

Field Evaluation of Advances in Energy- Efficiency Practices for Manufactured Homes

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ARIES Collaborative

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Field Evaluation of Advances in Energy-Efficiency Practices for Manufactured Homes

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The work presented in this report does not represent performance of any product relative to regulated minimum efficiency requirements.

The laboratory and/or field sites used for this work are not certified rating test facilities. The conditions and methods under which products were characterized for this work differ from standard rating conditions, as described.

Because the methods and conditions differ, the reported results are not comparable to rated product performance and should only be used to estimate performance under the measured conditions.

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Definitions

°F	Degree Fahrenheit
AC	Air conditioner
ACH50	Air change per hour at 50 Pascals
AIM-2	The Alberta Air Infiltration Model
ARIES	Advanced Residential Integrated Energy Solutions Collaborative
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
Avg.	Average (tables only)
BEopt V 2.3.0.2	Building Energy Optimization version 2.3.0.3
Btu	British thermal unit
CFH	Cubic feet per hour
cfm	Cubic feet per minute
COP	Coefficient of performance
d	Day
DHW	Domestic hot water
DOE	U.S. Department of Energy
EER	Energy efficiency ratio
EF	Energy factor
ft ²	Square feet
h	Hour
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
kBtu	One Kilo British thermal unit
kg	Kilogram
kJ	Kilojoule
kPa	Kilopascal
kW	Kilowatt
kWh	Kilowatt-hour
MEF	Modified Energy Factor

min	Minute
MMBtu	One million British thermal units
N/A	Not applicable
s	Second
SEER	Seasonal Energy Efficiency Ratio
SHGC	Solar heat gain coefficient
Temp.	Temperature (tables only)
UA value	Heat loss over the entire surface area
U-value	Overall heat transfer coefficient
W	Watt
yr	Year
ZEMH	Zero Energy Manufactured Home
ZERH	Zero Energy Ready Home

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Executive Summary

Through field-testing and analysis, this project evaluated whole-building approaches and estimated the relative contributions of select technologies toward reducing energy use related to space conditioning in new manufactured homes. Three lab houses of varying designs were built and tested side-by-side under controlled conditions in Russellville, Alabama. The tests provided a valuable indicator of how changes in the construction of manufactured homes can contribute to significant reductions in energy use.

The three side-by-side, unoccupied lab houses included heating, ventilating, and air-conditioning systems that simulated sensible loads. Each house represented a commonly specified set of features that impacts the energy used for space heating and cooling. The three homes had features that reflected the following levels of energy efficiency: a design meeting best-practice codes according to the U.S. Department of Housing and Urban Development (House A); a manufactured home conforming to typical ENERGY STAR[®] ratings (House B); and a design meeting the U.S. Department of Energy Zero Energy Ready Home criteria (House C).¹ Except for the differences in the energy-related measures, the prototypes were identical (floor plan, orientation, construction). (See Table 2 for specifications.) The side-by-side tests provided a wealth of comparative data that yielded the following observations:

- **Energy consumption.** During the cooling period, House C used half the energy of the other houses. House B used slightly less energy than House A for cooling. House B and House C consumed about the same amount of heating energy, which was nearly one-third of the heating energy consumed by House A.
- **Comfort.** All three homes operated within indoor comfort guidelines specified by the Air Conditioning Contractors of America ($\pm 3^{\circ}\text{F}$ temperature variation from the temperature at the thermostat during the cooling period) when the interior doors were open. House C used one-third of the heating energy of House A with superior comfort results. Some bedrooms in House A and House B had difficulty maintaining temperatures within $\pm 2^{\circ}\text{F}$ of the set point.
- **Propensity for moisture problems.** Wall cavity wood moisture content was slightly higher in House C than in House B, but it was still well within the safe limits. This was because of the addition of exterior foam insulation to the walls in House C. (The expanded polystyrene foam insulation reduces the overall vapor permeability of the walls.) However, compared to House B, the use of foam sheathing on House C also resulted in an average higher temperature (5.5°F) on the interior face of the foam during the heating season, mitigating condensation risk. Note that latent loads were not simulated.
- **Peak load performance.**² Peak demand in House C was significantly lower than it was in the other two houses throughout the year. House B had somewhat lower peaks than did House A during most months. House A and House B winter peaks were similar during the winter months, suggesting that House B's peak occurred when the heat pump was not operating and the house was relying solely on electric resistance backup. On average, during

¹ House C was the first manufactured home in the United States to qualify for Zero Energy Ready Home designation.

² Comments regarding peak load refer only to the demand resulting from the operation of the space-conditioning and ventilation equipment.

peak hours for the Tennessee Valley Authority system, House B had an 18% lower peak demand than House A, and House C had a 69% lower peak demand than House A.

- **Real-world performance of heat pumps.** The heating coefficient of performance (COP) was calculated for both heat pumps using two independent methods: (1) airflow and temperature measurements and (2) co-heat testing measurements.³ The co-heat testing method was deemed more reliable. Using this method, the COP of both the traditional split-system heat pump and the ductless mini-split averaged approximately 2.5. This was well below what would be expected for the mini-split based on manufacturer data. However, when the mini-split air handling unit fan was forced to run on high speed, its COP increased to 4.11, indicating that low airflow could have been a cause of poor heat pump performance with this unit.⁴
- **Estimate of costs and benefits.** The cost premiums to the manufacturer (counting only the energy-efficiency measures) to build House C compared to House A and House B were \$2,060 and \$1,166, respectively. The price premiums for the homebuyer to purchase House C compared to House A and House B would be \$6,607 and \$4,339, respectively, assuming that the houses were in regular production. The energy savings of House C compared to House A and House B range from 5.3 to 16.0 MMBtu/yr when they are located in International Energy Conservation Code Zone 3 and Zone 4. Within this range of conditions, the simple paybacks to the homebuyer associated with building House C compared to House A and House B ranged from approximately 8.8 to 17.5 yr.

³ Co-heat testing was used to measure whole-house heat loss and therefore effective envelope U-values. These data were used to tune the simulation parameters.

⁴ It was not determined whether this finding can be generalized to other mini-split heat pumps.

1 Introduction

Through field-testing and analysis, this project evaluated whole-building approaches and estimated the relative contributions of select technologies toward reducing energy use related to space conditioning in new manufactured homes. Three unoccupied lab houses of varying designs were built and tested side-by-side under controlled conditions in Russellville, Alabama (International Energy Conservation Code [IECC] Climate Zone 3).

The tests compared the performance of the three houses (referred to herein as House A, House B, and House C) built to different levels of thermal integrity. These tests allowed for a side-by-side comparison of actual whole-house performance focusing on the impacts that changes in envelope construction and heating, ventilating, and air-conditioning (HVAC) and distribution systems have on space-conditioning energy use. The houses had the following features: (1) House A was designed to best-practice thermal standards according to the U.S. Department of Housing and Urban Development (HUD) and equipped with an electric furnace and a split-system air conditioner; (2) House B was designed to comply with the requirements of the ENERGY STAR[®] program for manufactured homes, including an improved thermal envelope and a conventional split-system heat pump; and, (3) House C was designed to qualify for Zero Energy Ready Home (ZERH) designation according to the U.S. Department of Energy (DOE) and included a high-efficiency, single-point, ductless mini-split heat pump with a transfer fan distribution system instead of the traditional duct system for distribution. House A and House B used a standard duct system for air distribution.

2 Background and Literature Review

Energy is one of the major contributors to homeownership costs, and high energy costs create a pronounced financial burden on households that have modest incomes. Manufactured homes in particular are susceptible to excessive energy costs because industry energy standards, nationally promulgated by HUD, were last updated in 1994. Programs such as ENERGY STAR and DOE's ZERH showcase ways to improve home efficiency and reduce energy costs. These efforts often incur higher construction costs (associated with enhanced efficiency) to achieve lower energy bills, a combination designed to yield lower net monthly homeowner costs.

2.1 Literature Review

In pockets of activity throughout the nation, researchers have conducted or are conducting field tests to explore better ways to build affordable, durable, comfortable, energy-efficient homes. These activities include a few studies using manufactured homes. Some include field-testing homes that have varying envelope characteristics and an array of high-performance HVAC systems. A few of these efforts are described below.

- Research that sought to estimate the energy savings associated with changes in building envelope characteristics and HVAC system efficiency includes work by Oak Ridge National Laboratory (Gehl et al. 2012) comparing three site-built side-by-side homes in Farragut, Tennessee, that had similar floor plans. One home was a typical builder house, the second contained high-efficiency HVAC system components (including an air source heat pump and sealed and insulated attic ducts fully inside the conditioned volume), and the third was built using optimum-value framing construction and was equipped with photovoltaic panels and a

solar water-heating system. To simulate occupant load, the researchers programmed appliances, lights, and plug loads to run on a set schedule based on the Building America Research Benchmark protocol (Hendron and et al. 2006) (Hendron and Engebrent 2009). Compared to the standard home, Home 2 and Home 3 used 37% and 67% less energy, respectively.

- Similarly, Chasar, Chandra, and Parker et al. (2006) and Chasar, Chandra, and VanSchramm et al. (2010) compared three identical site-built homes of approximately 2,000 ft² in San Antonio, Texas, a location that experiences both semiarid and humid subtropical conditions. Temperature and humidity data were recorded, and energy use was measured at a subsystem level, including heating, cooling, hot water, and major appliances. One of the homes was built to builder standard practice and served as the control home, and the other homes contained high-performance features. Researchers recorded utility peak electric load comparisons to assess the merits of various envelope and equipment improvements. The control home used natural gas for space and water heating, whereas one of the improved homes was equipped with a high-efficiency heat pump. Energy ratings for the homes yielded E-Scales (analogous to a Home Energy Rating System index) of 86 for the control home, 54 for one improved home, and 37 for the third home. (The latter included a 2.4-kW photovoltaic array.) Relative to the control home, cooling energy savings ranged from 55% to 77% for the two improved homes. Total demand reductions observed from the control and improved homes ranged from 6 to 8 kW (62% to 83%) on the hottest day during the utility peak period. Peak air-conditioning loads in the improved homes on the same day were reduced by 1.2 to 2.9 kW (28% to 68%) compared to the control. A 2.4-kW grid-tied photovoltaic array on one home provided 60% of total electric energy needs during the hottest months, although the home was unoccupied during this period. During subsequent months when the home was occupied, the array provided approximately 30% to 40% of the total home electric needs.
- In a study performed in Grand Forks, North Dakota (Chasar et al. 2004), eight duplex units were built with the goal of achieving up to 50% energy savings compared to the 1993 Model Energy Code. The actual comparison of energy use was provided by simulation with key data (such as infiltration rates) supplied by the field study. Of the eight homes, four were tighter and had more efficient HVAC systems and appliances than the Model Energy Code design, and four were progressively tighter than the first group. The results were the basis for cost-effectiveness assessments. Combined gas and electric savings compared to the base case ranged from 25% (first group of homes) to 35% (second group).
- The Florida Solar Energy Center (2004) in association with North Carolina Agricultural and Technical State University compared the energy used by two manufactured homes placed side-by-side in a controlled field study. One home conformed to HUD standards; the other contained measures intended to reduce energy use by approximately 50%. Modeling with Energy Gauge software predicted heating, cooling, and domestic hot water (DHW) savings of 57%; and field measurements yielded 53% savings. The homes were unoccupied, and the internal loads were simulated by incandescent lights on timers.
- The Zero Energy Manufactured Home (ZEMH) project examined two occupied, 1,600 ft², double-section manufactured homes using identical floor plans (Lubliner and Hadley 2007). Remote monitoring equipment was installed to track the performance of each home during a

3-yr period. Both homes were built in 2002 by Kit Manufacturing. A comparison home was built to ENERGY STAR program requirements as part of the Northwest Energy Efficient Manufactured Housing Program. Total energy use of the ZEMH was 29.4 kWh/d, with 7.7 kWh/d was covered by the photovoltaic array. The ZEMH showed 9% less energy use per year than the ENERGY STAR home, with 44% less space-heating energy consumption despite having a higher occupant load than the ENERGY STAR home. The results were not normalized for occupant behavior.

- Pacific Northwest National Laboratory (2010) compared two identical, 1,500 ft² manufactured homes located side-by-side to measure the impact of a variety of energy-saving measures on energy use and peak demand. A baseline home served as a control, typifying an average existing home in the Pacific Northwest. The second home was used to test new technologies. Occupancy was simulated following the protocol developed by the Pacific Northwest National Laboratory (Hendron and Engebrent 2009). Among the technologies evaluated were highly efficient windows (R-5) projected to save approximately 13% of the total whole-house energy use and reduce cooling-season peak demand by approximately 25%. High-performance windows were also shown to improve thermal comfort through the moderation of radiant loads. However, the measure is expensive, and it has a long payback period (23 to 55 yr). The work continues, and the Pacific Northwest National Laboratory team plans to test other measures, including heat pump water heaters and low-emissivity storm windows. Table 1 summarizes the studies mentioned above and energy savings achieved.

Table 1. Summary of Past Lab House Research

Author(s)	Location	Description of Houses	Home Occupancy Size (ft ²)	Simulation (Yes/No)	Savings Relative to Case 1
Oak Ridge National Laboratory (2011)	Farragut, TN	<ol style="list-style-type: none"> 1. Typical builder house 2. Used energy-efficient appliances, windows, air-to-air heat pump, and insulated attic 3. Built using optimum-value framing construction with photovoltaic modules and solar water heating. 	2,400	Hendron's (2009) test protocol was used to simulate occupancy in all three houses throughout the testing period.	37% (2) 67% (3)
Chasar et al. (2010)	San Antonio, TX	<ol style="list-style-type: none"> 1. Typical builder house 2. High-performance home with 9.5 HSPF^a/18 SEER^b heat pump, tankless DHW boiler, 100% fluorescent lights on timers/occupancy sensors, and energy-efficient appliances 	2,000	All three houses were occupied during most of the testing time.	55% (2) 77% (3)

Author(s)	Location	Description of Houses	Home Size (ft ²)	Occupancy Simulation (Yes/No)	Savings Relative to Case 1
		3. Same as House 2 but with a tighter shell.			
Chasar et al. (2010)	Grand Forks, ND	1. Typical builder house (R-19 wall insulation) 2. As above with tighter shell (2.8 ACH50 ^c), 70% efficient energy recovery ventilator 3. As above with R-25 wall insulation, 2.4 ACH50 infiltration.	1,840	Only computer models were used for each house; a theoretical base case house built to local minimum standards was devised to determine the energy savings and cost-effectiveness.	25% (2) 35% (3)
Florida Solar Energy Center (2004)	Greensboro, NC	1. Basic HUD code standard 2. 50% more energy efficient.	1,528	The house included incandescent lights for 1.5 persons. Hot water draws were twice per day.	53%–58% (2)
Lubliner/ Florida Solar Energy Center (2007)	Cherry Lane, ID	1. ENERGY STAR HUD code home 2. ZEMH, built with highly efficient, cutting-edge technologies, including photovoltaic system, sun tempering, solar water heating, heat recovery ventilator, and other ENERGY STAR appliances.	1,600	No occupancy simulation or normalization was done. The number of occupants in the ZEMH was greater than it was in the ENERGY STAR home.	9% (2)

^a Heating seasonal performance factor

^b Seasonal Energy Efficiency Ratio

^c Air change per hour at 50 Pascals

2.2 Research Gaps and Relevance to Building America's Goals

Relatively few efforts aimed at improving energy performance have looked at the challenge from a factory-built perspective and in particular the manufactured-home perspective. The best approaches for reducing energy use, improving indoor air quality, and increasing durability in homes constructed on-site are sometimes different from those built off-site. For example, the selection, design, and installation of the HVAC equipment and related distribution system are unique to factory building, and when they are viewed from an energy-efficiency standpoint they are a weak link.

Building America has a goal of developing cost-effective home-technology solutions to reduce whole-house source energy consumption by 50% compared to the 2009 IECC. This effort aims to do so for space-conditioning energy in an affordable-housing segment. This work builds on knowledge related to the performance of point-source space conditioning for low-load homes, an area on which a number of Building America teams have focused. A thorough summary of the Building America literature on this subject is included in work by Building Science Corporation (Ueno and Loomis 2015). Additionally, this work examines the impact of changes in thermal performance on collateral issues, particularly moisture performance, continuing the systems approach deeply embedded in Building America research.

2.3 Research Questions

This project addressed the following research questions:

- Can factory-built, manufactured homes meet the ZERH program requirements as defined? If not, what changes are suggested in the ZERH specifications to better reflect and take advantage of the unique aspects of factory building?
- What are the added costs and energy savings for manufactured ZERHs using ductless mini-split heat pumps compared to homes that adhere to best-practice HUD codes and ENERGY STAR manufactured homes?
- How does point-source space conditioning perform in a ZERH in terms of comfort? How do operational variables, such as opening and closing interior doors and window blinds and the operation of transfer fans, impact performance in heating and cooling modes?
- What is the in-situ coefficient of performance (COP) in heating and cooling modes of a ductless mini-split heat pump?

3 Whole-Building Performance Assessment

This section profiles the whole-building assessment process conducted during a full heating and cooling season in Russellville, Alabama. The discussion covers the approach and process for setting up the experiment (house design and procurement, siting, commissioning, operation and monitoring protocol), results, and data analysis.

3.1 Approach

Three manufactured homes were built and colocated on a site adjacent to the SEhomes home-building facility in Russellville, Alabama. The houses varied only in the technologies being evaluated. They were operated for 15 months, and data was intensively monitored. The experiment assessed the following strategies for reducing energy use:

1. **Thermal envelope technologies.** The manufactured-home walls use exterior rigid foam insulation, enhanced air sealing, and highly insulated low-slope roofs utilizing a new dense-pack roof insulation technique.
2. **Site-installed heat pumps.** Site-installed, split-system heat pumps are used in the construction of manufactured homes in the Southeast, the location of the tests, and they are required for homes qualifying for ENERGY STAR designation. However, their contribution toward energy use reduction and the specific performance characteristics of heat pumps when used with homes adhering to HUD code have not been verified in the field.
3. **Ductless mini-split heat pumps.** Ductless heat pumps can be fully installed and commissioned during home manufacturing in the plant and transported to the site, ready for operation. They are used in place of a typical furnace and split-system cooling equipment; the latter are completed at the site during the home setup. Ductless heat pumps offer the following advantages:
 - A. By replacing central ducted systems, ductless heat pumps eliminate the associated distribution issues and comfort problems that often accompany poor supply register locations and losses due to duct leakage.
 - B. They typically incorporate inverter-driven compressors and variable-speed fan technology, which results in very high efficiencies of up to 27 SEER and 12.5 HSPF, much higher than standard heat pumps.
 - C. They free up space within the home usually devoted to heating and cooling equipment.
 - D. They provide both heating and cooling, offering a total space-conditioning solution in a single device.
 - E. They can be fully installed in the plant. This ensures that the home will have a heat pump, and it enables strict plant oversight and inspections that guarantee consistent installation quality and proper commissioning.
 - F. A single, wall-mounted outside compressor can be connected to numerous independently controlled air handling units, which facilitates flexible zoning of spaces and reduces over-/under-space conditioning.

- G. The compressor is mounted outside, which reduces interior noise compared to most compact, unitary air-conditioning/heat pump systems.
- H. They eliminate the possibility of mismatched indoor and outdoor equipment.
- I. They are available in small capacities (as low as 9,000 Btu/h) that are suitable for low-load buildings, such as small manufactured homes.

3.2 Experimental Method

3.2.1 Designs

Three side-by-side lab houses were built in Russellville, Alabama: one that conforms to best-practice HUD standards (House A), one that complies with the manufactured-home ENERGY STAR program (House B), and one that meets DOE’s ZERH criteria (House C). The prototypes were identical (floor plan, orientation, construction) except for the differences associated with the measures listed in Table 2. Construction drawings are provided in Appendix J.

Table 2. Lab House Specifications

Items	House A	House B	House C
Floor	R-14 fiberglass blanket	R-28 fiberglass blanket	R-28 fiberglass blanket
Flooring	60% carpet; 40% vinyl	60% carpet; 40% vinyl	60% carpet; 40% vinyl
Wall	½-in vinyl-covered drywall, R-11 fiberglass batts, ¼-in ThermalStar board (R-1) (perm rating >5), light-colored vinyl siding	½-in vinyl-covered drywall, R-13 fiberglass batts, ¼-in ThermalStar board (R-1) (perm rating >5), light-colored vinyl siding	½-in vinyl-covered drywall, R-13 + R-5 (1-in. extruded polystyrene) (perm rating 1.5), light-colored vinyl siding
Windows	U: 0.47, SHGC: 0.73; single pane, metal frame	U:0.31, SHGC: 0.33, double pane, vinyl frame, low-emissivity, air filled	U: 0.30, SHGC: 0.23, double pane, vinyl frame, low-emissivity, argon filled
Ceiling	R-22 blown fiberglass, vented roof cavity with asphalt shingles	R-33 blown fiberglass, vented roof cavity with asphalt shingles	R-45 blown fiberglass, dense-packed at eaves, vented roof cavity with asphalt shingles
Air Sealing	Standard plant practice including foaming ceiling penetrations, caulking under bottom plates and between top plates and ceiling, marriage line gasket	Standard plant practice including foaming ceiling penetrations, caulking under bottom plates and between top plates and ceiling, marriage line gasket	Standard plant measures plus foaming around window and door rough openings, patching/foaming floor penetrations, taping foam sheathing joints, caulking

Items	House A	House B	House C
			foam sheathing to top and bottom plates, spray foam along marriage line
Mechanical Ventilation	Fresh air duct to air handling unit, no mechanical damper	Fresh air duct to air handling unit, no mechanical damper	45 cfm exhaust fan
Space-Conditioning Distribution	Metal in-floor ducts sealed with mastic, R-8 crossover duct between home sections	Metal in-floor ducts sealed with mastic, R-8 crossover duct between home sections	BR2 and BR3: Tjernlund AS1 transfer fan, 21 W MBR and MBath: Tjernlund AS2 transfer fan, 33 W

Table 3. Cooling and Heating Equipment Specifications

	House A	House B	House C
Cooling Equipment	Intertherm DS4BD-024KB Cooling coil: C8 (B,D) AM01830U-B Cooling capacity: 23.4 kBtuh EER: 11.0 SEER: 13.0	Intertherm air source heat pump DT4BD-018K with backup electric resistance heaters Cooling coil: C8 (B, D) AM01830U-B Cooling capacity: 18 kBtuh	Mitsubishi variable-speed mini-split heat pump with outdoor unit assisted by temperature-controlled heaters when temperature falls below 69°F in the bedrooms
Heating Equipment	NORDYNE electric furnace, E3EB-010H, downflow set to low speed Resistance heating capacity: 35 kBtuh	EER:11.0 SEER: 13.0 Heating capacity at 47°F: 20.2 kBtuh; HSPF: 8.0 Heating capacity at 17°F: 11.5 kBtuh	Outdoor unit: MUZ-FH15NA Indoor unit: MSZ-FH15NA Cooling capacity: 15 kBtuh
Air Handling Unit	NORDYNE electric furnace, E3EB-010H, downflow set to low speed Resistance heating capacity: 10 kW Air handling unit wattage (heating elements + blower):10.4 kW	NORDYNE electric furnace, E3EB-010H, downflow set to low speed Resistance heat capacity: 10 kW Air handling unit wattage (heating elements + blower):10.4 kW	EER: 12.5 SEER: 22.0 Heating capacity at 47°F: 18 kBtuh; HSPF: 12.0 Heating capacity at 17°F: 11 kBtuh

The floor plan for House A and House B is shown in Figure 1. The only changes to House C were the elimination of the furnace, relocation of the water heater to the furnace closet (to reduce DHW pipe lengths), and enlargement of the Bedroom 2 closet to include the former water heater compartment.

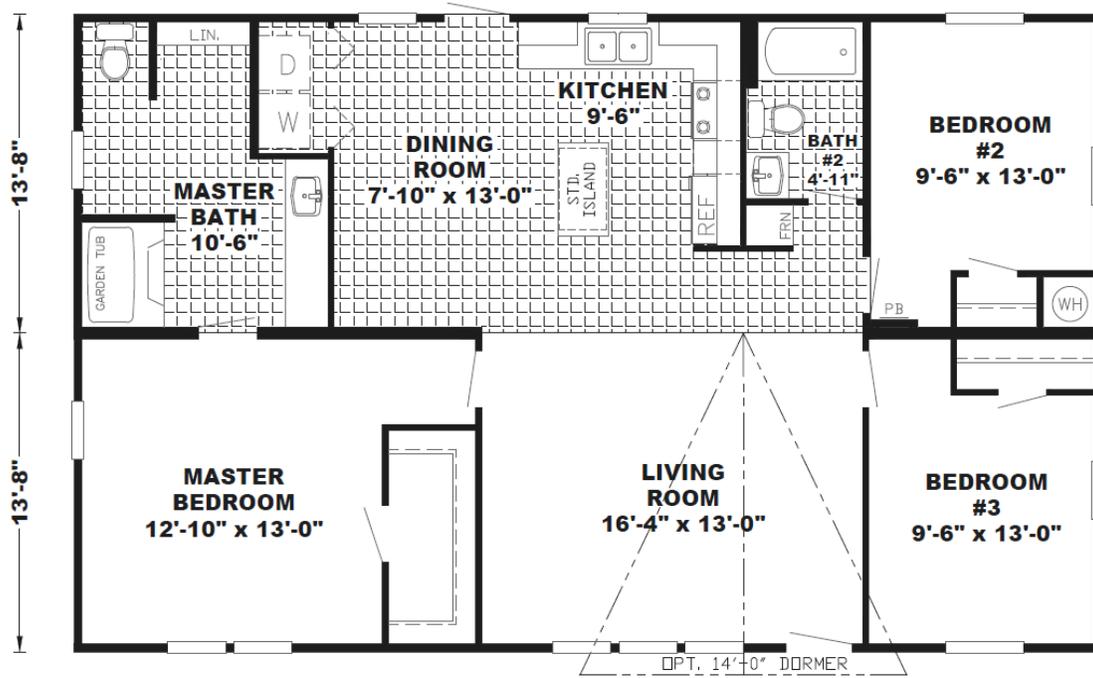


Figure 1. Floor plan

3.2.2 Acquisition

The homes were purchased from a single manufacturing plant and inspected during manufacture to ensure that the construction methods were consistent and uniform (Figure 2).



Figure 2. Lab house production

To ensure maximum protection against moisture and minimize envelope infiltration, window flashing systems were applied to the extruded polystyrene, as shown in Figure 3.



Figure 3. Window flashing system in House C

3.2.3 Siting

The homes were installed adjacent to the manufacturing plant for a period of approximately 15 months. The orientation and solar exposure were identical for all homes (Figure 4).



Figure 4. Lab house site

3.2.4 Commissioning

The homes were commissioned for the monitoring effort by establishing baseline conditions. A summary of commissioning test results is provided in Table 4, with additional details in Appendix A. Due to the influence of ENERGY STAR requirements on manufactured homes, most of the manufactured-home plants in the United States use similar techniques for reducing enclosure and duct leakage in all newly constructed homes. Therefore, the enclosure leakage in House A is nearly the same as it is in House B. Thus, compared to older manufactured homes in the field, House A is considered a best-practice HUD home.

Table 4. Commissioning Tests

Test	Equipment/Method	Results
Enclosure Leakage	Energy Conservatory blower door multipoint depressurization test	A: 4.7 ACH50 B: 4.6 ACH50 C: 3.8 ACH50
Duct Leakage	Energy Conservatory duct blaster depressurization test	A: 54 cfm25 to outside B: ~10 cfm25 to outside C: N/A
Ventilation Rate	Energy Conservatory FlowBlaster	A: 44 intermittent B: 32 intermittent C: 45 continuous
Air Handling Unit Air Flow	Energy Conservatory duct blaster	A: 980 cfm B: 1,000 cfm C: variable
Supply Register Air Flows	Energy Conservatory FlowBlaster	See Appendix A

Note that the air handling unit air flow rates for House A and House B are high relative to the loads for optimal humidity control, even at the lowest speed. Nevertheless, this was the plant’s standard air handling unit used for production homes. This would result in lower moisture removal in an occupied home; however, latent loads were not simulated in these unoccupied lab homes.

Air flow rates at a variety of power draws were measured for the variable-speed mini-split heat pump air handling unit in House C. These flows were later used in the investigation of heat pump COP. The resulting fan curve is shown in Figure 5.

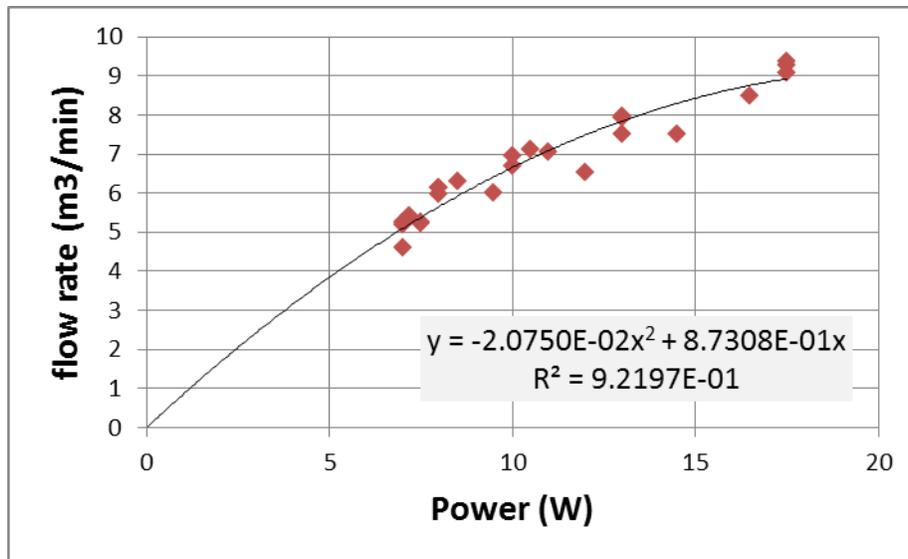


Figure 5. House C heat pump fan curve

3.2.5 Operation

The homes were operated in an identical manner, including interior temperatures and simulation of occupant loads.

- **Primary thermostat set points.** The heating set point was 71°F, and the cooling set point was 76°F. To maintain an average house temperature (average of all rooms) of 76°F during the cooling season, the heat pump in House C was placed on a setback schedule of 75°F from 8:00 p.m. to 10:00 a.m. and 73°F from 10:00 a.m. to 8:00 p.m. This was necessary to obtain average space temperatures comparable to House A and House B.
- **Secondary heating (House C) set point.** Auxiliary electric resistance heaters were used to supply backup heat to the House C bedrooms and master bathroom. These heaters were set to 9°F.
- **Transfer fans (House C) set point.** Through-wall transfer fans in House C were set on independent thermostats in each room to which the fans delivered air. These thermostats were set to activate the fan at an upper set point of 76°F during the cooling season and at 69°F during the heating season. Short-term tests were conducted with the fans deactivated to study their impact on room temperatures and energy use. Thermostat control was used so fans would run only when heating or cooling was needed in the rooms they served.
- **Occupant loads.** Sensible internal heat gain was simulated through the use of electric resistance heaters controlled by the data loggers. Figure 6 shows the schedule of loads.

Latent internal heat gains can also impact energy consumption and comfort; however, latent loading is more complex to simulate. Latent loads have been simulated through the use of ultrasonic humidifiers by the National Renewable Energy Laboratory (Fang et al. 2011). It was estimated that these homes would require approximately 1.5 gal/d of water delivered to the air. Because the homes were in a remote location and not attended regularly, there was limited opportunity to inspect the humidification equipment. Further, the homes were not connected to

water service. Because the systems would have had to run unattended for long periods, large water tanks (100–200 gal per home) using deionized water would have been required to reduce the need for humidifier maintenance, or frequent descaling would have been required. Risks associated with this approach include undetected leaks, malfunction of the humidification system, and biological growth in the water tanks. Therefore, internal latent loads were not simulated. However, a short-term humidification test was conducted in each home to assess the dehumidification capability of the equipment.

It is acknowledged that this may lead to greater variability in the results; eliminating latent internal loads could potentially impact the three houses differently. Additionally, measurements of moisture levels in the wall cavities were possibly underestimated along with attendant risks of condensation.

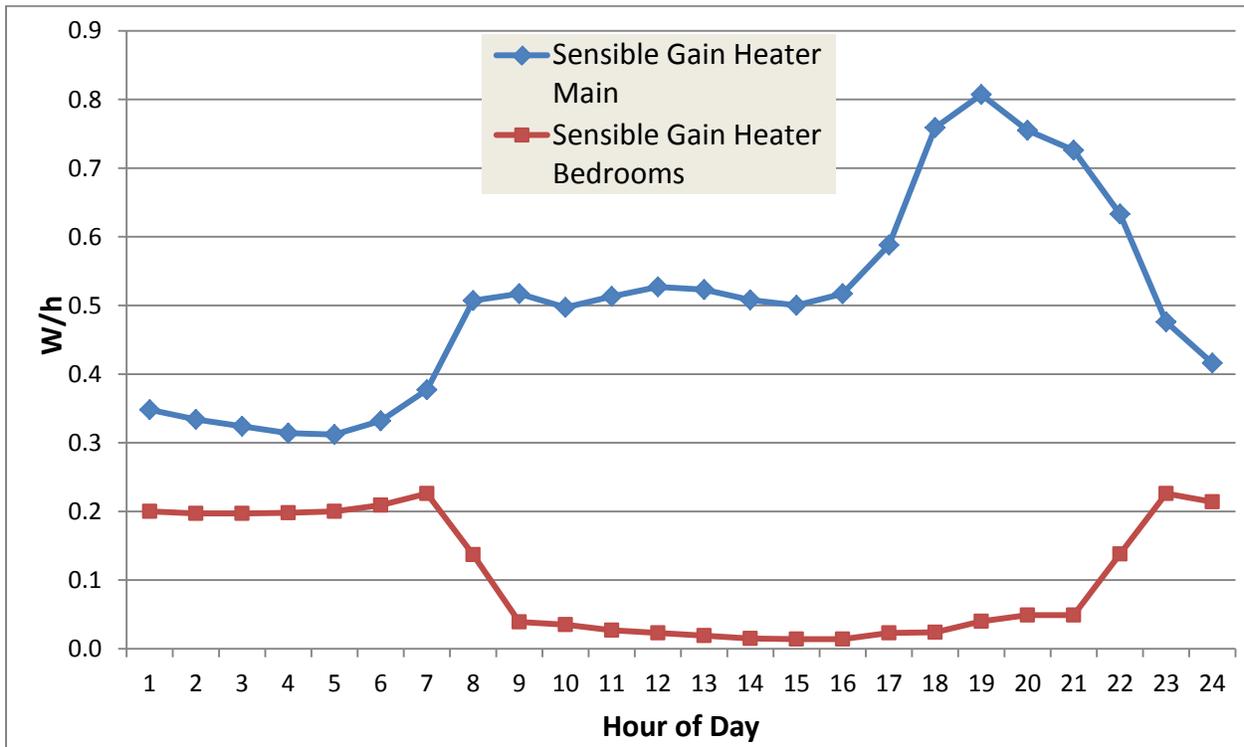


Figure 6. Sensible load profiles

- **Interior doors.** Because the field data was compared to single-zone Building Energy Optimization software (BEopt) model simulation, the interior doors were kept open for the majority of the testing period. To characterize the temperature distribution across the homes with the doors closed, short-term tests (5–10 d) were conducted during the cooling and heating seasons with the doors closed.
- **Window blinds.** For the majority of the monitoring period, the blinds were kept 50% closed. Short-term tests were conducted with the blinds fully open and fully closed.

3.2.6 Measurements

The homes were monitored for 15 months with 1-min data uploaded on a daily basis. CR1000 data loggers in conjunction with multiplexers were used for the data acquisition. Key metrics,

such as equipment operation and internal temperatures, were reviewed periodically to verify that the homes were operating as intended. Monitoring points and equipment included the following (see also Appendix B):

- **Air temperature.** Watlow T-type thermocouples in conjunction with aspirated shields were used to measure space air temperatures at five locations in each home at 5 ft above the floor (mounted on tripods) and at least 5 ft away from the heating/cooling/air distribution systems and exterior walls. A crawl space temperature sensor was placed at the center of the crawl space approximately 1.5 ft above grade. An attic air temperature sensor was placed at the center of the attic above the insulation.
- **Air temperature and relative humidity.** A Campbell Scientific CS215 temperature and relative humidity probe was used to measure the temperature and relative humidity at one location in each home. The same sensor type was also used to measure the outdoor air temperature and relative humidity at one location in all three homes. To protect the outdoor sensor from the elements, a protection shield was used.
- **Condensation.** In House B and House C, Campbell Scientific CS215 temperature and relative humidity probes were used to measure the air temperature and relative humidity of the surface between the exterior sheathing (extruded polystyrene insulation) and interior fiberglass insulation on a north-facing wall approximately 32 in. above the floor. An Omnisense S900-1 sensor was used to measure the moisture content of the lower exterior corner of a stud in the same north-facing wall. The data from these sensors was used to predict the propensity for moisture accumulation in the exterior wall.
- **Power consumption.** Power current transducers in conjunction with Wattnode energy-to-pulse transducers measured the power of HVAC equipment and total house power.
- **Status.** Hawkeye 300 (Veris H300) go/no-go current switches were used to measure the status of the reversing valves of the heat pumps.
- **Current.** The compressor and fan current were measured by Veris H721LC sensors.
- **Solar radiation.** A Campbell LI200X-L silicon pyranometer was used to measure the solar radiation at the site.

In addition to the long-term monitoring system, a series of short-term measurements were made:

- **Co-heat testing.** Short term co-heat testing in cold weather using portable electric resistance space heaters was conducted in all three houses to measure whole-house heat loss. The test data was used to obtain envelope U-value (overall heat transfer coefficient) information in each house and provide better quantification of simulation parameters. During the test, six portable heaters (one in each bedroom, one in the master bathroom, one in the living room, and one in the kitchen) were used to maintain 70°F throughout the test. Other heating equipment was switched off. Temperature sensors were used to turn the heaters on and off through the Campbell Scientific data loggers.
- **Tracer gas decay testing.** Infiltration was measured in all three homes utilizing tracer gas. The objectives of the tracer gas testing were to determine infiltration levels in all three

houses for calibrating models and quantifying the effectiveness of the ventilation systems in all three houses.

- **Humidification testing.** The dehumidification capability of the equipment was tested. The objective of this test was to measure the humidity levels in each house during near-peak humidity conditions to assess the capability of the cooling equipment to maintain comfort conditions.

3.3 Monitoring Results

This section describes the results of the tests noted above. It characterizes typical performance in terms of HVAC energy consumption, comfort, moisture, and peak energy demand. The analysis section that follows generalizes the data to annual whole-house energy consumption at various locations using calibrated energy models.

3.3.1 Cooling Season HVAC Energy

A representative cooling season period that exhibited a wide range of outdoor temperatures was selected for analysis. During this period—from August 29, 2014, through September 15, 2014—the interior doors were open and the window blinds were 50% closed.

A plot of cooling power relative to ambient temperature for this period shows that House C used less cooling energy than did the other houses at outdoor temperatures in excess of approximately 24°C (75°F) (Figure 7). House B used somewhat less cooling energy than did House A when temperatures rose above approximately 25°C. At lower temperatures, House A used less cooling energy than did House B. This inversion may have occurred because House A's lower insulation values and higher ventilation rate allowed more natural cooling, whereas the better insulated House B held the heat longer. (Both homes had similar airtightness.) This was especially prevalent after sunset when the lower solar heat gain through the windows of House B no longer provided an advantage over House A.

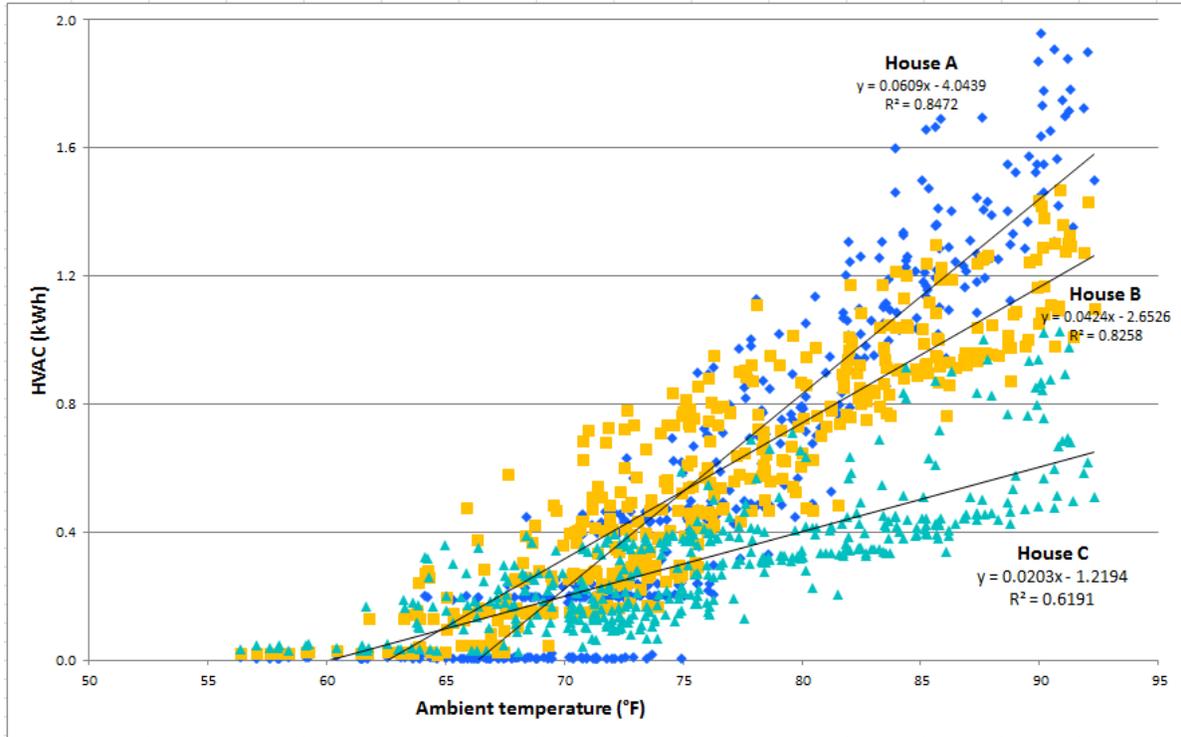


Figure 7. Cooling power relative to outdoor temperature for all three houses (Aug. 29–Sept. 15, 2014)

Table 5 summarizes the cooling performance of all three houses. During this typical summer period, House C used half as much HVAC energy as did the other houses. House B used slightly less than House A did for cooling. Average indoor temperatures were similar and close to thermostat set points. Average relative humidity was slightly higher in House C, but it was still within acceptable limits. Note that during this period House C’s fresh air ventilation rate was approximately 3.5 times higher than that of the other houses (meeting the American Society of Heating, Refrigerating, and Air-Conditioning Engineers [ASHRAE] Standard 62.2-2010), which contributed to the higher humidity. If ventilation rates in House A and House B were to have met the standard, the difference between the cooling energy consumption of House C and the others would have been greater. Also, although relative humidity was higher in House C, it is likely that other indoor air quality pollutant levels were lower because of the higher fresh-air ventilation rate.

Table 5. Cooling Statistics (Aug. 29–Sept. 7, 2014)

	House A	House B	House C
Total Cooling (Avg. kWh/d)	15.0	14.5	7.4
Average Indoor Temp. (°F)	76.4	75.9	75.4
Cooling Set Point (°F)	76.0	76.0	73.0–75.0
Avg. Relative Humidity (%)	46%	48%	59%
Air Handling Unit Fan Run Time	31%	37%	100%
Ventilation—Effective Continuous Rate (cfm)	14	12	45

3.3.1.1 Transfer Fan Operation

In House C, transfer fan thermostats were set to 76°F (24.4°C); that is, they would turn on when the thermostat controlling the fan reached that temperature. During these periods, the fans serving the rooms that had southern exposure (master bedroom and master bathroom) were on the highest percentage of the time (Table 6). The fan serving the northeast bedroom ran the least frequently.

Table 6. Transfer Fan Run Time (Aug. 29–Sept. 7, 2014)

Transfer Fan Location	Run Time (%)
Bedroom 2 (Northwest)	25
Bedroom 3 (Northeast)	12
Master Bedroom (Southeast)	55
Master Bathroom (Southwest)	49

3.3.1.2 Cooling Use When Outdoor Temperatures Were Below Indoor Cooling Target

The target temperature for all three houses during cooling was 76°F (24.4°C). The table and graphs below (Table 7, Figure 8, Figure 9, and Figure 10) identify hours during two weeks in the summer of 2014 when the cooling systems were operating coincident with outdoor air temperatures below this target cooling temperature. Opening windows during these conditions can provide free cooling and reduce overall HVAC energy consumption. As expected, the leakier and less thermally efficient the enclosure, the lower the proportion of cooling energy expended during these periods because daytime heat more rapidly dissipated to the outside. By opening windows and venting daytime heat that has built up, occupants would have had the opportunity to reduce cooling energy by up to 37% in House C, 33% in House B, and 22% in House A.

