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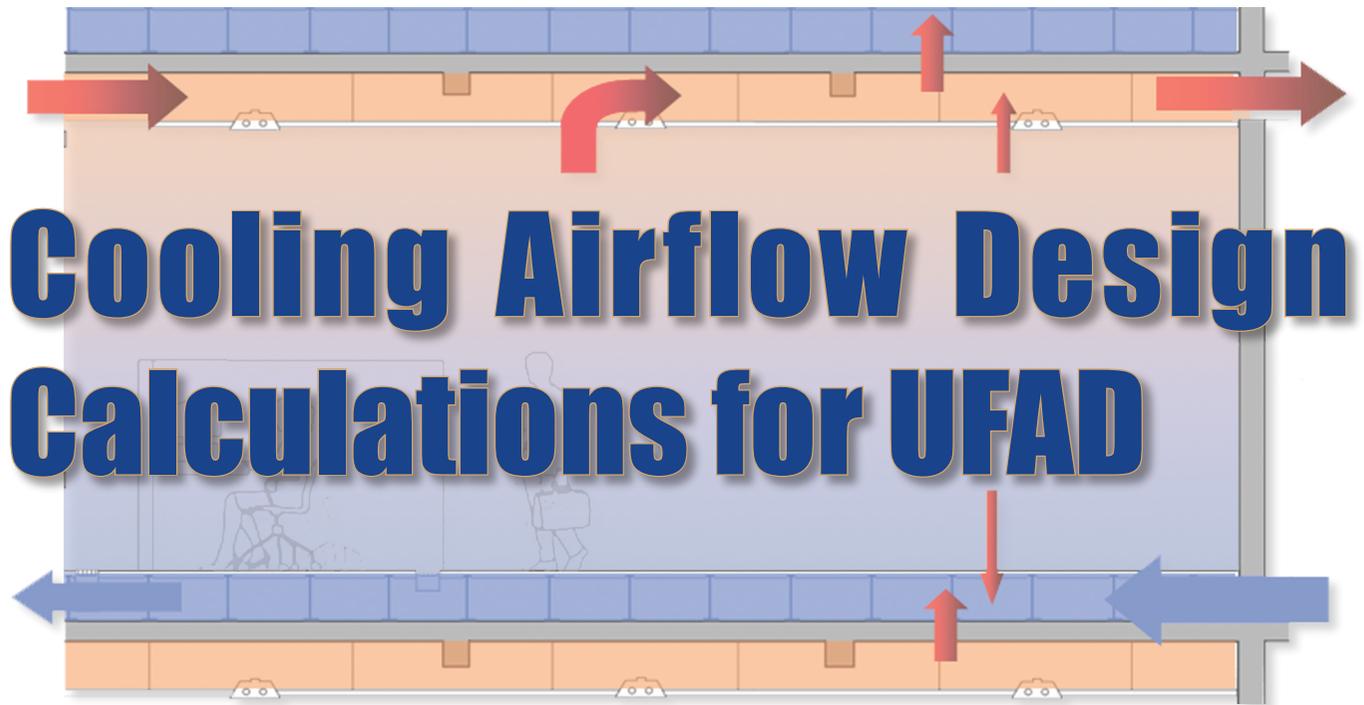
By **Fred Bauman, P.E.,** Member ASHRAE; **Tom Webster, P.E.,** Member ASHRAE, and **Corinne Benedek,** Student Member ASHRAE

During the past ten years as underfloor air distribution (UFAD) has begun to demonstrate significant growth in new commercial office building construction in North America, design engineers have often cited methods for airside design sizing as one of the most important unanswered questions regarding UFAD system design. The challenge in this regard has been how to accurately account for differences between a stratified UFAD environment and the familiar well-mixed environment produced by a conventional overhead (OH) variable air volume (VAV) air-distribution system.

In this article, we present new guidance from a recently developed practical and simplified design procedure to determine cooling airflow design requirements for interior occupied spaces of a building with a UFAD system. Preliminary design guidance for perimeter zones is also provided.

For decades engineers designing OH air-distribution systems have routinely calculated the amount of cooling airflow needed to remove sensible heat loads from a building space by using the following simple steady-state heat balance equation.

$$\text{CFM} = \frac{Q \times (hr \cdot cfm \cdot °F)}{1.1 \times Btu \times \Delta T} \quad (1)$$

Where:
CFM = total room airflow (cfm)
Q = total heat gains to room (Btu/h)
ΔT = temperature difference between return temperature (equal to room setpoint temperature) and the supply air temperature (°F)

The validity of this equation relies on two assumptions: (1) the room is fully mixed (i.e., uniform temperature dis-

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tribution) and (2) 100% of the net heat gain is removed from the space by airflow through the room. However, there are two key aspects of the design and cooling operation of UFAD systems that invalidate the assumptions of *Equation 1* and have important implications for the determination of UFAD cooling airflow quantities. These issues are room air stratification and the existence of a cool underfloor air supply plenum, as discussed further below.

