Underfloor Air Distribution: Thermal Stratification

By **Tom Webster, P.E.,** Member ASHRAE, **Fred Bauman, P.E.,** Member ASHRAE, and **Jim Reese, P.E.**

More than 130 underfloor air distribution (UFAD) systems are installed in North America today, and that number is growing. In comparison to classic displacement ventilation (DV)[1] systems that deliver air at low velocities, typical UFAD systems deliver air through floor diffusers with higher 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 have significant impacts on room air stratification and thermal comfort in the occupied zone.

The control and optimization of this stratification is crucial to system design and sizing, energy-efficient operation, and comfort performance of UFAD systems. To investigate these issues, a series of full-scale laboratory experiments were performed to determine room air stratification (RAS) for a variety of design and operating parameters. This article focuses on practical implications of RAS testing results for the control and operation of constant air volume (CAV) and variable air volume (VAV) UFAD systems. See the RAS Test Chamber Setup sidebar for more about the setup.

Stratification Theory

The theoretical behavior of UFAD systems is based on plume theory for DV systems. For DV, cool supply air is heated as it flows across the floor and then drawn upward primarily through entrainment by thermal plumes that develop over heat sources in the room. A stratification level is established that divides the room into two zones (upper and lower) having distinct airflow conditions. The lower zone, below the stratification level, has no recirculation and is close to displacement flow. The upper zone, above the stratification level, is characterized by recirculating flow producing a fairly well-mixed region. The height of this stratification level primarily depends on the room airflow rate relative to the magnitude of the heat sources.

In UFAD systems, the use of floor diffusers that introduce air with some momentum alters the behavior in the lower zone by increasing the amount of mixing and changing the temperature profile. If the diffuser throw is close to the stratification height or already exceeds it, the throw will penetrate into the warm upper layer bringing warm air down into the lower region.[3] The amount of air brought down influences the temperatures in the lower region. The amount of mixing in the lower layer influences the gradient. In the limit as throw and the amount of mixing is reduced, UFAD systems tend to approach the operation of DV systems. Higher throws that penetrate above the stratification height will result in warmer temperatures and less gradient in the lower region, all other conditions being constant.

Room Airflow

Figure 2 shows the impact of room airflow (total rate of airflow to the room/zone) on stratification performance for swirl diffusers operating at a nominal total heat input (heat input only from all sources including 100% of lighting input) of 5.2 W/ft^2 (56 W/m^2), supply air temperature (SAT) ~64°F (18°C), and room airflows of 1.0, 0.6 and 0.3 cfm/ft^2 (5.1, 3.0, and 1.5 L/s per m^2). (A summary of test conditions can be found in Reference 2.) This figure illustrates how stratification increases when room airflow is reduced for constant heat input. Although gradients in the occupied zone (OZ) of the space (from 4 in. to 67 in. [0.1 to 1.7 m]) ranged from 1.4°F to 6.8°F (0.8°C to 3.8°C), the range of average temperature change in the OZ was only 2.5°F (1.4°C). Similar results were

About the Authors

Tom Webster, P.E., and **Fred Bauman, P.E.,** are research specialists for the Center for the Built Environment, University of California, Berkeley, Calif.

Jim Reese, P.E., is vice president, Underfloor Air System Sales, York International, York, Pa.

found for perimeter zones. Other results[2] show that changes in SAT (for constant heat input and room airflow) do not change the fundamental shape of the profile but simply move it to higher or lower temperatures.

Diffuser Comparison – Interior Zones

The results presented here (*Figure 3*) show stratification profiles for simulated interior spaces operating at average OZ temperatures ($T_{OZ,AVG}$) in the range of 72°F to 74°F (22°C to 23°C) under a range of load conditions, room airflows, diffuser flow rates (represented by the flow rate as a percentage of design flow rate) and SATs shown in *Table 1*. Load is defined as the total zone load, i.e., the net sum of all net gains and conduction losses in the room. Load was determined by measuring actual heat removal under steady-state conditions using measured room airflow and room overall temperature difference (ΔT_{TOT} = RAT – SAT), where RAT is the zone return air temperature. For these tests, the system was not under thermostatic control so the results of parametric changes represent the natural response of the system.

