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Simplified Space Conditioning in Low-Load Homes: Results from Pittsburgh, Pennsylvania, New Construction Unoccupied Test House

Andrew Poerschke and Dave Stecher
IBACOS, Inc.

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Definitions

ACCA	Air Conditioning Contractors of America
ACH	Air changes per hour
AHU	Air handler unit
BEopt™	Building Energy Optimization (software)
Btu/h	British thermal units per hour
CFM	Cubic feet per minute
COP	Coefficient of performance
EER	Energy efficiency ratio
EF	Energy factor
ERV	Energy recovery ventilator
GSHP	Ground source heat pump
HVAC	Heating, ventilation, and air conditioning
PV	Photovoltaic

Executive Summary

IBACOS anticipates that houses achieving 50% whole-house source energy savings with respect to the Building America 2010 Benchmark (Hendron and Engebrecht 2010a) will be considered "low load." Low load is defined by IBACOS as a house with a thermal enclosure that yields a maximum space heating and cooling load of less than 10 Btu/h/ft^2 of conditioned floor area (31.5 W/m^2 [1,200 ft^2/ton]). These small loads can be met by systems other than today's typical ducted forced-air systems. For example, distributed fan coils with minimized ducts, terminal fan coil units, or point source units with buoyant force or ventilation-driven distribution may provide sufficient occupant comfort in a low-load home. These systems, which can have lower total installed costs than traditional ducted forced-air systems (Stecher 2011), allow the thermal enclosure characteristics of low-load houses to provide first-cost savings in addition to operational cost savings.

The purpose of this study is to help determine cost-effective solutions for heating and cooling houses that are designed to be energy efficient. This is done by testing the occupant comfort performance of some concepts that may already exist on the market but are not in use by production homebuilders. In some cases, the products are market available, but their use in housing may be a new application. The standards used to assess the performance of the systems in this study are Air Conditioning Contractors of America (ACCA) Manual RS (Rutkowski 1997) and ASHRAE Standard 55-2010 (ASHRAE 2010a).

This report addresses the following two research questions:

- To what extent do alternative space conditioning distribution strategies meet ACCA and ASHRAE guidelines for room-to-room temperature uniformity and stability, respectively? Under what weather conditions do indoor temperatures rise above or fall below the guidelines?
- How substantially do the alternative strategies differ from typical ducted forced-air systems in their ability to meet the ACCA and ASHRAE guidelines?

To address these questions, IBACOS performed energy modeling and created two low-load test facilities with instrumentation to enable the testing of several experimental alternatives to traditional forced-air distribution designs. One facility is a retrofit unoccupied test house in Fresno, California (Stecher and Poerschke 2013); the other is a new construction unoccupied test house in Pittsburgh, Pennsylvania. Several systems were tested in each house. This report outlines the results of the four systems tested in the Pittsburgh test house:

- Typical airflow volume ducted distribution system routed to all spaces requiring conditioning, with two thermostatically controlled zones—one for the first floor and one for the second floor—with ducts sized to meet peak system airflow output.
- Low airflow volume ducted distribution system to the bedrooms (19 to 23 CFM each) and a single point of delivery to the main living space; one thermostatically controlled zone for the bedrooms and one for the main living space.

- Over-door powered transfer fans to connect the bedrooms to the upstairs hallway, which is served by a single register on its own thermostat; a single register with its own thermostat served the first floor.
- No active systems—only open doors—to thermally connect the bedrooms to the upstairs hallway, which is served by a single register on its own thermostat; a single register with its own thermostat served the first floor.

The measured performance of the four systems in the Pittsburgh test house with respect to ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time) (ASHRAE 2010a) showed that failures according to that standard for all four systems tended to be cyclic in nature. The frequency of failure—that is, the room temperature change over time that is outside the bounds of the standard—and the rooms that failed varied from system to system. Room temperature change of greater than 2°F over durations of time greater than 15 min were rare in most cases; however, it was observed that these temperature changes were a result of specific bedrooms having a different load profile than the rest of the house. The transfer fan system was most successful in the bedrooms at the expense of frequent cyclic failure in the upstairs landing and living room, whereas the typical and low airflow systems were most successful at maintaining a high passing rate in all rooms throughout the house.

During heating mode, all systems failed to meet ACCA Manual RS guidelines (Rutkowski 1997) for room temperature variation from the thermostat set point at some point. The type of failure was indicative of each system type, with the typical airflow system both over- and underheating spaces, while the low volume and transfer fan system consistently underheated the bedrooms. Each system showed an 80% failure rate in at least one bedroom. During midseason heating conditions, all systems showed improved performance with smaller room-to-thermostat temperature differences and typically a lower failure rate.

The typical airflow system had the best ability to meet the ACCA Manual RS guidelines (Rutkowski 1997) and the ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time) requirements (ASHRAE 2010a). The low airflow system performed competitively enough to be worthy of future consideration. Although the transfer fan system was effective at meeting the requirements in the bedrooms, it did this at the expense of over-conditioning the second-floor landing. The success of leaving the bedroom doors open with no distribution system is indicative of one factor that may impact comfort complaints in homes with real occupants: leaving the interior doors open can help compensate for an ineffective distribution system.

Cutting-edge builders are achieving thermal enclosure performance that may make possible systems that use less distribution ductwork. This report offers insight into some strategies for reducing the amount of distribution ductwork. The report also provides heretofore unmeasured data for the range and frequency of potential thermal discomfort that occupants may experience when using each of these strategies. Thus, builders now can use this information to discuss space conditioning options with their clients to determine the level of potential discomfort the occupants are willing to accept to have a cost-optimized, cutting-edge, energy-efficient house.

1 Introduction and Background

IBACOS anticipates that houses achieving 50% whole-house source energy savings with respect to the Building America 2010 Benchmark (Hendron and Engebrecht 2010a) will be considered “low load.” Low load is defined by IBACOS as a house with a thermal enclosure that yields a maximum space heating and cooling load of less than 10 Btu/h/ft^2 of conditioned floor area (31.5 W/m^2 [1,200 ft^2/ton]). These small loads can be met by systems other than today’s typical ducted forced-air systems. For example, distributed fan coils with minimized ducts, terminal fan coil units, or point source units with buoyant force or ventilation-driven distribution may provide sufficient occupant comfort in a low-load home. These systems, which can have lower total installed costs than traditional ducted forced-air systems (Stecher 2011), allow the thermal enclosure characteristics of low-load houses to provide first-cost savings in addition to operational cost savings.

The purpose of this study is to help determine cost-effective solutions for heating and cooling houses that are designed to be energy efficient. This is done by testing the occupant comfort performance of some concepts that may already exist on the market but are not in use by production homebuilders. In some cases, the products are market available, but their use in housing may be a new application. The standards used to assess the performance of the systems in this study are ACCA Manual RS (Rutkowski 1997) and ASHRAE Standard 55-2010 (ASHRAE 2010a). These two standards also are what traditional ducted forced-air systems are required to meet. Although some may question if most field-installed traditional ducted forced-air systems actually do meet these standards, a large-scale study of the field-installed performance of traditional ducted forced-air systems is beyond the scope of this research. However, such a study should be considered by researchers in the future.

This report addresses the following research questions:

- To what extent do alternative space conditioning distribution strategies meet Air Conditioning Contractors of America (ACCA) and ASHRAE guidelines for room-to-room temperature uniformity and stability, respectively? Under what weather conditions do indoor temperatures rise above or fall below the guidelines?
- How substantially do the alternative strategies differ from typical ducted forced-air systems in their ability to meet ACCA and ASHRAE guidelines?

To answer these questions, IBACOS performed energy modeling and created two low-load test facilities with instrumentation to enable the testing of several experimental alternatives to traditional forced-air distribution designs. One facility is a retrofit unoccupied test house in Fresno, California (Stecher and Poerschke 2013); the other is a new construction unoccupied test house in Pittsburgh, Pennsylvania.

This report focuses on the results of the four systems tested in the Pittsburgh new construction unoccupied test house. Figure 1 shows a front view of the Pittsburgh test house. Based on the specifications implemented during construction of the Pittsburgh test house, resultant calculated design loads are 18,526 Btu/h (5,430 W) in heating and 11,236 Btu/h (3,293 W) in cooling. This corresponds to 6.7 Btu/ft^2-h (21.2 W/m^2) and 4.1 Btu/ft^2-h (12.8 W/m^2) on a per-unit area basis, respectively. Building Energy Optimization (BEopt™) modeling shows energy savings of 67.1% with respect to the Building America 2010 Benchmark (Hendron and Engebrecht 2010a). Full specifications are shown in Section 2 and were documented by Oberg (2010).

Figure 1. Front view of the Pittsburgh test house

The following systems were tested in the Pittsburgh test house:

- Typical airflow volume ducted distribution system routed to all spaces requiring conditioning, with two thermostatically controlled zones—one for the first floor and one for the second floor—with ducts sized to meet peak system airflow output.
- Low airflow volume ducted distribution system to the bedrooms (between 19 and 23 CFM for each bedroom) and a single point of delivery to the main living space; one thermostatically controlled zone for the bedrooms and one for the main living space.
- Over-door powered transfer fans to connect the bedrooms to the upstairs hallway, which is served by a single register on its own thermostat; a single register with its own thermostat served the first floor.
- No active systems—only open doors—to thermally connect the bedrooms to the upstairs hallway, which is served by a single register on its own thermostat; a single register with its own thermostat served the first floor.

2 Building and Distribution System Specifications

2.1 Specification Development and Energy Modeling

The focus of this research is not to validate whole-building energy consumption models. However, to develop the specifications for the Pittsburgh test house and to put the results in context with other Building America projects, Oberg (2010) created energy models of the Pittsburgh test house using both EnergyGauge USA (Version 2.8.03) and TRNSYS (Version 16.01). These models, input with the specifications shown in Table 1, predict that the house (excluding the photovoltaic [PV] system) will achieve approximately 67.1% whole-house source energy savings with respect to the 2008 Building America Benchmark (Hendron 2007).

Although many of the systems selected for the south-facing, two-story plus basement, 2,772-ft^2 Pittsburgh test house were chosen to answer research questions unrelated to the research discussed in this report, the thermal enclosure specifications were sufficient to provide for low heating and cooling loads, as discussed in detail in Section 2.2.

