisburg, PA	Source Energy Consumption (MMBtu/yr)	Whole-House Energy Related Costs, Annualized ($/yr)
Broan XB50	143	1,501
Broan HRV 70SE	139	1,555
Whisper Comfort 40 CFM	139	1,515
Whisper Comfort 100 CFM	140	1,566
Air King ES80	144	1,508
Non recovery type: balanced	154	1,639

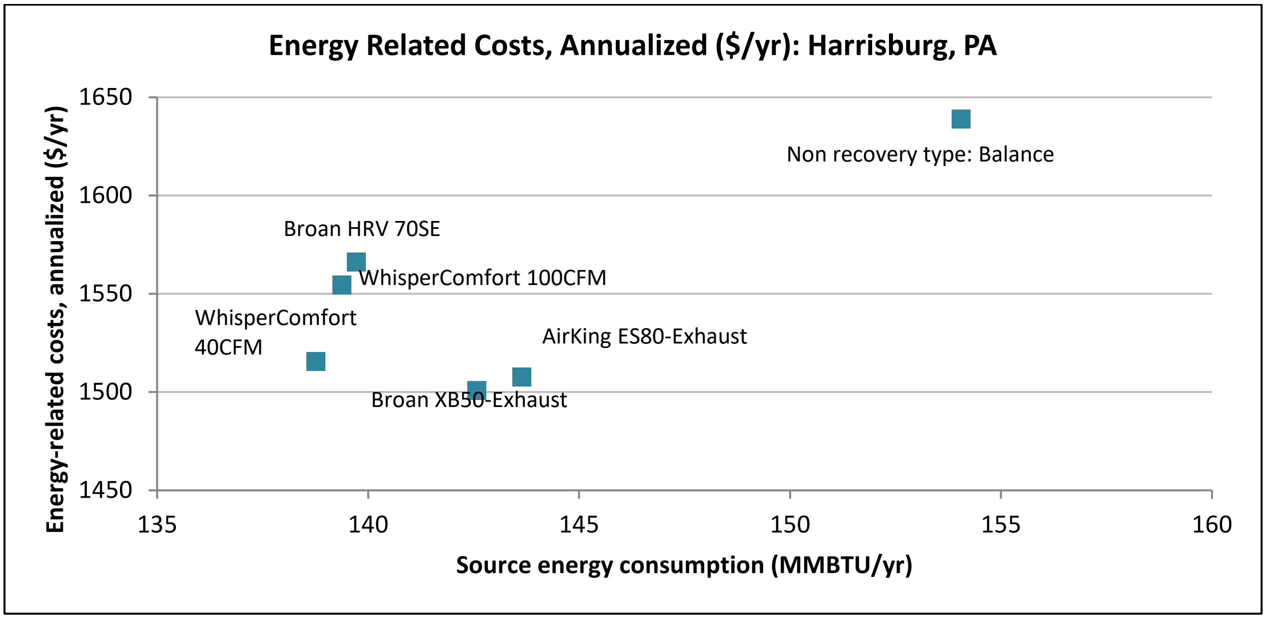


Figure B-3. Ventilation system design simulation results: Harrisburg, PA

Table B-4. Ventilation System Design Simulation Results: Fort Wayne, IN

Fort Wayne, IN	Source Energy Consumption (MMBtu/yr)	Whole-House Energy Related Costs, Annualized ($/yr)
Broan XB50	157	1,641
Broan HRV 70SE	155	1,705
Whisper Comfort 40 CFM	154	1,665
Whisper Comfort 100 CFM	155	1,717
Air King ES80	158	1,648
Non recovery type: balanced	171	1,803

