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Schiavon, Stefano
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Author:

[Schiavon, Stefano](#), Center for the Built Environment, University of California, Berkeley
[Lee, Kwang Ho](#), Center for the Built Environment, University of California, Berkeley
[Bauman, Fred](#), Berkeley
[Webster, Tom](#)

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Abstract:

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- a) transforms the zone design cooling load calculated for a standard overhead (OH) mixing system into the design cooling load for a stratified UFAD system, accounting for differences in design day cooling load profiles for OH and UFAD systems;
- b) splits the total UFAD cooling load into three fractions, supply plenum (SPF), zone, or room, (ZF), and return plenum (RPF);
- c) manages the thermal comfort in a vertically stratified environment;
- d) predicts the air temperature profiles and the setpoint temperature at the thermostat;

- e) models the air diffusers;
- f) predicts the design airflow rate; and
- g) models commonly used plenum configurations.



Development of a Simplified Cooling Load Design Tool for Underfloor Air Distribution (UFAD) Systems

Stefano Schiavon, Kwang Ho Lee, Fred Bauman, and Tom Webster
Center for the Built Environment
University of California
Berkeley, CA 94720-1839

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SUMMARY

This paper summarizes the assumptions and equations behind a new spreadsheet-based cooling load design tool for underfloor air distribution (UFAD) systems developed by the Center for the Built Environment at University of California, Berkeley. After briefly reviewing previous UFAD design tools, we describe in detail how the design tool:

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INTRODUCTION: OTHER UFAD DESIGN TOOLS

The most common cooling airflow design methods for UFAD systems used in practice have been described in the following papers: Sodec and Craig [1], Loudermilk [2], YORK [3] and Bauman et al. [4]. In this section we briefly review them (except the one developed by Bauman et al. [4].).

Displacement ventilation is a closely related, but older, ventilation strategy compared to UFAD, and as a result, has received considerably more research attention. There are many design guidelines for displacement ventilation [5-10]. Although many displacement ventilation concepts are used in UFAD design tools, there are two major differences: (1) the effect of heat transfer into the underfloor plenum on the cooling airflow design [4] and (2) the consequence of mixing with the upper layer when the vertical momentum flux from the cooling diffusers is sufficient to penetrate the density interface [11]. A design method for UFAD should be able to take into account these two phenomena. Lin and Linden [11] affirmed that models that neglect the effect of the vertical momentum on vertical air stratification are more appropriate for wall mounted diffusers at floor level (displacement ventilation).

Sodec and Craig [1] wrote a guideline for the selection, application and design of underfloor air distribution systems. The suggested design methods are preliminary and outdated, e.g., it is not able to predict the temperature stratification in the room and it does not consider the influence of the plenum on the airflow rate.

Loudermilk [2] proposed a method based on the separation of the conditioned space into two distinct horizontal zones, a mixing zone within the lower levels of the space and displacement type flow in the upper zone. Assumptions of the method are:

- a) the height of the mixing zone is equal to the height at which the supply outlet discharge velocity has been reduced to 50 fpm (0.25m/s), i.e., throw height of diffuser,
- b) the dimensionless temperature near the floor is equal to 0.4,
- c) convective heat gains that originate above the occupied zone may be neglected in calculation of the design airflow rate,
- d) the temperature gradient above the mixing zone is linear.

In order to take into account only the heat gain that affects the temperature in the occupied zone, Loudermilk [2] developed a detailed table where he specified, as a function of the vertical projection of the outlet and heat source type, the percentage of the total heat that should be considered in the calculation. Loudermilk [2] used the vertical projection of the outlet (throw height) as an input value of the design tool. However, Liu and Linden [12] theoretically and experimentally showed that the vertical projection of the outlet and the two layer stratification height are not equal and the former cannot be used to predict the latter. Moreover, the vertical projection is a function of the balance between the initial momentum flux and the buoyancy flux from the outlet [13]. Momentum and buoyancy fluxes depend on the airflow rate per diffuser, the outlet geometry and the temperature difference between the jet and the room. The throw height depends also on the type of environment, i.e. homogeneous or stratified, where the turbulent jet is injected [13]. The stratification height depends on the balance between these diffuser characteristics and the buoyancy plumes generated by the heat loads in the space.

Loudermilk [2] fixed the dimensionless temperature at the floor level equal to 0.4, which in fact is closer to what is expected for displacement ventilation systems. This is an approximation that greatly limits the applicability of this design approach for UFAD systems. Bauman [14] presented experimental data for two common UFAD diffusers (swirl and VAV Directional) that demonstrated all dimensionless floor temperatures fell in the range of 0.6-0.8. Other research has shown that the temperature near the floor can vary over a wide (Liu and Linden [12, 15, 16], and Webster et al. [17]).

How heat gains are considered is another limitation of this and other methods. The instantaneous cooling load is the rate at which heat energy is convected to the zone air at a given point in time. Computation of cooling load is complicated by the radiant exchange between surfaces, furniture, partitions, and other mass in the zone [18]. Most heat sources transfer energy by both convection and radiation. Radiative heat transfer introduces a time dependency to the process that is not easily quantified [18]. Radiation is absorbed by thermal mass in the zone and then later transferred by convection into the space. This process creates a time lag and dampening effect. The convective portion, on the other hand, is immediately transformed into cooling load in the hour in which that heat gain occurs [18]. The thermal storage effect is critical in differentiating between instantaneous heat gain for a given space and its cooling load at that moment. Accounting for the time delay is a major challenge in cooling load calculations. The sum of all space instantaneous heat gains at any given time does not necessarily (or even frequently) equal the cooling load for the space at that same time [19]. Thus, using the heat gains and not taking into account the thermal storage effect is not a proper way to calculate design airflow rates.

The UFAD outlet manufacturer YORK [3] prepared a guidebook for the installation, control, and design of UFAD systems. In the load calculation section the following advice was given: (1) the convective heat source located above the head level does not influence the air temperature in the occupied zone, thus they should not be taken into account in the heat load calculation; (2) for the first time, the problems related to the thermal decay in the plenum and its effect on the design airflow rate were describe and quantified by using a simplified steady-state heat transfer model; (3) thermal storage in building mass and load diversity issues were qualitatively but not quantitatively described; and (4) the effect of the temperature stratification on heat load calculations was raised but it was suggested that it should not be taken into account because, at that time, it was not possible to accurately quantify it. Other manufacturer's design guidebooks are available, but they are generally also quite simplified.

DESCRIPTION OF THE UFAD COOLING LOAD DESIGN TOOL

This section describes the fundamentals on which the design tool is based.

UFAD and OH cooling load calculation

In a simulation study using EnergyPlus, Schiavon et al. showed that the cooling load profiles for UFAD and OH are different [20]. Figure 1 shows a comparison between the predicted cooling load profiles for overhead (mixing) and UFAD systems for five zones of a middle floor from a 3-story prototype office building for a Baltimore, MD, summer design day. The HVAC system is operating between 5am and 8pm. During the night the HVAC system is off. The internal and external heat gains are almost the same for the two systems but the cooling load removed by the HVAC systems is different because, for the OH system, part of the heat is stored in the slab during the day and released at night when the system is off. For the UFAD system the presence of the raised floor reduces the ability of the slab to store heat as described by Schiavon et al. [21]. A more detailed description of the difference between the two systems can be found in [20].

While EnergyPlus is certainly capable of being used to make load calculations, from a practical point of view it is important to develop a simplified load calculation procedure for designers. Designers commonly use software that do not have the capability to accurately model a UFAD system (e.g., ASHRAE RTS method, Trace Load 700, etc.) for the calculation of peak cooling loads. To simplify the process, an index for transforming the peak design day cooling load for a well mixed

system into one for UFAD was developed. This index is designated UFAD Cooling Load Ratio (UCLR) and is defined by the following equation:

$$UCLR = \frac{CL^{UFAD}}{CL^{OH}} \quad (1)$$

UCLR is the ratio of the cooling load calculated for UFAD (CL^{UFAD}) to the cooling load calculated for a well mixed system (CL^{OH}). When UCLR equals 1, the two cooling loads are the same. UCLR greater than 1 means that the cooling load for UFAD is higher than for OH. For example, UCLR = 1.12 means that the UFAD cooling load is 12% higher than the OH cooling load. UCLR less than 1 means that the UFAD cooling load is less than the OH cooling load.

