Thermal Stratification Performance of Underfloor Air Distribution (UFAD) Systems

T. Webster* Fred Bauman†

M. Shi‡ J. Reese**

*Center for the Built Environment, University of California, Berkeley

†Center for the Built Environment, University of California, Berkeley

‡Center for the Built Environment, University of California, Berkeley

**York International

THERMAL STRATIFICATION PERFORMANCE OF UNDERFLOOR AIR DISTRIBUTION (UFAD) SYSTEMS

Tom Webster, Fred Bauman and Mingyu Shi
Center for the Built Environment, University of California, Berkeley, CA

James Reese
York International, York, PA

ABSTRACT

Tests were conducted to determine the impact of room airflow and supply air temperature (SAT) on the thermal stratification in interior spaces, and the effect of blinds in perimeter spaces for UFAD systems. Room airflow was varied over the range of 0.7-5.1 (L/s)/m^2 (0.3-1.0 cfm/ft^2) and SATs over 15-19°C (60-67°F) for constant nominal interior heat input of 55-59 W/m^2 (5.2-5.5 W/ft^2). Results show that spaces can be highly stratified when the airflow is reduced for a given load. When SAT is varied, the shape of the temperature profile does not change; it only moves to higher or lower temperatures. Perimeter space tests conducted at a heat load of 116 W/m^2 (10.7 W/ft^2) and constant room airflow of 5.1 (L/s)/m^2 (1.0 cfm/ft^2) with blinds opened or closed showed that room load is reduced when blinds are closed due to bypassing of window gains directly to the ceiling return via a convective plume.

INDEX TERMS

Underfloor air distribution, UFAD, thermal stratification, full-scale testing

INTRODUCTION

Despite the fact that UFAD systems are being applied in the field in increasing numbers, there is a strong need for an improved fundamental understanding of several key performance features of these systems. Most of the potential performance advantages of UFAD systems over conventional air distribution systems are related to the fact that conditioned air is delivered at or near floor level and is returned at or near ceiling level. Under cooling conditions, this upward movement of air in the room takes advantage of the natural buoyancy of heat gain to the space, producing a vertical temperature gradient, similar to that achieved in the displacement ventilation (DV) systems used commonly in Europe (Nielsen 1996). Unlike classic DV systems that deliver air at very low velocities, however, typical UFAD systems deliver air through floor diffusers with greater supply air velocities. In addition to increasing the amount of mixing and therefore potentially diminishing the ventilation performance compared to DV systems, these more powerful supply air conditions can also have significant impacts on room air stratification and thermal comfort in the occupied zone. The purpose of this paper is to report on a series of full-scale laboratory experiments investigating the impact of various system design and operating parameters on room air stratification. The control and optimization of this stratification can be crucial to proper design, system sizing, energy efficient operation, and comfort performance of UFAD systems.

METHODS

TEST CHAMBER AND INSTRUMENTATION

TEST CHAMBER - All tests were performed within a 5 m × 5 m (16 ft x 16 ft) test chamber, constructed with a 0.6 m (2 ft) high raised floor (Figure 1) and located inside an office building. Walls consist of steel studs 0.6 m (2 ft) o.c., insulated with Fiberglas with a thermal resistance of

0.52 W/m^2-K (R11) and covered with standard sheet rock inside and out. A suspended acoustical tile ceiling contains four ventilated parabolic fluorescent lighting fixtures. A 5 cm (2 in.) wide slot-return immediately above the curtain wall directs return air into a 0.6 m (2 ft) high return plenum above the ceiling tile. An insulated plastic covering tops the ceiling plenum. A 4.6 m (15 ft) wide × 3 m (10 ft) high section of curtain wall is installed between the test chamber and a weather chamber on the west side of the test chamber as illustrated in Figure 1. The curtain wall glazing is double pane low-e glass with a shading coefficient of 0.23 and an overall conductance of 3.95 W/m^2-°C (0.7 Btuh/(ft^2-°F)) (calculated by Window 5.0 window thermal analysis program (Mitchell, 2001)). The weather chamber is temperature controlled and contains a bank of high intensity lights that simulate the solar spectrum. The specified solar intensity based on floor area at the curtain wall of 2.9 W/m^2 (31.3 W/ft^2) was confirmed by an 18-point test with a solar pyranometer to within 1%. Besides the overhead lighting fixtures, internal heat gain was provided by laptop computers, CRT monitors, computer towers, printers, desk lamps, and occupants arranged as shown in Figure 1. Two types of floor diffusers were used; variable area (VA) with a design flow rate of 70 L/s (150 cfm) and swirl (SW) with a design flow rate of 42 L/s (90 cfm).