Room Air Stratification

Properly controlled UFAD systems produce temperature stratification in the conditioned space resulting in higher temperatures at the ceiling level that change the dynamics of heat transfer within a room, as well as between floors of a multistory building. Under these conditions, the temperature at the ceiling can no longer be assumed to be equal to the room setpoint temperature. *Figure 1* shows an example room air temperature profile for an interior zone for purposes of identifying key features in a stratified profile.[1] Previously, most concepts of UFAD cooling airflow design sizing followed guidelines developed for stratified displacement ventilation (DV) systems, as described by Chen and Glicksman.[2] This procedure attempted to account for stratification by determining the contribution of each load component to the occupied zone (the region below 67 in. [1.7 m] for standing and 42 in. [1.1 m] for seated occupants) and then assigning a design temperature difference to determine the airflow requirements (see discussions for UFAD systems[3,4]). However, as will be described later, although this load-assignment methodology works for traditional DV system designs, it does not fully characterize the thermal performance of any stratified system (UFAD or DV) that uses an underfloor air supply plenum. Furthermore, since stratified conditions of various magnitudes can exist in the occupied zone, the concept of determining the airflow quantity required to maintain the temperature at a 4 ft (1.2 m) high thermostat, which is assumed to represent a uniform well-mixed occupied zone temperature, is no longer valid. For purposes of allowing a comparison between cooling airflow quantities used by UFAD vs. well-mixed OH systems, we have defined an equivalent acceptable comfort condition for standing occupants in a stratified room as follows (refer to *Figure 1*):

- The average occupied zone temperature ($T_{oz,avg}$), calculated as the average of the measured temperature profile from ankle level (4 in. [0.1 m]) to head level (67 in. [1.7 m]), is equal to the desired setpoint temperature (as measured in a well-mixed OH system).
- The occupied zone temperature difference (ΔT_{oz}), calculated as the head-ankle temperature difference, does not exceed the maximum limit of 5°F (3°C), as specified by ANSI/ASHRAE Standard 55-2004, *Thermal Environmental Conditions for Human Occupancy*.

Underfloor Air Supply Plenums

A distinguishing feature of any UFAD system is the use of an underfloor plenum to deliver supply air through floor diffusers into the conditioned space. Cool supply air flowing through the underfloor plenum is exposed to heat gain from both the concrete slab (in a multistory building) and the raised floor panels. The magnitude of this heat gain can be quite high, resulting in undesirable temperature gain to the supply air in the plenum (sometimes referred to as thermal decay). While the amount of heat entering the underfloor plenum will not change the magnitude of the cooling load that must be removed at the system level, it does directly influence the required cooling airflow quantity by reducing the amount of heat gain to the room that must be removed by room air extraction, defined as heat gain removed by airflow through the room. A recent modeling study found that for a range of typical operating conditions, this supply plenum heat gain can amount to 30% to 40% of the total system heat gain (including 100% of overhead lights). This results in a reduction in the amount of heat load needing to be accounted for by the room air extraction rate.[5] More recently, full-scale laboratory experiments have verified the magnitude of this plenum heat gain.[6]

Figure 2 is a diagram that summarizes the results of an ongoing modeling study (using both a whole-building energy model and a first-law model[6,7] that includes a more detailed heat transfer analysis of the return plenum than was previously done. As indicated, the model is representative of a middle floor of a multistory building with UFAD and a hung ceiling.

The figure shows the calculated distribution of total system heat gain for the room, supply plenum and return plenum. For a hung ceiling, some portion of the system heat gain will be removed via airflow through the room where this airflow gains heat, and after exiting through the ceiling return grille, will lose heat to the slab in the return plenum. (It may be helpful here to recall that heat gain to the airflow represents *positive extraction* and heat loss, *negative extraction*.) In addition, although not shown explicitly, radiation plays an important role in the energy balance of the system. The results indicate that the room cooling load ratio, RCLR (defined as the room air extraction divided by the total system heat gain) is estimated to be 60% to 70%.

Figure 2 also illustrates how warmer temperatures at the ceiling and in the return plenum drive conductive heat transfer through the slab into the supply plenum for the floor above, as well as radiative heat transfer from the ceiling to the floor (and subsequently conduction through the raised floor into the supply plenum). Not indicated, but equally important, is the radiant contribution of the room loads. The calculated net heat gain to the supply plenum is 35% to 45% and net heat loss from the return plenum is 10% to 15%. This diagram depicts a typical interior zone configuration in which a common underfloor plenum serves both interior and perimeter spaces. Although the load distribution will remain about the same, the magnitude of the average temperature gain in the supply plenum will be based on the total airflow within the plenum. However, this airflow is governed by the demands of the room to control its temperature.

