A Systemic Short-Term Field Test for Residential HVAC Thermal Comfort Performance

William Rittelmann, IBACOS, Inc.

ABSTRACT

This paper proposes the development of a test protocol, based on American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE 1990) and ASHRAE (2004), that can be used to evaluate the thermal comfort performance of a residential HVAC system as installed within a single-family home. Thermal comfort criteria of both standards as well as guidelines from Air Conditioning Contractors of America (ACCA 1995) are discussed and additional criteria, necessary to evaluate the thermal comfort performance of a house through space and time, are proposed. The purpose is to enable designers to compare and optimize systems using a universal rating scale for thermal comfort performance. Measured data from various field investigations are used to illustrate issues with measurement techniques and analysis.

Introduction

Need for Additional Criteria

Performance standards such as ASHRAE (1990), ASHRAE (2004), and ISO (1994) are useful in defining thermal comfort for a group of people in a single room, but they are not intended to address the ability of a mechanical system to maintain consistent thermal comfort conditions in multiple spaces in a given building. ACCA (2004) states that residential forced-air systems should be designed to maintain room-to-room temperatures within 2.0°F (1.1°C) with a maximum of 4.0°F (2.2°C), but no protocol is suggested to define how and when these measurements should be taken. ASHRAE (1990) specifies the test procedure to determine the Air Diffusion Performance Index (ADPI) for a given room, but the index is loosely connected to thermal comfort and cannot be used to evaluate the systemic performance of a multi-room system. ASHRAE (2004) provides guidelines for the placement and specification of sensors, but does not define criteria for acceptable thermal comfort variations between rooms. By definition, the thermal comfort zone is intended to satisfy 80% of individuals in a group. If a house has rooms with thermal comfort conditions near the upper and lower boundaries of the thermal comfort zone, meeting the comfort criteria within each room only ensures that 60% of occupants would find whole-house thermal comfort performance acceptable. This is because the 20% Predicted Percent Dissatisfied (PPD) at the lower end of the thermal comfort range are not the same individuals in the 20% PPD at the upper end of the thermal comfort range. Without a method to equitably gauge the thermal comfort performance of various HVAC system types and configurations, decisions concerning the ability of a system to provide comfort are often based upon random, casual observations. Many builders, designers, and homeowners are unaware of the magnitude of temperature differences that occur within a house and the causes of these differences. Manufacturers of HVAC equipment promote systems that provide comfort, but because there are no standards to define this aspect of system performance, the performance of an installed system cannot be adequately measured, meaning that there is no real feedback for designers, installers, or manufacturers.

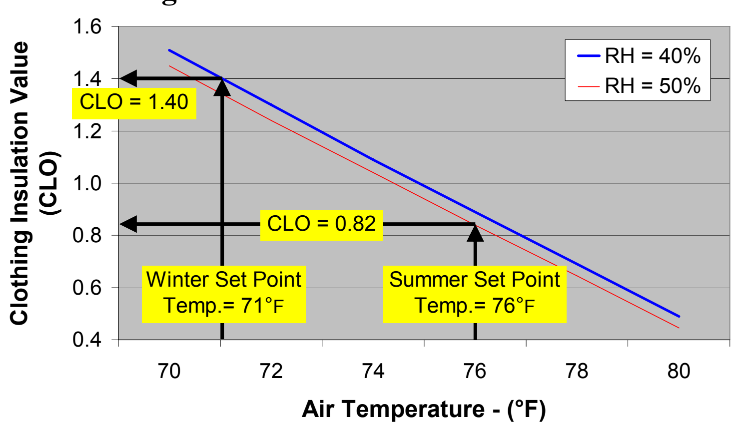
Measurement Criteria

Primary Factors of Thermal Comfort

The first step to understanding the difference between thermal comfort and thermal comfort performance of a system, is to review the ASHRAE (1990) and ASHRAE (2004) / ISO (1994) calculation procedures relative to the parameters of a residential HVAC system. The six primary factors in determining thermal comfort are; (1) Metabolic rate, (2) Clothing insulation, (3) Air temperature, (4) Radiant temperature, (5) Air speed, and (6) Humidity. ASHRAE (2004)/ ISO (1994) addresses all six criteria. ASHRAE (1990) only directly addresses air temperature and air speed, but stipulates other conditions must meet the criteria of ASHRAE (2004). The relevance and treatment of each variable are explained in the following paragraphs.