Diffuser performance is assessed by considering the combined impact of the average temperature and gradient in the occupied zone. For convenience in making comparisons, the difference between the average OZ temperature and SAT is used. We refer to this as the "average occupied zone temperature difference," ΔT_{OZ}. This difference helps in making comparisons by normalizing the effect of different SATs. The OZ gradient is expressed by the temperature difference between head at 67 in. (1.7 m) and foot at 4 in. (0.1 m).

Several tests were performed to investigate how the diffuser flow rate impacts performance in interior spaces. These tests were conducted by changing the number of diffusers while keeping both the total room airflow and heat input constant. *Figure 3* and *Table 1* show results for five typical tests: two diffusers each operated at two different diffuser flow rates, and a fifth test (VA-3) for the VA diffuser operating at about the

RAS Test Chamber Setup

The RAS testing was conducted in a 256 ft² (24 m²) full-scale chamber (*Figure 1*), where one wall is a curtain wall adjacent to a temperature-controlled chamber with a bank of lights that simulate solar gain. The chamber is configured to allow both interior and perimeter spaces to be simulated and is furnished with standard office equipment arranged in three workstations. Overhead lighting is provided by recessed parabolic florescent lighting fixtures ventilated to a return plenum space. Return air is exhausted from the room via a slot return mounted directly adjacent to, and above, the curtain wall. Stratification profiles were recorded using a movable tree with temperatures measured at 6 in. (0.15 m) increments. Temperature profiles shown here are the average of measurements taken with this tree at three locations in the test chamber (*Figure 1*). All measurements were taken outside of the direct influence of nearby diffusers (i.e., outside the clear zone, an imaginary cylinder centered on the diffuser within which discharge velocities may be greater than 50 fpm [0.25 m/s]). Experience has shown that these profiles are relatively consistent across the room. See Reference 2 for a complete description of the facility.

Performance tests were made by varying a single design or operating parameter to determine its effect on stratification profile. Two types of diffusers were used that represent two different classes of diffusers. Swirl (SW) diffusers are characterized by a swirling discharge airflow pattern that is intended to produce high induction and limited throws. Variable area (VA) diffusers (primarily used for VAV systems) operate by throttling the discharge area, which causes the discharge velocity to increase as airflow is decreased. For the particular VA type tested, throw and clear zone can be modified by changing the grille orientations to provide vertical, four-way or two-way directional "spread" (sometimes referred to as flair) discharge flow patterns. Throw can also be lowered (along with capacity) by reducing the plenum static pressure. All tests for VA diffusers shown here were conducted with the spread orientation and at constant plenum pressure, which is the manufacturer's recommended configuration. Design specifications are 90 cfm (42 L/s) at 0.08 in. w.c. (20 Pa) for the swirl diffuser and 150 cfm (79 L/s) at 0.05 in w.c. (12.5 Pa) for the VA diffuser.

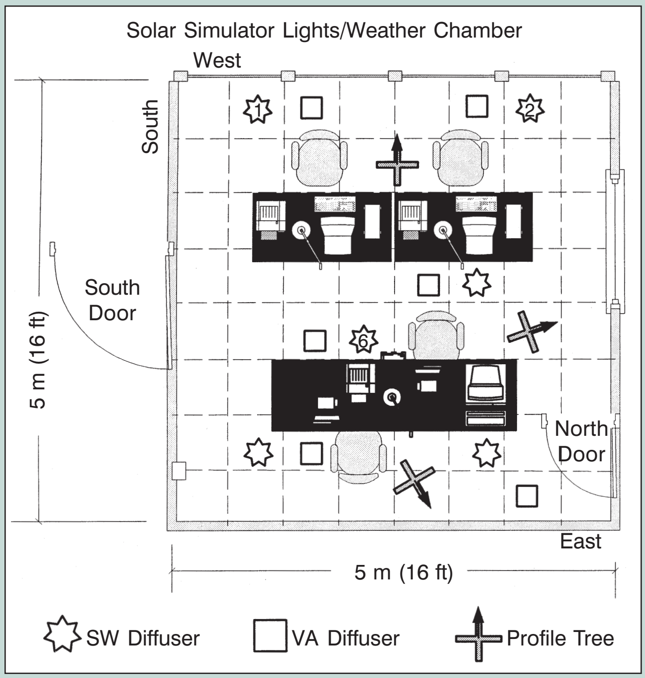

Figure 1: Test chamber plan layout.