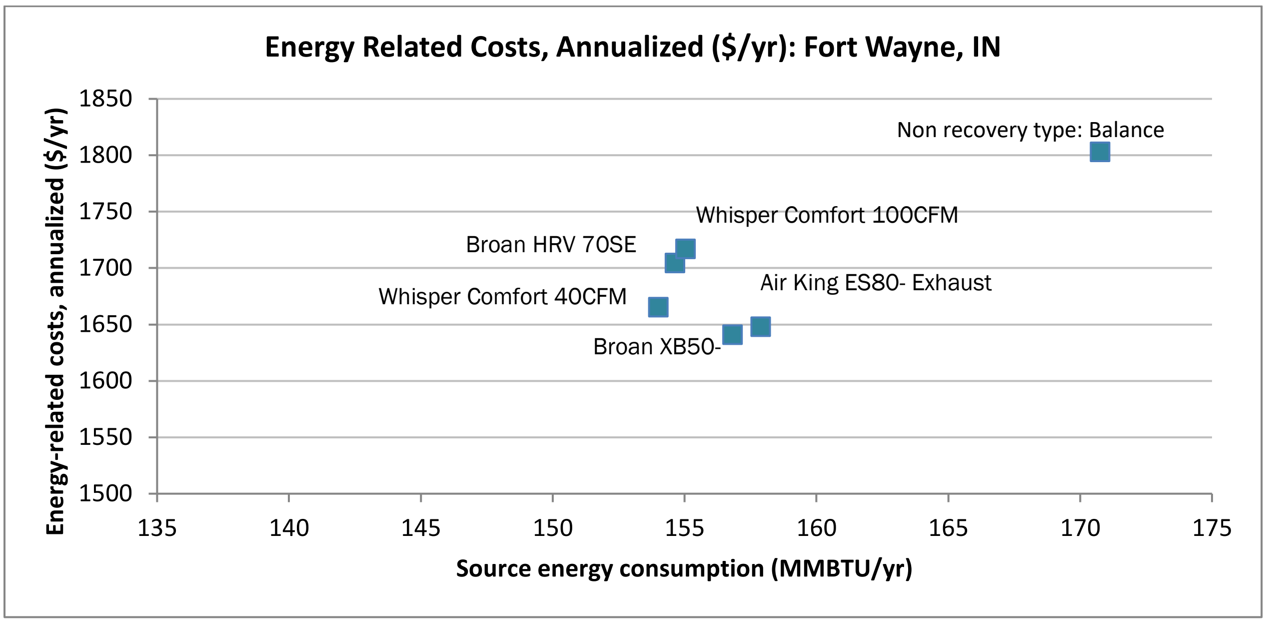


Figure B-4. Ventilation system design simulation results: Fort Wayne, IN

Appendix C. New Jersey Home Production and Installation

The New Jersey test home was produced in the Champion Homes plant in Claysburg, Pennsylvania. This plant typically produces single and double section manufactured (MHCSS) homes on metal chassis and with tape and texture drywall finish. The following summarizes production of the major features of the New Jersey home in chronological order.

Day 1

Figure C-1. Floor of "A" section inverted for electric and plumbing work

As with a typical home in this plant, the floor joists and OSB decking were assembled upside down to allow plumbing and electrical wiring to be more easily installed in and below the floor. At this stage in a standard home, ductwork would also be installed. The additional time spent adding double layers of batt insulation and checking for air-sealing leaks was recouped by not having to install the ductwork. During the production of the floor system, the department was short staffed by one worker but still easily met production rate targets with no extra hours worked.

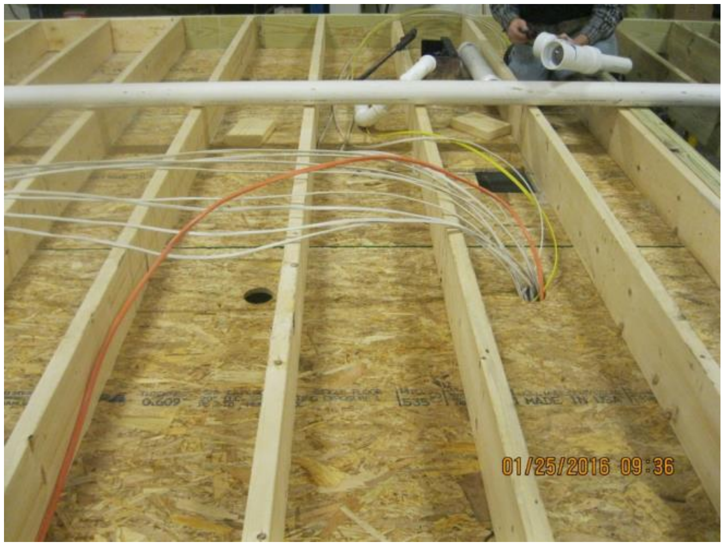


Figure C-2. Typical floor decking penetrations for electrical and plumbing service

After the plumbing pipes and electrical wiring were fed through floor penetrations, workers sprayed expanding foam around the penetrations, a standard task in the construction of the plant’s ENERGY STAR-compliant homes. As can be seen in the pictures below, the expanding foam does not entirely seal the penetrations.

