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Simplified Space Conditioning in Low-Load Homes: Results from the Fresno, California, Retrofit Unoccupied Test House

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IBACOS, Inc.

February 2014

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Office of Energy Efficiency and Renewable Energy

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Definitions

ACCA	Air Conditioning Contractors of America
BEopt	Building Energy Optimization (software)
Btu	British thermal units
Btu/h	British thermal units per hour
CFD	Computational fluid dynamics
CFM	Cubic feet per minute
HVAC	Heating, ventilation, and air conditioning
RMS	Root mean square

Acknowledgments

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Executive Summary

IBACOS anticipates that houses achieving 50% whole-house source energy savings with respect to the Building America 2010 Benchmark (Hendron and Engebrecht 2010) will be "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). 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 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?
- Where applicable, in what ways and by how much does the measured performance of the distribution system differ from the computer-modeled performance?

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; 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 three systems tested in the Fresno, California, house:

- Typical airflow volume ducted distribution system to the bedrooms and a single point of delivery into the main living space
- Low airflow volume ducted distribution system to the bedrooms and a single point of delivery into the main living space
- No ducted distribution to the bedrooms but over-door and bottom-of-door transfer grilles used to facilitate free movement of air when the bedroom doors are closed and a single point of delivery into the main living space.

The measured performance of the three systems in the Fresno test house with respect to ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time) (ASHRAE 2010a) showed that failures occurring over a time period greater than 15 min were rare for all three distribution systems. Failures tended to be cyclic in nature, with a temperature change greater than ± 2°F over a 15-min interval. Of the three systems, the system with no distribution ductwork to the bedrooms most frequently met these requirements in the bedrooms, whereas the typical airflow volume with ducted distribution to the bedrooms system was most successful in the kitchen, hallway, and hallway bathroom. The performance of the low airflow system ranked between the performance of the typical system and the non-ducted system for these rooms but had the best performance of the three systems in the living room.

Conversely, the system of typical airflow volume with ducted distribution to the bedrooms was the best at meeting ACCA Manual RS guidelines (Rutkowski 1997) for room temperature variation from the thermostat set point, with only the living room showing failures in cooling mode and with all rooms showing the fewest and least extreme failures in heating mode. The system with no distribution ductwork to the bedrooms performed the worst, with a high percentage of failures in the non-actively conditioned bedrooms. In the bedrooms, the non-ducted system failed approximately 50% of the time in cooling mode and almost 100% of the time in heating mode. By contrast, the typical system never failed in cooling mode and failed approximately 10% of the time in heating mode.

The typical airflow volume ducted distribution system to the bedrooms and a single point of delivery into the main living space was the most effective at meeting the ACCA guidelines in the bedrooms, although it did this at the expense of meeting the ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time) requirements (ASHRAE 2010a). It is notable that the system had difficulty in meeting both guidelines in the room that contained the thermostat—the living room. Based on its failure to meet ACCA Manual RS guidelines (Rutkowski 1997), the transfer grilles used with the non-ducted distribution system appeared to be ineffective at enabling heat transfer with the bedrooms via natural convection.

To allow confident use in other houses of the model developed for this study, the root mean square (RMS) error should be approximately 0.5°F to allow confident application of the ACCA Manual RS (Rutkowski 1997) and ASHRAE Standard 55-2010 (ASHRAE 2010a) criteria to the model for use in predicting the performance of over-door and bottom-of-door transfer grilles. The measured performance differs significantly from the modeled performance, with a bedroom RMS error ranging from 1.18°F to 4.51°F, depending on the iteration of the model and the measurement type. Therefore, the model should not be used in other studies without further refinement.

Cutting-edge builders are installing single point space conditioning systems with no means of providing conditioned air to bedrooms except via open doors. This report offers insight into a strategy for providing conditioned air to those rooms when the doors are closed and reports heretofore unmeasured data for the range and frequency of potential thermal discomfort that occupants may experience when this strategy is used. Builders can now 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 2010) will be "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). 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, or portions thereof, are also what traditional ducted forced-air systems can be required to meet (Masco Home Services 2013). 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 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?
- Where applicable, in what ways and by how much does the measured performance of the distribution system differ from the computer-modeled performance?

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 located in Fresno, California; the other is a new construction unoccupied test house located in Pittsburgh, Pennsylvania.

This report focuses on the results of the systems tested in the Fresno, California, retrofit unoccupied test house. Figure 1 shows a front view of the Fresno test house. The west-facing, slab-on-grade, 1,621-ft^2 house includes 2 × 4 wall construction with R-13 cellulose, a vented attic with R-60 blown-in cellulose, windows with a 0.30 U-value and 0.30 solar heat gain coefficient, and rigorous air-sealing measures. The house has well-shaded windows on the east and west sides due to large overhangs and adjacent fences, trees, and buildings.

Figure 1. Front view of the Fresno test house

Based on the specifications implemented during the retrofit of the Fresno test house, the resultant calculated design loads are 16,680 Btu/h in heating and 15,934 Btu/h in cooling. This corresponds to 10.3 Btu/ft^2-h and 9.8 Btu/ft^2-h on a per-unit area basis, respectively. Building Energy Optimization (BEopt) modeling shows energy savings of 50.1%. Full specifications for this house are documented by Stecher and Imm (2013).

The following systems were tested in the Fresno house:

- Typical airflow volume ducted distribution system to the bedrooms and a single point of delivery into the main living space
- Low airflow volume ducted distribution system to the bedrooms and a single point of delivery into the main living space
- No ducted distribution to the bedrooms but over-door and bottom-of-door transfer grilles used to facilitate free movement of air when the bedroom doors are closed and a single point of delivery into the main living space

Table 1 provides details about the three distribution systems outlined above for the Fresno test house. These three distribution systems were implemented in the Fresno test house by using zone dampers controlled by a central data logger. By opening and closing specific dampers, as detailed in Table 1, each distribution system could be operated for a specific duration of time (e.g., 1 week). The zone damper numbers in Table 1 correspond with the labels shown in Figure

2 and Figure 3. The detailed house and system design and commissioning were documented by Stecher and Imm (2013).

Table 1. Distribution System Control Dampers

Zone Dampers	1	2	3	4
System 1 – Typical Airflow	Open	Closed	Open	Closed
System 2 – Low Airflow	Open	Open 5/6 cycles; closed 1/6 cycles	Closed 5/6 cycles; open 1/6 cycles	Closed
System 3 – No Ducted Distribution	Open	Open	Closed	Closed

1.1 Typical Airflow Volume Ducted Distribution to the Bedrooms

Distribution system 1—typical airflow volume ducted distribution system to the bedrooms and a single point of delivery into the main living space—used ductwork sized according to ACCA Manual D (Rutkowski 2009) to provide airflows to meet ACCA Manual J loads (Rutkowski 2006, Stecher and Imm 2013). Figure 2 shows the distribution system layout.

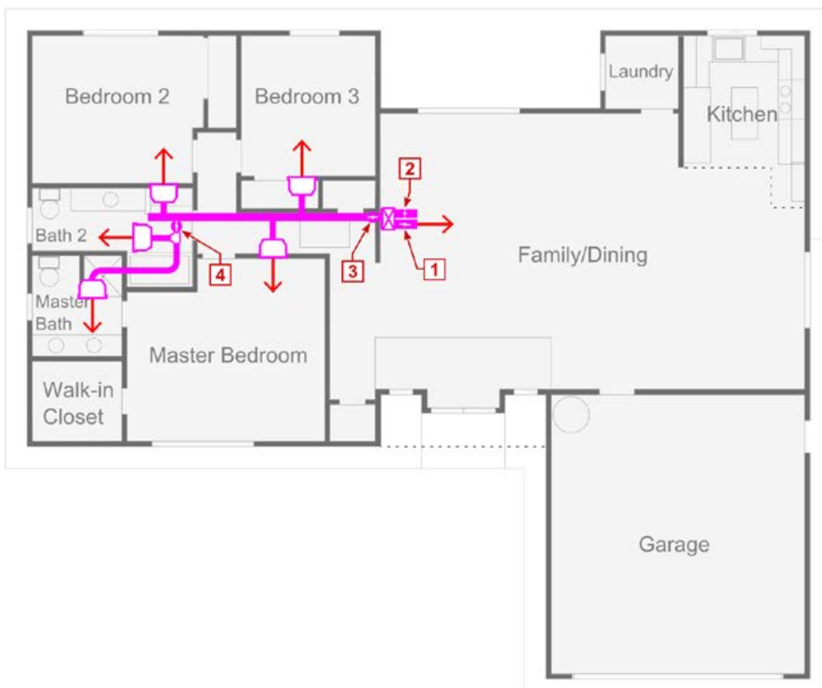

Figure 2. Typical airflow volume ducted distribution to the bedrooms (numbers correspond to zone damper numbers in Table 1)

1.2 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 Fresno house, the amount of ventilation air required for good indoor air quality was 40 CFM per ASRHAE Standard 62.2 (ASHRAE 2010b). Assuming a temperature difference between the supply air and room air of 25°F, the total delivered energy that 40 CFM can deliver is 1,084 Btu/h. This value is approximately 7% of the total peak load of the house (Stecher and Imm 2013), 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 three bedrooms of the house, the difference between the delivered load and the required load would be halved. The remaining portion of the house, which has an open floor plan, could be conditioned by a single supply register. The total ACCA Manual D (Rutkowski 2009) specified airflow for the three bedrooms is 235 CFM. Although still approximately six 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.