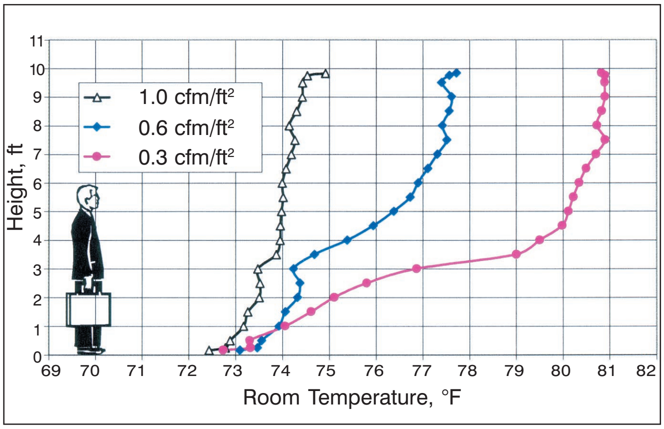

Figure 2: Effect of room airflow variation at constant heat input, swirl diffusers, interior zone.

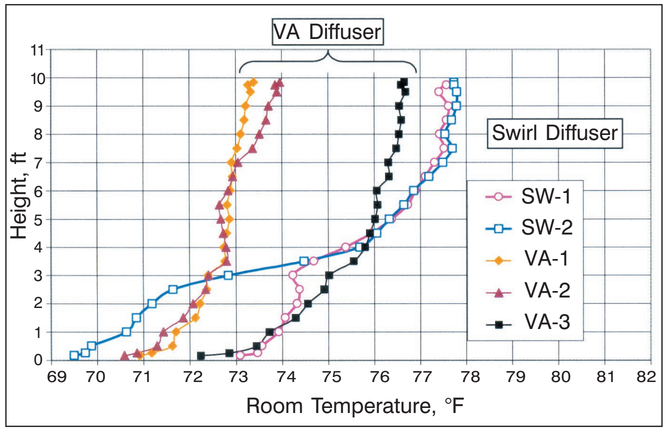

Figure 3: Impact of diffuser flow rate for two diffusers, interior zone.

same load to room airflow ratio as the swirl tests and at 40% diffuser flow rate. The gradient of the profiles for the swirl diffuser tests are greater than those for the VA diffuser (Tests VA-1 and VA-2) because the room airflow is lower relative to the load (resulting in a greater ΔT_{TOT}).

When diffuser flow rate is reduced for VA diffusers (Tests VA-1, VA-2) from 70% to 30% of design, ΔT_{OZ} and OZ gradient both increase by only 0.2°F (0.1°C). When the diffuser flow rate is reduced from 90% to 40% for the swirl diffuser (Tests SW-1 and SW-2), ΔT_{OZ} *decreases* by 0.8°F (0.4°C) while OZ gradient increases by 3.6°F (2°C) to a value of 6.8°F (3.8°C), 1.8°F (1°C) greater than acceptable limits specified by ASHRAE.[4] As expected, by comparing Test VA-3 to VA-1 and VA-2, the shape of the profile showed less sensitivity to reduced diffuser flow rate than the swirl diffuser.

For reference purposes, *Table 1* also shows projection height or throw (based on manufacturer's data) for these tests. The VA diffuser with grilles in the spread position produces four air jets that project outward at about a 30° angle from vertical, in comparison to the more vertical pattern for the swirl. Although airflow varies widely depending on orientation around the diffuser, for the test conditions shown here the vertical throw was estimated to be in the range of 6 to 7 ft (1.8 to 2 m). The throw is assumed to remain relatively constant as the discharge area is throttled. Although not shown here, the clear zone for the VA diffuser in spread position is considerably larger than that for the swirl diffuser. For the swirl diffuser, vertical throw is highly dependent on diffuser airflow, ranging from ~2 ft (0.6 m) at 40% flow up to ~4 ft (1.2 m) at 90% flow. These data indicate that throw may be related to ΔT_{OZ} and gradient, but it must be evaluated in relation to ΔT_{TOT} (i.e., the relationship between load and room airflow) and other aspects of the interaction between thermal plumes and diffuser flow (see previous discussion on theoretical behavior).