Table 7. Cooling Energy Consumption (Aug. 29–Sept. 15, 2014)

	House A	House B	House C
Total Cooling (kWh)	251	244	138
Time When Cooling Is Provided, When Outdoor Temp. Is Below 76°F (%)	11	21	51
Cooling Energy Consumption Not Required (% Cooling Energy During Hours as Defined in Row 2)	22	33	37

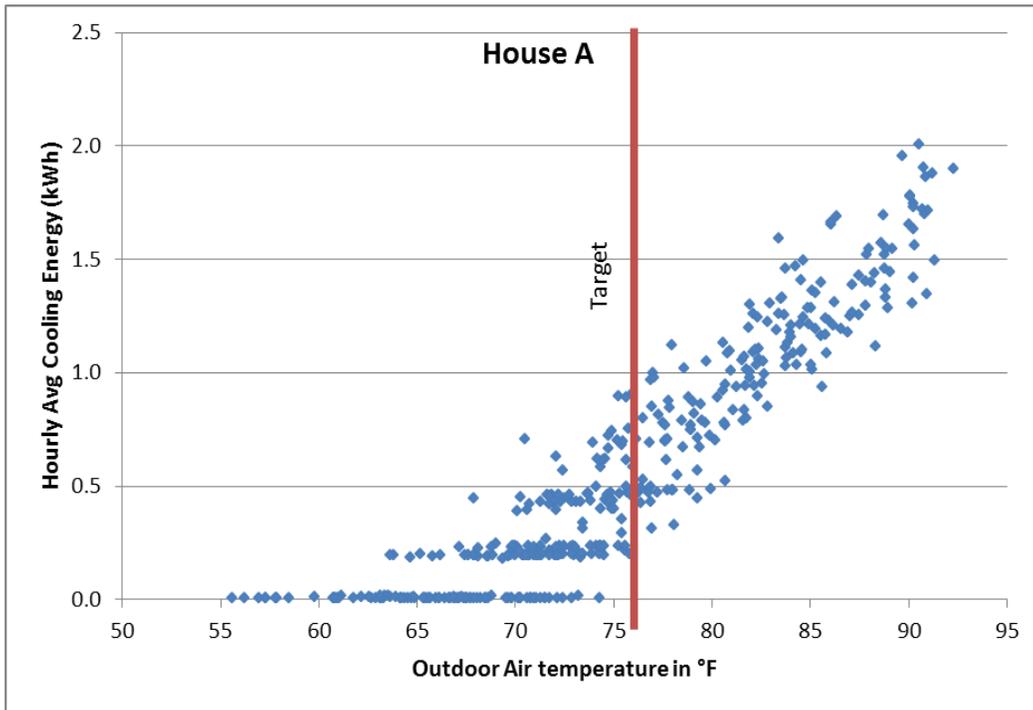


Figure 8. House A cooling energy consumption at various outdoor temperatures (Aug. 29–Sept. 15, 2014)

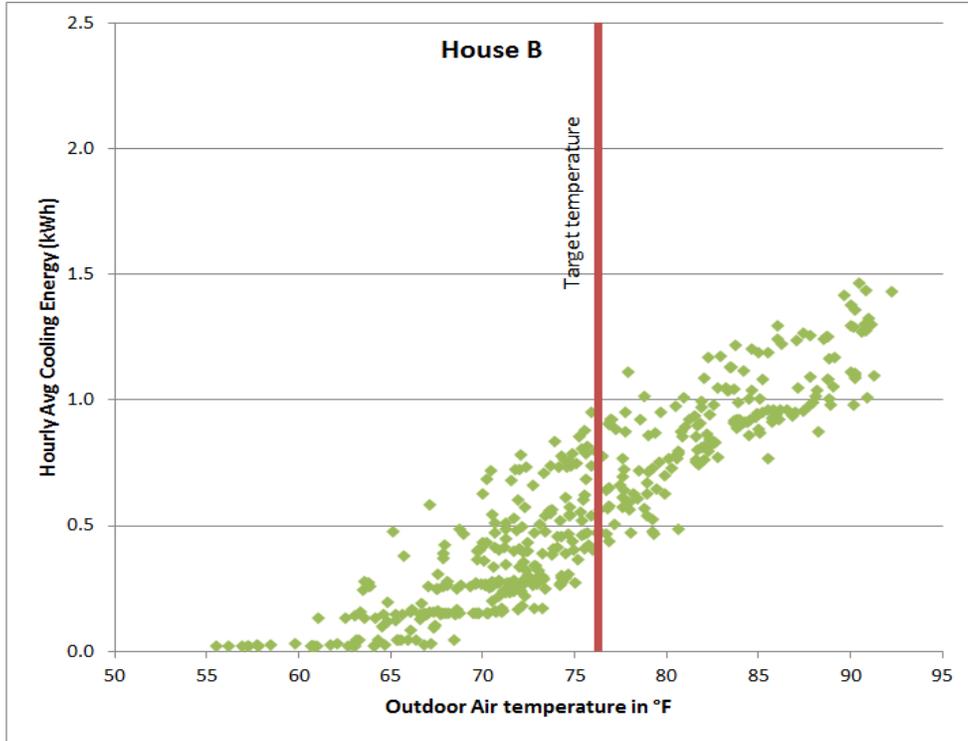


Figure 9. House B cooling energy consumption at various outdoor temperatures (Aug. 29–Sept. 15, 2014)

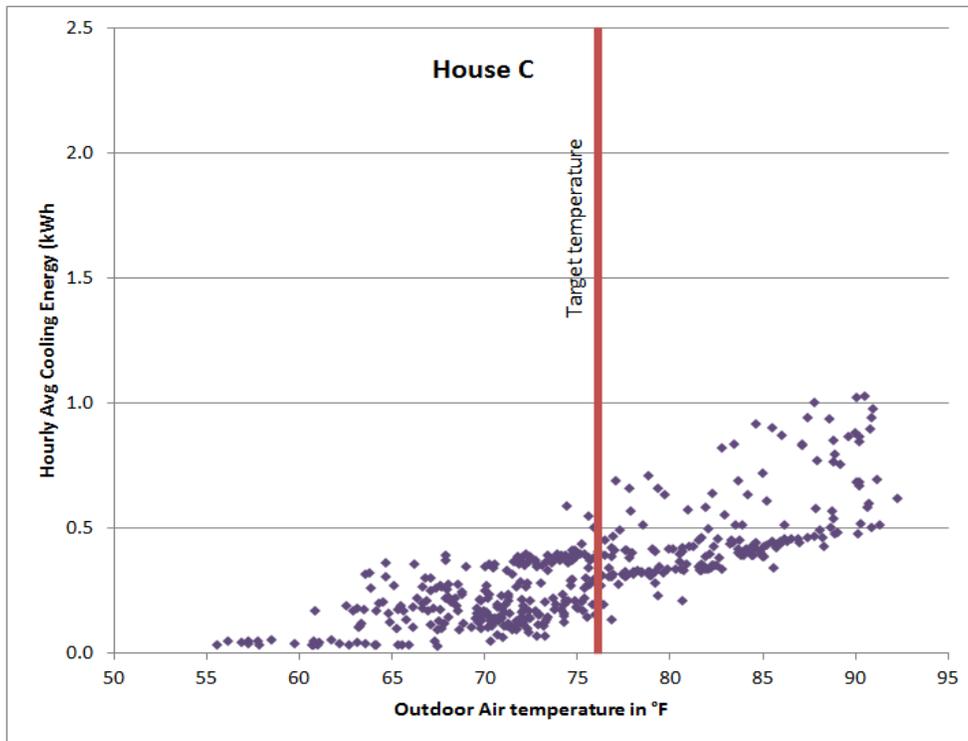


Figure 10. House C cooling energy consumption at various outdoor temperatures (Aug. 29–Sept. 15, 2014)

3.3.2 Cooling Season Comfort

Hourly room temperatures were monitored to provide an indicator of thermal comfort. The graphs below (Figure 11, Figure 12, and Figure 13) show the indoor and outdoor temperatures in each house for a typical week during the cooling season when daily high ambient temperatures reached approximately 90°F. The horizontal red line indicates the desired room temperature (76°F). The horizontal dashed lines indicate the upper and lower bounds of the +/-3°F temperature variation in cooling recommended by the Air Conditioning Contractors of America (1997). For the most part, all three houses were successful in maintaining indoor temperatures within this range. In House A, temperatures in the master bedroom and Bedroom 3 spiked the highest. Supply flows in these rooms were lower relative to the room sizes (Table 8). House B had the tightest range, indicating the best overall temperature control. House C showed the widest temperature fluctuation from one room to another, but only the master bathroom exceeded the upper bounds of the temperature range specified by the Air Conditioning Contractors of America. This room had the longest air pathway from the main living space, and it had southern exposure. The daily temperature fluctuations in the living and dining rooms are the result of the 2°F daily setback employed in House C.

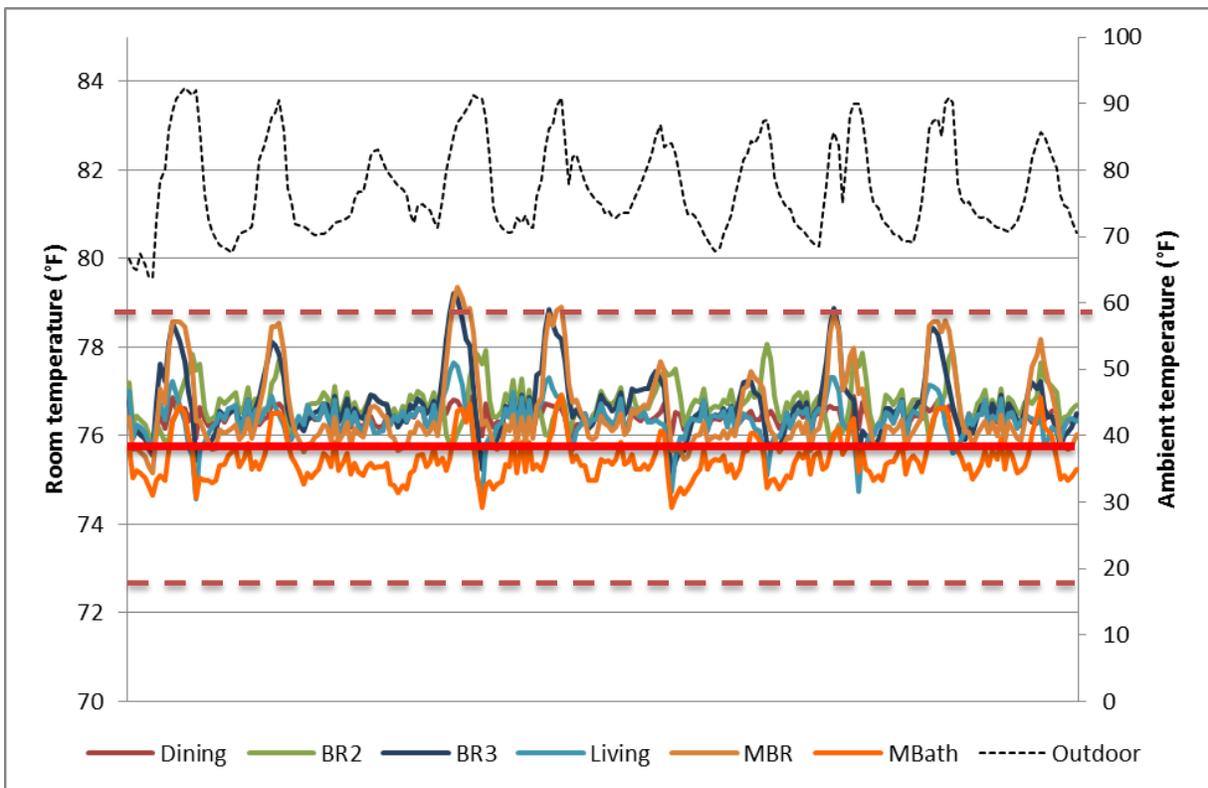


Figure 11. House A room and outdoor temperatures during the typical cooling season (Aug. 29–Sept. 7, 2014)

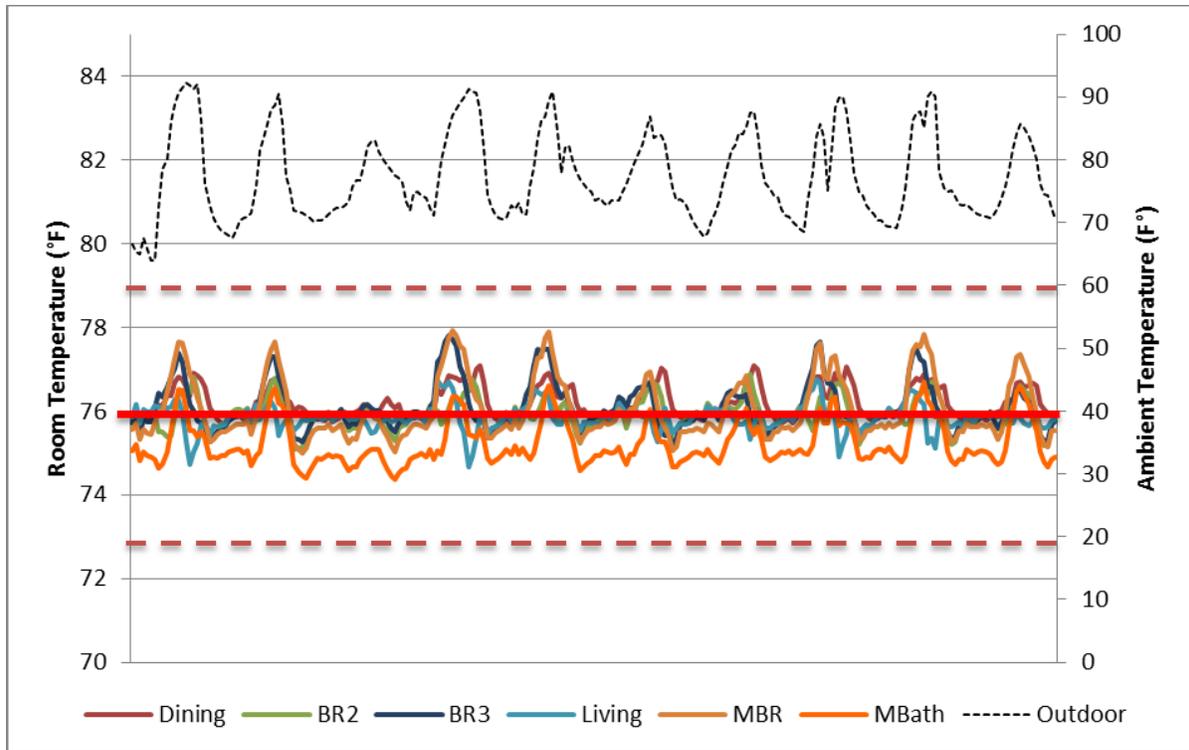


Figure 12. House B room and outdoor temperatures during the typical cooling season (Aug. 29–Sept. 7, 2014)

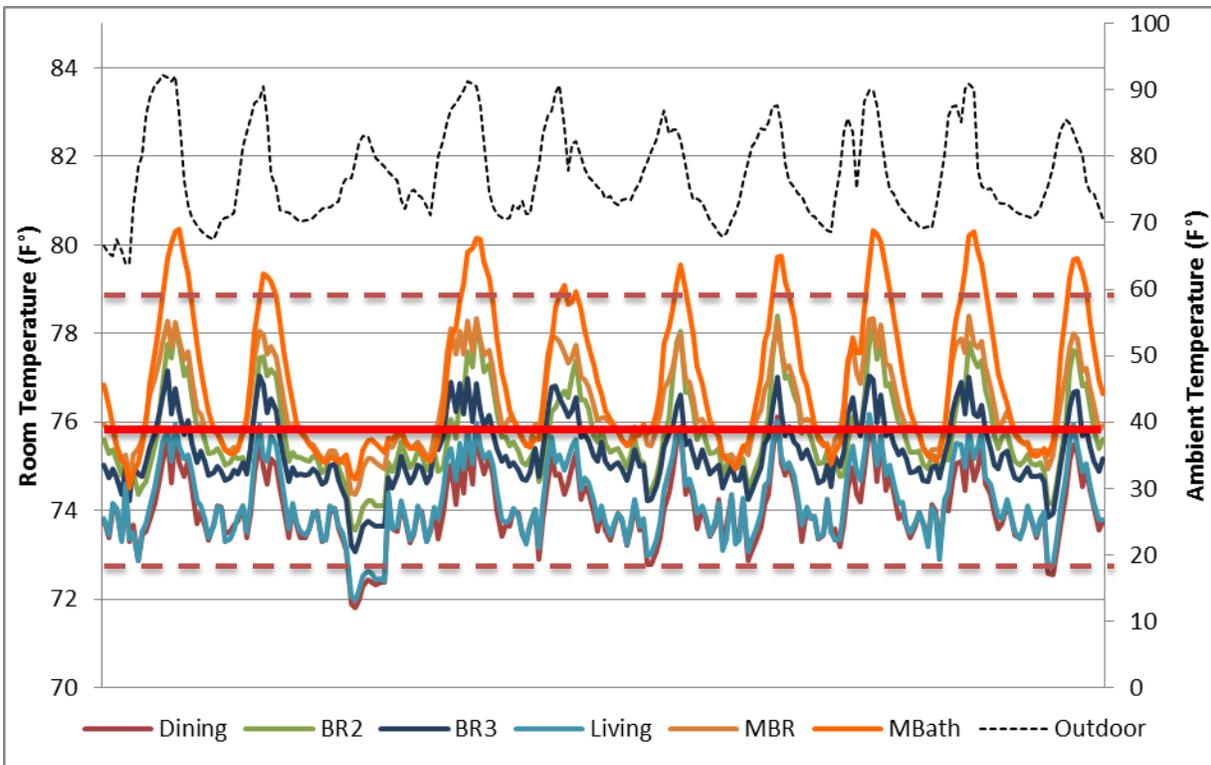


Figure 13. House C room and outdoor temperatures during the typical cooling season (Aug. 29–Sept. 7, 2014)

Table 8. House A Supply Flows

Register	Measured Supply Flow (cfm)	Room Size (sf)	Area Normalized Flow (cfm/sf)
Bedroom 2(Northwest)	136	142	1.0
Bathroom 2	Below measurement threshold	43	N/A
Kitchen	152	159	1.0
Dining Room	140	132	1.1
Master Bathroom	151	131	1.2
Master Bedroom	126	234	0.5
Living Room	116	232	1.0
Entry	124		
Bedroom 3 (Northeast)	109	137	0.8
TOTAL	1,054	1,210	0.87

3.3.3 Heating Season HVAC Energy

Table 9 summarizes the initial heating performance of all three houses in November 2014 with the interior doors open and the window blinds 50% closed. During this period, when average ambient temperature was 41.3°F, House C used 66% less HVAC energy than House A did and 8% less than House B did. House B used 63% less than did House A in heating. Average indoor temperatures were slightly lower than the thermostat set point in House B and House C. Average humidity was similar in all three homes. During this period, House C’s fresh air ventilation rate was approximately four times higher than that of the other houses (meeting the ASHRAE Standard 62.2-2010).

Table 9. Heating Statistics (Average Ambient Temperature of 41.3°F) (Nov. 12–17, 2014)

	House A	House B	House C
Total Heating (Avg. kWh/d)	48.7	18.1	16.6
Avg. Indoor Temp. (°F)	71.3	69.9	69.5
Heating Desired Temp. (°F)	71	71	71
Avg. Relative Humidity (%)	28%	30%	33%
Air Handling Unit Fan Run Time	22%	33%	100%
Ventilation—Effective Continuous Rate (cfm)	10	11	45

It was observed that bedrooms were not maintaining acceptable temperatures in House C. (See Section 3.3.4 on comfort below.) Therefore, resistance heaters with dedicated thermostats set to

69°F were added to those rooms. Table 10 summarizes the heating performance of all three houses from January 6–13, 2015, with the addition of resistance heat in House C. Interior doors were open, and window blinds were 50% closed. During this period, when average ambient temperature was 32.6°F, House B and House C used a similar amount of heating energy and slightly less than half the heating energy of House A.

Average indoor temperatures were close to the thermostat set point in all houses. Average humidity was similar in all three homes. During this period, House C’s fresh air ventilation rate was approximately two to three times higher than that of the other houses.

Table 10. Heating Statistics with Resistance Heat in House C Remote Rooms (Average Ambient Temperature of 32.6°F) (Jan. 6–13, 2015)

	House A	House B	House C
Total Heating (Avg. kWh/d)	66.6	32.3	31.8
Avg. Indoor Temp. (°F)	71.2	69.9	70.4
Heating Desired Temp. (°F)	71	71	71
Avg. Relative Humidity (%)	20%	21%	22%
Air Handling Unit Fan Run time	48%	47%	100%
Ventilation—Effective Continuous Rate (cfm)	21	15	45

Figure 14 shows heating power compared to ambient temperature. House B and House C used less heating energy than did House A at all outdoor temperatures. House B and House C used a similar amount of heating energy, but the trend lines indicate that as temperatures fell House C used less energy relative to House B. At colder temperatures, the heating system in House B relied more on electric resistance and less on the heat pump, whereas House C’s heat pump ran continuously (Figure 14). Figure 15 shows that in House B the backup electric resistance heat turned on when the outdoor air temperature fell below approximately 40°F, and it increased with decreasing temperatures. During this period, the heat pump showed stable and limited energy consumption.

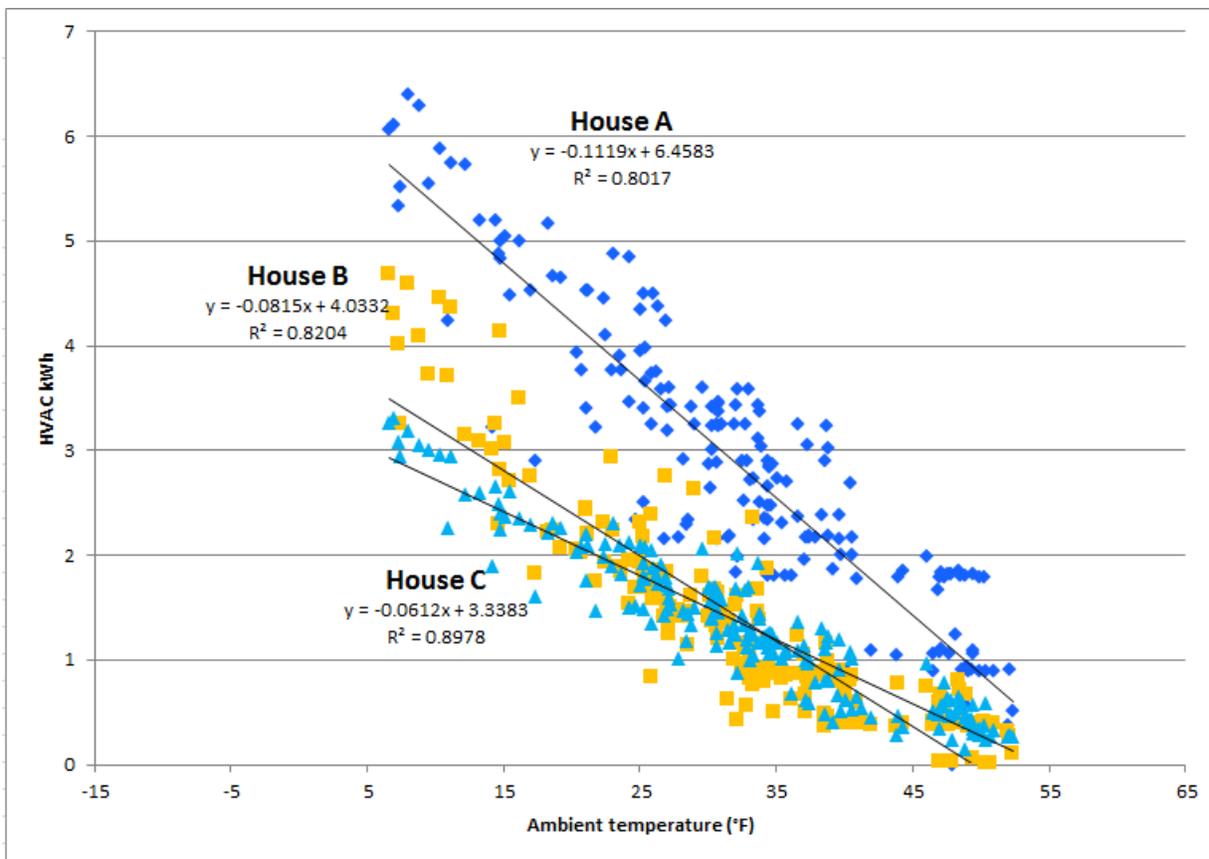


Figure 14. Heating energy compared to outdoor temperature for all three houses (Jan. 6–13, 2015)

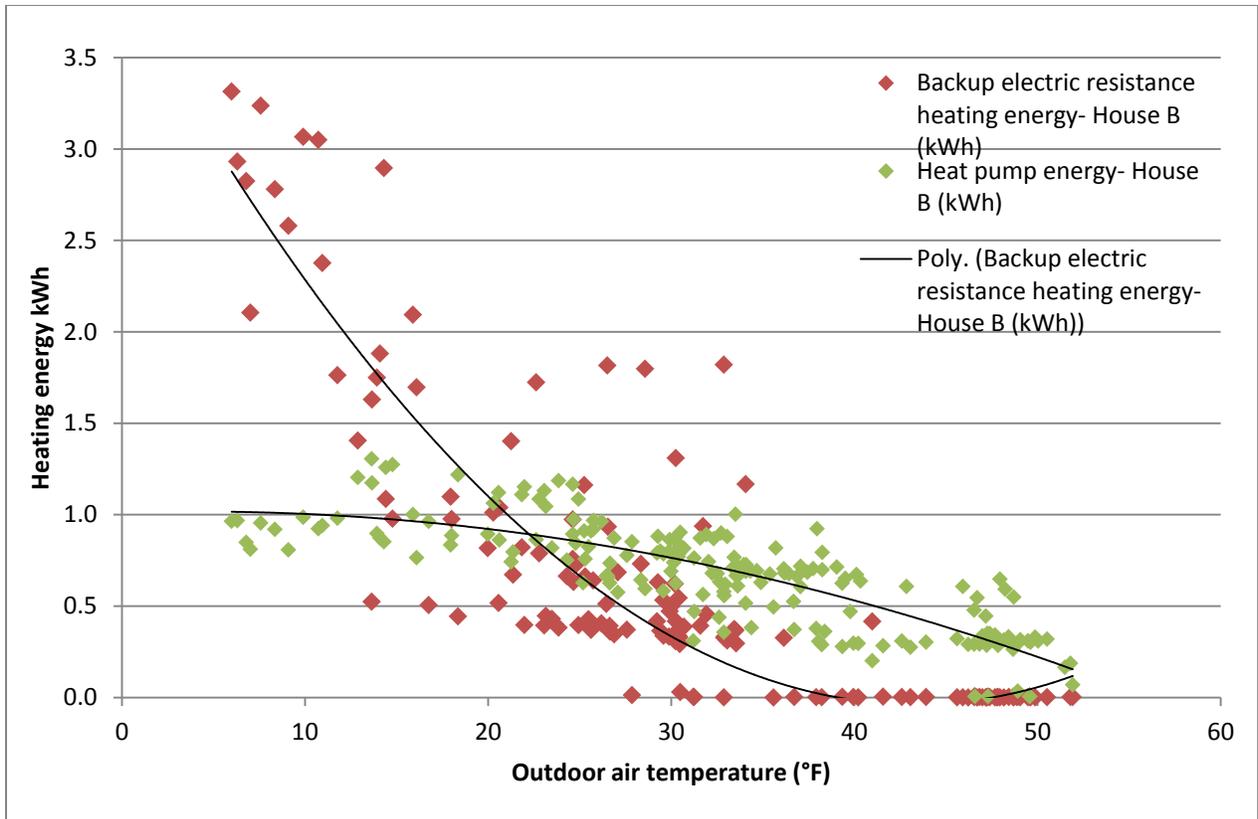


Figure 15. House B backup electric resistance heating energy compared to outdoor air temperature

Figure 16 shows average hourly electrical power used by the three components of the heating system in House C: the heat pump, the resistance heaters, and the transfer fans. The black line is the ambient temperature (°F). The heat pump and resistance heaters used similar amounts of heating energy, with resistance heating becoming a significant factor when the ambient temperatures fell below approximately 35°F. During this period, 46% of the space-conditioning energy use was attributable to the heat pump, 8% to the fans, and 46% to the resistance heaters.

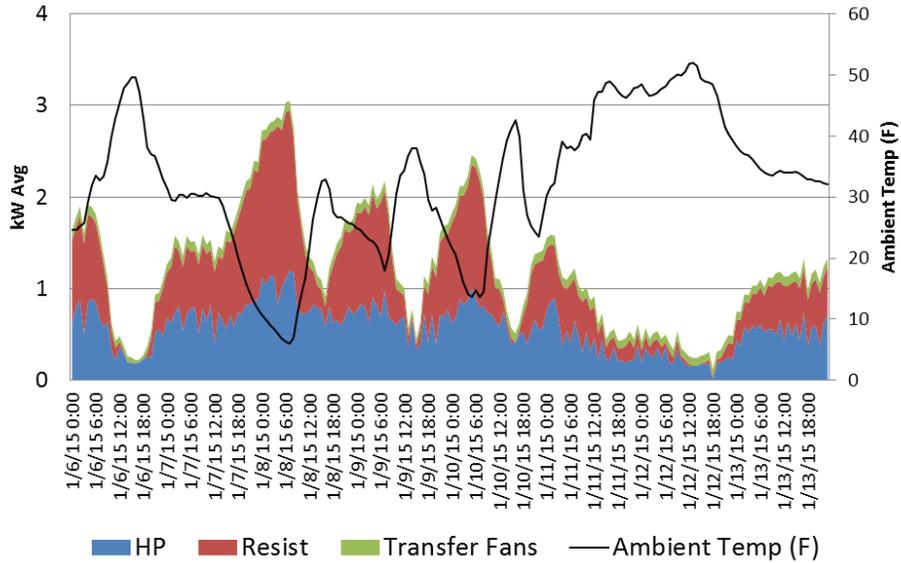


Figure 16. House C heat pump, transfer fan, and resistance heating energy compared to ambient temperature (Jan. 6–13, 2015)

House C’s backup resistance heaters helped maintain temperatures in the bedrooms and the master bathroom at 69°F.

Figure 17 shows that when the outdoor air temperatures were below approximately 40°F, the resistance heat consumption started climbing, and it surpassed the heat pump energy consumption when the ambient temperatures fell below approximately 25°F.

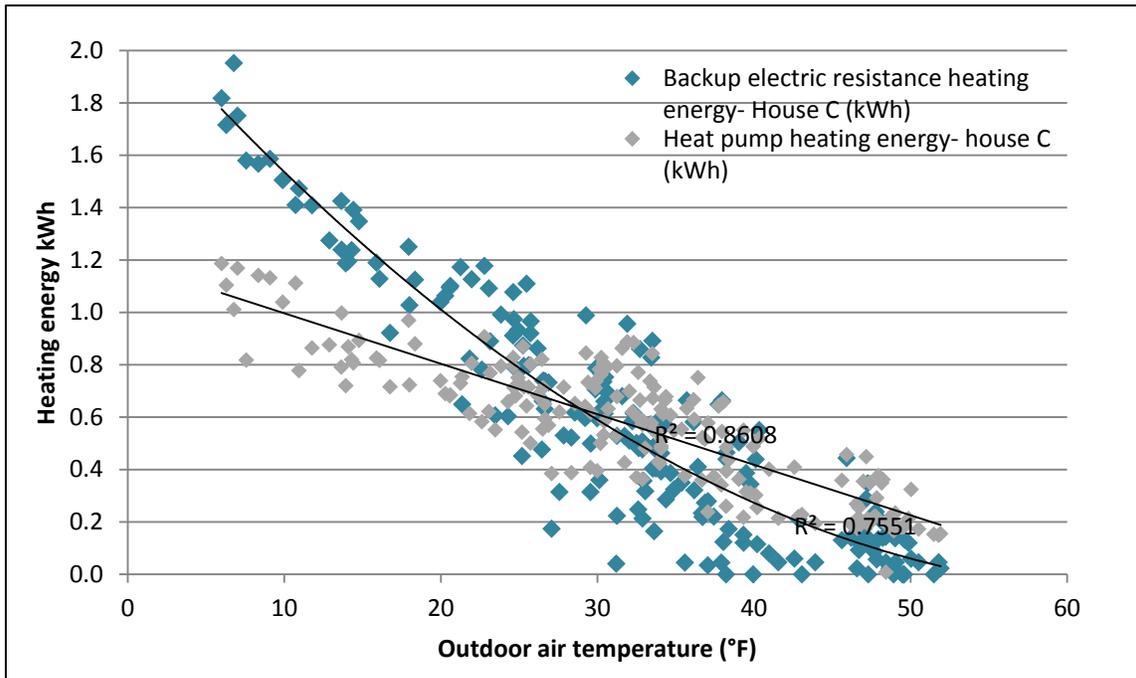


Figure 17. House C resistance heating energy compared to ambient temperature (Jan. 6–13, 2015)

3.3.4 Heating Season Comfort

As discussed above, an analysis of the indoor temperatures in House C revealed that the remote rooms were not maintaining acceptable temperatures (Figure 18). In particular, the master bathroom and master bedroom were falling well below the Air Conditioning Contractors of America acceptable heating variation from the set point limit ($\pm 2^{\circ}\text{F}$ from the set point in heating) when the outdoor temperatures fell below approximately 50°F . It was therefore decided to add electric resistance heat to the bedrooms and master bathroom controlled on independent thermostats to maintain at least 69°F in those rooms.

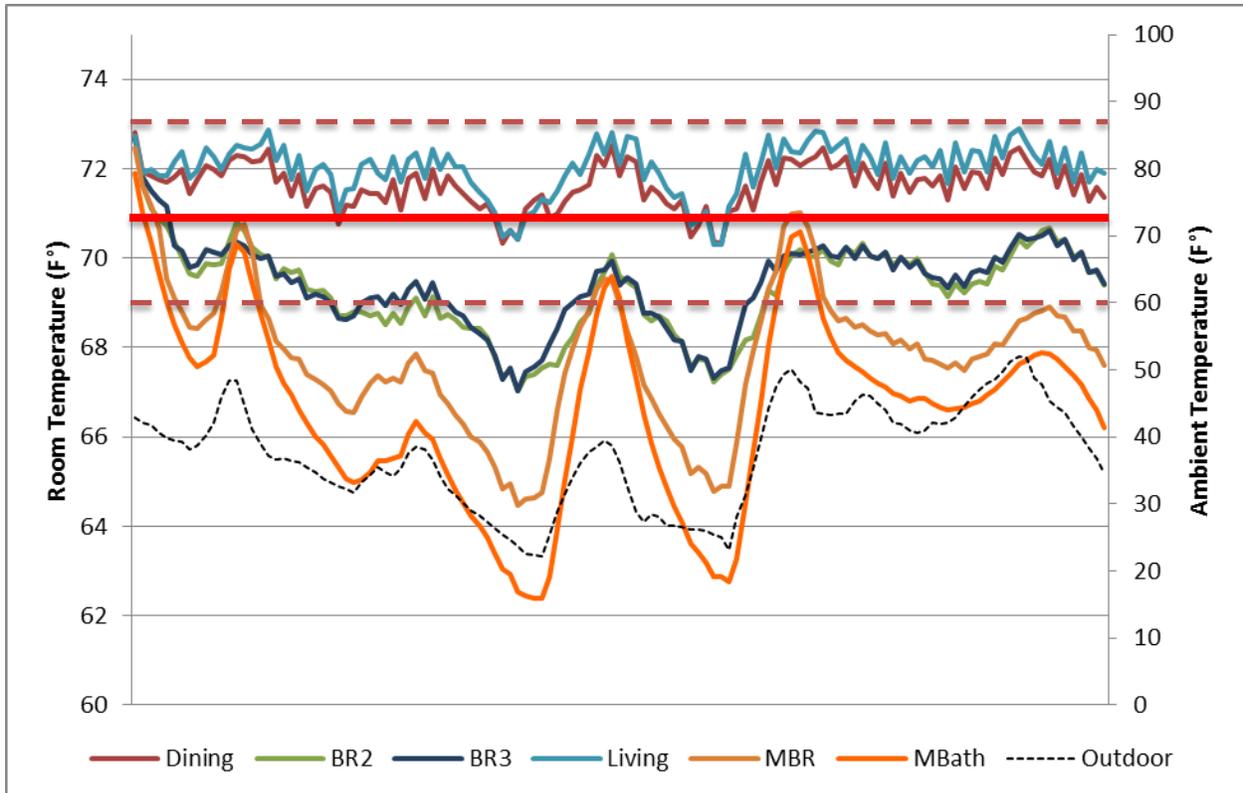


Figure 18. House C room and outdoor temperatures for the initial heating season analysis (Nov. 12–17, 2014)

After the resistance heat was activated, the bedrooms still remained cooler than the main space (at or above the 69°F electric heat set point); however, the comfort metrics met the Air Conditioning Contractors of America guidelines (Figure 19).

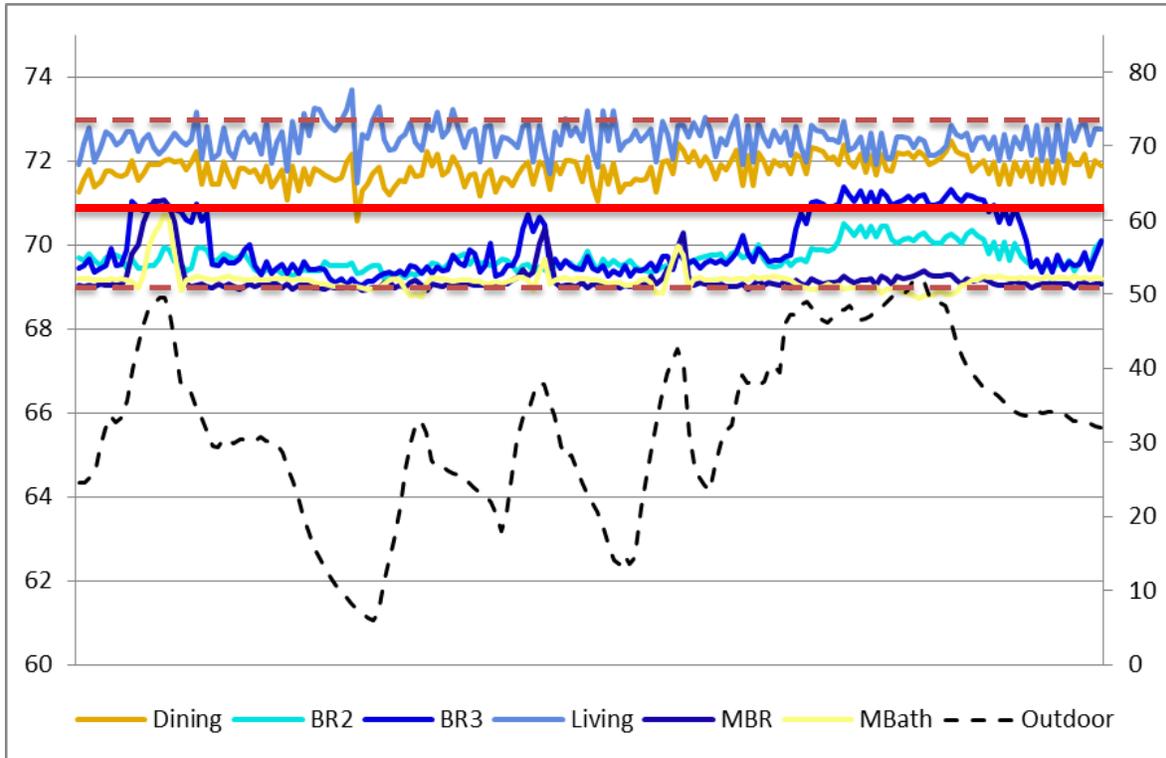


Figure 19. House C room and outdoor temperatures with resistance heat in remote rooms (Jan. 6–13, 2015)

For comparison purposes, the room temperatures in House A and House B are shown in Figure 20 and Figure 21, respectively. Bedroom 2, Bedroom 3, and sometimes the master bathroom struggled (sometimes unsuccessfully) to maintain 69°F in both House A and House B.

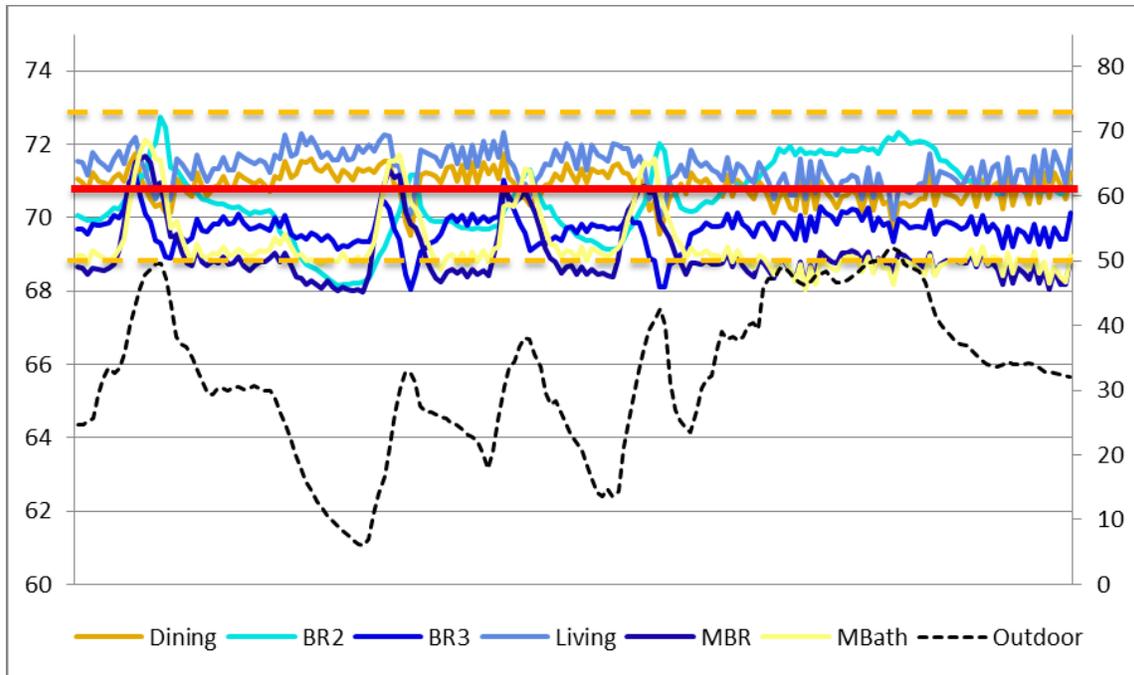


Figure 20. House A room and outdoor temperatures (Jan. 6–13, 2015)

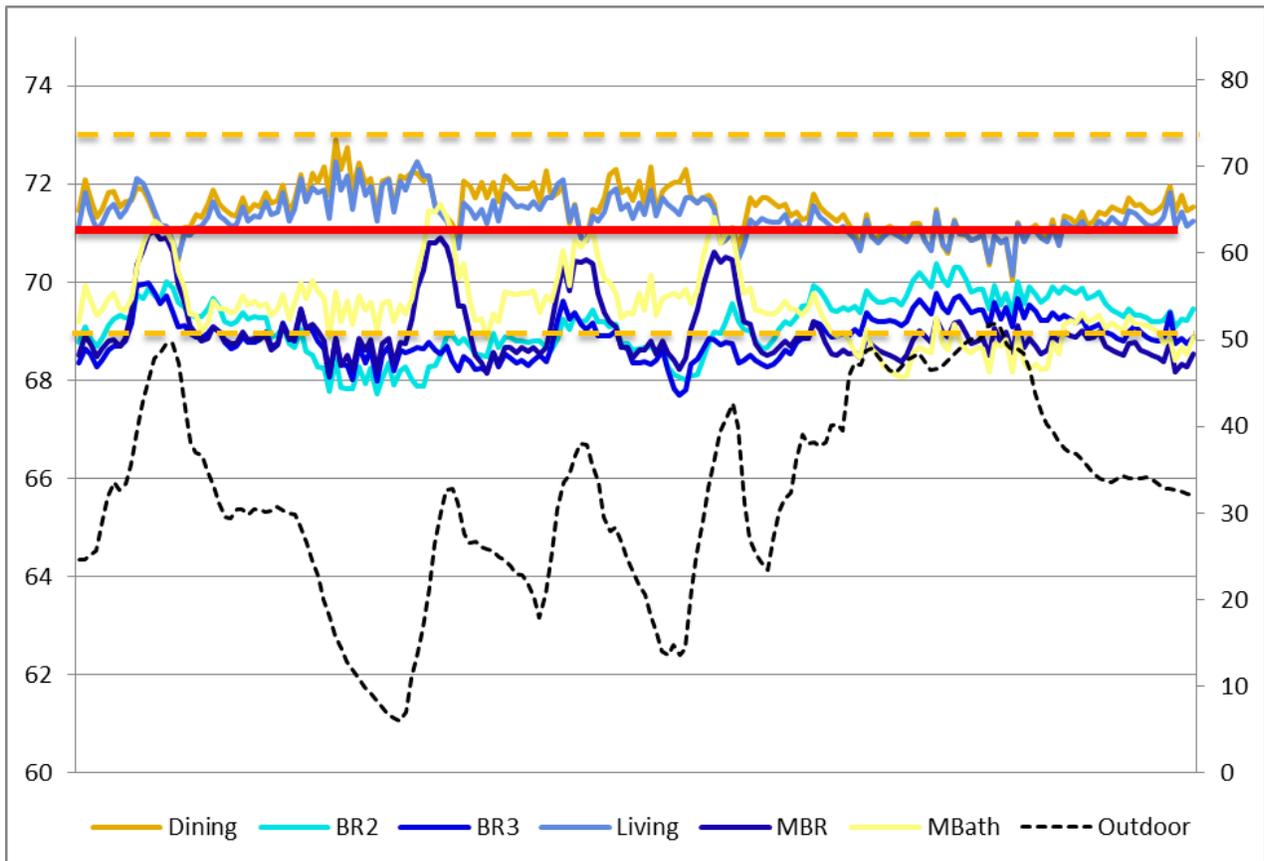


Figure 21. House B room and outdoor temperatures (Jan. 6–13, 2015)

3.3.5 Whole-House Ventilation

Whole-house ventilation in House A and House B is achieved through the air handling unit, as is typical for manufactured homes. A fresh-air intake at the roof is connected via a 4-in-diameter flexible duct to the return side of the air handling unit. There is no mechanical damper.

In House C, a continuously operating exhaust fan in the main living space provides ventilation in accordance with ASHRAE Standard 62.2-2010. This fan was initially planned for the master bathroom to help draw conditioned air from the living space to the master suite, but code required it to be in a central location.

