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Heat Transfer Pathways in Underfloor Air Distribution (UFAD) Systems

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Please note that the following two statements have been added as a result of the peer-review process for this paper: Results of heat gain shown in this theoretical steady-state model may be greater than those found in actual practice for UFAD systems. This may be for a number of reasons that are not now understood and should be the basis for further research and study.

ABSTRACT

This paper reports on a modeling study to investigate the primary pathways for heat to be removed from a room with underfloor air distribution (UFAD) under cooling operation. Compared to the standard assumption of a well-mixed room air condition, 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. A simplified first-law model has been used to estimate and compare the relative magnitudes of the heat being removed from a 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 is shown that up to 40% of the total room cooling load is transferred into the supply plenum and only about 60% is accounted for by the return air extraction rate. The implications for the design and operation of UFAD systems are discussed.

INTRODUCTION

For decades, engineers designing conventional overhead 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 in which design space heat gains (i.e., total room cooling loads) are considered 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 In a stratified underfloor air distribution (UFAD) environment, the assumption of perfect mixing is no longer valid, requiring a different way of thinking about energy flows and airflow quantities.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.

To date, most concepts of UFAD cooling airflow design sizing attempt 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 of UFAD systems in Loudermilk [1999] and Bauman [2003] and of displacement ventilation systems in Yuan et al. [1999]). While this method may have merit (mostly because it adheres to mixed system concepts), the current study will demonstrate that it may be oversimplified. In addition, ongoing research on room air stratification and energy performance of UFAD systems is investigating the rather complex combination of parameters that may influence stratification and cooling airflow quantities, including type and number of diffusers, diffuser throw, room cooling load, room air setpoint, and supply air temperature (CBE 2005).

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The modeling results described in this paper address the following key issues in the design and cooling operation of a UFAD system in a multi-story building.

1. Cool supply air flowing through the underfloor plenum is exposed to heat gain from both the concrete slab (conducted from the warm return air on the adjacent floor below the slab) and the raised floor panels (conducted from the warmer room above). While evidence of slab heat conduction into the underfloor plenum is common in field installations, quantitative measures of heat transfer through the raised floor have been discussed less frequently (Nagase et al. 1995; Webster et al. 2002a). 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*). These warmer supply air temperatures can make it more difficult to maintain comfort in the occupied space (without increasing airflow rates), particularly in perimeter zones where cooling loads reach their highest levels. How to predict plenum thermal performance is one of the key design issues practicing engineers face—evidence from completed projects indicates that excessive thermal decay can be a problem.

2. Properly controlled UFAD systems produce temperature stratification in the conditioned space, resulting in higher temperatures at the ceiling level than at the floor. The thermal-plume-driven airflow pattern and higher temperatures in the upper part of the room allow some portion of the convective heat gains to be more efficiently removed from the space. 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 by directly leading to higher rates of heat transfer into the underfloor plenum through two major pathways: (1) conduction from the warm return air through the slab into the supply plenum for the floor above and (2) radiation from the warm ceiling surface down to the floor and furniture of the space below. As will be shown, under normal conditions the radiant energy striking the floor makes up a large majority of the total heat transfer that is conducted through the raised floor panels into the underfloor plenum (in contrast, only a small amount of this total heat transfer is contributed by convection between the floor surface, usually carpet, and the room air near the floor). In addition, at higher ceiling temperatures, the radiant energy from the ceiling that impacts furniture, occupants, and other objects will largely reenter the occupied zone via convective heat transfer.

The simplified modeling study described in the next section was undertaken to improve the current understanding of heat removal from a stratified room with UFAD under cooling operation. While the steady-state model is approximate in nature and is not intended for use as a validated UFAD design tool, the goal of the study was to seek evidence that supports two surprising and widely observed thermal phenomena in UFAD systems: (1) the return air extraction rate, based on the temperature difference between the room return and diffuser supply air temperatures, is almost always noticeably less than the total room cooling load based on the sum of all space heat gains; and (2) temperature gain (thermal decay) in open underfloor supply plenums is often larger than expected. The modeling results are consistent with these observations and also provide preliminary guidance on approaches for reducing heat transfer into underfloor plenums.