This concept was developed after the test air handler unit and ductwork were already installed; thus, installing a reduced-capacity set of ductwork was not an option. Instead, to deliver the appropriate amount of airflow, the damper controlling the ductwork to the bedrooms was opened only once every six air handler unit cycles (#3 in Figure 3). This operation schedule—shown fractionally in Table 1 and graphically in Figure 3—was intended to ensure that the equivalent of 40 CFM would be delivered to the bedrooms for a given day of operation.

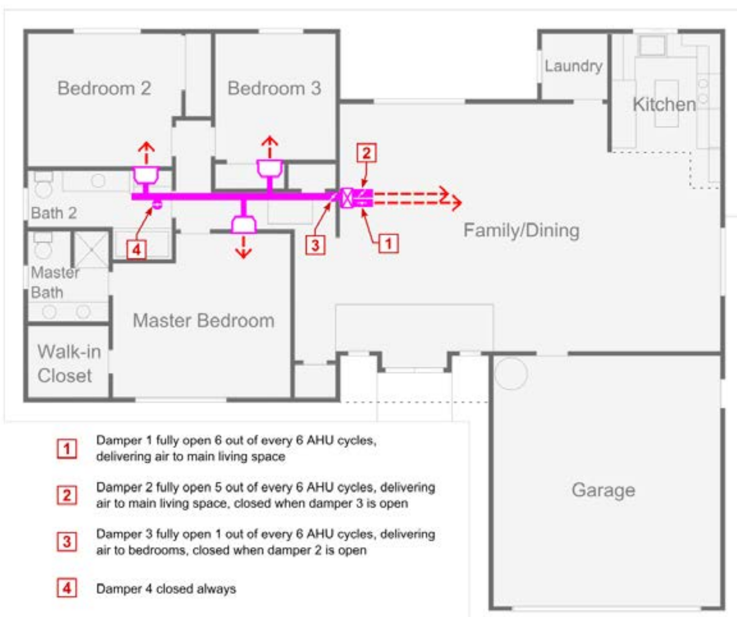

Figure 3. Low airflow volume ducted distribution to the bedrooms (numbers correspond to zone damper numbers in Table 1)

1.3 No Ducted Distribution to the Bedrooms

Distribution system 3—no ducted distribution to the bedrooms but over-door and bottom-of-door transfer grilles used to facilitate free movement of air when the bedroom doors are closed and a single point of delivery into the main living space—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 (CFD) 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 300–600 Btu/h through open doors and 0.3–0.6 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.

However, if interior partition doors are not always open, the potential for heat transfer to occur via natural convection can be reduced. Fortunately, Emery (1969) found that when a small, rectangular opening of height h is placed in a wall at the ceiling plane and one is placed at the floor plane on the order of $h/H \geq 0.05$—where $h = h_1 = h_2$, h_1 is measured down from the ceiling for the upper opening, h_2 is measured up from the floor for the lower opening, and H is the floor-to-ceiling height—no loss in flow occurs compared to the baseline case of a single clear opening of height H in the wall spanning from the floor to the ceiling (see Figure 4).

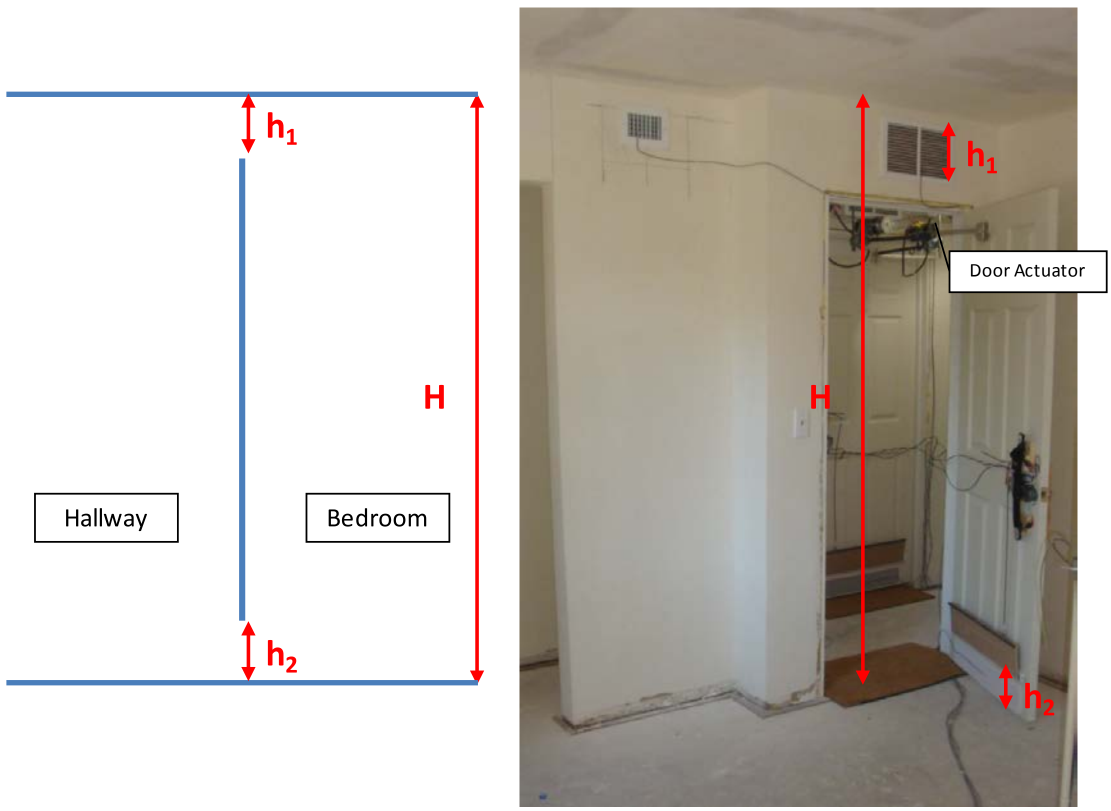

Figure 4. Implementation of the Emery (1969) study (left) in the Fresno test house (right)

This assumes the width of all openings is the same in both cases. Based on the findings from this work, conditioned air was delivered to a single location in the Fresno test house—the main living space—and a market-available, acoustically dampened, and light-shielding through-wall transfer grille was strategically located above and in the bottom of each bedroom door to enable airflow via natural convection from the main living space to the bedrooms (shown in Figure 4 and as green boxes with arrows in Figure 5), even when the bedroom doors are closed. Key changes from the Emery study as a result of implementation of off-the-shelf components include h_1 does not equal h_2, the opening widths are not equal, and the openings are not clear due to light and noise baffles and grilles. (The numbers on Figure 5 correspond to the zone damper numbers in Table 1.) Full specifications were documented by Stecher and Imm (2013).

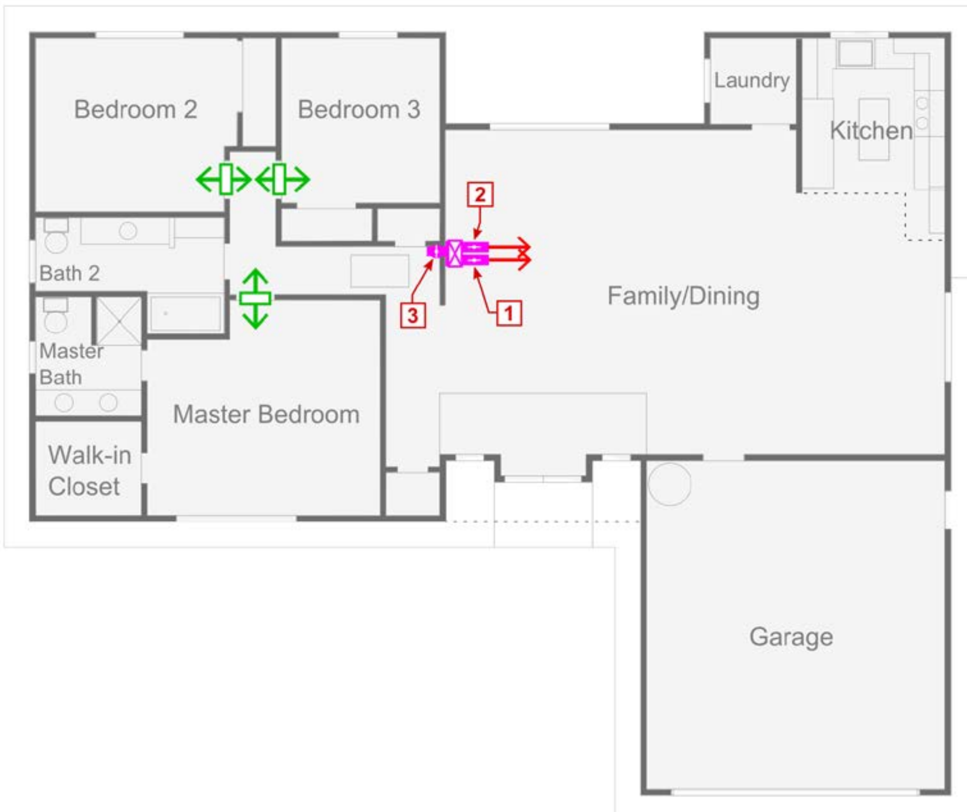

Figure 5. No ducted distribution to the bedrooms (numbers correspond to zone damper numbers in Table 1)

2 Field Test Methods

The research team performed field tests for 1 year in the Fresno retrofit unoccupied test house to determine the extent to which each of the systems did or did not meet established guidelines. To ensure the collection of data for each system throughout the entire year, each system was operated for 1 week at a time before switching to the next system. IBACOS 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 bare minimum requirement that a heating, ventilation, and air conditioning (HVAC) system must meet. Assuming some systems in this study are shown to be non-viable, future studies can focus on assessing the ability of the remaining systems to meet other relevant comfort criteria. Bedroom doors were closed during all system tests unless otherwise mentioned.