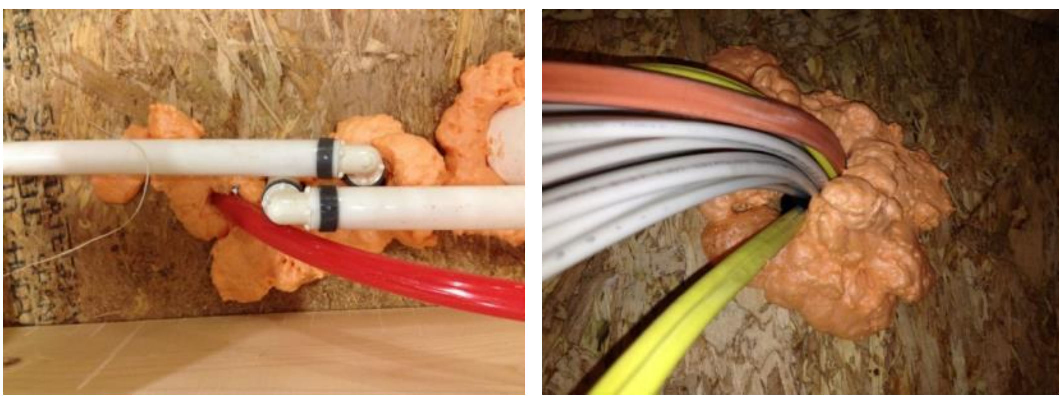


Figure C-3. Expanding foam with air gaps at floor decking penetrations

One layer of 14.5-in.-wide fiberglass batt was laid between joists (standard factory-built homes do not typically get this layer of insulation; the joist cavities are left open). Plastic strapping was stapled to the joists to keep the batts from shifting or falling out when the assembly is flipped back over (specially requested by the project team). An additional layer of 14-ft-wide fiberglass blanket insulation was rolled out atop the joists. This blanket insulation is the typical floor insulation used by the plant. After rolling out the blanket, the outer 3 feet (approximately) on both sides of each home section was sliced along the floor joists and pressed into the joist cavities. This became the “outrigger” section once the chassis was attached to the floor assembly. Once the insulation was installed, the bottom board (a plastic sheet) was rolled out to cover the entire floor.

Figure C-4. Batts being laid into joist cavities

Figure C-5. Plastic strapping to hold batts in place

Figure C-6. Rolling out blanket insulation

Figure C-7. Slicing blanket insulation in outrigger areas

Figure C-8. Rolling out the "bottom board"

From this station on the factory floor, the floor assembly was lifted and placed in the adjacent station, where the chassis and wheels were affixed.

Figure C-9. Completed floor assembly before flipping

Figure C-10. Floor assembly being lifted (refrigerant lines for heat pump are visible under the floor)

Concurrently with the floor deck assembly, the walls were constructed further down the line, as shown in Figure C-11.

Figure C-12. Wall build station

Day 2

On the second day of production, the home began in the flooring station and moved on to the wall set station, where items such as bathtubs, cabinets, and walls were installed on the floor. For the test home, vinyl flooring was laid on the bathroom floor areas of the two house sections only, because the remainder of the floors would receive laminate flooring on-site. Before the vinyl was installed, air-sealing tape was applied to all OSB seams on the floor deck (a custom measure), and through-floor penetrations were sealed on the top sides of the OSB with latex caulk. Electric resistance heat mat with foam underlayment was installed in the master bathroom beneath the vinyl flooring. Because the heat mat had to be custom cut and wires soldered to the electrical contacts, this took extra time and prevented the section from advancing to the next station for approximately 90 minutes. Some time was saved later in the day because the standard heating furnace did not need to be installed. For future applications, the heat mat should be pre-cut with wires pre-connected to speed this process.

