Arnold Schwarzenegger
Governor

ENERGY PERFORMANCE OF UNDERFLOOR AIR DISTRIBUTION SYSTEMS

Part VI: UFAD Cooling Airflow Design Tool

APPENDIX F

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ENERGY PERFORMANCE OF UNDERFLOOR AIR DISTRIBUTION (UFAD) SYSTEMS PART VI: UFAD COOLING AIRFLOW DESIGN TOOL

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1 INTRODUCTION

For decades engineers designing conventional overhead (OH) air distribution systems have routinely calculated the amount of cooling airflow needed to remove heat loads from a building space using the assumption of a well-mixed room air condition. The familiarity of designers with this relatively simple equation has led to a situation where design space heat gains (i.e., total room cooling load) are considered to be synonymous with return air extraction rates, and cooling airflows are found from the (mixed) room-supply temperature difference, typically assumed to be 11°C (20°F). In a stratified UFAD environment, the assumption of perfect mixing is no longer valid, requiring a different way of thinking about energy flows and airflow quantities. Room air stratification produces higher temperatures at the ceiling level that change the dynamics of heat transfer within a room, as well as between floors of a multi-story building.

Previously, most concepts of UFAD cooling airflow design sizing attempted to determine the contribution of each load component to the occupied zone (the portion of the zone below 6 ft) and then apply a design temperature difference to determine the airflow requirements to satisfy this load condition in the occupied zone (see discussions for UFAD systems [Loudermilk 1999, Bauman 2003] and displacement ventilation systems [Yuan et al. 1999]). While this method had merit (mostly because it adhered to mixed system concepts), more recent research has shown that it may be oversimplified. In addition, there is a lack of research-based guidelines for assigning heat load components to the lower and upper zones of the room, making the application of these earlier methods risky at best.

As part of the current effort to develop a UFAD version of EnergyPlus, members of the research team have investigated two key issues in the design and cooling operation of a UFAD system in a multi-story building. Both represent areas where UFAD differs from conventional overhead (OH) systems and both have important implications for the determination of cooling airflow quantities, as described briefly below.

1. *Thermal performance of underfloor plenums.* Cool supply air flowing through the underfloor plenum is exposed to heat gain from both the concrete slab and the raised floor panels. The magnitude of this heat gain can be quite high, resulting in undesirable loss of control of the supply air temperature from the plenum into the occupied space (sometimes referred to as thermal decay). To investigate this plenum heat gain, we created a simplified first-law model used to estimate and compare the relative magnitudes of the heat being removed from a stratified room. It was shown that under cooling operation in a multi-story building, heat leaves the room through two primary pathways: (1) heat extraction via warm return air exiting the room at ceiling level or through the return plenum, and (2) heat entering the underfloor supply plenum either through the slab from the floor below, or through the raised floor panels from the room above. Surprisingly it was shown that 30- 40% of the total room cooling load is transferred into the supply plenum and only about 60-70% is accounted for by the return air extraction rate, or cooling airflow quantity [Bauman et al. 2006].
2. *Room air stratification.* Properly controlled UFAD systems produce temperature stratification in the conditioned space resulting in higher temperatures at the ceiling level than at the floor. Current thinking for optimizing control of UFAD systems is to allow greater stratification to occur (thus reducing total airflow quantities and saving fan energy), while maintaining acceptable comfort conditions in the occupied zone (i.e., not exceeding the maximum limit (specified in ASHRAE Standard 55-2004 to be 5°F [3°C]) for vertical temperature difference in the occupied zone) [ASHRAE 2004]. However, as stratification is increased and temperatures near the ceiling rise, this condition aggravates the thermal decay problem described above in #1 by directly leading to higher rates of heat transfer into the underfloor plenum. Furthermore, since stratified conditions exist in the occupied zone, the concept of determining the airflow quantity required to maintain a uniform well-mixed occupied zone temperature (e.g., as controlled by a 4-ft high thermostat) is no longer valid. For purposes of allowing a comparison between cooling airflow

quantities used by UFAD vs. OH systems, we have defined an equivalent comfort condition for a stratified room as follows (see Figure 1for a schematic diagram identifying key features of a room air temperature profile):

- The average occupied zone temperature ($T_{oz,\ avg}$), calculated as the average of the measured temperature profile from foot level (4 in.) to head level (67 in.), 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-foot temperature difference, does not exceed the maximum limit specified by ASHRAE Standard 55 of 5°F.

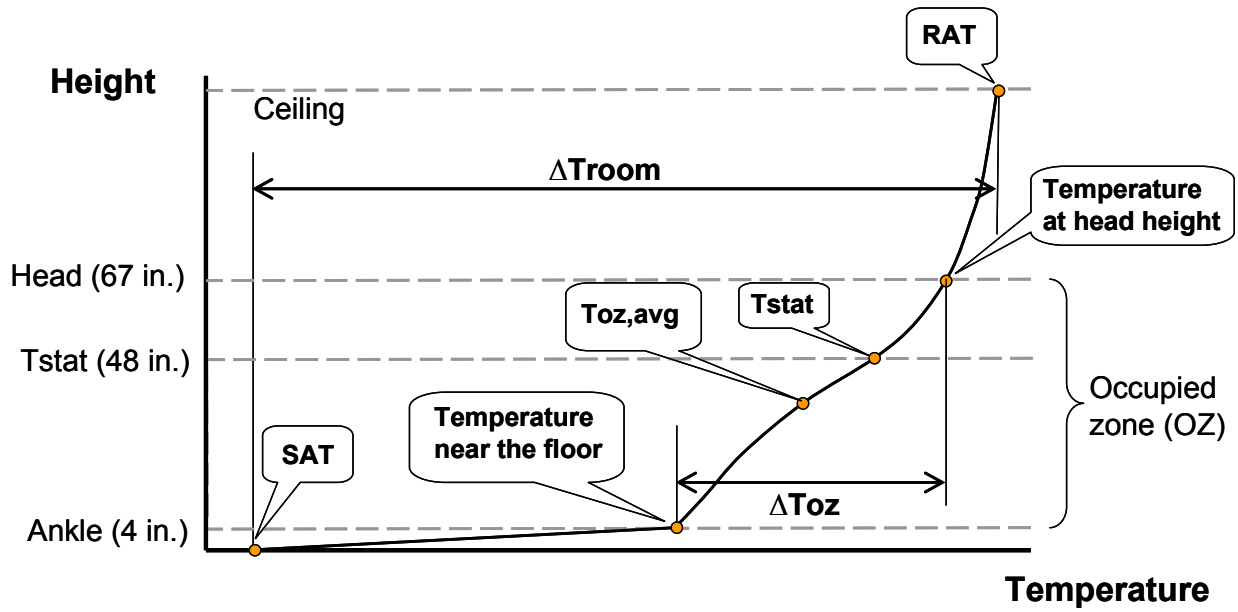

Figure 1. Schematic diagram of room air temperature profile in stratified UFAD system

The objective of this project reported below is to develop a practical and simplified design procedure for determining the amount of conditioned air required during cooling operation of a UFAD system in both interior and perimeter zones. Design engineers often cite methods for airside design sizing as one of the most important unanswered questions regarding UFAD system design. As described above, the determination of design cooling airflow quantities must take into account key differences between a thermally stratified space and a conventional well-mixed space. In the following report on the cooling airflow design tool, we describe (1) the development of the calculation methods that form the basis for the modeling engine, (2) user inputs and outputs, (3) preliminary validation by comparison to full-scale experimental data, (4) example results to demonstrate the behavior and sensitivity of the design tool to different user inputs and assumptions, (5) comparison to overhead system airflow quantities for equivalent comfort conditions, and (6) future work needed to improve and refine the design tool.

2 DEVELOPMENT OF DESIGN TOOL

The design tool has been developed to be a spreadsheet-based calculation procedure that in its final form will be easy to use by practicing design engineers. An important premise was to allow the user to apply standard cooling load calculation methods for conventional overhead systems, such as Trace, HAP, Elite software, and the new ASHRAE Radiant Time Series (RTS) procedure, as an input to the design tool. Figure 2 shows the current scheme for the design tool process. In addition to the load calculation, users will input several other parameters that define the design and desired operation of the UFAD system. For example, we know that airflow quantities will change with different levels of room air stratification, so

the user will be able to specify their target occupied zone temperature difference (ΔT_{oz}) as an input to the tool. Before proceeding to the main modeling engine, the calculated OH cooling load is modified based on an estimated Room Cooling Load Ratio, R, defined as the percentage of the OH cooling load that is to be assigned to the room in the UFAD airflow calculation. Since research has shown that about 1/3 of the load is transferred into the underfloor plenum [Bauman et al. 2006], this has the effect of reducing the amount of load that must be removed by the room airflow quantity.