The results presented below are based on a preliminary design tool developed as part of a larger research effort focusing
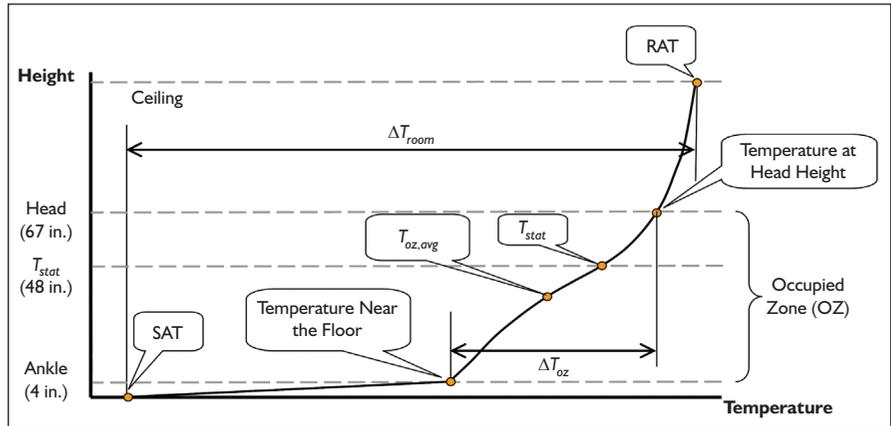
Figure 1: Example room air temperature profile in stratified UFAD system: SAT = room supply air temperature (diffuser discharge temperature); RAT = return air temperature at ceiling.
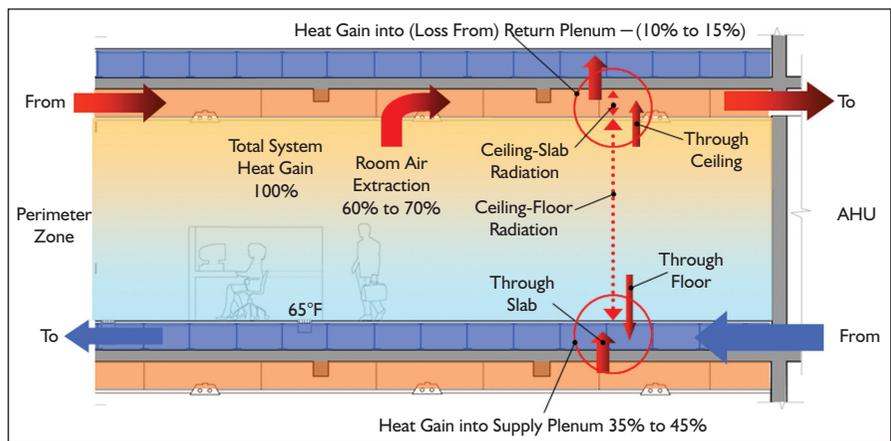

Figure 2: Predicted distribution of room cooling load in multistory building with UFAD: interior zone, total system heat gain = 2.8 W/ft^2 (31 W/m^2), room airflow = 0.6 – 0.7 cfm/ft^2, (3.1 – 3.6 L/s·m^2), diffuser discharge temperature = 65°F (18°C).

on developing energy simulation models for UFAD systems.[6] The design tool accounts for the key issues described previously: (1) room air stratification alters the assumption of well-mixed conditions, (2) the conventionally calculated room cooling load must be reduced by the net amount of system heat gain transferred to both the supply and return plenums and (3) the supply temperature to the room is greater than that for conventional OH design and depends on heat gain to the plenum.

Description of Design Tool

A design tool has been developed as a spreadsheet-based calculation procedure that in its final form will be easy to use by practicing design engineers. The tool is intended to allow the user to apply various commercially available cooling load calculation methods for conventional overhead systems, including the ASHRAE radiant time series (RTS) procedure. *Figure 3* shows a flow diagram of the anticipated process for using the design tool. In addition to the load calculation, users will input several other parameters that define the design and desired operation of the UFAD system. Before proceeding to the main modeling engine, the calculated total system heat gain is modified based on an estimated room cooling load ratio, RCLR, defined as the percentage of the total system heat gain (including 100% of lighting) that is to be assigned to the room in the UFAD airflow calculation. To date, most of the design tool development has focused on interior zone configurations. In this article, we will highlight recommendations for interior zone cooling airflow
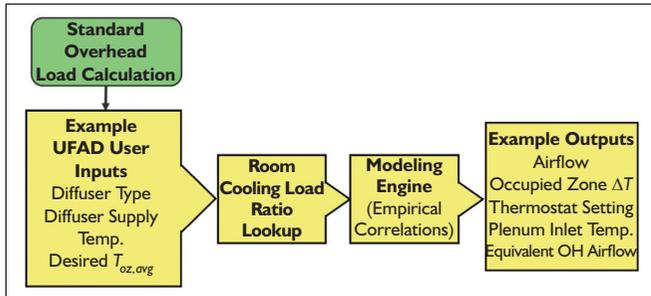
Figure 3: Design tool flow diagram.