Comparing Tests VA-1 and VA-2 with VA-3 for VA diffusers shows that as ΔT_{TOT} increases, the OZ gradient increases along with a small increase in ΔT_{OZ}. Comparing Tests VA-3 for the VA diffuser and SW-1 for the swirl where ΔT_{TOT} is essentially the same, indicates that mixing in the lower region is similar since the gradients are nearly equal, but that more warm air is brought down for the VA diffuser causing ΔT_{OZ} to increase by 1.6°F (0.9°C). Comparing the modest increase in ΔT_{OZ} to the substantial difference in throw for these two tests suggests that OZ temperatures are not strongly affected by throws in the range of conditions tested. Comparing Tests SW-1 with SW-2 for swirl diffusers demonstrates how the OZ gradient increases significantly at low diffuser flow rates as the mixing is confined to a region near the floor. Additional analysis of the test data set for swirl diffusers (not shown here) indicates that the typical range of increase in gradient for a 90% to 40% change in diffuser flow rate is about 2°F to 3°F (1°C to 2°C).

While load and room airflow are the primary drivers of the performance in the occupied zone, differences in diffuser type and operating characteristics appear to have a secondary impact. The testing results suggest that comfort performance is related to the interacting factors of load, room airflow, and diffuser flow rate. How these factors interact in practical situations will determine the relative performance differences between diffusers. For example, swirl diffusers have been shown to be sensitive to diffuser flow rate, so that as it is reduced for a given room airflow and load, the gradient in the OZ becomes more pronounced. Although this also results in cooler temperatures in the OZ and, therefore, offers the possibility of operating at reduced room airflow rates, the gradients can be significantly larger than recommended by ASHRAE[4] under high load conditions. Under light loads (for a given room airflow, e.g., CAV systems) the impact will be less significant because ΔT_{TOT} is less and thus, OZ gradient is reduced and will typically be within ASHRAE recommendations.

Analysis of data from tests representing typical operating conditions (ΔT_{TOT} ~15°F [8°C], SAT~63°F [17°C], $T_{OZ,AVG}$ ~ 74°F [23°C]) indicates that on average, $T_{OZ,AVG}$ for VA diffusers is about 8% of ΔT_{TOT} greater than that for swirls when gradients are equal. However, as explained previously, due to the sensitivity of swirl diffusers to diffuser flow rate, in practice, gradients for swirls at low diffuser flow rates can be larger

Test	Room Load W/ft² (W/m²)	Room Airflow cfm/ft² (L/s per m²)	ΔT_{TOT} °F (°C)	Diffuser Flow Rate (% of Design)	Throw* ft (m)	SAT °F (°C)	ΔT_{oz}** °F (°C)	OZ*** Gradient °F (°C)
Interior Zone Tests								
VA-1	2.6 (28)	0.8 (4.1)	10.8 (6.0)	70%	7 (2)	62.6 (17.0)	9.7 (5.4)	1.4 (0.8)
VA-2	2.9 (31)	0.8 (4.1)	11.7 (6.5)	30%	~7 (2)	62.3 (16.8)	9.9 (5.5)	1.6 (0.9)
VA-3	1.8 (19)	0.4 (2.0)	13.6 (7.6)	40%	~7 (2)	63.1 (17.3)	11.7 (6.5)	2.9 (1.6)
SW-1	2.5 (27)	0.6 (3.0)	13.3 (7.4)	90%	~4 (1.5)	64.4 (18.0)	10.1 (5.6)	3.2 (1.8)
SW-2	2.7 (29)	0.6 (3.0)	14.4 (8.0)	40%	~2 (0.5)	63.4 (17.4)	9.3 (5.2)	6.8 (3.8)
Perimeter Zone Tests								
VA-4	6.5 (70)	1.0 (5.1)	21.0 (11.7)	60%	~7 (2)	62.9 (17.2)	13.3 (7.4)	3.8 (2.1)
SW-3	6.5 (70)	1.0 (5.1)	20.3 (11.3)	100%	~5 (2)	62.7 (17.1)	12.8 (7.1)	5.1 (2.8)

*Maximum height at which velocities of 50 fpm (0.25 m/s) occur, manufacturer's data.
**Average occupied zone temperature minus SAT, 4 in. – 67 in. (0.1 m – 1.7 m).
***Temperature difference between 67 in. and 4 in. (1.7 m and 0.1 m).