Table 11 compares the effective ventilation rates in each house to the requirements of HUD (2010) and ASHRAE Standard 62.2-2010. The rates in House A and House B were lower than the HUD recommendation of 0.035 ft³ per ft², and all three houses were lower than the HUD minimum of 50 cfm.

Table 11. Effective Ventilation Rates

House	Whole-House Ventilation Flow (cfm)	
	Measured	Required ^a
A	22	50
B	13	50
C	45	50

^a The required whole-house ventilation rate should be 0.035 ft³ per square foot of the conditioned space or a minimum of 50 cfm. Conditioned area = 1,210 ft².

3.3.6 Moisture Measurements

One risk of adding exterior foam insulation to the walls in House C was that the reduced vapor permeability of the foam sheathing (extruded polystyrene insulation) could lead to higher moisture content in the wall cavity. To explore the potential for condensation within the wall cavities with exterior foam sheathing, air temperature and relative humidity in a cavity between the sheathing and interior fiberglass insulation on a north wall was measured to estimate the condensation potential in the cavity. A temperature and relative-humidity sensor was installed approximately 2.5 in. above the floor inside the cavity. The moisture content of a wood stud in the cavity was also measured at the same location using an Omnisense S900-1 moisture content sensor (Figure 22).



Figure 22. Moisture content and temperature/relative humidity sensors mounted in wall cavity

Figure 23, Figure 24, and Table 12 show the data from the moisture content sensors in the wall cavities in House B and House C. Wood moisture content was slightly higher in House C, but it was still well within the safe limits of 19% (Forest Products Laboratory 2015). Additionally, the House C wall cavity temperature was moderated by the exterior insulation, which resulted in a minimum temperature that was 5.5°F higher than it was in House B, mitigating condensation risk.

House C's moisture content tended to be somewhat higher in the summer, consistent with higher overall humidity in the house. House B had slightly higher readings in the winter. Note that the houses were unoccupied, and no moisture was added to simulate latent loads. Because the homes were newly constructed in the plant, wood was not subject to wetting, but it could have had residual natural moisture. Seasonal data seems to follow the expected trend—higher in the summer and lower in the winter. Note that the vinyl-covered wallboard used in all three homes has been associated with moisture problems in warm, humid climates (Moyer et al. 2001).

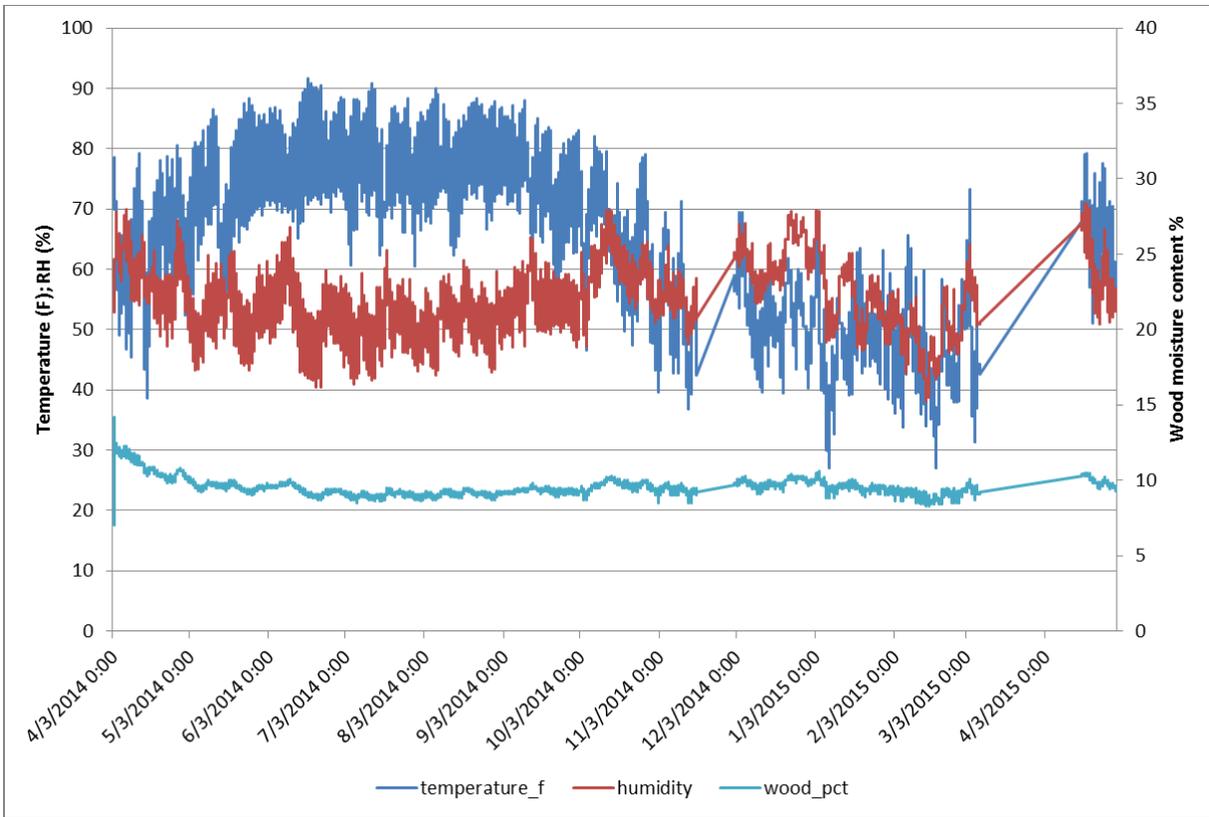


Figure 23. House B wall cavity conditions and wood moisture content (April 2014–April 2015)

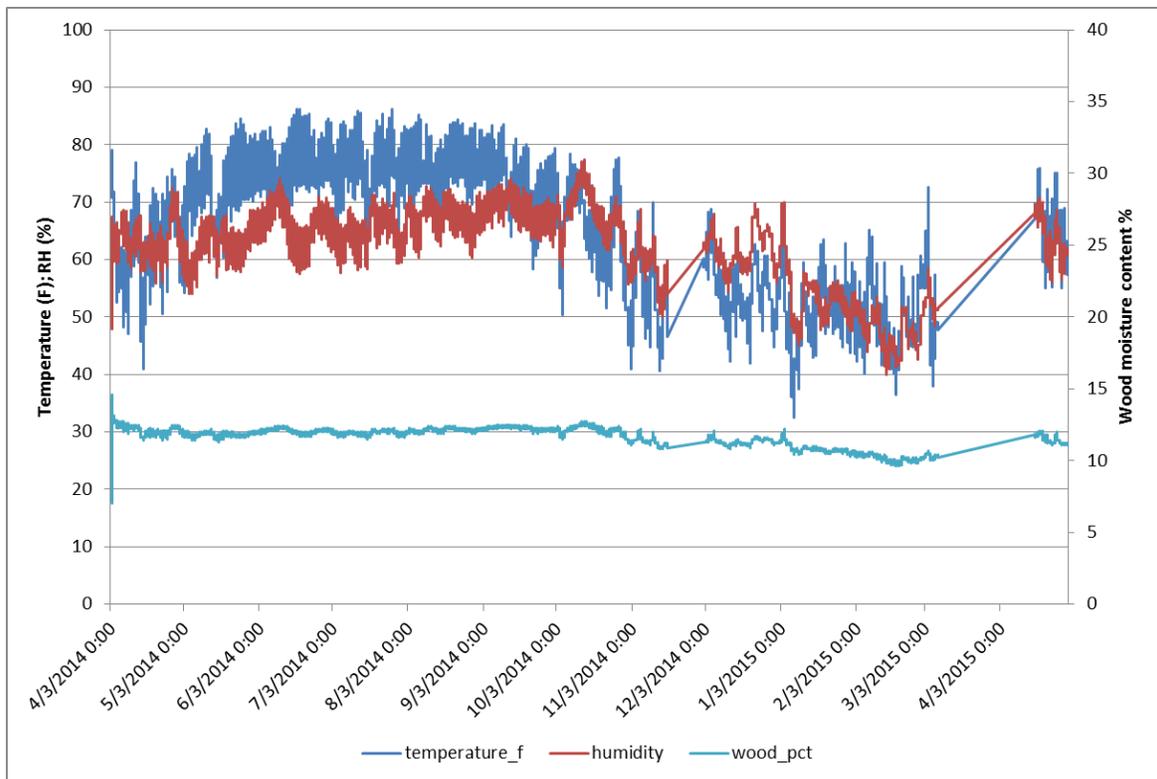


Figure 24. House C wall cavity conditions and wood moisture content (April 2014–April 2015)

Table 12. Wall Cavity Maximum, Minimum, and Average Conditions (April 2014–April 2015)

House		Temp. (°F)	Humidity (%)	Wood Moisture Content (%)	Dew Point (°F)
B	Maximum	91.6	71.0	14.2	67.2
	Minimum	27.0	38.2	7.0	7.9
	Avg.	64.8	54.7	9.5	48.0
C	Maximum	86.2	77.4	14.6	73.3
	Minimum	32.5	40.0	7.0	15.9
	Avg.	65.2	62.2	11.6	52.1

3.3.7 Peak Electric Demand

Peak electric demand is calculated by the utility serving Russellville (Tennessee Valley Authority) as the single half-hour period with highest average power (kW) draw in the month. The coincident peak hours of the Tennessee Valley Authority system are as follows:

- Winter: December–March, 6:00 a.m.–8:00 a.m.
- Summer: June–September, 2:00 p.m.–5:00 p.m.

Figure 25 shows the peak demand for the three houses using the criteria from the Tennessee Valley Authority. Some data is missing from February and March because experiments run during those months interfered with normal house operation. Peak demand in House C was significantly lower than the other two houses during all months of the year. The heat pump in House C operated nearly continuously, so there was much less variation in power draw. House B had somewhat lower peaks than did House A during most months, but some winter peaks were similar, indicating that House B’s peak occurred when the heat pump was not running and the house was relying solely on electric resistance backup. On average, during the peak hours of the Tennessee Valley Authority system, House B had an 18% lower peak demand than did House A, and House C had a 69% lower peak demand than did House A.

Interestingly, when broadening the range of hours to show peaks for any time of day, House B had higher winter peak demand than did House A during most months (Figure 26).

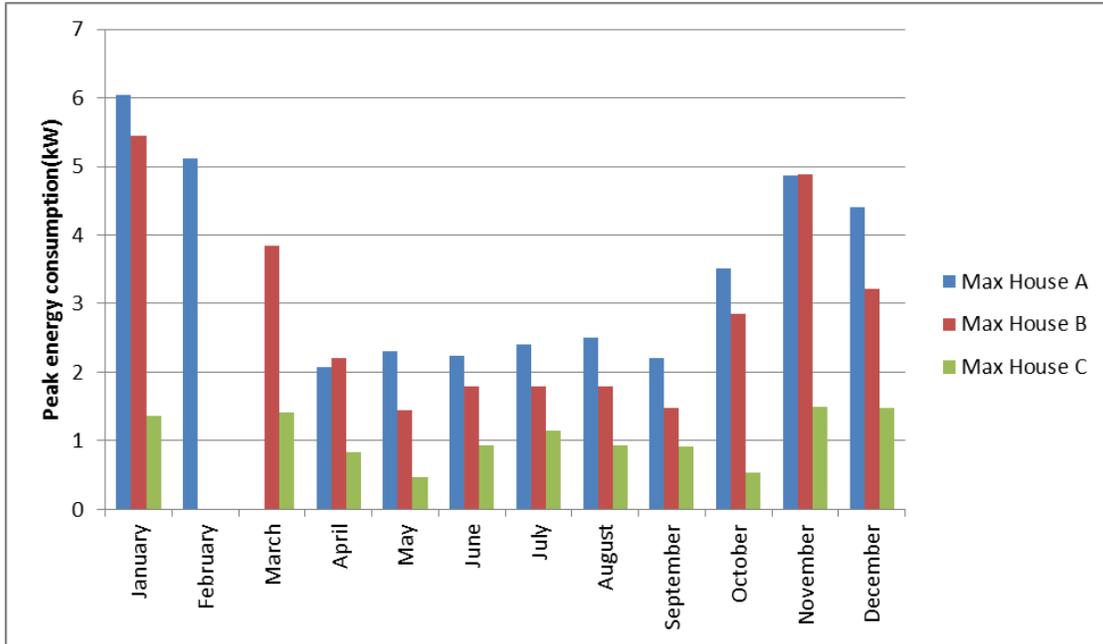


Figure 25. Monthly peak demand during coincident peak hours of the Tennessee Valley Authority system

Table 13. Summary of Average Peak Demand Reduction During Peak Hours of the Tennessee Valley Authority System

House	Avg. Monthly Peak Demand During Peak Hours	Avg. Demand Reduction Compared to House A
A	3.1	N/A
B	2.6	18%
C	1.0	69%

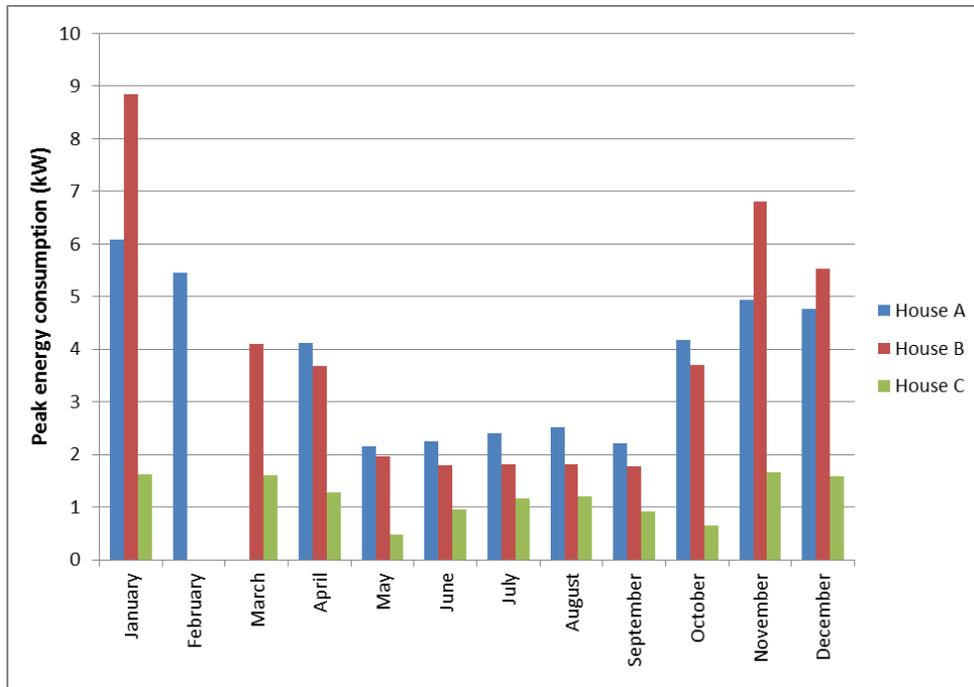


Figure 26. Monthly peak demand during all hours of the day

3.3.8 Co-heat and Tracer Gas Testing

Short-term co-heat testing in cold weather using portable electric resistance space heaters was conducted in all three houses to measure whole-house heat loss. The test data was used to obtain envelope U-value information that was used to tune the simulation parameters. During the test, six portable heaters (one in each bedroom, one in the master bathroom, one in the living room, and one in the kitchen) were used to maintain 70°F throughout the duration of the test. Other heating equipment was switched off. Temperature sensors were used to turn the heaters on and off through the Campbell Scientific data loggers. Tracer gas decay testing was conducted simultaneously with the co-heat testing to differentiate convective from conductive heat loss.

These tests were conducted during four days in December 2014 (December 8–11). Co-heat tests were run in each house on three nights. Tracer gas measurements were made in conjunction with some of the co-heat tests (Table 14).

Table 14. Testing Schedule

	A	B	C
Dec. 8–9	Co-heat	Co-heat	Co-heat
Dec. 9–10	Co-heat	Co-heat Tracer gas	Co-heat Tracer gas
Dec. 10–11	Co-heat Tracer gas	Co-heat	Co-heat Tracer gas
Dec. 11–12	Tracer gas	Tracer gas	

3.3.8.1 Co-Heat Testing

The co-heat test procedure was as follows:

1. The continuous exhaust fan in House C was turned off.
2. Heating systems in all houses were turned off.
3. Electric space heaters, controlled by the data logger based on each room’s indoor temperature, were used to maintain constant indoor temperature overnight in all rooms. Each room had its own heater and temperature sensor.
4. For two tests in each house, a simultaneous tracer gas test (see procedure below) was conducted to measure the infiltration rate during the co-heat testing period.
5. Fans were used to evenly distribute the heat and the tracer gas.
6. The energy consumed by the heaters, fans, and data collection equipment was measured continuously.
7. The heat loss of the building, inclusive of infiltration, was calculated via the following equation:

$$UA = IG / (T_{out} - T_{in})$$

Where:

UA = Heat loss in kW per degree C

IG = Total internal heat gain in kW

T_{out} = Ambient outdoor temperature (°C)

T_{in} = Average indoor temperature (°C).

3.3.8.2 Tracer Gas Testing

Infiltration was measured in all three homes utilizing tracer gas. The objectives of the tracer gas testing were to determine infiltration levels in all three houses for calibrating models and quantifying the effectiveness of the ventilation systems. Tracer gas testing was conducted simultaneously with co-heat testing so that envelope infiltration losses could be separately accounted for in the resulting heat loss estimates. The tracer gas test procedure was as follows:

1. The houses were set up per the co-heat procedure above on December 9–10 and December 10–11. For the December 11–12 test, the space-conditioning systems were on.
2. Air sampling tubes were run to each room and connected to a Brüel and Kjær Model 1302 Multi-Gas Monitor System. A Brüel and Kjær 1303 Multipoint Sampler was used in House C to separately measure the sulfur hexafluoride tracer gas concentration in each room (Figure 27). In House A and House B, the room sample tubes were connected to a common manifold.
3. Sulfur hexafluoride was released into the space to an approximate concentration of 3–4 ppm.
4. The tracer gas monitoring systems measured the sulfur hexafluoride concentration at 5-min intervals.
5. The decay rate of sulfur hexafluoride was converted into a natural air change rate via the following equation:

$$ACH_{natural} = \ln(C1/C2)$$

Where:

ACH_{natural} = natural air change rate per hour for hour n

ln = natural logarithm

C1 = Average sulfur hexafluoride concentration in parts per million for hour n-1

C2 = Average sulfur hexafluoride concentration in parts per million for hour n.



Figure 27. Tracer gas sampling equipment and sampling tube

The results of the tracer gas testing are summarized in Table 15 along with a comparison to the blower door testing and the natural air change rates predicted from the blower door test results using the Alberta Air Infiltration Model (AIM-2) incorporated into BEopt V 2.3.0.2 (Walker and Wilson 1998). The tracer gas testing data are provided in Appendix E. Results for the multiple

tests in House B and House C are consistent from one test to another, but they are higher than the annual average AIM-2 predictions.

Table 15. Tracer Gas Test Results Summary

	House A	House B	House C
Dec. 8–9		0.22	0.14
Dec. 9–10		0.22	0.14
Dec. 10–11	0.25	0.24	
Overall Avg.	0.25	0.23	0.14
Blower Door Air Change Rate (ACH50)	4.7	4.6	3.8
Air Change Rate Extrapolated from Blower Door Test^a	0.13 (annually)	0.12 (annually)	0.10 (annually)

^a Based on AIM-2 used in BEopt V 2.3.0.2

One set of tracer gas tests was conducted with space-conditioning and ventilation systems operating (Table 16). The continuous exhaust ventilation fan in House C (45 cfm) resulted in a higher air change rate than in House A and House B. The air change rate in House B was higher with systems operating than with systems off. However, there was no measured increase in the House A air change rate when the system was operating compared to when the system was off.

Table 16. Tracer Gas Test Results Summary—With Systems Operating

	House A Dec. 11–12	House B Dec. 11–12	House C Dec. 8, 8–10 a.m.
Estimated Continuous Effective Mechanical Ventilation Rate Based on Commissioning Tests and Air Handling Unit/Exhaust Fan Run Time (cfm)	22	13	45
ACH Natural from Tracer Gas Test	0.25	0.27	0.28
ACH Natural from Tracer Gas Tests Converted to cfm	40 cfm	44 cfm	45 cfm

The results of the daily co-heat testing are summarized in Table 17.

Table 17. Co-heat Test Results Summary (Btu/h/°F)

Test	A	B	C
Dec. 10	326	250	202
Dec. 11	299	237	212

The results of the co-heat testing are summarized in Table 18 along with a comparison to the UA value (heat loss over the entire surface area) predicted by the house designs. The co-heat testing data are provided in Appendix G. The results are in reasonable agreement.

Table 18. Co-Heat Test Results Summary Adjusted for Infiltration

House	Co-Heat Results (Incl. Infiltration Losses) (Btu/h/°F)	Avg. Infiltration Rate per Tracer Gas Test (ACH Natural)	Infiltration Heat Loss (Btu/h/°F)	Heat Loss per Co-Heat Test (Without Infiltration Loss) (Btu/h/°F)	Difference from Design UA Value (%)
A	333	0.25	43	290	-3.2
B	256	0.24	41	215	-8.5
C	208	0.14	24	184	0.4

The heat loss factors predicted by the results of the co-heat/tracer gas tests were used to calibrate the BEopt models and to calculate the heating system COPs.

3.3.9 Heating System COP

The COP of the heating system was calculated for all three houses using a co-heat method. For House B and House C, the COP of the heat pumps was also measured using airflow measurements.

3.3.9.1 Co-Heat Tests

Co-heat tests were conducted during two nights to determine each house’s actual heat loss (UA) inclusive of infiltration and duct leakage. The hourly monitored data from the co-heat tests is provided in Appendix G. Ventilation systems were not operating during the co-heat tests.

To calculate the system COPs, the UAs derived from the co-heat tests were used to calculate the heat losses during periods when the systems were operating. Because the derived UAs did not include the effects of fresh air ventilation, this was added to the heat loss based on measured ventilation flows and indoor-outdoor temperature differences. COPs were calculated for 1-h periods from 1:00 a.m. to 6:00 a.m. when the impact of solar gains would be smallest. The equation used to calculate simple heat loss during system operation was:

$$Q = UA * (T_{out} - T_{in}) + V * cp * (T_{out} - T_{in})$$

Where:

Q = Simple heat loss

UA = Area-weighted U value of the house’s envelope, including walls, roof, floor and fenestration, inclusive of infiltration

T_{out} = Outdoor ambient temperature

T_{in} = Average indoor temperature (average of all room temperature sensors)

V = Effective ventilation rate in cubic feet per minute—average for each hour

c_p = Specific heat of air approximated as 1.08⁵

Note that this equation assumes that there is no heat recovery in the ventilation heat flow through the building assemblies. COPs were then calculated using the following equation:

$$COP = (Q - IG) / Qc$$

Where:

Q = Simple heat loss in kW

IG = Internal gains in kW, measured

Qc = Heat pump compressor energy in kW, measured

3.3.9.2 Air-Side Measurements

The COPs in House B and House C were also calculated based on measurements of the supply and return air temperatures, fan power measurements, and onetime fan curve generated by measuring the air flow at a range of fan speeds. The following equations were used:

$$q_{c,tot} = m_{da}(hr - hs)$$

$$COP = q_{c,tot} / q_{elec}$$

Where:

q_{c,tot} = Total heating rate (sensible + latent) (kW)

q_{elec} = Electrical power drawn by outdoor unit (kW)

m_{da} = Dry air mass flow rate (kg/s)

hr = Enthalpy of return air (kJ/kg dry air)

hs = Enthalpy of supply air (kJ/kg dry air)

Enthalpy was calculated using an assumed total atmospheric pressure of 98.9 kPa and measurements of the air’s temperature and relative humidity. Correlations for enthalpy as a function of air properties are taken from the 2001 *ASHRAE Handbook—Fundamentals*, SI P6.9, Equation 32.

⁵ Converts cubic feet per minute into pounds per hour and multiplies times specific heat of air: 1 cfm * 60 minutes per hour = 60 CFH * .075 pound per cubic foot (standard air) = 4.5 * 0.24 specific heat of air = 1.08.

Air flow rate was calculated using a curve fit to the measured air flow rates as a function of electrical power draw of the fan. (See curve below.) The curve fit results in a prediction of volume flow rate. To calculate m_{da} , the density of the air on the supply side was calculated from the measured temperature and relative humidity and assumed total atmospheric pressure of 98.9 kPa. The mass flow rate of the dry air was then calculated:

$$m_{da} = \rho v_{ma} / (1 + \omega)$$

Where:

ρ = Density of moist air (kg/m^3)

v_{ma} = Volume flow rate of moist air (m^3/s)

ω = Humidity ratio of moist air (kg water/kg dry air)

The correlations for the density of air were taken from the 2001 *ASHRAE Handbook—Fundamentals*, Chapter 6, Equation 11 and Equation 28.

3.3.9.3 Results

The UAs calculated from the co-heat tests and average COPs for each house under normal operation are shown in Table 19. The hourly data is provided in Appendix G. An estimate for the COPs without the adjustment for ventilation is also included to provide a lower bound for the COP in the event that some natural heat recovery was present in the infiltration flow through the building assemblies.

Table 19. Summary of Heating System COPs

House	A	B	C
Equipment Make and Model Number	NORDYNE Electric Furnace (E3EB010H)	Intertherm Heat Pump (DT4BD-018KB & C8 (B,D)AM01830U-B)	Mitsubishi (MSZ-FH15NA & MUZ-FH15NA)
UA (Incl. Infiltration) Btu/h/°F	313	245	209
COP (Co-Heat Method) and Dates of Measurement (With Ventilation Adjustment)	1.10 Jan. 1–2	2.50 Jan. 1–2, 5, 7, 14	2.49 Jan. 8–12, 14, 16, 25
COP (Co-Heat Method) and Dates of Measurement (Without Ventilation Adjustment)	1.00 Jan. 1–2	2.26 Jan. 1–2, 5, 7, 14	1.63 Jan. 8–12, 14, 16, 25
COP (Air-Side Method) and Dates of Measurements	Not measured	1.37 Jan. 22–29	1.39 Jan. 20–26
Expected COP, Based on Manufacturer Data	1, less that lost via duct leakage (electric resistance)	3.2, less that lost via duct leakage	4.8

With the ventilation adjustment, the co-heat measured COP for House A was higher than 1.0, which would be the expected value for an electric resistance system with zero duct losses. (Without the ventilation adjustment, it was 1.0.) The co-heat method COP for House B was similar to what would be expected based on the manufacturer’s specifications. The air-side measurement was much lower.

Air-side measurements may be less reliable than measurements based on co-heat because of the uncertain uniformity of supply air measurements. It was noted that the heat pump heating coil in House B was nonuniform in temperature (Figure 28), which may have led to nonuniform supply air. Supply air was measured at a single supply register, which may not have been perfectly representative of the average supply air temperature. Other potential uncertainties include possible temperature differences from one room to another, higher convective airflow due to air handling unit operation than existed during the co-heat tests, and variations in ventilation rates from those estimated. Further, the COP calculations in House A and House B included the effects of duct leakage; therefore, the House A COP should be lower than 1. This is an indication of the magnitude of error inherent in this measurement process. Nevertheless, the COPs calculated by the co-heat method are taken to be close to actual performance.

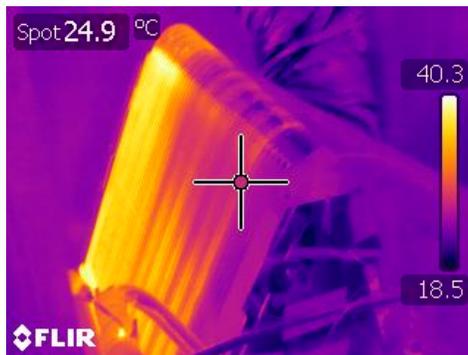


Figure 28. House B refrigerant coil in heating mode showing nonuniform temperatures

The co-heat and air-side COP measurements for House C are well below the expected COP based on manufacturer data. Discrepancies from these expectations are discussed in more detail below.

3.3.9.4 Mini-Split Heat Pump COP

One possible cause of low COP could be low fan speed and therefore low supply airflow. The fan speed was set to auto-speed mode, in which the air handling unit automatically selects the fan speed based on internal logic. During this period, however, the fan operated at low power (5–7 W) and commensurately low speeds (164–190 cfm) all the time in this mode, even when the heat loads were high.

To further investigate this phenomenon, additional testing was conducted with the fan placed in the high-speed setting. At high speed, the fan drew more power (9–16 W), and flows increased to 216–306 cfm. Figure 29 shows the air handling unit fan power for the auto- and high-speed settings. Figure 30 shows the fan curve (approximate flows at a range of power levels) established by taking onetime flow and power measurements.

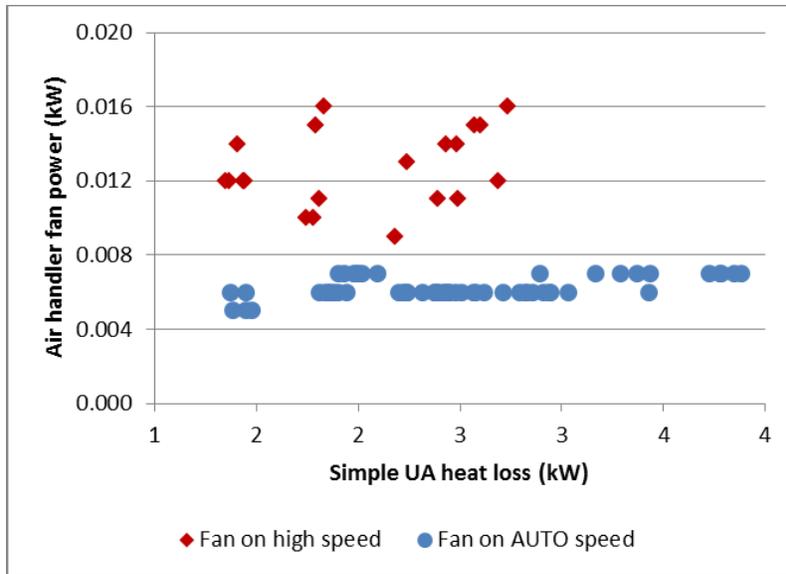


Figure 29. Air handling unit fan power for auto- and high-speed settings

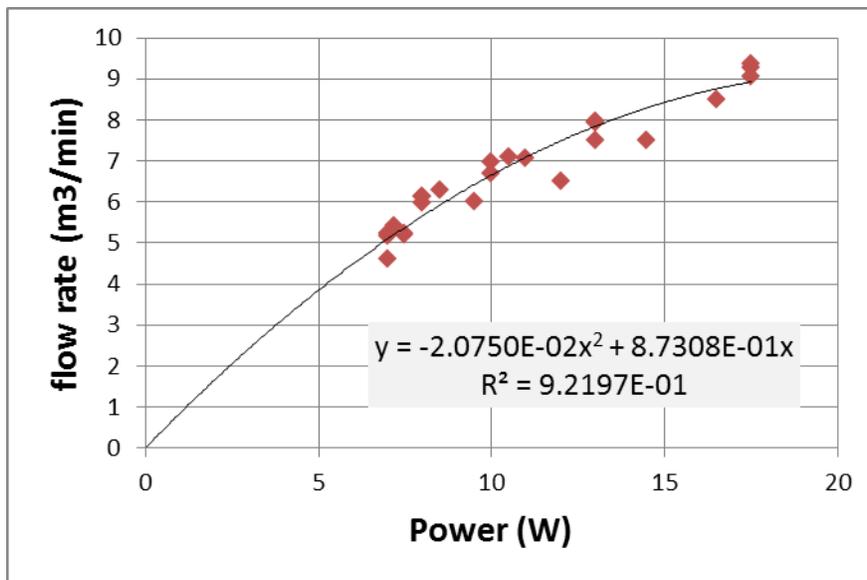


Figure 30. Fan curve based on onetime flow and power measurements

The average COPs calculated from the air-side and co-heat (with ventilation adjustment) methods while the fan was set on high speed (January 27–30) are provided in Table 20 along with a comparison to the auto-speed COPs.

Table 20. Mini-Split Heat Pump COPs at High and Low Fan Speeds

		High Fan	Auto Fan (Low)
Co-Heat Method	COP	4.11	2.49
	Avg. Ambient Temp. (°F)	36.8	30.7
Air-Side Method	COP	2.25	1.39
	Avg. Ambient Temp. (°F)	43.2	42.1

Forcing the fan to high speed increased the COP to an average of 4.11 (3.12 without ventilation adjustment) as measured by the co-heat method. The high fan speed COP as measured by the air-side method also increased but not as dramatically. It is possible that as airflows increased, the supply air temperature became less uniform across the outlet and the single-point air-side measurement did not capture a truly average reading. Note that average ambient temperature was 6.1°F higher during the high-speed period.

Plots of hourly COPs for both measurement methods with outdoor temperature for hours when the fan was on auto compared to high are provided in Figure 31 and Figure 32. The co-heat measurement method has fewer data points because only the hours of 1:00 a.m. to 6:00 a.m. were used.

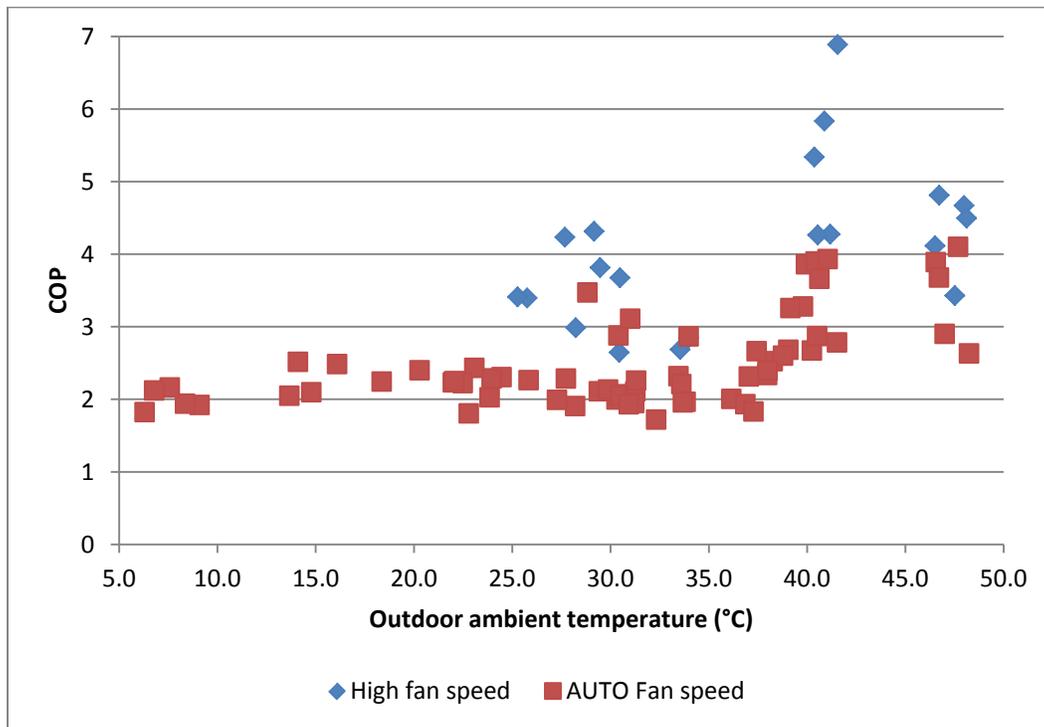


Figure 31. Comparison of mini-split COP (co-heat measurement method) with auto and high fan speeds

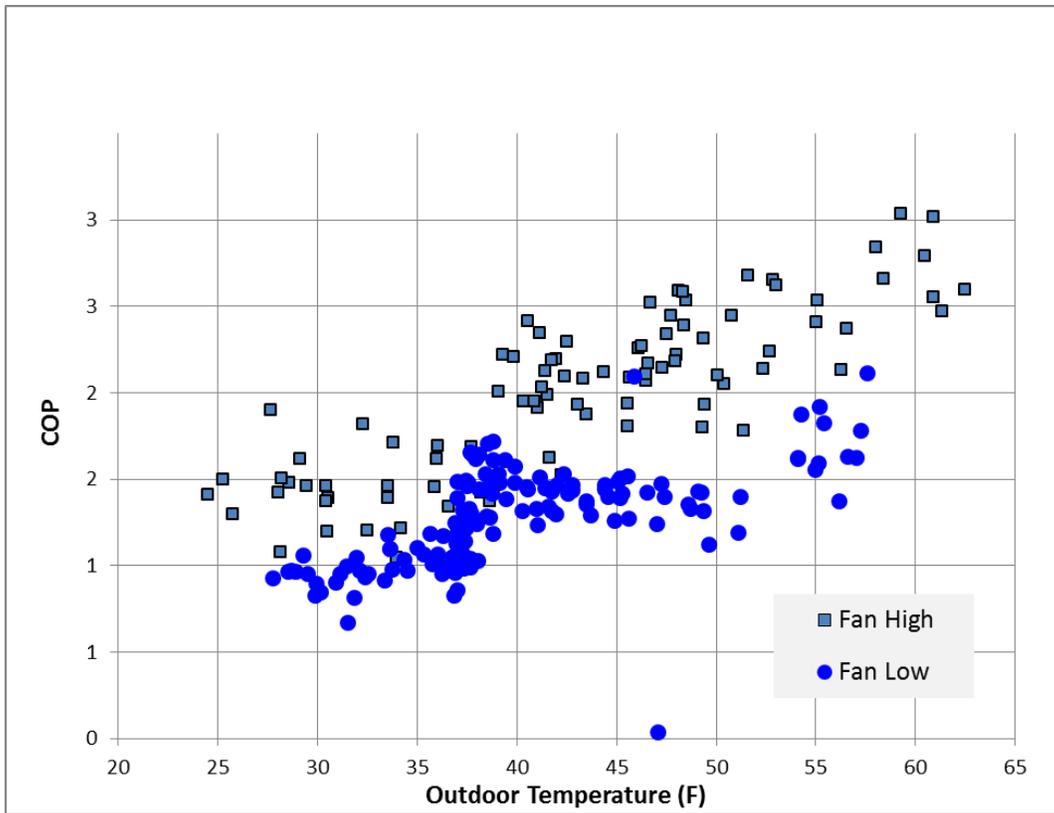


Figure 32. Comparison of mini-split COP (air-side measurement method) with low and high fan speeds

Another possible contributor to low COPs is higher than expected return air temperatures relative to average room temperature. During the period of January 6–13 (the winter sample period), the average temperature in the living space measured in the center of the room at approximately 60 in. above the floor was 70.2°F; the average return air temperature entering the heat pump was 74.8°F. There was clearly stratification of indoor air due to thermal buoyancy: the high-low thermocouples in the living room nearest the heat pump measured an average of 68.9°F at 12 in. above the floor and 75.4°C at 84 in. above the floor during this period.

3.3.10 Humidification Testing

One humidifier was placed in each house in the kitchen and set to operate continuously at its normal high output. The humidifiers were weighed at the beginning and end of the test to determine the average moisture injection rate during the test period. The test began at 8:00 a.m. on July 24, 2014, and ended at 6:30 a.m. on July 25. The average injection rates for each house, the average relative humidity, and ambient temperatures before and during the test are provided in Table 21.

Table 21. Humidification Test Results

House	24 h Before Humidification Test			During Humidification Test	
	Avg. Moisture Injection Rate ^a	Avg. Relative Humidity (%)	Avg. Ambient Temp. (°F)	Avg. Relative Humidity (%)	Avg. Ambient Temp. (°F)
A	195 g/h	38.5		39.4	
B	240 g/h	43.0	75.7	46.5	71.4
C	210 g/h	47.8		54.8	

^a The average moisture injection rate for a three-bedroom home is approximately 200 g/h per the Building America protocol.

In all three houses, there were only minor increases in measured indoor relative humidity. This indicates that the latent capacity of the cooling systems in each house was adequate to remove this rate of moisture injection. The sensible loads on the cooling systems were high because it was hot outside and the cooling systems were operating frequently to meet that load. The conclusion from this test is that this level of moisture removal (200 g/h) is within the capacity of the cooling equipment with the observed sensible heat ratio of approximately 0.75.

3.3.10.1 Impact of Transfer Fans in House C

The impact of transfer fans on the House C bedroom and master bathroom room temperatures was analyzed during both the cooling and heating seasons. Periods when all four fans were activated (i.e., being controlled by their dedicated thermostats) were compared to periods when they were all deactivated (Table 22).

Table 22. Analysis Period for the Impact of Transfer Fans on Room Temperatures in House C

House	Transfer Fans Active Period	Transfer Fans Deactivated Period
Cooling Season Dates	July 21–23, 2015	July 16–8, 2015
Heating Season Dates	Jan. 10–3, 2015	Jan. 18–21, 2015

3.3.10.1.1 Cooling Season

The purpose of the transfer fans during the cooling season was to deliver air from the living room, where the heat pump is located, to the bedrooms and master bathroom. The hypothesis is that the cooler living room air would reduce air temperatures in those rooms and that warmer air would be returned via open doors and transfer grilles back to the living room. Transfer fan flow rates were constant. (See Appendix C for flow rates.) Fans pulled air from the living room approximately 12 in. above the floor and delivered it to the bedrooms/master bathroom at approximately 12 in. from the ceiling. A vertical, sealed metal duct was constructed inside the 2 x 4 partition wall to transfer the air.

During this cooling analysis period, the average air temperature near the floor level of the living room (where the transfer fan inlet was) was 73°F. However, the average temperature at the master bedroom fan outlet was 76.6°F, indicating that the air might have picked up heat from the fan motor or other source. The average central room temperature in the master bedroom was 76.4°F, nearly the same as the air being delivered to the room. This is consistent with the comparison of room temperature data from fan-on and fan-off periods—there is no discernable difference in bedroom/master bathroom temperatures at similar outdoor temperatures. The series of graphs in Appendix C plot room temperatures compared to outdoor temperatures for both conditions. The only trend evident is a slight increase in room temperatures with increasing outdoor temperatures. Note that the interior doors were open during these measurements.



Figure 33. Vertical transfer fan duct

3.3.10.1.2 Heating Season

In House C, backup resistance heat was provided to the bedrooms and master bathroom. Therefore, it was not possible to examine the impact of transfer fans on bedroom temperature, which was always at least 69°F. The purpose of the fans in these conditions would be to reduce the need for resistance heat by bringing warmer living room air into the cooler bedrooms. Because the warmer living room air was heated by the more efficient heat pump, this would be desirable.

However, it was observed that the resistance heaters were on nearly all of the hours when the transfer fans were on, and they were off nearly all of times when the fans were off. Both were driven by outdoor temperature. Therefore, it was not possible to estimate the impact of transfer fans on the energy consumed by the temperature-controlled heaters during this period, but it would seem that the effect is negligible.

Note that the fans were configured for optimal cooling-season operation—i.e., they pulled air from near the floor level in the living room, which would be the coolest air in the room. The high-low temperature sensors showed consistent temperature stratification in the living room during the heating season, with the floor-level sensors reading 5.2°F lower than the ceiling-level sensors and 3.6°F lower than the central living room sensor.



Figure 34. High, low, and primary room temperature sensors

The heat transfer rate from the living room to the master bedroom and master bathroom due to transfer fans was calculated from flow and temperature measurements using the equation below. The results in Table 23 indicate that heat transfer through the fans was low.

$$Q_f = F * cp * (T_{lv} - T_{bd})$$

Where:

Q_f = Heat transferred to the bedroom in Btu per hour via the transfer fan

F = Fan flow rate in cubic feet per minute delivered to the bedroom/master bathroom

T_{bd} = Bedroom/master bathroom temperature (F)

T_{lv} = Living room temperature (F)

cp = Specific heat of air approximated as 1.086

⁶ Converts cubic feet per minute into pounds per hour and multiplies by specific heat of air: 1 cfm * 60 minutes per hour = 60 CFH * .075 pound per cubic foot (standard air) = 4.5 * 0.24 specific heat of air = 1.08

Table 23. Transfer Fan Heat Delivery

Room	Fan Flow (cfm)	Qf (Btu/h)	Fan Outlet (°F)	Room (°F)
Master Bedroom	106	103	70.0	69.0
Master Bathroom	115	89	69.8	69.0

3.3.11 Impact of Window Blinds

3.3.11.1 Cooling Energy

The impact of window blind position on cooling energy was investigated by comparing the cooling energy consumption of periods when the blinds were completely open to similar periods when the blinds were completely closed. Window treatments were white, horizontal venetian blinds (Figure 35). During the closed periods, the blinds covered 100% of the windows, and the blind vanes were angled downward toward the outside. The hypothesis was that closed blinds would reduce solar gain and therefore cooling energy. The periods analyzed are described in Table 24.



Figure 35. Window blinds 50% closed

Table 24. Window Blinds Analysis Period Conditions

House	Blinds Open	Blinds Closed
Dates	Aug. 17–23, 2014	Sept. 23–29, 2014
Avg. Ambient Temp. (°F)	78.6	67.1
Ambient Temp. Range (°F)	74.5–80.6	62.6–71.8
Avg. Indoor Temp. House A (°F)	75.7	74.8
Avg. Indoor Temp. House B (°F)	77.7	76.5
Avg. Indoor Temp. House C (°F)	74.7	73.0
Avg. Direct Normal Solar Radiation (Btu/ft ²)	1013	1102
Direct Solar Radiation Range (Btu/ft ²)	341–1,483	274–1,869

The direct factors affecting the cooling energy consumption during these periods are ΔT ($T_{\text{indoor}} - T_{\text{outdoor}}$) and solar radiation received. For “daily” cooling energy and temperature, we used the time periods just before sunrise on Day 1 to just before sunrise on Day 2 (6 a.m. to 6 a.m.), because the solar energy falling on the house during Day 1 would influence the cooling energy past midnight on Day 1, well into the early-morning hours. As shown in Table 24, the average ambient temperature during the period when the blinds were open is nearly 11.5°F higher than it was during the period when the blinds were closed. To eliminate the effect of cooler temperatures on cooling energy consumption, the data was normalized (see Appendix H). This was done by making a linear fit of the daily cooling energy as a function of ΔT ($T_{\text{indoor}} - T_{\text{outdoor}}$) and the daily totals of direct normal solar radiation during the period when the blinds were closed. This linear fit was used to predict the cooling energy consumption using identical outdoor temperature and direct normal radiation data as that used during the period when the blinds were open.