Figure 4: Simplified temperature profile vs. measured data: Swirl diffusers, cooling load = 3.1 W/ft² (33 W/m²), SAT = 65.6°F (18.7°C), $T_{setpoint}$ = 74.8°F (23.8°C), airflow = 0.60 cfm/ft² (3.1 L/s·m²).

	SW	VA
Number of Tests	18	8
$T_{oz,avg}$ Average Error (°F)	0.01	0.18
$T_{oz,avg}$ Standard Deviation (°F)	0.32	0.37
ΔT_{oz} Average Error (°F)	0.02	0.02
ΔT_{oz} Standard Deviation (°F)	0.52	0.24

Table 1: Comparison to test data.

calculations and will also provide preliminary guidance for perimeter zones. Research is ongoing to complete the design tool, including perimeter zone airflow calculations.[7]

The design tool uses a combination of a room energy balance and empirical correlations based on experimental data from a full-scale laboratory[6] to simulate a simplified temperature profile constructed of two line segments as shown in *Figure 4*. This assumes, for design purposes, that a controlled temperature profile passes through the thermostat setpoint. It also reflects the observation that most experimentally measured temperature profiles exhibit a change in slope between the lower and upper regions of the room. Beginning with the assumed diffuser discharge temperature (e.g., 65°F [18°C]) and an assumed airflow rate, the model calculates the air temperature at the ceiling using a heat balance equation based on the modified cooling load assigned to the room. The current design tool assumes a constant value of 0.7 for RCLR (i.e., 70% of the total system heat gain must be removed by room air extraction). Temperatures at the 4 in. (0.1 m) and 67 in. (1.7 m) heights are determined through empirical correlations. Using the simple profile, the tool derives the two comfort parameters, $T_{oz,avg}$ and ΔT_{oz}, and then determines the airflow that matches most closely the design conditions.

Comparison to Test Data

The experiments used to develop the design tool were conducted in a full-scale test chamber set up to represent an open plan office with realistic workstations and internal loads.[6] The applicability of the design calculations to real buildings is not currently known but is the subject of ongoing research. We do not expect that the profile correlations will be affected significantly by changes in system characteristics. RCLR correlations still need to be determined for different system configurations, but the model can easily accommodate changes in this parameter.

The accuracy of the empirical temperature profile was verified by comparison with the full-scale test data. This was done for interior zone loads and two types of diffusers, as described below.

Swirl (SW) diffuser: These round floor diffusers are one of the most commonly installed diffusers in UFAD systems; more models are commercially available than any other design. The swirl diffuser used in our tests is representative of typical UFAD applications, providing a design airflow of 80 cfm
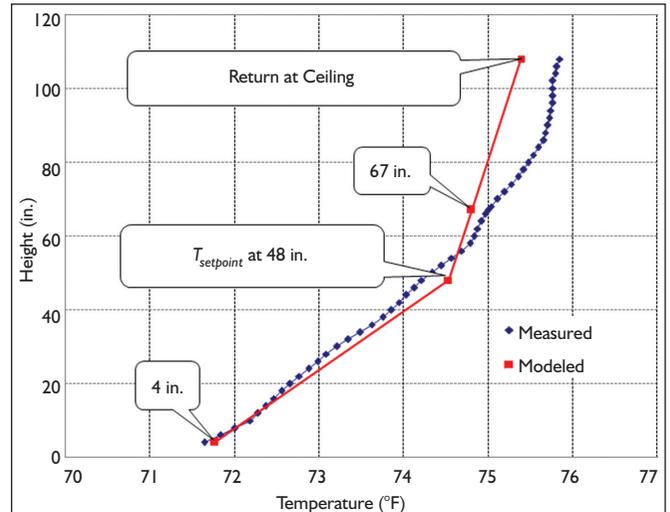
(38 L/s) at a plenum pressure of 0.05 in. H₂O (12.5 Pa). Since the stratification performance of swirl diffusers can change at different airflow rates, the concept of a diffuser design ratio (DDR) is introduced. DDR is defined as the ratio of actual airflow to design airflow (80 cfm [38 L/s]). As discussed below, for design calculations the use of DDR = 1.0 assumes that all diffusers are operating at their design airflow.

Variable-area (VA) diffuser: This square diffuser is designed for variable-air-volume operation. The unit we tested used an automatic internal damper to adjust the active area of the diffuser to maintain a nearly constant discharge velocity, even at reduced air volumes. The adjustable grilles were set in the manufacturer's recommended "spread" position for all tests. The VA diffuser provides a maximum design airflow of 150 cfm (71 L/s) at a plenum pressure of 0.05 in. H₂O (12.5 Pa).