A regression equation to predict UCLR was derived from a large number of EnergyPlus simulations. A detailed description of the method used to obtain the regression equation is reported by Schiavon et al. [20]. The influence on UCLR of the following variables was investigated using the large dataset generated by the EnergyPlus simulations: building floor level, building zone, supply air temperature, window-to-wall ratio, internal heat load, plenum configuration, climate, presence of floor carpet and structure type. The analysis showed that floor level and the zone type and orientation were the most influential variables.

The UFAD Cooling Load Ratio (UCLR) model is summarized by the following equation.

$$UCLR = 0.9528 + C_1X_1 + C_2X_2 \quad (2)$$

Where

X_1 = floor level: ground, middle and top.

$C_1=0$ if floor is the ground floor

$C_1=0.1572$ if floor is a middle floor

$C_1=0.2379$ if the floor is a top floor

X_2 = zone type: one interior zone and four perimeter zones, orientations east, south, west, and north.

$C_2=0$ if the zone is north oriented

$C_2=0.1739$ if the zone is east

$C_2=0.0999$ if the zone is south

$C_2=0.1349$ if the zone is west

$C_2=0.0802$ if the zone is an interior zone.

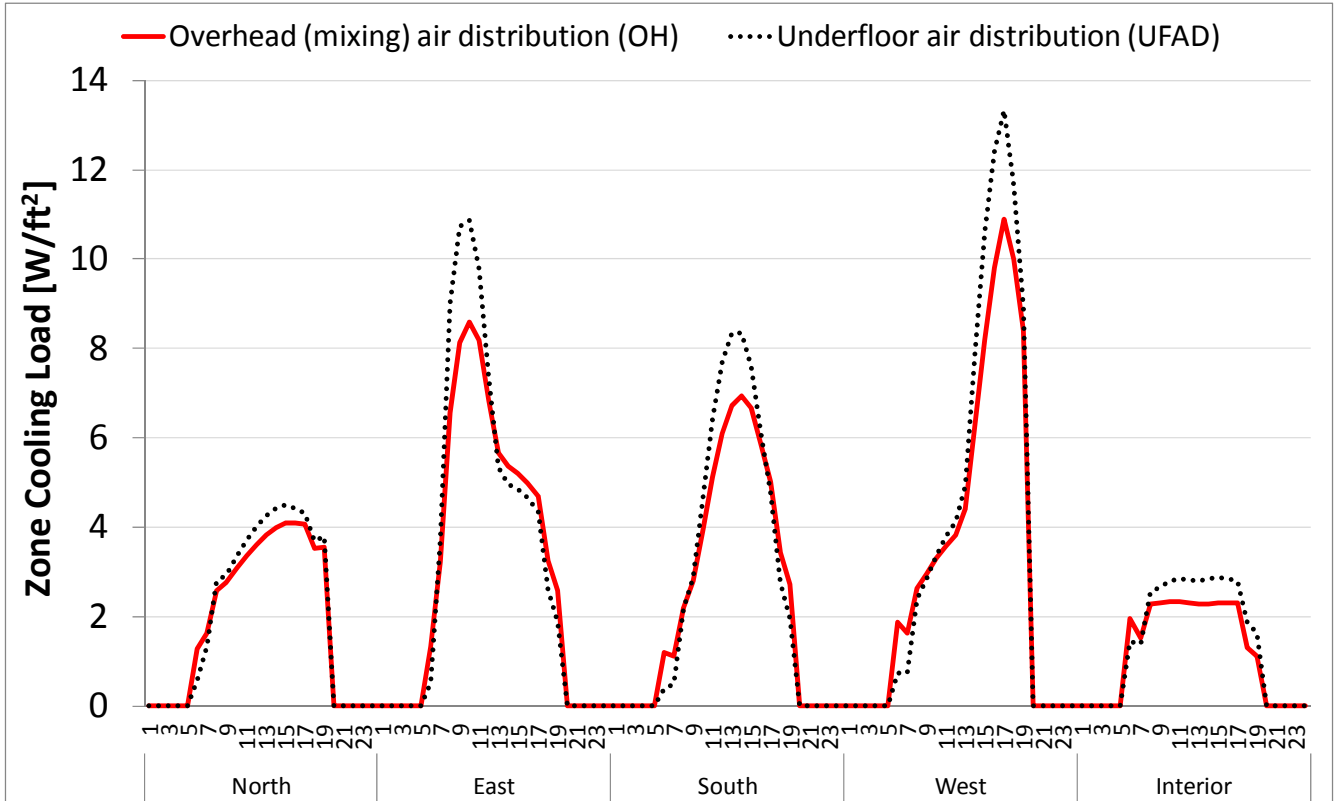


Figure 1. Cooling load profiles for overhead (mixing) and UFAD systems for four perimeter zones and an interior zone during the summer design day for Baltimore, MD. The HVAC system is operating between 5am to 8pm. During the night the HVAC system is off. The difference between the OH and UFAD is due to the ability of the OH slab to store heat during the day and release it at night when the system is off.

UFAD cooling load split (SPF, ZF, RPF)

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 affected by heat transfer from both the concrete slab (in a multi-story building) and the raised floor panels. Field measurements and computer fluid dynamic analyses [22 and 24] have shown that the supply air can warm up significantly, resulting in undesirable and uncontrolled air temperature discharged at the diffusers. This phenomenon is often referred to as thermal decay.

The amount of heat entering the underfloor plenum directly influences the design cooling airflow rate and the occupants' thermal comfort, and therefore must be accounted for by the design tool. Bauman et al. [22], by an analysis of heat transfer pathways in real underfloor plenums, has provided evidence that supports two widely observed thermal phenomena in UFAD systems: (1) part of the cooling load is removed in the underfloor supply plenum; and (2) temperature gain (thermal decay) in open underfloor supply plenums is often larger than expected. They estimated, based upon a simplified first-law model, that for typical multi-story building configurations (raised access floor on structural slab with or without suspended ceiling), 30-40% of the total room cooling load in the interior zone is transferred into the supply plenum and about 60-70% is removed in the zone. Bauman et al. [4], based on the above mentioned results, developed a practical design procedure for determining the amount of cooling air flow rate required in an interior zone of a UFAD system using a fixed split between the room and the supply plenum. In the current analysis, three indexes have been developed to split the total UFAD cooling load between the supply plenum, the zone (or room) and the return plenum.