The sun angles are not very different from August to September, so using a surrogate measure of solar energy such as direct normal radiation is reasonable. A complete cooling-season savings estimate would require modeling sun angles, which was beyond the scope of this analysis; but these data give a strong indication that closing the blinds led to a significant reduction in cooling energy.

The resulting predicted reduction in cooling energy due to closing window blinds in each house is shown in Table 25. Cooling energy reduction was less in House B and House C than it was in House A because those windows have a much lower SHGC than do the windows in House A.

Table 25. Cooling Energy Reduction Due to Window Blinds

	House A	House B	House C
Blinds Open Cooling Energy (kWh)	129.6	91.0	60.4
Blinds Closed Cooling Energy (kWh)^a	81.8	72.4	44.9
Savings (Blinds Closed vs. Blinds Open) (%)	36.9	20.4	25.7

^a Predicted using the best-fit equation derived from the regression of delta T and daily totals of solar radiation as the independent variables and cooling energy as the dependent variable.

3.3.11.2 Room Temperature

The impact of blind position (completely open or completely closed) on room temperatures in the cooling season was also investigated during the same two periods as described above. As expected, closed blinds led to lower room temperatures in House C in all the rooms (by an average of 2.6°F). In House A and House B, closed blinds did not result in significantly lower room temperatures because the space-conditioning distribution system overcame the effects of the blinds. Appendix C plots room temperature compared to ambient temperature under both blind configurations.

3.3.12 Impact of Interior Door Position

The impact of interior door position (open and closed) on room temperatures was investigated during the cooling and heating season. For the majority of the monitoring period, the doors were kept open; however, during selected periods, the doors were closed to compare the effects. These experiments were carried out during the periods shown in Table 26.

Table 26. Interior Door Position Analysis Periods

	Doors Open	Doors Closed
Cooling Season	June 22–28, 2015	July 2–8, 2015
Heating Season	Dec. 23–29, 2014	Dec. 13–19, 2015

3.3.12.1 Cooling Season

Bedroom 2 and Bedroom 3 in House A and House B showed no change to slightly lower temperatures when the doors were closed, indicating that the house had fairly well balanced cooling airflow. The master bedroom in House B showed somewhat elevated temperatures when the door was closed, which is consistent with the lower supply flow per square foot in that room. House A did not show this trend despite the same relatively low supply flow. The master bathrooms in House A and House B both showed lower temperatures when the doors were closed, which is consistent with their high supply flow per square foot and long return-air pathway. In House C, Bedroom 2 and Bedroom 3 showed significant warming when the doors were closed. The master bathroom did not show a difference with closed doors; the room is more distant from the central space, and less air was exchanged with the cooler living room when the doors were open. Likewise, the master bedroom’s larger volume and connection to the master bathroom diluted the effect of the open door on room temperature. Plots comparing the bedroom

temperatures to the ambient temperatures under both door configurations (open and closed) are provided in Appendix C. Table 45 shows the supply flows per square foot of room area.

3.3.12.2 Heating Season

Keeping the doors open resulted in room temperatures closer to the heating set point than they were when the doors were closed. In House A and House B, when the doors were closed at the ambient temperature ranges observed during these periods, the temperature in all rooms except the living room fell below the set point. The outdoor temperature had little effect on the indoor temperatures. This indicates an undersized distribution system for heating in these houses.

In House C, data was collected during periods when the resistance heat in the bedrooms and master bathroom were not active, which enabled an assessment of impact on room temperature. There was a clear trend of decreasing indoor temperatures in House C as outdoor temperatures decreased. The same trend as in cooling was evident: only Bedroom 2 and Bedroom 3 showed a significant change when the doors were closed (in this case, they became cooler). The master bathroom and small bathroom showed a negligible impact when the doors were closed. Plots comparing the bedroom temperatures to the ambient temperatures under both door configurations (open and closed) for heating are provided in Appendix D.

3.3.13 Convective Heat Transfer Through Open Doors

To calculate the convective heat flow through open doors in House C, two test periods were considered while the house was heated by electric resistance heaters and the exhaust fan and heat pump were off: (1) all rooms heated to 69°F and (2) bedrooms and master bathroom heated to 69°F with the living area heated to 71°F. The heating energy consumed by the heaters in the master bedroom, master bathroom, Bedroom 2, and Bedroom 3 were compared to each other during these periods (Table 27). The difference in the amount of energy consumed by the bedroom/master bathroom heaters between these two conditions was approximately equal to the heat flow into the bedrooms from the living room. To eliminate the effect of solar radiation, a 3-h nighttime period from 3 a.m. to 6 a.m. was considered during the course of 4 d as specified in Table 27, for a total of 12 h.

Table 27. Convective Heat Flow Through Open Doors

Temp. Set Point	69°F Living Room 69°F Bedrooms/ Master Bathroom	71°F Living Room 69°F Bedrooms/ Master Bathroom
Analysis Period	Feb. 3–6, 2015	Feb. 14–17, 2015
Avg. Outdoor Temp. (°F)	24.1	24.6
Avg. Crawl Space Temp. (°F)	39.6	33.6
Avg. Attic Temp. (°F)	23.9	27.5
Avg. Wind Speed (m/s)	0.6	0.9

Average ambient temperatures were similar between the two periods; however, there was a difference of 6°F in the crawl-space temperature and a difference of 3.6°F in the attic temperature between the two periods. The difference in attic temperature could be due to a

higher thermostat setting in the living room, which caused greater heat loss to the attic than during the period of February 3–6. The higher wind speed might have been the cause of the lower vented-crawl-space temperature of February 14–17.

The difference in the heat loss to the outside from these rooms during the two cases (71°F living room and 69°F living room) was adjusted based on the known U values of the building components by using the following equation (results are shown in Table 28):

$$Q = U \times A \times \Delta T$$

Where:

Q = heating energy consumed by the heaters in the rooms in Btu

U = U value of the surface area in Btu/h-F

A = area of the surface in square feet

ΔT = temperature difference between the rooms and space where heat transfer takes place.

The heat transfer into the bedrooms and master bathroom for all 12 h considered, when the living room was set to 71°F and 69°F, is shown in Table 29.

Table 28. Difference in Heat Loss to the Outside Over 12 Hours Between Living Room Set Point at 71°F and Living Room Set Point at 69°F

	Heat Loss Through Attic (Btu)	Heat Loss Through Floor (Btu)	Heat Loss Through Windows (Btu)	Heat Loss Through Walls (Btu)	Total Difference ^a
Bedrooms 2 and 3	-59	554	113	188	796
Master Bedroom	-103	972	169	247	1,286
Master Bathroom	-54	520	56	168	690

^a Calculated heat loss to the outside during the two test conditions

Table 29. Heat Transfer into Rooms During 12 h: Calculations

Row		BR2	BR3	MBR	Mbath
1	Heater energy: 71° living room; 69°F bedrooms (Btu)	6,620	3,879	10,927	10,625
2	Heater energy: 69°F all rooms (Btu)	12,946	8,605	19,439	11,331
3	Difference between rows 1 and 2 (Btu)	6,326	4,725	8,512	706
4	Adjustment due to difference in heat loss to outside (from Table 28) (Btu)	796	796	1,286	690
5	Heat transfer into rooms (Row 3 minus Row 4) (Btu)	5,548	3,948	7,259	29

Plots of hourly heater energy use in each room as a function of outdoor temperature are provided in Appendix I. Note that there is no direct doorway between the master bathroom and the living room; therefore, heat transfer into this room is low.

Additional calculations were made to separate the conductive heat flow through interior partitions from the convective heat flow through the open door (Table 30). The resulting effective airflow through the doors ranged from 140 to 281 cfm. This cfm translates into an air velocity of only about 20 ft/min (through half the area of an open door). This level of airflow is very difficult to measure and so was not measured directly. Note that there are a number of phenomena that lead to uncertainties with these calculations, including temperature stratification, imperfect mixing of air within rooms, imperfect estimates of component U values, averaging of temperatures, and variations in infiltration to the outside among the two periods.

Table 30. Convective Heat Transfer Through Open Doors: Calculations

Row		Units	Bedroom 2	Bedroom 3	Master Bedroom	Master Bathroom
1	Avg. ΔT ($T_{\text{bedroom}} - T_{\text{living}}$)	F	1.8	1.8	1.8	2.0
2	Avg. heat transfer into rooms per hour	Btu/h-F	256	182	335	1
3	Wall area	ft ²	90	90	92	131
4	Door area	ft ²	16.2	16.2	16.2	0
5	U value of interior wall	Btu/h-ft ² -F	0.34	0.34	0.34	0.34
6	UA interior wall	Btu/h-F	30.8	30.8	31.4	44.4
7	UA door opening (Row 2 less Row 6)	Btu/h-F	225	151	303	-44
8	Convective heat transfer through door	Btu/h	405	272	546	-87
9	Effective airflow through open door	cfm	209	140	281	N/A

3.4 Data Analysis

The objective of this analysis was to extrapolate from the measured data the impact on annual energy consumption of the primary measures installed in the lab houses in a range of climates in the Southeast living simulations.

3.4.1 Climate Locations

Three locations were selected to represent the range of climates within the service territory of the Tennessee Valley Authority. Table 31 lists the locations along with their respective IECC climate zone, HUD climate zone, manufactured home ENERGY STAR climate region, normal

heating degree days, and normal cooling degree hours. Maps illustrating the territory and climate zones of the Tennessee Valley Authority are provided in Figure 36.

Table 31. Climate Locations Selected for Modeling

Location	IECC Climate Zone	HUD Climate Zone	Manufactured Home ENERGY STAR Climate Region	Heating Degree Days (Base 65°F)	Cooling Degree Hours (Base 74°F)
Columbus, Mississippi	3	1	4	2,720	26,043
Knoxville, Tennessee	4	2	3	3,666	14,214
Bowling Green, Kentucky	4	2	2	4,314	14,745

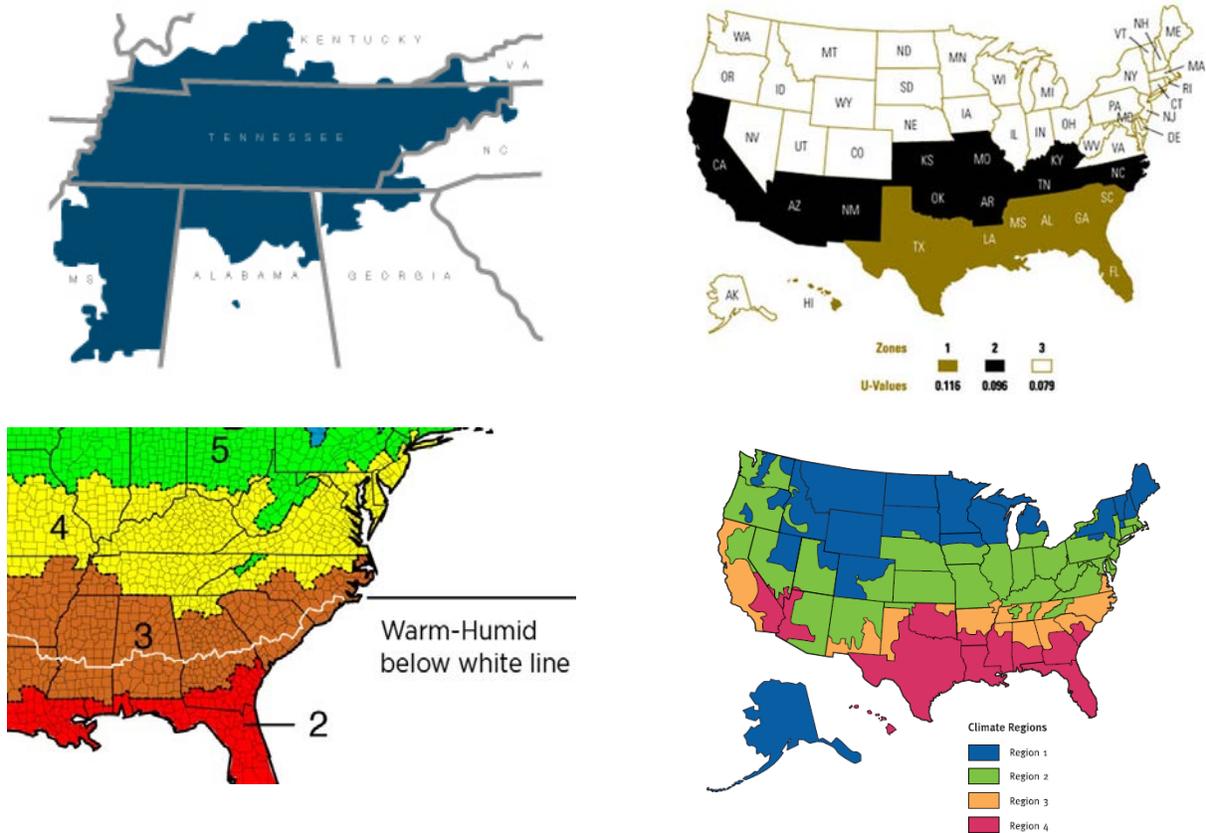


Figure 36. Map of climate locations

3.4.2 Model Development and Calibration

The three houses were modeled using BEopt version 2.3.0.2 with the Energy Plus (version 8-3-0) engine.

Models were calibrated by comparing their output to the measured field data for selected periods of time when the operating conditions were well known (i.e., during periods when resistance heat was used in all three homes). Aspects of the models were then adjusted to better align intermediate outputs (such as attic and crawl space temperatures) to their respective measured conditions and, finally, to compare the modeled to measured space-conditioning energy use of each home.

A measure of calibration accuracy is provided in Table 32. This indicates how accurately the calibrated model predicted overall space-conditioning energy for the selected periods. The periods were chosen when the houses were operating with consistent set points and equipment operation. The resistance heating period was calibrated first because the more precisely known heating energy input for all three houses reduced the number of variables to calibrate. After the resistance heating period was calibrated as closely as possible, the other two periods were calibrated, primarily by tuning the equipment efficiencies.

Table 32. Model Calibration Error^a in Space-Conditioning Energy

House	Calibration Period		
	Resistance Heating Only (March 7–17) (%)	As-Built Heating Systems (Jan. 6–13) (%)	Cooling (Aug. 29–Sept. 15) (%)
A	-0.56	-9	0
B	-2.3	1	-1
C	10.85	0.64 ^a	0

^a The model calibration error equals (modeled - measured)/measured.

^b When the energy consumed by the temperature-controlled heaters is predicted by the model.

The model inputs are provided in Table 34, Table 35, and Table 36, along with the basis for the inputs and an explanation of the calibration adjustment to the inputs, if any.

Table 33. Envelope Inputs for House A, B, and C

Type of Calibration	House A	House B	House C	Basis of Adjustment
Wall	U: 0.077	U: 0.077	U: 0.059	Modeled UA value for attic ceiling, floor, walls, windows, and door adjusted to match the co-heat test results
Attic Ceiling	U: 0.043	U: 0.031	U: 0.022	
Floor	U: 0.074	U: 0.043	U: 0.042	
Window	U: 0.47, SHGC: 0.73	U: 0.31, SHGC: 0.33	U: 0.3, SHGC:0.23	
Door	U: 0.4	U: 0.4	U: 0.4	
Interior Shading (Heating/Cooling Shade Multiplier)	0.75	0.75	0.75	0.75 indicates a 25% reduction in solar gain when the blinds cover 50% of the effective window area
Ratio of Effective Leakage Area (to Outside) in Unfinished Attic to Total Floor Area	0.0033 (BEopt default)	0.0033 (BEopt default)	0.00001	Adjusted House C to increase attic temperature to match measured data
Air Leakage	7.7 ACH50	7.7 ACH50	5 ACH50	Adjusted such that the AIM-II Model in BEopt would predict air leakage that matches the measured air leakage during the tracer gas test period in Dec.

Table 34. Model Inputs—House A

Model Input	Value	Basis and Calibration Adjustment (If Any)
Calibration Period: Jan. 6–13		
Electric Furnace	Capacity: 34.1 kBtu/h (10 kW)	Capacity: Autosize was modified to 10 kW
Mechanical Ventilation	21.56 supply	Based on flow measurement and air handling unit run time
Duct Leakage (To outside)	6.5 cfm ₂₅ /100 ft ² , all ducts insulated to R-12 located in pier and beam (approx. weighted avg. between R-8	Original measurement was 54 cfm ₂₅ to outside or 4.54 cfm ₂₅ /100 ft ² Total duct leakage: 115 cfm ₂₅ Original model: 17% ducts insulated to R-8 located in pier and beam, 83% ducts located in conditioned space

Model Input	Value	Basis and Calibration Adjustment (If Any)
	x-over duct + R-14 in-floor ducts	Changed to increase energy use
Heating Set Point	70.1 °F	Based on average measured indoor temperature during this period
Plug Loads	153 kWh	Sum of energy consumed by two sensible gain heaters and additional not measured internal gain
Calibration Period: Aug. 29–Sept. 15		
Central A/C		
Rated and Installed SEER	13	Rated value; input not used; not considered for model—EER used for calculations
EER (kBtu/kWh)	7.1	Reduced from 11 to account for higher measured cooling use during this period
Rated Air Flow Rate (cfm/t)	$705/2 = 353$	Rated standard airflow for this model is 705 cfm based on specifications from the Air-Conditioning, Heating, and Refrigeration Institute Tonnage = 2
Rated Supply Fan Power (W/cfm)	0.34	400 W/1,160 cfm
Installed Supply Fan Power (W/cfm)	0.68	$417/980 = 0.43$ Adjustment to increase the fan energy in the modeled output to match the measured fan energy
Mechanical Ventilation	21.56 cfm supply	Based on flow measurement and air handling unit run time
Duct Leakage (To outside)	6.5 cfm/100 ft ² All ducts insulated to R-12 located in pier and beam	Original measurement was 54 cfm to outside or 4.54 cfm/100 ft ² Original model: 17% ducts insulated to R-8 located in pier & beam, 83% ducts located in conditioned space Changed to increase energy use
Cooling Set Point	76.28°F	Based on average measured indoor temperature during this period
Plug Loads	277.5 kWh	Sum of energy consumed by two sensible gain heaters and additional not measured internal gain

Table 35. Model Inputs—House B

Model Input	Value	Basis and Calibration Adjustment (If Any)
Calibration Period: Jan. 6–13		
Air Source Heat Pump		
Rated HSPF (Btu/ W-h)	8	N/A
COP	3.0	COP = 3.2 based on Jon Winkler’s recommendation Measured COP is 1.6. Value adjusted to increase the heating energy consumption
Rated Air Flow Rate (cfm/t)	1160/1.68 = 690.47 heating	Rated standard airflow for the cooling coil = 580 cfm based on specifications from the Air-Conditioning, Heating, and Refrigeration Institute Air handling unit airflow per specification = 1,160 cfm
Rated Supply Fan Power (W/cfm)	0.34	400 W/1,160 cfm
Installed Supply Fan Power (W/cfm)	0.8	460/1,000 = 0.46 Adjustment to increase the fan energy in the modeled output to match the measured fan energy
Minimum Temp. Below Which Compressor Turns Off	29 °F	Reduced from 40°F to increase the efficiency of the heating system to ensure air source heat pump works for a longer period and matches measured data
Mechanical Ventilation	12.8 cfm supply	Based on flow measurement and air handling unit run time
Duct Leakage (To outside)	0.8 cfm/100 ft ² All ducts insulated to R-12 located in pier and beam	Original measurement was 9.5 cfm to outside, or 0.80 cfm/100 ft ² Original model: 17% ducts insulated to R-8 located in pier and beam, 83% ducts located in conditioned space
Heating Set Point	69.9 °F	Based on average measured indoor temperature during this period
Plug Loads	122.6 kWh	Sum of energy consumed by two sensible gain heaters and additional not measured internal gain
Calibration Period: Aug. 29–Sept. 15		
Air Source Heat Pump		

Model Input	Value	Basis and Calibration Adjustment (If Any)
Rated and Installed SEER	13	Rated value; input not used; not considered for model—EER used for calculations
EER (kBtu/kWh)	6.5	Reduced from 11 to account for lower efficiency measured during this period
Rated Air Flow Rate (cfm/t)	1,160/1.5 = 773 cooling	Rated standard airflow for the cooling coil = 580 cfm based on specifications from the Air-Conditioning, Heating, and Refrigeration Institute (did not seem to affect results) Air handling unit airflow per specification = 1,160 cfm
Rated Supply Fan Power (W/cfm)	0.34	400 W/1,160 cfm (did not affect results)
Installed Supply Fan Power (W/cfm)	0.80	460/1,000 = 0.46 Adjustment to increase the fan energy in the modeled output to match the measured fan energy
Mechanical Ventilation	12.8 cfm supply	Based on flow measurement and air handling unit run time
Duct Leakage (To outside)	0.8 cfm ₂₅ /100 ft ² All ducts insulated to R-12 located in pier and beam	Original measurement was 9.5 cfm ₂₅ to outside, or 0.80 cfm ₂₅ /100 ft ² Total duct leakage: 36 cfm ₂₅ Original model: 17% ducts insulated to R-8 located in pier and beam, 83% ducts located in conditioned space
Cooling Set Point	77.36°F	Based on average measured indoor temperature during this period
Plug Loads	277.3 kWh	Sum of energy consumed by two sensible gain heaters and additional not measured internal gain

Table 36. Model Inputs—House C

Model Input	Value	Basis and Calibration Adjustment (If Any)
Calibration Period: Jan. 6–13		
Mini-Split Heat Pump		
Rated HSPF (Btu/ W-h)	4.2	Reduced from 12 HSPF to account for measured HVAC energy use and to account for the temperature controlled heaters in the bedrooms

Model Input	Value	Basis and Calibration Adjustment (If Any)
Heating Capacity Offset (Btu/h)	3,000	Nominal heating capacity - nominal heating capacity = 18,000 - 15,000
Minimum Heating Capacity	0.29	Min. capacity/nominal rated capacity = 5,150/18,000
Maximum Heating Capacity	1.33	Max capacity/nominal rated capacity = 24,000/18,000
Min Heating Airflow (cfm/t)	134	Min. heating airflow: 201cfm, t: 1.5
Max Heating Airflow	331.33	Min heating airflow: 497cfm, t:1.5
Supply Fan Power (W/cfm)	0.18	Full load amps: 0.67, cfm: 410 (from the Air-Conditioning, Heating, and Refrigeration Institute)
Minimum Temp.	-13°F	Specifications indicate 62% heating capacity at -13°F
Heating Set Point	70.4°F	Based on average measured indoor temperature during this period
Plug Loads	157.5 kWh	Sum of energy consumed by two sensible gain heaters, 4 transfer fans, and additional not measured internal gain. Energy consumed by temperature controlled heaters taken out
Calibration Period: Aug. 29–Sept. 15		
Mechanical Ventilation	45 cfm exhaust	
Mini-Split Heat Pump		
Rated SEER (Btu/ W-h)	15.2	Reduced from 22 SEER to account for measured cooling energy
Minimum Cooling Capacity	0.436	Min. capacity/nominal rated capacity = 6,450/15,000
Maximum Cooling Capacity	1.27	Max. capacity/nominal rated capacity = 19,000/15,000
Min Cooling Airflow (cfm/t)	180	Min. cooling airflow: 225 cfm, t: 1.25

Model Input	Value	Basis and Calibration Adjustment (If Any)
Max Cooling Airflow	328.8	Min. cooling airflow: 411 cfm, t: 1.25
Supply Fan Power (W/cfm)	0.18	Full load amps: 0.67, cfm: 410 (from the Air-Conditioning, Heating, and Refrigeration Institute)
Minimum Temp.	-13°F	Specifications indicate 62% heating capacity at -13°F
Heating Set Point	74.12°F	Based on average measured indoor temperature during this period
Plug Loads	301.114 kWh	Sum of energy consumed by two sensible gain heaters, 4 transfer fans, and additional not measured internal gain

3.4.3 Model Runs

The calibrated models were then run for each of the three locations. Five models were prepared. Three of the models were created from directly measured data as described above. Two of the models were not built as test homes, so they were simulated by changing only the heating system type (electric resistance to heat pump or vice versa). Because the heating system efficiencies were well known from the measured data, this was done with a high degree of confidence. Note that the modeled home envelopes were not altered for code and program compliance in the different climate locations; rather, they were retained as built. A summary of the five models is provided in Table 37.

Table 37. Summary of Models

Model	Thermal Envelope	Space Conditioning	Data Source
A1	HUD code	Electric resistance furnace; split system AC ^a	Measured
A2	HUD code	Heat pump furnace; split system AC	Simulated
B1	Manufactured home ENERGY STAR	Electric resistance furnace; split system AC	Simulated
B2	Manufactured home ENERGY STAR	Heat pump furnace; split system AC	Measured
C	ZERH (IECC 2012)	Ductless mini-split heat pump	Measured

^a Air conditioner

The calibrated models had sensible gain heaters installed to account for the internal gains due to people, lights, and appliances. To estimate these internal gains in an occupied and functional home, some assumptions were made, as given in Table 38.

House C had transfer fans installed in the three bedrooms and the master bathroom; these turned on when the temperatures in those rooms fell below 69°F in heating or rose above 76°F in cooling. When analyzing the behavior of the transfer fans, it was found that they were on nearly 44% of the hours during the cooling season and 90% of the hours during the heating season. This energy was translated into BEopt as miscellaneous plug loads to include in the model the energy use and internal gains due to these fans.

Table 38. Assumptions for Model Runs

	Model A and A1	Model B and B1	Model C
Set Point	Heating: 68°F Cooling: 78°F	Heating: 68°F Cooling: 78°F	Heating: 68°F Cooling: 78°F
Bathroom Exhaust Rate (Default per Building America House Simulation Protocols)	32 cfm avg. 60 min/d (intermittent)	26 cfm avg. 60 min/d (intermittent)	50 cfm avg. 60 min/d (intermittent)
Bath Fan Power	0.3 W/cfm/fan	0.3 W/cfm/fan	0.3 W/cfm/fan
Kitchen Range Hood Exhaust (Changes in Code)	80 cfm	77 cfm	110 cfm 60 min/d (intermittent) (per HSP)
Water Heater	Electric benchmark, 50 gal, 0.90 EF	Electric standard, 50 gal, 0.92 EF	Electric premium, 50 gal, 0.95 EF
Lighting	100% incandescent	100% incandescent	80% compact fluorescent lighting
Refrigerator	Benchmark = 434 kWh/yr	0.8 X benchmark = 347 kWh/yr	0.8 X benchmark = 347 kWh/yr
Cooking Range	Benchmark (499 kWh/yr, electric)	0.8 X benchmark (399 kWh/yr, electric)	0.8 X benchmark (399 kWh/yr, electric)
Dishwasher	Benchmark (318 kWh/yr)	0.8 X benchmark (290 kWh/yr)	0.8 X benchmark (290 kWh/yr)
Clothes Washer	Standard (MEF ^a = 1.41)	ENERGY STAR (MEF = 2.47)	ENERGY STAR (MEF = 2.47)
Clothes Dryer	Electric (EF = 3.1)	Electric (EF = 3.1)	Electric (EF = 3.1)
Additional Plug Loads	None	None	Transfer fan energy as described above
Appliance Operating Schedules	Per the Building America House Simulation Protocols (National Renewable Energy Laboratory 2010)		

^a Modified Energy Factor

3.4.4 Modeling Results

Figure 37 through Figure 42 show the results of comparing the five models for each of the three climates. The results are presented in terms of annual site energy consumption in MMBtu/Yr and annual space-conditioning energy in MMBtu/yr (also site energy). All three climates showed similar relationships: Model A1 used the most space-conditioning and ventilation energy, followed by Model B1. The conventional heat pump models (A2 and then B2) were next in consumption. Model C had the least space-conditioning/ventilation consumption—generally less than half that of A1 and 20% to 35% less than the next most efficient model, B2. Because the mini-split heat pump in Model C is much more efficient than the conventional air-conditioning systems, the reduction in annual energy use was highest in the warmest climate (Columbus, Mississippi) and lowest in the coolest climate (Bowling Green, Kentucky).

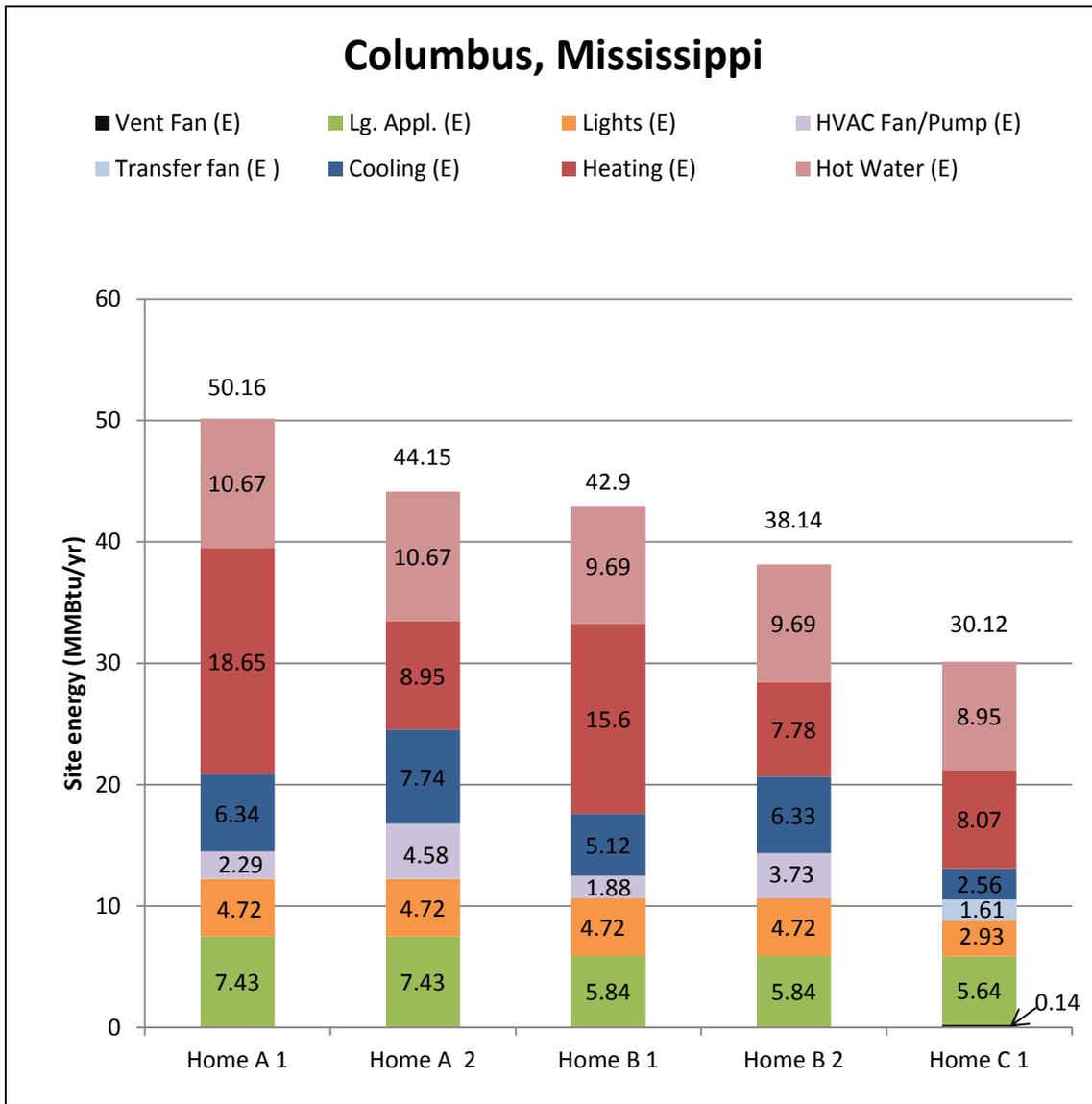


Figure 37. Annual energy consumption comparison—Columbus, Mississippi

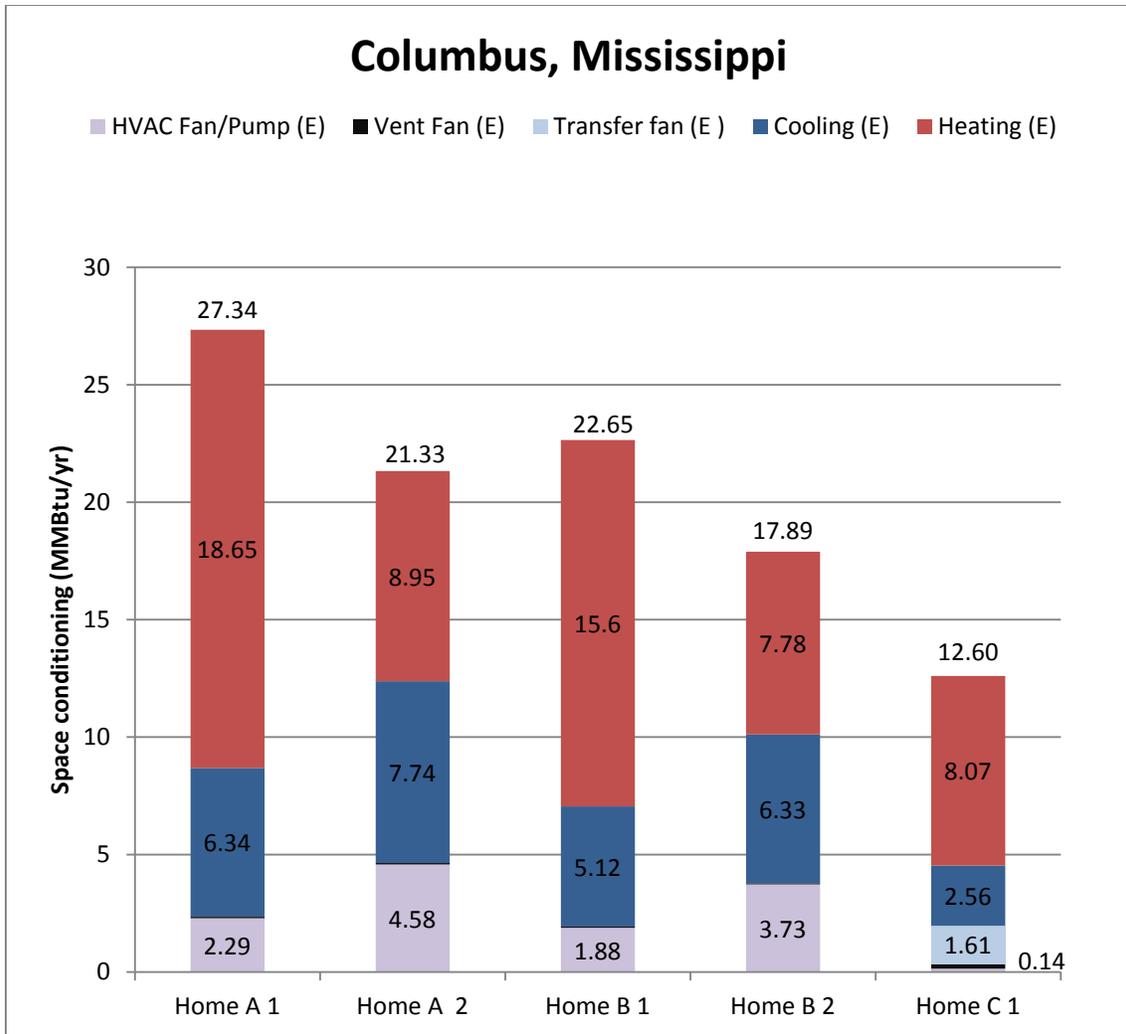


Figure 38. Annual space-conditioning comparison—Columbus, Mississippi

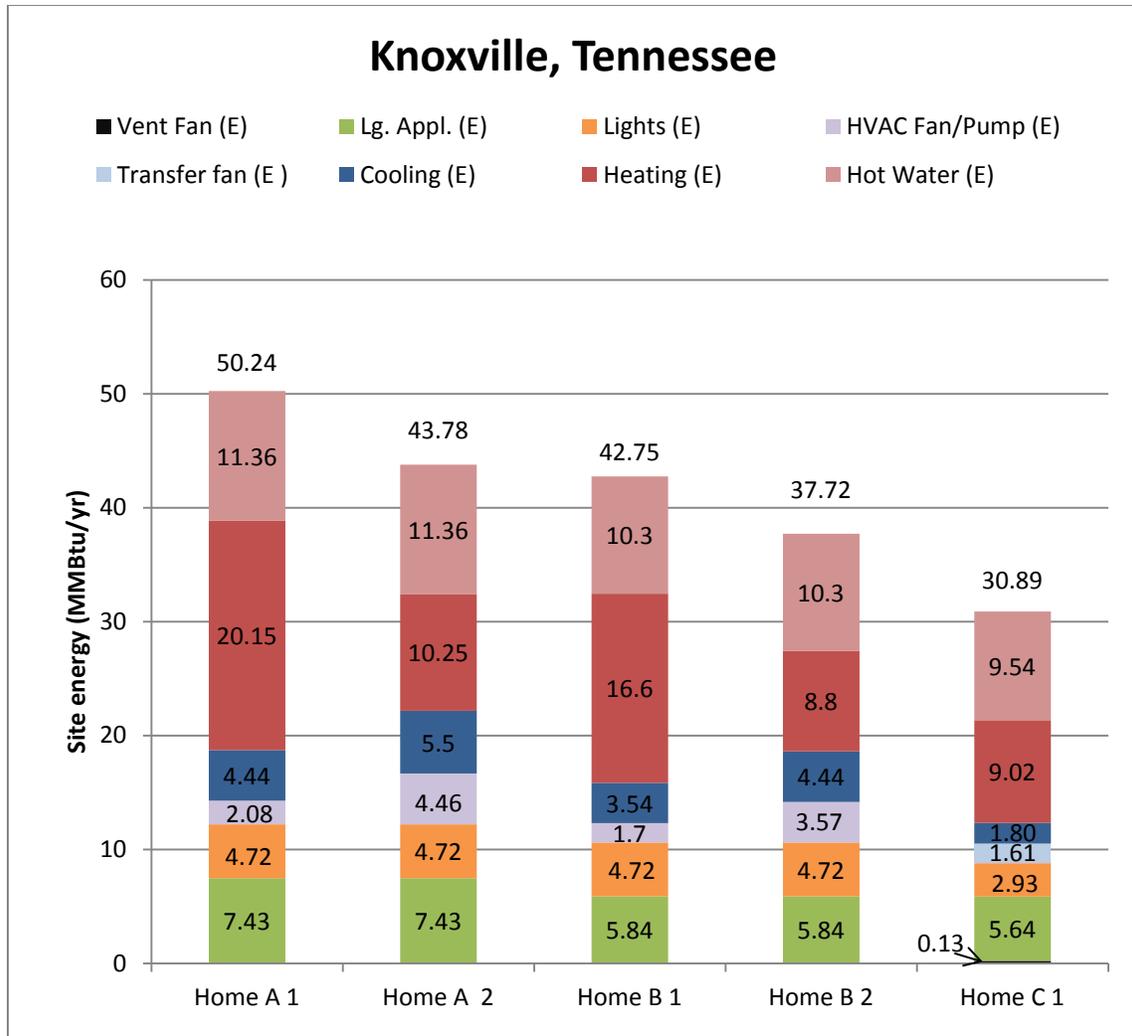


Figure 39. Annual energy consumption comparison—Knoxville, Tennessee

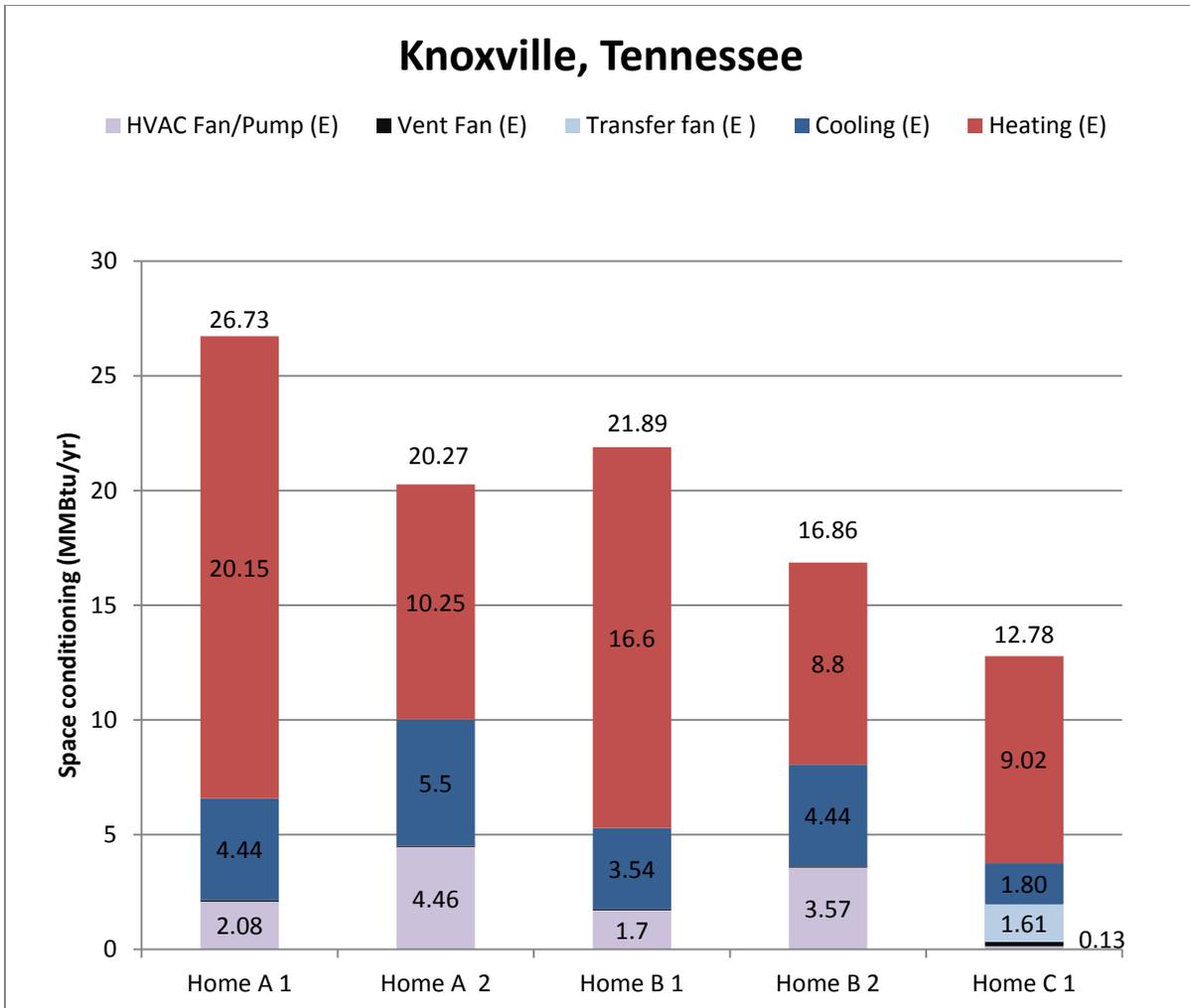


Figure 40. Annual space-conditioning comparison—Knoxville, Tennessee

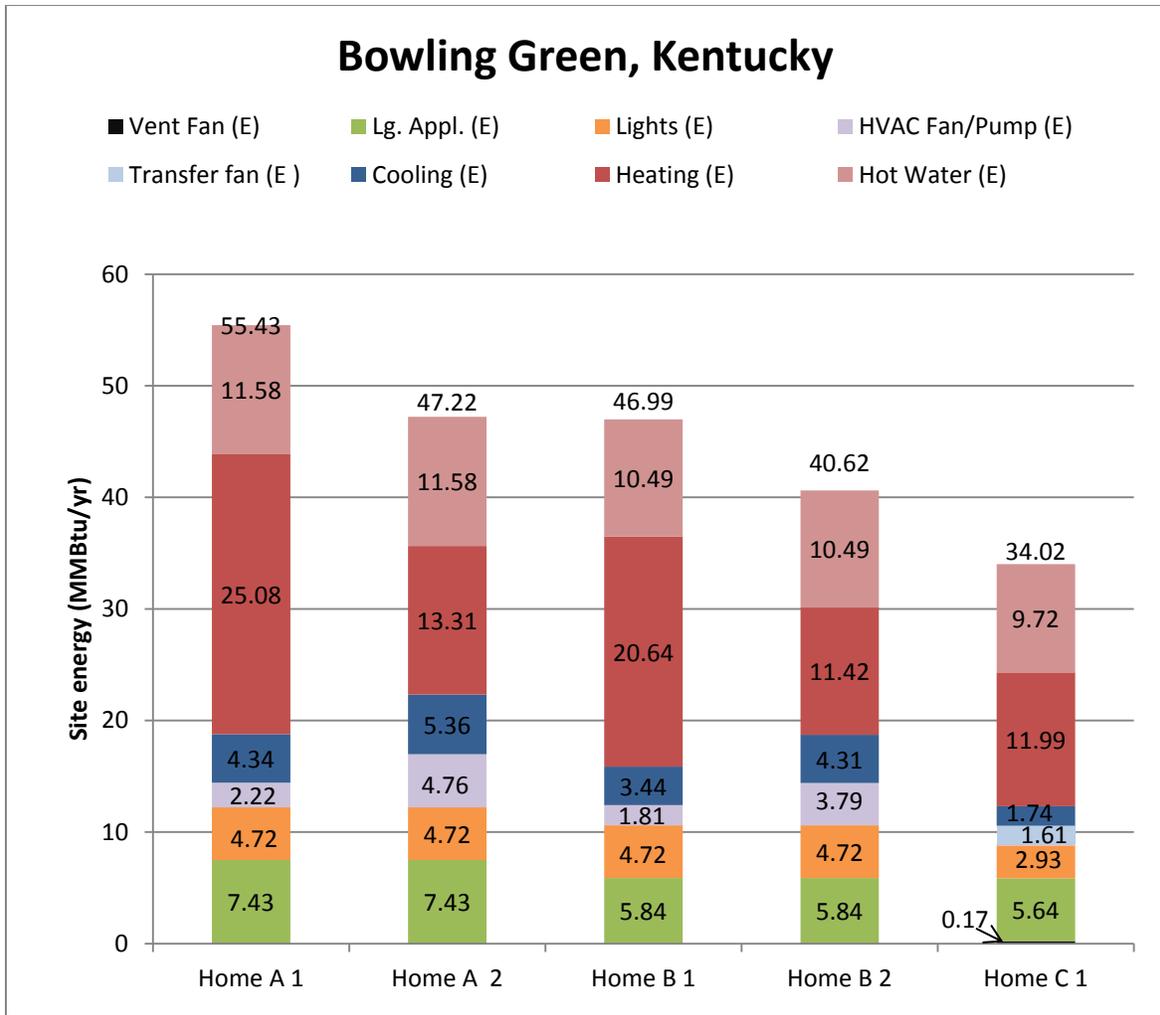


Figure 41. Annual energy consumption comparison—Bowling Green, Kentucky

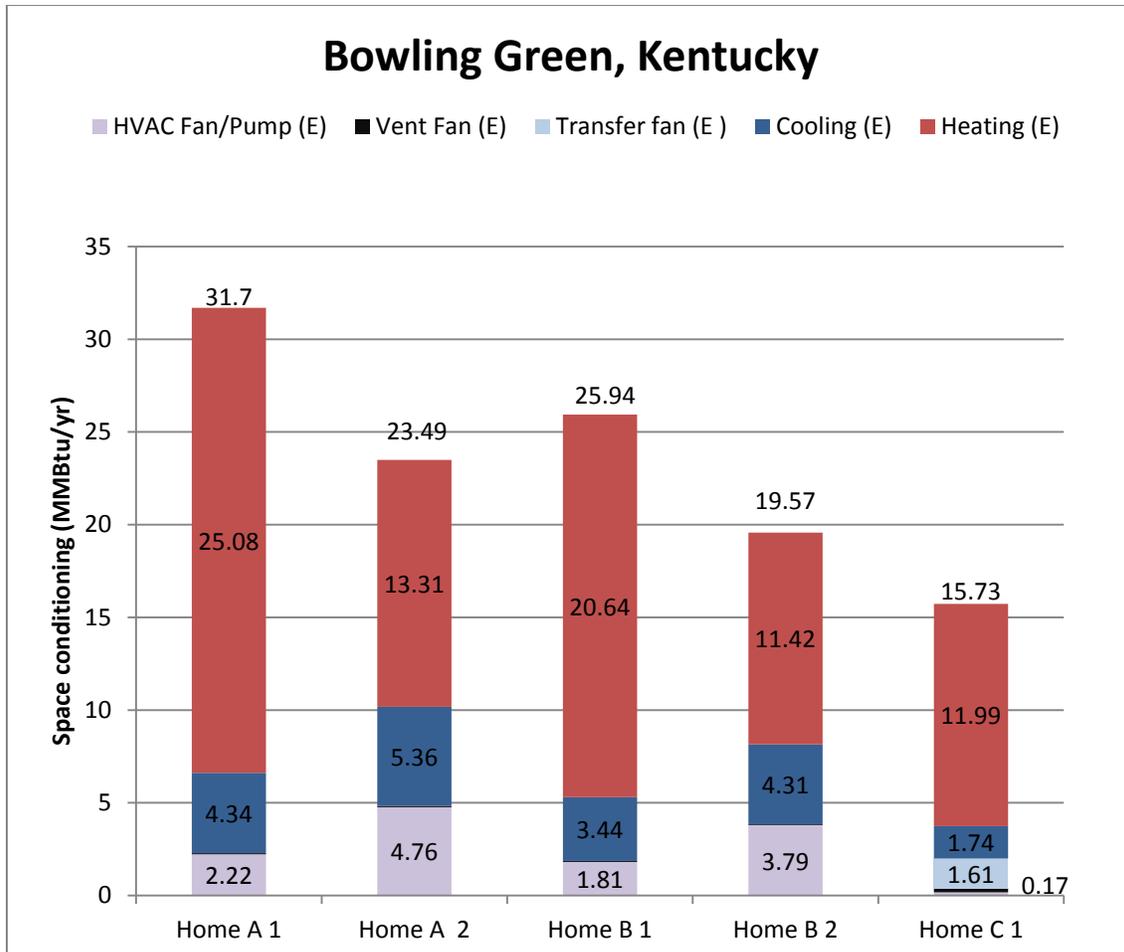


Figure 42. Annual space-conditioning comparison—Bowling Green, Kentucky

3.4.5 Cost Analysis

Table 39 shows the itemized costs for each component that differs among the three houses. The total incremental cost to build House C compared to House A was approximately \$2,500, including non-energy-related measures needed to comply with ZERH criteria. This translates to an approximate \$6,600 incremental price to the retail homebuyer.