Table 1: Test data summary.

than for VA diffusers by as much as 2°F to 3°F (1°C to 2°C).

Diffuser Comparison – Perimeter Zones

Figure 4 shows results for perimeter zone tests under peak load conditions and blinds closed. In general, the profiles are more stratified in the upper region than for interior zones. This is the result of the significantly higher loads for these tests, and in particular, the complex convection process caused by the strong thermal plume along the window.[3] *Table 1* shows results similar to those for the interior zone tests, namely that the VA diffuser operates with a slightly warmer OZ (0.5°F [0.3°C]) and less gradient (3.8°F [2.1°C] vs. 5.1°F [2.8°C]) than the swirl.

Although for the range of conditions tested the two diffusers yield differences in average OZ temperature and gradient, the differences are relatively small. This suggests that under normal operation with a properly designed and balanced system the differences in comfort performance at design loads would be imperceptible. Despite the differences in throw, the results indicate that somewhat comparable comfort conditions are produced for the same room airflow.

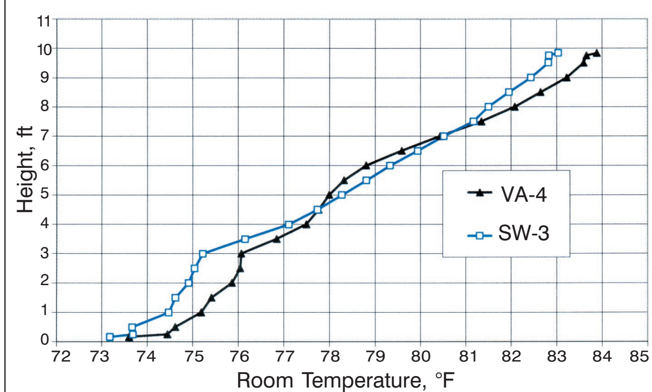

Figure 4: Comparing diffusers for perimeter zones.

Controls for UFAD Systems

CAV Systems

Many projects use CAV systems in interior zones and some use CAV in perimeter zones as well. Typically, CAV systems are controlled by varying SAT in response to thermostat signals. Even with proper design that promotes stratification at peak conditions, this type of control can result in a changing environment in the occupied region as load changes. Depending on the gradient at peak load, the average OZ temperature can be several degrees cooler than the thermostat temperature. (Thus, the thermostat setpoint should be set 2°F to 3°F [1°C to 2°C] greater than the desired occupied zone setting.) At light loads these temperatures will be close to one another because the profiles become more vertical (less stratification) as load decreases for the same room airflow. The SAT adjustment does not impact the shape of the profile, only its position on the temperature scale.[2]

As loads decrease from peak conditions for CAV systems, they will become progressively more over-aired, eventually virtually eliminating stratification. If the system is over-designed in the first place, stratification is likely never to be experienced in actual operation, which may explain why many projects in operation today report lack of stratification.

Many projects use CAV systems for large interior zones where the perimeter zones are served by supply air passing through the plenum of the interior zone. In addition, field experience shows that many interior CAV systems are designed with swirl diffusers with design flow rates of 90 to 100 cfm (42 to 47 L/s) that are assigned one to each workstation cubicle of ~50 to

100 ft^2 (5 to 9 m^2). If these were conservatively sized compared to actual loads and zone airflow is not properly adjusted during system balancing, then the zone will be over-aired. If the interior is over-aired, the SAT will increase to compensate, thereby compromising the system's ability to accommodate perimeter zone loads. On the other hand, if the system was balanced properly so that the zone airflow is adjusted (by reducing fan speed) to promote stratification at full load conditions, the swirl diffusers will operate at a fraction of their design flows, which will exacerbate the gradient in the OZ. A balance needs to be struck between OZ gradient and room airflow. For these reasons, swirl diffusers should be designed to operate close to their design flow rate whenever possible. Using fewer swirl diffusers, or diffusers with smaller peak capacity, in CAV systems may be advisable.