DESCRIPTION OF SIMPLIFIED MODEL

A simplified first-law model was used to evaluate and compare the relative proportions of heat fluxes leaving a room with UFAD under cooling operation through two primary pathways: (1) heat extraction via warm return air exiting the room at ceiling level or through the return plenum, after losing heat to the underside of the slab, and (2) total heat transferred to the underfloor supply plenum (through the slab and through the raised floor panels). To accomplish this, the model is configured to calculate (or use assumed values for) inlet and outlet air temperatures for two control volumes: (1) the room, bounded by the top of the raised floor and the underside of the slab (with or without insulation), and (2) the underfloor plenum, bounded by the top surface of the slab and the underside of the floor panels. Note that the suspended ceiling (when present) is modeled as an internal element of the room control volume. For the room/return plenum, the model performs a steady-state heat balance on each of the three major architectural layers of the heat transfer system: (1) the top surface of raised floor (carpet), (2) the bottom surface of slab (or slab insulation, if present), and (3) the suspended ceiling (if present, assumed to be a uniform temperature [infinite conductivity]). In addition, heat balances are also performed on the top and bottom surfaces of the supply plenum, representing the bottom surface of the raised floor panels and the top surface of the slab, respectively. These latter two heat balances did not involve radiative heat transfer, since radiation was neglected within the plenum based on previous experimental and numerical results, indicating that these effects were minimal. Equations 1–5 list the heat balance formulas for each of these five surfaces. Figures 1 and 2 show schematic diagrams of the two configurations considered, with and without a hung ceiling, respectively. An iterative solution was obtained when the difference in heat fluxes reaching and leaving each of these surfaces was zero.

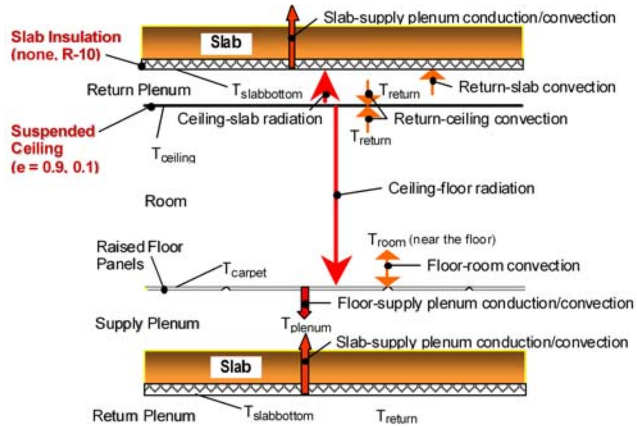

Figure 1 *Schematic of heat transfer pathways in a room with UFAD and a hung ceiling.*

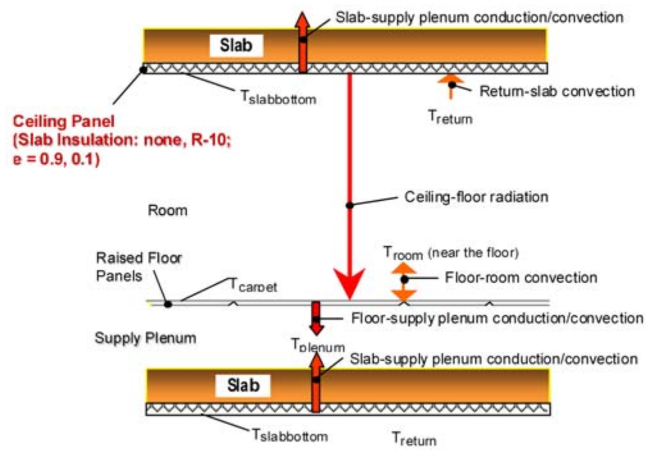

Figure 2 *Schematic of heat transfer pathways in a room with UFAD and no hung ceiling.*

Surface Heat Balance Equations

Top surface (carpet) of raised floor:
convection (either entering or leaving) + radiation (entering) = conduction (leaving),

$$h_{cn} \cdot (T_{nf} - T_{ft}) + \frac{F \cdot \sigma \cdot (T_{ca}^4 - T_{fta}^4)}{\frac{1}{\varepsilon_c} + \frac{1}{\varepsilon_f} - 1} = \frac{T_{ft} - T_{fb}}{\frac{t_f}{k_f}} \tag{1}$$