Table 1 shows the calculated errors when comparing the design tool calculations of the two key comfort parameters ($T_{oz,avg}$, ΔT_{oz}) with the full-scale experimental data. The model predictions were based on the measured airflow rate. The range of test conditions covered were the following: 1.9 W/ft² (20 W/m²) ≤cooling load ≤3.4 W/ft² (37 W/m²); 0.30 cfm/ft² (1.5 L/s·m²) ≤room airflow ≤0.85 cfm/ft² (4.3 L/s·m²); 60.8°F (16.0°C) ≤room SAT ≤68.2°F (20.1°C); 72°F (22.2°C) ≤$T_{setpoint}$ ≤76°F (24.4°C); 0.3 ≤DDR ≤2.0. *Figure 4* shows a representative comparison between the predicted and measured temperature profiles for one specific swirl diffuser test.

UFAD	UFAD or OH		UFAD						OH with SAT = 57°F	
Diffuser Discharge Temp. (°F)	Cooling Load* (W/ft²)	Diffuser Design Ratio†	$T_{oz,avg}$‡ = 73°F		$T_{oz,avg}$‡ = 74°F		$T_{oz,avg}$‡ = 75°F		$T_{setpoint}$†† = 73°F	$T_{setpoint}$†† = 75°F
			Airflow§ (cfm/ft²)	ΔT_{oz}** (°F)	Airflow§ (cfm/ft²)	ΔT_{oz}** (°F)	Airflow§ (cfm/ft²)	ΔT_{oz}** (°F)	Airflow (cfm/ft²)	Airflow (cfm/ft²)
Swirl 65°F	2.0	1.0	0.43	2.7	0.38	3.1	0.34	3.6	0.39	0.35
		0.5	0.40	4.5	0.36	5.2	0.32	5.8		
	3.0	1.0	0.65	2.6	0.58	3.0	0.52	3.4	0.59	0.53
		0.5	0.61	4.4	0.54	5.0	0.48	5.6		
	4.0	1.0	0.87	2.5	0.77	2.9	0.69	3.3	0.79	0.70
		0.5	0.81	4.3	0.72	5.0	0.64	5.6		
Swirl 67°F	2.0	1.0	0.58	1.9	0.49	2.3	0.43	2.7	0.39	0.35
		0.5	0.54	3.3	0.46	3.9	0.40	4.5		
	3.0	1.0	0.87	1.8	0.75	2.2	0.65	2.6	0.59	0.53
		0.5	0.81	3.2	0.69	3.8	0.61	4.4		
	4.0	1.0	1.17	1.7	1.00	2.1	0.87	2.5	0.79	0.70
		0.5	1.09	3.2	0.93	3.7	0.81	4.3		
VA 65°F	2.0	1.0	0.46	1.8	0.40	2.0	0.36	2.2	0.39	0.35
	3.0	1.0	0.70	1.9	0.62	2.1	0.55	2.3	0.59	0.53
	4.0	1.0	0.94	2.0	0.83	2.2	0.75	2.4	0.79	0.70
VA 67°F	2.0	1.0	0.62	1.5	0.53	1.6	0.46	1.8	0.39	0.35
	3.0	1.0	0.93	1.5	0.80	1.7	0.70	1.9	0.59	0.53
	4.0	1.0	1.25	1.6	1.07	1.8	0.94	2.0	0.79	0.70

*Total cooling load (system heat gain), including 100% of overhead lighting, as used for sizing conventional overhead (OH) systems.
†Diffuser design ratio (DDR) = (actual diffuser airflow)/(diffuser design airflow).
‡$T_{oz,avg}$ = average temperature in occupied zone (between head height, 67 in. [1.7 m], and ankle height, 4 in. [0.1 m]).
§Airflow = Total room airflow, including Category 2 leakage from supply plenum to room.
**ΔT_{oz} = temperature difference between head height, 67 in. (1.7 m), and ankle height, 4 in. (0.1 m).
††$T_{setpoint}$ = setpoint temperature measured at 4 ft (1.2 m) height.

Table 2: Design cooling airflow performance for UFAD and OH systems: Interior zones.

Design Tool Results: Interior Zones

Table 2 presents design tool predictions of UFAD cooling airflow rates and associated occupied zone temperature differences (ΔT_{oz}) for a range of typical interior zone design conditions. Also shown for comparison are predicted airflow rates for a conventional overhead (OH) system with a supply air temperature (SAT) of 57°F (14°C) and setpoint temperatures of 73°F (23°C) and 75°F (24°C).

To use the table, select the following design conditions: diffuser type and discharge temperature, cooling load, room setpoint temperature (for UFAD systems, this is equal to $T_{oz,avg}$) and DDR equal to 1.0. *Table 2* also includes information for swirl diffusers with DDR = 0.5, represent-

ing a design case where the airflow through each diffuser is 50% of design airflow (80 cfm [38 L/s]). Since the VA diffuser automatically maintains a consistent throw height, and a similar room temperature profile (if there is no significant Category 2 leakage [see below]), VA results are shown only for DDR = 1.0. The user should exercise care when extrapolating the airflow data in the table to other design conditions. It is recommended that the table not be used for airflow calculations outside of the following ranges of design conditions in interior zones: cooling load, up to 4 W/ft² (43 W/m²); room 4 ft (1.2 m) setpoint temperature, 73°F to 76°F (22.8°C to 24.4°C); and diffuser discharge temperature, 63°F to 68°F (17°C to 20°C).