The Supply Plenum Fraction (SPF), defined by Equation 3, is the ratio of the cooling load removed in the supply plenum to the total UFAD cooling load. SPF may vary between 0 and 1. If, for example, $SPF = 0.35$ this means that 35% of the UFAD cooling load is removed in the plenum.

$$SPF = \frac{CL_{SP}^{UFAD}}{CL^{UFAD}} \tag{3}$$

The Zone Fraction (ZF), defined by Equation 4, is the ratio of the cooling load removed in the zone to the total UFAD cooling load. ZF may vary between 0 and 1. If, for example, ZF = 0.60 this means that 60% of the UFAD cooling load is removed in the zone.

$$ZF = \frac{CL_Z^{UFAD}}{CL^{UFAD}} \quad (4)$$

The Return Plenum Fraction (RPF), defined by Equation 5, is the ratio of the cooling load removed in the return plenum to the total UFAD cooling load. RPF may vary between 0 and 1. For example, if RPF = 0.05 this means that 5% of the UFAD cooling load is removed in return plenum.

$$RPF = \frac{CL_{RP}^{UFAD}}{CL^{UFAD}} \quad (5)$$

From Equations 3, 4 and 5 it can be deduced that:

$$CL_{SP}^{UFAD} + CL_Z^{UFAD} + CL_{RP}^{UFAD} = CL^{UFAD} \quad (6)$$

and

$$SPF + ZF + RPF = 1$$

Similar to UCLR, three regression equations to predict SPF, ZF, RPF were derived from a large number of EnergyPlus simulations. A detailed description of the methods used is reported by Schiavon et al. (2010b). As for UCLR, the same range of variables was investigated for their influence on SPF, ZF, and RPF. The analysis showed that the most influencing variables were similar to UCLR: floor level and zone type (interior and perimeter, for all orientations). In the following, the regression equations for each split are summarized. The Supply Plenum Fraction model is defined by Equation 7.

$$SPF = (0.6179 + C_1X_1 + C_2X_2 + C_3X_1X_2)^2 \quad (7)$$

Where

X_1 = zone type: interior or perimeter

$C_1=0$ if the zone is an interior zone

$C_1=-0.2095$ if the zone is a perimeter

X_2 = floor level: ground, middle, top

$C_2=0$ if floor level is ground

$C_2=0.1242$ if floor level is middle

$C_2=-0.0896$ if floor level is top

$C_3=0$ if the zone is interior zone and the floor is the ground floor

$C_3= 0.0396$ if the zone is a perimeter zone and the floor a middle floor

$C_3= 0.1642$ if the zone is a perimeter zone and the floor level is a top floor.

The Zone Fraction model is summarized in Equation 8.

$$ZF = 1 - SPF - RPF \quad (8)$$

The Return Plenum Fraction model is summarized in Equation 9.

$$RPF = C_1X_1 \quad (9)$$

Where

X_1 = floor level: ground, middle, top.

$C_1= 0.01$ if floor level is ground

$C_1= 0.01$ if floor level is middle

$C_1= 0.30$ if the floor level top

Presented below is an example calculation of UCLR, SPF, ZF, and RPF, which depend on zone type, floor level and orientation of the zone. UCLR, SPF, ZF, RPF have been calculated for a perimeter zone with a south orientation located in a middle floor. The calculated indices are:

$$UCLR = 0.9528 + 0.1572 + 0.0999 = 1.21$$

$$SPF = (0.6179 - 0.2095 + 0.1242 + 0.0396)^2 = 0.337$$

$$RPF = 0.01$$

$$ZF = 1 - 0.337 - 0.01 = 0.653$$

Figure 2 illustrates how the modelling process works to transform the cooling load calculated for a well mixed zone into a UFAD cooling load and to split the UFAD cooling load into the three components.

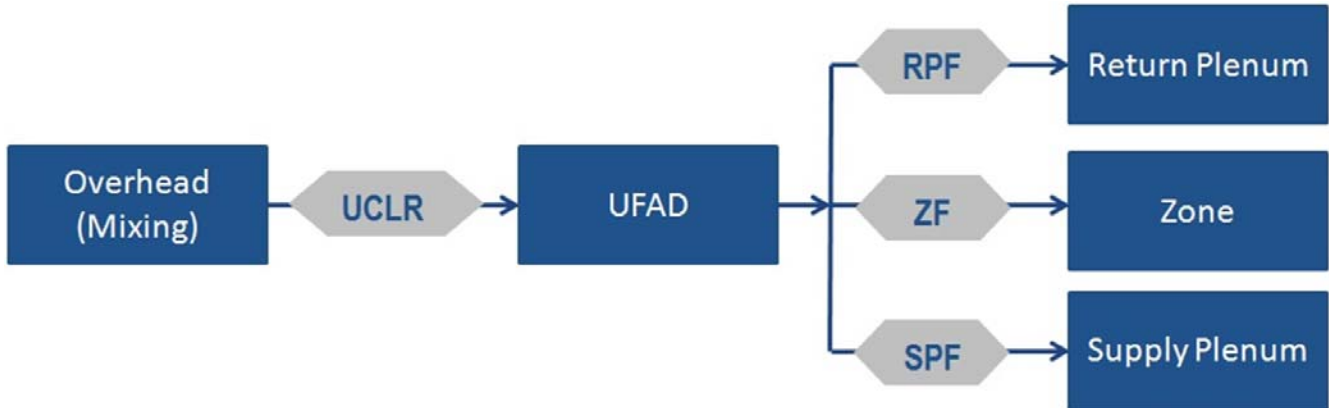


Figure 2 Schematic flow diagram of design tool showing transformation from cooling load calculated for an overhead mixing system into a UFAD cooling load, and then divided between the supply plenum, zone (room), and return plenum.

Vertical stratification and average occupied zone temperature

Properly controlled UFAD systems under cooling operation 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 multi-story building. Stratification also affects thermal comfort conditions. Under these conditions, the temperature at the ceiling can no longer be assumed to be equal to the room setpoint temperature. Figure 3 shows a schematic diagram of an example room air temperature profile for purposes of identifying key features in a stratified profile. As shown, when stratified conditions of various magnitudes 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, typically assumed for uniform well-mixed systems, is no longer valid.

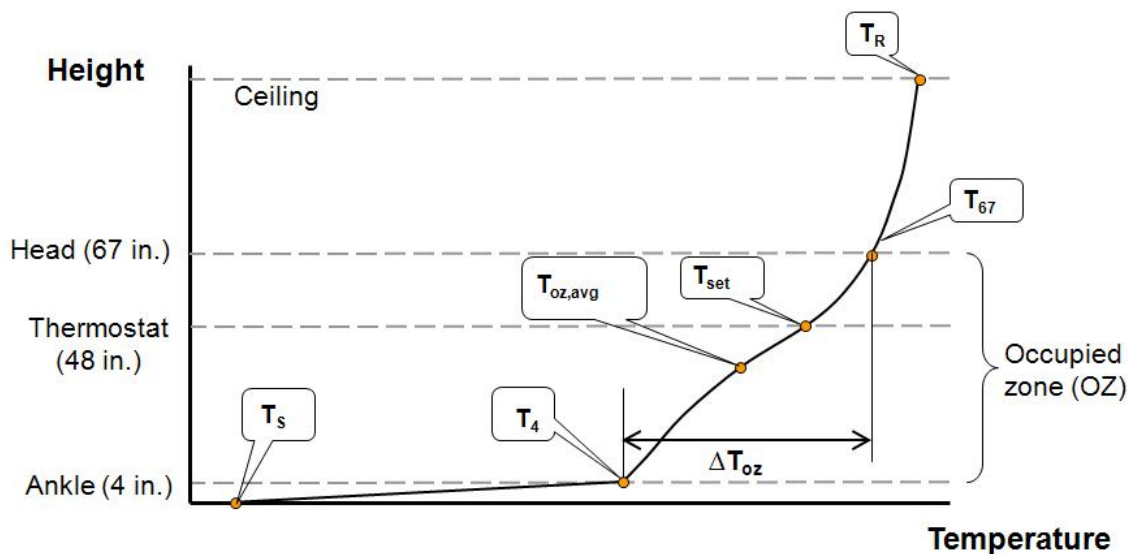


Figure 3. Example room air temperature profile in stratified UFAD room. (The meaning of the symbols used in this figure are reported in the nomenclature section.)

For purposes of comparison between cooling airflow quantities used by UFAD vs. well-mixed overhead OH systems, we have defined an equivalent acceptable comfort condition for standing occupants in a stratified room named the average occupied zone temperature ($T_{oz,avg}$); calculated as the weighted average of the measured temperature profile at ankle level (4 in. [0.1 m]), thermostat level (48 in. [1.2m]), and standing head level (67 in. [1.7 m]). $T_{oz,avg}$ is defined in Equation 10.

$$T_{oz,avg} = \left(\frac{1}{67-4}\right) \left[\left(\frac{67-48}{2}\right) (T_{67} + T_{set}) + \left(\frac{48-4}{2}\right) (T_4 + T_{set}) \right] \quad (10)$$

We assume that the thermal sensation perceived by an occupant exposed to a stratified environment is close to that of an occupant exposed to a uniform air temperature equal to the average occupied zone temperature ($T_{oz,avg}$) calculated according to Equation 10. Local and whole-body discomfort sensations is slightly affected by thermal gradient, but is strongly affected by average operative temperature [23]. Using $T_{oz,avg}$ is a better approximation than using the setpoint temperature at the thermostat height 48 in. (1.2 m).