Metabolic rate (MET) and clothing insulation value (CLO). For the purposes of comparing the performance of residential HVAC systems, metabolic rate (MET) and the clothing insulation value (CLO), should be set to constants, as done in ASHRAE (1990). Although individual occupants will adapt to seasonal comfort conditions in the house with various clothing levels, the assumption is that the house and HVAC system should be designed so that an individual does not need to change clothes to maintain comfort when moving from room to room. This adaptive comfort issue is not addressed in current standards, which are focused on the satisfaction of a group of people and not an individual. In the proposed protocol, metabolic rate is always assumed to be 1.0 and the clothing insulation value is adjusted between 0.5 and 1.5 according to the season and the set point temperature of the system as shown in Figure 1. The clothing insulation value is used to adapt the rating scale to various set point temperatures by setting it equal to a value that achieves a thermal sensation or a Predicted Mean Vote (PMV) of 0. This clothing insulation value is then used as a constant when analyzing the measured data for the duration of a field test. A relative humidity (RH) level of 50% is assumed as a reasonable target for summer conditions and 40% is assumed for the winter. However, if the HVAC system is not designed to influence humidity levels in the winter then additional criteria are required to calculate thermal comfort performance independent of humidity. This could simply be defined as the average dew point temperature measured during the test period, a concept similar to the calculation of the reference temperature correction calculation used in ASHRAE (1990).

Figure 1. Clothing Insulation Values for Neutral Thermal Comfort



Room air temperature. Air temperature is often times the only variable directly controlled by a residential HVAC system so it is critical that sensors be placed in representative locations within a room. Current ASHRAE (2004) guidelines suggest placing sensors either in the center of the room or 1.0 m (3.3 ft) inward from the center of each wall. In either case, it is suggested that the sensors be placed in the most extreme conditions within the room including near windows, diffusers, etc. It is also stated that measurements should be made sufficiently away from the occupied boundary of the room. Several of these criteria appear to be contradictory. The suggestions to place sensors in the most extreme conditions appears to contradict the suggestion to place sensors away from the occupied boundary of the room and placing sensors 1.0 m (3.3 ft) from the walls contradicts the placement in the center of the room. Also, the suggestion to place sensors near diffusers lacks specificity and would tend to skew results for rooms with low load densities.

ASHRAE (1990) contains much more specific criteria that is somewhat contradictory to ASHRAE (2004) and results in a minimum of 40 sensors located in two planes along the boundaries of the occupied zone. In addition, it stipulates that at least one plane shall be centered on the diffuser and, in the case of a room with a single diffuser, the length of the plane is equal to the width of the room perpendicular to the diffuser throw less 0.6 m (2 ft). The actual orientation of the plane is not clearly defined, but only implied to be perpendicular to the diffuser throw through the definition of the test plane length. This definition can work in a small room, but only if the diffuser is centered in the length of the wall. Another statement, that defines the orientation of the test plane for linear diffusers as being perpendicular to the diffuser, also seems to imply that the test plane orientation for all other diffuser types shall be perpendicular to the throw.

In addition to clarifying criteria and eliminating contradictions within and between existing standards, research is needed to determine a minimum number of sensors that would produce reasonable results. It is questionable whether 40 sensors are needed to quantify the thermal comfort performance of a small room.

Air temperature measurement criteria for the development of a thermal comfort performance standard would need to address the following criteria:

- Sensor range and accuracy – ASHRAE (2004) does not directly specify these criteria, but instead references ASHRAE (1991), ASHRAE (1990) and ISO (1998) for these criteria. This approach creates some confusion due to the fact these references do not all adhere to the same specifications.
- Sensor location – ASHRAE (1990) provides a good model, but distances to diffusers need to be specified and the quantity of locations needs to be relative to floor area.
- Sensor heights – ASHRAE (2004) defines four heights, but fewer vertical locations would reduce testing costs, possibly with little reduction in accuracy. Additional research is needed.
- Radiant shielding – This is not mentioned in any of the standards and the lack of proper shielding leads to misleading results. Additional research is needed.
- Measurement interval, time-averaging, and duration – Neither ASHRAE (1990) or ASHRAE (2004) directly address these criteria for air temperature measurement. To characterize thermal comfort performance that would be meaningful to a full-time occupant, the duration would need to be a minimum of 24 hours. Time-averaged values of one hour seem logical relative to changes in system loads and data management, but

these would not capture cyclical discomfort due to the operation of oversized systems. This would either require much smaller time-averaged values or measurement of maximum and minimum values within a specified time period. A measurement interval of one minute would be sufficient to capture maximums and minimums.