Table 1. Pittsburgh Test House Specifications

Assembly	Specifications
Concrete Slab	R-10 continuous below slab
Basement/Crawlspace Walls	R-25 finished portion of basement, R-19.5 unfinished portion of basement
Above-Grade Exterior Walls	2 × 4 studs staggered within a 2 × 8 wall thickness, R-30 cavity insulation, R-10 continuous exterior sheathing with recessed furring strips, 5/8-in. drywall, framing fraction of 15%, whole-wall U-value = 0.024 Btu/h/ft^2
Overhanging Floors	N/A
Roof (Location of Insulation)	R-60 blown insulation in the floor of the vented attic
Exterior Doors	R-5
Windows	306 ft^2, U-value = 0.24, solar heat gain coefficient = 0.22
Building Airtightness	0.54 ACH at 50 Pa actual (0.6 ACH at 50 Pa target)
Mechanical Ventilation	Per ASHRAE Standard 62.2 (ASHRAE 2010b) continuous requirement via self-balancing ERV with exhaust from baths, kitchen, and laundry with supply into AHU return
Heating and Cooling	Water-to-air GSHP with integral desuperheater, ASHRAE/ARI/ISO 13256-1 rated part-load cooling, EER of 26.0, part-load heating COP of 4.6, full-load cooling EER of 18.5, full-load heating COP of 4.0, two stages of heating and cooling, two types of ground loops, traditional vertical well and experimental horizontal loop, below-the-basement slab assembly
Ductwork	Three strategies for evaluation: 1) standard system using duct board and insulated flex duct; 2) reduced volume distribution using polyvinyl chloride piping; 3) reduced distribution locations with one supply on each floor, all ductwork located within conditioned space, targeted leakage rate of 0% leakage to outdoors, and tested total system leakage of 15% to indoors and 0% leakage to outdoors
Water Heater	GSHP desuperheater serving a 400-L preheat storage tank with an EF of 0.91, which serves a conventional 200-L electric water heater with an EF of 0.94; cross-linked polyethylene piping water distribution with a home-run piping layout for the hot water supply and a distributed remote manifold system for the cold water supply
Appliances	ENERGY STAR®-rated refrigerator and clothes washer with the lowest energy consumption ratings from within the builder's standard product line, dryer matched to washer, dishwasher is builder's standard ENERGY STAR-rated dishwasher but does not have the lowest energy consumption rating in its class
Miscellaneous Electric Loads	No strategies were incorporated to reduce this load area
Fluorescent Lighting	100% energy-efficient, high efficacy lighting design with a wireless light control system
PV System	3.8-kW solar PV array with a rated conversion efficiency of greater than 99%, consisting of 16 240-W panels, each panel equipped with its own microinverter

ACH is air changes per hour. AHU is air handler unit. COP is coefficient of performance. EER is energy efficiency ratio. EF is energy factor. ERV is energy recovery ventilator. GSHP is ground source heat pump.

2.2 Load Calculation and System Selection—ACCA Manual J and Manual S

To establish the low-load nature of the house, the research team performed ACCA Manual J (Rutkowski 2006) calculations for peak total and room-by-room design loads based on the specification package shown in Table 1. The calculations were performed using Wrightsoft Right-Suite Universal 8.0.11 software (Wrightsoft). The input assumptions included the following:

- One occupant in each bedroom and one occupant in the family room (five total occupants)
- Peak internal gains of 2,400 Btu/h (703 W) according to ACCA Manual J, Section 9, Appliance, Equipment and Lighting Load Scenario 1, which assumes the house contains "a refrigerator, range with vented hood, dish washer, clothes washer and vented clothes dryer, and contains electronic equipment and lighting allowances." The peak internal gains are distributed as follows: "1000 Btu/h (293 W) for the kitchen, 500 Btu/h (147 W) for the utility room, and 900 Btu/h (264 W) for a TV or computer and a few lighting fixtures" (Rutkowski 2006).

The resultant whole-house design loads were 18,526 Btu/h (5,430 W) in heating and 11,236 Btu/h (3,293 W) in cooling. This corresponds to 6.7 Btu/ft^2-h (21.2 W/m^2) and 4.1 Btu/ft^2-h (12.8 W/m^2) on a per-unit area basis, respectively. These values are within the low-load threshold of 10 Btu/ft^2-h (31.5 W/m^2).

Individual room loads are shown in Table 2. Using these loads and ACCA Manual S protocol (Rutkowski 1995), IBACOS selected the equipment to be installed in the test house based on the manufacturer's equipment data. Due to other research needs unrelated to this study, a GSHP system was specified for the house. Because the total calculated airflow value for the house was smaller than the smallest size of AHU available from the manufacturer, the ACCA Manual S oversizing limit could not be followed, and the system as described in Table 1 was oversized by approximately a factor of 2.

Table 2. Individual Room Loads

Room	Heating Load (Btu/h)	Cooling Load (Btu/h)
Finished Basement	2,296	916
Basement Bath	85	27
Unfinished Basement	2,381	808
Breakfast Room	1,191	593
Kitchen	255	108
Dining Room	1,021	862
Family Room	1,914	997
Laundry Area	510	135
Powder Room	0	0
Foyer	2,041	593
Bedroom 2	1,191	889
Bedroom 3	893	2,102
Bedroom 4	1,616	1,617
Bath 2	213	108
Master Bedroom	1,616	1,186
Master Bath	468	216
Master Walk-in Closet	213	108
Upper Hallway	213	162
Total House	**18,117**	**11,427**

2.3 Distribution System Development—ACCA Manual D and Manual T

IBACOS implemented the four distribution systems in the Pittsburgh test house by manually connecting different ductwork systems to the central AHU. The team installed three different ductwork systems in the house, along with one transfer fan strategy. The detailed heating, ventilation, and air conditioning (HVAC) system design and commissioning for each of the four systems are documented as follows.

The ductwork used in each of the four distribution systems in the Pittsburgh test house was evaluated in the ACCA Manual D (Rutkowski 2009a) design process:

- **Distribution system 1:** Typical airflow volume ducted distribution system routed to all spaces requiring conditioning, with two thermostatically controlled zones—one for the first floor and one for the second floor—with ducts sized to meet peak system airflow output.
- **Distribution system 2:** Low airflow volume ducted distribution system to the bedrooms (19 to 23 CFM each) and a single point of delivery to the main living space; one thermostatically controlled zone for the bedrooms and one for the main living space.
- **Distribution system 3:** Over-door powered transfer fans to connect the bedrooms to the upstairs hallway, which is served by a single register on its own thermostat; a single register with its own thermostat served the first floor.

- **Distribution system 4:** No active systems to connect the bedrooms to the upstairs hallway, which is served by a single register on its own thermostat; a single register with its own thermostat served the first floor.

Using ACCA Manual D protocol (Rutkowski 2009a), the duct systems were sized based on the individual room loads calculated by Wrightsoft Right-Suite Universal 8.0.11 (Wrightsoft). Trunks were sized to enable 700 to 900 fpm (3.56 to 4.57 m/s) and branches at 400 to 500 fpm (2.03 to 2.54 m/s). The design pressure drop between each branch was maintained to within ±0.01 in. of water column (±2.5 Pa).

The supply registers in the bedrooms and bathrooms were sized using ACCA Manual T protocol (Rutkowski 2009b). The distance from the register face to the exterior wall was measured, and 2 ft were added to ensure that the register had sufficient throw to reach the exterior wall and turn down to create air circulation within the space. IBACOS used these throw distances and ACCA Manual T noise criteria requirements (Rutkowski 2009b) to select the registers. Supply registers for all systems are located high on interior partition walls, a well-established method to provide cooling energy. IBACOS has determined, through research in cold climates, that this register placement provides equal occupant comfort when compared to perimeter floor registers for heating distribution (Rittelmann 2008).

The measured flow volume from each supply register was determined using a low-flow balometer flow hood with an accuracy of ±3% + 5 CFM (TSI 2012). A study by Wray et al. (2002) indicates actual error for these devices could be up to 30%. This error can have an impact on the results from this study and the actual real-world installation of HVAC equipment. Due to the lower overall magnitude of the airflow, the risk of measurement error suggesting a dramatically different amount of energy provided to a room is lower than that of a typical house. Other factors contributing to energy gain or loss offer a relatively greater percentage of the error because airflow from the space conditioning system is smaller relative to other internal thermal transfer means within this house. Due to the lack of sufficient flat wall area surrounding some registers, the opening of the flow hood was made smaller using duct mask such that only the register face would fit through the opening.

2.3.1 Typical Airflow Volume Ducted Distribution to the Bedrooms

Distribution system 1—typical airflow volume ducted distribution system to the bedrooms and main living space—used ductwork sized according to ACCA Manual D (Rutkowski 2009a) to provide airflows (Table 3) to meet ACCA Manual J loads (Rutkowski 2006). Figure 2 shows the distribution system layout. Bedroom doors were closed during the testing of this system. There were two thermostatically controlled zones. The first floor and basement were controlled using a thermostat located in the center hallway; the second floor was controlled using a thermostat located in the master bedroom. A single electronically controlled damper for each zone was used to prevent or enable airflow. Ductwork supplying the bedrooms ran through the walls and ceiling, obstructing access and preventing dampers from being used to balance the system. The measured airflow values deviate significantly from the specified flows, but this is typical for a field-installed system.

Table 3. System 1—Design and Measured Room Airflows

Room	Heating Specified CFM	Cooling Specified CFM	Cooling Measured CFM
Finished Basement	64	56	52
Unfinished Basement	61	53	52
Breakfast Room	43	37	60
Dining Room	52	45	25
Family Room	58	50	56
Foyer	49	43	25
Bedroom 2	48	42	17
Bedroom 3	75	65	55
Bedroom 4	74	64	35
Master Bedroom	63	55	30
Master Bath	14	12	28
Total	601	522	435

*Heating airflow not measured.