Dimensionless temperature near the floor

The dimensionless temperature, Phi, at a height in the room is generally defined by the following equation:

$$\Phi = \frac{T - T_s}{T_R - T_s} \quad (11)$$

where T , T_s , and T_R , are, respectively, the point, supply and return air temperatures. The dimensionless temperature ratio at the ankle level, Φ_4 , and at the head level for a standing person, Φ_{67} , are defined in Equations 12 and 13.

$$\Phi_4 = \frac{T_4 - T_s}{T_R - T_s} \quad (12)$$

$$\Phi_{67} = \frac{T_{67} - T_s}{T_R - T_s} \quad (13)$$

For displacement ventilation systems according to Brohus and Ryberg [25], Φ_4 varies between 0.3 and 0.65. According to Chen and Glicksman [10] Φ_4 varies between 0.2 and 0.7. Mundt [9] developed a model for the prediction of Φ_4 for displacement ventilation systems that is a function of the airflow rate and it is based on a heat transfer model between the ceiling and the floor. Mundt's equation is used in the cooling airflow design model developed by Chen and Glicksman [10]. Lin and Linden [11] and Liu and Linden [12, 15] theoretically developed and experimentally tested (in a small-scale salt-tank model) a prediction of Φ for underfloor air distribution system as a function of the a non-dimensional parameter, Gamma (Γ). Their model was also tested in full-scale experiments by Webster et al. [17]; the model relationships developed are reported in the section "Gamma-Phi equations" below. The model has been implemented in EnergyPlus [26] and [27] by using a slightly different definition of Gamma.

Plenum configurations and plenum simplifications

The supply plenum is a key component of a UFAD system, it is responsible of the generally higher peak cooling load of UFAD compared to a traditional overhead system and, as explained above, it offsets part of the room cooling load. Due to thermal decay, the air leaving the diffusers is warmer than the air supplied to the plenum. The supply air may take many different paths in the plenum. To determine plenum temperature distributions, CFD simulations are required.

In the design tool four plenum configurations were modelled: series, reverse series, independent and common. The most commonly used plenum configuration is the "Series" where the supply air from the interior zone flows to the perimeter zone, as schematically represented in Figure 4. This strategy has the main disadvantage of overcooling the interior zone in order to keep the perimeter zone comfortable. To overcome this problem the system may be designed first to supply air to the perimeter zone, where the coldest air is needed, and then to the interior zone. We named this plenum configuration "Reverse series"; its schematic is shown in Figure 5. Supplying air to the perimeter zone in a Reverse series manner can be accomplished using, for example, air highways, and/or other ducting. In the two plenum configurations described above the plenum is open. It is also possible to create a partitioned plenum where the air is supplied independently to each partitioned plenum zone. In some designs, air is supplied to each plenum zone at different temperatures. We named this configuration "Independent". A schematic of the "Independent" plenum configuration is shown in Figure 6. In the fourth option, named "Common", it is assumed that the air is distributed into one open plenum in such a way (combination of ductwork or higher inlet velocities) that the average temperature in the interior zone plenum is equal to that of the perimeter zone plenum. This

characteristic temperature distribution has been observed in some operational UFAD buildings. A scheme of the "Common" plenum configuration is shown in Figure 7.

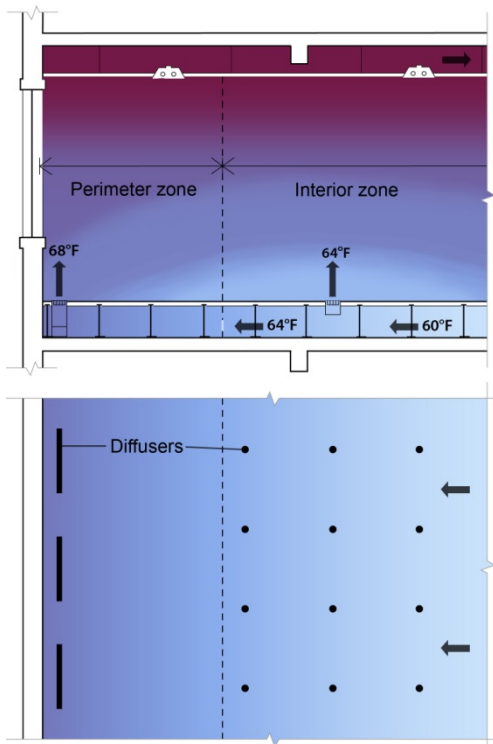


Figure 4. Series plenum configuration.

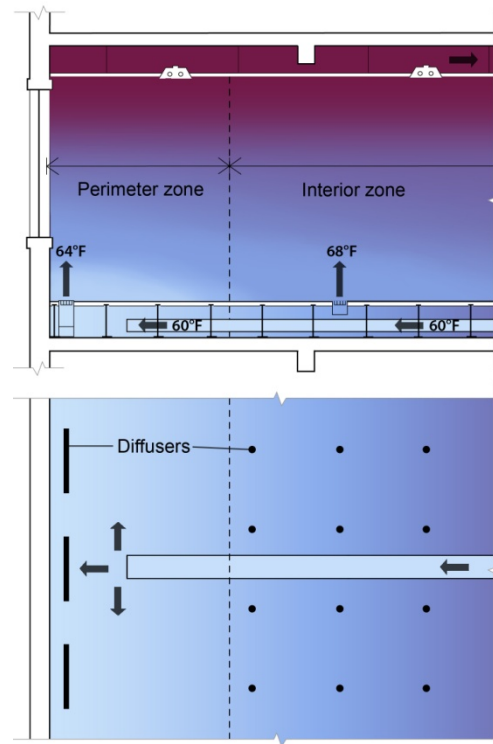


Figure 5. Reverse series plenum configuration.

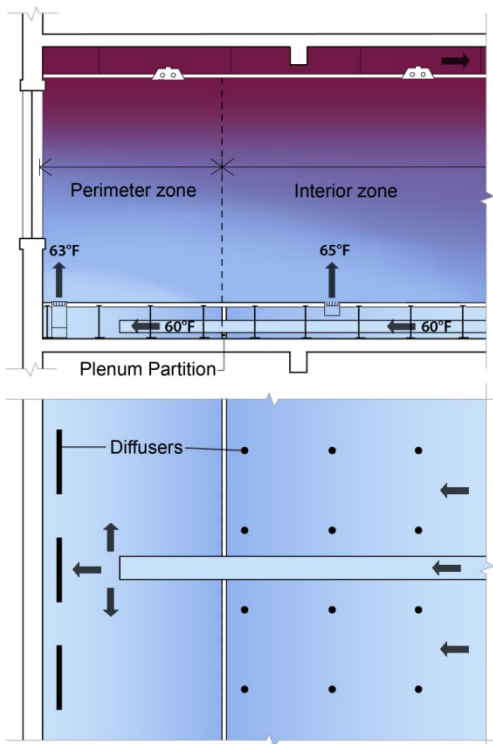


Figure 6. Independent plenums configuration. The interior and perimeter zones are designed as independent zones.