INSTRUMENTATION - All air temperatures were measured with 100 ohm RTDs and all surface temperatures were measured with Type K or T thermocouples. RTDs have a published accuracy of ±0.3°C (0.5°F) and were calibrated with a precision mercury thermometer with 0.06°C (0.1°F) resolution. In addition to 23 sensors on the profile tree and pressures and airflow measurements, 27 temperatures were measured by RTDs and thermocouples placed throughout the test chamber. SAT was measured in three locations in the supply plenum and return temperatures in three locations each in the return slot and plenum. All RTDs were connected to a modular 64-channel data acquisition system. Channels were sampled at 3-second intervals for 2 minutes yielding 40 readings averaged for each temperature measurement at the end of each test. Diffuser flow rates were measured manually with a calibrated balometer. Room airflow rates were later confirmed with a calibrated orifice meter installed in the ductwork to the supply plenum. The range of estimated measurement uncertainties at 95% confidence for temperatures is 0.1-1.0°C (0.18-1.8°F) and 4-7% for measured heat removal (room airflow multiplied by overall delta T (supply to return temperature difference). Heat flux measurements (and associated inside/outside surface temperatures) were made to check wall surface-to-surface heat conductances (U-values). The results found the U-values to be very close to the assumed values from ASHRAE (ASHRAE, 2001).

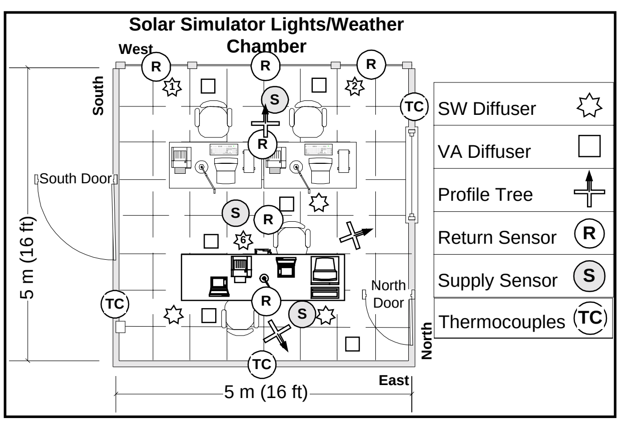

FIGURE 1. Test facility plan and layout

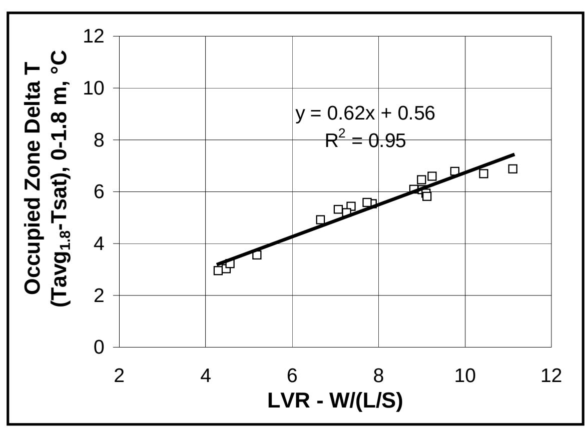

FIGURE 2. Average room temperature (0-1.8 m) difference vs. LVR (interior loads, ~17°C SAT)

PROCEDURES AND ANALYSIS

PROCEDURES Before each series of tests the facility was setup and all RTDs calibrated. The access floor panel joints were all taped to eliminate floor leakage as an uncontrolled variable. Profile measurements were made at three locations in the room at the end of each test. The tree locations were adjusted as necessary to maintain a 0.6-0.9 m (2-3 ft) clearance from the nearest diffuser. As a result, the stratification profiles presented here represent temperature conditions in the ambient space outside of the direct influence of the supply diffusers. The range of test conditions were 1.6-5.1 (L/s)/m^2 (0.3-1.0 cfm/ft^2) for room airflow, 25%-120% of design for diffuser airflow (diffuser air flow ratio), and 15-19°C (60-67°F) for SAT. For most tests nominal total heat input representative of peak load conditions was used. Interior space heat input range was 55-59 W/m^2 (5.1-5.5 W/ft^2) including 100% of overhead lighting; perimeter spaces including solar gain was 116 W/m^2 (10.7 W/ft^2). Tests were normally run for 1.5 to 4 hours to achieve steady state. Tests were conducted in pairs or triplet to study the effect of one parameter while other variables and heat input were held constant.