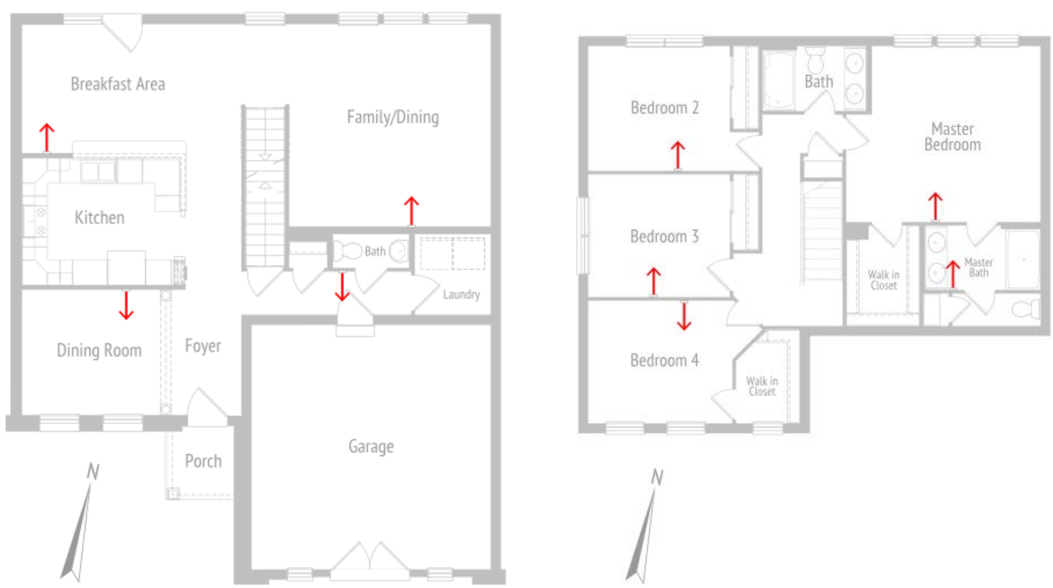

Figure 2. Typical airflow volume ducted distribution to the bedrooms and all rooms: first floor (left) and second floor (right). (The partially open door position shown in the graphic does not represent the door position used in the house during testing.)

2.3.2 Low Airflow Volume Ducted Distribution to the Bedrooms

Distribution system 2—low airflow volume ducted distribution system to the bedrooms and a single point of distribution delivery into the main living space—was inspired by the Passivhaus ventilation systems in Europe, whereby the entire peak heating load of a residence is delivered via the volume of ventilation air required for good indoor air quality (Feist et al. 2005). In the case of the Pittsburgh test house, the amount of ventilation air required for good indoor air quality was 80 CFM per ASHRAE Standard 62.2 (ASHRAE 2010b). Assuming a temperature difference between the supply air and room air of 25°F, the total delivered energy that 80 CFM can deliver is 2,168 Btu/h. This value is approximately 19% and 12% of the total peak cooling and heating loads of the house, respectively (see Section 2.2), which are most likely inadequate for sufficient occupant comfort if used as the sole source of space conditioning. However, if the conditioned ventilation supply air were delivered to only the four bedrooms of the house, the delivered load would be approximately 35% of the peak loads in both heating and cooling for those rooms. The remaining portion of the house, which has an open floor plan, could be conditioned by a single supply register (Table 4 and Figure 3). Although the peak load on the bedrooms was approximately three times greater than what the test system would deliver, it was anticipated that the substantial difference would reveal a clear success-to-failure transition point at certain load conditions on the house.

Table 4. System 2—Design and Measured Room Airflows

Room	Heating	Cooling	
	Specified CFM	Specified CFM	Measured CFM
Family Room	300	300	287
Bedroom 2	20	20	20
Bedroom 3	20	20	23
Bedroom 4	20	20	23
Master Bedroom	20	20	19
Total	380	380	372

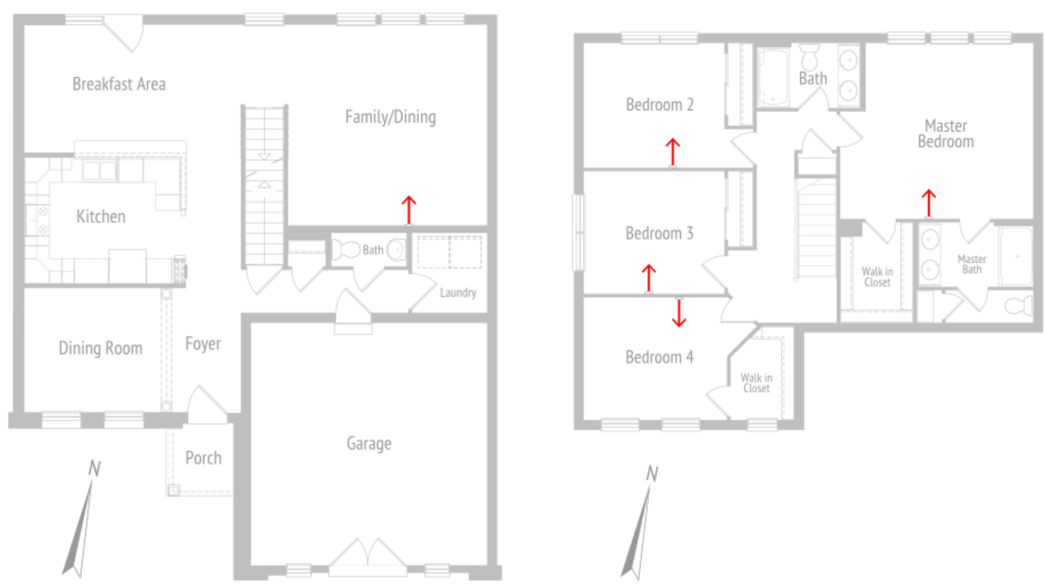

Figure 3. Low airflow volume ducted distribution to the bedrooms and full volume single point to the main floor: first floor (left) and second floor (right). (The partially open door position shown in the graphic does not represent the door position used in the house during testing.)

To deliver the appropriate amount of airflow, a set of low-capacity ductwork leading to the bedrooms was connected to the second-floor zone output of the test AHU. Because the required airflow to the second-floor zone was significantly lower than the amount provided by the AHU, an additional duct leading to the unfinished basement was attached at the zone damper and incorporated a reheat device to neutralize the thermal energy impact of the air coming out of this duct during cooling mode.

Bedroom doors were closed during the testing of this system.

2.3.3 Over-Door Powered Transfer Fans to the Bedrooms

Distribution system 3 is over-door powered transfer fans to connect the bedrooms to the upstairs hallway, which was served by a single register on its own thermostat. In this system, a single register with its own thermostat served the first floor, whereas another single register with its own thermostat served the second-floor landing, with the specified airflows shown in Table 5 and shown by the red arrows in Figure 4. Bedroom doors were closed during the testing of this system. To facilitate the movement of conditioned air from the landing to the bedrooms, a through-wall transfer fan was installed above each bedroom door (green arrows in Figure 4). These transfer fans are market available and are designed to be used in the manner installed in the Pittsburgh test house—to help circulate conditioned air to non-directly conditioned rooms. Each transfer fan (Figure 5) was rated by the manufacturer to provide 75 CFM of airflow and was controlled by an individual thermostat in the room it served. The use of an individual thermostat for each transfer fan was intended to enable each transfer fan to respond to the conditions in the room it served. All thermostats in the house—transfer fan and HVAC system—were set to the same temperature: 71°F in heating and 76°F in cooling. Return air pathways were provided in the bottom of each bedroom door.

Table 5. System 3—Design and Measured Room Airflows

Room	Heating	Cooling	
	Specified CFM	Specified CFM	Measured CFM
Family Room	300	300	287
Bedroom 2	75*	75*	not measured
Bedroom 3	75*	75*	not measured
Bedroom 4	75*	75*	not measured
Master Bedroom	75*	75*	not measured
Upper Hallway	300	300	273
Total	600	600	560

*This is air circulated from the hallway to the bedroom.

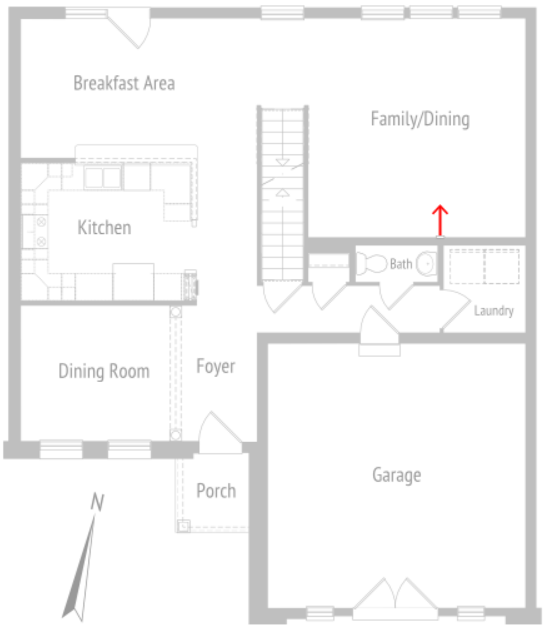

Figure 4. Over-door powered transfer fans to the bedrooms and full volume single point to the main floor: first floor (left) and second floor (right). (The partially open door position shown in the graphic does not represent the door position used in the house during testing.)

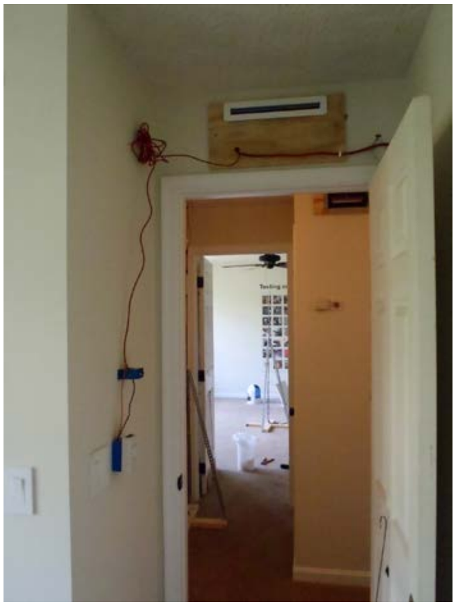

Figure 5. Over-door powered transfer fans to bedroom 2

2.3.4 No Ducted Distribution to the Bedrooms with Doors Open

Distribution system 4—no active systems to connect the bedrooms to the upstairs hallway—was served by a single register on its own thermostat. A single register with its own thermostat served the first floor (Figure 6, Table 6, and Figure 7). The decision to leave the bedroom doors open to facilitate free movement of air from the hallway was based on research performed by Barakat (1985). Barakat's research indicated that the steady-state heat transfer through an open doorway at a 2.7°F temperature difference was 1,050 Btu/h via natural convection, with 48 Btu/h via radiation. Building on this work, multizone airflow calculations and computational fluid dynamics simulations performed by Feist et al. (2005) of dwellings meeting the Passivhaus energy standard indicate that, at a temperature difference of 1.8°F between spaces, expected heat transfer rates are 300 to 600 Btu/h through open doors and 0.3 to 0.6 Btu/h/ft^2 of internal partition wall area. In a low-load home, these values may provide a substantial contribution to satisfying an individual room load before providing active conditioning. Field test data obtained by IBACOS (2008, 2010a, 2010b) from a Passivhaus in climate zone 5 also support this hypothesis.