Mean radiant temperature. The greatest distinction between thermal comfort performance of an HVAC system and thermal comfort of an individual is evident when considering the influence of mean radiant temperature. Figure 2 shows that MRT has almost as much influence on thermal comfort as air temperature. This means that it must be measured if an HVAC system is capable of directly influencing it, such as a radiant floor system. However, forced-air systems do not directly influence MRT nor are they controlled by it. Because of this, the thermal performance of the system can be obscured by changes in MRT, such as direct solar gains, if it is included in the thermal comfort calculation. This is primarily why ASHRAE (1990) does not factor MRT into the calculation of ADPI. A universal test method for thermal comfort performance needs to address MRT so that the performance of a forced-air system can be compared to a radiant floor system. Figure 3 illustrates the relationship between MRT and air temperature near a low-E window during a 48-hour period in the winter. MRT and air temperature are very similar for most of the time period with the exception of a few hours of direct sunlight. If the measurement of MRT were excluded from the test procedure it appears that it would be quite reasonable to set MRT equal to the air temperature minus 1.8°F (1.0°C) in the winter and plus 1.8°F (1.0°C) in the summer.

Figure 2. Influence of MRT and Air Temperature on Thermal Comfort

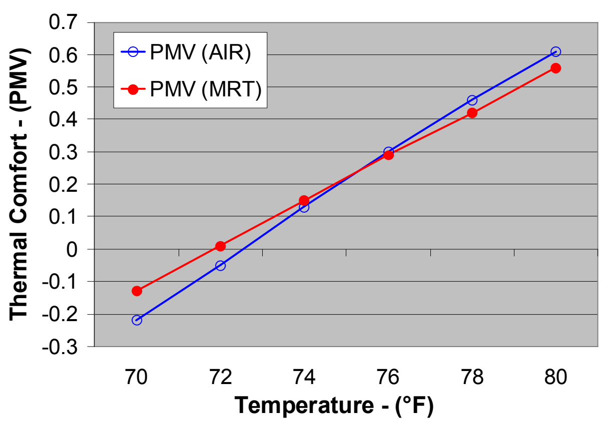
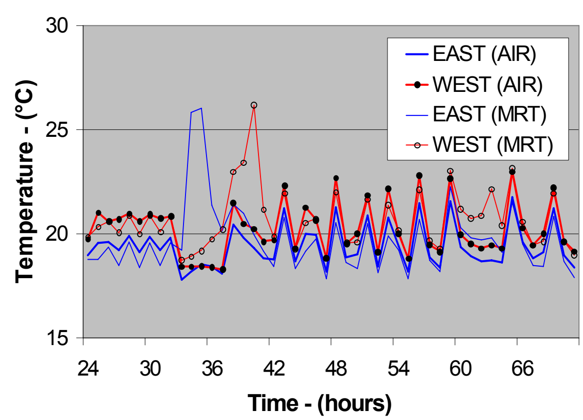


Figure 3. MRT and Air Temperatures in Rooms with Low-E Windows



Mean radiant temperature measurement criteria for the development of a thermal comfort performance standard would need to address the following criteria:

- Sensor range and accuracy – Identical to air temperature criteria.
- Sensor location – Needs to be defined relative to windows and radiant surfaces. Direct solar gain should probably be avoided so that high MRT measurements do not obscure MRT changes due to the system. Quantity needs to be specified relative to floor area.
- Sensor heights – ASHRAE (2004) defines four heights, but one measurement may suffice for the calculation of MRT at other points if the interior surface temperatures of windows

and exterior walls can be accurately estimated or measured. Equations 50 through 53 in Chapter 8 of ASHRAE (2005) may be used.

- Sensor enclosure configuration – This is not defined in current standards. Olesen (1989) suggests using an ellipsoid-shaped globe
- Asymmetric radiant measurements – ASHRAE (2004) defines the heights for seated and standing occupants, but criteria for horizontal locations needs to be developed to determine the minimum number required to produce reasonably accurate results.
- Measurement interval, time-averaging, and duration – Similar to air temperature measurements.

Room air velocities. Room air velocities are probably the most problematic measurements required by ASHRAE (1990) and ASHRAE (2004) due to the high cost associated with multiple low-velocity sensors. Requirements for the measurement of room air velocities to determine thermal comfort performance of residential systems need to be carefully evaluated. The current three-minute time-averaged value specified by current standards appears to be a practical method for characterizing air velocities, but additional criteria are needed to specify the measurement interval and perhaps allow the measurements to be made independent of the time interval used for temperature measurements. This would enable the use of a single hand-held air velocity meter to be used. Measurements would also need to be made coincident with the heating and cooling cycles of a forced-air system, but further research is needed to determine the need for measurement of air velocity during isothermal operation at the end of a heating cycle.