Bottom surface of raised floor:
conduction (entering) = convection (leaving),

$$\frac{T_{ft} - T_{fb}}{\frac{t_f}{k_f}} = h_{cf} \cdot (T_{fb} - T_p) \tag{2}$$

Top surface of slab with insulation (if no insulation, $R = 0$):
conduction (entering) = convection (leaving),

$$\frac{T_{sb} - T_{st}}{\frac{t_s}{k_s} + R} = h_{cf} \cdot (T_{st} - T_p) \tag{3}$$

Bottom surface of slab with insulation (if no insulation, $R = 0$):
convection (entering) + radiation (entering) = conduction (leaving),

$$h_{cnu} \cdot (T_r - T_{sb}) + \frac{\sigma \cdot (T_{ca}^4 - T_{sba}^4)}{\frac{1}{\varepsilon_c} + \frac{1}{\varepsilon_s} - 1} = \frac{T_{sb} - T_{st}}{\frac{t_s}{k_s} + R} \tag{4}$$

Suspended ceiling:
convection (entering) = radiation (leaving),

$$h_{cnu} \cdot (T_r - T_c) + h_{cnd} \cdot (T_r - T_c) = \frac{\sigma \cdot (T_{ca}^4 - T_{sba}^4)}{\frac{1}{\varepsilon_c} + \frac{1}{\varepsilon_s} - 1} + \frac{F \cdot \sigma \cdot (T_{ca}^4 - T_{fta}^4)}{\frac{1}{\varepsilon_c} + \frac{1}{\varepsilon_f} - 1} \tag{5}$$

The following assumptions were employed in the model. The average forced convection coefficient on both the top and bottom surfaces of the underfloor air supply plenum was assumed to be 5.0 W/m^2·K (0.88 Btu/h·ft^2·°F), based on previous full-scale experiments (Jin et al. 2006). Natural convection (still-air) convection coefficients were assumed on all other horizontal surfaces, including the bottom of the slab, top and bottom of the suspended ceiling, and top of the raised floor (4.04 W/m^2·K [0.71 Btu/h·ft^2·F] for upward heat flow and 0.92 W/m^2·K [0.16 Btu/h·ft^2·°F] for downward heat flow). The thermal conductivity for the 15 cm (6 in.) concrete slab was 0.93 W/m·K (6.45 Btu·in./h·ft^2·°F) and for the raised floor (including carpet) was 0.14 W/m·K (0.97 Btu·in./h·ft^2·°F). In this simplified model, heat exchange with the walls was neglected, as heat transfer only in the vertical direction was considered, thereby simulating an interior zone. The same principles would hold for an exterior zone, but solar gain and wall conduction would have to be considered. In addition, to account for the effect of furniture and other objects in the room that would block radiative heat transfer between the ceiling and floor, a view factor of 0.5 was applied to the net radiant exchange between these surfaces. The remaining 50% of the net radiative heat transfer leaving the ceiling was assumed to impact and be absorbed by these miscellaneous objects located primarily in the occupied zone. Since this radiant load reentered the occupied zone, we did not include it in our calculations, which focused on estimating the amount of energy leaving the room. Sensitivity of the model predictions to the above assumptions are discussed later in the paper.

Model Configurations

To investigate possible strategies for reducing the magnitude of energy transferred into the underfloor plenum, the model was configured to include different combinations of low-emissivity ceiling and slab insulation. These ceiling thermal properties were applied to both a suspended ceiling and an open, exposed slab (no hung ceiling), as described below.

1. ***Suspended ceiling***. In conventional suspended ceiling construction, the ceiling was configured in two separate layers: a suspended layer of low-e ceiling panels and a slab-insulation layer attached to the underside of the structural slab, representing the top of the return air plenum (Figure 1).
2. ***Open, exposed ceiling***. In an open, exposed slab configuration, there is no suspended ceiling (the ceiling plenum is eliminated). In this configuration, the slab insulation layer also had a low-emissivity coating on its lower, exposed surface and was installed directly on the underside of the structural slab or structural layer between floors of a multi-story building (Figure 2).