2.1 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 experiment 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 house 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 house (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. Although ASHRAE Standard 55-2010 also provides guidance on room-to-room temperature differences, the ACCA standard was chosen because it was more straightforward to use for the data collected in this study.

ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time) 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).

To assess the criteria outlined in ASHRAE Standard 55-2010 Section 5.2.5, Section 7.3.2 (Temperature Cycles and Drifts) and Section 7.4 (Measuring Conditions) are provided in the standard (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). In this case, the climate of Fresno, California, is predominantly sunny throughout the year, even in winter; therefore, the exception allowing for conditions representative of the climate was followed.

Section 7.4 (Measuring Conditions) of ASHRAE Standard 55-2010 (ASHRAE 2010a) requires only two hours of data collection to assess system performance. In this study, the analysis was performed 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). For contrast, the analysis also was performed 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).

2.2 Instrumentation Setup

To assess the performance of the three distribution systems with respect to the ACCA Manual RS (Rutkowski 1997) and ASHRAE Standard 55-2010 (ASHRAE 2010a) guidelines, instrumentation was installed throughout the Fresno test house. The research team installed shielded, aspirated thermocouples in each room of the house to measure air temperature (Figure 6). This style of aspirated thermocouple is commonly used in Building America field tests and has been shown to accurately measure dry bulb temperature. The construction of the shield consists of an inner piece and outer piece of PVC pipe, with foil adhered to the adjacent faces acting as a shield. A small fan positioned above the pipe draws air upward and past the thermocouple, further reducing the error associated with direct sunlight on the shield. Supply temperature measurements were made using unshielded thermocouples. Although no analysis was conducted to ensure the air velocity is adequate to mitigate any radiation exchange with duct surfaces, this effect is assumed to have minimal impact on the results of the research. Measurement of the outdoor conditions comprised temperature, humidity, and incident solar radiation. Measurements from 20-s scans were averaged over 1 min and recorded (Stecher and Imm 2013). The data used for this analysis were collected from the systems installed in the Fresno retrofit unoccupied test house from June 1, 2012, to March 1, 2013.

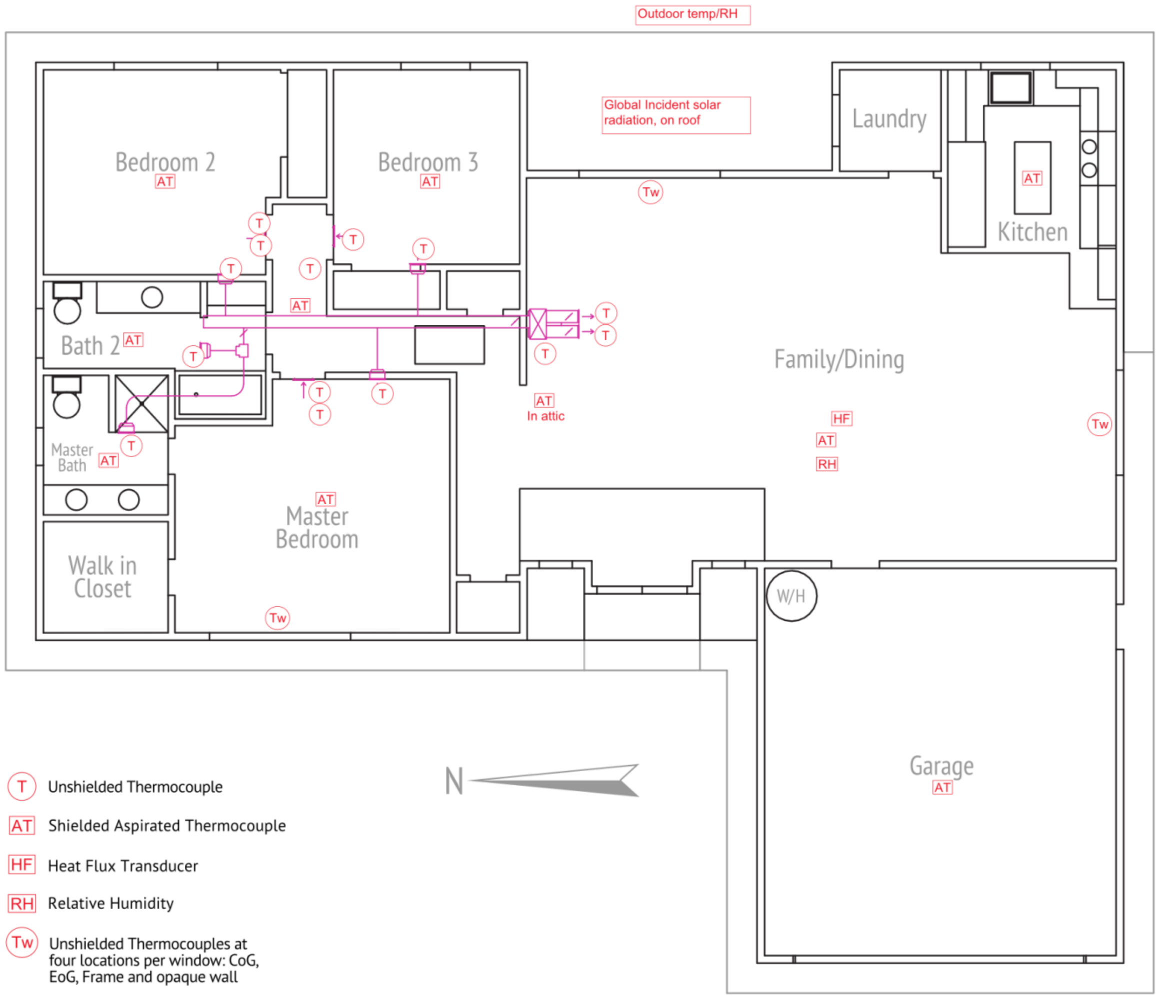

Figure 6. Sensor locations

3 Field Test Results

Each system of the Fresno test house was operated for 1 week at a time throughout the course of a year to capture system operation in each season. Measured temperature data from each room were analyzed to determine the capability of the system in meeting ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time), Section 7.3.2 (Temperature Cycles and Drifts), and Section 7.4 (Measuring Conditions) (ASHRAE 2010a) and by ACCA Manual RS (Rutkowski 1997) as described in Section 2.1 of this report.

3.1 ASHRAE Standard 55-2010 Analysis

Results from the ASHRAE Standard 55-2010 analysis are shown in Table 2 through Table 7. For each design condition—ASHRAE heating, ASHRAE cooling, and all other conditions—there is a table for each operational mode (typical airflow, low airflow, 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.

In cooling mode, the typical airflow system had a high passing rate for the kitchen, hallway, master bathroom, and hallway bathroom. The primary mode of failure for the other rooms was cyclic, with fluctuations of greater than 2°F occurring every 15 minutes (Table 2). The low airflow system had a high passing rate for the master bathroom, east bedroom, and northeast bedroom. All other rooms had predominantly cyclic failure, although the kitchen showed a greater than 10% failure rate under the 30-minute ramp or drift mode (Table 3). During the operation of the no ducted distribution system, the master bedroom, master bath, east bedroom, and northeast bedroom had zero failures in any of the modes defined by ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time). However, the living room failed due to cyclic variation 100% of the time, and the kitchen, hallway, and hallway bathroom also had frequent failure rates due to cycles induced by the operation of the HVAC system (Table 4). Failures via 60-, 120-, or 240-min drifts or ramps were rare to non-existent.