Figure C-13. Applying adhesive to edge of bathroom footprint receiving vinyl flooring; note air-sealing tape at OSB floor deck seams

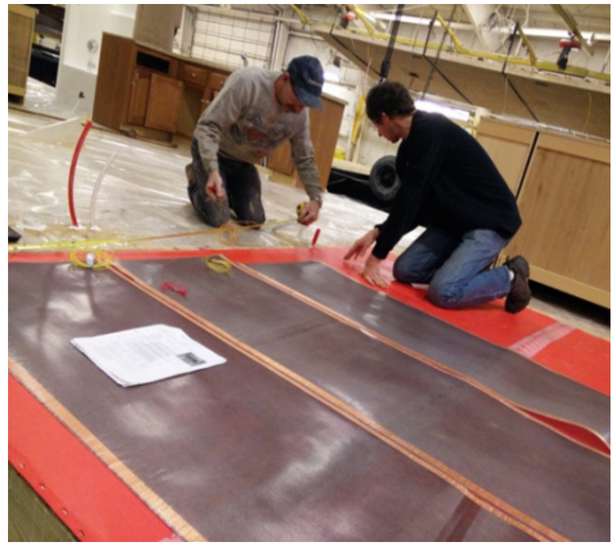


Figure C-14. Laying out and wiring electric heat floor mats in master bathroom

Countertops, sinks, and both interior and exterior walls were installed on the floor decks at the next station. A 4-in.-wide foam airsealing gasket (Owens Corning RimSealR) was fastened to the tops, bottoms, and sides of exterior wall sections before setting them onto the floor deck. This was done in lieu of the standard .5-in.-width adhesive-backed foam gasket the plant normally uses.

Figure C-15. Kitchen cabinets are placed on floor early in production process

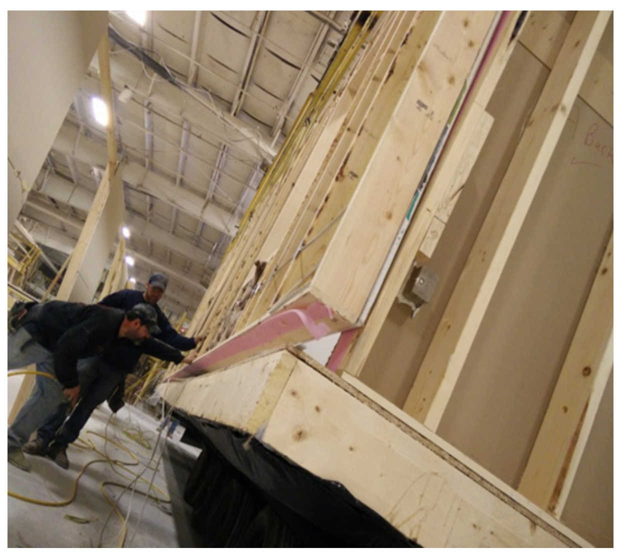


Figure C-16. Wall section is installed with foam gasket along bottom edge/end of adjoining wall

Moisture-resistant cement wallboard was installed in the wall behind the one-piece fiberglass shower instead of standard drywall to comply with ENERGY STAR 3.0 requirements.

Figure C-17. Moisture resistant cement board (purple) used behind tubs

Concurrent with the work done on the main assembly line on day 2 was the fabrication of the roof assembly. The purchaser of the home required a sprinkler system that was installed in the factory by an outside contractor.

Figure C-18. Sprinkler contractors at roof table

Day 3

Day 3 began with air-sealing all exterior wall penetrations (mostly electrical outlet boxes) with caulk and expanding foam. The connection joint between the floor and bottom plate of all exterior walls was sealed with air-sealing tape (Owens Corning HomeSealR).