Table 1 lists the four different ceiling configurations that were simulated using the simplified model. The baseline and three cases represent different combinations of slab insulation and low-emissivity ceiling. The ceiling emissivity was applied to the top and bottom surfaces of the hung ceiling (if present) or to the bottom surface of the slab or slab insulation (if no hung ceiling).

As indicated in Figures 1 and 2, the calculations by the simplified model consider the following heat transfer processes:

- conduction through the slab and floor panels and into the supply plenum via convection;
- radiation from the ceiling to the raised floor (and from the top of the hung ceiling to the underside of the slab);
- convection between the return air and the suspended ceiling and/or the ceiling/slab (the return temperature, after losing heat to the slab, is assumed to be the same as the temperature within the well-mixed ceiling plenum [when present] as well as near the top of the room [with or without suspended ceiling]);
- convection between the room air near the floor and the raised floor panels (carpet surface).

Model Calculations

Table 2 shows calculated temperatures and heat fluxes related to the major heat transfer pathways in a multi-story building using UFAD with a hung ceiling (both underfloor and overhead plenums). Table 3 shows the same set of calculations for the room configuration with an exposed slab (no hung ceiling and no overhead plenum).

Initial air temperatures, representing baseline conditions, were taken to represent a typical vertical temperature profile for a well-stratified interior zone of a building, such as those described by Webster et al. (2002a, 2002b). As shown in the column labeled "Conditions" in both Tables 2 and 3, the temperature near the ceiling, representing the return air temperature (T_r), was assumed to be 25.6°C (78.0°F), and the temperature near the floor (T_{nf}) was assumed to be 22.2°C (72.0°F), producing a 3.4°C (6.0°F) temperature gradient from floor to ceiling. Since the model focused on calculating the heat being transferred out of the room, the air temperature in the middle of the room (e.g., the thermostat temperature) was not directly calculated. In addition, the model can be considered representative of standard floor-to-ceiling heights found in multi-story office buildings. Even if the ceiling height is slightly higher (for example, if the suspended ceiling is removed), the calculated return temperature will be the same for each case. This is because once the model has iteratively calculated what portion of the room load is leaving the room via return air extraction (this calculation is independent of room height in the model), the return temperature will be fixed based on the assumed constant values of supply temperature entering the room and room airflow rate. For rooms of different heights, the slope of the temperature gradient will be slightly altered.

The average supply air temperature leaving the underfloor plenum (T_p) was 18.3°C (65.0°F), and the room airflow rate was 3.05 L/s·m^2 (0.6 cfm/ft^2) and was held constant for all cases. This airflow rate is similar to expected rates for comparable overhead systems for interior zones, an observation that has been supported by current experience in the field. All other temperatures and heat fluxes were calculated by the model. To provide consistency in the comparison between different cases, it was required that the total room cooling load (return air extraction plus heat transfer into plenum) was held constant for the same room geometry. The tables also list total heat transfer into the supply plenum (W/m^2 [Btu/h·ft^2]), which is the sum of heat entering the supply plenum through the slab and through the raised floor panels.

Table 1. Properties of Ceiling Configurations Investigated with Simplified Model

Ceiling Condition	Baseline	Case 1	Case 2	Case 3
Emissivity (ε_c) = 0.9	X		X	
No slab insulation	X	X		
Emissivity (ε_c) = 0.1		X		X
Slab insulation = R-10			X	X

Table 2. Calculated Temperatures and Heat Fluxes in Room with UFAD and Hung Ceiling; Room Airflow Rate = 3.1 L/s·m² (0.6 cfm/ft²); All Values Calculated Unless Otherwise Indicated