VAV Systems

Under VAV control, as load changes room airflow and diffuser flow rate will change. Tests comparing different diffuser flow rates were not conducted for perimeter zones, but it is assumed that behavior similar to that shown for interior zones would be exhibited. We anticipate that under VAV operation the characteristic profiles for each diffuser would persist and be relatively consistent at least for moderate reductions in load. However, swirl diffusers could lead to larger OZ gradients at low loads due to their sensitivity to flow rate. Further

Controlling Stratification

General

• Promoting and maintaining room air stratification is critical to successful design and operation of UFAD systems. The objective is to minimize energy use (reduce room airflow) while maintaining comfort (acceptable temperatures and stratification in the occupied zone). Overall room air stratification is primarily driven by room airflow rate relative to load. As room airflow is reduced for constant heat input, stratification will increase. Gradients can exceed ASHRAE standards when room airflow is too low relative to load.

• When room airflow is reduced for a given load, the change in average OZ temperature is relatively small compared to the change in gradient. The change in average OZ temperature is about half the change at typical thermostat heights.

• Changing the supply air temperature (for constant load and room airflow) does not change the fundamental shape of the stratification profile, but simply moves it to higher or lower temperatures.

Diffuser type

Test results for interior and perimeter zones indicate that diffuser type and operating characteristics have a secondary impact on stratification performance (the combination of average OZ temperature and gradient).

• For swirl diffusers, occupied zone gradient increases and average OZ temperature is reduced as diffuser flow rate is reduced for constant load and room airflow. For this reason, it is recommended that sizing and number of swirl diffusers be carefully considered so that they operate near their design flow rate whenever possible.

• For VA diffusers in interior zones, occupied zone gradient was found to be insensitive to diffuser flow rate, remaining nearly constant as diffuser flow rate was reduced. VA diffusers operating at peak interior loads and under normal stratified conditions yield an average OZ temperature slightly warmer than that of swirl diffusers (assuming swirl diffusers operating near design flow rate). Although the clear zone for VA diffusers in four-way spread configuration is larger than for swirl diffusers, the grilles can be configured in a variety of ways to change the clear zone in a given direction, resulting in differences in how the two diffusers can be applied.

• Throw does not appear to have a major impact on comfort performance or room airflow requirements for the test conditions studied.

• Under full load conditions studied, both swirl and VA diffusers exhibit very comparable performance in terms of their ability to maintain acceptable comfort in the occupied zone in perimeter and interior zones operating under similar room airflow and load conditions.

Systems and Control

• Due to changes in the stratification profile, the control of CAV systems at a single thermostat location will produce some variation in environmental conditions in the OZ as load changes.

• For CAV and VAV systems, it is recommended that thermostat settings be increased 2°F to 3°F (1°C to 2°C) above the desired, average OZ temperature.

• CAV systems using swirl diffusers that are oversized and are not properly adjusted during the balancing process tend to be over-aired at some or all operating conditions thereby limiting stratification. Due to sensitivity of OZ gradient to diffuser flow rate for swirl diffusers, over-sizing will require tradeoffs between room airflow and OZ gradient to minimize fan energy consumption while maintaining comfort at peak loads; at low loads the impact is lessened due to the reduction in overall room gradient. Over-aired interior CAV systems may result in supply air temperatures too high to serve perimeter zones properly.

• UFAD design airflow requirements in perimeter zones have been estimated to be in the range of 25% less to 15% greater than an equivalent overhead system, depending on the amount of stratification, as well as other operating conditions. This range indicates that design and operating decisions can have a significant impact on overall performance.

testing will be necessary to confirm how these two diffusers compare when operating over a full range of load conditions under CAV and VAV control.

For VAV systems, single-point thermostatic control may be effective in maintaining comfort conditions in the OZ if temperature profiles remain relatively constant throughout the turndown range. However, more sophisticated control techniques that control both average OZ temperature and gradient may also be required.

UFAD System Sizing

Initial results from this testing[2] indicate that zone loads are different for UFAD than for overhead systems and can be significantly impacted by the following factors:

Floor heat transfer. Heat transfer through the floor represents a significant difference between UFAD and overhead systems. This factor can account for as much as 0.6 to 1.2 W/ft^2 (6.4 to 12.9 W/m^2). This reduces the zone airflow requirements but not the system load since this heat transfer is a gain to the supply air and appears as a load at the system level.