Air Leakage

Supply plenum leakage is one of the most important issues facing the UFAD industry. Experience in the field has shown that Category 2 leakage (from the plenum into the room) can often be in the range of 10% to 20% of design airflow. To account for leakage in design calculations, the predicted airflow from *Table 2* should include the estimated (or measured) air leakage rate at design conditions. Although the results of *Table 2* were developed assuming no leakage, additional full-scale experiments were conducted to investigate the impact of leakage on stratification.[6] These experiments demonstrated that for swirl and VA diffusers with DDR close to 1.0, the

increase in ΔT_{oz} and decrease in $T_{oz,avg}$ will be minimal. These guidelines are applicable for Category 2 design air leakage rates up to about 20%. Category 2 leakage rates above 20% may impact the amount of stratification in the occupied zone, as well as the system's ability to control room temperature. Note that while Category 1 leakage (from the plenum to the outside or other zones in the building) will not impact room air temperature profiles, it will directly impact the airflow (and, therefore, energy use) of the air handler.

Comparison to Overhead (OH) System

For standard room operating conditions (UFAD $T_{oz,avg}$ = OH $T_{setpoint}$ = 74°F [23.3°C]; UFAD SAT = 65°F) at all load levels, *Table 2* shows that airflow quantities for swirl and VA diffusers are predicted to range from 0% to 10% higher than OH airflows. At the higher room SAT of 67°F (19.4°C), UFAD airflows range from 32% to 43% higher than OH (using 57°F [14°C] SAT). The maximum predicted ΔT_{oz} for all design conditions shown in the table for swirl (DDR = 1.0) and VA diffusers is 3.6°F (2.0°C), indicating that excessive stratification will not be a problem at design load. The impact of increasing stratification (produced by raising the room setpoint for a given room SAT) on reducing airflow rates is demonstrated by the values for a higher design setpoint temperature of 75°F (24°C). At this setpoint and with room SAT = 65°F (18°C), predicted UFAD airflow rates are equal for swirl diffusers and slightly higher (7%) for VA diffusers, compared to OH airflow rates at all load levels.

Diffuser Type

The design tool results allow a comparison between the two types of diffusers. VA diffusers, with their higher vertical throw were previously thought to require higher airflow rates due to reduced stratification, while swirl diffusers with throws that vary with airflow were thought to provide excessive stratification at low load conditions. The design tool shows that under design flow conditions (swirl DDR = 1.0), airflow quantities are quite similar, with swirl airflows slightly less than VA (7% less at room SAT = 65°F [18°C], load = 4 W/ft² [13 W/m²], $T_{oz,avg}$ = 74°F [23°C]). VA diffusers, with their variable air volume control, produce consistent stratification (ΔT_{oz}), although approximately 1°F (0.5°C) less than that for swirls, over all load levels for the same room SAT and $T_{oz,avg}$. Swirl diffusers exhibit higher stratification under part load conditions (DDR = 0.5), although nearly all predicted values for ΔT_{oz} are less than or equal to 5°F (3°C) (maximum limit specified by Standard 55-2004), except at room SAT = 65°F (18°C) and $T_{oz,avg}$ = 75°F (24°C).

Example Design Tool Calculation

Figure 5 presents design tool results from *Table 2* for swirl diffusers supplying 65°F (18°C) air with DDR = 1.0 in an interior zone with a cooling load of 3 W/ft² (32 W/m²) and an RCLR = 0.7. The figure plots predicted cooling airflow rates and ΔT_{oz} as a function of $T_{oz,avg}$ over the range of 72°F to 76°F (22.2°C to 24.4°C). Calculations are highlighted for two design conditions that cover the range of $T_{oz,avg}$ between 73°F (23°C) and 75°F (24°C). At each setpoint temperature (read on the left-hand y-axis), the first step is to read the airflow rate on the x-axis, and then for the same airflow the second step is to read the ΔT_{oz} on the right-hand y-axis, as shown. The results indicate that to design the system to maintain the average occupied zone temperature in the range of 74 ±1°F, the system must be designed for an airflow in the range of 0.52 to 0.65 cfm/ft² (2.6 to 3.3 L/s·m²), resulting in an occupied zone temperature difference in the range of 2.6°F to 3.4°F (1.4°C to 1.9°C).