Conditions	Baseline Ceiling ε_c = 0.9 No Slab Insulation	Case 1 Ceiling ε_c = 0.1 No Slab Insulation	Case 2 Ceiling ε_c = 0.9 Slab Insulation R-10	Case 3 Ceiling ε_c = 0.1 Slab Insulation R-10
T_{sb}, °C (F)	23.1 (73.6)	23.3 (73.9)	26.3 (79.3)	27.5 (81.6)
T_r, °C (°F)	25.6 (78.0)*	26.2 (79.2)	27.5 (81.6)	28.5 (83.4)
T_c, °C (°F)	23.8 (74.8)	25.6 (78.1)	26.0 (78.8)	28.0 (82.4)
Ceiling-floor radiation,† W/m² (Btu/h·ft²)	5.7 (1.80)	1.6 (0.51)	8.9 (2.83)	2.3 (0.73)
Slab-supply plenum conduction, W/m² (Btu/h·ft²)	13.1 (4.16)	13.5 (4.27)	3.7 (1.19)	4.3 (1.38)
T_{fi}, °C (F)	21.4 (70.5)	20.1 (68.1)	22.4 (72.2)	20.3 (68.5)
T_{nf},‡ °C (°F)	22.2 (72.0)	22.2 (72.0)	22.2 (72.0)	22.2 (72.0)
Floor-supply plenum conduction, W/m² (Btu/h·ft²)	6.4 (2.03)	3.6 (1.14)	8.4 (2.66)	4.1 (1.30)
Floor-room convection, W/m² (Btu/h·ft²)	0.8 (0.24)	2.0 (0.62)	–0.5 (–0.16)	1.8 (0.57)
T_p,** °C (°F)	18.3 (65.0)	18.4 (65.0)	18.3 (65.0)	18.3 (64.9)
Total heat transfer into supply plenum, W/m² (Btu/h·ft²)	19.5 (6.19)	17.1 (5.40)	12.2 (3.85)	8.4 (2.67)
% change from baseline	N/A	–12.7%	–37.8%	–56.9%
Return air extraction rate, W/m² (Btu/h·ft²)	26.4 (8.38)	28.9 (9.17)	33.8 (10.72)	37.6 (11.90)
% change from baseline	N/A	9.4%	27.9%	42.0%
Room airflow rate,†† L/s·m² (cfm/ft²)	3.1 (0.6)	3.1 (0.6)	3.1 (0.6)	3.1 (0.6)
T_{pi}, °C (°F)	13.0 (55.4)	13.7 (56.7)	15.0 (59.0)	16.0 (60.8)

* Assumed to be 25.6°C (78.0°F) for the baseline.
† Calculated based on a radiation view factor of 0.5.
‡ Assumed to be 22.2°C (72.0°F) for the baseline and held constant for all other cases.
** Assumed to be 18.3°C (65.0°F) for the baseline and held constant for all other cases.
†† Assumed to be 3.1 L/s·m² (0.6 cfm/ft²) for the baseline and held constant for all other cases.

The return air extraction rate is calculated according to the following equations:

$$Q = \rho \times Cp \times \dot{V} \times \Delta T \qquad \text{(SI units)} \qquad (6a)$$

$$Q = 60 \times \rho \times Cp \times \dot{V} \times \Delta T \qquad \text{(I-P units)} \qquad (6b)$$

where

Q = return air extraction rate, W/m² (Btu/h·ft²)
ρ = density of air, 1.2 kg/m³ (0.075 lb/ft³)
Cp = specific heat of air, 1.0 kJ/kg·K (0.239 Btu/lb·°F)
$\dot{V}$ = the airflow moving through the room, 3.05 L/s·m² (0.6 cfm/ft²)
ΔT = the temperature difference between the return air temperature (T_r) and the supply air temperature (T_p), °C (°F)

The room air temperature near the floor was held constant at 22.2°C (72.0°F) for all cases simulated, representing conditions that would be expected to occur when the room is under setpoint temperature control. To estimate changes in the plenum inlet temperature (temperature of air entering the plenum from the air handler) the following method was used. An energy balance on the supply plenum was performed using Equation 6 for the baseline case, with Q equal to the total heat transfer into plenum and ΔT equal to the difference between the average (well-mixed) plenum temperature (18.3°C [65.0°F]) and the plenum inlet temperature. This equation was then solved to calculate the required plenum inlet temperature to maintain a constant mixed air average temperature of 18.3°C (65.0°F).[1] For the baseline cases, for

Table 3. Calculated Temperatures and Heat Fluxes in Room with UFAD and Exposed Slab (No Hung Ceiling); Room Airflow Rate = 3.1 L/s·m^2 (0.6 cfm/ft^2); All Values Calculated Unless Otherwise Indicated