Room air humidity. Humidity (RH), although not directly controlled by most residential systems, can vary due to the type or control of the HVAC system and the ratio of latent to sensible loads. Because there is typically not a great variation in the partial vapor pressure (PA) within the house, current standards allow it to be measured using one sensor. Regardless of the system's ability to directly or indirectly control humidity, it should be measured and included in the thermal comfort calculations. If it is controlled, a target value should be used to determine the CLO value used in the calculations and the measured values should be incorporated into the thermal comfort calculations to determine PMV. If it is not controlled, the measured values should be used to calculate an average value for the period of the test, which can then be used to determine the CLO value and used in the PMV calculations.

Cyclical temperature variation. The relevance of establishing criteria for cyclical variations in room temperature is dependent on the type of comparison being made with the test results. While part of the variation may be due to the configuration of a system, another part is simply due to the responsiveness of the thermostat, which can sometimes obscure the thermal comfort impact due to the system configuration. ASHRAE (1990) addresses this phenomenon by providing a calculation method to compensate all measured temperatures relative to fluctuations measured at the center of the room. This provides a good relative comparison between systems, excluding the impact of the controls, but it does not give a good absolute indication of thermal comfort. On the other hand, ASHRAE (2004) defines the maximum allowable operative temperature change for a 15-minute period as 2.0°F (1.1°C). It is important to remember that this is in terms of operative temperature and that MRT is unlikely to change much in 15-minutes. Under many circumstances, the threshold translates into an allowable air temperature change of 4.0°F (2.2°C), which appears to be excessive, especially given the fact that the width of the summer thermal comfort zone, in terms of operative temperature, is only 6.0°F (3.3°C). While it would be good to have more stringent limits on cyclical temperature variation, it could also be argued that they could be done away with by simply making measurements and calculating PMV-PPD at a high resolution. Either way, an equitable test method should distinguish systems that provide better thermal comfort through variable capacity or integrated thermal mass from those that produce large cyclical temperature variations.

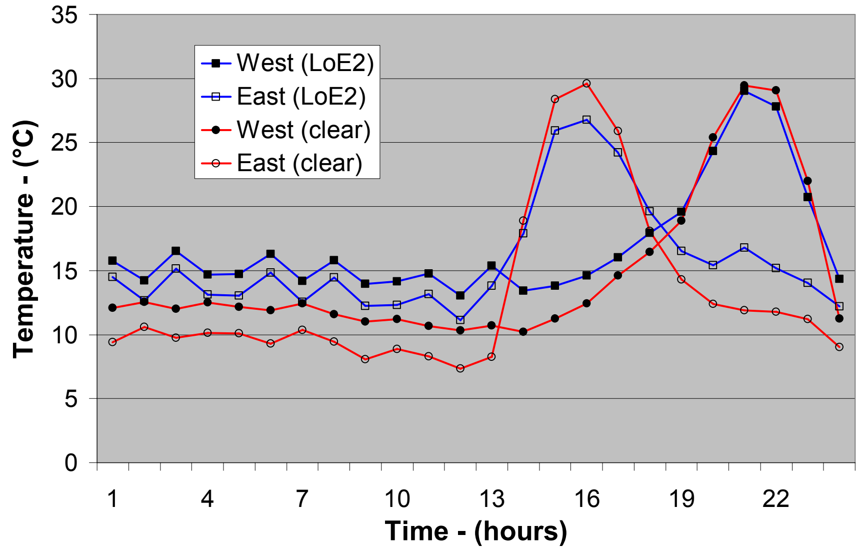
Local Discomfort Factors

Factors affecting local discomfort include radiant temperature asymmetry, draft, vertical temperature difference and floor surface temperature. The relationships between each component and the Percent Dissatisfied (PD) of a control group are given in ASHARE (2004), but additional specificity, software tools and research are needed to provide a comprehensive evaluation method for whole-house thermal comfort performance. Although Berglund and Fobelets (1987) concluded that the PD due to radiant asymmetry and draft are independent and additive, ASHRAE (2004) only implies that the PD for each of these components is additive to one another and the overall PPD. In addition, the maximum allowable PD limits listed in Table 5.2.4 of ASHRAE (2004) appear to contradict the basis of the PMV-PPD index, which only allows for an additional 10% PD due to the sum of local discomfort components. This does not appear to make sense when the maximum PD for draft alone is listed as 20% and the sum of all four allowable discomfort factors is 40%.