Table 2. Failure Mode Cooling Typical Airflow—2 Days

			Kitchen	Family/ Dining	Hallway	Bath 2	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom	Master Bath
Pass %			89.5	0	98.7	100	36.7	61	1.1	100
Fail %	Mode	Cyc.	10.5	93.5	1.3	0	58.9	39	98.1	0
		30	0	6.2	0	0	4.4	0	0.7	0
		60	0	0.3	0	0	0	0	0	0
		120	0	0	0	0	0	0	0	0
		240	0	0	0	0	0	0	0	0

Table 3. Failure Mode Cooling Low Airflow—8 Days

			Kitchen	Family/ Dining	Hallway	Bath 2	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom	Master Bath
Pass %			10	5.3	49.7	39	83.4	92.1	55.1	96.8
Fail %	Mode	Cyc.	77.8	93	49.2	60.3	11	7.3	41.6	3.2
		30	12.1	1.6	0.8	0.7	2.9	0	3	0
		60	0.1	0.1	0.3	0	1.8	0.3	0.1	0
		120	0	0	0	0	0.5	0.1	0.2	0
		240	0	0	0	0	0	0	0	0

Table 4. Failure Mode Cooling No Ducted Distribution—25 Days

			Kitchen	Family/ Dining	Hallway	Bath 2	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom	Master Bath
Pass %			0.5	0	30.6	25.1	100	100	100	100
Fail %	Mode	Cyc.	88.6	100	68.4	74.7	0	0	0	0
		30	10.9	0	1.1	0.2	0	0	0	0
		60	0.1	0	0	0	0	0	0	0
		120	0	0	0	0	0	0	0	0
		240	0	0	0	0	0	0	0	0

In heating mode, the typical airflow system had a high passing rate for the master bedroom, master bathroom, and hallway bathroom. The primary mode of failure for the other rooms was cyclic (Table 5). The low airflow system had a high passing rate for the master bedroom, master bathroom, east bedroom, and northeast bedroom. All other rooms had predominantly cyclic failure (Table 6). During the operation of the no ducted distribution system, the master bedroom, master bath, east bedroom, and northeast bedroom had zero failures in any of the modes defined by ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time) (ASHRAE 2010a). Similar to the case of the cooling mode, the living room, kitchen, hallway, and hallway bathroom had frequent failure rates due to cyclic variation in temperature (Table 7).

Table 5. Failure Mode Heating Typical Airflow—19 Days

			Kitchen	Family/ Dining	Hallway	Bath 2	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom	Master Bath
Pass %			0	0.1	13.5	93.8	45.9	19	86.1	99.6
Fail %	Mode	Cyc.	86	90.7	76.9	6.2	48.9	76.4	13.8	0.4
		30	10.8	9	9.2	0	5.1	4.6	0.1	0
		60	3.3	0.2	0.4	0	0.1	0	0	0
		120	0	0	0	0	0	0	0	0
		240	0	0	0	0	0	0	0	0

Table 6. Failure Mode Heating Low Airflow—28 Days

			Kitchen	Family/ Dining	Hallway	Bath 2	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom	Master Bath
Pass %			12.2	12.2	16	33.4	88.5	91.5	92.4	94.5
Fail %	Mode	Cyc.	85.8	86.6	78.7	63.8	6.9	7.2	7.4	5.3
		30	1.6	1.2	4.9	2.8	2.1	0.8	0.2	0.2
		60	0.4	0	0.4	0	2	0.2	0	0
		120	0	0	0	0	0.5	0.3	0	0
		240	0	0	0	0	0	0	0	0

Table 7. Failure Mode Heating No Ducted Distribution—20 Days

			Kitchen	Family/ Dining	Hallway	Bath 2	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom	Master Bath
Pass %			1	1	2.6	35.6	100	100	100	100
Fail %	Mode	Cyc.	98.7	98.9	92.1	61.6	0	0	0	0
		30	0.2	0	5	2.8	0	0	0	0
		60	0	0	0.3	0	0	0	0	0
		120	0	0	0	0	0	0	0	0
		240	0	0	0	0	0	0	0	0

In addition to the analysis performed during the conditions called out by ASHRAE Standard 55-2012 Section 7.4 (Measuring Conditions) (ASHRAE 2010a), the analysis also was performed 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). These data are tabulated for each system for each mode: cooling or heating, H_1.

As shown in Table 8 through Table 10 for the cooling mode, the typical airflow system had a greater than 80% passing rate for the kitchen, hallway, master bathroom, and hallway bathroom. The primary mode of failure for the other rooms was cyclic (Table 8). The low airflow system had a greater than 80% passing rate for the master bathroom, east bedroom, and northeast bedroom. The hallway and master bedroom were slightly less than 80% passing, and most of their failures were due to cyclic variation. The kitchen and living room failed most of time due to cyclic variation (Table 9). During the operation of the no ducted distribution system, the master bedroom, master bath, east bedroom, and northeast bedroom had almost zero failures in any of the modes defined by ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time). The living room, kitchen, hallway, and hallway bathroom had frequent failure rates due to cyclic variation in temperature (Table 10).

Table 8. Failure Mode Cooling Typical Airflow—Non-ASHRAE Conditions—19 Days

			Kitchen	Family/ Dining	Hallway	Bath 2	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom	Master Bath
Pass %			91.7	16.3	98.6	100	59.6	77.6	29.1	100
Fail %	**Mode**	**Cyc.**	8.3	76.7	1.4	0	38.1	21.8	69.9	0
		30	0	6.8	0	0	2.4	0.5	1	0
		60	0	0.1	0	0	0	0	0	0
		120	0	0	0	0	0	0	0	0
		240	0	0	0	0	0	0	0	0

Table 9. Failure Mode Cooling Low Airflow—Non-ASHRAE Conditions—27 Days

			Kitchen	Family/ Dining	Hallway	Bath 2	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom	Master Bath
Pass %			35.8	32.8	73.8	55.7	88.5	92.3	78.6	91.2
Fail %	**Mode**	**Cyc.**	56.4	66	25.6	43.8	10.5	7.5	19.9	8.3
		30	7.7	1.2	0.6	0.6	0.7	0.2	1.4	0.5
		60	0.1	0	0	0	0.3	0	0	0
		120	0	0	0	0	0	0	0	0
		240	0	0	0	0	0	0	0	0

Table 10. Failure Mode Cooling No Ducted Distribution—Non-ASHRAE Conditions—83 Days

			Kitchen	Family/ Dining	Hallway	Bath 2	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom	Master Bath
Pass %			33.2	31.7	51	48.6	100	99.9	100	100
Fail %	Mode	Cyc.	59.4	68.3	47	50.2	0	0.1	0	0
		30	7.4	0	2	1.2	0	0	0	0
		60	0	0	0	0	0	0	0	0
		120	0	0	0	0	0	0	0	0
		240	0	0	0	0	0	0	0	0

As shown in Table 11 through Table 13 for the heating mode, the typical airflow system had a greater than 90% passing rate for the master bedroom, master bathroom, and hallway bathroom. The primary mode of failure for the other rooms was cyclic (Table 11). The low airflow system had a greater than 90% passing rate for the master bedroom, master bathroom, east bedroom, and northeast bedroom. All other rooms were closely divided between passing and cyclic failure (Table 12). During the operation of the no ducted distribution system, the master bedroom, master bath, east bedroom, and northeast bedroom had zero failures in any of the modes defined by ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time). The living room, kitchen, hallway, and hallway bathroom had frequent failure rates due to cyclic variation in temperature (Table 13). In all system operation modes, the rate of failure was lower than that observed during ASHRAE Standard 55-2010 Section 7.4 (Measuring Conditions) guidelines.

Table 11. Failure Mode Heating Typical Airflow—Non-ASHRAE Conditions—26 Days

			Kitchen	Family/ Dining	Hallway	Bath 2	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom	Master Bath
Pass %			15.6	16.8	38.6	99.7	66.6	54.8	98.3	99.6
Fail %	Mode	Cyc.	74.4	74.6	55.1	0.3	31.9	43	1.7	0.4
		30	9.6	8.6	6.3	0	1.5	2.2	0	0
		60	0.5	0	0	0	0	0	0	0
		120	0	0	0	0	0	0	0	0
		240	0	0	0	0	0	0	0	0

Table 12. Failure Mode Heating Low Airflow—Non-ASHRAE Conditions—39 Days

			Kitchen	Family/ Dining	Hallway	Bath 2	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom	Master Bath
Pass %			39.5	39.9	46.5	66.7	92.6	91.7	92.5	95.8
Fail %	Mode	Cyc.	59.4	59.4	49.5	32	6.4	7.5	7.2	4.2
		30	1	0.7	3.9	1.3	0.9	0.8	0.2	0
		60	0.1	0	0.1	0	0.1	0	0	0
		120	0	0	0	0	0	0	0	0
		240	0	0	0	0	0	0	0	0

Table 13. Failure Mode Heating No Ducted Distribution—Non-ASHRAE Conditions—37 Days

			Kitchen	Family/ Dining	Hallway	Bath 2	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom	Master Bath
Pass %			33.3	33.4	40	66.9	100	100	100	100
Fail %	Mode	Cyc.	66.7	66.6	56.5	32.5	0	0	0	0
		30	0	0	3.5	0.6	0	0	0	0
		60	0	0	0	0	0	0	0	0
		120	0	0	0	0	0	0	0	0
		240	0	0	0	0	0	0	0	0

In general, the data collected during the conditions meeting ASHRAE Standard 55-2010 Section 7.4 (Measuring Conditions) included time periods with greater duration and frequency of HVAC system operation, with correspondingly greater failure rates due to cyclic temperature movement than the data collected during time periods that did not meet the measurement standard. In all cases, rooms with no active conditioning tended to most frequently meet the cycles, drifts, and ramps standard (Table 14). When the peak load of the entire house was supplied into the living space, it caused a high rate of failure not only in that space but also in immediately adjacent spaces connected by open doorways, such as the hallway and the hallway bathroom. The performance of the low airflow system is between the performance of the typical airflow system and the no ducted distribution system, consistent with its operation methodology of delivering less conditioned air to the bedrooms than the typical airflow system. Failures due to drifts or ramps 30 min or greater were rare in all cases. During conditions not meeting ASHRAE Standard 55-2010 Section 7.4 (Measuring Conditions), all systems had a higher passing rate (Table 15).