Conditions	Baseline Ceiling ε_c = 0.9 No Slab Insulation	Case 1 Ceiling ε_c = 0.1 No Slab Insulation	Case 2 Ceiling ε_c = 0.9 Slab Insulation R-10	Case 3 Ceiling ε_c = 0.1 Slab Insulation R-10
T_{sb}, °C (F)	22.1 (71.8)	22.6 (72.8)	24.5 (76.2)	26.3 (79.3)
T_r, °C (°F)	25.6 (78.0)*	25.7 (78.3)	26.9 (80.5)	27.6 (81.8)
Ceiling-floor radiation,† W/m^2 (Btu/h·ft^2)	3.5 (1.09)	0.8 (0.26)	6.7 (2.14)	1.8 (0.57)
Slab-supply plenum conduction, W/m^2 (Btu/h·ft^2)	10.4 (3.31)	11.7 (3.71)	2.9 (0.93)	3.8 (1.19)
T_{ft}, °C (F)	20.7 (69.2)	19.8 (67.7)	21.8 (71.2)	20.1 (68.2)
T_{nf},‡ °C (°F)	22.2 (72.0)	22.2 (72.0)	22.2 (72.0)	22.2 (72.0)
Floor-supply plenum conduction, W/m^2 (Btu/h·ft^2)	4.9 (1.55)	3.0 (0.95)	7.2 (2.27)	3.8 (1.19)
Floor-room convection, W/m^2 (Btu/h·ft^2)	1.4 (0.45)	2.2 (0.70)	0.4 (0.14)	2.0 (0.62)
T_p,** °C (°F)	18.3 (65.0)	18.4 (65.1)	18.3 (65.0)	18.3 (64.9)
Total heat transfer into supply plenum, W/m^2 (Btu/h·ft^2)	15.3 (4.85)	14.7 (4.67)	10.1 (3.20)	7.5 (2.38)
% change from baseline	N/A	–3.8%	–34.0%	–51.0%
Return air extraction rate, W/m^2 (Btu/h·ft^2)	26.4 (8.38)	27.0 (8.56)	31.7 (10.03)	34.3 (10.86)
% change from baseline	N/A	2.2%	19.7%	29.5%
Room airflow rate,†† L/s·m^2 (cfm/ft^2)	3.1 (0.6)	3.1 (0.6)	3.1 (0.6)	3.1 (0.6)
T_{pi}, °C (°F)	14.1 (57.4)	14.3 (57.7)	15.5 (59.9)	16.2 (61.2)

* Assumed to be 25.6°C (78.0°F) for the baseline.
† Calculated based on a radiation view factor of 0.5.
‡ Assumed to be 22.2°C (72.0°F) for the baseline and held constant for all other cases.
** Assumed to be 18.3°C (65.0°F) for the baseline and held constant for all other cases.
†† Assumed to be 3.1 L/s·m^2 (0.6 cfm/ft^2) for the baseline and held constant for all other cases.

example, this produces a plenum inlet temperature of 13.0°C (55.4°F) for Table 2 and 14.1°C (57.4°F) for Table 3. This energy balance calculation on the supply plenum was then used to estimate the new plenum inlet temperature. All other temperatures were allowed to vary during the iterative solution technique. The constant supply air temperature leaving the plenum (18.3°C [65°F]) and predicted return air temperatures were used to calculate the new return air extraction rate for these other cases.

1. This method represents a hypothetical situation where a fixed average supply temperature entering the room at the diffusers was maintained by the HVAC control system by controlling the temperature of the air entering the plenum. In comparison, current practice typically controls to a constant plenum inlet temperature, thereby producing a supply temperature entering the room that will vary depending on the amount of heat transferred to the plenum.