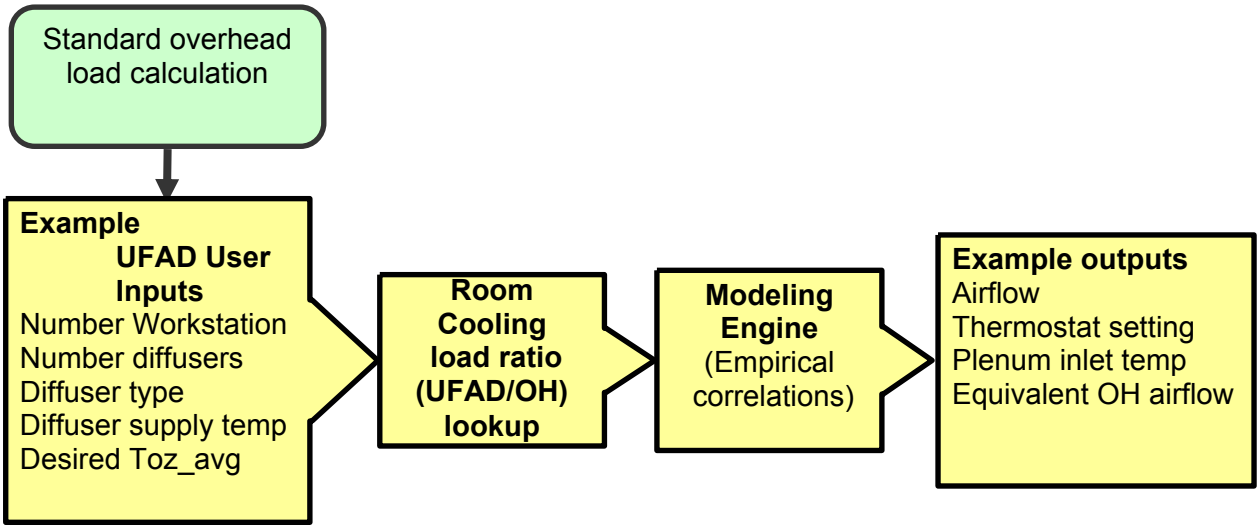

Figure 2. Design Tool Schematic

The main modeling engine then calculates a range of airflows that satisfy the input conditions. As described further below, this involves the prediction of a simplified room air temperature profile from which the key comfort parameters ($T_{oz,\ avg}$, ΔT_{oz}) can be derived. In addition to the cooling airflow quantity, the design tool is also capable of predicting additional helpful design information, including:

- Thermostat setting – Since most building zones are still controlled with a 4-ft high thermostat, this will provide a recommended thermostat setpoint temperature that will maintain the desired average occupied zone temperature under the input conditions of the design tool.
- Plenum entering air temperature – Using the Room Cooling Load Ratio defined above, the model will assign the remaining portion of the OH cooling load (~1/3) to the underfloor plenum. This will allow the calculation of the plenum inlet temperature (similar to a coil leaving temperature in the air handler) required to maintain the desired average diffuser supply air temperature (e.g., 65°F).
- Equivalent OH airflow – For comparison purposes, the model will also calculate the cooling airflow quantity for a conventional overhead system based on the same OH cooling load and equivalent comfort conditions (i.e., OH thermostat setpoint temperature = UFAD $T_{oz,\ avg}$).

In its current form, the tool simulates a simplified temperature profile constructed of two line segments that are connected at the 4-ft thermostat height. This assumes, for design purposes, that a controlled temperature profile passes through the thermostat setpoint. The bottom point of the profile represents the 4-in. ankle height temperature and the top point represents the return air temperature at the ceiling (see Figure 3).

In order to derive this temperature profile, we have constructed the tool to perform its steady state calculations for a range of possible airflows covering the expected airflows for typical interior office spaces. As described later, future work will address airflow calculations for perimeter zones and may also include an iterative procedure. Presently, for each airflow rate, the model calculates the return air temperature using a heat balance equation based on the modified cooling load assigned to the room. Temperatures at the 4-in. and 67-in. heights are determined through empirical correlations with our full-

scale experimental data (see section on RAS full-scale testing in final report [Bauman, Webster et al. 2006]. From the return air temperature and the temperature at a height of 67 in., we calculate the temperature at 48 in. (thermostat) assuming that the temperature profile runs directly through it (see Figure 3). We then calculate the temperature difference in the occupied zone and the average temperature in the occupied zone based on this derived profile. To determine the temperature of air entering the supply plenum, we use a heat balance on the plenum based on an assumed 65°F temperature leaving the supply plenum through the diffusers (future refinements to the design tool will allow alternative diffuser supply temperatures) and the portion of the OH room cooling load estimated to be entering the supply plenum.

Figure 3 shows an example of the simplified temperature profile calculated by the design tool in comparison to full-scale measured data. The vertical temperature difference between head height (67 in.) and ankle height (4 in.), ΔT_{oz} should not exceed 5 °F, as specified by ASHRAE Standard 55-2004 [ASHRAE 2004]. The average temperature in the occupied zone, $T_{avg,\ oz}$ is determined based on the temperature profile between 4 and 67 inches.

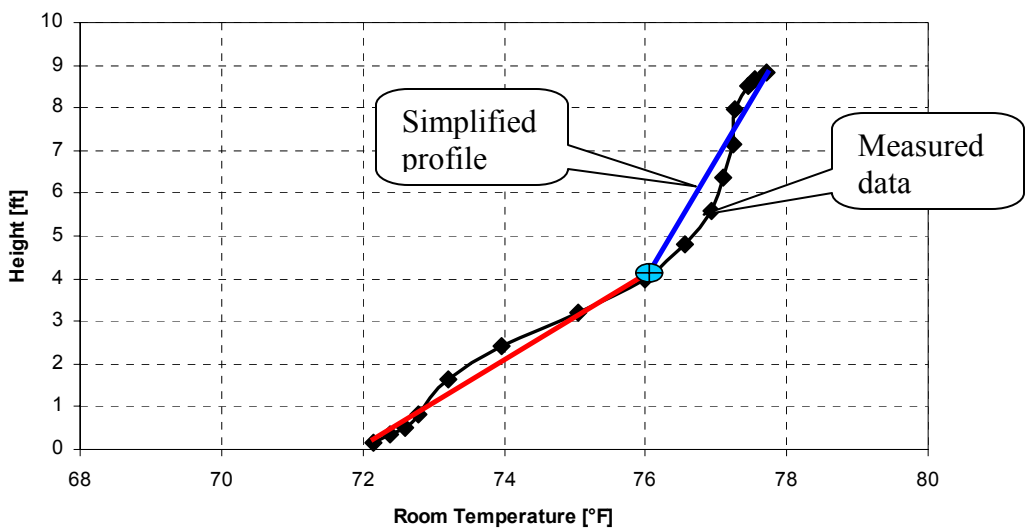

Figure 3. Simplified Temperature Stratification Profile

2.1 DETAILED MODEL PROCESS

2.1.1 INPUTS

A_f	floor area, [ft^2]
W	heat load, total heat gain as calculated by standard cooling load design tools (and assuming there is no supply plenum) [W]
ws	number of workstations
Diffuser type	SW (swirl), HD (horizontal discharge swirl), or VA (variable area)

2.1.2 CALCULATION PROCESS

In the current setup of the model we begin with the above user inputs and calculate the results for a range of airflows (~0.3-0.8 cfm/ft^2). For each airflow rate, we use the following equations to solve for T_R, Γ, Φ_4, Φ_{67}, T_4, and T_{67}.