Figure 6. First-floor (left) and second-floor (top) supply register locations

Table 6. System 4—Design and Measured Room Airflows

Room	Heating	Cooling	
	Specified CFM	Specified CFM	Measured CFM
Family Room	300	300	287
Upper Hallway	300	300	273
Total	600	600	560

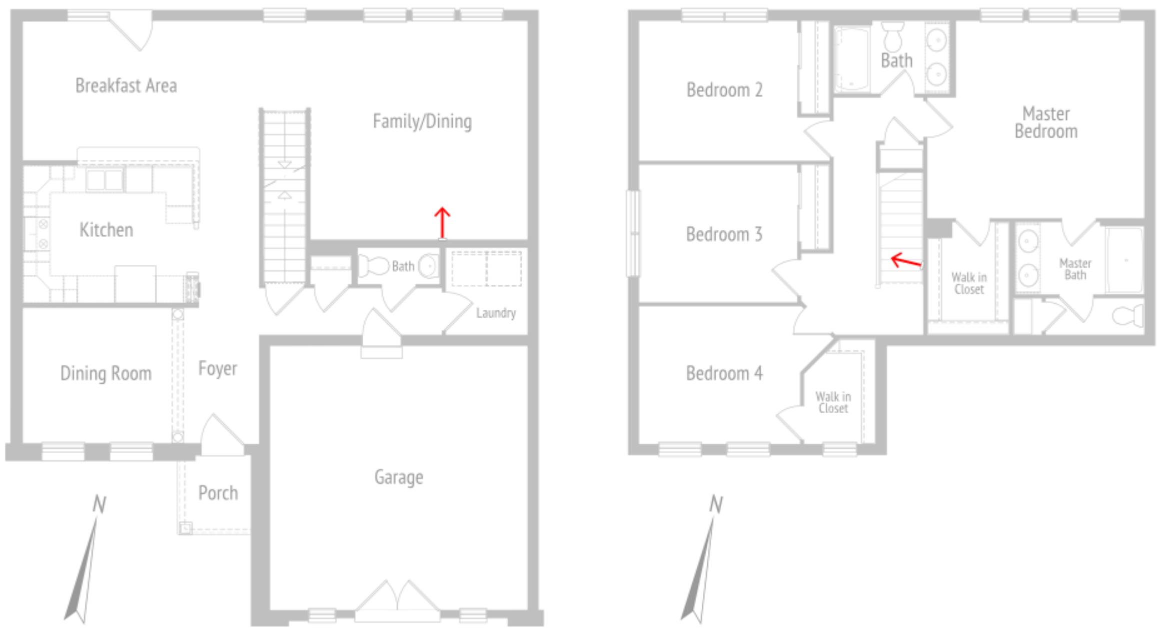

Figure 7. No ducted distribution to the bedrooms: first floor (left) and second floor (right). (The partially open door position shown in the graphic does not represent the door position used in the house during testing.)

3 Field Test Methods

IBACOS performed field tests for 1 year in the Pittsburgh test house to determine the extent to which each system did or did not meet established guidelines. To ensure the collection of data for each system throughout the entire year, IBACOS operated each system for 12 days at a time, with 2 days allotted for changeover to the next system. The research team then used the collected data in an analysis incorporating the relevant comfort criteria. In this case, the criteria were limited to temperature—specifically, ACCA Manual RS (Rutkowski 1997) and ASHRAE Standard 55-2010, Sections 5.2.5 (Temperature Variations with Time), 7.3.2 (Temperature Cycles and Drifts), and 7.4 (Measuring Conditions) (ASHRAE 2010a)—because maintaining an acceptable indoor temperature represents the minimum requirement that an HVAC system must meet. Assuming some systems in this study are shown to be viable, future studies can focus on assessing the ability of these systems to meet other relevant comfort criteria.

3.1 Comfort Criteria

Occupant comfort is based on a combination of factors, including occupant clothing and activity level, room air temperature and humidity, mean radiant temperature, and room air velocities (ASHRAE 2010a). In this study, dry bulb temperature was determined to be the primary factor of consideration because most residential HVAC systems turn on and off based solely on the dry bulb temperature measured by the thermostat. To fail in this area indicates a fundamental failure of the system; to succeed in controlling temperature prompts follow-up questions of performance in other areas (e.g., humidity control) that require different experimental setups. The relevant standards for room air temperature are defined by ACCA Manual RS (Rutkowski 1997) and ASHRAE Standard 55-2010 Sections 5.2.5 (Temperature Variations with Time), 7.3.2 (Temperature Cycles and Drifts), and 7.4 (Measuring Conditions) (ASHRAE 2010a).

ACCA Manual RS (Rutkowski 1997) requires the dry bulb temperature measured within any room of the thermostatically controlled zone to be within ±3°F of the thermostat setting during the cooling season. Similarly, the temperature during the heating season in any room must be within ±2°F of the thermostat set point temperature. The temperature difference measured between any two rooms in the zone (also known as the room-to-room temperature difference) should be no greater than 4°F in the heating season and no greater than 6°F in the cooling season. Multizone systems are used in this study; however, because each thermostat uses the same set point, the house is treated as one zone for assessment with respect to this standard.

ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time) (ASHRAE 2010a) outlines three types of fluctuations that are unsatisfactory to occupants: cyclic variations, drifts, and ramps. Cyclic variations involve temperature changes of greater than ±2°F over a 15-min interval. Drifts are temperature changes that are due to internal or external loads acting on the building, whereas ramps are caused by the space conditioning system. The temperature change during a drift or ramp can be positive or negative, and the allowable change in temperature is based on the duration of time over which it occurs. Specifically, for any 15-min period, no more than 2°F change is allowed; for any 30-min period, no more than 3°F is allowed. For any 60-min period, no more than 4°F is allowed. For any 120-min period, no more than 5°F is allowed, and for any 240-min period, no more than 6°F is allowed (ASHRAE 2010a).

ASHRAE Standard 55-2010 provides the researcher with Section 7.3.2 (Temperature Cycles and Drifts) and Section 7.4 (Measuring Conditions) in order to assess the criteria in Section 5.2.5 (ASHRAE 2010a). Section 7.3.2 (Temperature Cycles and Drifts) provides the equations to follow to determine the difference in temperature over a given period of time. Section 7.4 (Measuring Conditions) provides the conditions in which to perform the measurements. Specifically, the indoor-to-outdoor temperature difference during the measurement period must be greater than 50% of the indoor-to-outdoor temperature difference used for the heating or cooling design calculation. Additionally, for the heating season, the weather should be cloudy; for the cooling season, the weather should be sunny (ASHRAE 2010a). Table 7 summarizes the design conditions for the Pittsburgh test house.

Table 7. Design Conditions Used for Determining ASHRAE 7.3.2 Compliance

	Design Temperatures from Wrightsoft
Winter Design Conditions	1.8°F (Cloudy) (50% design: 36.4°F)*
Summer Design Conditions	89.1 °F (Sunny) (50% design: 82.55°F)*

* Indoor temperature assumed to be 71°F during winter and 76°F during summer.

Section 7.4 (Measuring Conditions) of ASHRAE Standard 55-2010 (ASHRAE 2010a) requires only 2 hours of data collection to assess system performance. In this study, IBACOS performed the analysis for every day in which the average outdoor and indoor temperatures met Section 7.4 requirements (e.g., for all the minutes of all the days with conditions at or above 50% of the design conditions). To better capture aberrant behavior, the research team analyzed whole-day data rather than data during a single 2-hour period. For contrast, the team also performed analysis using the data from the time periods not meeting the conditions required for Section 7.4 (e.g., for all the minutes of all the days with conditions less than 50% of the design conditions). Solar irradiance is another important driver of the house load. IBACOS manually screened days so that periods considered as meeting Section 7.4 requirements for thermal loads also met the requirement of having design condition irradiance.

3.2 Instrumentation Setup

To assess the performance of the four distribution systems with respect to the ACCA Manual RS (Rutkowski 1997) and ASHRAE Standard 55-2010 (ASHRAE 2010a) guidelines, IBACOS installed instrumentation throughout the Pittsburgh test house. The research team installed thermocouples in each room in the house to measure air temperature (Table 8). Then the team also made supply temperature measurements using thermocouples. Measurement of the outdoor conditions comprised temperature, humidity, and global incident solar radiation. Measurements from 20-sec scans were averaged over 1 min and were recorded. The data used for this analysis were collected from the systems installed in the Pittsburgh test house from April 15, 2012, to January 15, 2013.

Table 8. Long-Term Measurement

Measurement	Equipment Used	Measurement Uncertainty
Air temperature at 43 in. from the floor	Unshielded Type-T thermocouples (sensors exposed to direct sunlight were shielded)	±1.1°F
AHU runtime averaged temperature via unshielded thermocouple at each supply location at point of maximum velocity	Unshielded Type-T thermocouples with maximum velocity location determined by hot wire anemometer trace	±1.1°F
AHU runtime averaged temperature at central return via unshielded thermocouple	Unshielded Type-T thermocouples	±1.1°F
Air temperature at each over-door and bottom-of-door transfer grille location	Unshielded Type-T thermocouples	±1.1°F
Runtime of HVAC system: AHU and heat pump outdoor unit	Continental Control System Wattnode	0.5%
Runtime of ERV	Continental Control System Wattnode	0.5%
Global incident solar radiation on site	LI-COR 200 silicon pyranometer	5.0%
Outdoor temperature and relative humidity	Vaisala HMP60 in shielded enclosure	±0.6°F, ±3.0% relative humidity

Internal sensible and latent gains according to the simulated occupancy schedule occurred through the monitoring period. Other loads and occupancy that were simulated using humidifiers and electric resistance heaters to match the simulated loads were based on the Building America House Simulation Protocols (Hendron and Engebrecht 2010b) and were implemented by one heater and one humidifier on the main floor and two small humidifiers and one heater in each of three of the four bedrooms. Three large humidifiers and one heater were located in the master bathroom to more accurately simulate the output intensity of moisture generated during showers. The research team simulated different loads by varying the duration of time of operation of the fixed output heaters and humidifiers. An ERV provided continuous ventilation to the house at ASHRAE Standard 62.2 rates (ASHRAE 2010b).