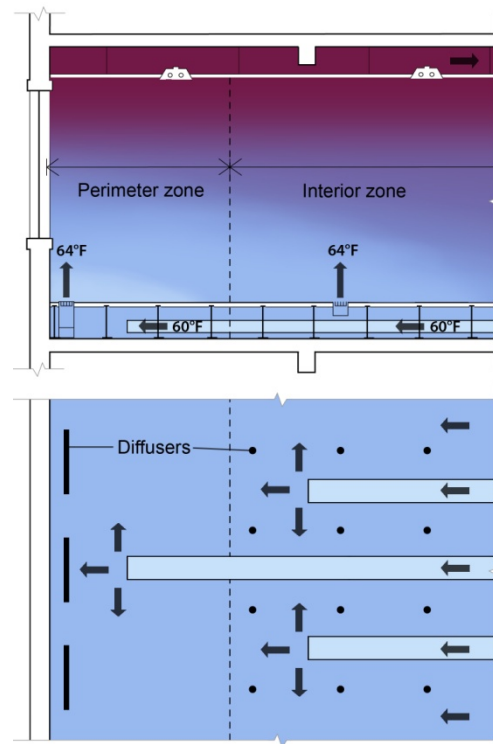


Figure 7. Common plenum configuration.

Even though it is well known that temperature variations will occur across the floor plate in the plenum, the design tool assumes that the air in each plenum zone is well mixed, i.e., it has a uniform air temperature. Previous research has shown that this simplification provides acceptable accuracy for energy and load calculation purposes. The assumptions and equations describing the plenum behavior for each type are shown in Table 1.

Table 1. Assumption and equations related to the modeled plenum configurations.

	Assumption and equations*
Series	T_{plenum}^I is a user input $T_{\text{plenum}}^P = T_s^I$ $Q_{\text{room}}^T = Q_{\text{room}}^I + Q_{\text{room}}^P$ $T_s^I = T_{\text{plenum}}^I + \frac{(3.412 \text{ SPF}^I W^I)}{(1.08 Q_{\text{room}}^T)}$ $T_s^P = T_s^I + \frac{(3.412 \text{ SPF}^P W^P)}{(1.08 Q_{\text{room}}^P)}$
Reverse series	T_{plenum}^P is a user input $T_{\text{plenum}}^I = T_s^P$ $Q_{\text{room}}^T = Q_{\text{room}}^I + Q_{\text{room}}^P$ $T_s^P = T_{\text{plenum}}^P + \frac{(3.412 \text{ SPF}^P W^P)}{(1.08 Q_{\text{room}}^T)}$ $T_s^I = T_s^P + \frac{(3.412 \text{ SPF}^I W^I)}{(1.08 Q_{\text{room}}^I)}$
	T_{plenum}^j (for j=interior and perimeter) are user inputs $T_s^j = T_{\text{plenum}}^j + \frac{(3.412 \text{ SPF}^j W^j)}{(1.08 Q_{\text{room}}^j)}$ $Q_{\text{room}}^j = Q^j + A_f q_{\text{leak},II}^j$
Common	$T_s^I = T_{\text{plenum}}^I + \frac{(3.412(\text{SPF}^I W^I + \text{SPF}^P W^P))}{(1.08 Q_{\text{room}}^T)}$ $T_s^P = T_s^I$

* the meaning of the symbols used in this table are reported in the nomenclature section.

Diffuser types

The user may chose between three types of diffusers:

1. Swirl diffuser for interior zones
2. Variable Air Volume directional diffuser for interior and perimeter zones
3. Linear bar grill diffuser for perimeter zones

A detailed description of the different diffusers can be found in Webster et al. [17].

Not all the diffuser types on the market can currently be modeled in the tool because the diffuser fluid-dynamic characteristics affect the temperature stratification in the room. Thus, before a particular diffuser can be added to the tool, detailed field or laboratory measured data is needed to obtain the Gamma-Phi relationships (see next paragraph) required to predict stratification in the room. If a manufacturer would like to add any of their diffusers, it is easy to implement once the Gamma-Phi relationship is known. The results of the design tool are strictly valid only for the type of diffusers tested. The choice of the diffuser type implies determination of: the diffuser effective area (A_d), the angle factor specific to the diffuser

type ($\cos \theta$), and the non-dimensional air temperature (Φ_k^j) at the k -in. height for the j zone where $j=I, P$ (Interior, Perimeter); $k=4, 67$ in., and θ is the angle from vertical to the discharge angle of the diffuser.

A_d and $\cos \theta$ for the diffuser types used in the design tool are listed in Table 2. The equations for the determination of Φ_k^j are reported in the paragraph “Gamma-Phi equations”.

Table 2. Geometric diffuser characteristics needed for the Gamma-Phi calculations.

Diffuser Type	Zone Type	Diffuser Effective Area, A_d [in ² (m ²)]	Angle Specific to Diffuser Type, θ [°]	Angle Factor Specific to Diffuser Type, $\cos \theta$
Swirl	Interior	17.05 (0.011)	28	0.883
VAV Directional	Interior and perimeter	54.25 (0.035)	45	0.707
Linear Bar Grill	Perimeter	38.75 (0.025)	15	0.966

The diffusers that are currently available in the design tool are briefly described below.

Swirl

Swirl floor diffusers are a commonly installed type of diffuser in the interior zone of UFAD systems. Swirl diffusers are generally installed as passive diffusers, requiring a pressurized underfloor plenum. The Gamma-Phi equations for the swirl diffuser are valid for swirl diffusers with an effective area of 17.05 in² (0.011 m²) and a discharge angle of 28°. These values were derived from bench testing and correspond closely to results from catalogue data. The design airflow rate for this diffuser is 75 cfm.

VAV directional for interior and perimeter zone

These diffusers operate with constant underfloor plenum pressure (i.e., 0.05 iwc (12.5 Pa) during laboratory tests) and constant outlet velocities leaving the diffusers. In order to vary the amount of airflow entering the room, there are two options: 1) varying the effective area of the diffuser by moving a damper plate; 2) varying the time ratio between when the diffuser is fully open versus when it is fully closed. The Gamma-Phi equations were obtained for the first option. Recent laboratory tests have confirmed that the same Gamma-Phi correlations are also valid for the second option. For the diffusers tested, when fully open the design airflow rate is roughly 150 cfm (255 m³/h). If the area of a fully open diffuser is 54.25 in² (0.035 m²) the outlet velocity at the grille is about 400 fpm (2 m/s). The discharge angle of the tested diffuser is estimated to be 45°, corresponding to the “spread” grille arrangement.

Linear bar grilles

These diffusers are standard products that are routinely used in OH systems and are commonly employed in UFAD systems for heating and cooling in the perimeter zone. The Gamma-Phi equations for the linear bar grilles diffuser are for diffusers with an effective area of 0.083 ft²/lineal foot (0.025 m² per lineal meter) a discharge angle of 15°. They come in various lengths and widths and usually have a lip on the bars that directs the flow away from the window.

Gamma-Phi equations

Lin and Linden [11] showed that in a UFAD system, the buoyancy flux of the heat source B and the momentum flux of the cooling jets M are the primary parameters controlling stratification. Therefore, according to Liu [16] a non-dimensional parameter Γ (Gamma) can be defined for single-diffuser, single-source cases from B , M and the effective area of a diffuser A_d as:

$$\Gamma = \frac{(M)^{\frac{3}{4}}}{(A_d)^{\frac{1}{2}} \cdot (B)^{\frac{1}{2}}} \quad (14)$$

Physically, Γ represents the relative strengths between buoyancy and momentum forces; B is the source of buoyancy generated by convective thermal plumes while M represents the effects of turbulent airflow from the diffusers that cause mixing. For the same geometry of the diffusers, large Γ means that the mixing dominates, and for small Γ buoyancy dominates and we expect more stratification in the space.

Gamma-Phi for the interior zone

For cases with multi-diffusers and multi-heat sources and for real full-scale interior zones, Γ is described by the following equation. How this representation of Γ is determined from the definition above can be found in Liu [16].

$$\Gamma = \frac{(Q \cdot \cos \varphi)^{\frac{3}{2}}}{m \cdot \left(\frac{n}{m} \cdot A_d\right)^{\frac{5}{4}} \cdot (0.0281 \cdot W)^{\frac{1}{2}}} \quad (15)$$

Where:

Q = room airflow [m³/s]

A_d = Diffuser effective area; Cos θ = discharge angle for diffuser flow [m²]

n = number of diffusers; m = number of plumes (i.e., occupants)

W = zone cooling load (supply and return plenum cooling loads are not included plenum) [kW].