RESULTS

Tables 4 and 5 present a summary of the room cooling load distribution with hung ceiling and without hung ceiling, respectively. Figures 3 and 4 present the same information graphically. The figures allow an easy comparison of the relative proportions of energy leaving the room through the two primary pathways: (1) heat extraction via warm return air exiting the room at ceiling level or through the return plenum and (2) total heat transferred to the underfloor supply plenum (through the slab and raised floor panels). Note that in Figures 3 and 4, 100% represents the heat gain to the room (i.e., the source of the loads) that leaves via return air extraction and heat transfer to the plenum. It is synonymous with the total energy leaving the *system*, composed of the room and supply plenum. In Tables 4 and 5, we refer to this as the “Total Room Cooling Load.” These results are discussed further in the following sections.

Table 4. Summary of Room Cooling Load Distribution with UFAD and Hung Ceiling; Room Airflow Rate = 3.1 L/s·m² (0.6 cfm/ft²)

Energy Flow	Baseline Ceiling ε_c = 0.9 No Slab Insulation		Case 1 Ceiling ε_c = 0.1 No Slab Insulation		Case 2 Ceiling ε_c = 0.9 Slab Insulation R-10		Case 3 Ceiling ε_c = 0.1 Slab Insulation R-10	
	W/m² (Btu/h·ft²)	% of Total	W/m² (Btu/h·ft²)	% of Total	W/m² (Btu/h·ft²)	% of Total	W/m² (Btu/h·ft²)	% of Total
Total room cooling load (= total heat transfer leaving room)	46.0 (14.57)	100	46.0 (14.57)	100	46.0 (14.57)	100	46.0 (14.57)	100
Heat transfer through slab into plenum	13.1 (4.16)	28.5	13.5 (4.26)	29.3	3.7 (1.19)	8.1	4.3 (1.37)	9.4
Heat transfer through floor into plenum	6.4 (2.03)	14.0	3.6 (1.14)	7.8	8.4 (2.66)	18.3	4.1 (1.30)	8.9
Total heat transfer into supply plenum	19.5 (6.19)	42.5	17.1 (5.40)	37.1	12.2 (3.85)	26.4	8.4 (2.67)	18.3
Return air extraction rate	26.4 (8.38)	57.5	28.9 (9.17)	62.9	33.8 (10.72)	73.6	37.6 (11.90)	81.7

Table 5. Summary of Room Cooling Load Distribution with UFAD and Exposed Slab (No Hung Ceiling); Room Airflow Rate = 3.1 L/s·m² (0.6 cfm/ft²)

Energy Flow	Baseline Ceiling ε_c = 0.9 No Slab Insulation		Case 1 Ceiling ε_c = 0.1 No Slab Insulation		Case 2 Ceiling ε_c = 0.9 Slab Insulation R-10		Case 3 Ceiling ε_c = 0.1 Slab Insulation R-10	
	W/m² (Btu/h·ft²)	% of Total	W/m² (Btu/h·ft²)	% of Total	W/m² (Btu/h·ft²)	% of Total	W/m² (Btu/h·ft²)	% of Total
Total room cooling load (= total heat transfer leaving room)	41.8 (13.23)	100	41.8 (13.23)	100	41.8 (13.23)	100	41.8 (13.23)	100
Heat transfer through slab into plenum	10.4 (3.31)	25.0	11.7 (3.71)	28.1	2.9 (0.93)	7.0	3.8 (1.19)	9.0
Heat transfer through floor into plenum	4.9 (1.55)	11.7	3.0 (0.95)	7.2	7.2 (2.27)	17.2	3.8 (1.19)	9.0
Total heat transfer into supply plenum	15.3 (4.86)	36.7	14.7 (4.66)	35.3	10.1 (3.20)	24.2	7.5 (2.38)	18.0
Return air extraction rate	26.4 (8.38)	63.3	27.0 (8.56)	64.7	31.7 (10.03)	75.8	34.3 (10.85)	82.0

Baseline (No Slab Insulation, Ceiling ε_c = 0.9)

A review of the baseline results in Tables 2 and 5 and Figures 3 and 4 is instructive to understanding the relative magnitudes of the major heat transfer processes in a typical room with a stratified UFAD system. The major findings are as follows:

- As shown for this baseline case, approximately 40% (37%–43%) of the total room cooling load leaving the space exits into the supply plenum and only about 60% is accounted for by the return air extraction rate. This is a good demonstration of why temperature gain in supply plenums can be a problem. In addition, these results indicate that the return air extraction rate based on the temperature difference between the room return and supply air temperatures in a stratified UFAD system will always be significantly less than the total room cooling load based on the sum of all space heat gains.
- The amount of energy leaving the underside of the ceiling and radiating down to the room below (the amount shown in the table is half of the total, due to the view factor of 0.5) is slightly less but still similar in magnitude to the amount of heat conducted up through the slab to the supply plenum on the floor above.