Figure C-19. Electrical boxes sealed from back with caulk

Figure C-20. View of batt insulation in exterior wall stud bays

Figure C-21. Floor-wall joint sealed with tape; two beads of silicone caulk seal OSB wall sheathing to rim joist

Figure C-22. Owens Corning RimSealR foam gasket is stapled to OSB wall sheathing (left);
Rigid foam insulation is stapled to sheathing and taped with Owens Corning HomeSealR Tape (right)

Figure C-23. Owens Corning representatives trained plant employees on airtight window flashing installation with Owens Corning FlashSealR Tape

As the exterior walls were air-sealed and insulated, electrical and air-sealing work was ongoing at the roof. In addition to the normal wiring that is run above the ceiling, plant employees and research team members routed electrical connections and sensor wiring; the additional wiring caused about an hour delay because the sections could not move forward to the rooftop insulation station until the wiring work was complete. Much of this additional wiring was so that individual fans and other items would be on dedicated circuits for ease of monitoring and would not be required in normal production. Some plant employees stayed for 2 hours after normal closing in order to continue window and some siding installation so that the following day's schedule would not be delayed. The team simultaneously air-sealed the sprinkler ceiling penetrations with ad-hoc rigid foam boxes. Once at the roof insulation station, plant staff was trained to dense pack the blown insulation at the roof eave using a custom-made jig.

Within the house sections, fabrication of the transfer fan soffits and ducting also slowed progress because this was the first experience the plant had in doing these tasks.

Figure C-24. Plant electrical workers route wiring above ceiling

Figure C-25. Ceiling sprinkler penetrations air-sealed with rigid foam boxes and caulk

Figure C-26. Dense-packing blown insulation at eaves

Figure C-27. Rooftop OSB decking laid over vent channel material

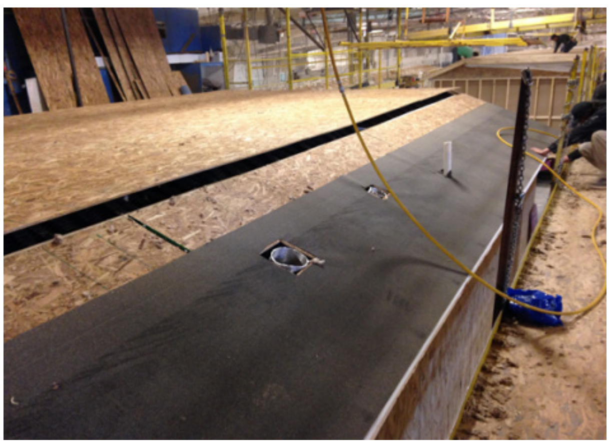


Figure C-28. Roof showing exhaust fan penetrations and self-sealing bituminous membrane along eave

Figure C-29. Transfer fan soffit and duct installation

Day 4

Day 4 was lighter in scope than the prior three days. The transfer fan soffits and ducts were completed and sealed; researcher team members installed door closure sensors in bedroom and bathroom door frames. Standard plant work was completed concurrently, including completion of the window flashing and installation on the second home section, vinyl siding (done with extra-long staples to accommodate the rigid foam insulation outboard of the wall sheathing), and interior electrical and drywall finishing.

Figure C-30. Transfer fan soffit completed except for grille

Figure C-31. Transfer fan viewed from bedroom closet

Figure C-32. Electric resistance backup heater installed in bedroom

Figure C-33. Low-expansion foam being applied as interior window frame seal

Figure C-34. Vinyl siding and soffit installation

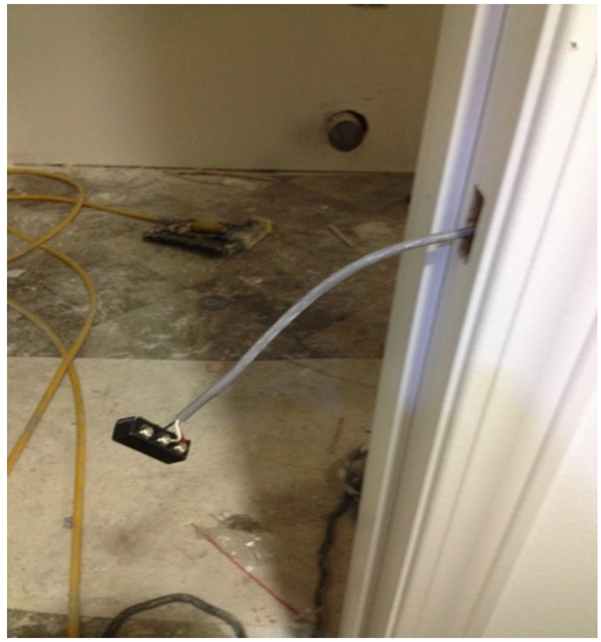


Figure C-35. Partially installed door closure sensor and wiring at doorframe

Day 5-Completion

Remaining work at the plant consisted of interior and exterior finishes and trim, including:

- Interior paint
- Kitchen and bathroom backsplash tiling
- Lighting and ceiling fan installation and other finish electrical
- Finish plumbing
- Appliance installation
- Other interior finish items such as window blinds and closet shelving
- Exterior trim
- Plumbing water testing
- Electrical testing
- Loading ship loose items
- Wrapping the house for shipment.

Figure C-36. Installation at the final site in New Jersey

Appendix D. Occupant Interview Questions

Address:	**Interviewer:**
Date:	**Interviewees:**

Focus the responses on thermal comfort, indoor air quality, energy costs, maintenance of energy, and ventilation-related equipment:

1. How would you rate the overall comfort of the home, compared to previous houses in which you have lived?
   - In winter
   - In summer
   - During swing seasons
2. Please describe the evenness of temperature across the various rooms in the home.
3. Was high humidity ever a problem inside the home? If so, please describe extent, duration, and time of year.
4. What aspects of the home are you most pleased with?
5. What aspects of the home are you least pleased with?
6. How do the utility bills for the home compare with what you expected?
7. What is your awareness and satisfaction level of the following specific features?
   - Heating system
   - Cooling system
   - Ventilation system (indoor air quality)
   - Transfer fan distribution system
   - Controls for mechanical systems
   - Air sealing (drafts)
   - Lighting
   - Hot water system
   - Appliances

8. Compared to your previous residences, what is your opinion of the home with respect to?
    - Street noise
    - Maintenance
    - Durability
    - Transmission of odors between rooms
9. In your opinion, are there non-energy (quality-related) benefits reasons to increase energy efficiency even beyond what can be paid for by energy savings?
10. How did you operate your systems (set points, setbacks, etc.):
    - Heating: what thermostat settings?
    - Cooling: what thermostat settings?
    - Ventilation fans
    - Transfer fans
    - Backup heaters
    - Did you adjust the thermostat set points when the home was unoccupied? Please describe.
11. When did you occupy the house?
12. How many people on average occupied the house?

Appendix E. Regression Model for New Jersey Home

During the simulated occupancy period of the New Jersey home, internal gains (sensible and latent) were implemented to simulate occupancy. The heat pump set point was set at 69°F, and resistance heaters in the bedrooms were set at 68°F. Transfer fans were left on. Resistance heating energy usage and ambient temperature were measured from January through February 2017 and used to generate a regression model based on a power function with ambient temperature as the independent variable and resistance heating energy usage as the dependent variable.

TMY temperature data was then used as the input to the regression model to predict resistance heating energy usage over the full heating season. Considering the maximum capacity of the resistance heaters, hourly site energy usage of the predicted results was capped at 1.5 kW. Measured data points, predicted data points, the regression function, the regression trendline, and R-squared value are shown in Figure E-1. Although the model has a good fit at lower ambient temperatures, the R^2 of 0.315 indicates a poor overall correlation, mainly due to the fact that the regression model assumes the ambient temperature is the only independent variable while omitting other important factors such as the internal and solar gains, which have a greater proportional impact at higher ambient temperatures. The regression model predicted 224 kWh annual site energy use of the resistance heaters, while the load calculation method described in Appendix F estimated 424 kWh. Considering the performance of the regression model, results from the load calculation (Appendix F) were used as input to the as-built model.