Table 39. Incremental Costs

Item	House A	\$ Home-owner	House B	\$ Home-owner	House C	\$ Home-owner	Estimated Incremental Cost A to B	Estimated Incremental Cost B to C	Estimated Incremental Cost A to C
Cooling Equipment	AC installed, 2 t	\$2,498	HP installed, 1.5 t	\$3,015	Mini-split heat pump	\$3,300	\$518	\$285	\$803
					Mini-split heat pump installation	\$0	N/A	\$0	\$0
Heating Equipment	Furnace installed + thermo	\$333	Furnace installed + thermo	\$333	Included in cooling	\$0	\$0	(\$333)	(\$333)
		\$0		\$0			\$0	\$0	\$0
Distribution	Interior ducts, material	\$232	Interior ducts, material	\$232	Transfer fans with fan center and duct material	\$1,454	\$0	\$1,222	\$1,222
	Interior ducts, plant labor	\$0	Interior ducts, plant labor	\$0	Transfer fan thermostat	\$176	\$0	\$176	\$176
	Crossover duct, material	\$92	Crossover duct, material	\$92	Transfer fans, fan thermostat, wiring and duct installation	\$0	\$0	(\$92)	(\$92)
	Crossover duct, site labor	\$38	Crossover duct, site labor	\$38	N/A	\$0	\$0	(\$38)	(\$38)
	R-14 blown fiberglass	\$404	R-28 blown fiberglass	\$579	R-45 dense-pack eaves, blown	\$1,089	\$175	\$510	\$685

Item	House A	\$ Home-owner	House B	\$ Home-owner	House C	\$ Home-owner	Estimated Incremental Cost A to B	Estimated Incremental Cost B to C	Estimated Incremental Cost A to C
					fiberglass				
Insulation—Walls	R-11 batt fiberglass		R-13 batt fiberglass	\$670	Foam + FG batt R-13	\$1,606	\$670	\$936	\$1,606
Insulation—Floor	R-14 blanket fiberglass	\$687	R-28 blanket fiberglass	\$1,374	R-28 blanket fiberglass	\$1,374	\$687	\$0	\$687
Air-Tightening Measures	N/A	\$0	N/A	\$0	Prescriptive measures	\$0	\$0	\$0	\$0
						\$53	\$0	\$53	\$53
Windows	Clayton single plus storm	\$1,382	Clayton vinyl, low-emissivity air filled	\$1,600	Clayton vinyl, low-emissivity, argon filled	\$2,200	\$218	\$600	\$818
Doors	Standard	\$769	Standard	\$769	One door, low U-value	\$507	\$0	(\$263)	(\$263)
Roof Ventilation	N/A	\$0	N/A	\$0	AccuVent baffles	\$169.88	\$0	\$170	\$170
Interior Framing	2 x 3 framing	\$0	2 x 3 framing	\$0	Up-charge for 2 x 4 framing for transfer fan walls	\$220	\$0	\$220	\$220
Range Hood	Standard	\$56	Standard	\$56	BROAN 30-in range hood 403001 white	\$73.39	\$0	\$18	\$18
Water Heater	Standard	\$0	Standard	\$0	Standard		\$0	\$0	\$0

Item	House A	\$ Home-owner	House B	\$ Home-owner	House C	\$ Home-owner	Estimated Incremental Cost A to B	Estimated Incremental Cost B to C	Estimated Incremental Cost A to C
Carbon Monoxide Alarms	N/A	\$0	N/A	\$0	Carbon monoxide alarms	\$176	\$0	\$176	\$176
Ventilation Fans	Ventline	\$46	Ventline	\$46	Air King x 2	\$184.80	\$0	\$138	\$138
Gutters	N/A	\$0	N/A	\$0	Gutters	\$450	\$0	\$450	\$450
Refrigerator	Standard	\$775	Standard	\$775	ENERGY STAR model	\$774.97	\$0	\$0	\$0
Dishwasher	Standard	\$378	Standard	\$378	ENERGY STAR model	\$378.42	\$0	\$0	\$0
Lights	N/A	\$0	N/A	\$0	Compact fluorescent lighting in 80%	\$110	\$0	\$110	\$110

Because House C was designed to comply with ZERH criteria, it included some items that were not strictly related to energy efficiency, such as gutters, ventilation fans, and carbon monoxide alarms. Not counting these items, the premiums to build House C compared to House A and House B were \$5,843 and \$3,575, respectively (price to homebuyer). The modeled energy savings of House C compared to House A and House B ranged from 5.29 to 15.97 MMBtu/yr depending on climate location.

Table 40 through Table 42 compare the annual utility bill savings in dollars at the average statewide Tennessee electricity rate of 10.59¢/kWh (U.S. Energy Information Administration 2015) with the costs for energy measures only.

As shown in the modeling results and in Table 40, the energy savings from House C compared to House B was the highest in the warmest climate (Columbus), which resulted in a payback period of 14.4 yr. House B and House C showed payback periods that ranged from 4.9 to 9.7 yr across the climates compared to House A. In Bowling Green, House C achieved the largest savings—\$664/yr—compared to House A. House B, the ENERGY STAR home, had the shortest payback period in all climates compared to House A.

Table 40. Energy Savings and Payback—Columbus, Mississippi

House	Annual Utility Cost	Savings Compared to House A	Incr. Retail Cost Compared to House A	Payback Compared to House A (yr)	Savings Compared to House B	Incr. Retail Cost Compared to House B	Payback Compared to House B (yr)
A	\$1,653	N/A	N/A	N/A	N/A	N/A	N/A
B	\$1,280	\$373	\$2,268	6.1	N/A	N/A	N/A
C	\$1,031	\$622	\$5,843	9.4	\$249	\$3,575	14.4

Table 41. Energy Savings and Payback—Knoxville, Tennessee

House	Annual Utility Cost	Savings Compared to House A	Incr. Retail Cost Compared to House A	Payback Compared to House A (yr)	Savings Compared to House B	Incr. Retail Cost Compared to House B	Payback Compared to House B (yr)
A	\$1,656	N/A	N/A	N/A	N/A	N/A	N/A
B	\$1,263	\$393	\$2,268	5.8	N/A	N/A	N/A
C	\$1,055	\$601	\$5,843	9.7	\$208	\$3,575	17.2

Table 42. Energy Savings and Payback—Bowling Green, Kentucky

House	Annual Utility Cost	Savings Compared to House A	Incr. Retail Cost Compared to House A	Payback Compared to House A (yr)	Savings Compared to House B	Incr. Retail Cost Compared to House B	Payback Compared to House B (yr)
A	\$1,816	N/A	N/A	N/A	N/A	N/A	N/A
B	\$1,356	\$460	\$2,268	4.9	N/A	N/A	N/A
C	\$1,152	\$664	\$5,843	8.8	\$204	\$3,575	17.5

4 Conclusions

Following are the research questions and responses:

1. *Can factory-built, manufactured homes meet the ZERH requirements as defined? If not, what changes are suggested in the ZERH specifications to better reflect and take advantage of the unique aspects of factory building?*

House C was built in compliance with the HUD code and DOE ZERH criteria. The use of a ductless heat pump simplified the compliance with ENERGY STAR version 3 HVAC requirements. Thermal envelope, ventilation, and indoor air quality requirements were not a barrier, although they did add costs.

2. *What are the added costs and energy savings for manufactured ZERHs using ductless mini-split heat pumps compared to the HUD code and ENERGY STAR manufactured homes?*

HVAC energy savings in House C achieved a 50% reduction compared to the HUD-code home (House A) despite the need for supplemental resistance heat. Room for improvement exists in the form of better air distribution during the heating season, reducing the need for supplemental resistance heating, and improved ductless mini-split COPs, achieved perhaps by increasing airflow rates.. Equipment improvements had a larger impact than envelope improvements. The premium (counting only the energy-efficiency measures) to build House C compared to House A and House B was \$2,060 and \$1,166, respectively, at manufacturer cost. The energy savings of House C compared to House A and House B ranged from 5 to 16 MMBtu/yr depending on climate location. Paybacks to the homeowner for House C compared to House A and House B ranged from 8.8 to approximately 17.5 yr based on retail costs.

3. *How does point-source space conditioning perform in a ZERH in terms of comfort, and how do operational variables such as opening and closing interior doors and window blinds and the operation of transfer fans affect that performance in heating and cooling modes?*
 - A. **Thermal comfort.** The point-source heat pump in House C performed reasonably well in cooling mode. House C showed the widest temperature fluctuation from one room to another, but only the master bathroom exceeded the upper bounds of the Air Conditioning Contractors of America temperature range (with the interior doors open). In heating mode, the bedrooms did not maintain acceptable temperature in House C; therefore, resistance heaters with dedicated thermostats set to 69°F were added to those rooms. This increased energy consumption when the ambient temperature was below freezing.
 - B. **Relative humidity.** Relative humidity was within acceptable limits in all houses all year. Latent loads were not simulated; however, short-term humidification testing in summer revealed little impact on indoor relative humidity despite the significant moisture loads being introduced into the buildings, which indicates that the cooling equipment had sufficient excess capacity to handle the latent loads during hot weather.

- C. **Transfer fan operation impact on room temperatures.** A comparison of fan-on to fan-off periods in House C did not show a difference in bedroom/master bathroom temperatures at similar outdoor temperatures during both seasons. The interior doors were open during these measurements so the heat transfer through the open doors may have dwarfed that through the fans.
- D. **Blind position impact on cooling energy.** The data showed a strong indication that closing blinds reduced cooling energy needs. Cooling energy was reduced by 20.4% and 25.7% in House B and House C, respectively. House A cooling energy was reduced by 36.9%. The higher savings in House A is theorized to be due to the higher solar heat gain coefficient in the House A windows and a proportionally greater impact on solar gain of the blinds in those circumstances.
- E. **Blind position impact on room temperatures in cooling.** Closing blinds during the cooling season led to lower room temperatures in House C in all the rooms. In House A and House B, closing the blinds did not result in significantly lower room temperatures, presumably because the space-conditioning distribution system overcame the effects of the blinds on room temperature.
- F. **Impact of door position on room temperatures.** In House C during the heating season, keeping the doors open resulted in a lower difference in room temperature between the main space and the bedrooms. Bedrooms were 0.4–1.8°F lower than the heating set point with the doors open and 4.0–5.4°F lower with the doors closed. During cooling season, there was no clear impact on room temperatures of closing interior doors.
- G. **Convective heat transfer through open doors.** Heat transfer through open doors with a two-degree F temperature difference between the main living space and bedrooms was estimated by measuring the run time of the electric resistance heater and adjusting it based on partition U-values. Effective airflow rates of approximately 140 to 281 cfm were calculated.

4. *What is the in-situ COP in heating of a ductless mini-split heat pump?*

The heating coefficient of performance was calculated for both heat pumps using two independent methods: (1) airflow and temperature measurements and (2) co-heat testing measurements. The co-heat testing method was deemed more reliable. Using this method, the COP of both the conventional split-system heat pump and the ductless mini-split were approximately 2.5. This was well below the expectation for the mini-split based on manufacturer data. However, when the mini-split air handling unit fan was forced to run on its high-speed setting, its COP increased to 4.11, indicating that low airflow could have been a cause of the low heating efficiency of this unit.

5. *Other conclusions*

- A. **Moisture measurements.** Wood moisture content was slightly higher in House C compared to House B, but it was well within safe limits. Higher moisture content was due to the addition of exterior foam insulation to the walls in House C that reduced the vapor permeability of the foam sheathing (extruded polystyrene

insulation). However, it also resulted in a 5.5°F higher minimum condensation surface temperature than in House B, mitigating condensation risk.

- B. **Peak loads.** Peak demand in House C was significantly lower than it was in the other two houses during all months of the year. House B had somewhat lower peaks than did House A during most months, but some winter peaks were similar, indicating that House B's peak occurred when the heat pump was not running and the house was relying solely on electric resistance backup. On average, House B had an 18% lower peak demand than did House A, and House C had a 69% lower peak demand than did House A during the utility's peak hours.

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Appendix A: Commissioning Test Results

Table 43. Duct Leakage

Duct Leakage (cfm)		
House	Total	To Outside
A	115 cfm	54 cfm
	Fan pressure: -50.3 Pa	Fan pressure: -8.9 Pa
B	36 cfm	8-11 cfm (low flow)
	Fan pressure: -31.9 Pa	Fan Pressure: -1.2 Pa
C	N/A	N/A

Table 44. Total System Flow

Pressures Measured at Kitchen Register	House A	House B
Normal System Operating Pressure	46 pa	42.5 pa
Operating Pressure with Duct Blaster Connected	34 pa	29 pa
Matched System Pressure	45.5 pa	42 pa
Return Flow at Matching System Pressure	980 cfm	1,000 cfm
Fan Pressure		-80.2

Table 45. Supply Register Flows

Register	House A (cfm) (July 23, 2014)	House B (cfm)	Room Area (ft ²)	House A Supply (cfm/ft ²)	House B Supply (cfm/ft ²)
Northwest Bedroom	136	120	142	1.0	0.8
Bathroom 2	LO	76	43	LO	1.8
Kitchen	152	130	159	1.0	0.8
Dining Room	140	133	132	1.1	1.0
Master Bathroom	151	156	131	1.2	1.2
Master Bedroom	126	123	234	0.5	0.5
Living Room	116	118	232	1.0	1.0
Entry	124	104			
Northeast Bedroom	109	93	137	0.8	0.7
TOTAL	1,054	1,053	1,210	0.87	0.87

Table 46. Ventilation Flows

House	Whole-House Ventilation Flow (cfm)	Exhaust Fan Flow (cfm)		Range Hood (cfm)
		Master Bathroom	Bathroom 2	
A	44	46	18	80
B	32	40	12	77
C	45	50	49	110

Table 47. House A Envelope Leakage

House A						
Volume: 9,617 ft³		Test: Depressurization				
Baselines (Pa)	Target House Pressure (Pa)	Fan Pressure (Pa)	Flow @ Pressure (cfm)	Ring	Actual House Pressure (Pa)	ACH
0.65	19	45.1	390	B	-18.9	2.4
-0.7	29	87.7	550	B	-29.1	3.4
-0.62	39	113	634	B	-39.2	4.0
-1.5	49	167	748	B	-49	4.7
0.07	59	205	857	B	-59	5.3
House B						
Volume: 9,617 ft³		Test: Depressurization				
Baselines (Pa)	Target House Pressure (Pa)	Fan Pressure (Pa)	Flow @ Pressure (cfm)	Ring	Actual House Pressure (Pa)	ACH
0.34	19	44.4	388	B	-19	2.4
0.36	29	90.1	538	B	-29	3.4
0.28	39	116.5	649	B	-39.1	4.0
0.62	49	150	740	B	-49.4	4.6
0.22	59	196	818	B	-59.8	5.1
-0.22						
House C						
Volume: 9,617 ft³		Test: Depressurization				
Note: With range hood damper installed						

Baselines (Pa)	Target House Pressure (Pa)	Fan Pressure (Pa)	Flow @ Pressure (cfm)	Ring	Actual House Pressure (Pa)	ACH
0.84	19	27.2	268	B	-19.8	1.7
0.19	29	41.2	384	B	-29.7	2.4
-0.25	39	64.7	475	B	-39	3.0
-0.4	49	106.5	616	B	-48.9	3.8
-0.12	59	151	716	B	-59.3	4.5

Table 48. Transfer Fan Flows—House C

Room	Original Fans (March 30, 2014)		Original Fans (July 22, 2014)		New Fans Max. Speed in Master Bedroom and Bathroom July 22, 2014	
	Fan Side (cfm)	Supply (cfm)	Fan Side (cfm)	Supply (cfm)	Fan Side (cfm)	Supply (cfm)
Master Bedroom	66	48	57	45	133	106
Master Bathroom	51	48	51	45	125	115
Northwest Bedroom	42	38	Not measured	37	N/A	N/A
Northeast Bedroom	56	48	55	49	N/A	N/A

Appendix B: Monitoring Points and Equipment

Table 49. Inventory of Monitoring Equipment, Data Loggers, and Sensors

Quantity	Description/ Purpose	Make and Model of Monitoring Equipment (or Equal)	A	B	C	Outdoors
3	Data logging equipment	Campbell Scientific data logger, model: CR1000	1	1	1	
3		Campbell Scientific data logger, power supply	1	1	1	
3		Campbell Scientific, multiplexer model: AM16/32B	1	1	1	
15	Air temperature measurement	Watlow thermocouples	5	5	5	
2	Electric furnace power (100 amps)	Continental Control Systems, ACT-0750-100	2			
7	Air temperature/relative humidity	Campbell CS215 probe	1	4	1	1
19	Protection for outdoor air temperature and relative humidity sensor	Campbell Scientific, 41303-5A radiation shield	6	6	6	1
1	Solar radiation sensor	Campbell LI200X-L silicon pyranometer				1
4	Moisture content of wood	Omnisense S900-1		2	2	
6	Compressor and fan current measurements	Veris, H721LC (10 amps)	2	2	2	
2	Status heat pump reversing valve	Veris, H300		1	1	
6	Heat pump/AC power (20 amps)	Continental Control Systems, ACT-0750-20	2	2	2	
6	Total house power (150 amps)	Continental Control Systems, ACT-0750-150	2	2	2	
7	Wattnode power meter	Continental Control Systems, WNB-3Y-208-P	3	2	2	

Quantity	Description/ Purpose	Make and Model of Monitoring Equipment (or Equal)	A	B	C	Outdoors
1	Bath fan run time	Veris, H300			1	
1	Electric furnace resistance element status	Veris, H300		1		
3	Serial-to-Ethernet interface for wireless communication	Campbell Scientific, NL201	1	1	1	
4	Monitor run time of through- wall air transfer fans	Veris, H300				4

Appendix C: Impacts of Transfer Fans

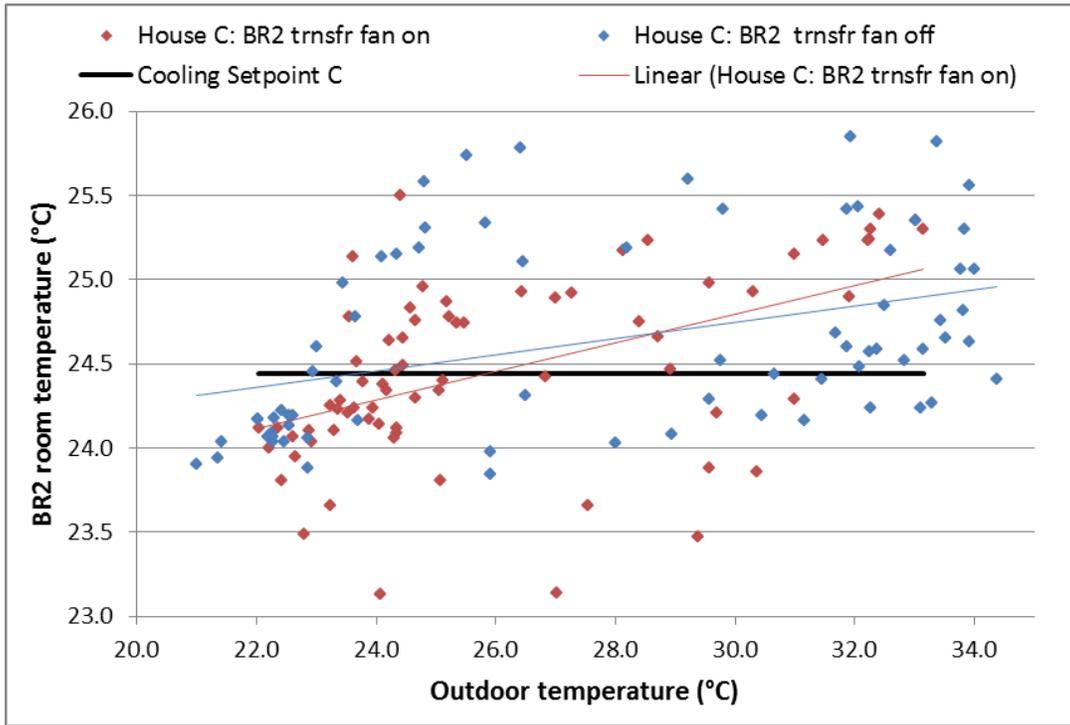


Figure 43. House C Bedroom 2 temperatures with and without transfer fan operation (cooling season)

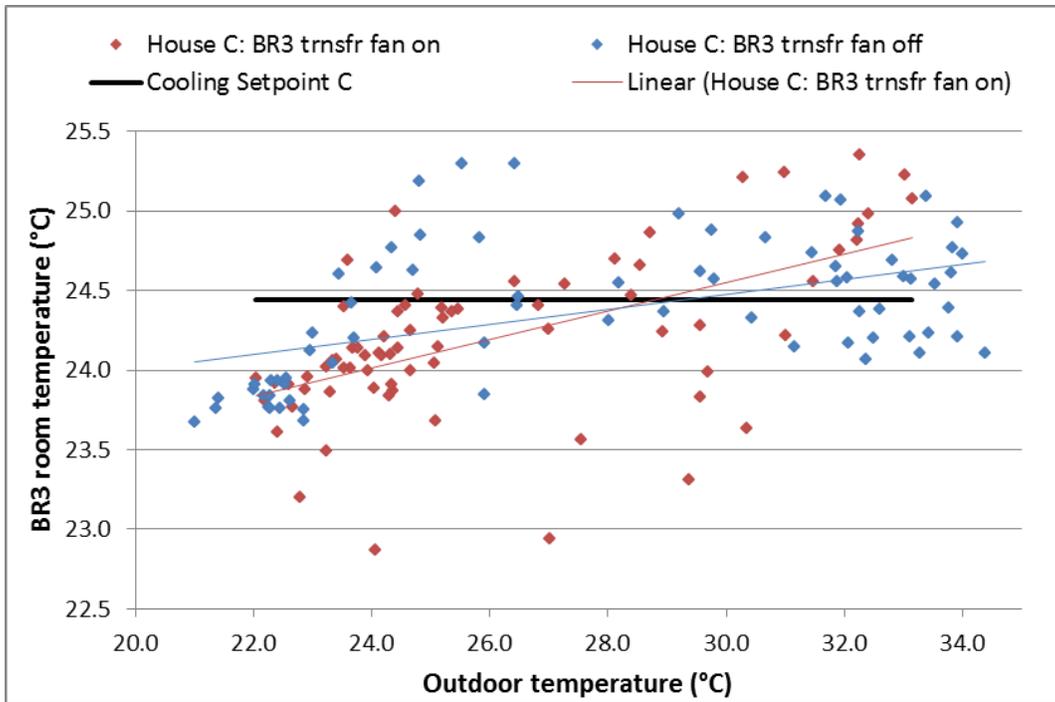


Figure 44. House C Bedroom 3 temperatures with and without transfer fan operation (cooling season)

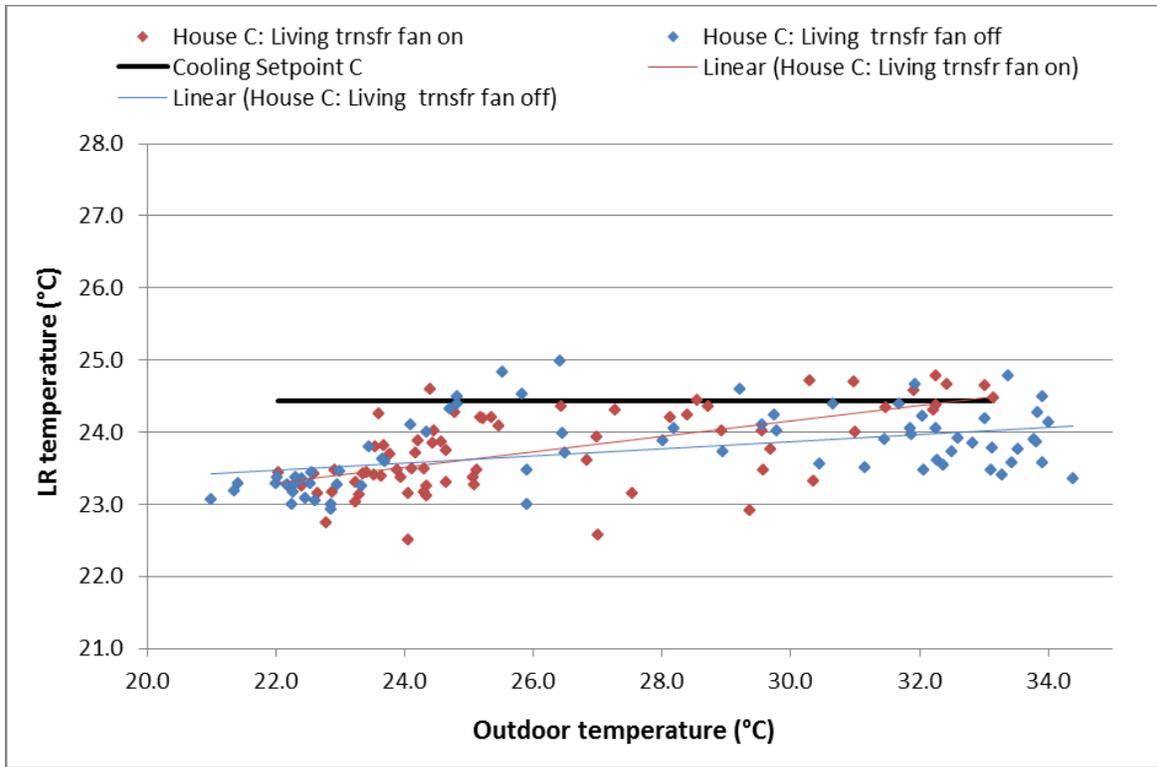


Figure 45. House C living room temperatures with and without transfer fan operation (cooling season)

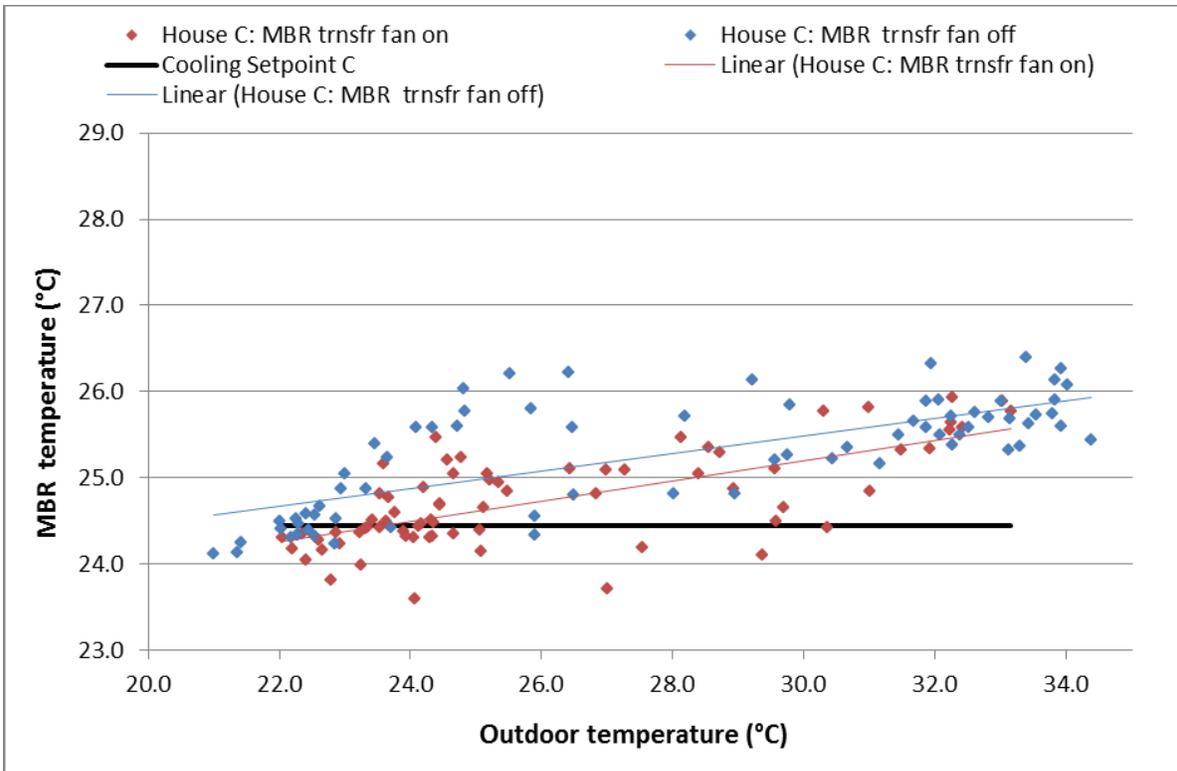


Figure 46. House C master bedroom temperatures with and without transfer fan operation (cooling season)

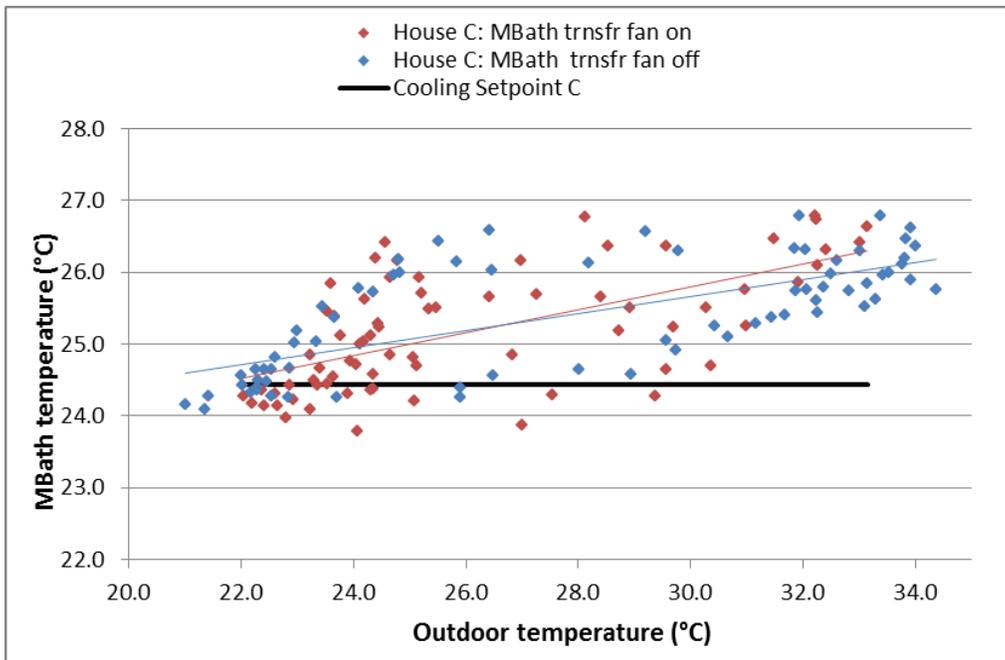


Figure 47. House C master bathroom temperatures with and without transfer fan operation (cooling season)

Appendix D: Impacts of Blind Position

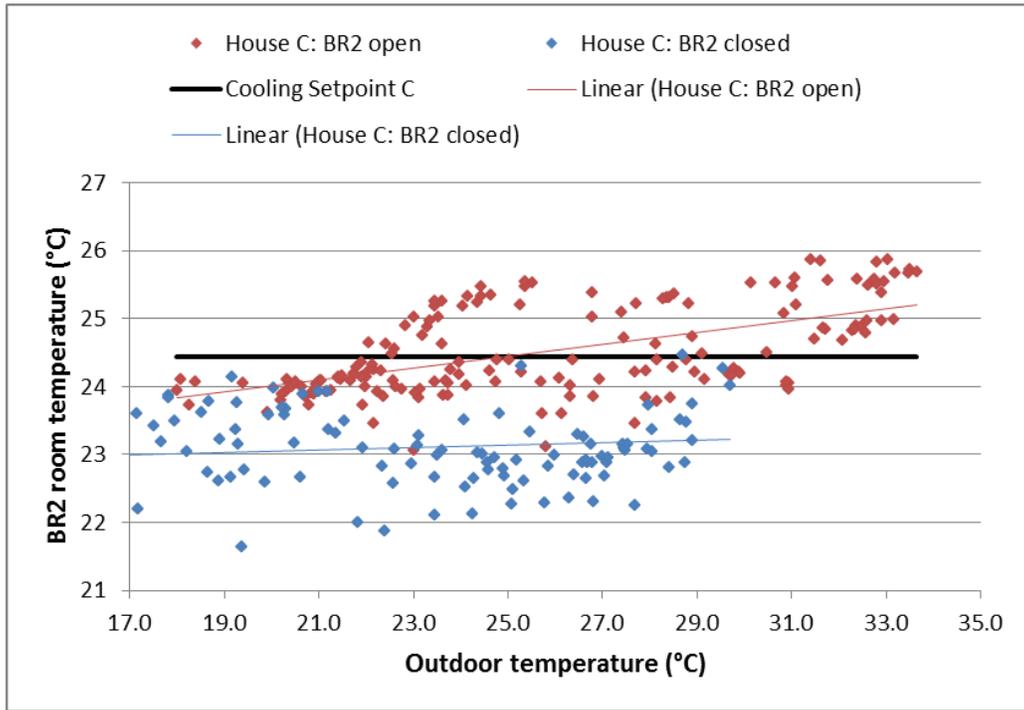


Figure 48. House C impacts of blind position in Bedroom 2 on room temperature compared to outdoor temperature (cooling season)

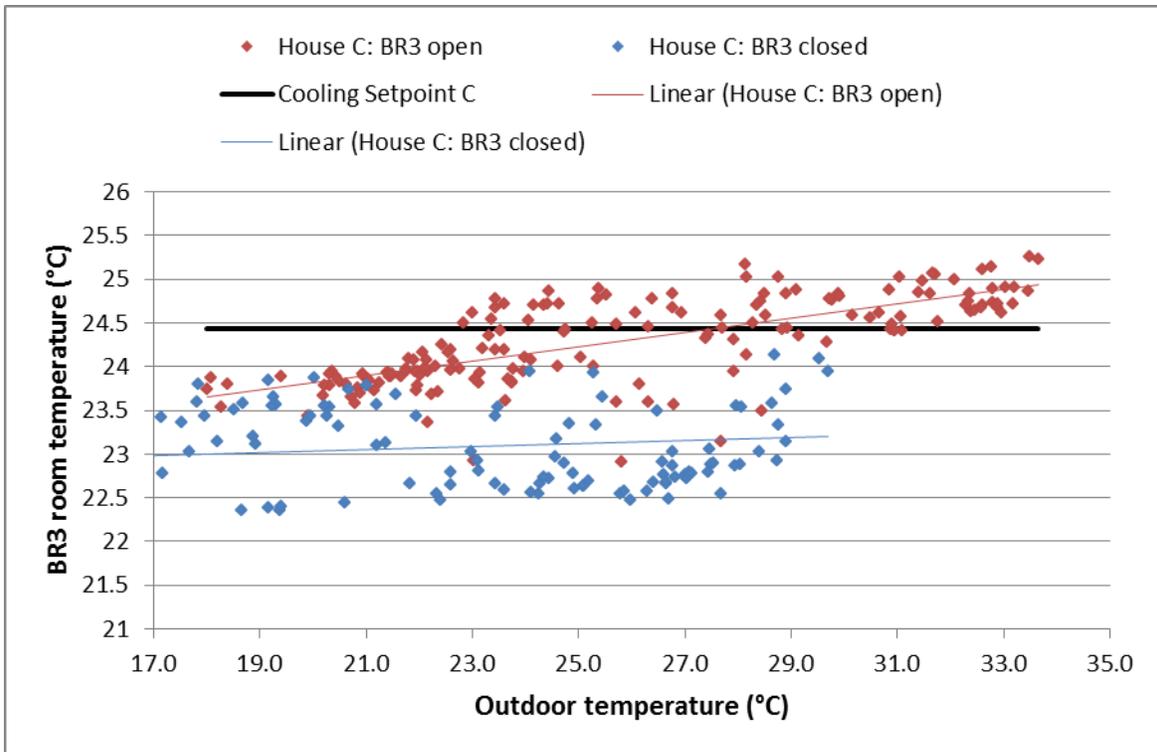


Figure 49. House C impacts of blind position on Bedroom 3 on room temperature compared to outdoor temperature (cooling season)

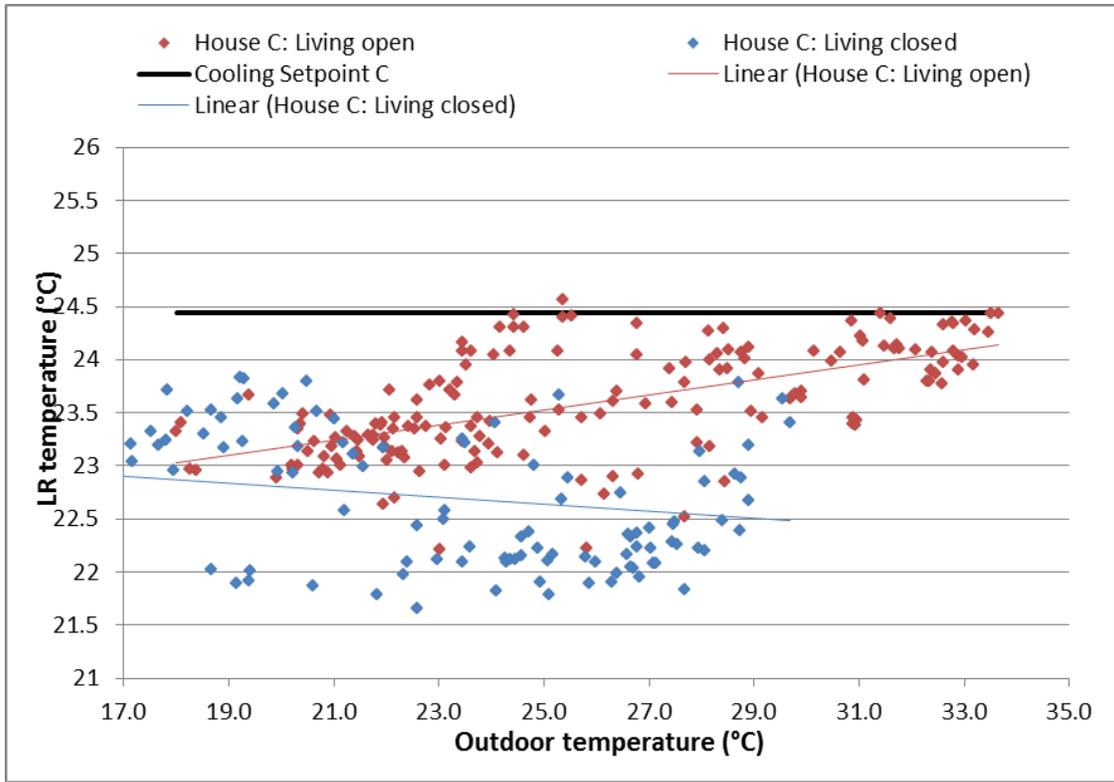


Figure 50. House C impacts of blind position on living room temperature compared to outdoor temperature (cooling season)

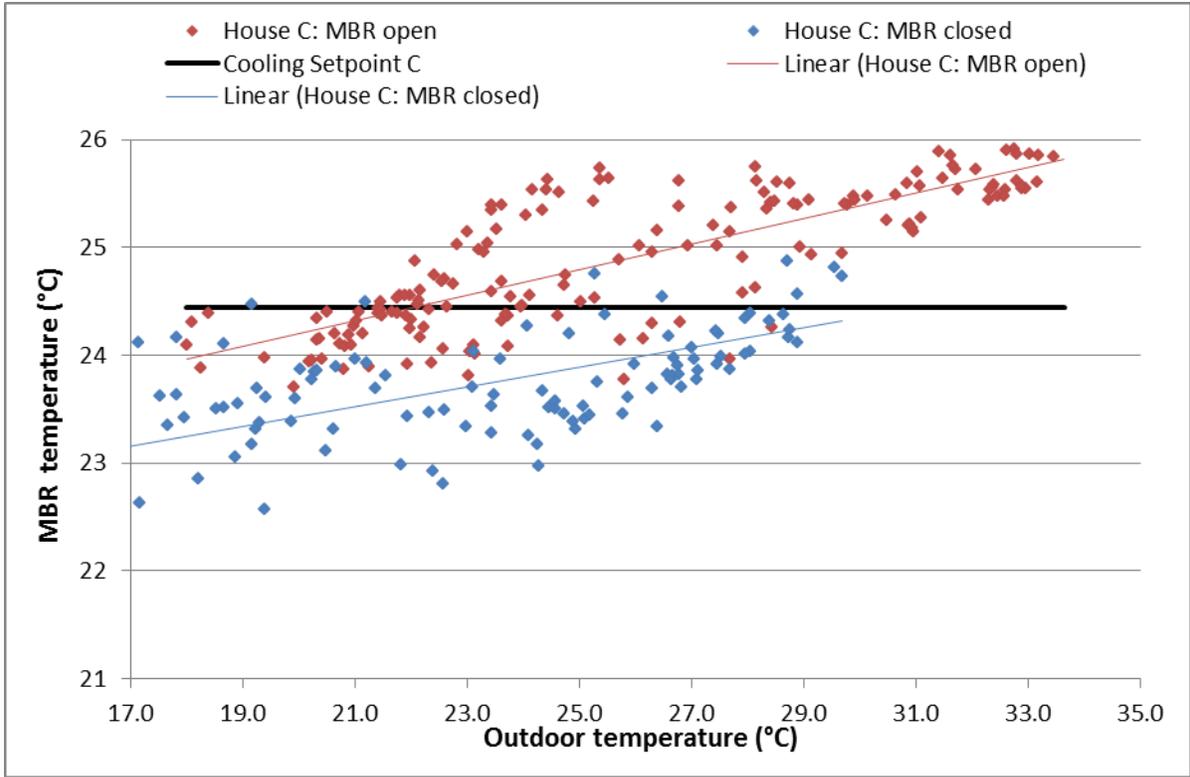


Figure 51. House C impacts of blind position on master bedroom temperature compared to outdoor temperature (cooling season)