IBACOS took the following measurements throughout the course of one year: indoor conditions (temperature and relative humidity) measured at the thermostat, and temperature measured at 43 in. from the floor by thermocouples placed at the locations shown in Figure 8. It has been shown that radiant asymmetry is negligible during cold weather; therefore, the team used unshielded thermocouples in this study.

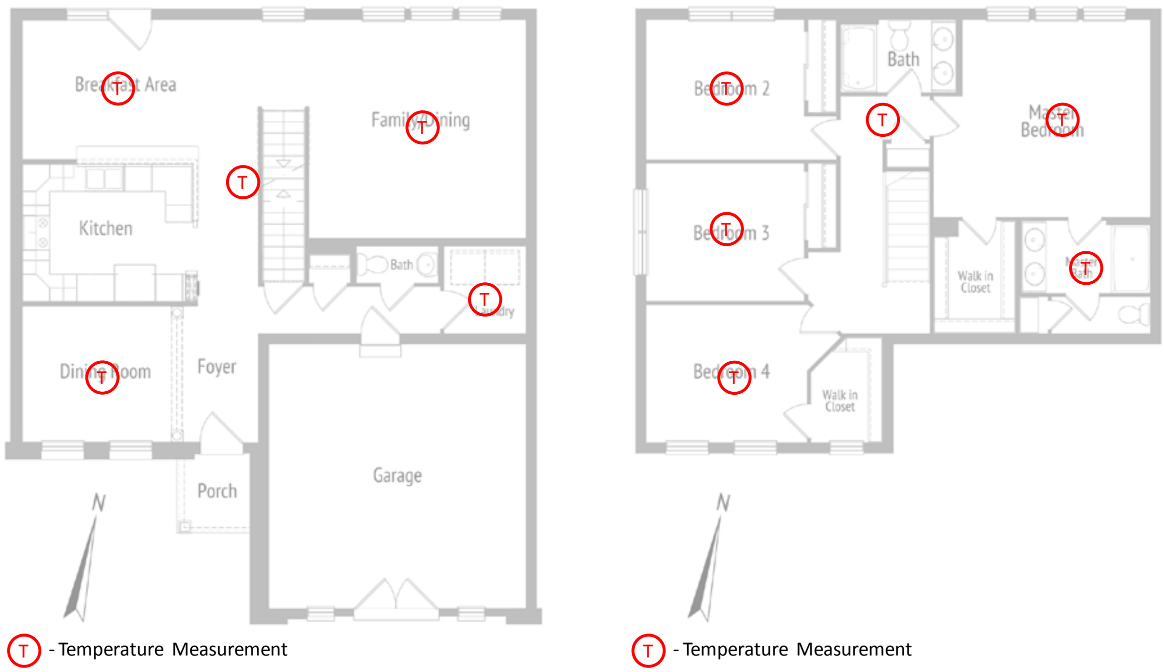

Figure 8. Sensor locations: first floor (left) and second floor (right). (The partially open door position shown in the graphic does not represent the door position used in the house during testing.)

Sensor uncertainty is important, given the ±2°F margin of the ACCA analysis. To determine the likelihood of drawing an incorrect conclusion from the measured results, the team performed an uncertainty analysis using measurements sampled from a normal distribution. The measured data indicated any individual room's temperature to be normally distributed, with a typical standard deviation around 3.0. Additionally, each sensor's measurement is assumed to be of a standard distribution around the true value, with a standard deviation of 0.25. Using these assumptions, the team sampled one million theoretical measurements and determined the rate at which the sensor error caused an incorrect conclusion whether a room was within the comfort criteria. This analysis indicates an 87% accuracy; that is, there is a 13% chance that any individual conclusion is inaccurate.

4 Field Test Results

IBACOS operated each system in the Pittsburgh test house for 12 days at a time throughout the course of a year to capture system operation in each season. The team analyzed temperature data from each room to determine the capability of the system in meeting room-to-thermostat temperature difference guidance by ASHRAE Standard 55-2010 Sections 5.2.5 (Temperature Variations with Time), 7.3.2 (Temperature Cycles and Drifts), and 7.4 (Measuring Conditions) (ASHRAE 2010a) and by ACCA Manual RS (Rutkowski 1997) as described in Section 3.1 of this report.

Due to issues encountered during testing—specifically, conflict with other testing occurring in the house during the cooling season and controls problems—data were not available for all systems during the cooling season. Limited data were available for the transfer fan and low volume system for the upstairs bedrooms. The results that follow are entirely for the heating season.

4.1 ASHRAE Standard 55-2010 Analysis

One potential source of occupant dissatisfaction is if the temperature of a room changes too rapidly or by too large of a magnitude. The magnitude of temperature change that an occupant will tolerate is impacted by the duration of time over which it occurs. Although ASHRAE Standard 55-2010 Section 5.2.5 (ASHRAE 2010a) allows room air temperature to change by up to 6°F over the course of 4 hours, only 2°F are allowed during periods of 15 min or less. For the systems tested in this study, the research team expected that the low airflow and transfer fan systems would see failures over a long duration due to inability to meet the individual room load, while the typical airflow system would see failures over short periods of time when it provided too much conditioning for the given load on a room.

To determine the ability of each system to meet ASHRAE Standard 55-2010 Section 5.2.5, IBACOS collected data during the heating test condition per Section 7.4 of ASHRAE Standard 55-2010 (ASHRAE 2010a). These data are summarized in Table 9 and Table 10.

Table 9. Passing Rate (%) of Each System for ASHRAE 55-2010 Section 5.2.5 During Section 7.4 Measuring Conditions

System	Number of Days	Living	Breakfast	Dining	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Typical Airflow, Doors Closed	4	100	100	100	100	100	100	94	100	100	100
Low Airflow, Doors Closed	13	100	100	100	100	99	100	99	100	100	100
Over-Door Transfer Fans, Doors Closed	2	8	21	100	100	0	100	100	100	100	100
No Ducted Distribution, Doors Open	1	1	1	81	100	100	100	100	100	100	100

Table 10. Failure Rate (%) of Each System via Cyclic Temperature Change of More Than 2°F During ASHRAE 55-2010 Section 7.4 Measuring Conditions

System	Number of Days	Living	Breakfast	Dining	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Typical Airflow, Doors Closed	4	0.0	0.0	0.0	0.0	0.0	0.0	1.7	0.0	0.0	0.0
Low Airflow, Doors Closed	13	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Over-Door Transfer Fans, Doors Closed	2	86.5	75.0	0.0	0.0	97.3	0.0	0.0	0.0	0.0	0.0
No Ducted Distribution, Doors Open	1	98.3	97.6	19.5	0	0	0	0	0	0	0

The assumption that underconditioned rooms would fail over a long time frame held true. However, this is not obvious from the passing rate results shown in Table 9 because only one of the bedrooms showed a failure during the operation of the low volume system, and no bedroom failures occurred during the operation of the transfer fan system. Bedroom 4, the bedroom that did fail to meet the standard, failed 1% of the time due to a temperature change of greater than 4°F in 1 hour and 5°F in 2 hours. This bedroom also failed to meet the standard a total of 4% of the time during the operation of the typical airflow system due to a temperature change of 5°F over 2 hours and 6°F over 4 hours. Furthermore, this bedroom failed via a temperature change of greater than 2°F over 15 min (Table 10).

Although none of the bedrooms failed ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time) (ASHRAE 2010a) during the operation of the transfer fan system, the second-floor landing did, failing 97% of the time due to temperature change of greater than 2°F over 15 min (Table 10). This occurred because the amount of conditioned air necessary to condition the entire second floor was delivered to one small space. Although the transfer fans appeared to be effective at maintaining the air temperature of the bedrooms over time, they did this at the expense of overheating the second-floor landing each time the system cycled on.

Results of the cycling analysis are shown in Table 11 through Table 14 for data collected during heating, per Section 7.4 of ASHRAE Standard 55-2010 (ASHRAE 2010a), as well as data collected for all other conditions—there is a table for each operational mode (typical airflow, low airflow, transfer fans, and no ducted distribution). The percentage values in each table are based on the total amount of time—measured in days—used for the assessment.

Table 11. Failure Mode Heating Typical Airflow Volume Ducted Distribution to the Bedrooms with Doors Closed (1.2)—ASHRAE Conditions—4 Days

			Living	Breakfast	Dining	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Pass %			100.0	100.0	100.0	100.0	100.0	100.0	94.3	100.0	100.0	100.0
Fail %	Mode*	Cyc.	0.0	0.0	0.0	0.0	0.0	0.0	1.7	0.0	0.0	0.0
		30	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		60	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		120	0.0	0.0	0.0	0.0	0.0	0.0	1.5	0.0	0.0	0.0
		240	0.0	0.0	0.0	0.0	0.0	0.0	2.5	0.0	0.0	0.0

Cyc. indicates a 2°F drift in 15 min, 30 min a 3°F drift, 60 min a 4°F drift, 120 min a 5°F drift, and 240 min a 6°F drift.

Table 12. Failure Mode Heating Low Airflow Volume Ducted Distribution to the Bedrooms with Doors Closed (3.1)—ASHRAE Conditions—13 Days

			Living	Breakfast	Dining	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Pass %			100.0	100.0	100.0	100.0	99.0	100.0	99.3	100.0	100.0	100.0
Fail %	Mode	Cyc.	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		30	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		60	0.0	0.0	0.0	0.0	0.0	0.0	0.2	0.0	0.0	0.0
		120	0.0	0.0	0.0	0.0	0.0	0.0	0.5	0.0	0.0	0.0
		240	0.0	0.0	0.0	0.0	1.0	0.0	0.0	0.0	0.0	0.0

Cyc. indicates a 2°F drift in 15 min, 30 min a 3°F drift, 60 min a 4°F drift, 120 min a 5°F drift, and 240 min a 6°F drift.