The problem of incorporating local thermal discomfort factors is further compounded by the fact that the criteria presented in ASHRAE (2004) are only intended to be applied to thermal comfort scenarios in which the occupants are lightly clothed (0.5 clo – 0.7 clo). It is stated that the results are more conservative for clothing insulation levels higher than 0.7 clo when the whole body comfort condition is near neutral, but it also states that the results "may underestimate acceptability at the lower temperature limits of the thermal comfort zone." This information appears to be somewhat conflicting because higher clothing insulation levels are needed to achieve a neutral comfort condition at lower temperatures. Further research of local thermal discomfort relative to clothing insulation levels at the lower end of the comfort zone is needed.

Radiant temperature asymmetry. With the exception of window surface temperatures needed to determine radiant temperature symmetry, the criteria previously discussed for the measurement of air temperatures can be applied to all other temperature measurements required. Window surface temperature measurements are unique in that it is impossible to disaggregate the direct solar component from a measurement taken with a sensor that is physically placed on the surface of the glass. The temperature either needs to be measured using a remote infrared thermometer, or a protocol needs to be developed to address this issue using a simplified calculation procedure based upon surface mounted temperature measurements taken on the north or opposite side of the house. Figure 4 illustrates the divergence due to direct solar radiation on temperature sensors mounted directly to the surface of the glass. Data are shown for both clear and low emissivity double-pane windows facing East and West. Interior air temperatures were approximately 68ºF (20ºC) while outdoor air temperatures were approximately 28ºF (-2ºC). The data also illustrate the potential maximum radiant temperature asymmetry for points close to a window and the ability of advanced glazing materials to improve comfort. The surface temperature of the clear East-facing window reaches a low of 45ºF (7ºC), which could translate to a radiant temperature asymmetry of 23ºR (13ºK) and a PD of 10% according to Table 5.4.2.1 of ASHRAE (2004). The LoE2 window surface temperature only reaches a low of 52ºF (11ºC), which could translate to a radiant temperature asymmetry of 16ºR (9ºK) and a PD of only 4%.

Figure 4. Glass Interior Surface Temperatures (Winter Day, Feb 2, 2006)



Draft. Research by Fanger and Christensen (1986) and Fanger et. al. (1989) concluded that individuals are more sensitive to draft about the head and neck region at lower temperatures. The relationships between air temperature, turbulence intensity, and allowable mean air velocities are defined in ASHRAE (2004). For a typical turbulence intensity of 35% the allowable mean air velocity ranges from 34 fpm (0.17 m/s) at 68ºF (20ºC) up to 55 fpm (0.28 m/s) at 80ºF (26.7ºC). ASHRAE (1990), while not intended as a "compliance" standard, suggests that "For winter (heating) conditions … the test zone average air velocity shall not exceed 30 fpm (0.15 m/s) and for summer (cooling) conditions shall not exceed 50 fpm (0.25 m/s)."

Vertical temperature differences. ASHRAE (2004) only implies that this is an independent and additive component of local thermal discomfort. Clarification of existing criteria is needed.

Floor temperature. ASHRAE (2004) only implies that this is an independent and additive component of local thermal discomfort. It is also stated that the criteria is only applicable to individuals wearing lightweight indoor shoes. Clarification of existing criteria is needed and continued research is needed to define criteria for barefoot individuals or those wearing lighter footwear such as socks or sandals.

Lessons from Field Investigations

Overview

The following section describes the apparatus, measurement conditions, calculation procedures and lessons learned from various field investigations of thermal comfort performance in single family houses in a mixed climate. The houses were unoccupied during testing and are characterized as high-performance homes built to Energy Star® performance levels.

Apparatus

Apparatus consists of a central data acquisition system wired to multiple temperature sensors and a system status sensor. The data acquisition system is a model CR10X manufactured by Campbell Scientific Instruments. Multiplexers are used to expand the number of channels to as many as 170. Type T thermocouples are used to measure space air temperatures 0.1 m (4 in.) and 1.7 m (67 in.) above the floor at in accordance with ASHRAE (2004). Horizontal locations deviate somewhat from ASHRAE (1990) and ASHRAE (2004) criteria due to time and equipment constraints, but sensors are generally located in the center of rooms and at the boundary of the occupied zone in an effort to get a representative sample. Locations near diffusers are avoided, which consequently results in sensors being placed away from windows. Radiant shields are placed around thermocouples that are in the path of direct sunlight. Except for thermocouple arrays placed in the center of a room, horizontal locations were confined to a zone that was at least 24 inches (0.6 m) from, but no more than 39.4 inches (1.0 m) away from a wall. Locations near supply air outlets were avoided. Arrays were also spaced apart so that at least 90% of the occupied zone was no more than 9.8 ft (3.0 m) from a temperature array. Figure 5 and Figure 6 illustrate a typical setup in a 22 ft (6.7 m) by 35 ft (10.7 m) room with floor diffusers.