ANALYSIS The approach used to analyze results is to examine stratification profile shape differences between two or more tests where a parametric change was made for a given variable (e.g., room airflow rate, diffuser type). Following the load factor concept of Nielson (Nielson, 1996) the primary factor for distinguishing room operating conditions is the load to volume ratio (LVR), the ratio of measured heat removal to room airflow rate, which is directly proportional to overall delta T. Figure 2 is a chart of occupied zone average temperature difference (average from floor to 1.8 m (6 ft)) minus SAT) plotted against LVR for interior loads (the slope of this line represents Nielson's load factor). Included in this chart are tests that represent a broad range of room operating conditions, two diffuser types operating at different diffuser design ratios, and SATs of about 17°C (63°F). Certain tests where high stratification was intentionally elicited were not included. This shows that performance varied widely for these tests but the high correlation (R^2=0.95) indicates little effect due to diffuser characteristics. (The effect of diffusers on performance is reported in more detail in a companion paper (Webster, In press).)

HEAT BALANCE

To achieve a heat balance in the chamber calculated loads should equal the measured load. Calculated loads were based on measured surface and air temperatures, measured internal gains, and standard ASHRAE conductances. However, no definitive data was obtained for the contribution of lighting heat to room load. Therefore, load calculations were made for assumed fractions of 10% to 40%. The best correlation with measured heat removal occurred at 20% where the curve fit (R^2 =0.94) was very close to heat balance (i.e., slope=1.025).

RESULTS

EFFECT OF ROOM AIRFLOW

Figure 3 shows an example of interior load stratification profiles where room airflow rates were varied for constant heat input. Additional data for these tests are reported in Table 1.

From the results the following observations can be made:

- As the total room airflow increases, room air stratification decreases producing a more vertical profile characteristic of a well-mixed space.
- As airflow increases, the measured heat removal rate also increases. (See Discussion.)
- For the range of conditions tested, the temperature near the floor remains relatively constant for all room airflows; i.e., the profile "pivots" around a fixed temperature.

- The gradients in the occupied zone of the space (between the floor and 1.2 m (4 ft) for a seated or 1.8 m (6 ft) for a standing occupant) increased as the room airflow is reduced resulting in a relatively large change in temperature at the typical thermostat level (1.5 m (5 ft)). Table 1 shows the temperature stratification in the occupied zone (0.1 m to 1.7 m (4 in. to 67 in.)) exceeds the upper limit of 3°C (5°F) recommended by ASHRAE (ASHRAE, 1992) at the lowest room airflow rate (Test 2-2c). Also shown in Table 1 is the fact that as the airflow changes, the change in temperature at 1.5 m (5 ft) is about twice that of the change in average occupied zone temperature.

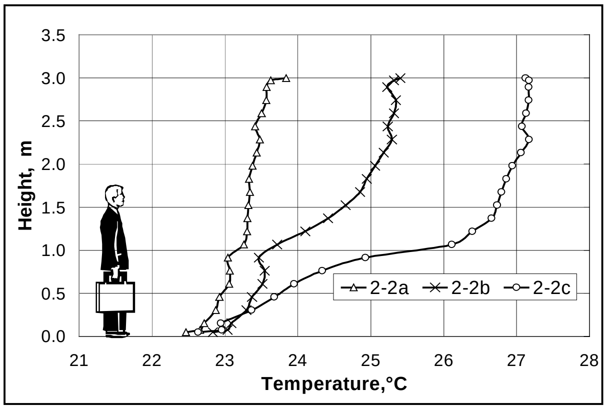

FIGURE 3. Effect of room airflow variation, interior loads (see Table 1)

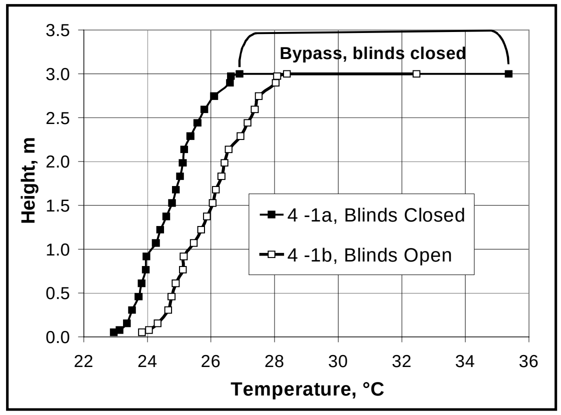

FIGURE 5. Effect of blinds, perimeter loads (see Key)