1. The return air temperature, T_R:

$$T_R = T_S + \frac{(W \times R \times 3.412)}{(1.08 \times Q_{room})} \qquad (1)$$

Where

T_R	= Return air temperature, [°F]
T_S	= temperature of air supplied at diffuser, [°F]
W	= heat load, [W]
R	= room cooling load ratio
Q_{room}	= airflow (through diffusers plus category II leakage), [cfm]

2. The non-dimensional parameter, Γ, representing the ratio of buoyancy to inertia forces in the room:

$$\Gamma = \frac{(Q\cos\Theta)^{3/2}\left(\frac{0.0281 \times W \times R}{1000}\right)^{1/2}}{ws \times \left(\frac{n}{ws} \times A_d\right)^{5/4}} \qquad (2)$$

Where

Γ	= non-dimensional parameter, Gamma
Q	= airflow (through diffusers), [m^3/s]
Cos Θ	= angle factor specific to diffuser type
ws	= number of workstations
n	= number of diffusers
A_d	= diffuser effective area, [m^2]

3. The non-dimensional temperature at the 4-in. height, Φ_4:

For SW and HD (based on correlation of experimental data) [Bauman, Webster et al. 2006]:

$$\Phi_4 = 0.2667 \times \Gamma^{0.3099} \qquad (3a)$$

For VA:

$$\Phi_4 = 0.486 \times \Gamma^{0.1176} \qquad (3b)$$

Where

Φ = non-dimensional temperature: $\frac{T_4 - T_S}{T_R - T_S}$

4. The air temperature at 4 in., T_4:

$$T_4 = \Phi_4 (T_R - T_S) + T_S \tag{4}$$

Where

T_4 = temperature at 4", [°F] (bottom of occupied zone)

5. The non-dimensional temperature at the 67-in. height (based on experimental data), Φ_{67}:

$$\Phi_{67SW} = 0.94 \tag{5a}$$
$$\Phi_{67HD} = 0.93 \tag{5b}$$
$$\Phi_{67VA} = 0.96 \tag{5c}$$

6. The air temperature at 67 in., T_{67}:

$$T_{67} = \Phi_{67} (T_R - T_S) + T_S \tag{6}$$

We then use the values for T_4, T_{67}, and T_R to solve for T_{set}, $T_{oz,\,avg}$, and ΔT_{oz}

7. The setpoint temperature (air temperature at 48 in.), T_{set}:

$$T_{set} = \left(\frac{T_R - T_S}{108 - 67} \right) \times (48 - 108) + T_R \tag{7}$$

8. The average temperature in the occupied zone, $T_{oz,\,avg}$:

$$T_{oz,avg} = \left(\frac{1}{67-4} \right) \times \left(\left((T_{67} + T_{set}) \times \left(\frac{67-48}{2} \right) \right) + \left((T_{set} + T_4) \times \left(\frac{48-4}{2} \right) \right) \right) \tag{8}$$

9. The temperature stratification in the occupied zone (4 in. to 67 in.), ΔT_{oz}:

$$\Delta T_{oz} = T_{67"} - T_{4"} \tag{9}$$

Where

ΔT_{oz} = vertical temperature difference in occupied zone, [°F]

We can then use the room cooling load ratio, R, and T_s to find the air temperature entering the underfloor plenum:

$$T_{splenum} = T_S - \left(\frac{W \times (1 - R) \times 3.413}{1.08 \times Q_{room}} \right) \quad (10)$$

Where

$T_{splenum}$ = temperature of air entering supply plenum, [°F]

Currently the airflow used in this calculation is the same as the room airflow (Q_{room}). However, in many systems the plenum supplies air to both interior and perimeter zones so that the thermal decay will be produced by a combination of these airflow quantities, which would tend to decrease the magnitude of temperature gain. In the future we will add elements to the model to consider this important factor.

2.1.3 ACCOUNTING FOR LEAKAGE

The model currently uses two separate terms for airflow. Q is the airflow that passes through the diffusers in the raised floor. Q_{room} is the total airflow to the room and is equal to Q plus Category II leakage. Category II leakage is defined as uncontrolled air leakage from the pressurized underfloor plenum (through gaps between panels, electrical floor outlets, etc.) that enters the room and can still contribute to the removal of the heat load. We plan to use both terms at different points in the process. In the current version we are using the flow through the diffusers only (Q). In future versions, we will use a leakage percentage (entered by the user with guidance from the tool) to calculate the Q_{room}. In the current model development, in order to compare to our experimental test data, we have included the calculation of Q_{room} based on our measurements of floor leakage in the experimental chamber. These experiments were conducted with the floor panel joints sealed with tape so the leakage rates are relatively small. For real buildings the leakage can be significant and highly variable, project to project. Also, our experimental work demonstrates how the airflow divides between leakage and diffusers, and their combined effect on stratification in the occupied zone. It also appears, in general, to affect VA systems differently than SW systems. In the future we will develop procedures to properly account for these effects.

Further research is needed to study the effect of leakage on room air stratification, though we expect that higher leakage will result in colder temperatures near the floor and possible higher stratification in the occupied zone.

3 MODEL BEHAVIOR

As the design tool development progressed, we began to investigate how it performs compared to our current understanding of underfloor air distribution systems when subjected to a variety of conditions in order to ensure that the tool is producing intuitive and accurate results.

3.1 COMPARISON TO TEST DATA

We have compared the output of the design tool in its current form to a number of tests conducted in the York test chamber [Bauman, Webster et al. 2006] to determine how close the calculated temperature profile reflects the actual temperature profile. Test data exists for swirl, horizontal discharge swirl, and variable area diffuser types. It is this same test data that is used to develop the correlations on which the tool is based.

3.1.1 SWIRL

Figure 4 shows a comparison between a measured temperature profile (test data) and the calculated temperature profile from the design tool for one particular test. Comparisons for the whole set of experiments we used produced similar results. The lower portion of the profile tends to line up well, and the return air temperature seems to consistently be under-predicted by approximately 0.5°F by the design tool. (See error analysis below.)

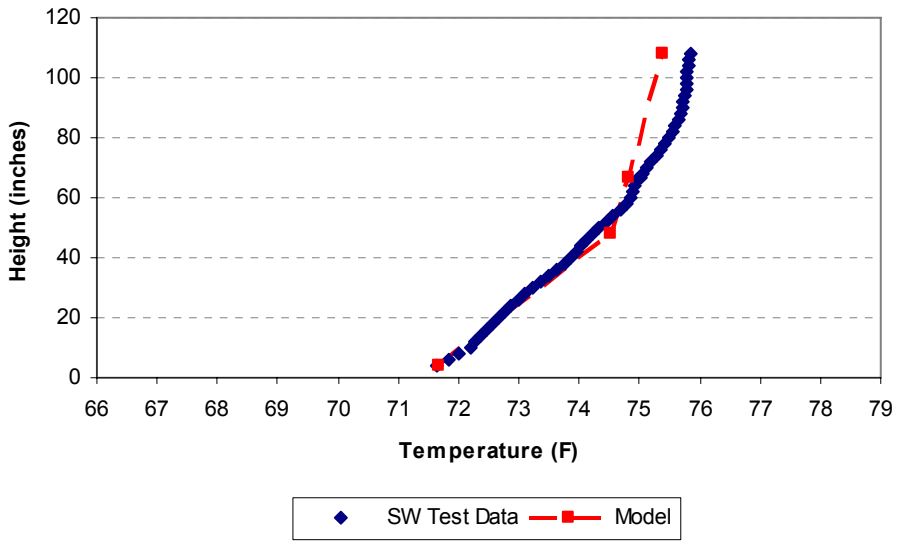

Figure 4. Swirl: Calculated and Measured Temperature Profile

3.1.2 HORIZONTAL DISCHARGE SWIRL

Figure 5 shows a typical comparison of the measured (test data) and calculated profiles for a horizontal discharge swirl system. This figure is representative of the behavior we observed for the design tool compared to the test data for all of the HD diffuser tests. The calculated temperature at the 4" and 67" heights are close to the measured, and the return air temperature is consistently under-predicted, but by a small margin (less than 0.5 °F). The average temperature of the occupied zone, however, is consistently over-predicted as can be seen by the gap between the measured and calculated profiles in the occupied zone (4" to 67").