Table 14. Passing Rate for All Systems and Modes—ASHRAE Conditions

	Mode	System	Kitchen	Family/ Dining	Hallway	Bath 2	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom	Master Bath
ASHRAE Conditions	Cooling	Typical Airflow	89.5	0	98.7	100	36.7	61	1.1	100
	Cooling	Low Airflow	10	5.3	49.7	39	83.4	92.1	55.1	96.8
	Cooling	No Ducted Distribution	0.5	0	30.6	25.1	100	100	100	100
	Heating	Typical Airflow	0	0.1	13.5	93.8	45.9	19	86.1	99.6
	Heating	Low Airflow	12.2	12.2	16	33.4	88.5	91.5	92.4	94.5
	Heating	No Ducted Distribution	1	1	2.6	35.6	100	100	100	100

Table 15. Passing Rate for All Systems and Modes—Non-ASHRAE Conditions

	Mode	System	Kitchen	Family/ Dining	Hallway	Bath 2	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom	Master Bath
Non-ASHRAE Conditions	Cooling	Typical Airflow	91.7	16.3	98.6	100	59.6	77.6	29.1	100
	Cooling	Low Airflow	35.8	32.8	73.8	55.7	88.5	92.3	78.6	91.2
	Cooling	No Ducted Distribution	33.2	31.7	51	48.6	100	99.9	100	100
	Heating	Typical Airflow	15.6	16.8	38.6	99.7	66.6	54.8	98.3	99.6
	Heating	Low Airflow	39.5	39.9	46.5	66.7	92.6	91.7	92.5	95.8
	Heating	No Ducted Distribution	33.3	33.4	40	66.9	100	100	100	100

3.2 ACCA Manual RS Analysis

Results from the ACCA Manual RS (Rutkowski 1997) analysis using the methodology described in Section 2.1 are shown in Table 16 through Table 21. Again, there is one table for each system (typical airflow, low airflow, and no ducted distribution). For consistency of data, the analysis was performed only on days that met the ASHRAE Standard 55-2010 Section 7.4 (Measuring Conditions) test conditions standard (ASHRAE 2010a).

In cooling mode, the typical airflow system maintained all rooms except the living room within ± 3°F of the temperature at the thermostat. Room-to-room temperature differences are all within ACCA Manual RS (Rutkowski 1997) standards (Table 16). The low airflow system showed greater extremes in temperature and had a higher percentage of time outside the ± 3°F band for the living room. The northeast bedroom also spent 64% of the time outside the acceptable range. Room-to-room temperature differences were more extreme and more frequent in occurrence, with all rooms spending some time outside the ± 6°F band (Table 17). The no ducted distribution system showed the greatest extremes in temperature, and the percentage of time outside the ± 3°F band increased for all rooms compared to the low airflow system. The northeast bedroom again had the greatest duration of time outside the acceptable range at 87%, along with the most extreme temperature: 8.7°F above the thermostat set point. Room-to-room temperature differences were also more extreme and more frequent in occurrence, with all rooms spending at least 30% of the time outside the ± 6°F band (Table 18). The apparent effect of outdoor temperature is unclear, with no obvious patterns and extreme indoor conditions occurring for outdoor temperatures between 70°F and 108°F.

Table 16. Typical Airflow Volume System During Cooling Mode

		Kitchen	Family/ Dining	Hallway	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom
Room-to-Thermostat Set Point Temperature Difference	Largest positive difference (°F)	0.9	0.5	2.8	2.8	0.8	0.8
	Largest negative difference (°F)	–2.7	–4.5	–0.8	–2.1	–2.6	–3.1
	Percentage of time outside ± 3°F band	0	31	0	0	0	0
Room-to-Room Temperature Difference	Average (°F)	–0.3	–1.5	1.5	1.2	–0.3	–0.6
	Maximum (°F)	–2.6	–4.8	4.8	4.3	–3.0	–3.6
	Percentage of time outside ± 6°F band	0	0	0	0	0	0

Table 17. Low Airflow System During Cooling Mode

		Kitchen	Family/ Dining	Hallway	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom
Room-to-Thermostat Set Point Temperature Difference	Largest positive difference (°F)	1	0.7	3.4	8.3	5.9	4.1
	Largest negative difference (°F)	–3.6	–6.1	–1.2	–1.3	–2.4	–2.6
	Percentage of time outside ± 3°F band	5	45	0	64	13	4
Room-to-Room Temperature Difference	Average (°F)	–2.2	–3.6	0.4	3.5	1.6	0.3
	Maximum (°F)	–9.9	–13.8	–8.7	13.8	11.5	9.4
	Percentage of time outside ± 6°F band	25.4	40.2	6.6	40.9	28.3	15.2

Table 18. No Ducted Distribution System During Cooling Mode

		Kitchen	Family/ Dining	Hallway	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom
Room-to-Thermostat Set Point Temperature Difference	Largest positive difference (°F)	1.7	0.4	3.4	8.7	6.9	4.8
	Largest negative difference (°F)	–4.3	–6.9	–3.9	–4.3	–3.3	–3.8
	Percentage of time outside ± 3°F band	18	49	2	87	58	24
Room-to-Room Temperature Difference	Average (°F)	–3.2	–4.8	–1.1	4.8	3.2	1
	Maximum (°F)	–12.3	–14.6	–9.8	14.6	12.7	10.1
	Percentage of time outside ± 6°F band	57.8	67.7	36.7	68.1	55.9	35.2

In heating mode, the typical airflow system had infrequent but significant temperature excursions outside the ACCA required ± 2°F of the thermostat set point temperature. An exception to this is the master bedroom, which was outside this temperature range 80% of the time, with a maximum temperature of 6°F below the thermostat set point. Room-to-room temperature differences were significant but infrequent, occurring less than 15% of the time (Table 19). The master bedroom was similarly out of bounds for the low airflow system. The low airflow system also showed greater extremes in temperature and had a higher percentage of time outside the ± 2°F band for all rooms. In particular, the northeast bedroom and the east bedroom spent at least 70% of the time outside the ± 2°F band. Room-to-room temperature differences were more extreme and more frequent in occurrence, with all rooms spending at least 30% of the time outside the ± 4°F band (Table 20). The no ducted distribution system showed the greatest extremes in temperature, and the percentage of time outside the ±3°F band increased for all rooms compared to the low airflow system. The master bedroom, northeast bedroom, and east bedroom were outside the ± 2°F band at least 95% of the time, with temperature extremes of approximately 9°F below the thermostat set point. Room-to-room temperature differences also were more extreme and more frequent in occurrence, with all rooms spending at least 60% of the time outside the ± 4°F band (Table 21). The apparent effect of outdoor temperature is unclear, with no obvious patterns and extreme indoor conditions occurring for outdoor temperatures between 32°F and 62°F.

Table 19. Typical Airflow Volume System During Heating Mode

		Kitchen	Family/ Dining	Hallway	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom
Room-to-Thermostat Set Point Temperature Difference	Largest positive difference (°F)	2.8	2	3.3	2	1.9	0.3
	Largest negative difference (°F)	–3.4	–4.1	–2.6	–4.4	–5.2	–6.0
	Percentage of time outside ±2°F band	41	54	5	7	13	80
Room-to-Room Temperature Difference	Average (°F)	–0.1	–0.8	1.5	0.8	0.4	–1.9
	Maximum (°F)	7.8	–3.2	5.5	4.9	5.3	–5.7
	Percentage of time outside ±4°F band	1.7	0.1	13.1	5.2	4.1	13.8

Table 20. Low Airflow System During Heating Mode

		Kitchen	Family/ Dining	Hallway	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom
Room-to-Thermostat Set Point Temperature Difference	Largest positive difference (°F)	6.2	4.3	3.8	2.3	4.3	4.4
	Largest negative difference (°F)	–4.0	–6.4	–3.5	–7.8	–7.3	–8.4
	Percentage of time outside ± 2°F band	49	56	15	71	81	85
Room-to-Room Temperature Difference	Average (°F)	1.7	1.1	2.3	–1.7	–1.4	–1.9
	Maximum (°F)	12.9	11.8	11	–11.6	–11.7	–12.8
	Percentage of time outside ± 4°F band	38.8	31.8	50.5	44.8	43.5	53.3

Table 21. No Ducted Distribution System During Heating Mode

		Kitchen	Family/ Dining	Hallway	Bedroom 2 (NE)	Bedroom 3 (E)	Master Bedroom
Room to Thermostat Set Point Temperature Difference	Largest positive difference (°F)	5.8	4.7	3.4	–0.7	–1.6	–2.2
	Largest negative difference (°F)	–4.1	–5.1	–4.1	–9.8	–9.3	–9.1
	Percentage of time outside ± 2°F band	61	66	24	96	99	100
Room-to-Room Temperature Difference	Average (°F)	3.6	2.8	3.3	–3.0	–3.2	–3.6
	Maximum (°F)	15	13.5	12.2	–14.5	–14.3	–14.1
	Percentage of time outside ± 4°F band	71.4	63.2	77	75.4	75.7	80

4 Discussion

In general, the typical airflow volume ducted distribution system to the bedrooms and a single point of delivery into the main living space was the most 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 the guidelines in the room that contained the thermostat. Ostensibly, this should be the easiest room in which to meet the requirements because it contains the control; however, this indicates the inherent challenge in supplying sufficient airflow and temperature difference to ensure the room load is met while minimizing the perceived temperature movement by the occupant. This room did not meet the guidelines because the shielded aspirated thermocouple for the living room temperature measurement was able to detect the partially mixed airflow moving through the room. Using a register that allowed more precise airflow placement could have helped to mitigate this problem.