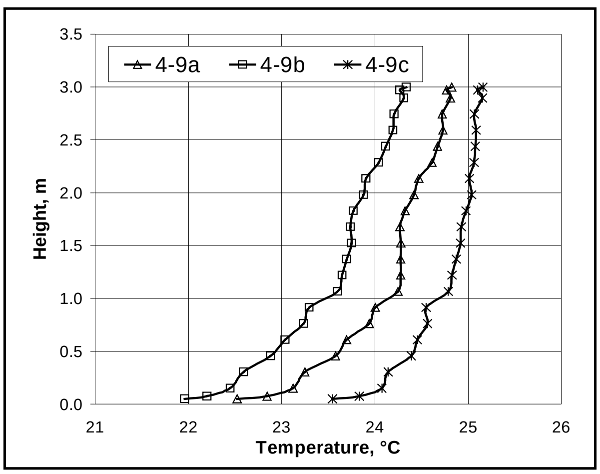

FIGURE 4. Effect of SAT, interior loads (see Key)

Test Conditions Key (Figures 4, 5)

Nominal Total Heat Gain:
Interior: 59.0 W/m^2 (5.48 W/ft^2)
Perimeter: 116 W/m^2 (10.76 W/ft^2)

Test	Load Type	Room Airflow (L/s/m^2)	Ht. Rem. Rm. [Tot.] (W/m^2)	SAT (°C)
4-9a	Int	2.7	25.2	15.8
4-9b	Int	2.7	29.1	17.4
4-9c	Int	2.7	20.1	19.3
4-1a	Per	5.1	61.5 [117]	17.1
4-1b	Per	5.1	73.2 [99]	17.0

TABLE 1. Test conditions for room airflow variation

Test	2-2a	2-2b	2-2c
Nominal total heat input, W/m^2 (W/ft^2)	56.6 (5.2)	56.6 (5.2)	56.6 (5.2)
LVR, W/(L/s) (W/cfm)	7.3 (3.3)	9.1 (4.1)	11.1 (5.0)
Room airflow, (L/s)/m^2 (cfm/ft^2)	5.0 (1.0)	3.1 (0.6)	1.7 (0.3)
Room temperature – @Thermostat,°C (°F)	23.4 (74.1)	24.9 (76.8)	26.7 (80.1)
Average room temp, 0 - 1.8 m, °C (°F)	23.1 (73.6)	23.8 (74.8)	25.1 (77.2)
Average room temp, 0 - 1.2 m, °C (°F)	23.0 (73.4)	23.4 (74.1)	24.2 (75.6)
Head-foot temp. diff., 0.1 m - 1.7 m, °C (°F)	0.7 (1.3)	1.8 (3.2)	3.8 (6.8)

EFFECT OF SUPPLY AIR TEMPERATURE (SAT)

Figure 4 shows results for tests where SAT was varied over the range of 17-19°C (60-67°F) for constant heat input and room airflow rate for a simulated interior space. As shown, the temperatures of the profiles increase or decrease, but keep approximately the same shape.

PERIMETER SPACES – ROOM LOAD BYPASSING

Figure 5 shows the difference in performance between perimeter space tests with blinds closed and open for the same total heat input (solar plus internal gains) and room airflow. Note the large temperature difference between the ceiling of the room (measured at 0.05 m (2 in.) below the ceiling) and the slot return temperature for both cases, and how much greater it is with blinds closed. This indicates that the use of blinds increases the strength of the window thermal plume, thereby facilitating "bypassing" of convective energy directly to the return and reducing the total gain to the room. The bypassed energy is about 26% and 48% of the total heat removal for blinds open and closed, respectively, almost twice as much for lowered blinds. The key for Figure 5 confirms that the measured room heat removal is lower for closed blinds, which would be expected if the total gain is reduced. Note that the total heat removal (measured heat removal using the slot return temperature) is greater for *closed* blinds (Test 4-1a). This represents the total load on the HVAC system. (See Discussion.)