Table 13. Failure Mode Heating Over-Door Powered Transfer Fans to the Bedrooms with Doors Closed (4.5)—ASHRAE Conditions—2 Days

			Living	Breakfast	Dining	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Pass %			8.0	21.3	100.0	100.0	0.0	100.0	100.0	100.0	100.0	100.0
Fail %	Mode	Cyc.	86.5	75.0	0.0	0.0	97.3	0.0	0.0	0.0	0.0	0.0
		30	5.5	3.7	0.0	0.0	2.4	0.0	0.0	0.0	0.0	0.0
		60	0.0	0.0	0.0	0.0	0.3	0.0	0.0	0.0	0.0	0.0
		120	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		240	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Cyc. indicates a 2°F drift in 15 min, 30 min a 3°F drift, 60 min a 4°F drift, 120 min a 5°F drift, and 240 min a 6°F drift.

Table 14. Failure Mode Heating No Ducted Distribution to the Bedrooms with Doors Open (4.4)—ASHRAE Conditions—1 Day

			Living	Breakfast	Dining	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Pass %			1.3	1.2	80.5	100.0	100.0	100.0	100.0	100.0	100.0	100.0
Fail %	Mode	Cyc.	98.3	97.6	19.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		30	0.4	1.2	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		60	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		120	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		240	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Cyc. indicates a 2°F drift in 15 min, 30 min a 3°F drift, 60 min a 4°F drift, 120 min a 5°F drift, and 240 min a 6°F drift.

In addition to the analysis performed during the conditions specified by ASHRAE Standard 55-2012 Section 7.4 (Measuring Conditions) (ASHRAE 2010a), IBACOS also performed the analysis during weather conditions that did not meet the requirements of Section 7.4. This occurred when the outdoor temperature was lower than the 50% design summer (82.55°F) temperature and greater than the 50% design winter (36.4°F) temperature (e.g., non-peak and midseason conditions). These data are tabulated for each system for heating mode. Table 15 provides a summary of these results. The low passing rate of the first-floor living/breakfast area and the second-floor landing is a result of the dual-point air distribution system supplying all conditioning air directly to these zones.

Table 15. Passing Rate (%) of Each System for ASHRAE 55-2010 Section 5.2.5 During Time Not Meeting Section 7.4 Measuring Conditions

System	Number of Days	Living	Breakfast	Dining	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Typical Airflow, Doors Closed	13	100	100	100	100	100	100	87	92	97	94
Low Airflow, Doors Closed	2	100	100	100	100	100	100	83	100	100	100
Over-Door Transfer Fans, Doors Closed	15	11	9	98	100	15	100	100	100	100	100

As shown in Table 16 through Table 18, rooms receiving the greatest conditioning air volume suffered significant cycling discomfort. Otherwise, during these mild conditions, each room had a high passing rate.

Table 16. Failure Mode Heating Typical Airflow Volume Ducted Distribution to the Bedrooms with Doors Closed (1.2)—Non-ASHRAE Conditions—13 Days

			Living	Breakfast	Dining	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Pass %			100.0	100.0	100.0	100.0	100.0	99.5	86.6	91.9	97.0	93.6
Fail %	Mode	Cyc.	0.0	0.0	0.0	0.0	0.0	0.4	5.2	4.9	2.6	5.9
		30	0.0	0.0	0.0	0.0	0.0	0.0	0.3	0.5	0.2	0.2
		60	0.0	0.0	0.0	0.0	0.0	0.1	1.2	0.7	0.2	0.4
		120	0.0	0.0	0.0	0.0	0.0	0.0	2.0	0.0	0.1	0.0
		240	0.0	0.0	0.0	0.0	0.0	0.0	4.6	2.0	0.0	0.0

Cyc. indicates a 2°F drift in 15 min, 30 min a 3°F drift, 60 min a 4°F drift, 120 min a 5°F drift, and 240 min a 6°F drift.

Table 17. Failure Mode Heating Low Airflow Volume Ducted Distribution to the Bedrooms with Doors Closed (3.1)—Non-ASHRAE Conditions—2 Days

			Living	Breakfast	Dining	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Pass %			100.0	100.0	100.0	100.0	100.0	100.0	83.1	100.0	100.0	100.0
Fail %	Mode	Cyc.	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		30	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		60	0.0	0.0	0.0	0.0	0.0	0.0	3.7	0.0	0.0	0.0
		120	0.0	0.0	0.0	0.0	0.0	0.0	8.0	0.0	0.0	0.0
		240	0.0	0.0	0.0	0.0	0.0	0.0	5.3	0.0	0.0	0.0

Cyc. indicates a 2°F drift in 15 min, 30 min a 3°F drift, 60 min a 4°F drift, 120 min a 5°F drift, and 240 min a 6°F drift.

Table 18. Failure Mode Heating Over-Door Powered Transfer Fans to the Bedrooms with Doors Closed (4.5)—Non-ASHRAE Conditions—15 Days

			Living	Breakfast	Dining	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Pass %			11.4	9.2	98.3	99.8	15.4	100.0	100.0	100.0	100.0	100.0
Fail %	Mode	Cyc.	87.5	90.1	1.6	0.2	84.0	0.0	0.0	0.0	0.0	0.0
		30	1.1	0.7	0.1	0.0	0.6	0.0	0.0	0.0	0.0	0.0
		60	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		120	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		240	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Cyc. indicates a 2°F drift in 15 min, 30 min a 3°F drift, 60 min a 4°F drift, 120 min a 5°F drift, and 240 min a 6°F drift.

4.2 ACCA Manual RS Analysis

Although the analysis in Section 4.1 focused on temperature changes over time within a given room, this Section 4.2 focuses on the difference in temperature between any given room and the thermostat, as well as the temperature difference between any one room and any other room. Guidance for this analysis is given by ACCA Manual RS (Rutkowski 1997). The purpose of evaluating the temperature difference between any room and the thermostat is to determine the effectiveness of the system at executing the intent of the user. The range of temperature allowed by ACCA Manual RS for a given room—±2°F from the thermostat set point during heating mode—represents the tolerance that the system can have without the user noticing. Results from the ACCA Manual RS (Rutkowski 1997) analysis using the methodology described in Section 2.1 are shown later in this section in Table 19 through Table 22.

Again, there is one table for each system (typical airflow, low airflow, transfer fans, and no ducted distribution). For consistency of data, IBACOS performed the analysis only on days that met the ASHRAE Standard 55-2010 Section 7.4 (Measuring Conditions) test conditions standard (ASHRAE 2010a).

In heating mode during ASHRAE measuring conditions, the typical airflow system was the most successful at keeping the rooms within the ±2°F from the thermostat set point specified by ACCA Manual RS (Rutkowski 1997), as shown in Table 19. However, even this system failed at least 80% of the time to maintain the temperatures of bedroom 2 and bedroom 4 with a maximum temperature difference of 8.3°F below the thermostat set point. It is unclear why only bedroom 2 and bedroom 4 showed these failures while bedroom 3 was successful. Because the data were collected during ASHRAE Standard 55-2012 Section 7.4 measuring conditions, outdoor conditions were cloudy—less than 2 hours of direct sunlight per day—with a temperature below 36.4°F. Bedroom 3 and bedroom 4 were stable during the ASHRAE cycles, drifts, and ramps analysis of the same data. Closer examination of the data as shown in Table 20 reveals that bedroom 2 and the master bedroom never reached the set point temperature. These two rooms, along with bedroom 4, also were underconditioned by 50% to 60%, based on their measured airflow rates as listed in Table 4. Bedroom 4 did reach the set point and surpassed it at times, but this was largely due to incident solar gain through its south-facing windows. This room experienced the most variability, further highlighting the difficult nature of conditioning low-load homes with large amounts of southern glazing and minimal thermal communication with the rest of the house. Greater conditioning air would be necessary to maintain comfort on cold nights; however, during sunny days, the room would overheat. Potential strategies to mitigate this problem might include greater thermal communication via open doorways, jump ducts, or central AHU cycling, or through a modulating HVAC system capable of supplying differing amounts of air to each zone, dependent upon the load. In effect, there is no way for a conventional HVAC system to adequately condition these types of zones. The second-floor landing, where the thermostat for this zone was located, was within ±2°F from the thermostat set point 90% of the time.

Table 19. Percentage of Time Outside ±2°F of the Thermostat Set Point During Heating Mode—ASHRAE Conditions

System	Number of Days	Living	Breakfast	Dining	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Typical Airflow, Doors Closed	**4**	7	15	6	7	7	35	80	19	85	9
Low Airflow, Doors Closed	**13**	47	52	39	50	46	26	83	11	46	29
Over-Door Transfer Fans, Doors Closed	**2**	34	33	44	83	58	100	87	10	100	95

Table 20. Typical Airflow Volume Ducted Distribution to the Bedrooms with Doors Closed (1.2) During Heating Mode—ASHRAE Conditions

	Room	Living	Dining	Breakfast	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Room-to-Thermostat Set Point Temp. Difference	Largest positive difference (°F)	4.4	4.0	3.6	4.4	6.9	–1.0	5.7	3.8	–1.1	1.3
	Largest negative difference (°F)	–5.0	–4.7	–5.3	–4.9	–4.7	–7.0	–8.3	–5.6	–7.4	–4.7
	Percentage of time outside ±2°F band	7.0	14.6	6.1	7.1	7.2	35.1	80.0	18.5	84.9	8.9

One might speculate that stratification from the first floor to the second-floor landing caused a false satisfied reading for the second-floor thermostat or that the significantly lower airflow in three of the four bedrooms caused their persistent deviation from the thermostat set point. However, during the testing of the low airflow system (Table 21), the master bedroom, bedroom 2, and bedroom 3 all improved, while bedroom 4 nominally remained the same. This trend was noticed, despite the system delivering approximately the same amount (20 CFM) of conditioned air to each room, as shown in Table 4. Furthermore, this system was operated as a single zone for the whole house, with the thermostat located on the first floor. As shown in Table 19, the hallway showed an increased percentage of failure, and this was indeed due to temperatures consistently at or above the set point (Table 20). So thermostat location may have been a factor in the low temperatures observed in bedroom 2 and the master bedroom. However, if this were the sole factor, the failure rate for bedroom 3 would have increased when its delivered energy advantage was eliminated during the low-flow system testing. It did not. In fact, the failure rate of bedroom 3 decreased.