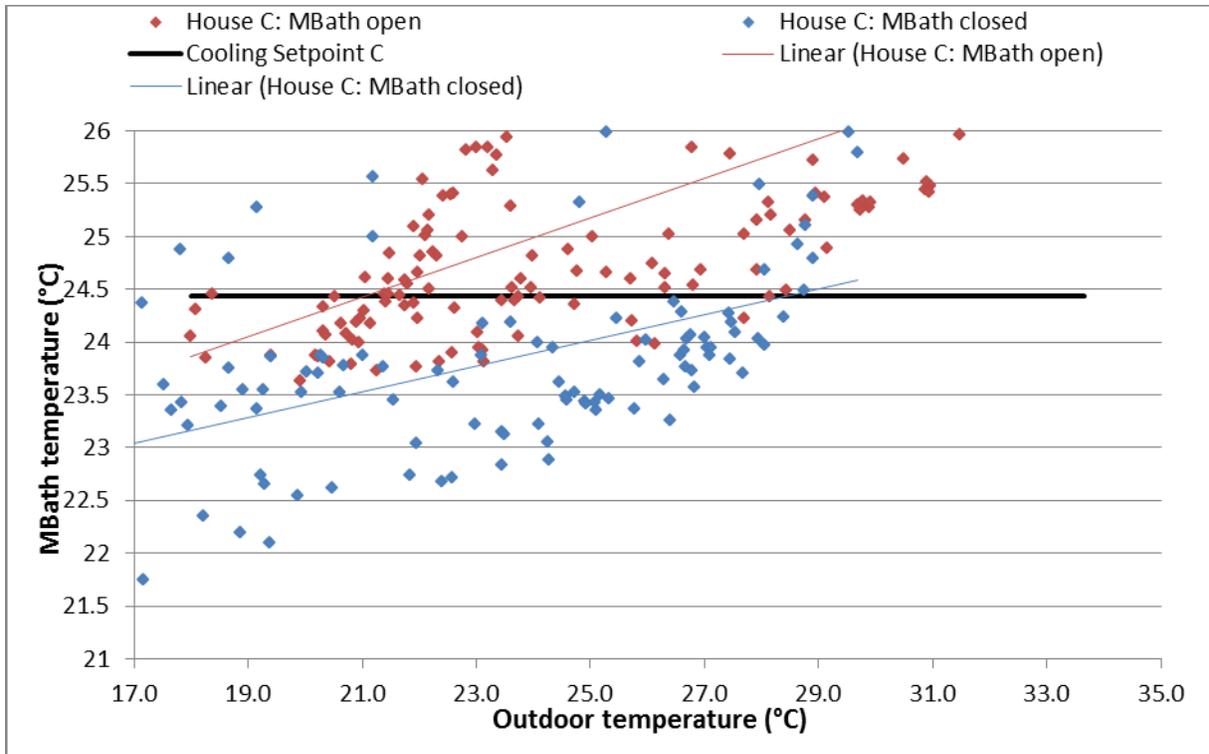


Figure 52. House C impacts of blind position on master bathroom temperature compared to outdoor temperature (cooling season)

Appendix E: Impacts of Door Position

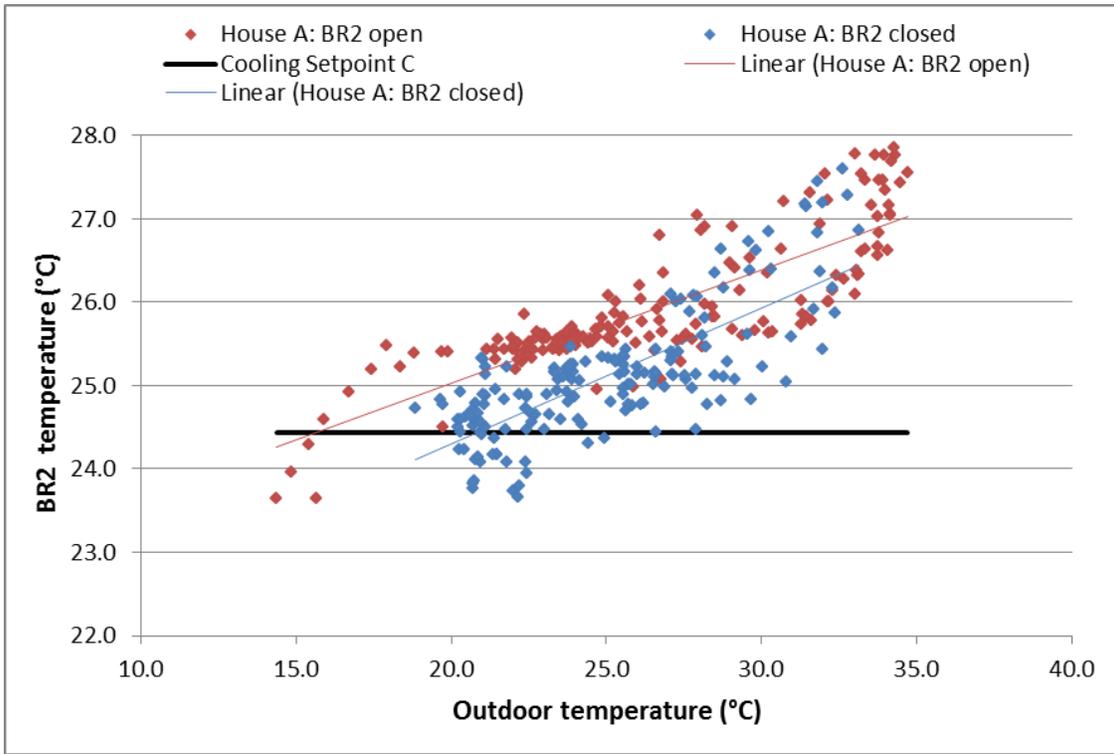


Figure 53. House A impacts of door position on Bedroom 2 temperature compared to outdoor temperature (0.8 cfm/ft²) (cooling season)

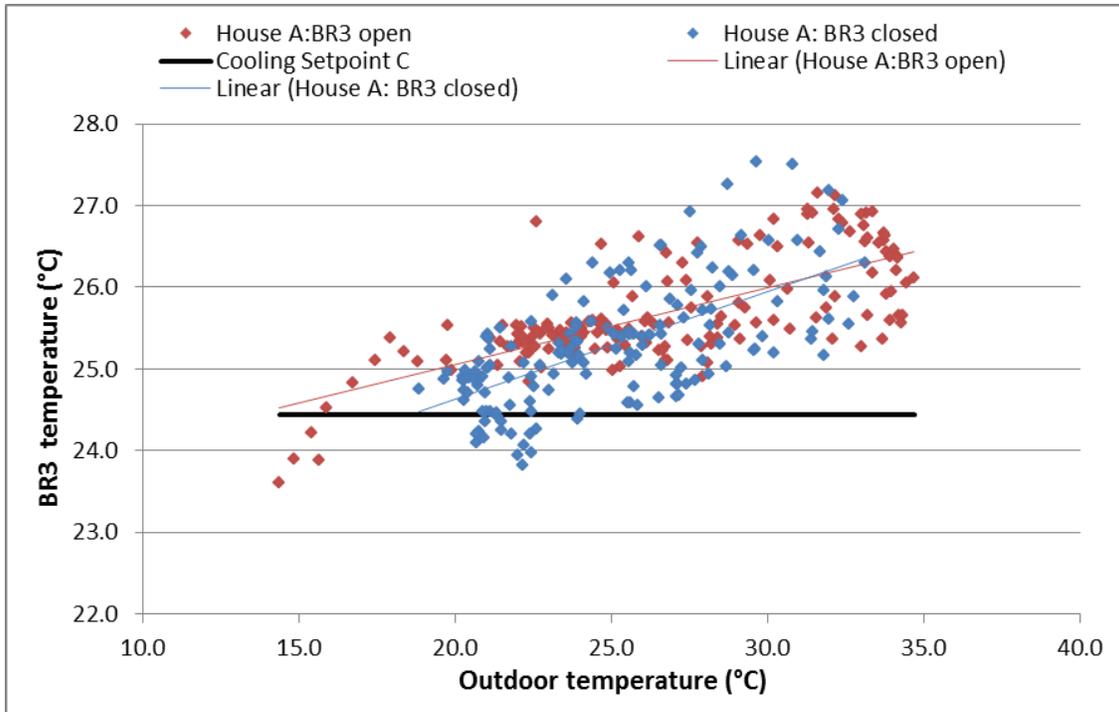


Figure 54. House A impacts of door position on Bedroom 3 temperature compared to outdoor temperature (1 cfm/ft²) (cooling season)

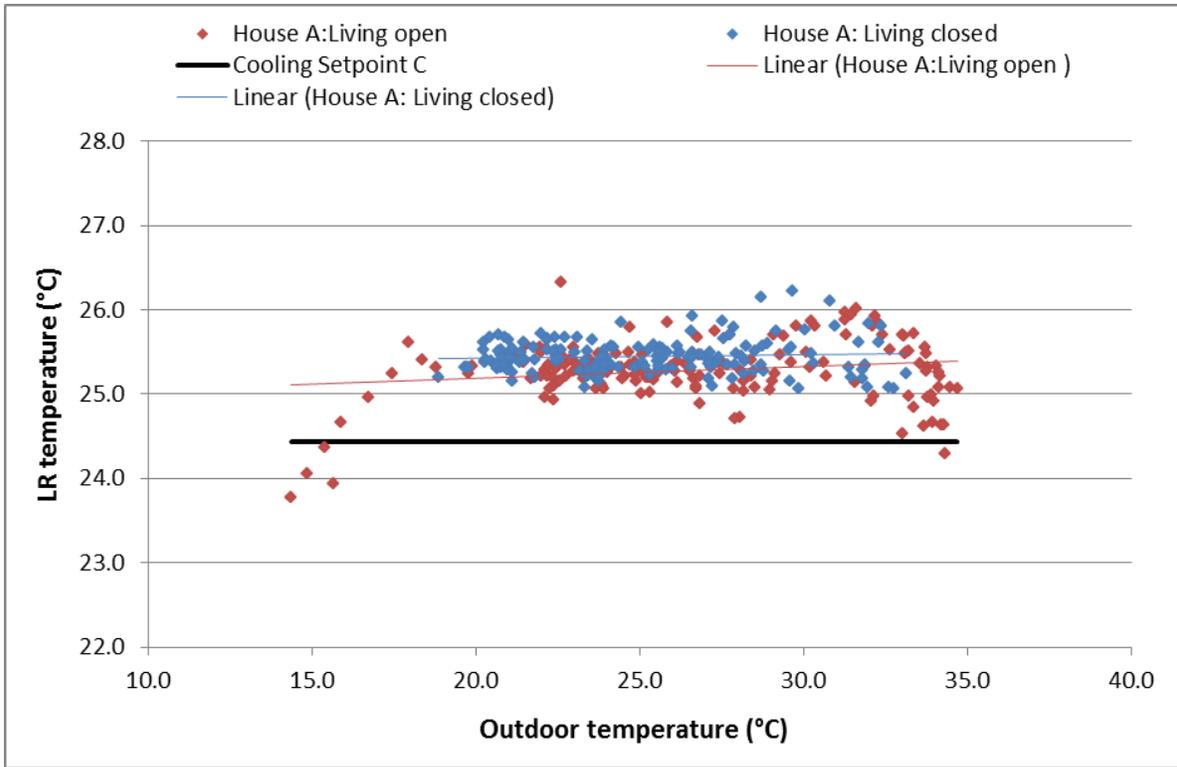


Figure 55. House A impacts of door position on living room temperature compared to outdoor temperature (cooling season)

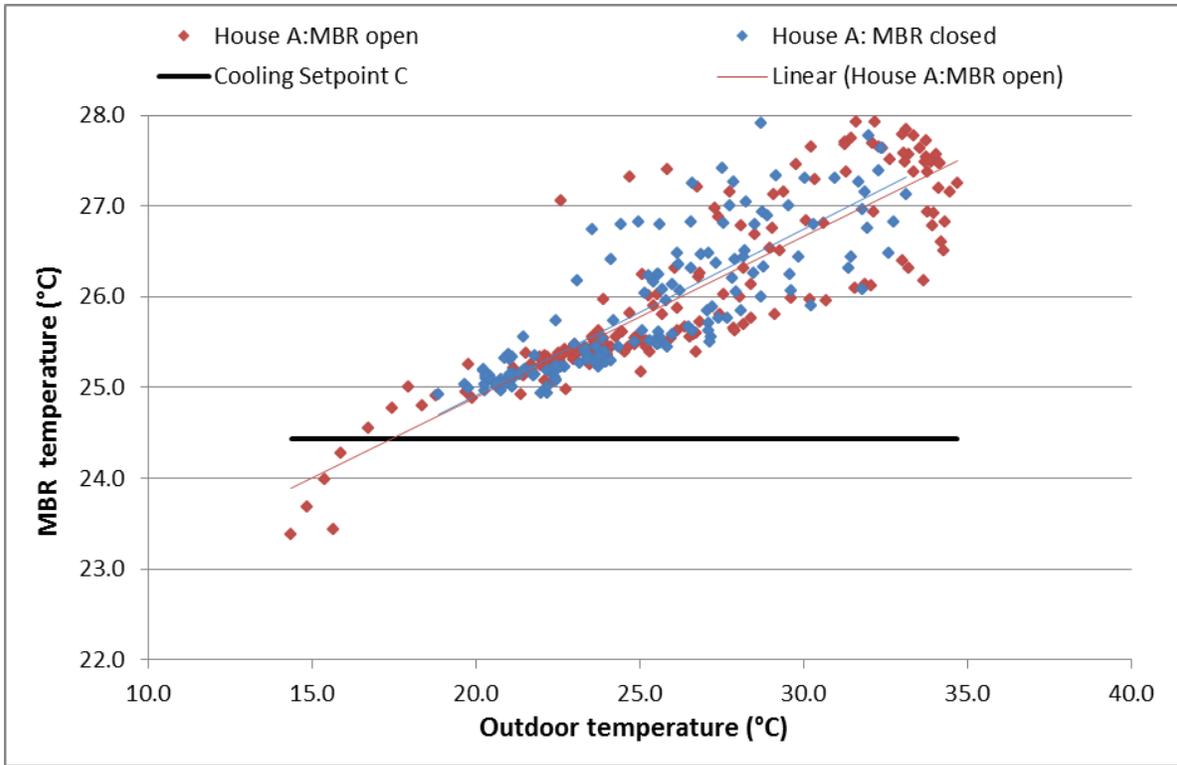


Figure 56. House A impacts of door position on master bedroom temperature compared to outdoor temperature (0.5 cfm/ft^2) (cooling season)

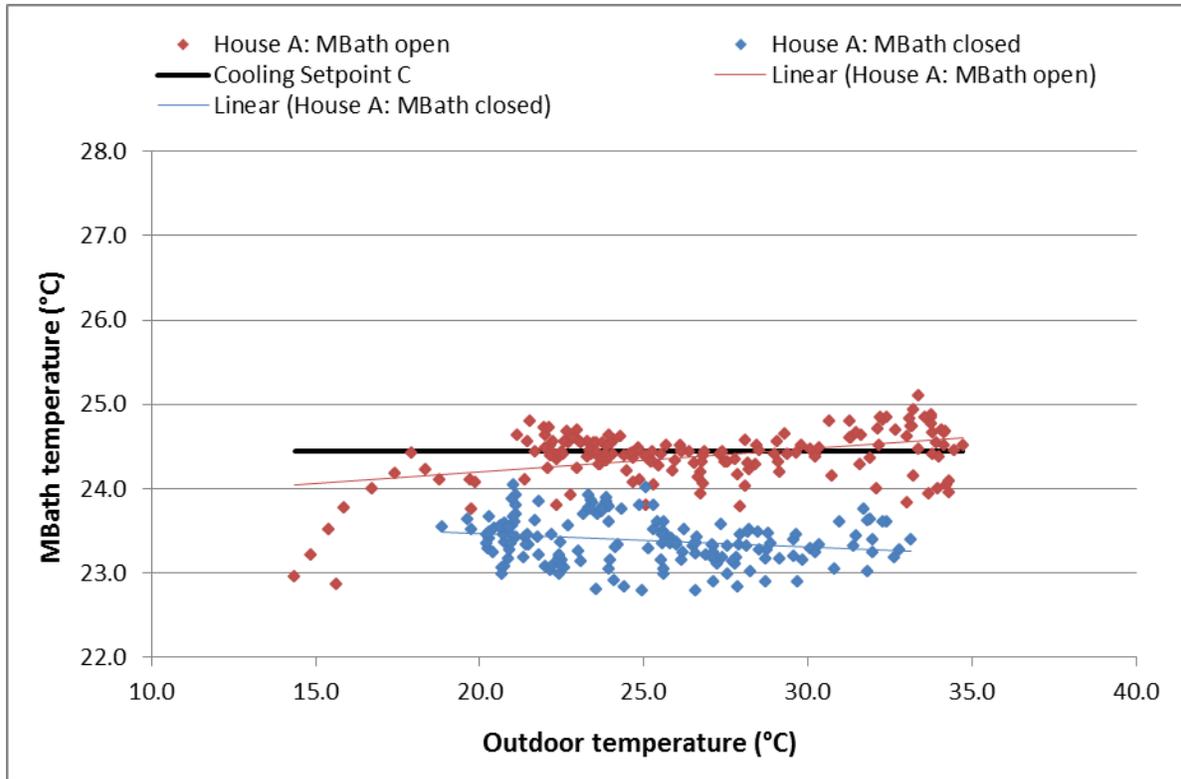


Figure 57. House A impacts of door position on master bathroom temperature compared to outdoor temperature (1.2 cfm/ft²) (cooling season)

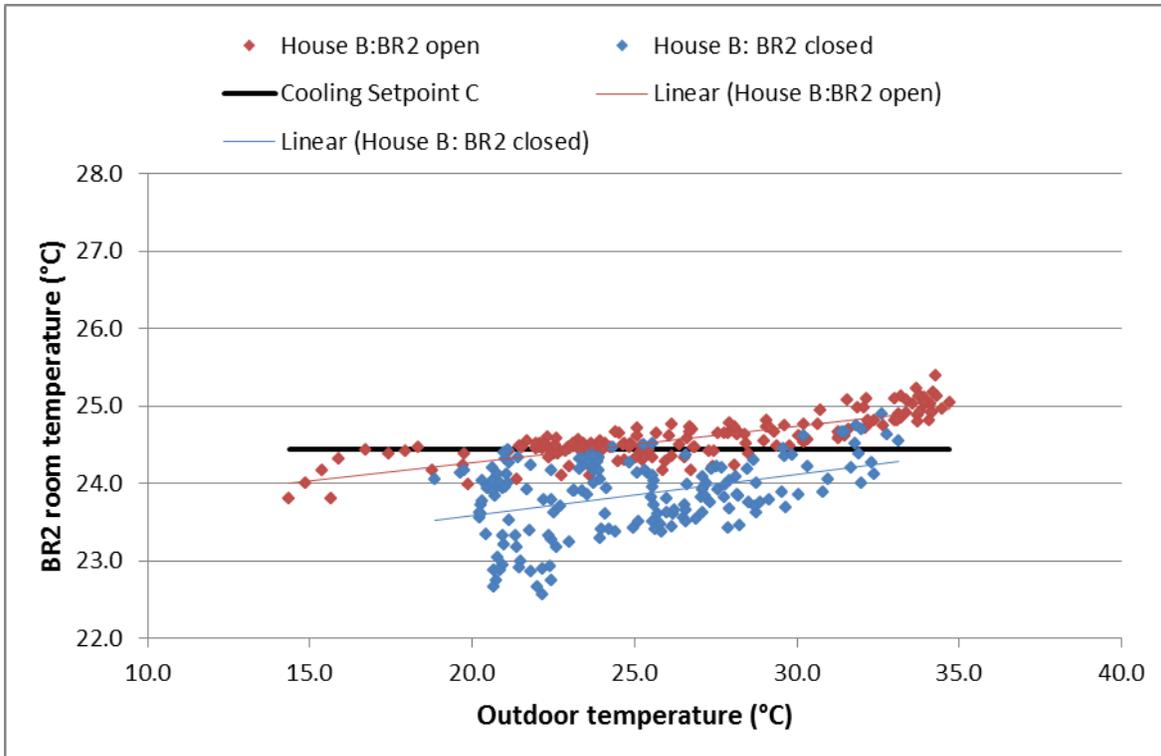


Figure 58. House B impacts of door position on Bedroom 2 room temperature compared to outdoor temperature (0.7 cfm/ft²) (cooling season)

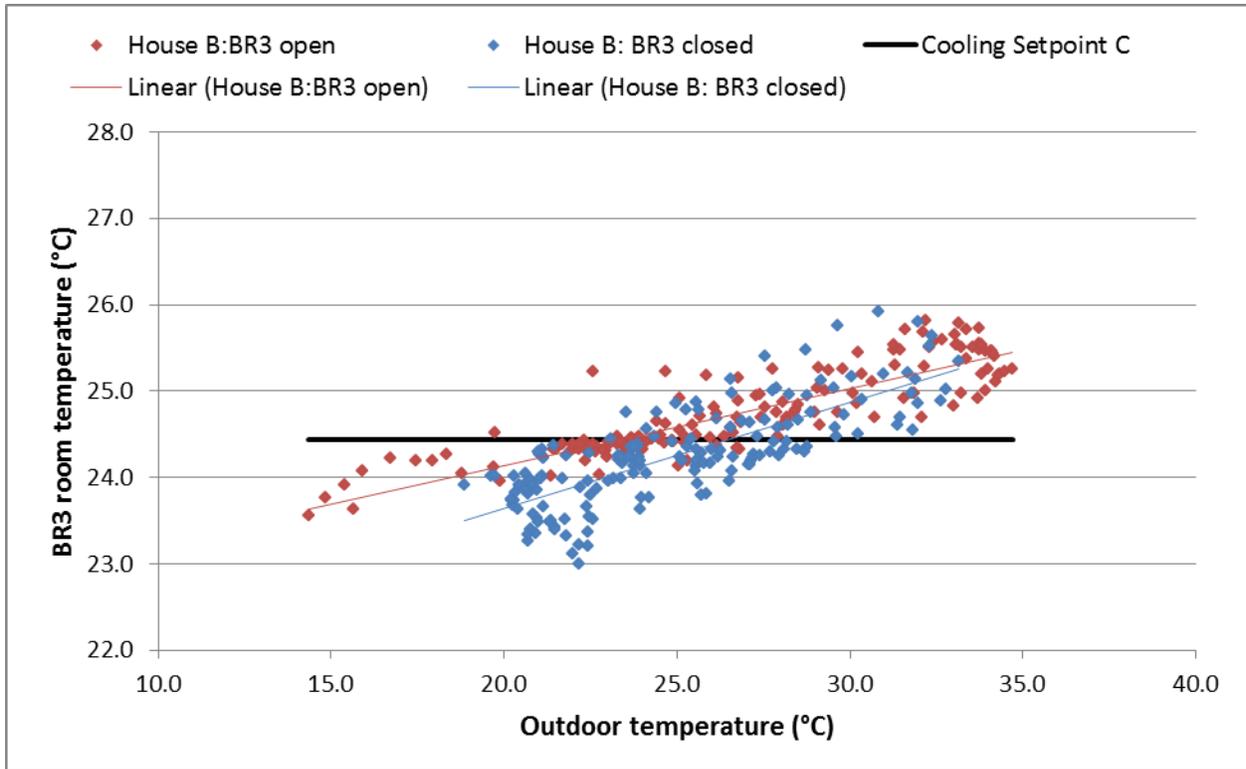


Figure 59. House B impacts of door position on Bedroom 3 room temperature compared to outdoor temperature (0.8 cfm/ft²) (cooling season)

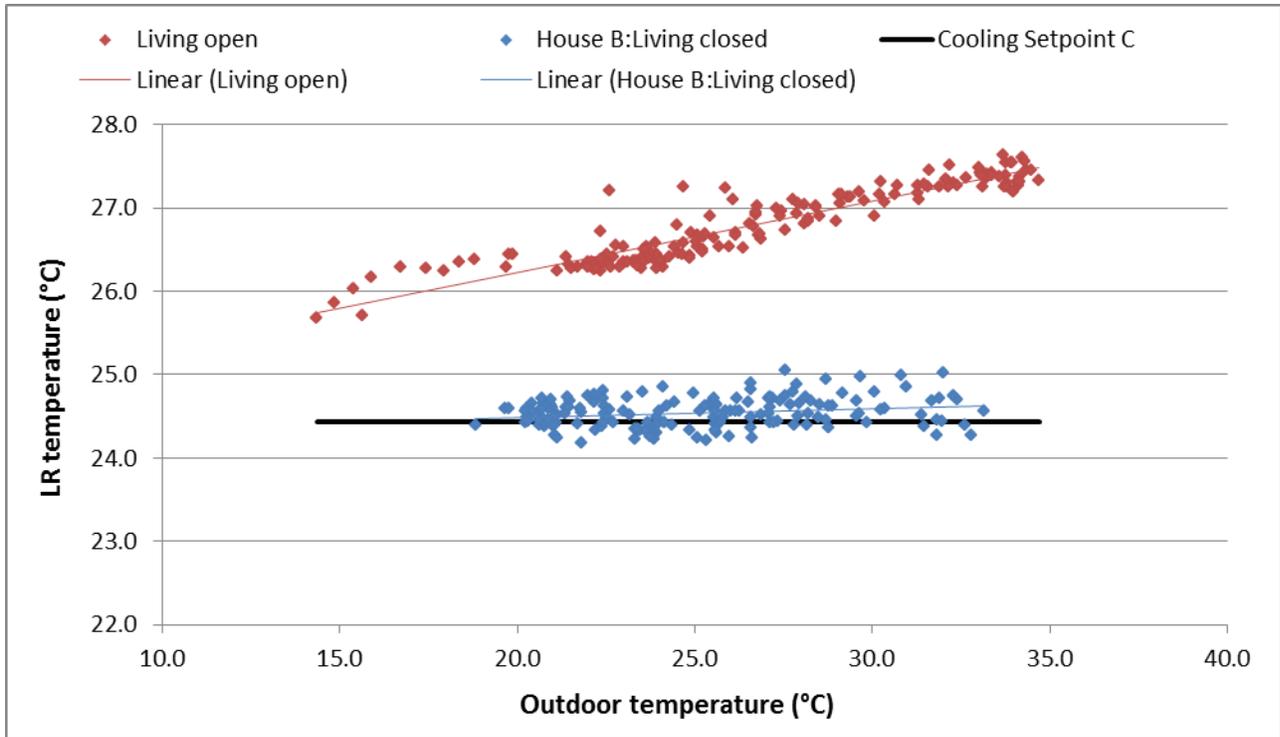


Figure 60. House B impacts of door position on living room temperature compared to outdoor temperature—air conditioner failure open period (cooling season)

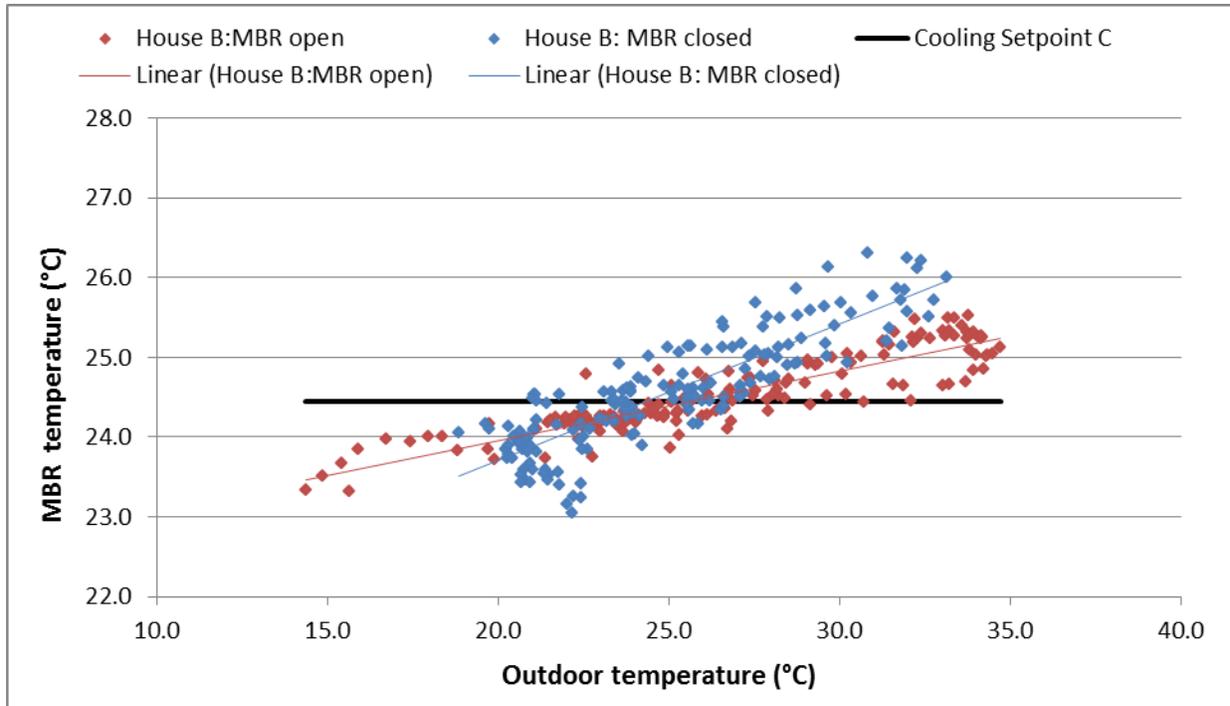


Figure 61. House B impacts of door position on master bedroom temperature compared to outdoor temperature (0.5 cfm/ft²) (cooling season)

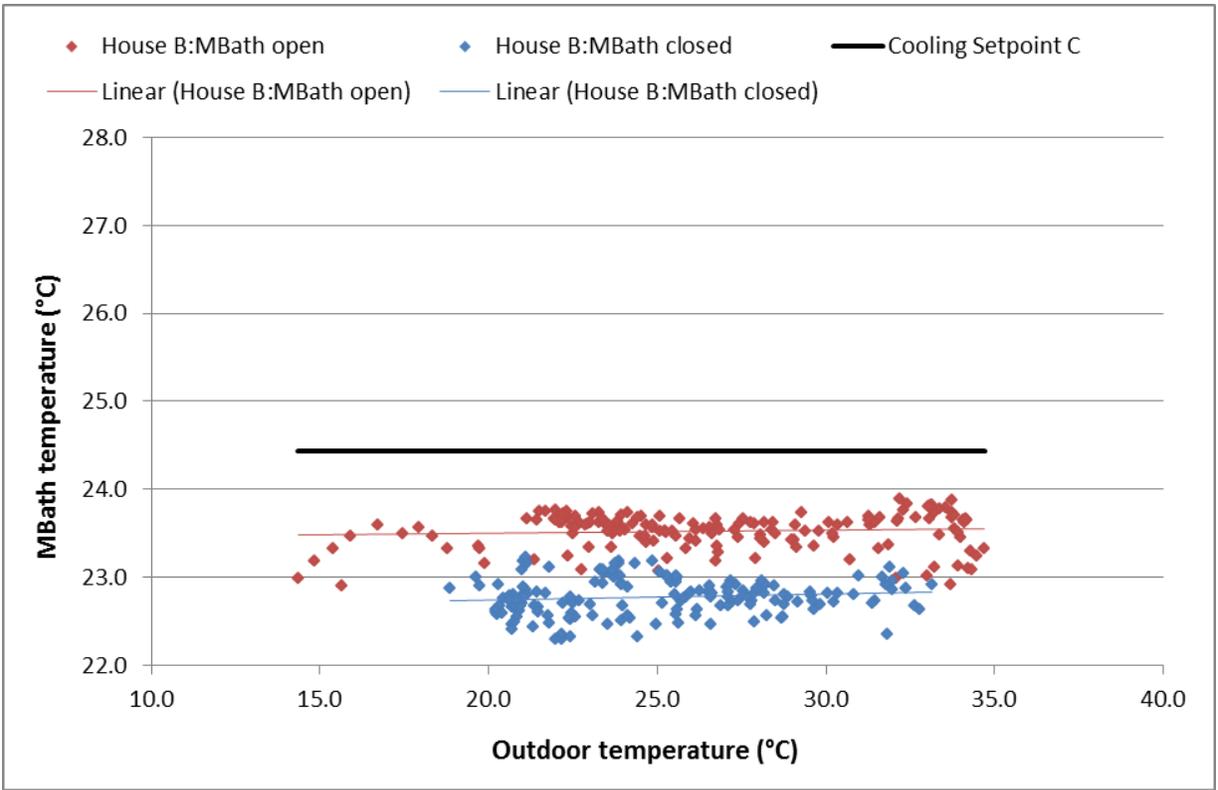


Figure 62. House B impacts of door position on master bathroom temperature compared to outdoor temperature (1.2 cfm/ft²) (cooling season)

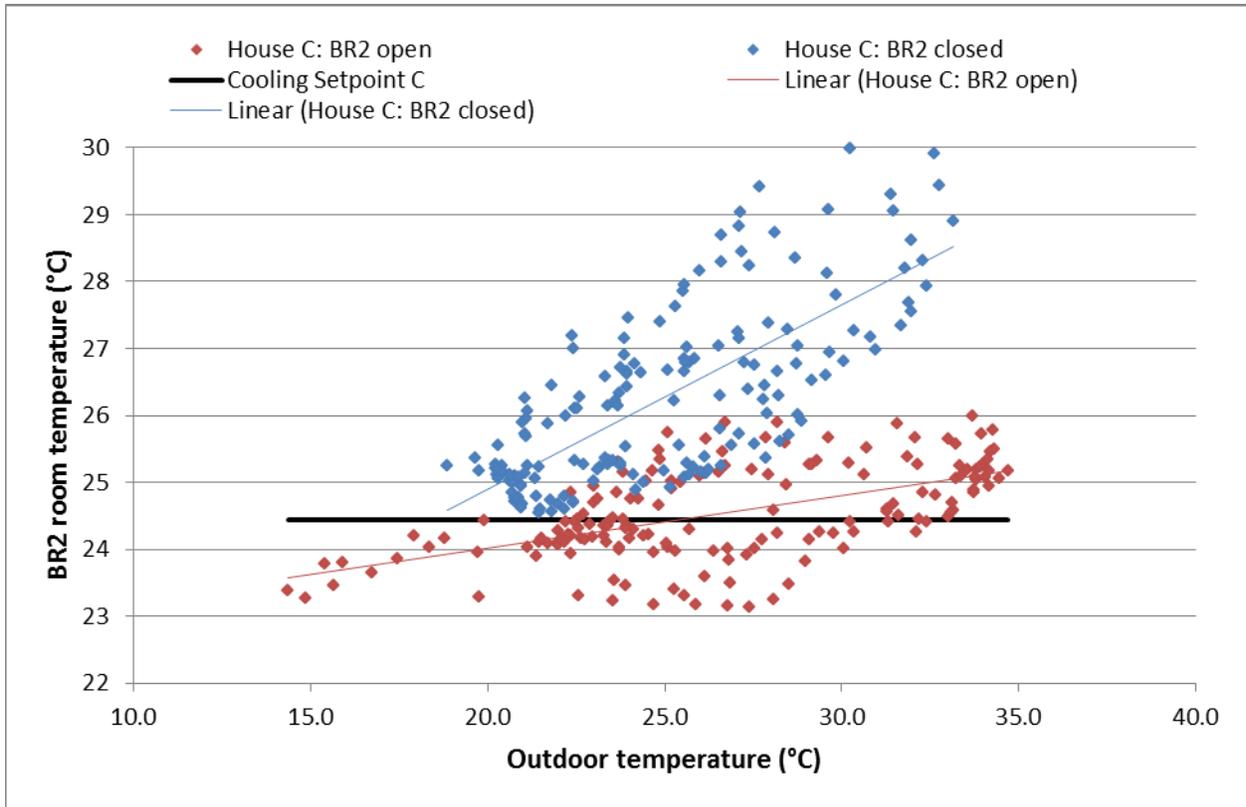


Figure 63. House C impacts of door position on Bedroom 2 room temperature compared to outdoor temperature (cooling season)

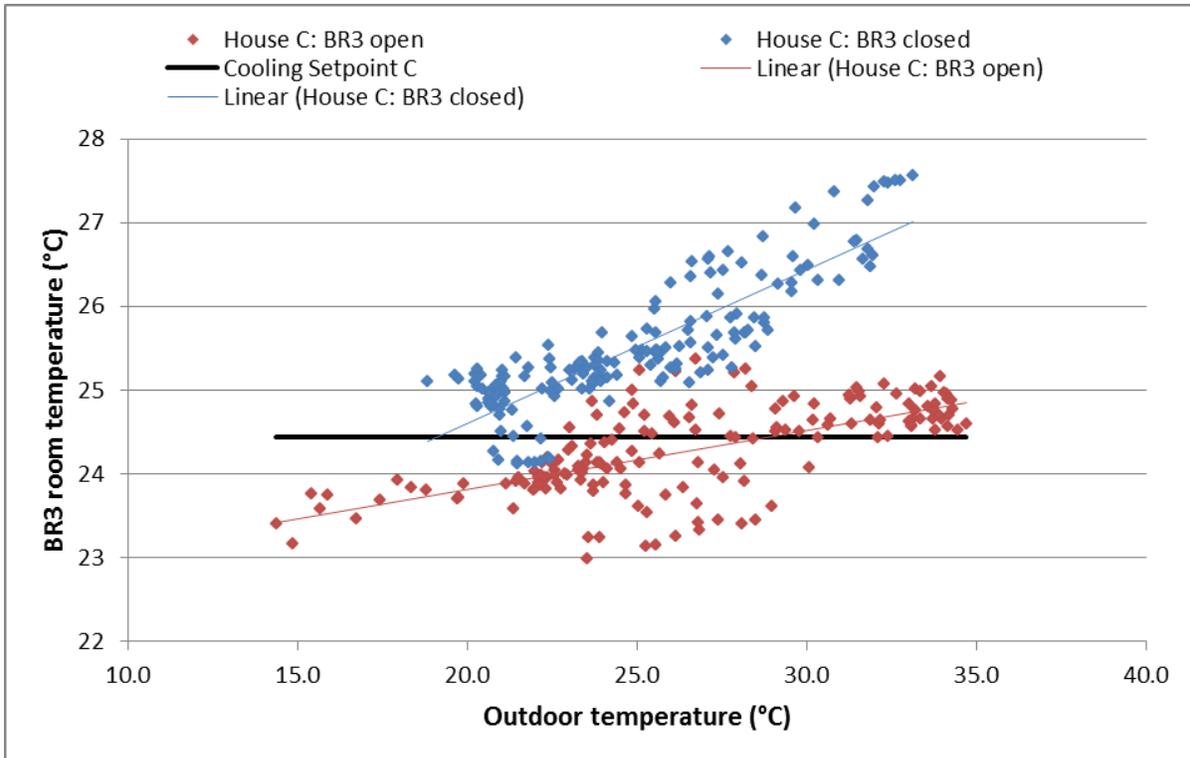


Figure 64. House C impacts of door position on Bedroom 3 room temperature compared to outdoor temperature (cooling season)

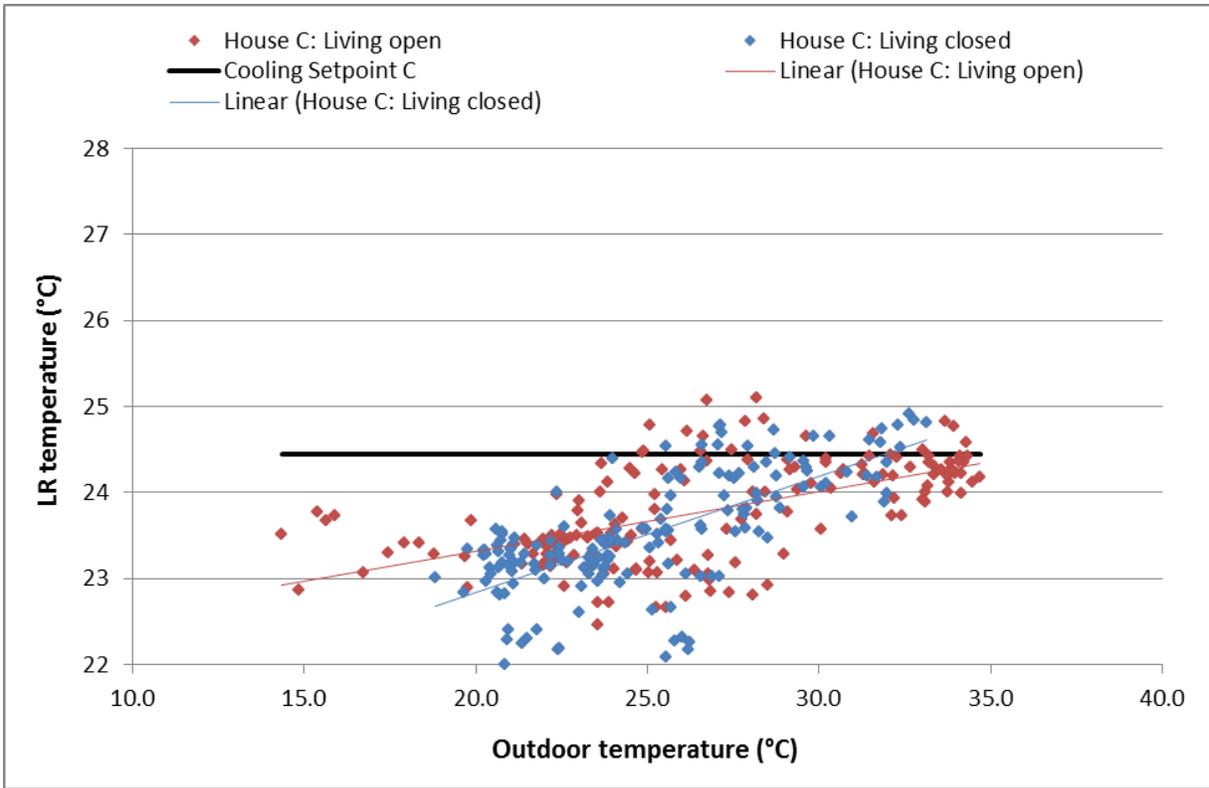


Figure 65. House C impacts of door position on living room temperature compared to outdoor temperature (cooling season)

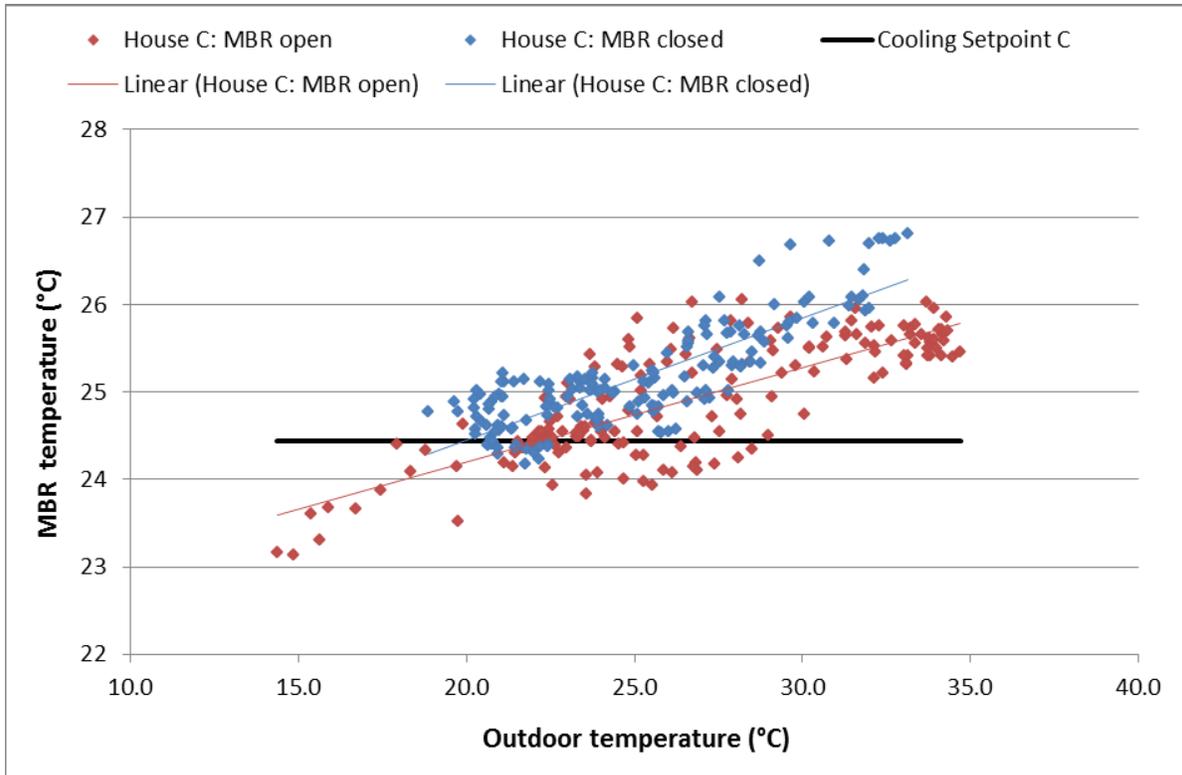


Figure 66. House C impacts of door position on master bedroom temperature compared to outdoor temperature (cooling season)