Thermostat Setting

In conventional overhead systems that aim to maintain a uniform well-mixed environment, the assumption is made by building operators that the 4 ft (1.2 m) thermostat setpoint is the same as the average temperature experienced in the occupied zone. However, as indicated in *Figure 1,* depending on the amount of stratification that produces cooler temperatures near the floor, the average temperature in the occupied zone ($T_{oz,avg}$)
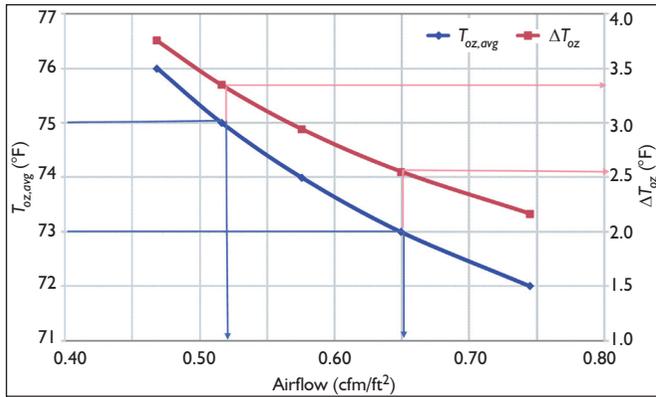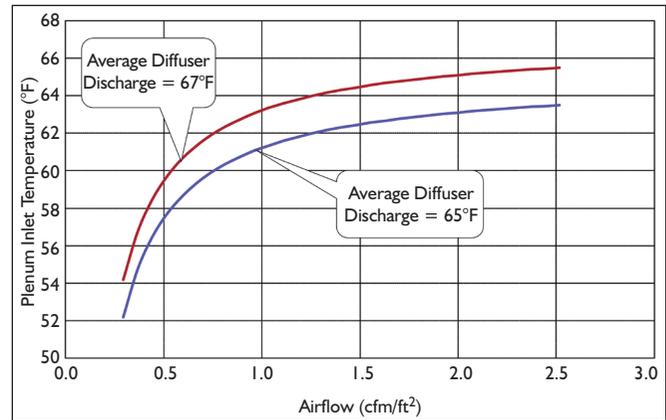
Figure 5: Example design cooling airflow performance: interior zone, cooling load = 3.0 W/ft² (32 W/m²), swirl diffusers with DDR = 1.0, diffuser discharge temperature = 65°F (18°C).

Figure 6: Predicted plenum inlet temperature vs. total plenum airflow for different average diffuser discharge temperatures: Cooling load = 3.0 W/ft² (32 W/m²).

will be lower than the single point thermostat temperature. To maintain equivalent comfort conditions in a stratified environment (e.g., UFAD $T_{oz,avg}$ = OH $T_{setpoint}$), it is recommended to raise the 4 ft (1.2 m) thermostat setpoint by 1°F (0.5°C) for ΔT_{oz} values around 3°F (2°C) and by 0.5°F (0.3°C) for ΔT_{oz} values around 2°F (1°C). For example, to maintain an average occupied zone temperature of 74°F (23.3°C) for swirl diffusers (DDR = 1.0) supplying 65°F (18°C) air to the space at any load level, the 4 ft (1.2 m) thermostat should be set at 75°F (24°C). When the final design tool is released, it will include an automatic calculation for the 4 ft (1.2 m) thermostat setting.

Plenum Inlet Temperature

Figure 6 shows the predicted plenum inlet temperature as a function of total plenum airflow for two different average diffuser discharge temperatures (65°F, 67°F [18.3°C, 19.4°C]) for a cooling load of 3.0 W/ft² (32 W/m²). The predictions are based on a simple steady state energy balance assuming that 40% of the cooling load enters the supply plenum. Note that while results are shown for an average room supply temperature, measured diffuser discharge temperatures across the floor plate will be both higher and lower, depending on the airflow distribution within the plenum.[6,8] Actual performance in the field may vary, but the trends are very clear. The results show that plenum thermal decay can be a problem at lower plenum airflow rates. If coil leaving temperatures in the air handling unit (AHU) have to be reduced to maintain room supply temperatures, potential economizer energy savings (in suitable climates) may be eroded. As illustrated in *Figure 2,* an underfloor plenum serving an interior zone is typically configured as a larger open plenum that also serves perimeter spaces, thus increasing the total airflow through the plenum and reducing the temperature gain. The above considerations point to the importance of ongoing research to develop improved plenum design guidelines and to assess the energy impact of thermal decay on economizer performance.

Perimeter Zones

Perimeter zones differ from interior zones due to the type and magnitude of the heat sources. In cooling mode, the dominant heat source creates a large thermal plume that rises vertically at the window and under peak load conditions the perimeter heat gain is several times that of total internal gains. Research is ongoing to complete the development of a perimeter zone cooling airflow design tool that will be modeled after the interior zone tool described earlier.[6,7] Interim guidance is discussed briefly below.