Table 21. Low Airflow Volume Ducted Distribution to the Bedrooms with Doors Closed (3.1) During Heating Mode—ASHRAE Conditions.

	Room	Living	Dining	Breakfast	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Room-to-Thermostat Set Point Temp. Difference	Largest positive difference (°F)	5.3	7.1	4.8	5.1	6.8	–0.6	10.0	6.1	–0.1	1.3
	Largest negative difference (°F)	–2.2	–1.8	–2.4	–1.9	–0.8	–3.6	–7.7	–2.6	–3.7	–3.2
	Percentage of time outside ±2°F band	46.9	52.1	39.0	50.2	45.5	26.0	82.7	10.6	46.3	28.6

To shed more light on the issue, an analysis of the transfer fan system is necessary. These data are presented in Table 22. During the operation of the transfer fan system, the same trends were observed as during the operation of the typical airflow system: underheating of the northern bedrooms, bedroom 4 frequently out of bounds with both over- and underheating, and bedroom 3 within specifications most of the time. In this instance, the second-floor landing was overheating most of the time, indicating that the thermostat—in this case, located in the master bedroom—was calling for heating, but the transfer fans were unable to deliver it to the rooms. Upon reviewing the delivered air temperatures from the transfer fans, IBACOS determined that bedroom 3 received significantly warmer air because of its location near the second-floor landing supply register (Figure 4). That is, bedroom 3 received air that was 6°F to 10°F warmer than air received by the other bedrooms, which, assuming equal airflow to each room, would typically result in 10% more energy transferred relative to the other bedrooms. IBACOS' review of the solar data indicated that bedroom 4 did receive some solar gains coincident with its temperature rising above the thermostat set point. During operation of the transfer fan system, the whole house had a reduced performance, with a high rate of failure in the first-floor spaces.

Table 22. Over-Door Powered Transfer Fans to the Bedrooms with Doors Closed (4.5) During Heating Mode—ASHRAE Conditions

	Room	Living	Dining	Breakfast	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Room-to-Thermostat Set Point Temp. Difference	Largest positive difference (°F)	1.6	1.5	1.2	–1.0	8.9	–2.1	5.8	5.7	–2.8	–0.7
	Largest negative difference (°F)	–2.8	–3.0	–3.1	–3.3	–0.5	–4.3	–9.0	–1.7	–5.8	–4.9
	Percentage of time outside ±2°F band	33.7	33.4	43.7	82.8	58.1	100.0	87.3	9.6	100.0	95.4

IBACOS collected data for a scenario with single-point distribution and bedroom doors open; however, only 1 day fit the ASHRAE measurement conditions standard. A single day is an inadequate dataset from which to draw any conclusions; therefore, these data were omitted.

Bedroom 2 and the master bedroom also faced a large, grass-covered hill and had potentially lower infiltration (something a guarded blower door test may have indicated). Additionally, during clear sky, 50% of the view factor was taken up by the hillside. Bedroom 4 had a full sky view factor and no hill to block the wind. Also, three of the four walls of bedroom 4 were on the exterior of the home. This bedroom consistently had greater nighttime downward drift than did the other rooms. Bedroom 3 had the least amount of exterior wall area—only one of four walls. As a result, bedroom 3 had the smallest nighttime downward drift.

A summary graphic of this analysis is provided in Figure 9. The percentage of time each room is more than 2°F different from the thermostat provides a binary indication of comfort. As a scalar measure of the magnitude of each excursion, a degree hour metric has been calculated for each room—that is, the temperature difference beyond 2°F multiplied by the number of hours for each excursion. If a room was 4°F greater than the thermostat for 3 hours, this would result in 6 degree hours deviation (4°F – 2°F * 3). In this graphic, the number degree hours has been averaged over the number of days for each test period.

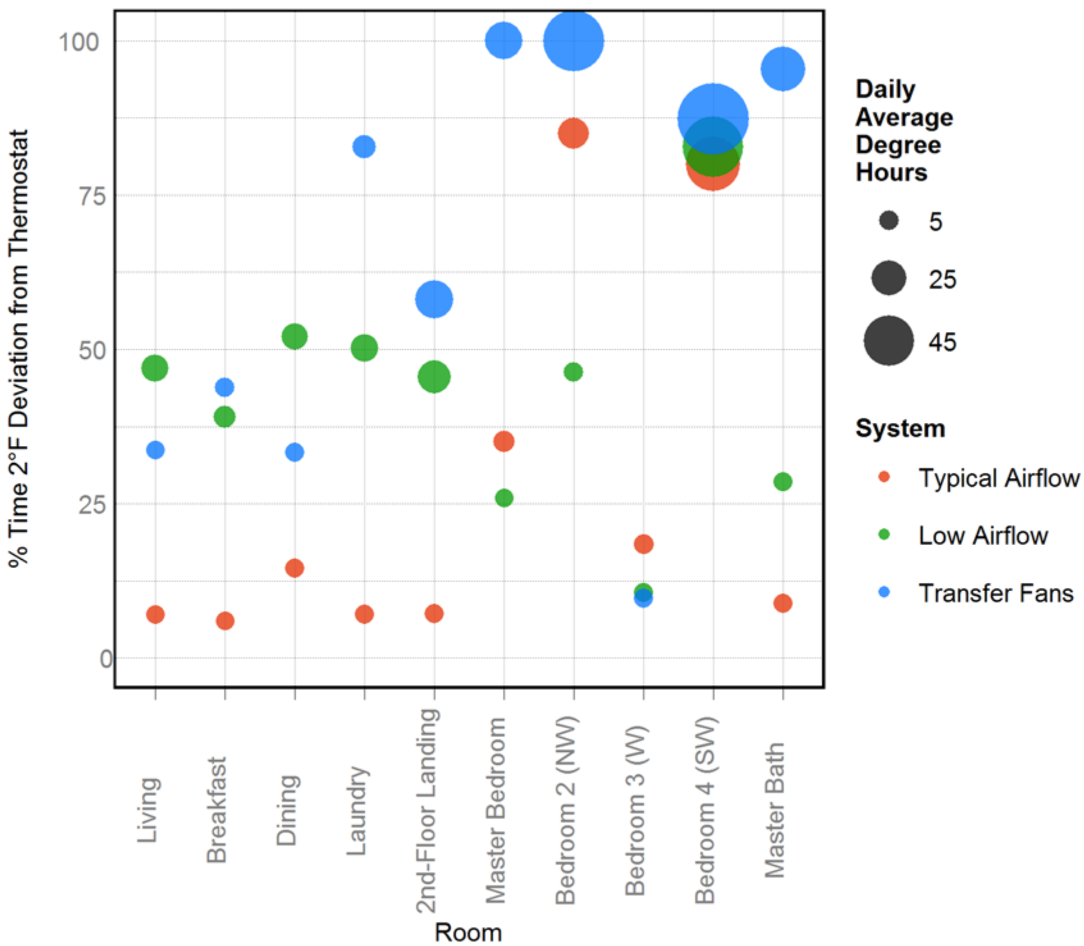

Figure 9. Percentage of failure time, and extent in degree hours, by room for ASHRAE conditions

From Figure 9, it is evident the typical airflow system was most successful at maintaining thermal uniformity; however, all systems struggled to adequately condition the varied loads in the southern, sun-tempered bedroom.

In addition to the analysis performed during the conditions specified by ASHRAE Standard 55-2012 Section 7.4 (Measuring Conditions) (ASHRAE 2010a), IBACOS performed the analysis during weather conditions that did not meet the requirements of Section 7.4. This occurred when the outdoor temperature was lower than the 50% design cooling temperature and greater than the 50% design heating temperature (e.g., non-peak and midseason conditions). Table 23 and Table 24 present a summary of these results.

Table 23. Percentage of Time Outside ±2°F of the Thermostat Set Point During Heating Mode—Non-ASHRAE Conditions

System	Number of Days	Living	Breakfast	Dining	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Typical Airflow, Doors Closed	**13**	9.5	2.1	7.8	0.2	0	1.6	53.6	15.6	2.1	9.3
Low Airflow, Doors Closed	**2**	0	8.4	0	0	0	0.7	60.8	35.6	1.1	0
Over-Door Transfer Fans, Doors Closed	**15**	27.8	40.6	41.7	70.1	15.2	73.6	96.3	61	88.8	84.5

Table 24. Typical Airflow Volume Ducted Distribution to the Bedrooms with Doors Closed (1.2) During Heating Mode—Non-ASHRAE Conditions

	Room	Living	Dining	Breakfast	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Room-to-Thermostat Set Point Temp. Difference	Largest positive difference (°F)	1.5	3.7	1.2	1.7	1.6	1.8	9.6	6.8	2.8	5.3
	Largest negative difference (°F)	–2.5	–1.9	–2.4	–2.1	–1.9	–2.6	–4.6	–4.6	–2.5	–0.3
	Percentage of time outside ±2°F band	9.5	2.1	7.8	0.2	0.0	1.6	53.6	15.6	2.1	9.3

During non-ASHRAE conditions, the impact of high solar gain can be seen in bedroom 3 and bedroom 4, with significant deviations above the thermostat set point with the limited distribution (Table 25) and over-door transfer fan systems (Table 26). The performance of the north-facing bedrooms (bedroom 2 and the master bedroom) improved during non-ASHRAE conditions. Again, the reduced flow rate of the low airflow system did not appear to impact its performance compared to that of the typical airflow system. During the operation of the transfer fan system, the whole house had a reduced performance, with a high rate of failure in the first-floor spaces, consistent with behavior during ASHRAE conditions.