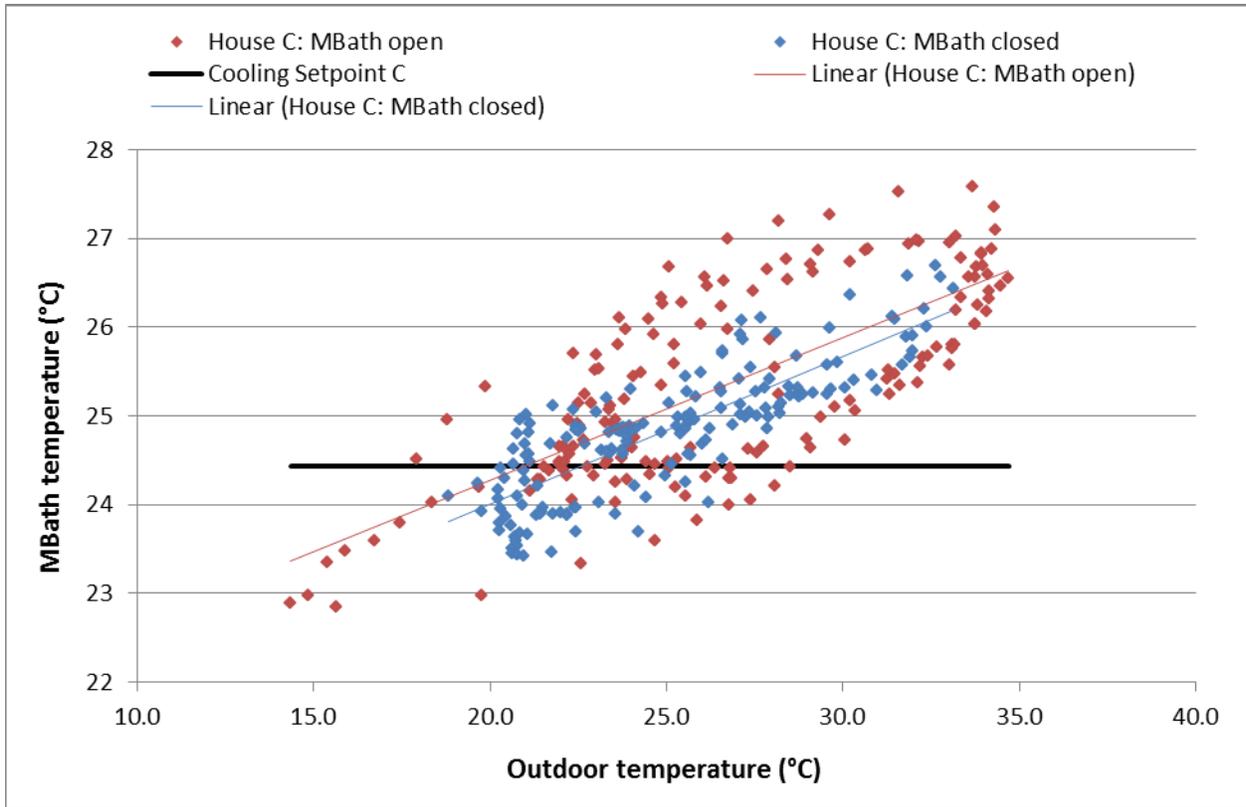


Figure 67. House C impacts of door position on master bathroom temperature compared to outdoor temperature (cooling season)

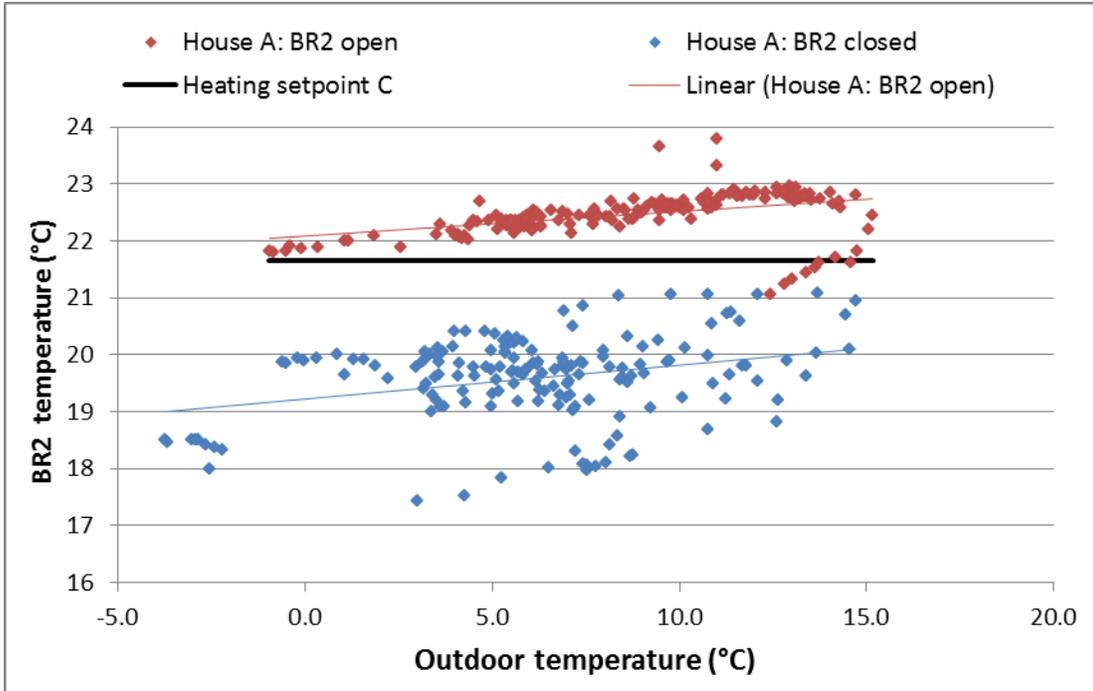


Figure 68. House A impacts of door position on Bedroom 2 temperature compared to outdoor temperature (heating season)

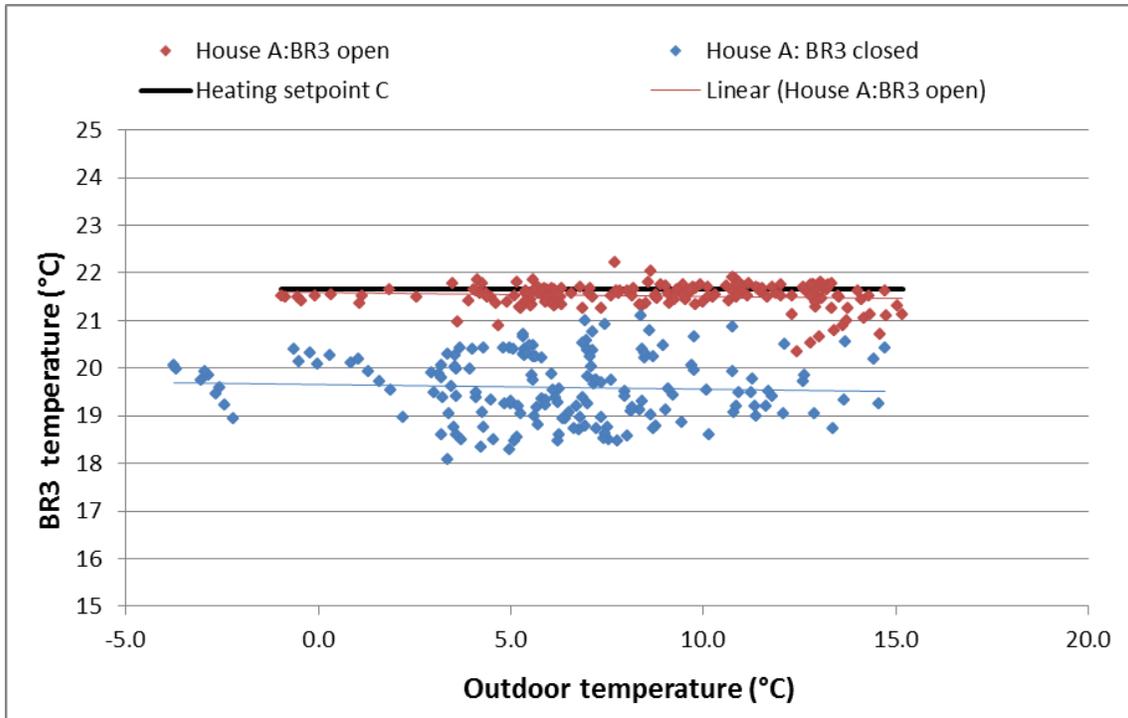


Figure 69. House A impacts of door position on Bedroom 3 temperature compared to outdoor temperature (heating season)

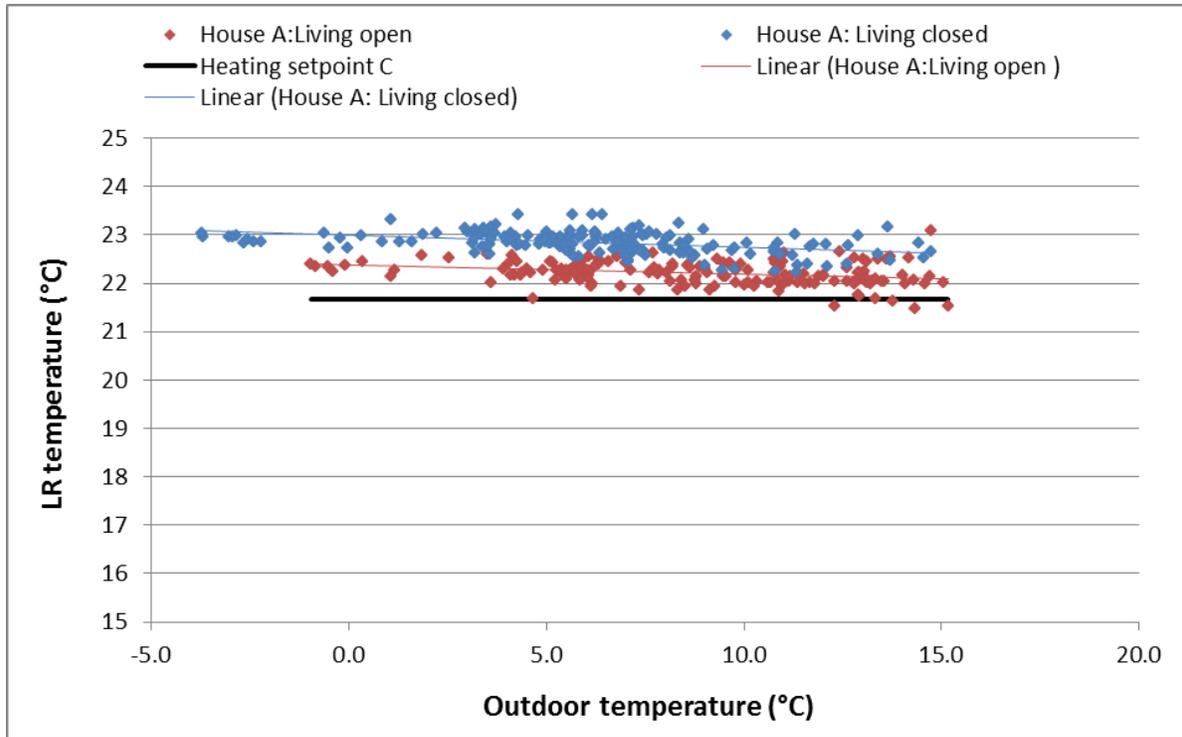


Figure 70. House A impacts of door position on living room temperature compared to outdoor temperature (heating season)

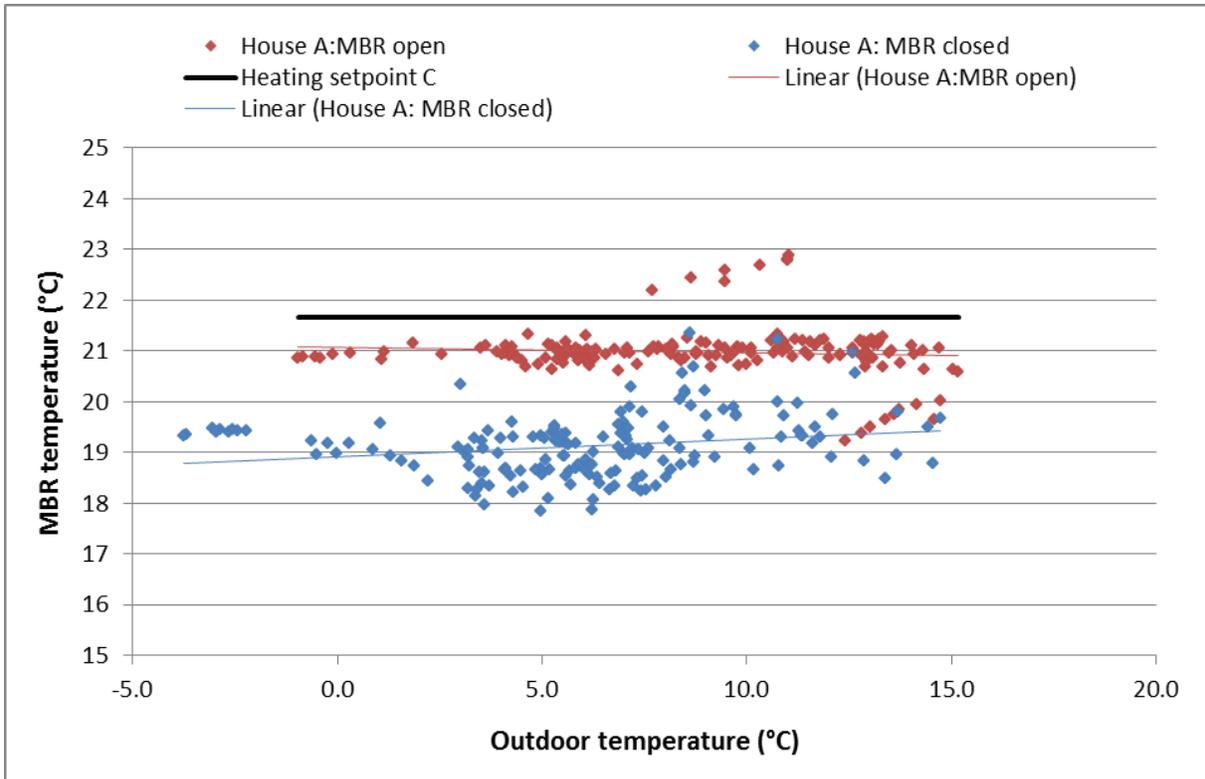


Figure 71. House A impacts of door position on master bedroom temperature compared to outdoor temperature (heating season)

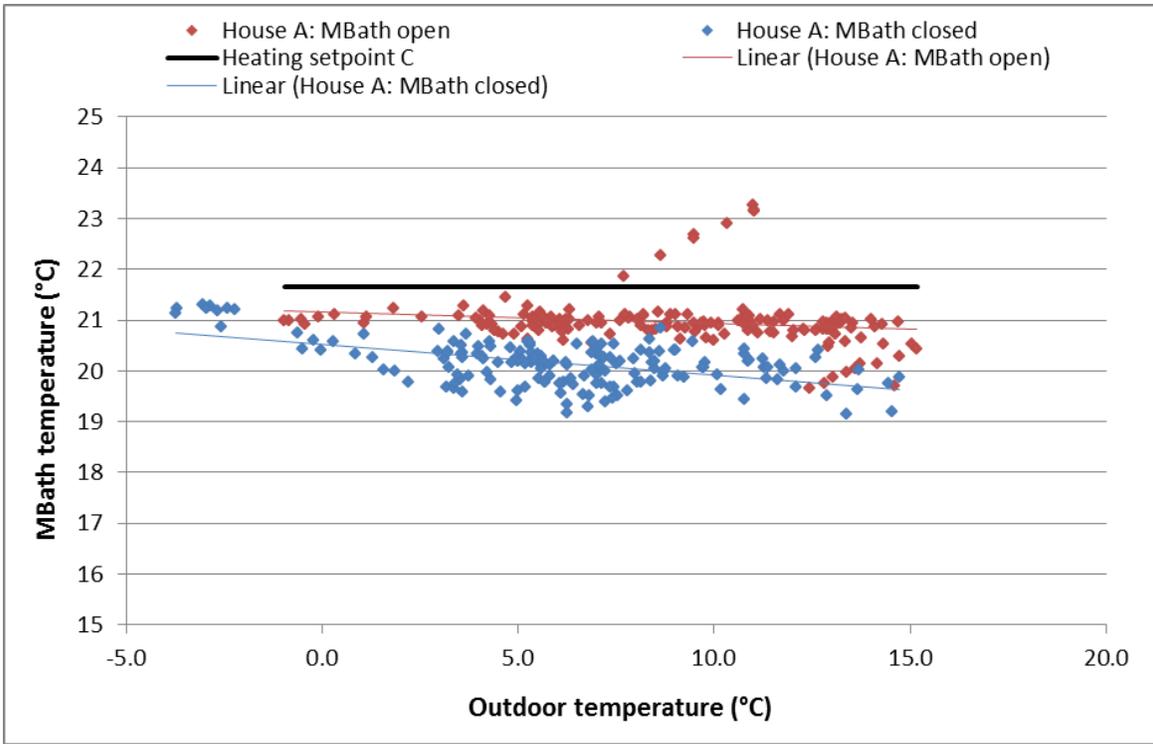


Figure 72. House A impacts of door position on master bathroom temperature compared to outdoor temperature (heating season)

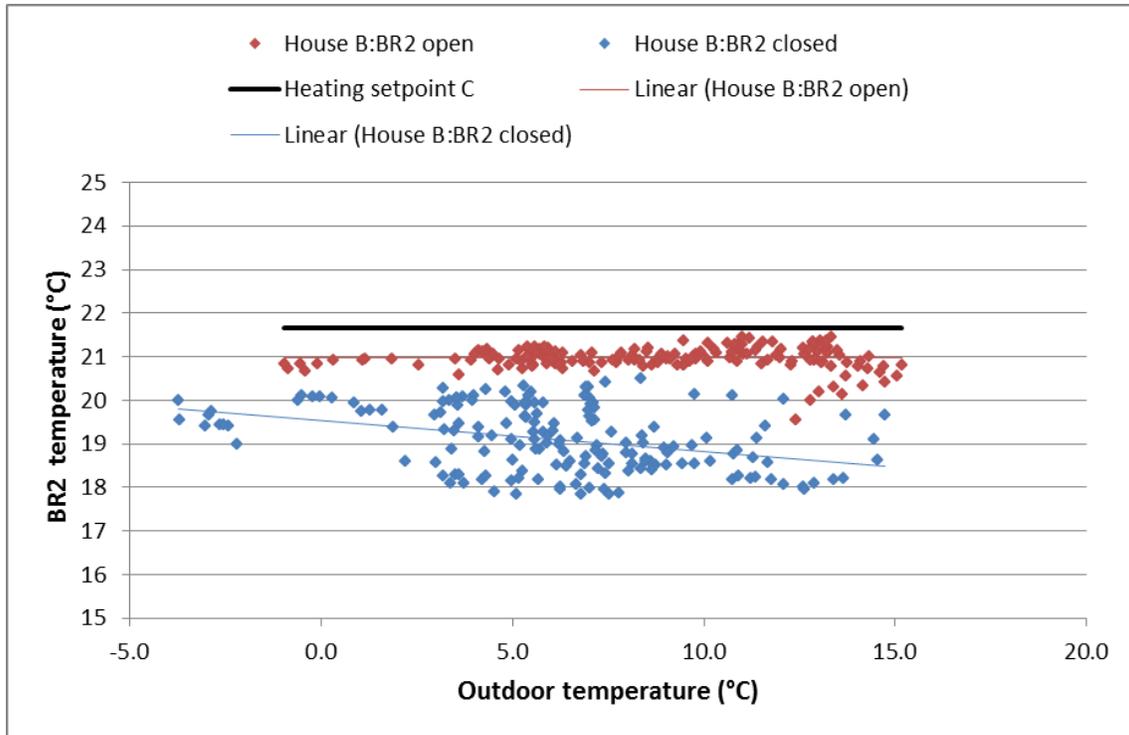


Figure 73. House B impacts of door position on Bedroom 2 temperature compared to outdoor temperature (heating season)

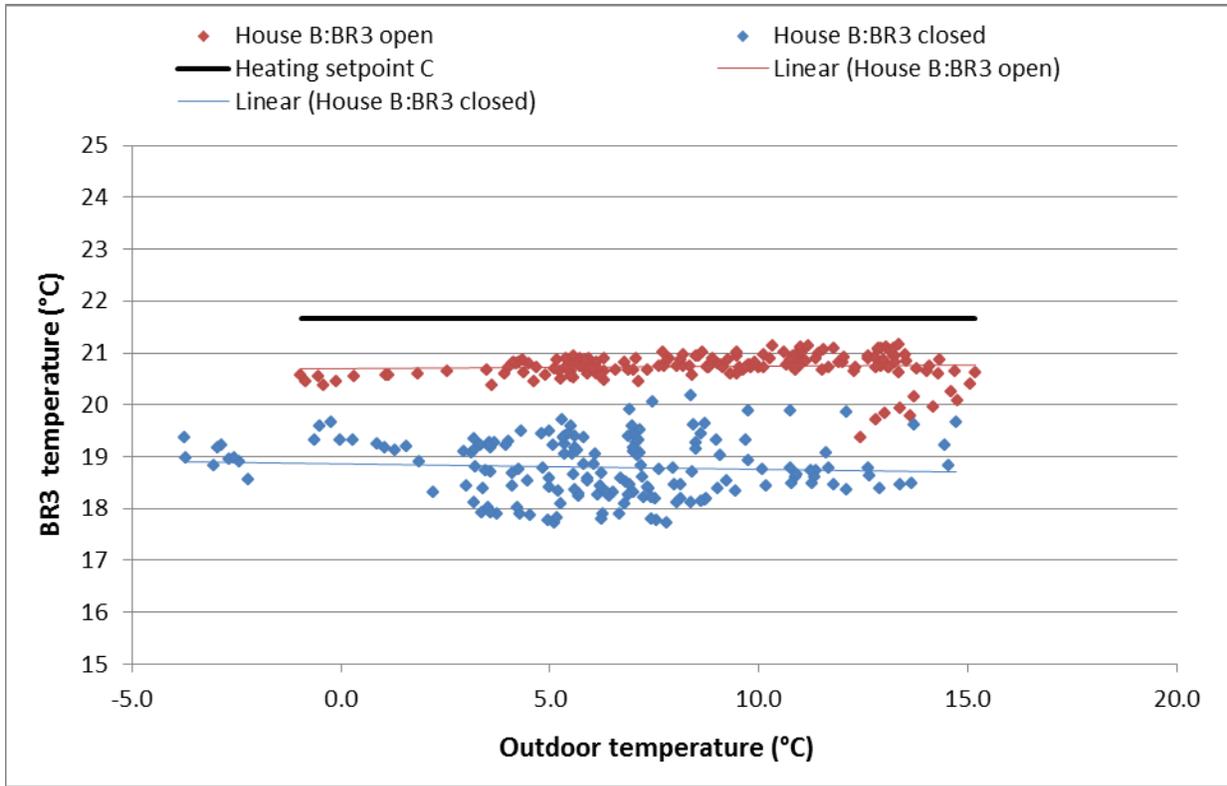


Figure 74. House B impacts of door position on Bedroom 3 temperature compared to outdoor temperature (heating season)

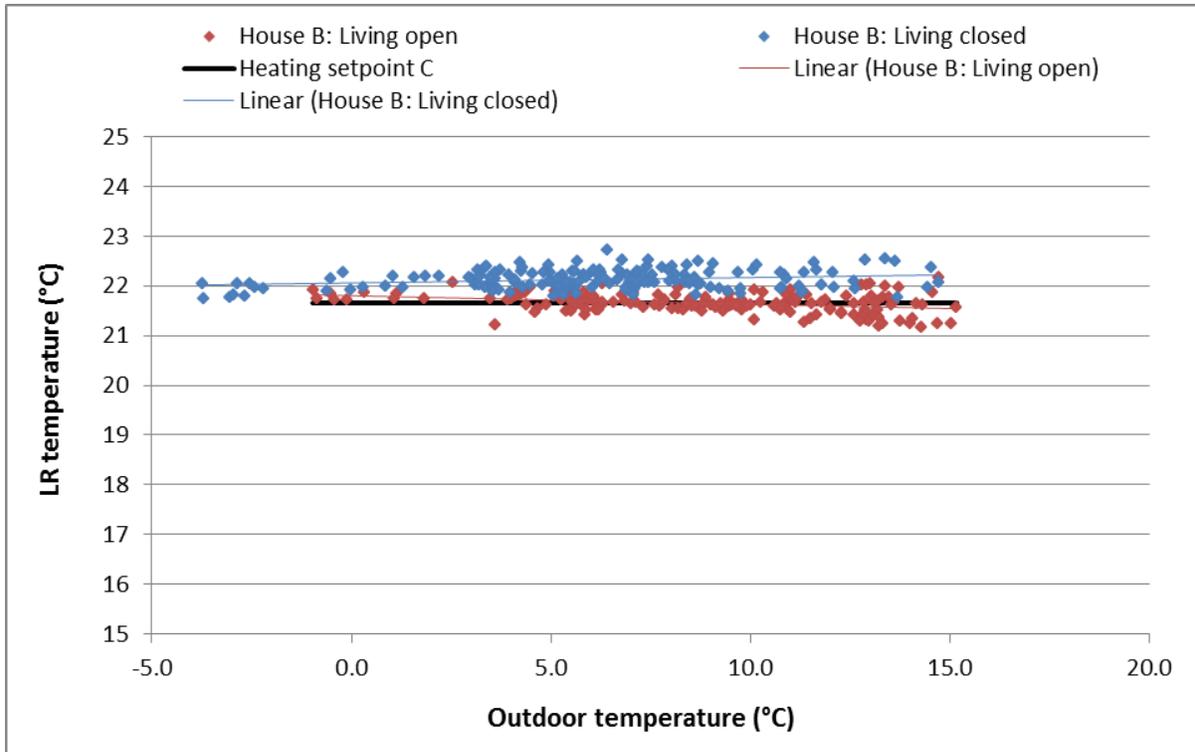


Figure 75. House B impacts of door position on living room temperature compared to outdoor temperature (heating season)

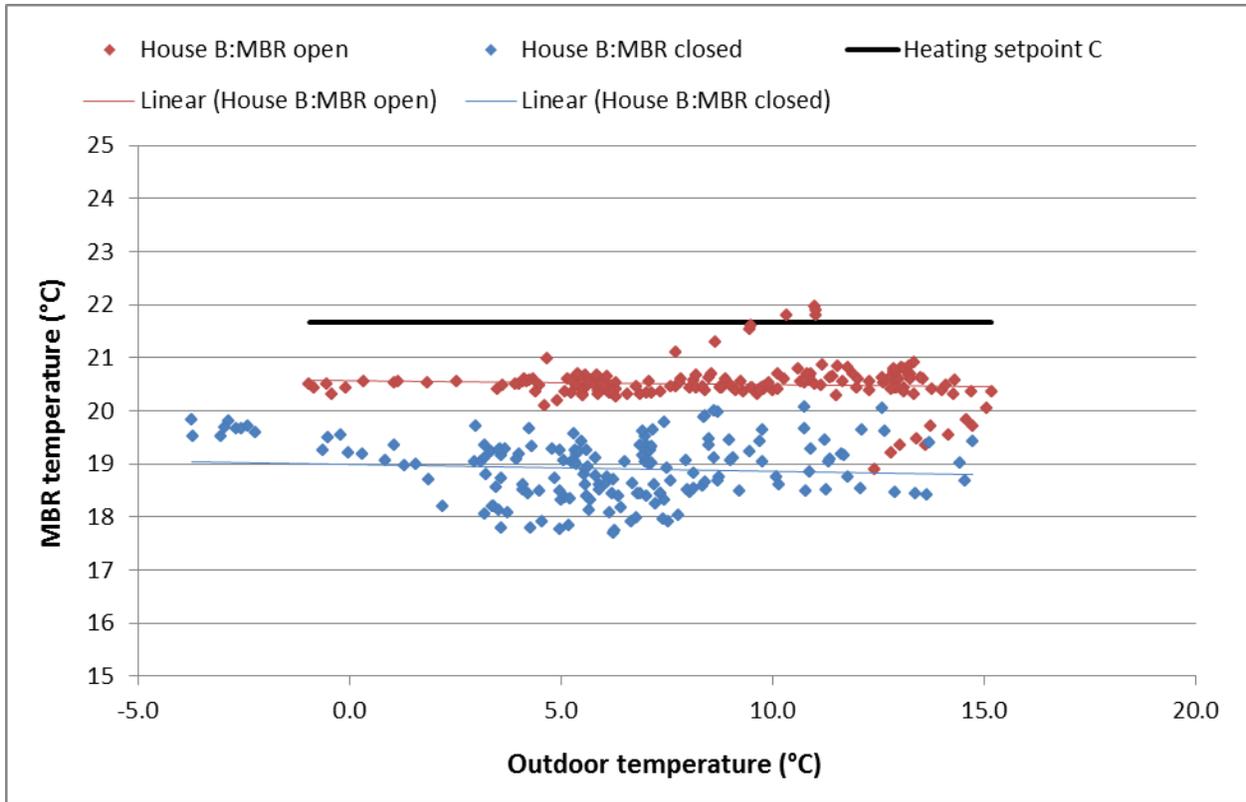


Figure 76. House B impacts of door position on master bedroom temperature compared to outdoor temperature (heating season)

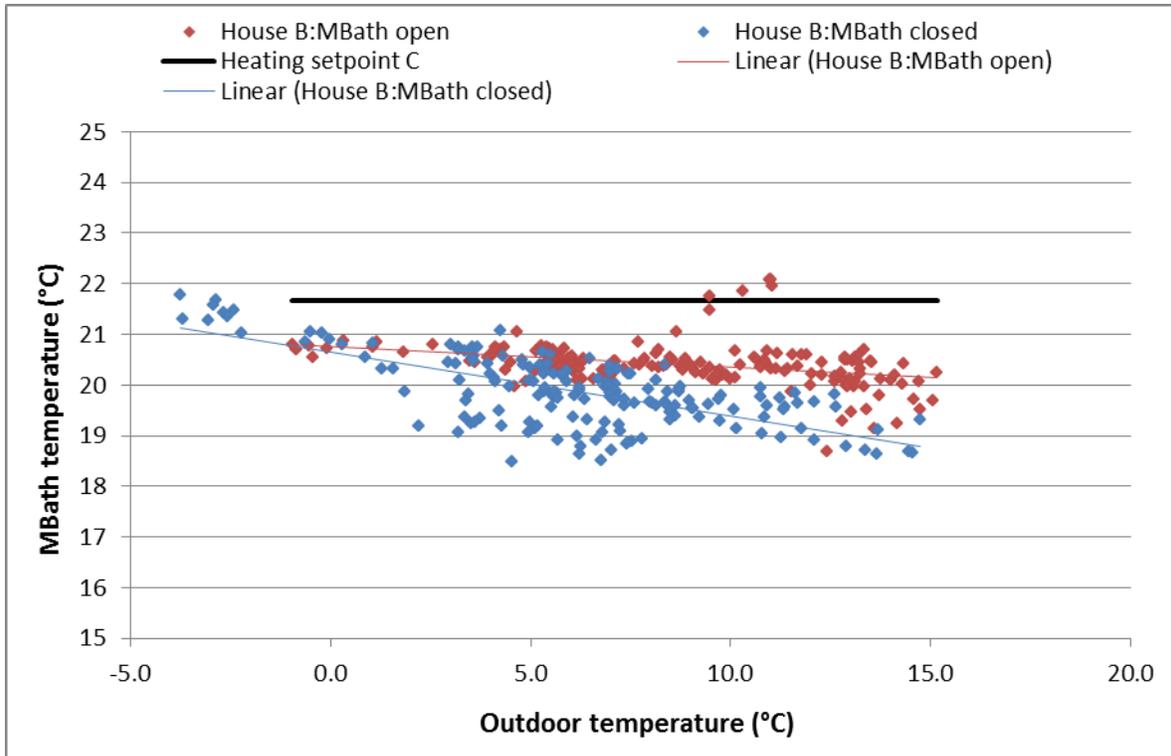


Figure 77. House B impacts of door position on master bathroom temperature compared to outdoor temperature (heating season)

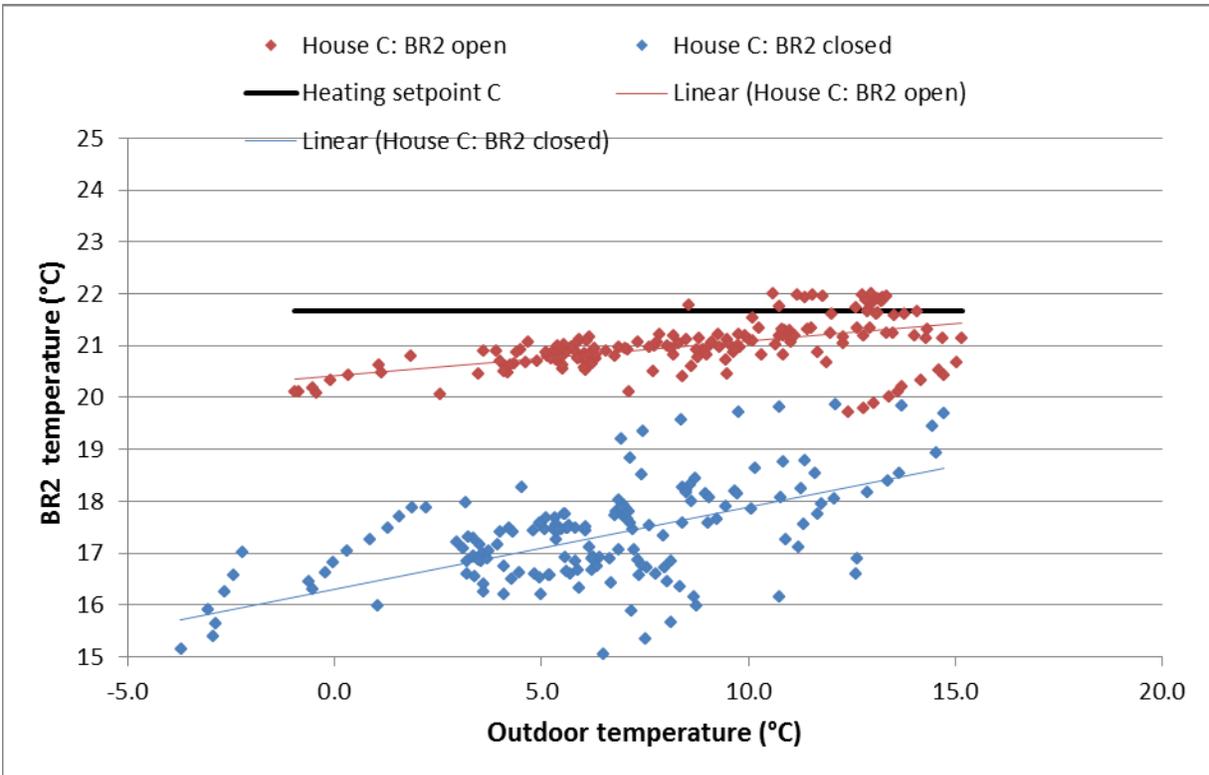


Figure 78. House C impacts of door position on Bedroom 2 temperature compared to outdoor temperature (heating season)

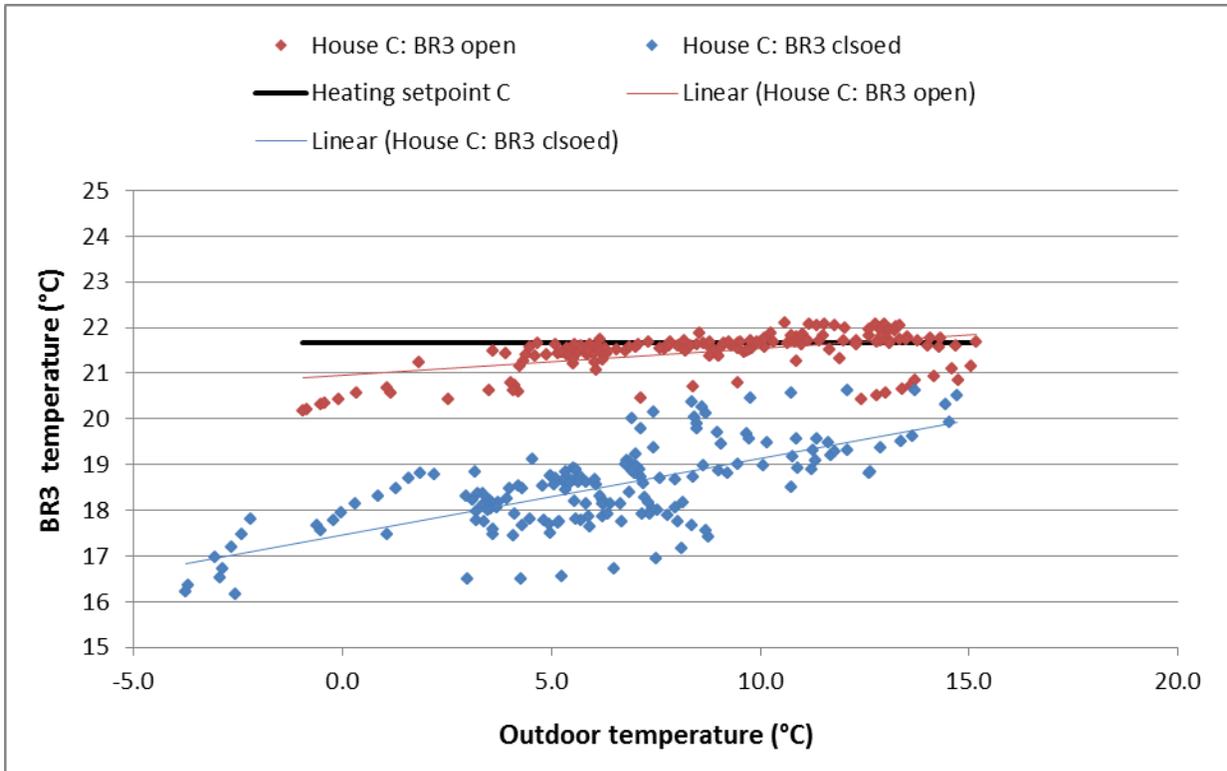


Figure 79. House C impacts of door position on Bedroom 3 temperature compared to outdoor temperature (heating season)

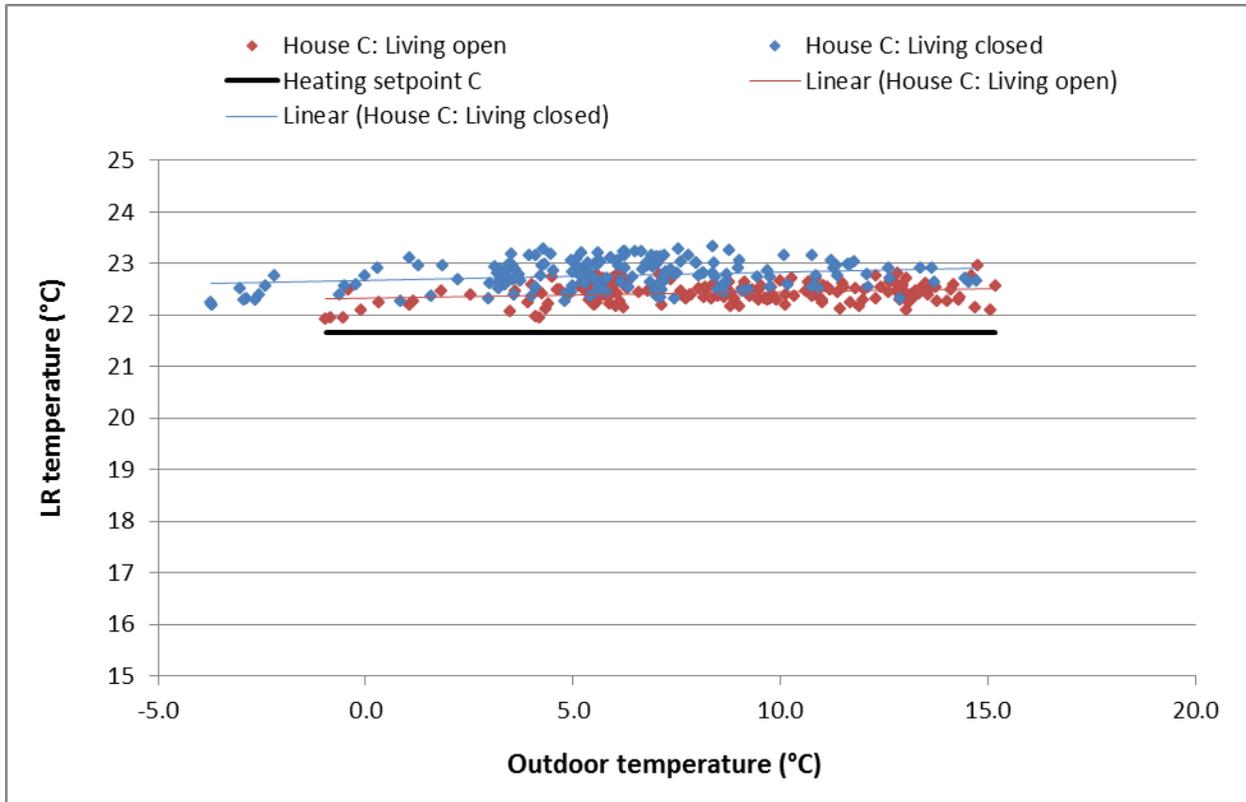


Figure 80. House C impacts of door position on living room temperature compared to outdoor temperature (heating season)

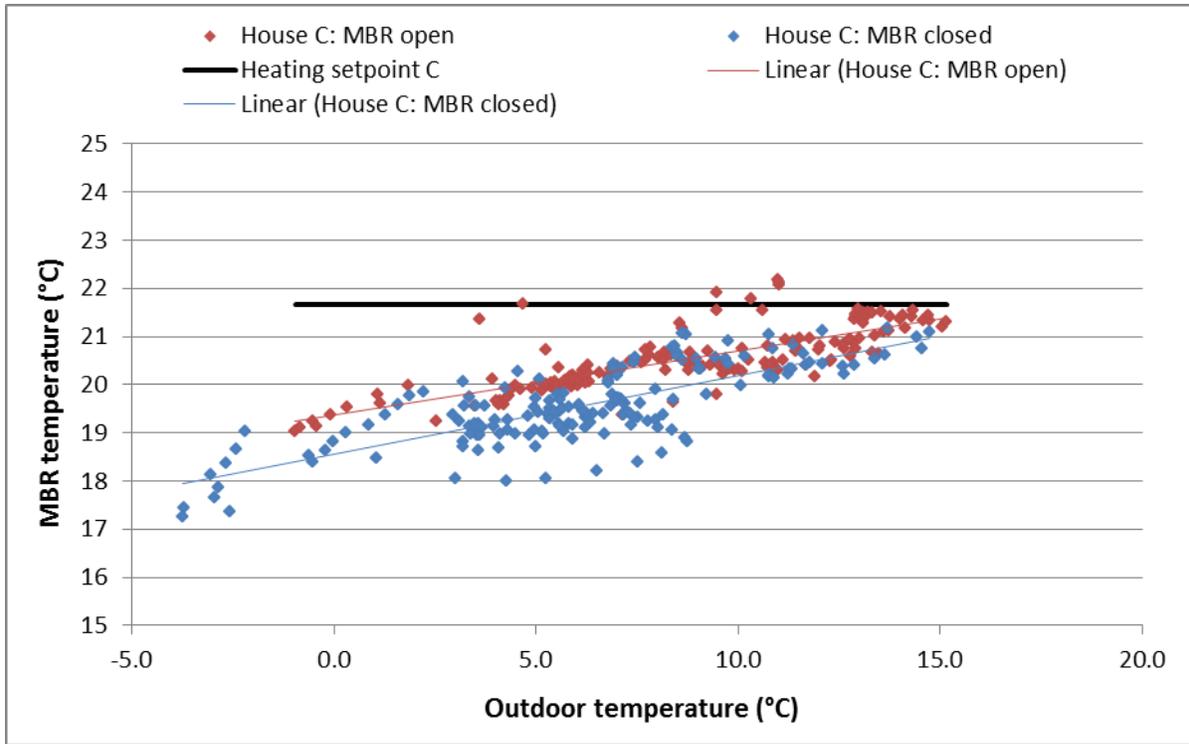


Figure 81. House C impacts of door position on master bedroom temperature compared to outdoor temperature (heating season)