Although the typical airflow system performed almost perfectly in cooling mode, no system was perfectly effective in heating mode. The discrepancy between cooling mode and heating mode performance can be partially attributed to the difference in temperature criteria in ACCA Manual RS (Rutkowski 1997). When the same temperature boundaries are used for each mode, the performance is similar. Why two different temperature boundaries are used for the heating mode and cooling mode is unclear, and its applicability in energy efficient, low-load houses could be a topic of future research.

Additionally, there is no specified amount of time that a room needs to be out of bounds. In this study, which used data averaged every minute, even temperature extremes occurring for a very short period of time would trigger a failure. This prompts the question of the minimum acceptable duration of time that a temperature extreme needs to occur before failing the ACCA Manual RS guidelines (Rutkowski 1997). The application of ASHRAE Standard 55-2010 (ASHRAE 2010a) to a time period greater than that outlined in Section 7.4 (Measuring Conditions) of that standard shows there is a slight risk to researchers in the field of getting a result during a short-term test that is not necessarily representative of the performance of the house as a whole. The study also showed that, in this particular house, there is a lower risk of occupant discomfort during midseason part-load conditions than during conditions closer to peak. This conclusion should be taken with caution because the house in this study had no internal gains and little time-dependent solar gain due to well-shaded windows. Both of these variables are likely to have a significant impact in most houses.

One unexpected result of the method used for operating the low airflow volume ducted distribution system to the bedrooms and a single point of distribution delivery into the main living space was that the system delivered more energy into the bedrooms than originally anticipated. During the system cycles when airflow was provided to the bedrooms, the system ran for approximately 40% longer than during the cycles when no airflow was provided to the bedroom. This increase in delivered energy should be considered when looking at the results for the low airflow system. Although this system yielded sufficient performance in many cases, the methods used to create a low average flow rate—operating the bedroom registers at the standard airflow for a reduced period of time—introduced variations to the data significant enough such that the results from this study should not be used as a generalization of the performance of low airflow systems. The method of operation likely exaggerated the swings in temperature with respect to the ASHRAE 55-2010 analysis (ASHRAE 2010a) and created the opportunity for more significant excursions outside the acceptable ACCA Manual RS values (Rutkowski 1997). This made the performance of the system appear worse than that of what would likely be installed in the field—a low airflow system that delivered a small amount of airflow every system cycle.

The transfer grilles used on the no ducted distribution system appeared to be ineffective at enabling heat transfer with the bedrooms via natural convection. To help verify this conclusion, the doors to the bedrooms were left open for a 2-week period, with the no ducted distribution system operational. The results showed that the bedrooms were almost the same temperature as the hallway and were closer in temperature to the main living space than with the doors closed. However, there remained a significant temperature difference between the bedrooms and the main living space because the cross-sectional area of the hallway acted as a bottleneck to the amount of airflow that could occur via natural convection. Factors that could have influenced the effectiveness of this strategy include the cross-sectional area, shape, and length of the hallway, the placement of the single point of conditioned air supply to the house, and its horizontal throw parameters. The transfer grilles themselves enabled less airflow than previously anticipated due to several possible factors, including the sound-deadening material used inside them and the angle and shape of the fins on the grilles. To determine the energy contribution of the transfer grilles to the bedrooms, a CFD analysis was performed and is documented in this report in Section 5.

The lack of a clear and obvious association of failures with specific outdoor conditions indicates that a time series analysis should be considered in future research. For example, if the most extreme temperature deviations occurred after a very cold night followed by a mild day, there may be a delay in the load reaching a room.

5 Calibration of a Computational Fluid Dynamics Model for Transfer Grille Airflow

To determine the expected flow rate through the over-door and bottom-of-door transfer grilles used in the air distribution strategy with no ductwork to the bedrooms (distribution system 3) at a given temperature difference between the spaces on either side of the transfer grilles, the IBACOS research team created a CFD model. The team then used the field test results to calibrate this model.

5.1 Computational Fluid Dynamics Model Description

The IBACOS team created a finite element CFD model using ANSYS CFX (ANSYS). The purpose of the model was to determine the expected flow rates through the over-door and bottom-of-door transfer grilles at a given temperature difference between the hallway and bedroom. The CFD model consists of a single fluid material (air) encompassing the volume of the hallway, hallway bathroom (henceforth referred to as “bathroom”), and northeast bedroom (henceforth referred to as “bedroom”) of the Fresno retrofit unoccupied test house (see Figure 7). The boundary conditions are represented in the figure as follows: green lines are the exterior walls, gray lines are the interior walls, red lines are the windows, and the curved line is a curved air boundary that allows air to enter and leave the model from the remainder of the house and is where temperature data from the living room space are input as values.

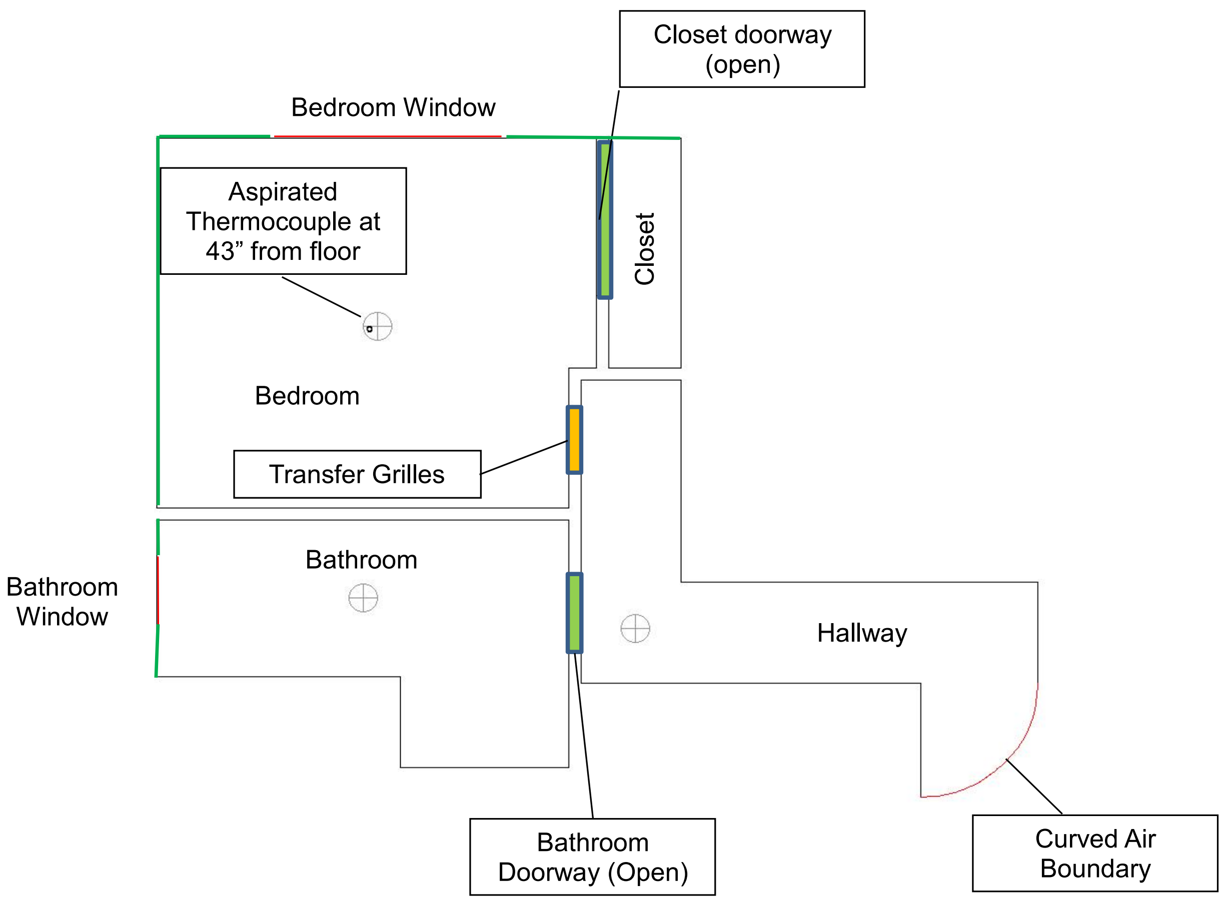

Figure 7. Plan view of the CFD model, showing the upper and lower transfer grilles and fan locations

The model is of only the air; the interior and exterior walls and floor are not included in the model but instead are represented as boundary conditions. The interior partition walls and floor assembly were assumed to be adiabatic in the model. The exterior wall, ceiling, and window boundaries were treated with convection boundary conditions, based on specified R-values (Table 22) and outside air temperatures from monitored data. A curved-face, free-flowing air boundary layer was established at the end of the hallway where it meets the main living space. This type of boundary layer enables convective heat transfer across it while ensuring that the amount of air in the model remains constant. This boundary condition used inputs of temperature data measured in the living room. The mesh was primarily tetrahedral, but hexahedrals, prisms, and pyramids also were used when appropriate. It contains 61,151 nodes and 201,663 elements. The residuals value of the mesh was less than 2×10^{-5}. Mesh size was the smallest at the transfer grille locations (Figure 8) and served as a limit for a computational time step, which was 2 s.