Table 25. Low Airflow Volume Ducted Distribution to the Bedrooms with Doors Closed (3.1) During Heating Mode—Non-ASHRAE Conditions

	Room	Living	Dining	Breakfast	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Room-to-Thermostat Set Point Temp. Difference	Largest positive difference (°F)	0.8	2.9	0.8	1.0	1.6	0.0	7.9	5.0	0.7	1.0
	Largest negative difference (°F)	–1.1	–0.6	–1.3	–0.8	–0.6	–2.1	–3.1	–0.4	–2.3	–1.2
	Percentage of time outside ±2°F band	0.0	8.4	0.0	0.0	0.0	0.7	60.8	35.6	1.1	0.0

Table 26. Over-Door Powered Transfer Fans to the Bedrooms with Doors Closed (4.5) During Heating Mode—Non-ASHRAE Conditions

	Room	Living	Dining	Breakfast	Laundry	2nd-Floor Landing	Master Bedroom	Bedroom 4 (SW)	Bedroom 3 (W)	Bedroom 2 (NW)	Master Bath
Room-to-Thermostat Set Point Temp. Difference	Largest positive difference (°F)	4.3	5.0	4.4	4.5	5.8	5.4	7.8	8.2	4.6	6.1
	Largest negative difference (°F)	–2.5	–3.6	–2.7	–3.0	–2.2	–4.2	–6.2	–3.6	–6.0	–4.3
	Percentage of time outside ±2°F band	27.8	40.6	41.7	70.1	15.2	73.6	96.3	61.0	88.8	84.5

Several conclusions can be drawn from the summary graphic, Figure 10. Bedroom 2 and the master bedroom/bath continue to have significant comfort failure with the transfer fan system because the supply air is entering the opposite end of the hall, better conditioning bedroom 3. Bedroom 4 continues to suffer the most severe comfort failure due to its southern glazing.

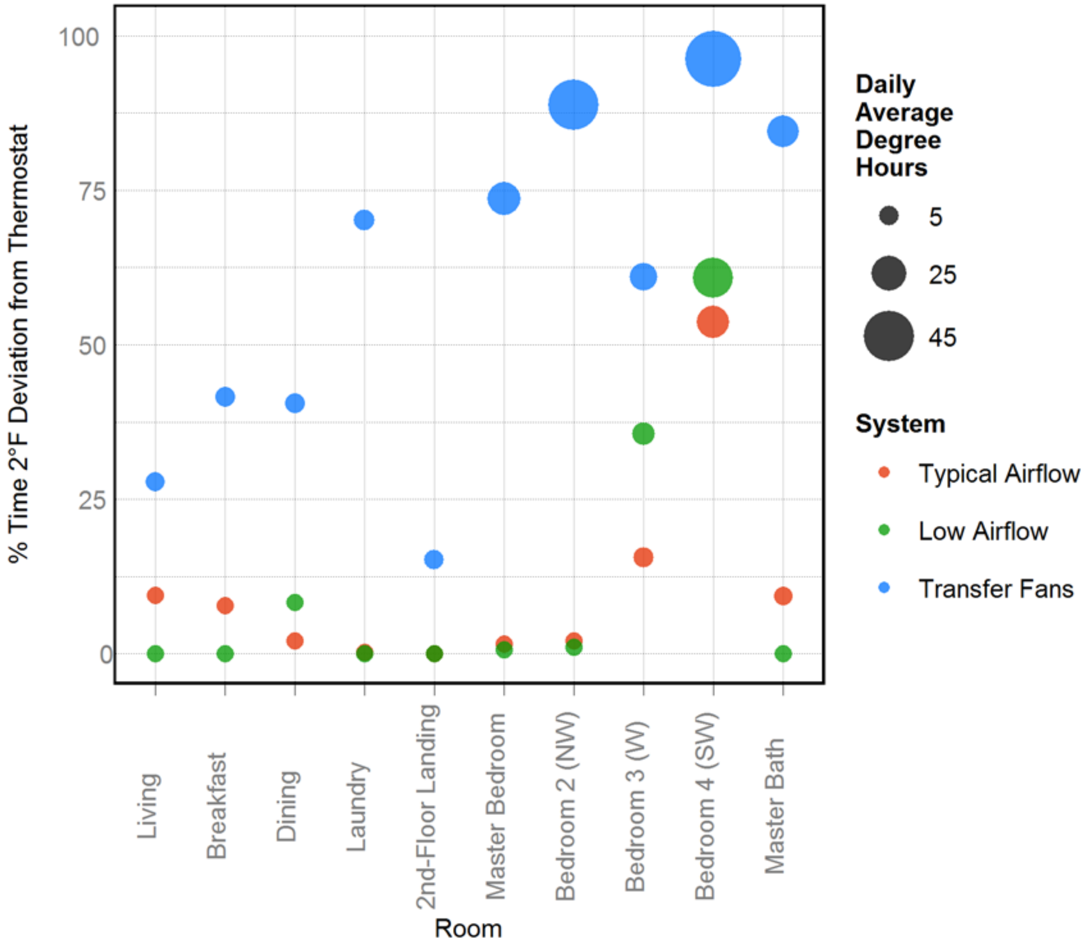

Figure 10. Percentage of failure time, and extent in degree hours, by room for non-ASHRAE conditions

5 Discussion

In general, the standard airflow volume distribution system routed to all spaces requiring conditioning, with two thermostatically controlled zones, was effective at meeting the ACCA Manual RS (Rutkowski 1997) and ASHRAE Standard 55-2010 (ASHRAE 2010a) guidelines. It is notable that the system had difficulty in meeting both sets of guidelines in the upstairs bedrooms. The isolation between the bedrooms and the remainder of the house—caused by the closed bedroom doors—prevented the thermostat from responding to unique solar and internal gains. Additionally, the isolation allowed over-conditioning of the rooms whenever the loads in a room were less than the loads on the rest of the house. That the low airflow system did better than the typical airflow system in those bedrooms during heating mode could be attributed to the reduced load as a result of solar gains in those rooms that did not occur in the remainder of the house. This is consistent with the behavior of the two systems in cooling mode, where the bedrooms were consistently cooler with the typical airflow system than with the low airflow system. As a result, the rooms with high solar gains more frequently met ACCA Manual RS guidelines (Rutkowski 1997); however, the rooms with low solar gains more frequently failed the guidelines. This brings up the inherent compromise of a space conditioning system that does not provide room-by-room control. The question remains: Can a system capable of providing variable air volume to the bedrooms use an algorithm to optimize its response such that one system and one control can simultaneously satisfy rooms experiencing different fractions of their peak loads?

The system using open bedroom doors in conjunction with the single register in the upstairs hallway helped confirm expectations that the door opening could allow a substantial amount of energy transfer between the actively conditioned hallway and the non-actively conditioned bedrooms. Although not presented in this report due to scarcity of data, the results do point toward this conclusion.

The relative success of all systems in meeting the ASHRAE 55-2010 Section 5.2.5 standards (ASHRAE 2010a) can be attributed primarily to the quality of the thermal enclosure. The only remaining challenge is the impact of direct solar gains on occupant comfort. Although the solar heat gain coefficient values for the windows in this house were relatively low (0.22), solar gain was still significant. The potential occupant discomfort in individual rooms due to solar gains must be balanced against the windows' apparent wintertime energy benefits in whole-house models.

6 Conclusions

Two research questions to be answered by this report are as follows:

- To what extent do alternative space conditioning distribution strategies meet ACCA and ASHRAE guidelines for room-to-room temperature uniformity and stability, respectively? Under what weather conditions do indoor temperatures rise above or fall below the guidelines?
- How substantially do the alternative strategies differ from typical ducted forced-air systems in their ability to meet the ACCA and ASHRAE guidelines?

As shown in Section 4.1, failures according to ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time) (ASHRAE 2010a) for all four systems in the Pittsburgh test house tended to be cyclic in nature. The frequency of failure and the rooms that failed varied from system to system. Failures due to drifts or ramps greater than 15 min were rare in most cases; however, the research team observed that drifts or ramps were a result of specific bedrooms having different load profiles than those of the rest of the house. The transfer fan system was most successful in the bedrooms at the expense of frequent cyclic failure in the second-floor landing and living room, whereas the typical and low airflow systems were most successful at maintaining a high passing rate in all rooms throughout the house.

As shown in Section 4.2, during heating mode, all systems failed—at least at some point—to meet ACCA Manual RS guidelines (Rutkowski 1997) for room temperature variation from the thermostat set point. The type of failure was indicative of each system type, with the typical airflow system both over- and underheating spaces, whereas the low volume and transfer fan systems consistently underheated the bedrooms. The failure of the traditional system to adequately condition the space highlights the difficulty low-load homes present for HVAC designers. Reduced envelope load results in a greater impact from irregular internal gains and solar loads.

During midseason heating conditions, all systems showed improved performance with smaller room-to-thermostat temperature differences and typically a lower failure rate. There was a relative lack of acceptable data for the cooling mode analysis; however, data were available for the typical airflow system, which showed a substantial improvement compared to heating mode, with a failure rate of no more than 13%. This can be attributed to the different criteria used in ACCA Manual RS (Rutkowski 1997) for cooling mode room-to-thermostat analysis: ±3°F instead of the ±2°F specified for heating mode analysis.

The typical airflow system had the best ability to meet the ACCA guidelines and the ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time) requirements (ASHRAE 2010a). The low airflow system performed competitively enough to be worthy of future consideration. Although the transfer fan system was effective at meeting the requirements in the bedrooms, it did this at the expense of overconditioning the second-floor landing. The success of leaving the bedroom doors open with no distribution system is indicative of one factor that may impact comfort complaints in homes with real occupants: leaving the interior doors open can help compensate for an ineffective distribution system.

Ongoing research is focused on translating the measured results into a multizone energy model that can accurately compare many systems and control strategies. The results from this model will direct researchers toward the ideal control and air delivery strategies for a low-load home in order to balance efficient system operation with comfort in rooms with highly variable loads.

Cutting-edge builders are achieving thermal enclosure performance that may make possible systems that use less distribution ductwork. This report offers insight into some strategies for reducing the amount of distribution ductwork. The report also provides heretofore unmeasured data for the range and frequency of potential thermal discomfort that occupants may experience when using each of these strategies. Thus, builders now can use this information to discuss space conditioning options with their clients to determine the level of potential discomfort the occupants are willing to accept to have a cost-optimized, cutting-edge, energy-efficient house.

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