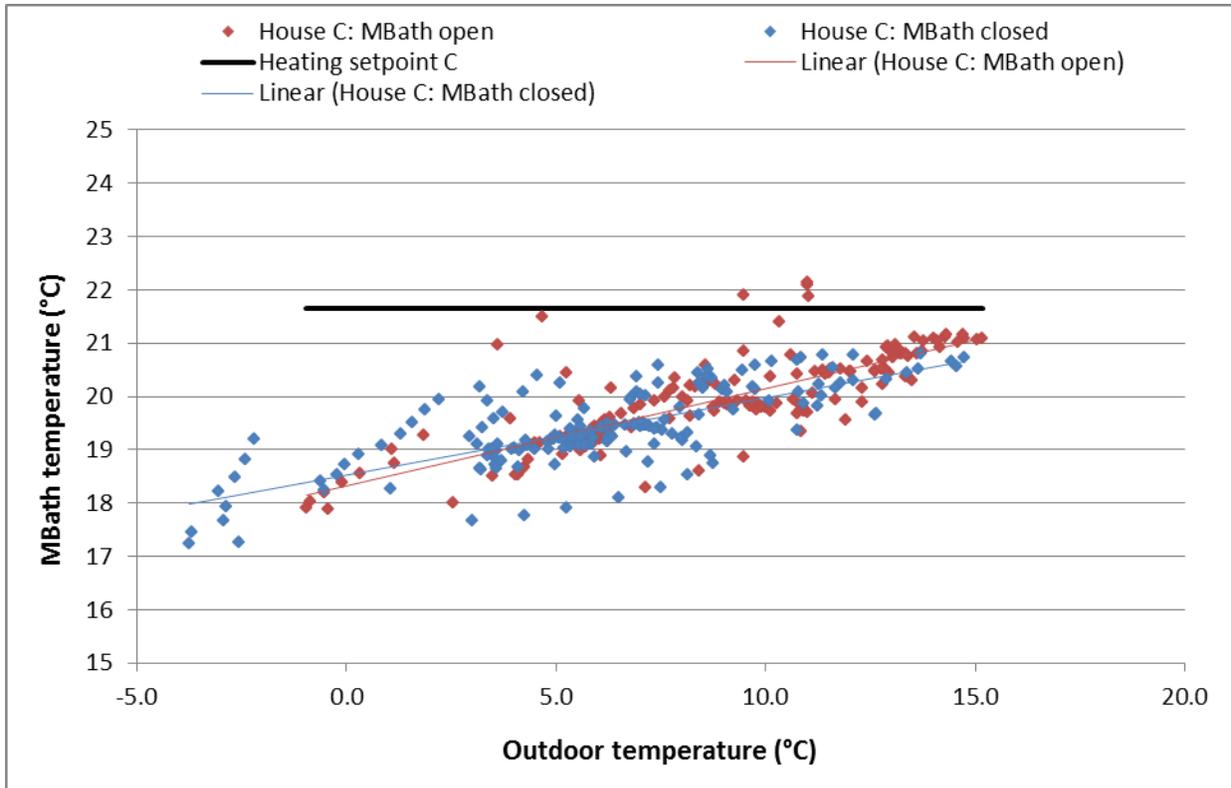


Figure 82. House C impacts of door position on master bathroom temperature compared to outdoor temperature (heating season)

Appendix F: Tracer Gas Test Data

Table 50. Tracer Gas Test Data

2014	Hour	House C		House B		House A		Notes
		ppm	ACH	ppm	ACH	ppm	ACH	
Dec. 8	17	3.01		3.06				Normal operation of HVAC in all three houses. House C exhaust fan off
	18	2.62	0.14	2.51	0.20			
	19	2.31	0.13	2.07	0.19			
	20	2.03	0.13	1.64	0.23			
	21	1.81	0.11	1.34	0.20			
	22	1.61	0.12	1.1	0.20			
	23	1.43	0.12	0.867	0.24			
	0	1.27	0.12	0.696	0.22			
Dec. 9	1	1.11	0.13	0.553	0.23			
	2	0.972	0.13	0.443	0.22			
	3	0.851	0.13	0.339	0.27			
	4	0.743	0.14	0.273	0.22			
	5	0.655	0.13	0.214	0.24			
	6	0.573	0.13	0.171	0.22			
	7	0.493	0.15	0.145	0.16			
	8	0.409	0.19					House C exhaust fan on starting at 7:30 a.m.
	9	0.307	0.29					House C exhaust fan on
	10	0.235	0.27					House C exhaust fan on
	11							House C exhaust fan off until Dec 11
	12							
	13							
	14							
	15							
	16							

	House C		House B		House A		Notes
	17		3.09				
	18	3.1	2.44	0.24			Co-heating in all three houses
	19	2.68	0.15	1.91	0.24		
	20	2.31	0.15	1.5	0.24		
	21	2.06	0.11	1.2	0.22		
	22	1.73	0.17	0.953	0.23		
	23	1.51	0.14	0.759	0.23		
	0	1.31	0.14	0.609	0.22		
Dec. 10	1	1.15	0.13	0.489	0.22		
	2	0.971	0.17	0.39	0.23		
	3	0.84	0.14	0.307	0.24		
	4	0.718	0.16	0.242	0.24		
	5	0.621	0.15	0.197	0.21		
	6	0.535	0.15	0.16	0.21		
	7	0.468	0.13	0.13	0.21		
	8	0.407	0.14	0.11	0.17		
	9	0.355	0.14				
	10	0.315	0.12				
	11	0.268	0.16				
	12	0.236	0.13				
	13						
	14						
	15						
	16						
	17		3.42		3.95		
	18		2.68	0.24	3.08	0.25	Co-heating in all three houses
	19		2.11	0.24	2.41	0.25	
	20		1.65	0.25	1.87	0.25	
	21		1.3	0.24	1.46	0.25	

	House C	House B	House A	Notes
Dec. 11	22	1.01	0.25	1.12 0.27
	23	0.784	0.25	0.872 0.25
	0	0.607	0.26	0.675 0.26
	1	0.477	0.24	0.525 0.25
	2	0.371	0.25	0.408 0.25
	3	0.292	0.24	0.321 0.24
	4	0.23	0.24	0.252 0.24
	5	0.183	0.23	0.199 0.24
	6	0.147	0.22	0.151 0.28
	7	0.117	0.23	0.121 0.22
	8			
	9			
	10			
	11			
	12			
	13			
	14			
15				
16				
17				Normal operation of HVAC in all three houses. House C exhaust fan on
18	3.21		2.82	
19	2.51	0.25	2.23 0.23	
20	1.85	0.31	1.75 0.24	
21	1.43	0.26	1.34 0.27	
22	1.15	0.22	1.06 0.23	
23	0.859	0.29	0.815 0.26	
0	0.658	0.27	0.637 0.25	
Dec. 12	1	0.504	0.27	0.502 0.24
	2	0.384	0.27	0.392 0.25

	House C	House B	House A	Notes
3	0.299	0.25	0.299	0.27
4	0.233	0.25	0.236	0.24
5	0.159	0.38	0.182	0.26
6	0.126	0.23	0.148	0.21
7	0.109			

Appendix G: Co-Heat Testing Data

Table 51. House A Co-Heat Testing Data

Time Stamp	Outdoor Temp.	Avg. Indoor Temp.	Sensible Gain Heater	Sensible Gain Heater	Main Panel	Outdoor Unit	Air Handling Unit Fan	Total Internal Gain	Calculated Simple UA
	(°C)	(°C)	(kW)	(kW)	(kW)	(kW)	(kW)	(kW)	(kW/°C)
12/10/2014 2:00	3.1	23.0	0.304	0.197	3.467	0.011	0.001	3.467	0.174
12/10/2014 3:00	3.2	23.0	0.297	0.197	3.435	0.011	0.002	3.435	0.174
12/10/2014 4:00	3.2	23.0	0.283	0.197	3.417	0.011	0.002	3.417	0.172
12/10/2014 5:00	3.2	23.0	0.281	0.201	3.405	0.011	0.001	3.405	0.172
12/10/2014 6:00	3.3	23.0	0.300	0.209	3.268	0.011	0.001	3.268	0.166
12/11/2014 2:00	-1.7	23.0	0.001	0.002	3.933	0.000	0.001	3.933	0.159
12/11/2014 3:00	-1.3	23.0	0.001	0.002	3.934	0.000	0.001	3.934	0.162
12/11/2014 4:00	-1.9	23.0	0.001	0.002	3.936	0.000	0.001	3.936	0.158
12/11/2014 5:00	-2.8	23.0	0.001	0.002	4.043	0.000	0.001	4.043	0.156
12/11/2014 6:00	-3.1	23.0	0.001	0.002	4.037	0.000	0.001	4.037	0.154

Table 52. House B Co-Heat Testing Data

Time Stamp	Outdoor Temp.	Avg. Indoor Temp.	Sensible Gain Heater	Sensible Gain Heater	Main Panel	Outdoor Unit	Air Hand-ling Unit Fan	Total Internal Gain	Calculated Simple UA
	(°C)	(°C)	(kW)	(kW)	(kW)	(kW)	(kW)	(kW)	(kW/°C)
12/10/2014 2:00	3.1	23.0	0.301	0.197	2.617	0.006	0.008	2.617	0.132
12/10/2014 3:00	3.2	23.0	0.292	0.197	2.680	0.006	0.008	2.680	0.135
12/10/2014 4:00	3.2	23.0	0.281	0.196	2.603	0.006	0.008	2.603	0.132
12/10/2014 5:00	3.2	23.0	0.279	0.199	2.603	0.006	0.008	2.603	0.132
12/10/2014 6:00	3.3	23.0	0.298	0.208	2.562	0.006	0.008	2.562	0.130
12/11/2014 2:00	-1.7	23.0	0.000	0.002	3.128	0.007	0.008	3.128	0.126
12/11/2014 3:00	-1.3	23.0	0.000	0.002	3.144	0.007	0.008	3.144	0.130
12/11/2014 4:00	-1.9	23.0	0.000	0.002	3.147	0.007	0.008	3.147	0.126
12/11/2014 5:00	-2.8	23.0	0.000	0.002	3.186	0.007	0.008	3.186	0.123
12/11/2014 6:00	-3.1	23.0	0.000	0.002	3.169	0.007	0.008	3.169	0.121

Table 53. House C Co-Heat Testing Data

Time Stamp	Outdoor Temp.	Avg. Indoor Temp.	Sensible Gain Heater	Sensible Gain Heater	Main Panel	Outdoor Unit	Air Handling Unit Fan	Simple UA Heat Loss	Total Internal Gain	Calculated Simple UA
	(°C)	(°C)	(kW)	(kW)	(kW)	(kW)	(kW)	(kW)	(kW)	(kW/°C)
12/10/2014 2:00	3.1	23.0	0.287	0.196	2.158	0.024	0.004	2.184	2.134	0.107
12/10/2014 3:00	3.2	23.0	0.277	0.196	2.144	0.025	0.004	2.165	2.119	0.107
12/10/2014 4:00	3.2	23.0	0.267	0.196	2.128	0.024	0.004	2.171	2.104	0.106
12/10/2014 5:00	3.2	23.0	0.264	0.200	2.147	0.022	0.004	2.171	2.125	0.107
12/10/2014 6:00	3.3	23.0	0.283	0.208	2.162	0.022	0.004	2.165	2.140	0.108
12/11/2014 2:00	-1.7	23.0	0.001	0.001	2.565	0.032	0.004	2.713	2.533	0.110
12/11/2014 3:00	-1.3	23.1	0.001	0.001	2.633	0.037	0.004	2.663	2.596	0.113
12/11/2014 4:00	-1.9	23.0	0.001	0.001	2.647	0.035	0.004	2.731	2.612	0.113
12/11/2014 5:00	-2.8	23.0	0.001	0.001	2.590	0.032	0.004	2.833	2.558	0.111
12/11/2014 6:00	-3.1	23.0	0.001	0.001	2.624	0.030	0.004	2.867	2.594	0.113

Appendix H: Impacts of Window Blind Position

Table 54. Data Used for Multi-Variable Regression—Blinds Closed Period (Sept. 23–29)

	Avg. Temp. (°C)	Delta T (T _{indoor} - T _{outdoor}) House A (°C)	Delta T (T _{indoor} - T _{outdoor}) House B (°C)	Delta T (T _{indoor} - T _{outdoor}) House C (°C)	Direct Normal Radiation Total (Wh/m ²)	Total Cooling Energy House A (kWh)	Total Cooling Energy House B (kWh)	Total Cooling Energy House C (kWh)
Tue. 09/23/14	17.30	-5.9	-6.9	-5.1	5,895.29	4.03	3.37	2.63
Wed. 09/24/14	18.50	-5.1	-6.0	-4.1	5,665.66	4.97	4.23	3.10
Thu. 09/25/14	20.15	-3.7	-4.6	-2.7	4,716.51	6.49	5.46	3.70
Fri. 09/26/14	20.17	-3.8	-4.6	-2.6	1,619.95	4.76	4.57	3.40
Sat. 09/27/14	19.50	-4.5	-5.3	-3.3	864.44	3.79	3.82	2.82
Sun. 09/28/14	22.05	-2.2	-2.9	-0.7	1,044.04	5.03	5.17	3.78
Mon. 09/29/14	21.10	-3.3	-4.0	-2.1	4,535.68	8.29	7.08	4.73

Table 55. Predicted Cooling Energy Calculation with Normalized Data—Blinds Open Period (Aug. 17–23)

	Avg. Temp. (°C)	Direct Normal Radiation Total (Wh/m ²)	Predicted Total Cooling Energy House A- if blinds were closed (kWh)	Predicted Total Cooling Energy House B- blinds closed period (kWh)	Predicted Total Cooling Energy House C (kWh)
Sun. 08/17/14	24.07	1,075.37	8.21	7.77	5.09
Mon. 08/18/14	24.13	1,936.92	8.69	8.02	5.10
Tue. 08/19/14	25.99	2,874.63	11.57	10.28	6.39

	Avg. Temp. (°C)	Direct Normal Radiation Total (Wh/m ²)	Predicted Total Cooling Energy House A- if blinds were closed (kWh)	Predicted Total Cooling Energy House B- blinds closed period (kWh)	Predicted Total Cooling Energy House C (kWh)
Wed. 08/20/14	26.91	3,969.03	13.25	11.52	7.04
Thu. 08/21/14	26.85	3,798.98	13.06	11.39	6.99
Fri. 08/22/14	26.74	4,678.43	13.43	11.62	7.11
Sat. 08/23/14	27.20	4,043.29	13.61	11.83	7.20
		Total cooling (kWh)	81.83	72.42	44.91

Table 56. Window Blinds Regression

SUMMARY OUTPUT House A

Regression Statistics

Multiple R	0.848477
R Square	0.719914
Adjusted R Square	0.579871
Standard Error	1.014776
Observations	7

ANOVA

	df	SS	MS	F	Significance F
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SUMMARY OUTPUT House A

Regression	2	10.5874	5.2937	5.1406	0.078448			
				64				
Residual	4	4.119078	1.0297					
			7					
Total	6	14.70648						
	Coefficients	Standard error	t Stat	P-value	Lower 95%	Upper 95%	Lower 95.0%	Upper 95.0%
Intercept	8.275608	1.449644	5.7087	0.0046	4.250751	12.30047	4.250751	12.30047
			17	56				
Delta T House A	1.241755	0.415736	2.9868	0.0404	0.087486	2.396025	0.087486	2.396025
			81	63				
Direct Normal Radiation Total (W/m²)	0.000609	0.000228	2.6681	0.0559	-2.5E-05	0.001242	-2.5E-05	0.001242
			73	12				

SUMMARY OUTPUT House B

Regression Statistics

Multiple R	0.887946
R Square	0.788448
Adjusted R Square	0.682672
Standard Error	0.696203
Observations	7

ANOVA										
	df	SS	MS	F	Significance F					
SUMMARY OUTPUT House B										
Regression	2	7.225825	3.612912	7.453945	0.044754					
Residual	4	1.938792	0.484698							
Total	6	9.164617								
	Coefficients	Standard error	t Stat	P-value	Lower 95%	Upper 95%	Lower 95.0%	Upper 95.0%		
Intercept	8.543596	1.113623	7.671891	0.001552	5.451682	11.63551	5.451682	11.63551		
Delta T House B	1.025916	0.265947	3.857594	0.018184	0.287528	1.764303	0.287528	1.764303		
Direct Normal Radiation Total (W/m²)	0.000376	0.000158	2.384163	0.075646	-6.2E-05	0.000815	-6.2E-05	0.000815		
SUMMARY OUTPUT House C										
Regression Statistics										
Multiple R	0.893938									
R Square	0.799126									
Adjusted R Square	0.698688									

Standard Error	0.388353
Observations	7

SUMMARY OUTPUT House C

ANOVA								
	df	SS	MS	F	Significance F			
Regression	2	2.399962	1.199981	7.956467	0.040351			
Residual	4	0.603273	0.150818					
Total	6	3.003236						
	Coefficients	Standard error	t Stat	P-value	Lower 95%	Upper 95%	Lower 95.0%	Upper 95.0%
Intercept	4.386133	0.369196	11.88024	0.000287	3.361082	5.411185	3.361082	5.411185
Delta T House C	0.577592	0.144814	3.988512	0.016285	0.175524	0.979659	0.175524	0.979659
Direct Normal Radiation Total (W/m²)	0.00022	9.1E-05	2.413245	0.073297	-3.3E-05	0.000472	-3.3E-05	0.000472

Appendix I: Impacts of Controlled Heater Energy

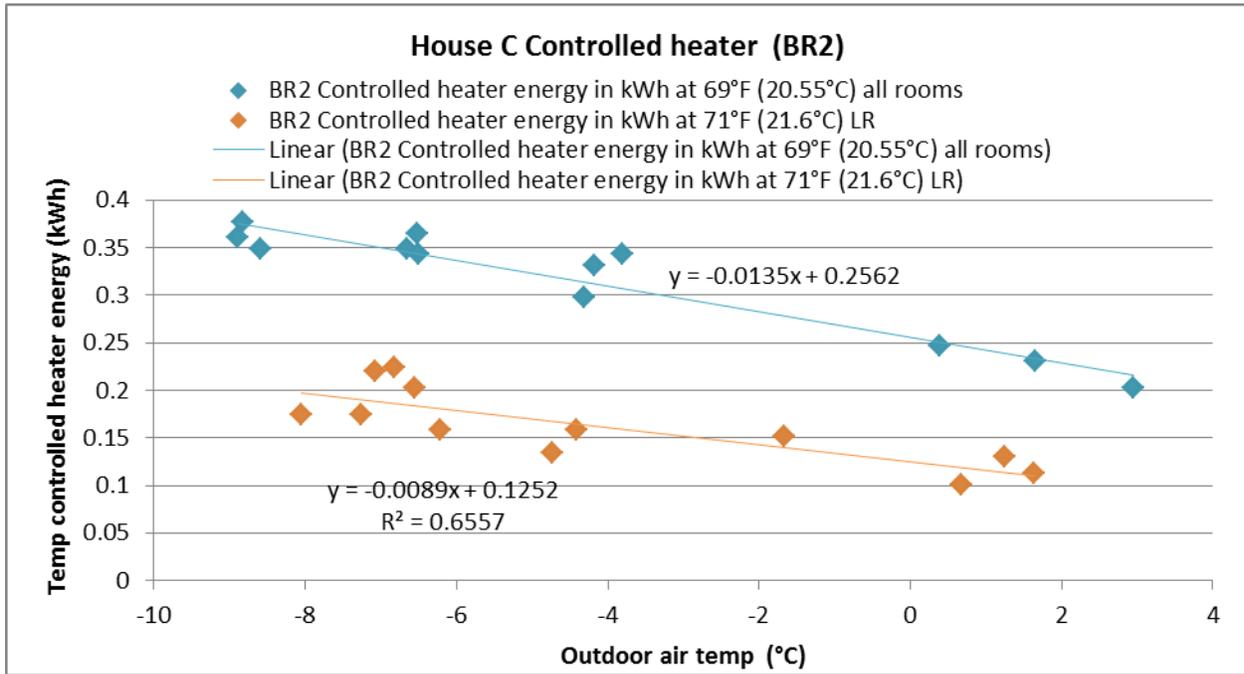


Figure 83. House C Bedroom 2 impacts of controlled heater energy

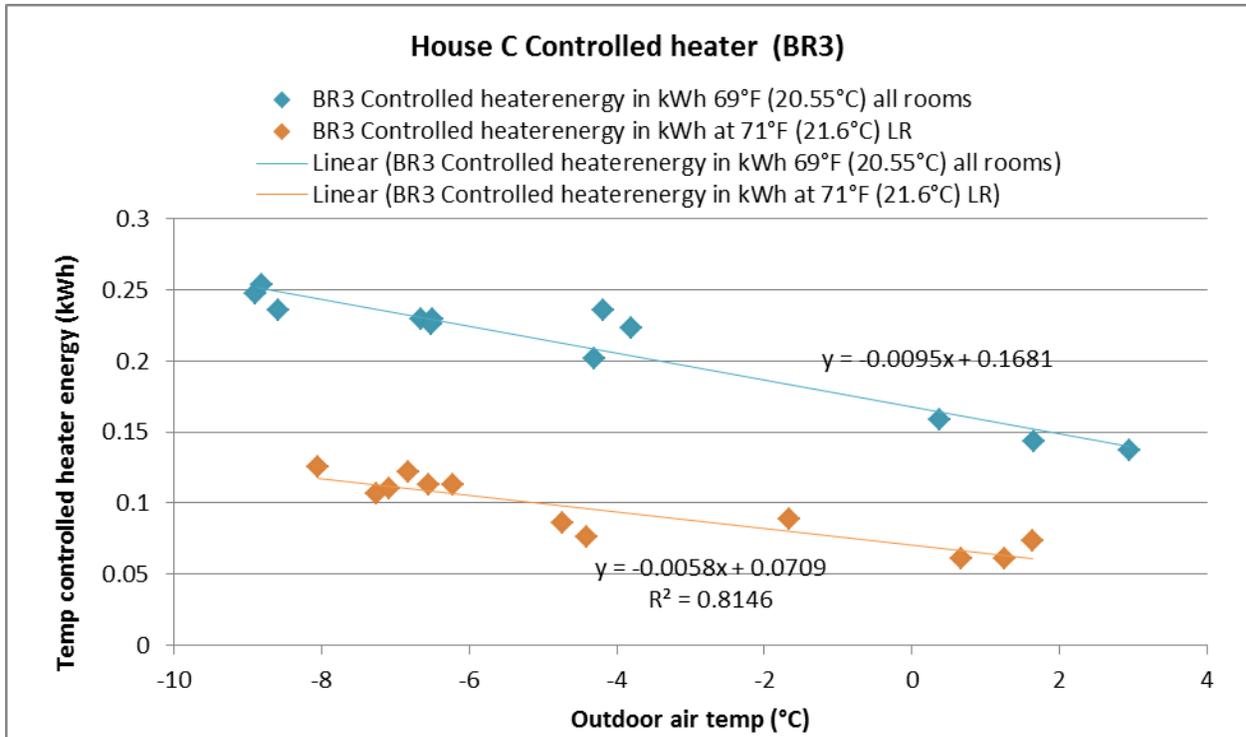


Figure 84. House C Bedroom 3 of controlled heater energy

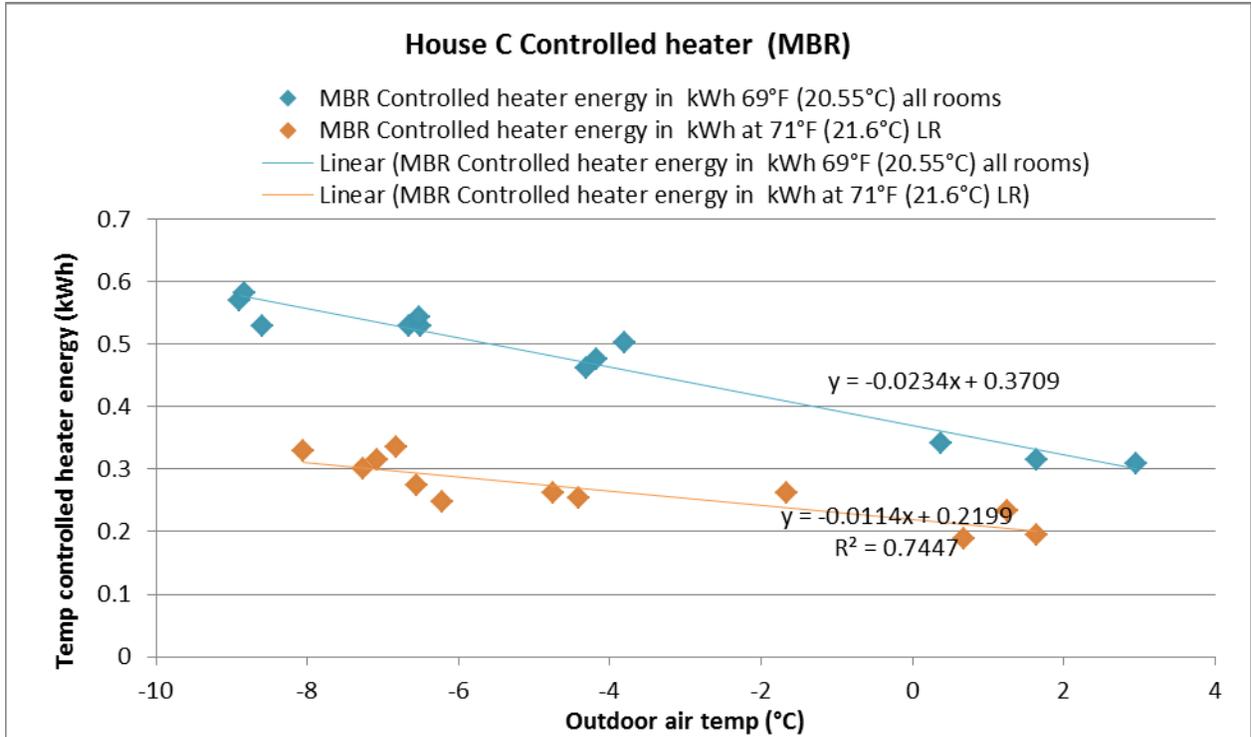


Figure 85. House C master bedroom impacts of controlled heater energy

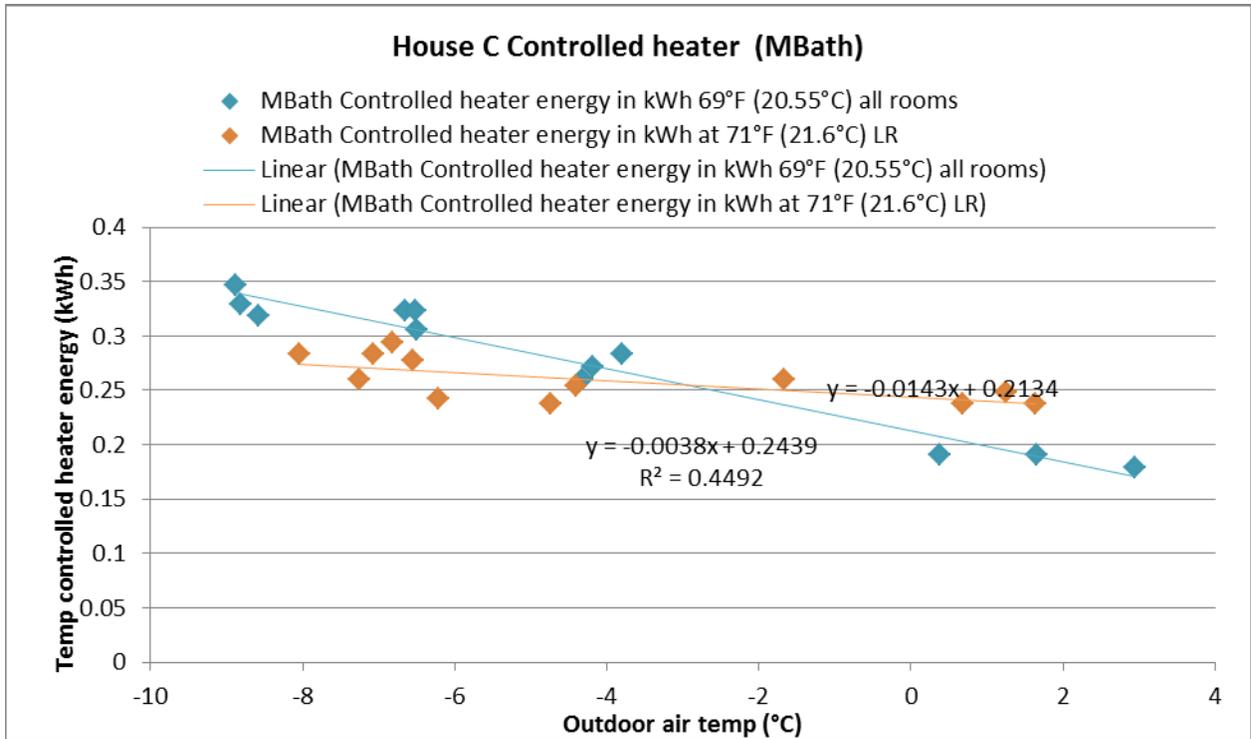


Figure 86. House C master bathroom impacts of controlled heater energy

Appendix J: Construction Drawings

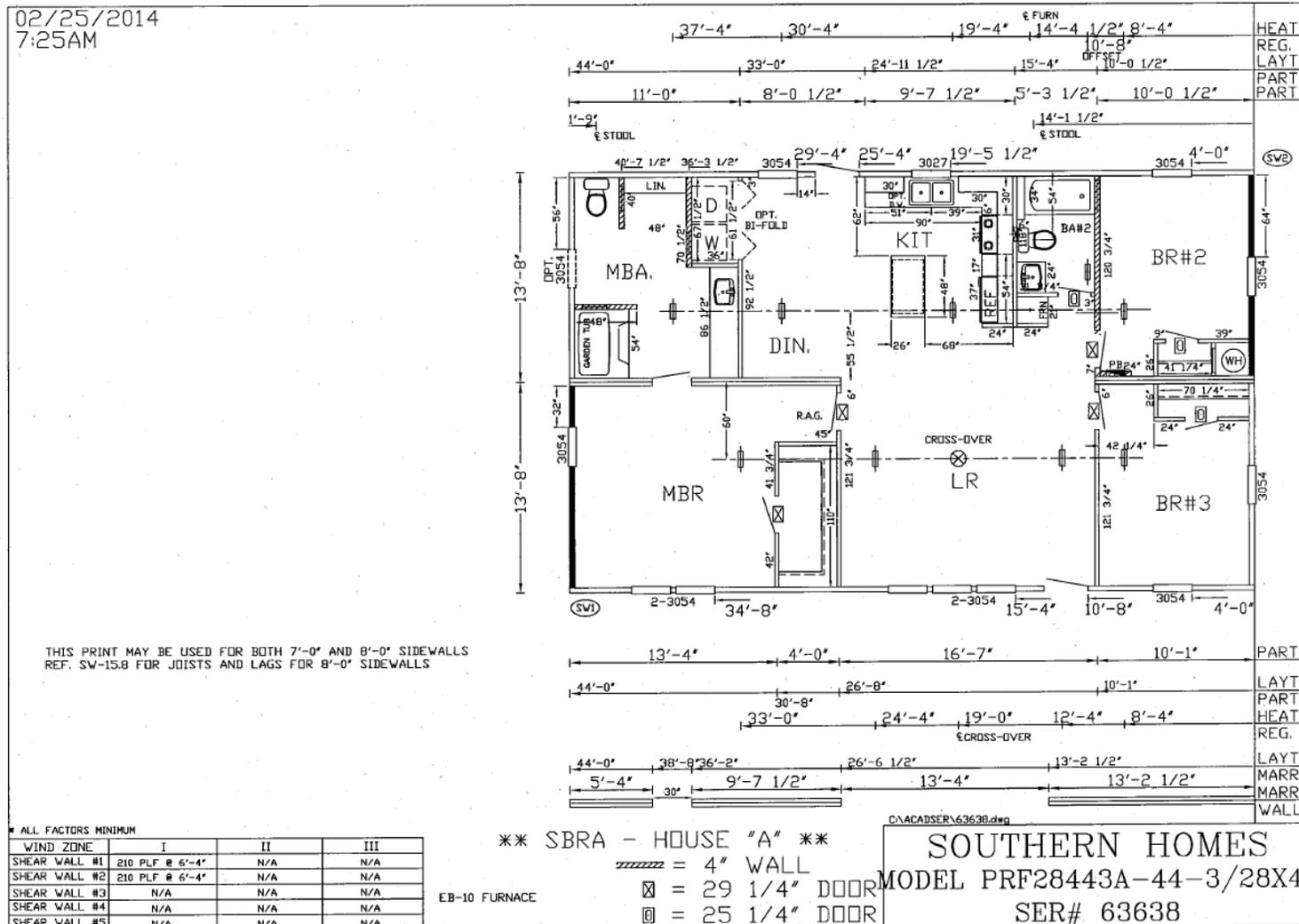


Figure 87. House A production print (House B is identical.)

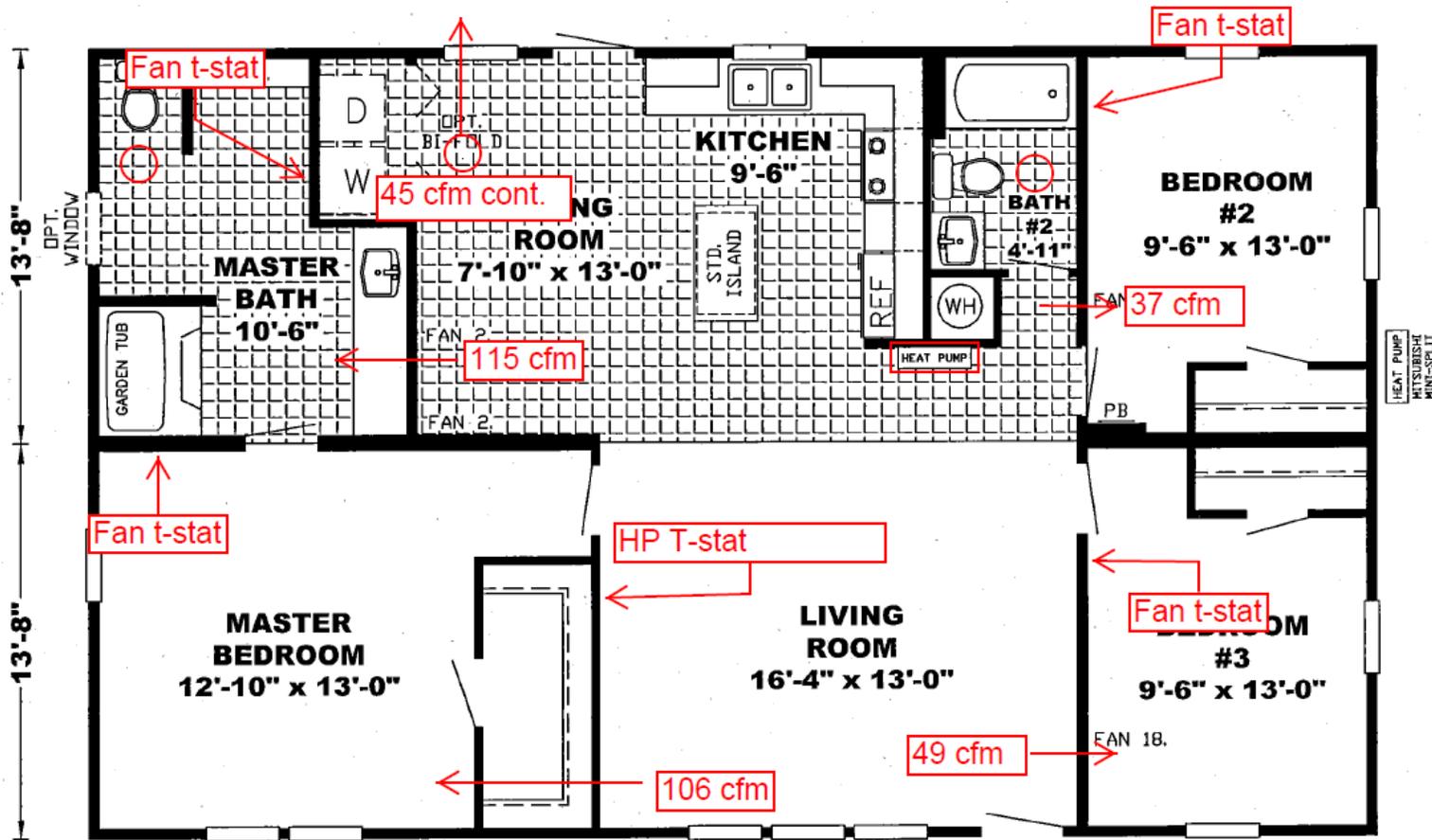


Figure 88. House C mechanical system schematic plan

Wall and Roof Insulation Details (House C)

Figure 89 through Figure 93 show construction details indicating how the exterior continuous insulation was incorporated into the wall assembly.

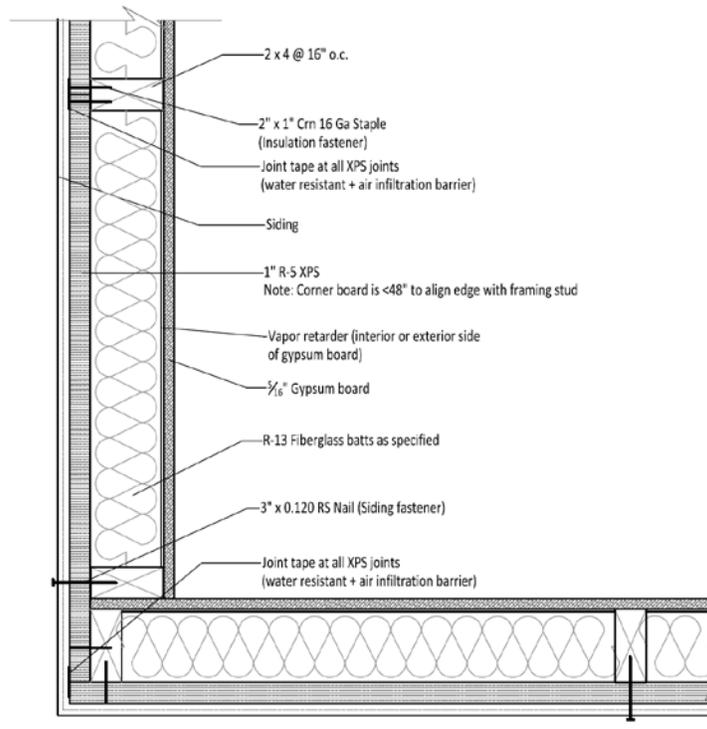


Figure 89. Plan view of wall

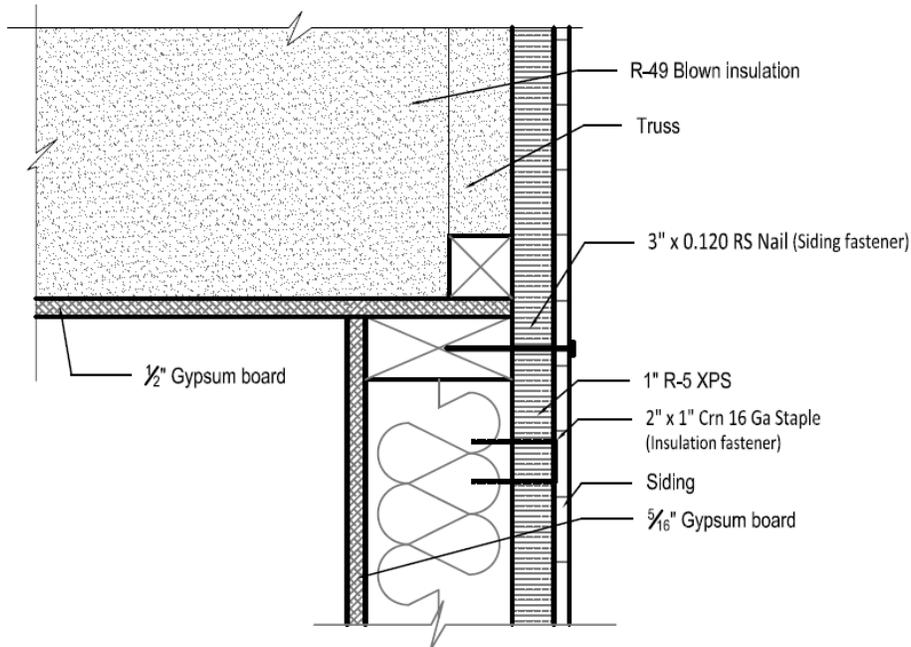


Figure 90. Top plate (gable wall section)

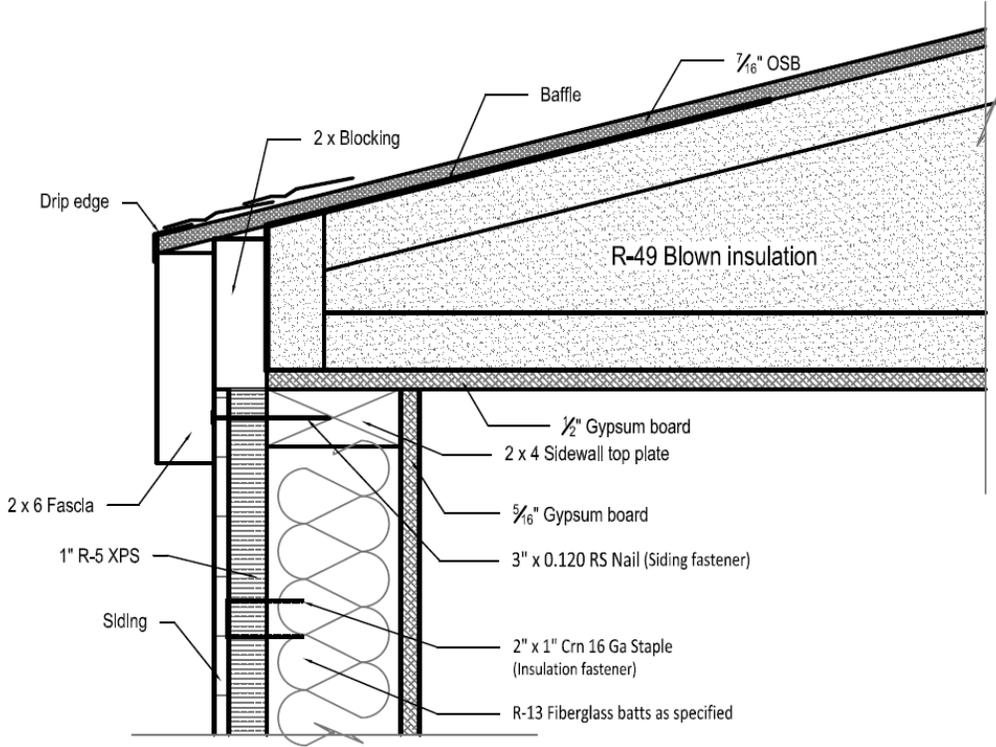


Figure 91. Roof-to-wall connection (side wall section)

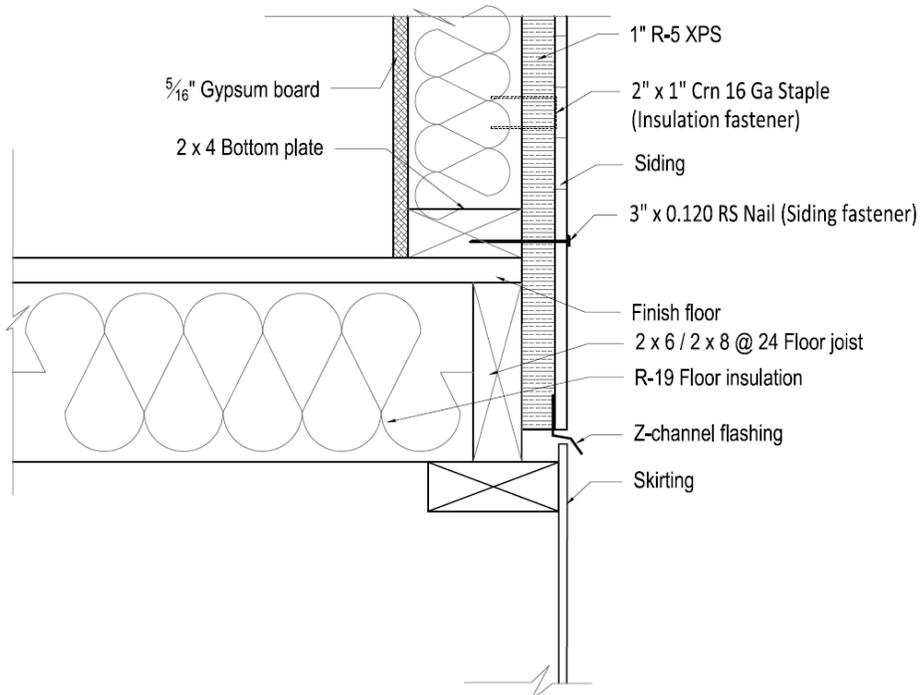


Figure 92. Floor-to-wall connection (side wall section)

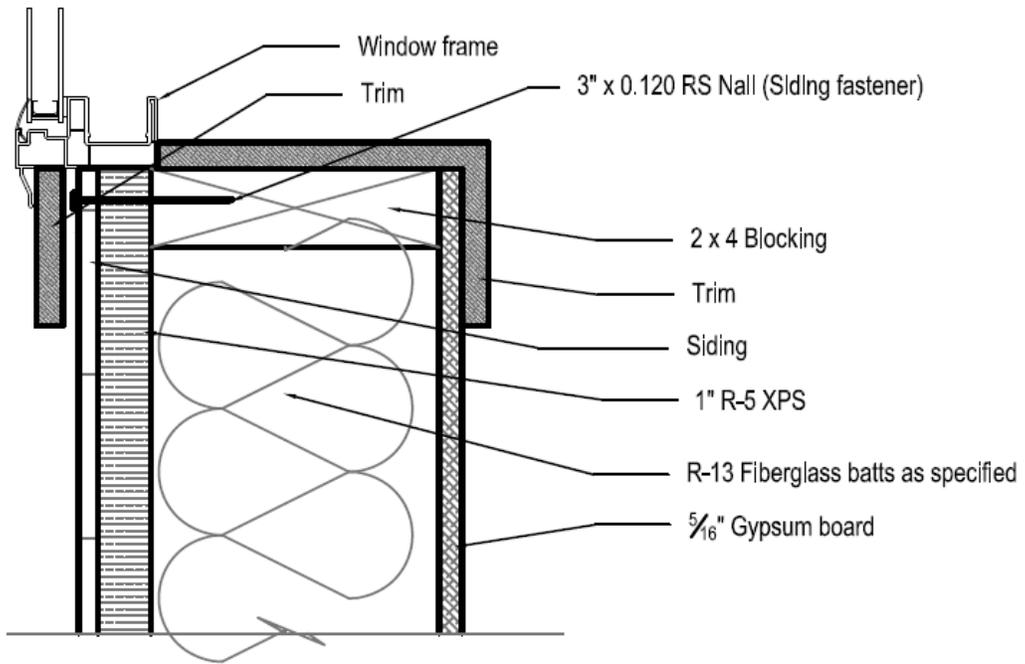


Figure 93. Window sill

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