DISCUSSION

ROOM AIRFLOW AND SAT The room airflow tests exemplify how the magnitude of the load can vary between tests. Although the heat input from solar and internal sources was constant between tests, the measured heat removal (the best measure of actual load) was not. Differences in load will occur to the extent that differences in room profiles affect the heat transfer across surfaces. Some of this variability is due to heat transfer through the chamber non-adiabatic chamber walls combined with uncontrolled outside conditions. For real interior spaces this change in load would occur primarily as a function of changes in the ceiling and floor heat transfer; thus we would expect the differences in load to be smaller than those shown. For perimeter spaces, changes in heat transfer at the outside wall also would affect the load. Similar arguments can be made for the variability associated with SAT change. In real interior spaces there would be no outside walls so conduction differences would be confined to ceiling and floor. The sensitivity likely would be greater than shown here. Increasing SAT will always decrease conduction loads; i.e., for outside conditions both higher (wall delta T decreases thus reducing heat gain) and lower (wall delta T increases thus increasing heat loss and lowering net heat input) than room temperature.

PERIMETER SPACE BYPASSING The perimeter space tests with blinds open and closed demonstrate a potential fundamental difference between UFAD and overhead systems. In overhead systems virtually all the solar and conduction gain appears in the room load due to mixing produced by ceiling diffusers. For UFAD the entire heat balance of the room is different for closed vs. open blinds. Some heat gain components increase (e.g., ceiling heat gain increases significantly when blinds are closed) while others decrease. The net result, however, is lower room load for closed blinds. The fact that bypass energy is shown for open blinds indicates that conduction heat gains also create thermal plumes. In addition, it seems counter intuitive that system load increases with blinds closed while room load decreases. Lower total load for open blinds can be partly explained by direct solar gain absorbed at the floor being lost to the supply plenum where it is not accounted for as part of the system load; i.e., the SAT was adjusted to be the same for both tests. It is unlikely that this factor explains all of the difference suggesting that a complex thermal exchange is occurring at the window. Decreasing room load can be a significant benefit since the number of diffusers and fan sizes can be reduced, but there may be a tradeoff in terms of total cooling energy.

CONCLUSIONS AND IMPLICATIONS

The following are conclusions from full-scale experiments of UFAD systems conducted to investigate the effect on temperature stratification in simulated interior and perimeter spaces.

INTERIOR SPACES When room airflow was reduced over the range of 5.1-1.7 (L/s)/m^2 (0.9- 0.3 cfm/ft^2) with constant total heat input, the temperature difference at 0.1 m-1.7m (4 in.- 67 in) increased from 0.7°C to 3.8°C (1.3 to 6.8°F). The change in average temperature in the occupied zone (floor to 1.2 m (4 ft)) was about half that at the thermostat height. Changing SAT over the range of 15-19°C (60-67°F) did not change the profile shape; it only moved to higher or lower temperatures.

PERIMETER SPACES Tests conducted at 116 W/m^2 (10.8 W/ft^2) total input for both closed and open blinds, resulted in lower room load for closed blinds due to bypassing directly to the ceiling return via a convective plume. However, system load (total load on the HVAC system) was found to increase. This bypass phenomenon is one of the factors that distinguish UFAD load conditions from overhead systems. Other important factors include floor loss (the underfloor plenum provides a heat sink that reduces room load by 5.4-13.5 W/m^2 (0.5-1.3 W/ft^2) and stratification (reduces airflow requirements).

The results of this work, along with planned future research, will be used to develop design and operating guidelines for optimizing stratification performance in UFAD systems to improve thermal comfort, reduce energy use, and reduce systems costs.

ACKNOWLEDGMENTS

This work was sponsored by the Center for the Built Environment (CBE), a NSF/Industry/University Cooperative Research Center at the University of California, Berkeley. We gratefully acknowledge the support of our CBE Partners (please see www.cbe.berkeley.edu) and York International for supporting the use of the test facility. This material is based upon work supported in part by the National Science Foundation under Grant No. 9711282.

REFERENCES

ASHRAE (1992). "Thermal environmental conditions for human occupancy," ANSI/ASHRAE Standard 55-1992, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.

ASHRAE (2001). 2001 ASHRAE Handbook, Fundamentals. Atlanta: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.

Mitchell, R., C. Kohler, et al. (2001). "Window 5.0 User Manual," LBNL-44789, Lawrence Berkeley National Laboratory, November.

Nielsen, P. V. (1996). "Displacement Ventilation - Theory and Design," ISSN 0902-8005 U9306, Department of Building Technology and Structural Engineering, Aalborg University.

Webster, T., F.S. Bauman, J. Reese and M. Shi. (In press). "Underfloor Air Distribution (UFAD): Thermal Stratification Performance," ASHRAE Journal, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.