Thermal bypass. In addition to the thermal bypassing of convective loads that occurs above the stratification level, testing indicates that in perimeter zones thermal bypassing of convective energy from window solar and conduction gains also contributes to a reduction in zone loads.[2] Lowering the blinds significantly increases this effect. Exploiting this phenomenon may substantially reduce fan energy consumption and first costs.

SAT. Increased SAT for given comfort control setpoints tends to increase the airflow requirements and thus fan energy consumption. This is especially important for perimeter zones where the impact of thermal decay (i.e., potential heat gain to the plenum supply air as it comes in contact with warm slabs and access floor panels) can result in elevated SATs.

Stratification. Promoting stratification in design and operation is key to minimizing supply airflow requirements. However, the degree of stratification has to be balanced with comfort considerations.

When these factors are considered together, the difference between overhead and UFAD design airflows can be significant. An accurate determination of these differences will only be possible when a detailed simulation and design tool is available that models UFAD systems properly. Estimates based on the results of this testing indicate that the difference between UFAD and overhead room airflow requirements for perimeter zones can be in the range of –25% to + 15% for the same heat input, depending on operating conditions. The size of this range indicates that design and operating decisions can have a significant impact on overall performance.

Conclusions

This article discusses recent research results from a series of full-scale experiments investigating room air stratification in underfloor air distribution systems. Tests were conducted to compare the cooling performance of two types of floor diffusers in both interior and perimeter zones. Major conclusions from testing results are shown in the Controlling Stratification sidebar.

Results from an ongoing UFAD testing program indicate that there are stratification performance issues that must be carefully considered to realize the full potential of this technology. Additional information from this research program is available.[6,7] Planned work will address these issues: part load operation of CAV and VAV systems, comparison of airflow requirements between UFAD and overhead VAV systems, performance of linear bar grille diffusers (commonly used in perimeter zones), and more detailed comfort and control analyses.

Acknowledgments

We would like to especially acknowledge the contributions made by York International in support of this research by providing the facility and operating personnel to conduct testing. We would also like to express our appreciation of Mingyu Shi for assistance with test preparation and data analysis. This work was also sponsored in part by the Center for the Built Environment (CBE), an NSF/Industry/University Cooperative Research Center at the University of California, Berkeley. We would like to gratefully acknowledge the support of our CBE Partners: Armstrong World Industries, Arup, California Department of General Services, California Energy Commission, HOK, International Facility Management Association, Pacific Gas & Electric Co., Skidmore Owings and Merrill, Tate Access Floors, Inc., Taylor Engineering Team (Taylor Engineering, Southland Industries, Swinerton Builders, Engineering Enterprise), The Trane Company, United Technologies, U.S. Department of Energy, U.S. General Services Administration, the Webcor Team (Webcor Builders, Critchfield Mechanical, and Rosendin Electric), York International, the National Science Foundation, and the Regents of the University of California. This material is based upon work supported in part by the National Science Foundation under Grant No. 9711282.

References

1. Nielsen, P.V. 1996. *Displacement Ventilation - Theory and Design*. Department of Building Technology and Structural Engineering, Aalborg University.

2. Webster, T.L., F.S. Bauman, J. Reese, and M. Shi. In press. “Thermal stratification performance of underfloor air distribution (UFAD) systems.” To be presented at Indoor Air 2002, Monterey, Calif., June.

3. Linden, P. 2001. Personal Communication, University of California, San Diego, Department of Mechanical and Aerospace Engineering.

4. ANSI/ASHRAE Standard 55-1992, *Thermal Environmental Conditions for Human Occupancy.*

5. Webster, T., E. Ring, and F. Bauman. 2000. “Supply fan energy use in pressurized underfloor air distribution systems.” Center for the Built Environment. University of California, April.

6. Bauman, F. and T. Webster. 2001. “Outlook for underfloor air distribution.” *ASHRAE Journal* 43(6):18–27.

7. Bauman, F., K. Powell, R. Bannon, A. Lee, and T. Webster. 2000. *Underfloor Air Technology* Web site: www.cbe.berkeley.edu/underfloorair. Center for the Built Environment, University of California.