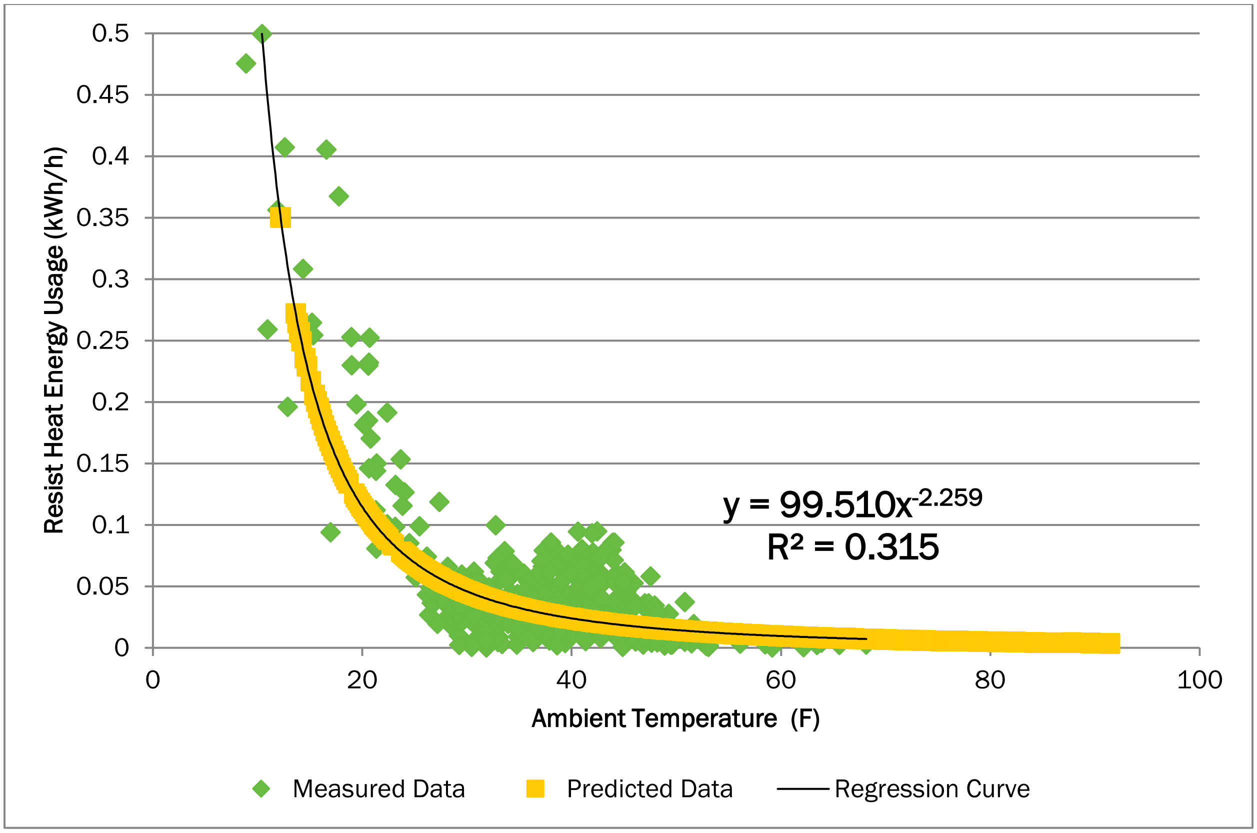


Figure E-1. Regression model for New Jersey home supplemental heating energy

Appendix F. Supplemental Heating Load Calculations

BEopt has no built-in function capable of modeling the impact of the transfer fans on the conditioned space and cannot simulate energy usage of supplemental resistance heaters when used in combination with a primary space conditioning system such as a ductless heat pump. The Maryland and Massachusetts homes had no unoccupied monitoring data available from which to generate projections. Therefore, a load calculation during the heating season was conducted for these homes to estimate the annual supplemental resistance heat needed for the supplemental heating case models.