Table 22. Exterior Boundary R-Values and Temperatures

Assembly	R-Value
Exterior Wall	13
Ceiling	60
Window	3
Interior Wall	Adiabatic
Floor	Adiabatic

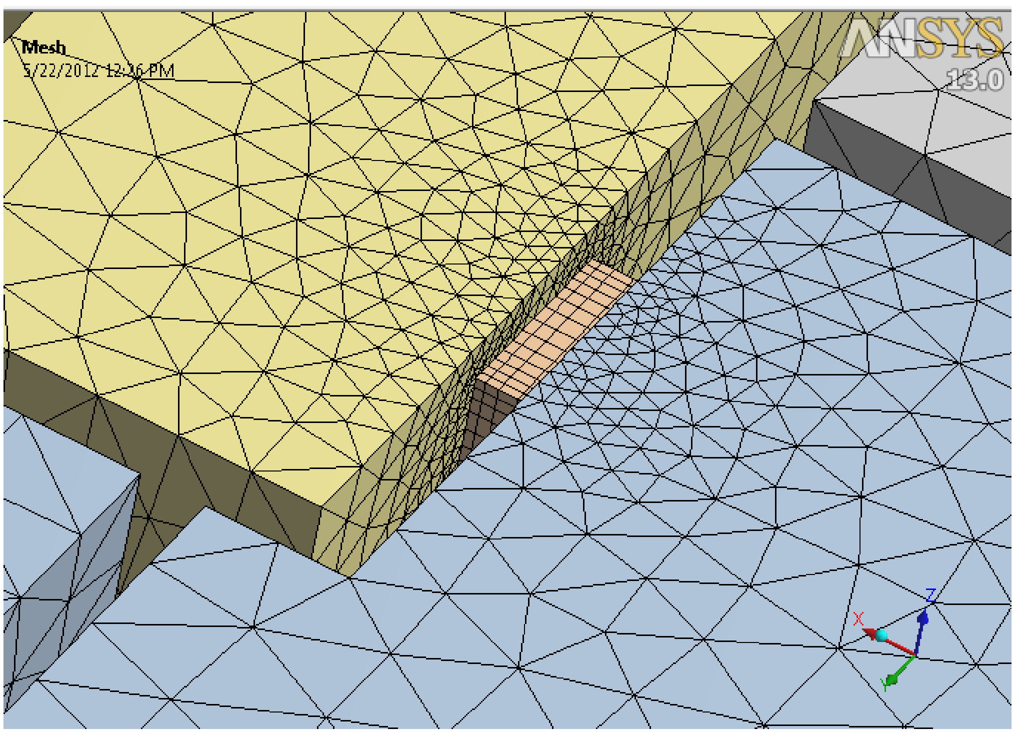

Figure 8. Close-up of the CFD model mesh, showing the upper transfer grille (pink) connecting the bedroom (light blue) and hallway (tan)

5.2 Methods

To refine the model based on measured data, average temperatures were taken for the three air spaces: the hallway, the bathroom, and the bedroom. Point temperature measurements also were taken at the locations coinciding with the measurements used in the actual unoccupied test house. These locations included the center of the bedroom at 43 in. from the floor, the center of the bathroom at 43 in. from the floor, and the hallway at 43 in. from the floor. In the actual test house, each of these measurements was shielded from radiation and was mechanically aspirated. The model also had a temperature data point at the geometric center of each transfer grille to coincide with the unshielded non-aspirated measurements used in the actual house. Data collected on December 2, 2011, for the 12-h period between 12:00 a.m. and 12:00 p.m. were used for all iterations of the model.

Three iterations of the model were performed to assess the influence of two factors: the aspirated thermocouple fan and the transfer grille pressure drop versus flow coefficient. The temperatures at the center of the room, the upper transfer grille, and the lower transfer grille were used as the primary factors for comparison between measured results and modeled performance for each iteration (also referred to as a case). Case 1 was the base case model. In Case 2, to assess the impact of the thermocouple aspiration fan used in the built test house, the research team incorporated into the CFD model an assembly consisting of the radiation shielding tubes and a 7-CFM fan pointing toward the ceiling, and then the team recalculated the results with all other input data, the same as those used in the first run. In Case 3, to assess the impact of the assumed—based on a curve fit to a data point for each transfer grille obtained from the manufacturer—pressure versus flow coefficients, the team evaluated the transfer grilles in a separate CFD model to determine a more accurate pressure drop to flow rate coefficient. The research team then input this value into the main model, which included the fan assembly used in Case 2. In all cases, the interior walls and the floor were assumed to be adiabatic, while the exterior walls, windows, and the ceiling were determined to have the R-values shown in Table 22.

5.3 Results

Three iterations of the model were performed. Their resulting bedroom air temperatures are shown in Figure 9. The bedroom air temperature for the second iteration (Case 2)—which included the aspiration fan—more closely follows measured data than the first iteration. The third iteration (Case 3) differs most from the measured data.

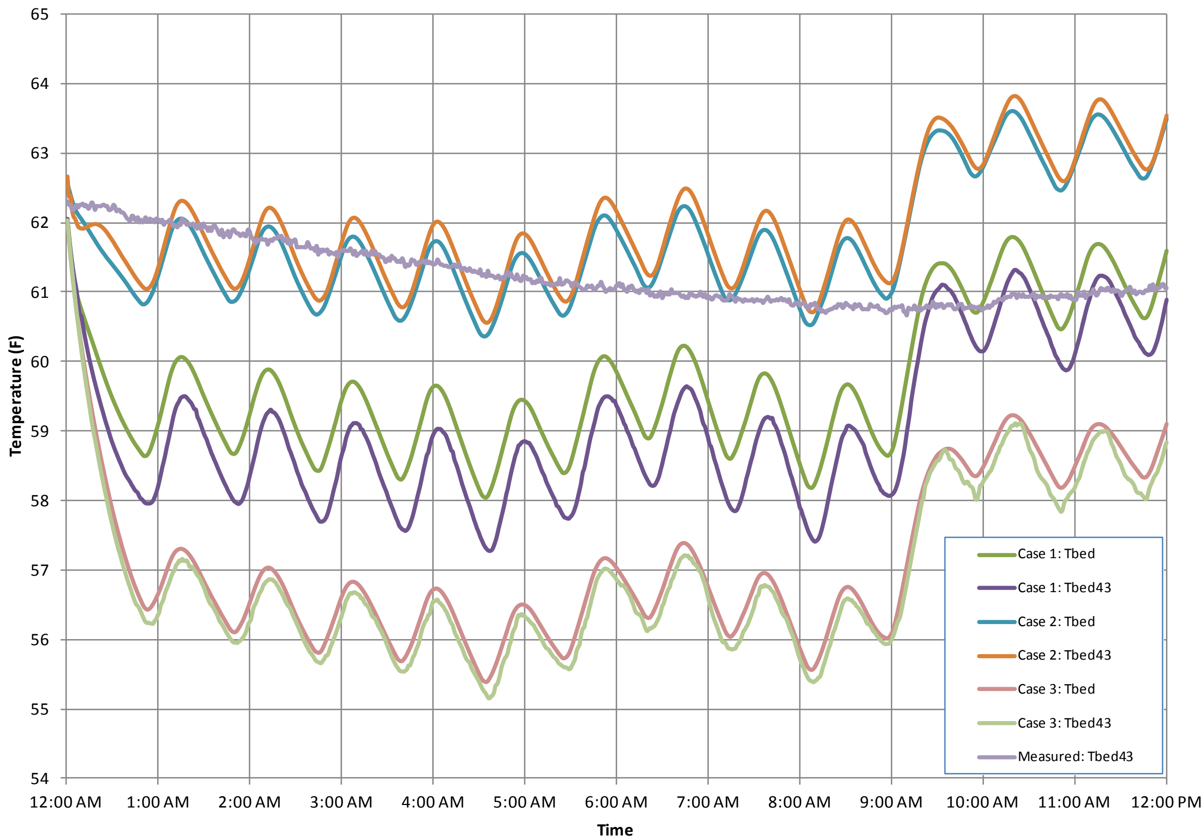

Figure 9. Plot comparing temperatures of the three runs

The room air temperature results also show a difference in the two different methods of determining air temperature in the model. The first is represented in Figure 9 by labels in the legend that do not contain the number 43. These values are determined in the model using the volume averaged air temperature of the room. The labels that do contain the number 43 are values that used a single measurement point in the model meant to emulate the actual sensor location used in the test house. Case 1 shows a larger difference between the two methods than Case 2 or Case 3 shows. Case 2 shows that the single point of measurement is generally slightly warmer than the volume weighted average, whereas Case 1 and Case 3 are cooler. The RMS error for the two measurement types shows that the volume weighted average is generally closer in temperature to the measured data, as shown in Table 23. This table also shows that the RMS error for the volume average room air temperature improved 39% from 1.95°F for Case 1 to 1.18°F for Case 2. A relatively high RMS error remained for the lower transfer grille at 5.58°F (Table 23). This value increased to 7.13°F when the updated flow to pressure coefficient was input into the model in Case 3.