Figure 7 presents the results of three full-scale laboratory experiments investigating room air stratification in a perimeter zone. The 26 ft (7.9 m) square test room was configured to simulate an open plan office containing six workstations with linear bar grilles next to the perimeter window/wall and swirl diffusers elsewhere. A solar simulator in a weather chamber provided direct solar gain through windows on one wall with a window to wall ratio of 0.74. The simulated perimeter zone load for all tests in *Figure 7,* including an interior load level of 3.5 W/ft² (38 W/m²), was
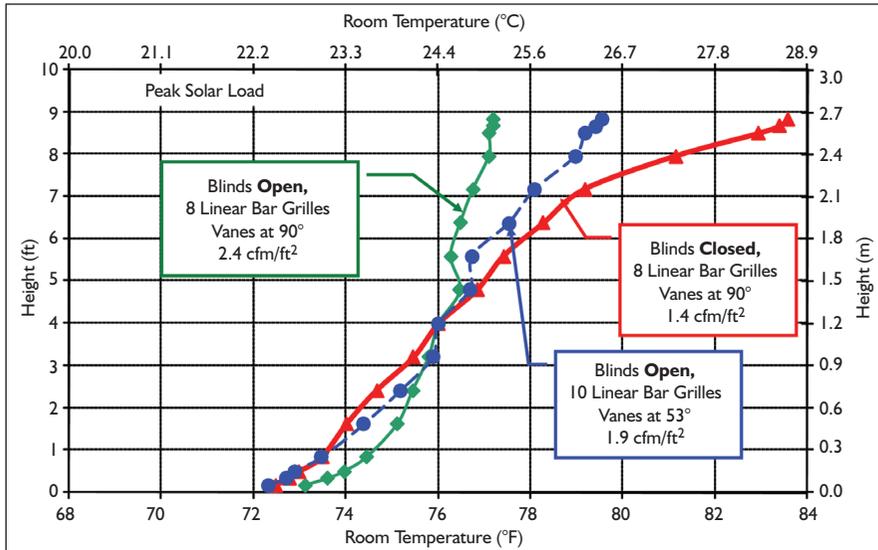
Figure 7: Room air stratification temperature profiles for perimeter zones at constant perimeter zone load of 14.8 W/ft² (159 W/m²), supply temperature of 65°F (18°C), and control setpoint of 76°F (24.4°C) at 4 ft (1.2 m).

14.8 W/ft² (159 W/m²), based on the 15 ft (4.6 m) wide floor area near the window. These test conditions are representative of a west-facing zone in 40° north latitude on July 21 with a window with solar heat gain coefficient (SHGC) = 0.37. All tests used a diffuser supply temperature of 65°F (18°C) and were controlled to a 4 ft (1.2 m) thermostat setpoint of 76°F (24.4°C), meaning that average occupied zone temperatures were 75°F (24°C) or below. Key findings are summarized below.

• The test producing the least amount of stratification required the highest airflow rate: 2.4 cfm/ft² (12.2 L/s·m²), based on the 15 ft (4.6 m) wide perimeter zone. This test used 8 bar grilles with vertical vanes, producing the highest throw and mixing.

• When the number of bar grilles was increased to 10 and the vanes were inclined to 53°, thereby reducing the throw and mixing of the diffusers, stratification increased and the required airflow decreased to 1.9 cfm/ft² (9.7 L/s·m²). This demonstrates the potential benefits of using lower-throw diffusers, although care must be taken to maintain acceptable comfort conditions.

• The lowest airflow requirements were obtained for the test when window blinds were closed, producing the highest stratification with an airflow rate of only 1.4 cfm/ft² (7.1 L/s·m²). This reduction in

airflow rate is the result of several factors, including attenuation of the incident solar radiation entering the space, formation of a stronger thermal plume around the blinds, and changes in the radiant energy distribution (more diffuse). The use of blinds at peak load has profound implications for design although considerable difference of opinion exists about whether to assume blinds open or closed for design calculations.

• For comparison, an overhead system supplying 55°F (12.8°C) air and maintaining a setpoint temperature of 75°F (24°C) would require 2.3 cfm/ft² (11.7 L/s·m²) for this same load condition. If UFAD supply air temperatures increase above 65°F (18°C) due to thermal decay in the plenum, required UFAD airflow rates would also increase.

Summary

In this article we have presented new design guidance for UFAD cooling airflow calculations in interior zones based on a recently developed design tool. The design tool predicts required cooling airflow rates and the amount of stratification in the occupied zone for a range of design conditions. Results are shown for swirl and variable-area diffusers, two of the most frequently installed diffusers in interior zones. The design tool also compares UFAD airflows with conventional overhead (OH) airflows

for equivalent design and comfort conditions. The design tool accounts for heat transfer to the underfloor supply plenum, which can be a significant portion of the total cooling load. This allows the calculation of the plenum inlet temperature (air handler leaving temperature) to maintain a desired diffuser supply temperature.

Preliminary guidance for perimeter zones is presented based on full-scale testing results. Ongoing research is scheduled to complete the development of the cooling airflow design software tool in 2008. In particular, the research will investigate the development of improved models for predicting RCLR and the effects of air leakage. The completed design tool will include a perimeter zone model, user interface and further validation.

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