Figure 5. Typical Sensor Placement

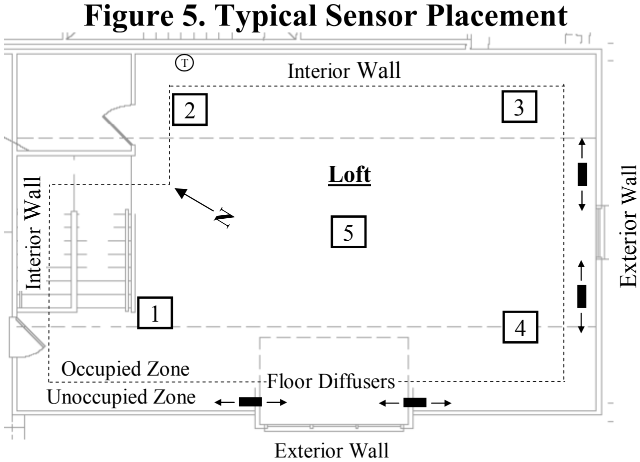


Figure 6. Loft Showing Sensor Placement

Figure 7 and Figure 8 show the raw data collected during the field test and the variations between the different horizontal locations shown in **Error! Reference source not found.**. The noticeable spikes in the data in **Error! Reference source not found.** are due to direct solar gains and a lack of a radiant shield in the case of Location 5. The spikes in data from Locations 1 and

3 are believed to be indirectly caused by solar radiation that causes the wooden base of the sensor stand to heat up and radiate heat up through the open end of the thermocouple shield.

Figure 7. Air Temp. - 4 in. Above Floor

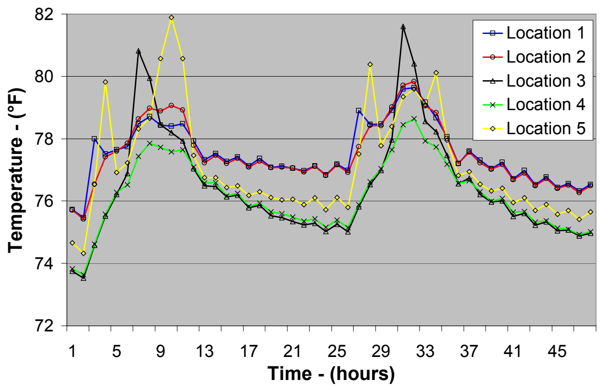
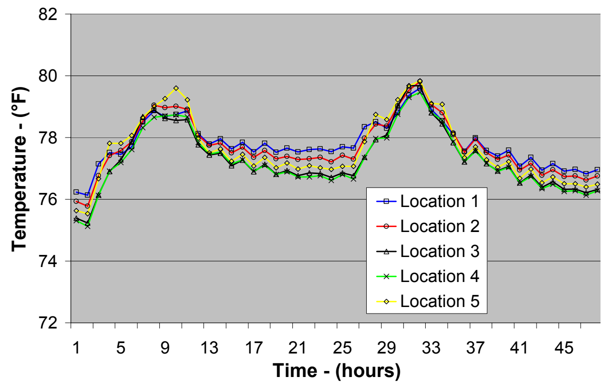
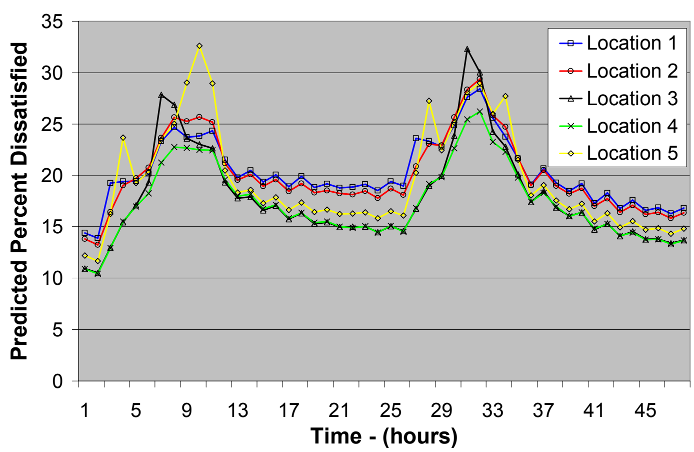


Figure 8. Air Temp. - 67 in. Above Floor



The order of the data in Figure 7 and Figure 8 relative to the sensor placement appears to be logical, with cooler temperatures measured near the exterior walls and warmer temperatures measured near the interior walls. With the exception of excursions due to solar gains, temperatures measured in the center of the room are predictably between those measured near the interior and exterior walls. Figure 9 shows the resulting PMV-PPD. The 48-hour average for Location 5 (center of room) is 19.5% while the average of all five locations is 18.6% - less than 1% difference. The variation in local thermal discomfort was even less, 10.4% vs. 10.3%. Additional study is needed, but this suggests that the center of the room may provide an adequate representation of the thermal comfort conditions of the whole room in high performance houses.