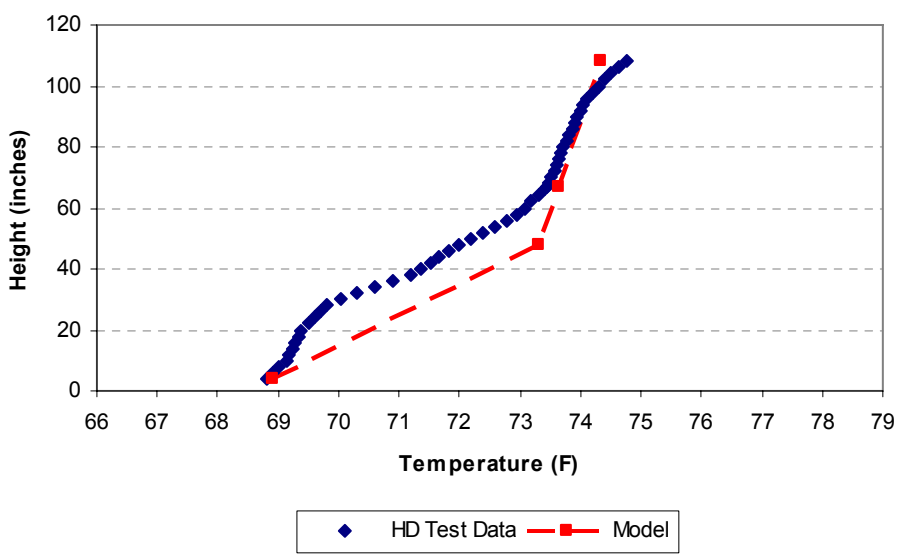

Figure 5. Horizontal Discharge Swirl: Calculated and Measured Temperature Profile

3.1.3 VARIABLE AREA

In the variable area diffuser case, the calculated profiles tend to fall close to the measured, but seem to be often under predicting temperatures at all heights. The temperature at the 4" height is not as closely

predicted in the variable area case as it is for the swirl and HD swirl systems. In this case, the temperature difference in the occupied zone (ΔToz) is well predicted, as the measured and calculated profiles have the same shape. However, the calculated average temperature in the occupied zone is generally slightly lower than the measured value as shown.

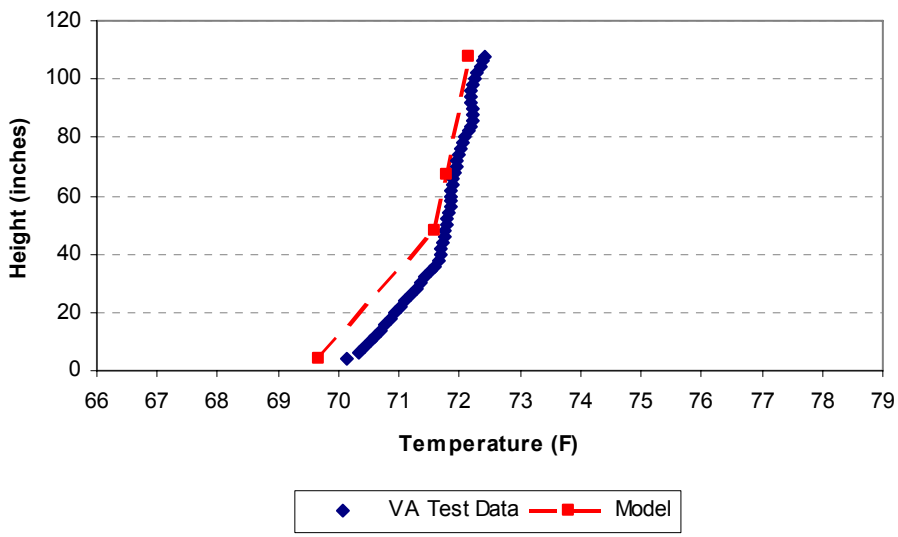

Figure 6. Variable Area: Calculated and Measured Temperature Profile

3.1.4 OVERALL ERROR ANALYSIS

Table 1 shows the calculated error when comparing the design tool calculations to all of our test data. The first two rows show that, on average, the design tool is predicting $T_{oz,avg}$ for swirl diffusers very accurately (within 0.01°F) on average. The standard deviation of this error is 0.32°F. The stratification of the occupied zone, ΔT_{oz}, for swirl diffusers is also predicted very accurately (within 0.02°F) with a standard deviation of this error of 0.52°F. Comparison results for the HD and VA diffusers are also summarized in Table 1.

Table 1. Comparison to Test Data

	SW	HD	VA
$T_{oz,avg}$ Average Error [°F]	0.01	-0.66	0.38
$T_{oz,avg}$ StDev of Error [°F]	0.32	0.80	0.32
ΔT_{oz} Average Error [°F]	0.02	-0.68	-0.75
ΔT_{oz} StDev of Error [°F]	0.52	0.53	0.35

Comparing the results of the HD swirl and the standard swirl demonstrate the importance of the *shape* of the profile in the occupied zone. This suggests that future versions of the model may require a way to adjust the simulated profile based on expected shape. More work will be necessary to fully characterize these various shapes.

3.2 SENSITIVITY STUDIES

3.2.1 VARIATION OF LOAD

One of the user inputs to the design tool will be the cooling load as derived from a standard load calculation procedure. If we presume that the goal of the designer is to use an airflow that would yield an

average occupied zone temperature ($T_{oz,\ avg}$) of 73°F to 75°F and an occupied zone stratification (ΔT_{oz}) of 3°F to 5°F then we can use the model to explore a range of load conditions to determine what airflows will satisfy these comfort criteria. In Figure 7 through Figure 9, $T_{oz,\ avg}$ and ΔT_{oz} are plotted against airflow (cfm/sf) for three different load conditions in the space based on design tool results assuming a swirl diffuser, one diffuser per workstation, and a 65°F diffuser discharge temperature (Ts). The blue shaded region is the range of airflows that satisfy the $T_{oz,\ avg}$ design condition (73°F to 75°F) and the red shaded region is the range of airflows that satisfy the ΔT_{oz} design condition (3°F to 5°F). Where the two overlap is the range of airflows that satisfy both design conditions. We assumed a Room Cooling Load Ratio of 70% for all of these studies.