Gamma – Phi for the perimeter zone

For perimeter zones, Gamma is developed based on the theory of line plumes generated by heat gains from exterior windows and walls. In this case Gamma reduces to:

$$\Gamma = \frac{(Q \cdot \cos \varphi)}{(n \cdot A_d) \cdot (0.0281 \cdot W_L)^{\frac{1}{3}}} \quad (16)$$

Where:

Q = total perimeter zone airflow [m³/s]

A_d = Diffuser effective area [m²]; Cos θ = cosine of discharge angle for diffuser flow

n = number of diffusers;

W_L = zone extraction rate per unit length of zone [kW/m]. W_L is the zone cooling load (supply and return plenum cooling loads are not included) divided by the length of the external wall of the perimeter zone considered. For example, if the zone cooling load is 2 kW and the length of the perimeter zone is 20 m, then W_L = 2/20 = 0.1 kW/m.

In Figure 8 and Figure 9 are reported measured data (from laboratory full-scale experiments) of Gamma-Phi for three diffuser types (swirl diffuser for the interior zone, VAV directional diffuser for the interior and for the perimeter zones, and linear bar grille diffuser for the perimeter zone) used in the design tool at the two heights (4 in. and 67 in.), respectively. The description of the measurements is reported by Webster et al. [17].

Note that for VAV directional diffusers Gamma is relatively constant. This is due to the fact that these diffusers are designed to deliver nearly constant discharge velocity as the flow varies by adjusting the discharge area. For the test results we assumed a constant average discharge velocity of 2 m/s and derived the effective area from this assumption and measured airflow; using equations (15) and (16) this results in nearly constant Gamma. The results demonstrate the intent of the design which is to keep stratification relatively constant (although limited) under varying load conditions.

Figure 9 shows that the non-dimensional temperature at head height for a standing occupant, Φ_{67} , is very close to the return temperature at ceiling height. Varying Gamma does not significantly change Φ_{67} ; for this reason in Table 3, Φ_{67} is kept constant and equal to the average value calculated for the tested diffuser type. Moreover, this also demonstrates that there is often little stratification above head height. For linear bar grilles the average Φ_{67} includes some values with large discharge angles so the results will be conservative with respect to predicted stratification; i.e., slightly more stratification might be indicated in the design tool if commonly employed diffusers characteristics are used. When more data is available, we expect that this Phi value will be Gamma dependent.

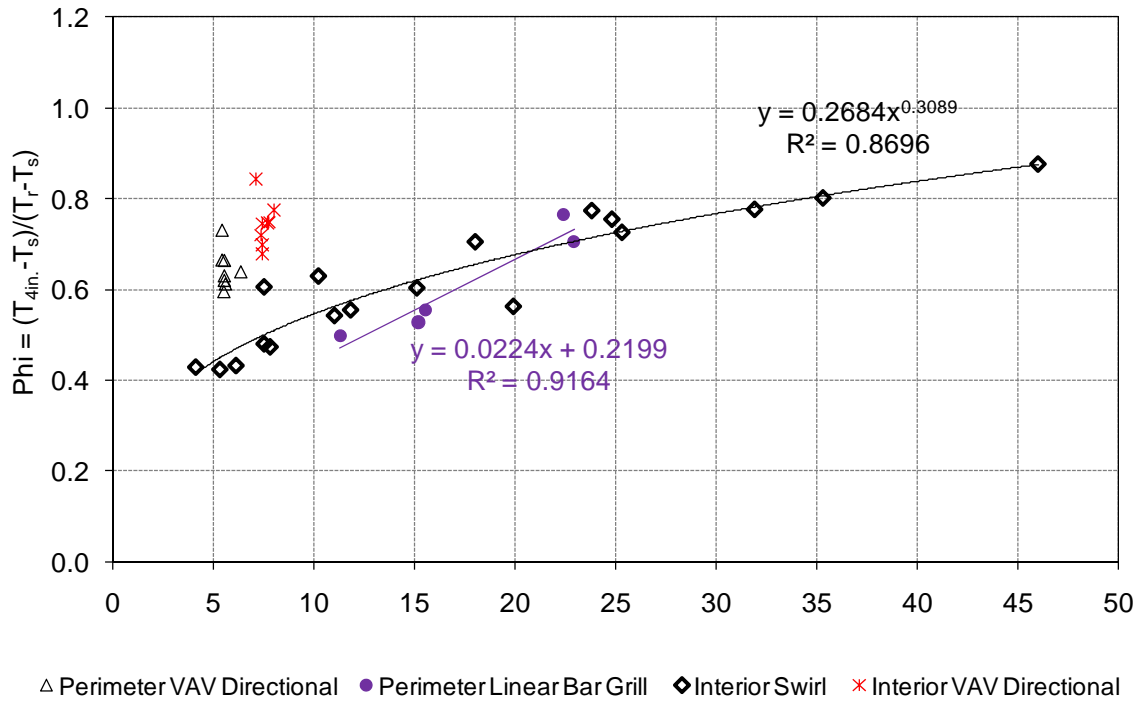


Figure 8. Measured data of Gamma and Phi at 4 in. for the three types of diffusers used in the design tool.

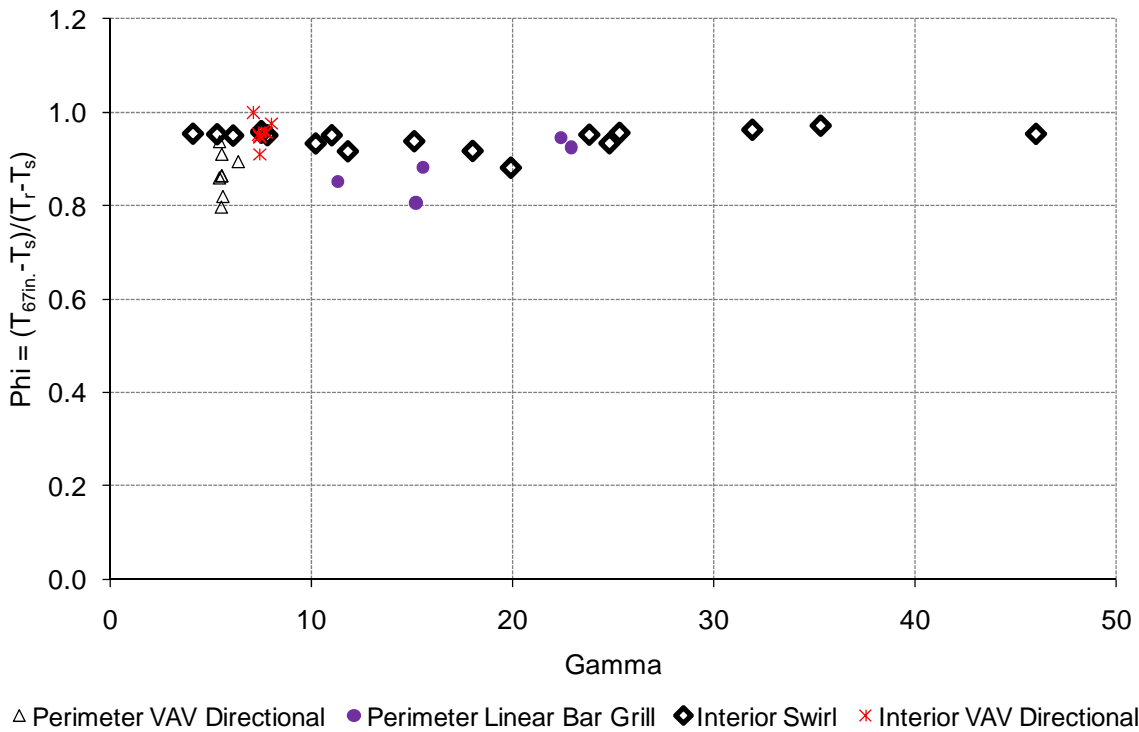


Figure 9. Measured data of Gamma and Phi at 67 in. for the three types of diffusers used in the design tool.