Table 23. RMS Error Values for Modeled Temperatures at Each Location

Location	RMS Error (°F)		
	Case 1	Case 2	Case 3
Bedroom (Volume Average)	1.946	**1.182**	4.328
Bedroom (at 43 In. from Floor)	2.511	**1.260**	4.505
Hallway (Volume Average)	1.513	**0.786**	1.457
Hallway (at 43 In. from Floor)	0.933	**1.237**	0.916
Bathroom (Volume Average)	0.394	0.982	**0.335**
Bathroom (at 43 In. from Floor)	**0.511**	1.987	0.542
Lower Transfer Grille	**5.415**	5.583	7.130
Upper Transfer Grille	0.554	1.222	**0.553**

The change in the flow to pressure difference coefficient reduced the modeled airflow into the room between Case 2 and Case 3 by 39%. Average airflow in Case 2 was approximately 30 CFM at an average temperature difference between the hallway and bedroom of 6.2°F. Average airflow in Case 3 was approximately 22 CFM at an average temperature difference between the hallway and bedroom of 9.6°F. In each of the two cases, the rate of airflow followed the same trend and remained consistent over the 12-h period, varying by approximately 10%. Based on these flow rates and temperature differences, the transfer grilles in Case 2 provided a continuous 270 Btu/h of heat to the room; in Case 3, they provided 228 Btu/h.

5.4 Discussion

The results from the model are telling, in that the amount of energy transported is low, regardless if the flow rate is 30 CFM or 22 CFM. The flow rates also occur at room-to-hallway temperature differences that are outside the bounds of ACCA Manual RS (Rutkowski 1997) heating requirements. These temperature differences also are 3.4 to 5.3 times greater than the 1.8°F that Feist et al. (2005) used to observe between 300 Btu/h and 600 Btu/h of convective heat transfer through an open door. The amount of heat transfer occurring through the transfer grilles via a 1.8°F temperature difference—a temperature difference that also would meet the requirements of ACCA Manual RS—would be correspondingly lower. Thus, this strategy is not viable in buildings where ACCA Manual RS guidelines must be met. Given the low energy transfer rates in the model and the test house, it is not surprising that the aspiration fan had a significant impact on the output of the CFD model. Figure 10 visualizes the whole-room air mixing induced by the 7-CFM fan (i.e., the size of fan typically used in a laptop computer).

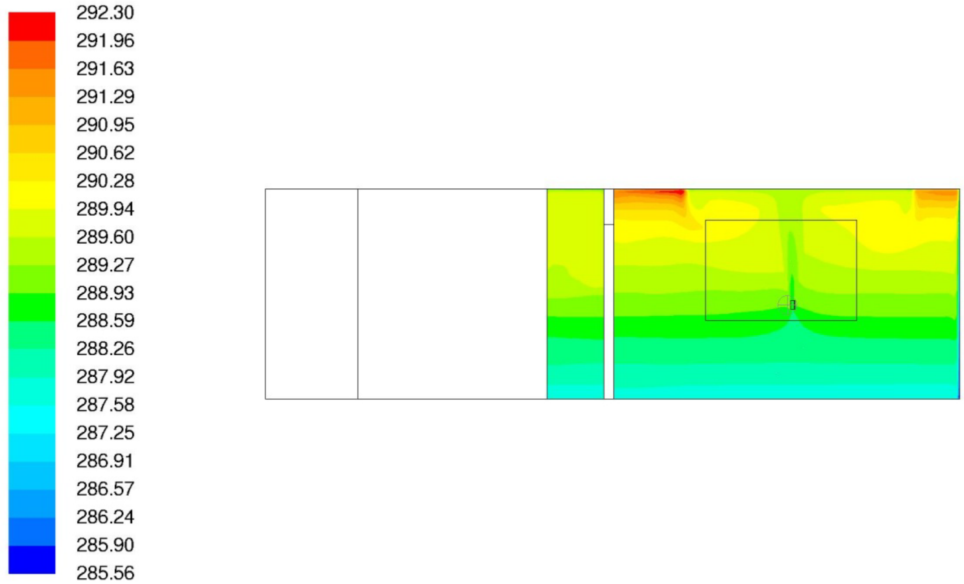

Figure 10. Vertical section view of the bedroom through the fan centerline, showing fan-influenced air movement (scale is air temperature in Kelvin)

Although the Case 2 modeled room air temperature tracks the closest to measured data, the observed RMS values of 1.18°F and the fluctuations observed in Figure 9 indicate an issue remains with some aspect of the model. One possible explanation for the fluctuations may be due to the lack of thermal mass in the model that is in the test house, such as drywall and the concrete floor. Furthermore, that the ostensibly "more accurate" CFD determined pressure versus flow ratio used in Case 3 resulted in a bedroom temperature that was significantly colder than measured could indicate that the CFD results are inaccurate. On the other hand, it could indicate that other factors, such as conduction through the interior partition walls, are significant. The impact of conduction through interior partition walls and thermal mass of the structure should be assessed before using this model elsewhere.

6 Conclusions

The first two research questions answered by the 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 ACCA and ASHRAE guidelines?

As shown in Section 3.1, failures according to ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time) (ASHRAE 2010a) for all three systems tended to be cyclic in nature. Failures due to drifts or ramps greater than 15 min were rare in all cases. All systems had a greater than 85% failure rate in the living room. This was due, in part, to the sensor location and the amount of energy delivered to the space. The no ducted distribution system was most successful in the bedrooms; because no conditioned air was supplied directly to them, they showed no cyclic failure. The typical airflow system was most successful in the kitchen, hallway, and hallway bathroom. The performance of the low airflow system ranked between the performance of the typical airflow system and the no ducted distribution system for these rooms but had the best performance of the three systems in the living room.

As shown in Section 3.2, the typical airflow system was the best at meeting ACCA Manual RS guidelines (Rutkowski 1997) for room temperature variation from the thermostat set point, with only the living room showing failures in cooling mode and with all rooms showing the fewest and least extreme failures in heating mode. The no ducted distribution system performed the worst, with a high percentage of failures in the non-actively conditioned bedrooms. In the bedrooms, this system failed approximately 50% of the time in cooling mode and almost 100% of the time in heating mode. By contrast, the typical airflow system never failed in cooling mode and failed approximately 10% of the time in heating mode.

The typical airflow system was the most effective at meeting the ACCA Manual RS guidelines (Rutkowski 1997) in the bedrooms, although it did this at the expense of meeting the ASHRAE Standard 55-2010 Section 5.2.5 (Temperature Variations with Time) requirements (ASHRAE 2010a). It is notable that the system had difficulty in meeting both guidelines in the room that contained the thermostat—the living room. Based on the system's failure to meet ACCA Manual RS guidelines (Rutkowski 1997), the transfer grilles used with the no ducted distribution system appeared to be ineffective at enabling heat transfer with the bedrooms via natural convection. Despite this, the lack of any failure in the non-actively conditioned bedrooms via drifts, ramps, or cycles during the operation of this system indicates the success of the thermal enclosure at mitigating the effects of external loads placed on the house.

The third research question answered by the report is as follows:

- Where applicable, in what ways and by how much does the measured performance of the distribution system differ from the computer-modeled performance?

The measured performance differs significantly from the modeled performance, with a bedroom RMS error ranging from 1.18°F to 4.51°F, depending on the iteration of the model and the measurement type as shown in Section 5.3 in this report. To reliably use this model, the RMS error should be approximately 0.5°F to allow confident application of the ACCA Manual RS (Rutkowski 1997) and ASHRAE Standard 55-2010 (ASHRAE 2010a) criteria to the model to use for predicting the performance of over-door and bottom-of-door transfer grilles in conjunction with a single point space conditioning system.

Cutting-edge builders are installing single point space conditioning systems with no means of providing conditioned air to bedrooms except via open doors. This report offers insight into a strategy for providing conditioned air to those rooms when the bedroom doors are closed. The report also provides heretofore unmeasured data for the range and frequency of potential thermal discomfort that occupants may experience when using this strategy. The results from this study of a single-story, three-bedroom house in Fresno, California, should not be generalized for other house types and climates; instead, it is information for the research community to use as it works with builders toward the end goal of being able 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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