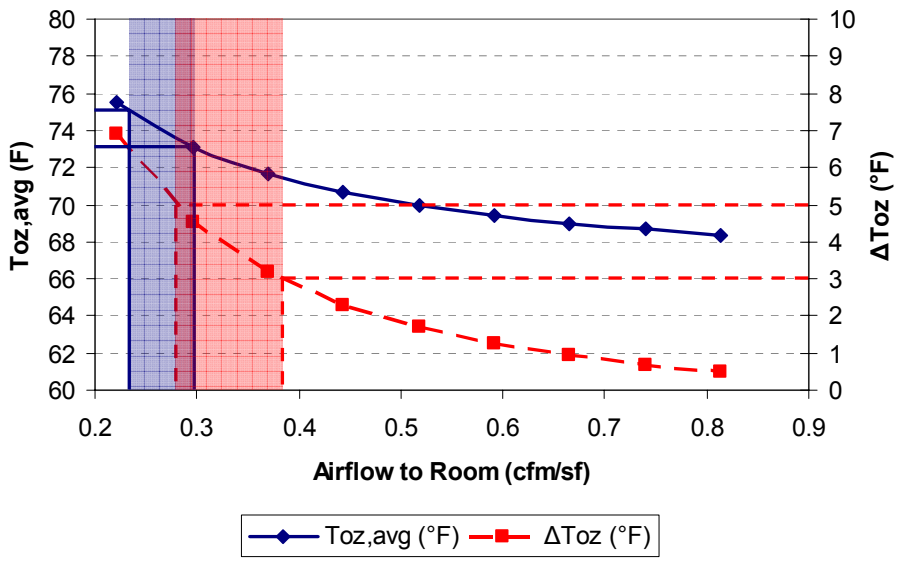

Figure 7. Design Conditions vs. Airflow, at a Cooling Load of 1.5 W/sf

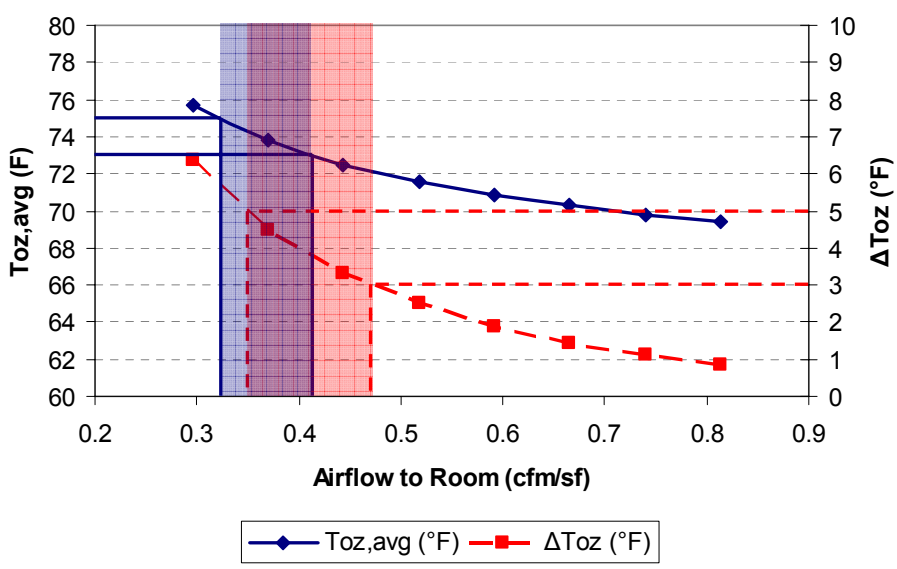

Figure 8. Design Conditions vs. Airflow, at a Cooling Load of 2.0 W/sf

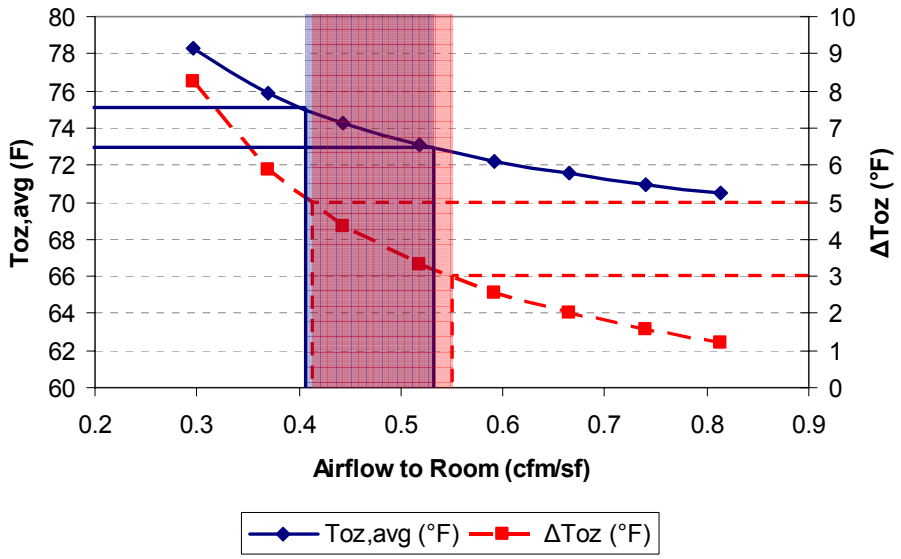

Figure 9. Design Conditions vs. Airflow, at a Cooling Load of 2.5 W/sf

Note that raising the load in the space alters the location of the $T_{oz,\ avg}$ line indicating that as the load is raised, more air is needed to keep the average occupied zone temperature at the same level. Figure 7 shows that at a load of 1.5 W/sf, an airflow of approximately 0.3 cfm/sf would be needed to satisfy the design conditions, whereas Figure 9 shows a situation where approximately 0.4 to 0.5 cfm/sf would be required to meet design conditions. As the load is increased (from 1.5 W/sf in Figure 7 to 2.5 W/sf in Figure 9), more air is needed to keep the $T_{oz,\ avg}$ within design limits, but the temperature stratification is less affected by the higher loads, as indicated by the relatively lesser movement of the red column with respect to the blue column. Horizontal discharge and variable area systems show the same pattern.

3.2.2 VARIATION OF NUMBER OF DIFFUSERS

The design tool also allows the user to input the number of diffusers per workstation in the space. As more diffusers are added for the same total room airflow, each diffuser delivers less air, thereby reducing the diffuser throw and amount of mixing in the room. For most UFAD installations it would be expected that one diffuser/workstation, plus a few more for corridors and other open use spaces, would result in values between 1-1.5 diffusers/workstation. Figure 10 through Figure 13 show the consequence of varying this parameter assuming the same conditions and design limits as discussed in Section 3.2.1 above.