The Gamma-Phi relationships used in the design tool are summarized in Table 3.

Table 3. Gamma-Phi relationships at 4 and 67 in for the diffuser types used in the design tool

Zone	Diffuser Type	Γ - Φ_{4in}	Γ - Φ_{67in}
Interior	Swirl	$\Gamma < 4.3 \Phi_{4in} = 0.4212$ $4.3 < \Gamma < 46 \Phi_{4in} = 0.2684 \Gamma^{0.3089}$ $\Gamma > 46 \Phi_{4in} = 0.875$	$\Phi_{67in}=0.944$
Interior	VAV Directional	$\Phi_{4in} = 0.745$	$\Phi_{67in}=0.956$
Perimeter	Linear Bar Grill	$\Gamma < 11 \Phi_{4in} = 0.47$ $11 < \Gamma < 29.56 \Phi_{4in} = 0.2199+0.0224 \Gamma$ $\Gamma > 29.56 \Phi_{4in} = 0.882$	$\Phi_{67in}=0.882$
Perimeter	VAV Directional	$\Phi_{4in} = 0.631$	$\Phi_{67in}=0.874$

MODEL EQUATIONS

In this section are reported the equations used in the software for the three plenum configurations: series, reverse series, parallel/independent plenums and common.

Series

When the plenum is in series the temperature of the air entering the supply plenum in the interior zone, T_{plenum}^I , is known. In order to get stable solutions the solver is run several times. For each run, the solver first varies Q^P in order to get $T_{oz,avg}^P = T_{oz,avg,d}^P$ and then with this first estimate for Q^P , it varies Q^I in order to have $T_{oz,avg}^I = T_{oz,avg,d}^I \cdot Q^P$. Q^P is calculated first because it affects the amount of airflow in the plenum of the interior zone ($Q_{room}^P + Q_{room}^I$), thus the thermal decay in the interior zone depends also on Q^P . The equations used for the calculation of cooling design airflow rate when a series plenum is used are reported below.

$j=l,P$	$Q_{room}^j = Q^j + A_f^j q_{leak,II}^j$	(17)
	$Q_{room}^T = Q_{room}^I + Q_{room}^P$	(18)
	$T_s^I = T_{plenum}^I + \frac{(3.412 \text{ SPF}^I W^I)}{(1.08 Q_{room}^T)}$	(19)
	$T_s^P = T_s^I + \frac{(3.412 \text{ SPF}^P W^P)}{(1.08 Q_{room}^P)}$	(20)
$j=l,P$	$T_R^j = T_s^j + \frac{(3.412 ZF^j W^j)}{(1.08 Q_{room}^j)}$	(21)
$j=l,P$	$T_{RP}^j = T_R^j + \frac{(3.412 RPF^j W^j)}{(1.08 Q_{room}^j)}$	(22)
For $j=l$	$\Gamma^I = \frac{(Q^I \cos^{i,I} \theta)^{1.5}}{m \left(\frac{n^I}{m} A_d^{i,I} \right)^{5/4} \left(\frac{0.0281 ZF^I W^I}{1000} \right)^{0.5}}$	(23)
For $j=P$	$\Gamma^P = \frac{(Q^P \cos^{i,P} \theta)}{(n^P A_d^{i,P}) \left(\frac{0.0281 ZF^P W^P}{1000 L^P} \right)}$	(24)
$j=l,P; k=4, 67 \text{ in.}$	Φ_k^{ij} see Table 3	(25)
$j=l,P; k=4, 67 \text{ in.}$	$T_k^j = \Phi_k^{ij} (T_R^j - T_s^j) + T_s^j$	(26)
$j=l,P$	if $T_{set}^j < T_4^j$ then $T_{set}^j = T_4^j$ else $T_{set}^j = \left(\frac{T_R^j - T_{67}^j}{H_{in.}^j - 67} \right) (48 - H_{in.}^j) + T_R^j$	(27)
$j=l,P$	$\Delta T_{oz}^j = T_{67}^j - T_4^j$	(28)
$j=l,P$	$T_{oz,avg}^j = \left(\frac{1}{67 - 4} \right) \left[\left(\frac{67 - 48}{2} \right) (T_{67}^j + T_{set}^j) + \left(\frac{48 - 4}{2} \right) (T_4^j + T_{set}^j) \right]$	(29)

with $j=l,P$ (Interior, Perimeter); i = diffuser types; $k=4, 67 \text{ in.}$

Reverse series

When the plenum is in reverse series the temperature of the air entering the supply plenum in the perimeter zone, T_{plenum}^I , is known. In order to get stable solutions the solver is run several times. For each run the solver first varies Q^I in order to get $T_{oz,avg}^I = T_{oz,avg,d}^I$ and then, with this first estimate for Q^I , it varies Q^P in order to have $T_{oz,avg}^P = T_{oz,avg,d}^P$. Q^I is calculated first because it affects the amount of airflow in the plenum of the perimeter zone ($Q_{room}^P + Q_{room}^I$), thus the thermal decay in the perimeter zone depends also on Q^I . The equations for the reverse series plenum option are reported below.

$j=l,P$	$Q_{room}^j = Q^j + A_f^j q_{leak,II}^j$	(30)
	$Q_{room}^T = Q_{room}^I + Q_{room}^P$	(31)
	$T_s^P = T_{plenum}^P + \frac{(3.412 SPF^P W^P)}{(1.08 Q_{room}^T)}$	(32)
	$T_s^I = T_s^P + \frac{(3.412 SPF^I W^I)}{(1.08 Q_{room}^I)}$	(33)
$j=l,P$	$T_R^j = T_s^j + \frac{(3.412 ZF^j W^j)}{(1.08 Q_{room}^j)}$	(34)
$j=l,P$	$T_{RP}^j = T_R^j + \frac{(3.412 RPF^j W^j)}{(1.08 Q_{room}^j)}$	(35)
For $j=l$	$\Gamma^I = \frac{(Q^I \cos^{i,l} \theta)^{1.5}}{m \left(\frac{n^I}{m} A_d^{i,l} \right)^{5/4} \left(\frac{0.0281 ZF^I W^I}{1000} \right)^{0.5}}$	(36)
For $j=P$	$\Gamma^P = \frac{(Q^P \cos^{i,P} \theta)}{(n^P A_d^{i,P}) \left(\frac{0.0281 ZF^P W^P}{1000 L^P} \right)}$	(37)
$j=l,P; k=4, 67 \text{ in.}$	$\Phi_k^{i,j}$ see Table 3	(38)
$j=l,P; k=4, 67 \text{ in.}$	$T_k^j = \Phi_k^{i,j} (T_R^j - T_s^j) + T_s^j$	(39)
$j=l,P$	if $T_{set}^j < T_4^j$ then $T_{set}^j = T_4^j$ else $T_{set}^j = \left(\frac{T_R^j - T_{67}^j}{H_{in.}^j - 67} \right) (48 - H_{in.}^j) + T_R^j$	(40)
$j=l,P$	$\Delta T_{oz}^j = T_{67}^j - T_4^j$	(41)
$j=l,P$	$T_{oz,avg}^j = \left(\frac{1}{67-4} \right) \left[\left(\frac{67-48}{2} \right) (T_{67}^j + T_{set}^j) + \left(\frac{48-4}{2} \right) (T_4^j + T_{set}^j) \right]$	(42)

with $j=l,P$ (Interior, Perimeter); i = diffuser types; $k=4, 67 \text{ in.}$

Independent (Design Interior and Perimeter as Independent Zones)

When the two plenums are independent and the user can input different supply air temperatures T_{plenum}^j for $j=I,P$ (Interior, Perimeter) are known. The solver varies Q^I or Q^P in order to obtain $T_{oz,avg}^I = T_{oz,avg,d}^I$ and $T_{oz,avg}^P = T_{oz,avg,d}^P$. The two calculations (Q^I and Q^P) are independent and the solver is run only once for each configuration.