Figure 9. Hourly PMV-PPD - Loft



Measurement Conditions

As is evident in the preceding figures, a minimum test period of 48 hours, rather than 24 hours, was selected to characterize the thermal comfort performance of the system due to the variability of the weather conditions. The tests were conducted during conditions that exceeded 50% of the design conditions, in compliance with ASHRAE (2004). For each test period, average hourly and peak outdoor temperatures were recorded to serve as a reference when comparing different cases. Total horizontal insolation was also measured, but a sensor failure voided the use of the data. Measurements were taken at 15-second intervals and stored as one-minute and one-hour averages.

The thermostat set point temperature was set to 73.0°F (22.8°C) for cooling tests and 70.0°F (21.1°C) for heating tests 24 hours prior to the test and for the duration of the test period. Other temperatures may be used provided the clothing value in the thermal comfort calculation procedure is adjusted to achieve a neutral PMV of zero at the set point temperature. Because closed doors and limited fan operation present the most extreme operating conditions, the system fan was set to cycle and all interior doors were closed for the tests.

Thermal Comfort Calculations

Temperature calculations. Individual temperature measurements (15-second intervals) were first averaged over one-hour periods before they were averaged together with measurements taken in an individual room. Temperatures for PMV calculations are linearly interpolated from the average temperature values of the high and low thermocouples in an individual room to estimate the temperature at 34 inches (1.1 m) above the floor. Accuracy may be improved by performing PMV and PPD calculations for each temperature array location within a room, but only if conditions are close to neutral with some arrays producing negative PMV values and others resulting in positive PMV values.

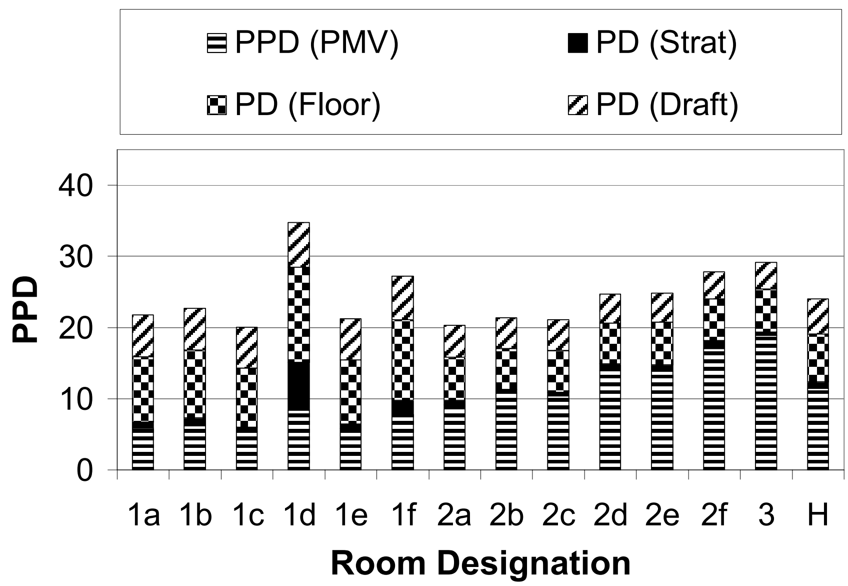
Predicted mean vote. The Predicted Mean Vote (PMV) and resulting Predicted Percent Dissatisfied (PPD) for individual rooms were calculated using a spreadsheet and macro based on the calculation method outlined in Appendix D of ASHRAE (2004) and Annex D of ISO (1994). Whole-house values for PMV-PPD values were calculated as a weighted average of the individual room PMV-PPD values using floor area. Inputs for the model are listed in Table 1. Clothing values were set to correspond to a neutral PMV of 0.0 at the set point temperature of the HVAC system and a relative humidity of 50% for the summer and 40% during the winter. These fixed relative humidity values used to determine the clothing values should not be confused with relative humidity input data requirements of the calculation procedure listed in Table 1.