For each room, the heat loss rate of every surface was calculated, which is based on the equation below:

$$Q = UA\Delta T$$

Where:

Q = the heat loss of the conditioned space (Btu/h)

U = the overall heat transfer coefficient (Btu/(ft^2·F·h))

A = heat transfer surface area (ft^2)

ΔT = difference in temperature (F)

Heat loss rate from infiltration was calculated based on the equation below:

$$Q_{infiltration} = \rho V c \Delta T$$

Where:

$Q_{infiltration}$ = heat loss from the infiltration (Btu/h)

ρ = density (lbs/ft^3)

V = infiltration plus ventilation rate (ft^3/h)

c = heat capacity (Btu/lbs·F)

ΔT = difference in temperature (F)

Total heat loss of each room was the sum of heat loss of every surface and heat loss from the infiltration:

$$Q_{total} = Q_{floor} + Q_{attic} + Q_{wall} + Q_{window} + Q_{infiltration}$$

Where:

Q_{total} = total heat loss of the conditioned space (Btu/h)

Q_{floor} = heat loss from the floor (Btu/h); temperature under the floor was assumed to equal ambient temperature

Q_{attic} = heat loss from the attic (Btu/h); attic temperature was calculated by the BEopt model

Q_{wall} = heat loss from the wall (Btu/h)

Q_{window} = heat loss from the window (Btu/h)

$Q_{infiltration}$ = heat loss from the infiltration (Btu/h)

Required supplemental resistance heat was calculated as the balance of total heat loss and heat gain:

$$Q_{resist} = Q_{total} - Q_{solar} - Q_{internal}$$

Where:

Q_{total} = total heat loss of the conditioned space (Btu/h)

Q_{resist} = resistance heat required for the conditioned space (Btu/h)

Q_{solar} = solar heat gain (Btu/h)

$Q_{internal}$ = internal heat gain (Btu/h)

The following assumptions were made in the load calculation:

1. Average air temperatures of rooms where heat pump was located were assumed to be 70°F.
2. Average air temperatures of other bedrooms and bathrooms were assumed to be 68°F.
3. Supply air temperature from transfer fan was assumed to be equal to the air temperature at the ceiling of the room where heat pump was located; assumed to be 71°F.
4. Transfer fan airflow and infiltration assumptions:

   The blower door test result was converted to natural infiltration rate of the house using Sherman's model (Sherman, 1987). Infiltration of each room was assumed to be proportional to the volume of the room.

TMY hourly ambient temperature was used to calculate the hourly averaged heat loss and required supplemental heat, which was summed for the heating season to estimate the annual supplemental resistance heat.

Table F-1. New Jersey Home Fan Flows and Infiltration/Ventilation Rates

Room	Transfer Fan Airflow (cfm)	Infiltration and Ventilation (cfm)
Master Bedroom	150	2.7
SW Bedroom	150	2.0
NW Bedroom	150	1.9
Master Bathroom	110	0.7
Family Bathroom	110	1.5

Table F-2. Massachusetts Home Fan Flows and Infiltration/Ventilation Rates

Room	Transfer Fan Airflow (cfm)	Infiltration and Ventilation (cfm)
Master Bedroom	150	2.8
SW Bedroom	150	2.2
NW Bedroom	150	2.0
Master Bathroom	110	0.7
Family Bathroom	110	1.6
Living Room	0	9.4

Table F-3. Maryland Home Fan Flows and Infiltration/Ventilation Rates

Room	Transfer Fan Airflow (cfm)	Infiltration and Ventilation (cfm)
Master Bedroom	150	2.0
Bedroom 1	150	2.4
Bedroom 2	150	1.9
Master Bathroom	110	0.8
Second Bathroom	110	0
Living Room	0	8.1

Annual supplemental heating load calculation results for the houses are shown in the tables below:

Table F-4. New Jersey Home Annual Supplemental Heating Load Calculation Results

Room	Required Supplemental Resistance Heat (kWh/yr)
Master Bedroom	221
SW Bedroom	185
W Bedroom	18
Master Bathroom	0
Family Bathroom	0
Total	424

Table F-5. Massachusetts Home Annual Supplemental Heating Load Calculation Results

Room	Required Supplemental Resistance Heat (kWh/yr)
Master Bedroom	435
SW Bedroom	372
W Bedroom	61
Master Bathroom	0
Family Bathroom	0
Living Room	0
Total	868

Table F-6. Maryland Home Annual Supplemental Heating Load Calculation Results

Room	Required Supplemental Resistance Heat (kWh/yr)
Master Bedroom	137
Bedroom 1	109
Bedroom 2	23
Master Bathroom	0
Second Bathroom	0
Living Room	0
Total	269

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