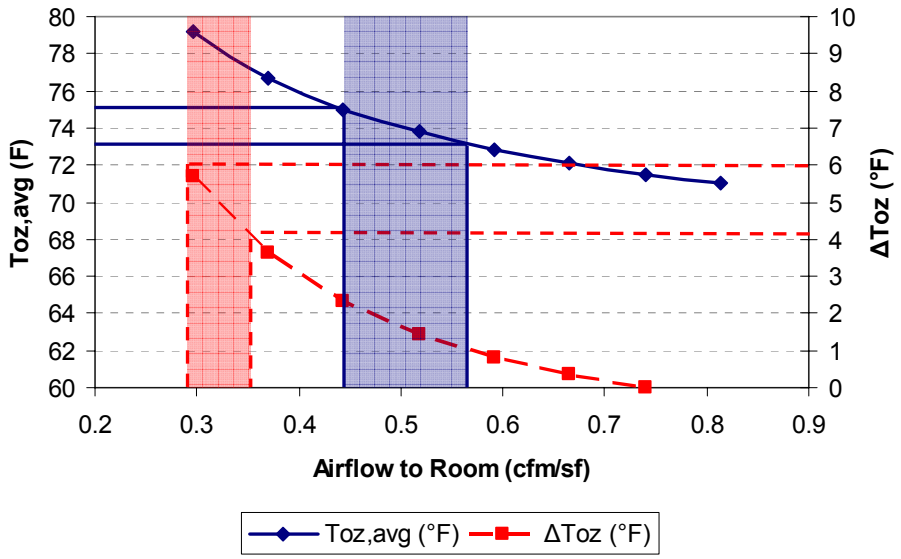

Figure 10. Design Conditions vs. Airflow, 0.5 Diffusers / Workstation

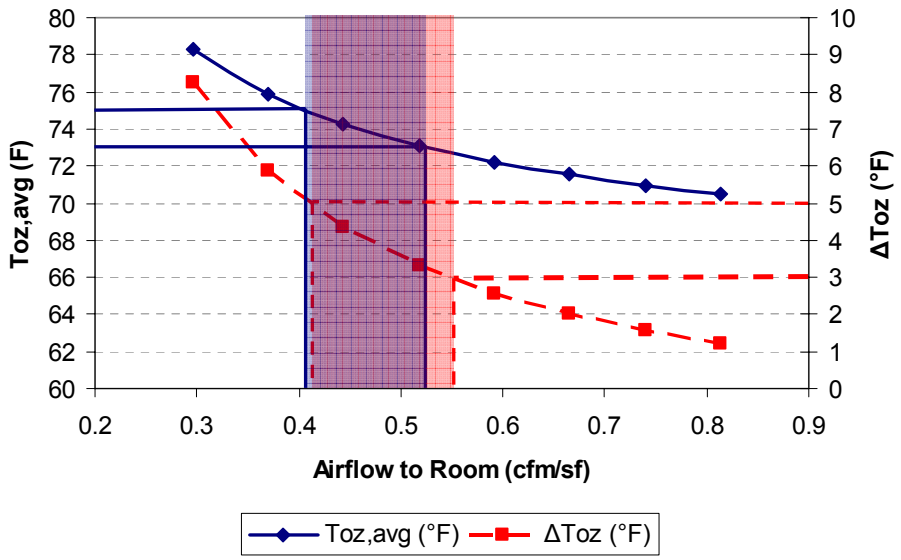

Figure 11. Design Conditions vs. Airflow, 1.0 Diffusers / Workstation

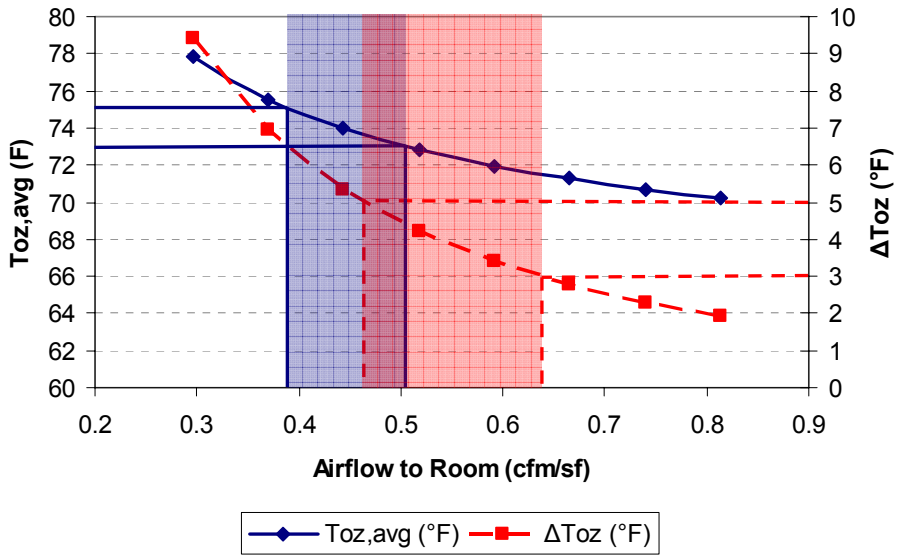

Figure 12. Design Conditions vs. Airflow, 1.5 Diffusers / Workstation

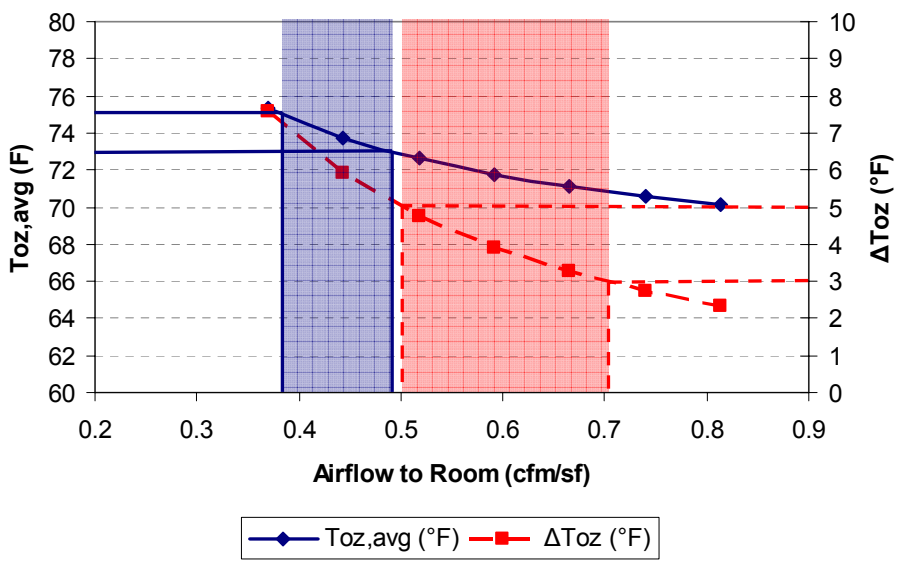

Figure 13. Design Conditions vs. Airflow, 2.0 Diffusers / Workstation

In this series of figures, we see that as more diffusers are added to the space, less airflow is needed to maintain the desired average temperature in the occupied zone, but more airflow is needed to maintain the stratification desired. At the low end of only 0.5 diffusers / workstation and at the high end of 2.0 diffusers / workstation, there is no single airflow quantity that would meet both design conditions. However, Figure 11 and Figure 12 show that for 1.0 and 1.5 diffusers / workstation, there is a range of airflows that will satisfy both. Again, this pattern is the same for the horizontal discharge and variable area conditions, though the values vary.

3.2.3 VARIATION OF SUPPLY AIR TEMPERATURE

The current version of the design tool assumes that the average air temperature leaving the supply plenum (at the diffusers, Ts) is set at 65°F. Figure 14 through Figure 16 show the consequence of varying this

supply temperature. As the supply temperature is increased, the stratification curve is unaffected and we find that more airflow is required to satisfy the average temperature design condition. It is easy to see that if we raise the supply air temperature much more we will again no longer be able to satisfy both design conditions. Additionally, as the supply air temperature is increased, we find that the $T_{oz,avg}$ vs. airflow curve flattens out, meaning that higher supply air temperatures decrease the sensitivity of average temperature to changing airflow.