Table 1. Inputs for Thermal Comfort Computer Model

Description	Variable	Units	Heat	Cool
Clothing	CLO	clo	1.5	1.0
Metabolic rate	MET	met	1.0	1.0
External Work	WME	met	0.0	0.0
Air temperature	TA	C	Varies	varies
Mean radiant temp.	TR	C	TA – 1	TA + 1
Relative air velocity	VEL	m/s	0.075	0.075
Relative humidity	RH	%RH	40%	varies

Field data indicate that local thermal discomfort during the heating season can account for as much as 80% of the aggregate PPD in some rooms and as little as 33% in others. Figure 10 illustrates the room-by-room disaggregated thermal comfort indices for a well-insulated, three-story home in Pittsburgh, PA during the winter. PD due to radiant temperature asymmetry is not included, but would likely add as much as 6% PD to the rooms on the first floor based on the temperature difference between the floor and the ceiling. If that were the case, none of the rooms would be in compliance with ASHRAE (2004). The uncomfortable conditions on the first floor (rooms 1a–1f) are due to a combination of poor mixing of supply air from ceiling diffusers, combined with outdoor air infiltration from a leaky fireplace that contributed to a minimum local hourly air temperature at the floor of 56ºF (13.4ºC). This increased the 24-hour average discomfort due to stratification to a PD of 6.7% and discomfort due to cold floors to a PD of 13.3%. Poor comfort conditions on the upper floors were a consequence of the poor air diffusion on the first floor and the subsequent migration of that hot air up through the stairwell, which led to longer heating cycles and overheating. These conclusions can not be made from the results of the thermal comfort calculations, but are only evident through evaluation of raw temperature measurements.

Figure 10. Aggregate Thermal Comfort Results - Heating



Conclusions

Much of the criteria needed to measure thermal comfort performance exists in current ASHRAE standards, but discrepancies and limitations of scope suggest that there is a need to develop a complimentary standard to define a single test method that can be used to determine an in-situ absolute comfort rating for residential applications. Discrepancies between the definitions of "occupied zone," as well as the increased use of better-insulated windows and walls, points to a need for research to redefine the "occupied zone." Criteria such as the horizontal placement of sensors, measurement intervals, and test duration need to be added or clarified. Criteria used to determine radiant temperature asymmetry need to be added and the measurement of glass surface temperature measurements needs to be specified to exclude solar radiation. A procedure is proposed to determine a test condition benchmark with a neutral comfort condition (PMV=0). Research needs to be done to investigate the possibility of modifying the air velocity measurement procedures to determine the mean air velocity of a room with a single hand-held meter. Calculation procedures are needed to clearly show that the individual components of thermal discomfort are independent and additive with respect to each other and the PMV-PPD.

Measurements from field investigations were used to illustrate problems with using contact sensors to measure glass surface temperatures, direct solar gain on black globe temperature sensors, and indirect solar radiation on partially shielded air temperature sensors. Results from field investigations were used to illustrate thermal comfort variations from room to room and emphasize the possible need to utilize a positive/negative rating scale, such as PMV, in an attempt to point to the fundamental cause of discomfort.

References

[ACCA] Air Conditioning Contractors of America. 1995. *Manual D, Residential Duct Systems*. Arlington, Va.: Air Conditioning Contractors of America.

[ASHRAE] American Society of Heating, Refrigerating and Air-Conditioning Engineers. 1990. "Method of Testing for Room Air Diffusion." *ANSI/ASHRAE Standard* 113-1990. Atlanta, Ga.: American Society of Heating, Refrigerating and Air-Conditioning Engineers.

_____. 1991. "Method of Testing for Rating the Performance of Air Outlets and Inlets." *ANSI/ASHRAE Standard* 70-1991. Atlanta, Ga.: American Society of Heating, Refrigerating and Air-Conditioning Engineers.

_____. 2004. "Thermal Environmental Conditions for Human Occupancy." *ANSI/ASHRAE Standard* 55-2004. Atlanta, Ga.: American Society of Heating, Refrigerating and Air-Conditioning Engineers.

Baskin, E., and E. Vineyard. 2003. "Thermal Comfort Assessment of Conventional and High-Velocity Distribution Systems for Cooling Season." *ASHRAE Transactions* CH-03-7-5, January 2003. Chicago, Ill.

Fanger, P.O., and N.K. Christensen. 1986. "Perception of draught in ventilated spaces." Ergonomics 29(2):215-35.

ISO 7730:1994. "Moderate Thermal Environments - Determination of the PMV and PPD Indices and Specification of the Conditions for Thermal Comfort."

[NASEO] National Association of State Energy Officials. 2000. *National Home Energy Rating Technical Guidelines*. Alexandria, Va.: National Association of State Energy Officials.

Olesen, B.W., J. Rosendahl, L.N. Kalisperis, L.H. Summers, and M. Steinman. 1989. "Methods for measuring and evaluating the thermal radiation in a room." *ASHRAE Transactions* 95(1).