$j=I,P$	$Q_{room}^j = Q^j + A_d^j q_{leak,II}^j$	(43)
$j=I,P$	$T_R^j = T_s^j + \frac{(3.412 ZF^j W^j)}{(1.08 Q_{room}^j)}$	(44)
$j=I,P$	$T_s^j = T_{plenum}^j + \frac{(3.412 SPF^j W^j)}{(1.08 Q_{room}^j)}$	(45)
$j=I,P$	$T_{RP}^j = T_R^j + \frac{(3.412 RPF^j W^j)}{(1.08 Q_{room}^j)}$	(46)
	$\Gamma^I = \frac{(Q^I \cos^{i,I} \theta)^{1.5}}{m \left(\frac{n^I}{m} A_d^{i,I} \right)^{5/4} \left(\frac{0.0281 ZF^I W^I}{1000} \right)^{0.5}}$	(47)
	$\Gamma^P = \frac{(Q^P \cos^{i,P} \theta)}{(n^P A_d^{i,P}) \left(\frac{0.0281 ZF^P W^P}{1000 L^P} \right)}$	(48)
$k=4,67 in$	Φ_k^{ij} see Table 3	(49)
$k=4,67 in$	$T_k^j = \Phi_k^{ij} (T_R^j - T_s^j) + T_s^j$	(50)
$j=I,P$	if $T_{set}^j < T_4^j$ then $T_{set}^j = T_4^j$ else $T_{set}^j = \left(\frac{T_R^j - T_{67}^j}{H_{in.}^j - 67} \right) (48 - H_{in.}^j) + T_R^j$	(51)
$j=I,P$	$\Delta T_{oz}^j = T_{67}^j - T_4^j$	(52)
$j=I,P$	$T_{oz,avg}^j = \left(\frac{1}{67-4} \right) \left[\left(\frac{67-48}{2} \right) (T_{67}^j + T_{set}^j) + \left(\frac{48-4}{2} \right) (T_4^j + T_{set}^j) \right]$	(53)

with $j=I,P$ (Interior, Perimeter); $i=$ diffuser types; $k=4, 67 in$.

Common

When the plenum is treated as a common well mixed environment, T_{plenum}^j for $j= I,P$ (Interior, Perimeter) are known and $T_{plenum}^I = T_{plenum}^P$. In order to get stable solutions the solver is run several times. For each run the solver varies Q^I and Q^P in order to obtain $T_{oz,avg}^I = T_{oz,avg,d}^I$ and $T_{oz,avg}^P = T_{oz,avg,d}^P$. The equations for this option are reported below.

$j=I,P$	$Q_{room}^j = Q^j + A_f q_{leak,II}^j$	(54)
	$Q_{room}^T = Q_{room}^I + Q_{room}^P$	(55)
	$T_s^I = T_{plenum}^I + \frac{(3.412(SPF^I W^I + SPF^P W^P))}{(1.08 Q_{room}^T)}$	(56)
	$T_s^P = T_s^I$	(57)
$j=I,P$	$T_R^j = T_s^j + \frac{(3.412 ZF^j W^j)}{(1.08 Q_{room}^j)}$	(58)
$j=I,P$	$T_{RP}^j = T_R^j + \frac{(3.412 RPF^j W^j)}{(1.08 Q_{room}^j)}$	(59)
<i>For j=I</i>	$\Gamma^I = \frac{(Q^I \cos^{i,I} \theta)^{1.5}}{m \left(\frac{n^I}{m} A_d^{i,I} \right)^{5/4} \left(\frac{0.0281 ZF^I W^I}{1000} \right)^{0.5}}$	(60)
<i>For j=P</i>	$\Gamma^P = \frac{(Q^P \cos^{i,P} \theta)}{(n^P A_d^{i,P}) \left(\frac{0.0281 ZF^P W^P}{1000 L^P} \right)}$	(61)
$j=I,P; k=4, 67 \text{ in.}$	Φ_k^{ij} see Table 3	(62)
$j=I,P; k=4, 67 \text{ in.}$	$T_k^j = \Phi_k^{ij} (T_R^j - T_s^j) + T_s^j$	(63)
$j=I,P$	if $T_{set}^j < T_4^j$ then $T_{set}^j = T_4^j$ else $T_{set}^j = \left(\frac{T_R^j - T_{67}^j}{H_{in.}^j - 67} \right) (48 - H_{in.}^j) + T_R^j$	(64)
$j=I,P$	$\Delta T_{oz}^j = T_{67}^j - T_4^j$	(65)
$j=I,P$	$T_{oz,avg}^j = \left(\frac{1}{67-4} \right) \left[\left(\frac{67-48}{2} \right) (T_{67}^j + T_{set}^j) + \left(\frac{48-4}{2} \right) (T_4^j + T_{set}^j) \right]$	(66)

NOMENCLATURE

Symbols

A_f^j	Floor area of the j -zone, [ft ²], $j=I,P$
H_t^j	Room height [t=in or ft], $j=I,P$
L^P	Length of the perimeter zone [ft]
Q^j	Airflow (through diffusers) in the j -zone, [cfm], $j=I,P$
Q_{room}^j	Airflow (through diffusers plus category II leakage) in the j -zone, [cfm], $j=I,P$
T_R^j	Return air temperature of the j -zone, [°F], $j=I,P$
T_{RP}^j	Plenum return air temperature of the j -zone, [°F], $j=I,P$
T_S^j	Temperature of air supplied at diffuser of the j -zone, [°F], $j=I,P$
T_k^j	Air temperature at k -in. height, [°F], $k=4, 67$ in.
$T_{oz,avg,d}^j$	Design average temperature in the occupied zone of the j -zone, [°F], $j=I,P$
$T_{oz,avg}^j$	Average temperature in the occupied zone of the j -zone, [°F], $j=I,P$
T_{plenum}^j	Temperature of air entering supply plenum of the j -zone, [°F], $j=I,P$
T_{set}^j	Setpoint temperature at the thermostat level(48 in.) of the j -zone, [°F], $j=I,P$
W_{OH}^j	Design cooling load calculated for a mixing system for the j -zone, [Btu/(hr ft ²) or kBtu/hr], $j=I,P$
W^j	Design cooling load for UFAD of the j -zone, [W], $j=I,P$
n^j	Number of diffusers in the j -zone, $j=I,P$
$q_{leak,2}^j$	Estimated Category 2 leakage in the j -zone, [cfm/ft ²], $j=I,P$
m	Number of occupant in the interior zone
OH	Overhead (well-mixed) system
RPF_j	Return plenum fraction for the j -zone, [-], $j=I,P$
SPF_j	Supply plenum fraction for the j -zone, [-], $j=I,P$
$UCLR$	UFAD cooling load ratio
$UFAD$	Underfloor air distribution
ZF_j	Zone fraction for the j -zone, [-], $j=I,P$

Greek symbols

ΔT_{oz}^j	Temperature difference between the head and the ankle of a standing person in the occupied zone of the j -zone (from 4 in. to 67 in.), [°F], $j=I,P$
Φ_k^j	Non-dimensional temperature at the k -in. height, [°F], $k=4, 67$ in.
Γ^j	Non-dimensional parameter representing the ratio of buoyancy to inertia forces in the j -zone, $j=I,P$

Superscript and Subscript

j	Zone type $j=I,P$ (Interior, Perimeter)
k	Height, $k=4, 67$ in
i	Type of diffuser, $i=$ types of diffusers listed in the section "Diffuser types".

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