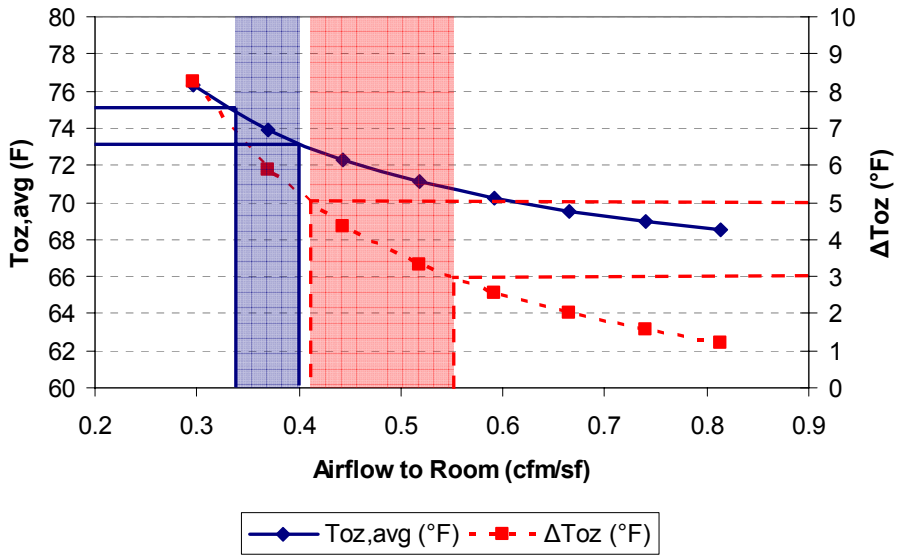

Figure 14. Design Conditions vs. Airflow, 63°F Air Leaving Supply Plenum

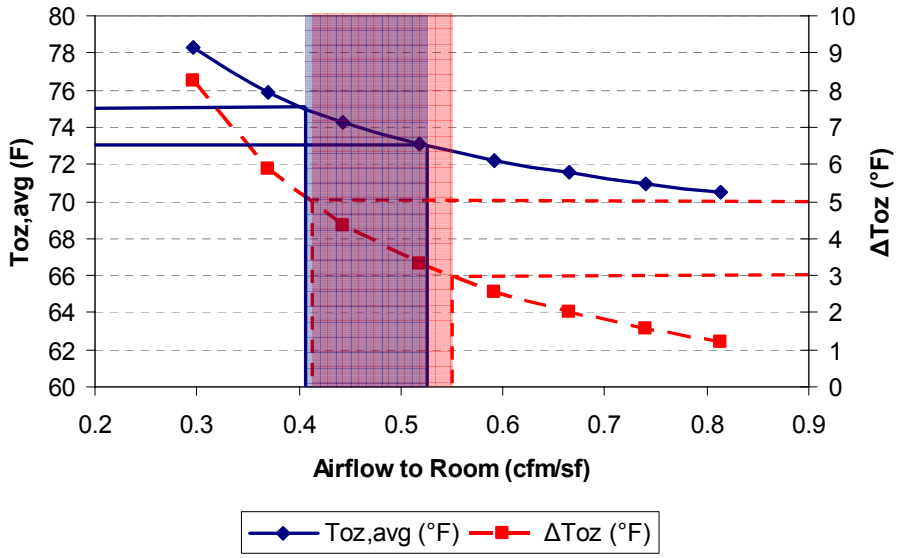

Figure 15. Design Conditions vs. Airflow, 65°F Air Leaving Supply Plenum

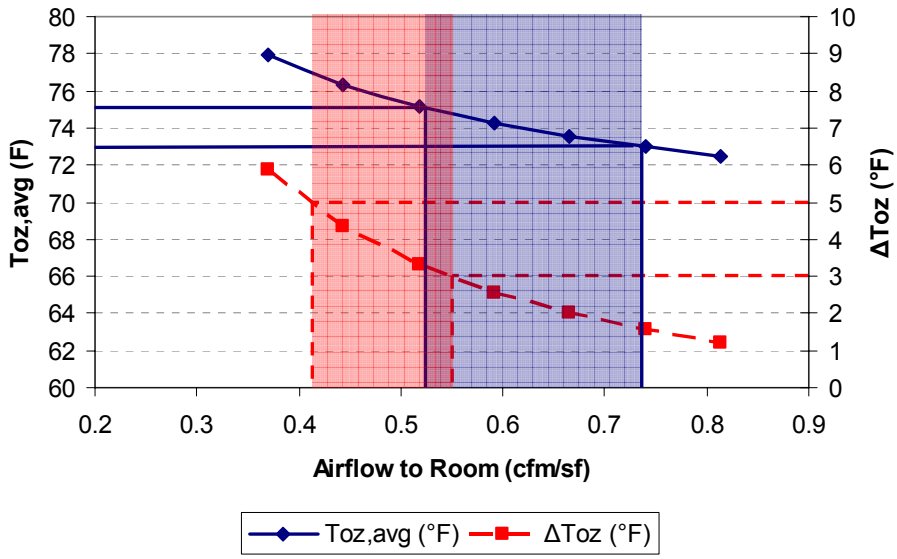

Figure 16. Design Conditions vs. Airflow, 67°F Air Leaving Supply Plenum

Figure 17 shows variations in the supply plenum entering temperature (Tsplenum) as a function of changes in the diffuser discharge temperature (Ts) and airflow rate. By combining the information from Figure 14 through Figure 16, we see that in order to satisfy design conditions we can either use a supply temperature (diffuser discharge temperature, Ts) of 65°F at an airflow of 0.42 cfm/sf to 0.53 cfm/sf OR a supply temperature of 67°F at an airflow of 0.53 cfm/sf to 0.55 cfm/sf. Taking this information to Figure 17, we see that the 65°F condition requires that the air temperature entering the plenum (Tsplenum) be between 59.5°F to 60.5°F whereas a 67°F condition requires a Tsplenum between approximately 62.2°F to 62.5°F. In short, the design tool shows that we can meet the design conditions by either using a smaller quantity of cooler air, or a larger quantity of warmer air.

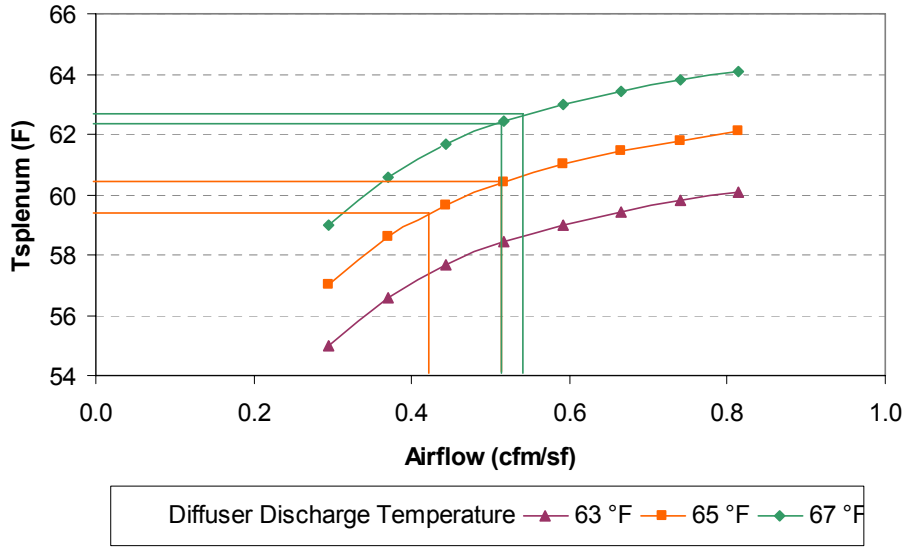

Figure 17. Air Temperature Entering Supply Plenum (Tsplenum) vs. Airflow for different diffuser discharge temperatures (not including airflow to the perimeter)

3.2.4 VARIATION OF ROOM COOLING LOAD RATIO

Future research will be done to assist in determining appropriate estimates of the Room Cooling Load Ratio, or the fraction of the load that will be applied to the room air. The remaining load will be applied to the supply plenum. As we vary R, the zone of acceptable airflow varies just as it would if the load itself were varied, as discussed in Section 3.2.1 of this report. The difference, however, between varying the load itself and varying R, occurs in the resulting Tsplenum. Figure 18 shows the consequence of varying R on the required plenum supply air temperature. As more load is applied to the plenum (lower R), the temperature of the air supplied to the plenum must be colder, though the effect is slightly less pronounced at higher airflow rates.

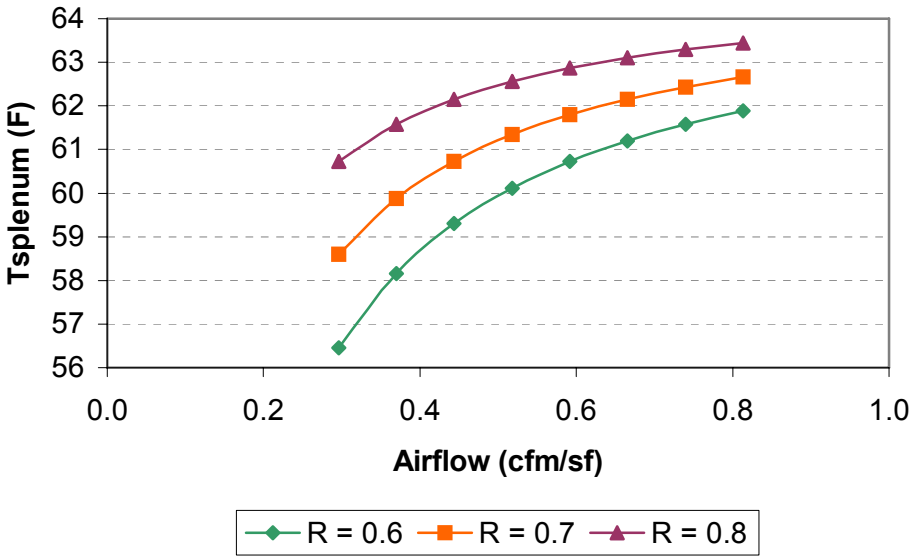

Figure 18. Tsplenum vs. Airflow, Varying R

3.2.5 THERMOSTAT SETPOINT VS. AVERAGE TEMPERATURE OF THE OCCUPIED ZONE

In conventional overhead systems that aim to maintain a uniform well-mixed environment, the assumption is made by building operators that the thermostat setpoint is the same as the average temperature experienced in the occupied zone. Under the stratified conditions of underfloor air systems, this assumption is no longer correct. From the design tool, we are able to derive the difference between these two values. This information can be used to determine what 4-ft thermostat setpoint temperature should be maintained (assuming that the space temperature is controlled using a traditional single-point measurement at the 4-ft height) to provide the desired average occupied zone temperature. Figure 18 shows the general pattern of this relationship, based on an example of a swirl diffuser at a load of 2.5 W/sf, 1 diffuser / workstation, and a supply plenum leaving temperature of 65°F. This pattern remains similar for other system types, diffuser densities and supply temperatures, though the values and distance between the curves do vary. As seen here, the average temperature in the occupied zone tends to be lower than the single temperature at a height of 48 in., but as more airflow is supplied and the system operates closer to a well-mixed system, the two become closer and closer together.

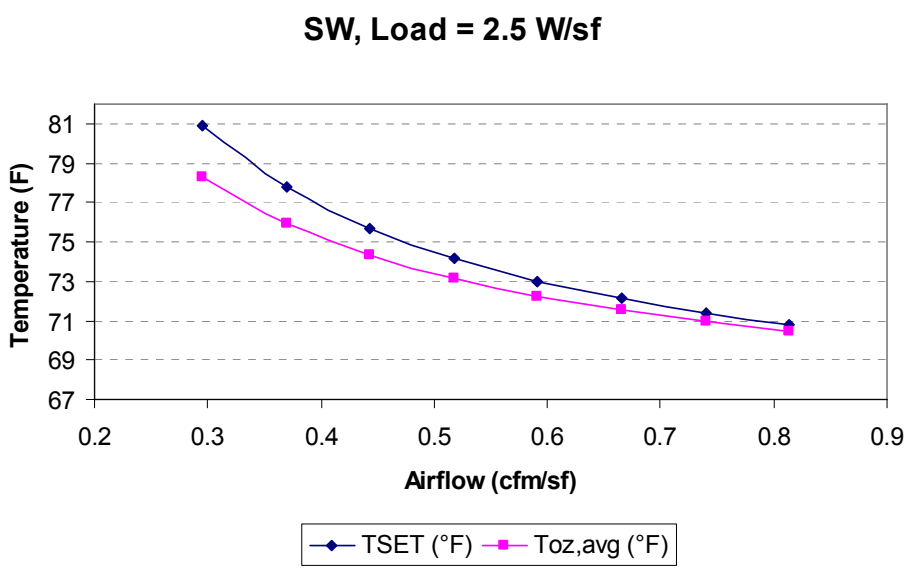

Figure 19. Example of 4-ft Thermostat Setpoint vs. Average Temperature in the Occupied Zone

3.3 COMPARISON TO OVERHEAD AIRFLOW

The design tool will also be capable of calculating the range of airflows that would be used in a comparable overhead system subject to the same loads and attempting to meet the same average temperatures as that of the occupied zone in the UFAD case. Initial studies have shown that this airflow range is close to the predicted airflow required to meet the design conditions for UFAD and often overlaps as shown in the example in Figure 20. In this figure, the dashed box represents the overhead airflow needed to achieve an average temperature in the room of 73°F to 75°F assuming a supply air temperature of 55°F to 57°F and a load of 2 W/sf. The UFAD case assumes a diffuser discharge (Ts) temperature of 65ºF. Although difficult to read in the figure, the zones of acceptable airflow are as follows: UFAD system: 0.35-0.41 cfm/sf, and OH system: 0.31-0.39 cfm/sf.

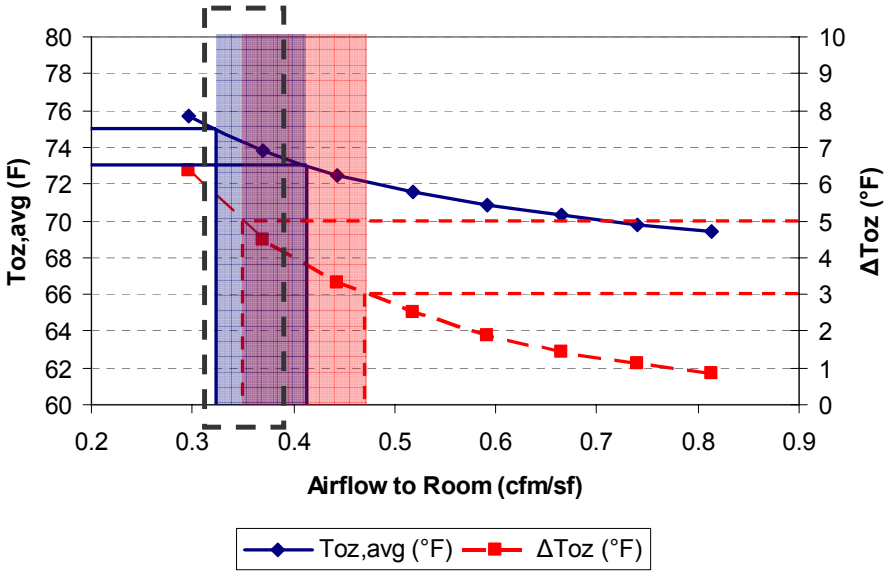

Figure 20. Overhead Airflow Needed Under Same Load Conditions

4 SUMMARY AND CONCLUSIONS

In this report, we have described the development of a practical and simplified design procedure for determining the amount of conditioned air required during cooling operation of a UFAD system. To date, such a design tool has not been available to the building design community. The calculation of design cooling airflow quantities for UFAD systems must take into account key differences between a thermally stratified space and a conventional well-mixed space. As described elsewhere in our final report [Bauman, Webster et al. 2006], there are two primary considerations that distinguish the design and cooling operation of a UFAD system in a multi-story building from that of a conventional overhead (OH) system: (1) thermal performance of underfloor plenums, and (2) room air stratification. Both have important implications for the determination of cooling airflow quantities.

The design tool is a spreadsheet-based calculation procedure that in its final form will be easy to use by practicing design engineers. To make the tool more compatible with existing techniques used by the industry, the model will accept as an input the results of standard cooling load calculation methods for conventional OH systems, such as Trace, HAP, Elite software, and the new ASHRAE Radiant Time Series (RTS) procedure. Additional user inputs include diffuser type, diffuser supply air temperature, number of workstations (or other load configuration information), and desired comfort conditions. For a stratified UFAD environment, comfort conditions are defined based on the following two parameters:

- The average occupied zone temperature ($T_{oz,\ avg}$), calculated as the average of the measured temperature profile from foot level (4 in.) to head level (67 in.); and
- The occupied zone temperature difference (ΔT_{oz}), calculated as the head-foot temperature difference.

The development of the design tool was based on empirical correlations derived from the full-scale tests conducted in the York test chamber (see Part II of final report [Bauman, Webster et al. 2006]). These tests investigated three different diffuser types (swirl, horizontal discharge swirl, and variable-area), which are all included in the model. The design tool also will be able to account for the amount of Category II air leakage entering the conditioned space. The design tool predicts a range of acceptable cooling airflows, within which the target comfort criteria are satisfied for the design input assumptions of the model. In addition, the design tool is also capable of providing additional design information, including: (1) recommended 4-ft thermostat setpoint temperature that will maintain the desired average occupied zone temperature, (2) plenum entering air temperature, and (3) for comparison purposes, the cooling airflow quantity for a conventional OH system based on the same cooling load and equivalent comfort conditions.

Preliminary comparisons of design tool predictions with full-scale test data showed reasonable agreement. A series of sensitivity studies were conducted to demonstrate how the design tool model behaved in response to variations in cooling load, number of diffusers, diffuser supply air temperature, and room cooling load ratio (portion of OH system cooling load that is assigned to the room for UFAD airflow calculations). A comparison between predicted UFAD cooling airflow with that of an OH system for equivalent design conditions yielded very similar air quantities for the example shown.

The current preliminary version of the design tool is suitable for interior zone airside sizing calculations. As described below, additional work is needed to improve and refine the design tool, including adding perimeter zone airflow calculations, conducting supporting whole-building energy simulations with new refined versions of UFAD EnergyPlus, and completing further model validation.

5 FUTURE WORK

The development work completed to this point represents an initial simplified version of a validated and fully functioning UFAD cooling airflow design tool that can be used by the design community. Further work is required in a number of areas to complete the intended development of the design tool, including:

- Studies to determine how to assign the room cooling load ratio, R. The current preliminary model assumes a fixed value of 70%. Supporting simulation studies using the newly developed version of EnergyPlus/UFAD are needed to investigate the room cooling load ratios over a wider range of realistic building configurations. Implementation of this feature will require additional user inputs describing the underfloor plenum configuration (e.g., plenum in multi-story building, ground floor plenum on top of slab-on-grade, etc.).
- Add the capability to determine perimeter zone cooling airflow quantities. The current version was based only on empirical data for interior zone conditions.
- Conduct a more complete validation of the design tool based on full-scale laboratory data, as well as available field data from operational UFAD installations.
- Develop a more comprehensive supply plenum model to consider effects of perimeter zone airflow and interaction between plenum temperatures and room cooling load ratio.
- Develop a Category II leakage model and a guide to assist designers in selecting an appropriate value.
- Work with designers to ensure closer correlation between the language and units used in the design tool and standard load calculation procedures.
- Incorporate a list of available diffusers and their associated effective areas and discharge angles.
- Consider ways to account for different load configurations representative of other building types (e.g., libraries, schools, etc.).
- Complete user interface development:
    - Finalization of inputs / outputs
    - Provision of reasonable input parameter ranges and warnings when outputs are out of range
    - Display of outputs

6 NOMENCLATURE

A_f	= floor area, [ft^2]
W	= heat load, [W]
ws	= number of workstations
Diffuser type	= SW (swirl), HD (horizontal discharge swirl), or VA (variable area)
A_d	= diffuser effective area, [m^2]
T_R	= Return air temperature, [°F]
T_S	= temperature of air supplied at diffuser, [°F]
W	= heat load, [W]
R	= room cooling load ratio
Q_{room}	= airflow, [ft^3/min]
Γ	= non-dimensional parameter
Cos Θ	= angle factor specific to diffuser type
ws	= number of workstations

n	= number of diffusers
Φ	= non-dimensional temperature
$T_{4"}$	= temperature at 4", [°F] (lower bound of ASHRAE occupied zone)
$T_{67"}$	= temperature at 67", [°F] (upper bound of ASHRAE occupied zone)
T_{set}	= setpoint temperature at 48", [°F]
$T_{oz, avg}$	= average temperature in the occupied zone, [°F]
ΔT_{oz}	= temperature range in occupied zone, [°F]
$T_{splenum}$	= temperature of air supplied to the underfloor plenum, [°F]

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