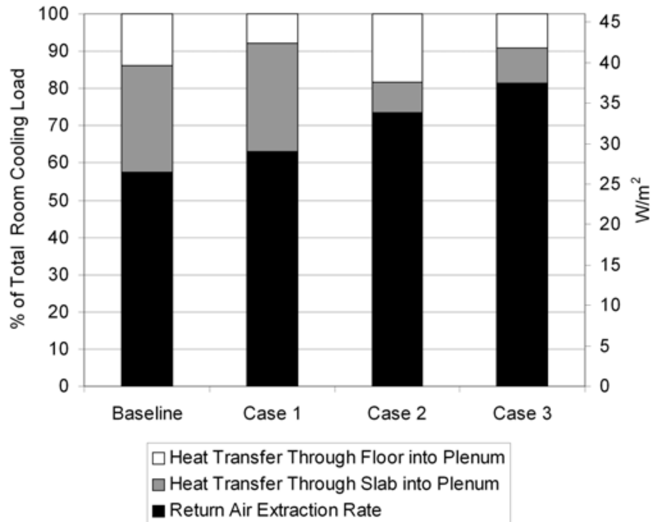

Figure 3 *Predicted percentage of total room cooling load and amount (W/m^2) of energy flows leaving a room with UFAD and hung ceiling; room airflow rate = 3.1 L/s·m^2 (0.6 cfm/ft^2).*

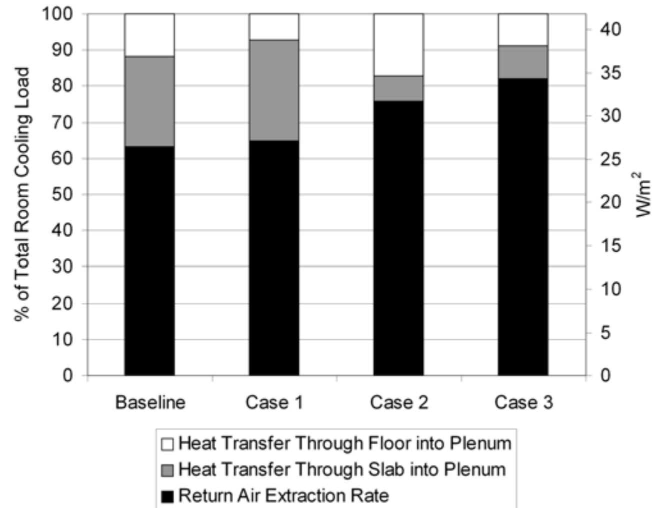

Figure 4 *Predicted percentage of total room cooling load and amount (W/m^2) of energy flows leaving a room with UFAD and exposed slab (no hung ceiling); room airflow rate = 3.1 L/s·m^2 (0.6 cfm/ft^2).*

- The radiation striking the floor makes up the majority of the total heat transfer that is conducted through the raised floor panels into the underfloor plenum. In comparison, only a small amount of this total heat transfer through the floor panels is contributed by convection between the room air near the floor and the floor surface (carpet).
- For both configurations (with and without hung ceiling), about twice as much of the total energy entering the supply plenum enters through the slab as through the floor panels.
- In Table 2, for the hung ceiling configuration, the plenum inlet temperature required to maintain the average plenum temperature of 18.3°C (65.0°F) is calculated to be 13.0°C (55.4°F), meaning that on average the air entering the plenum increases in temperature by 5.3°C (9.5°F) before exiting the plenum. In Table 3, for the exposed slab configuration, the plenum inlet temperature is 14.1°C (57.4°F), meaning the average temperature gain is 4.2°C (7.6°F).
- As shown in Table 2, the suspended ceiling serves as a radiator at a slightly higher temperature (compared to the exposed slab in Table 3), thereby producing larger heat transfer downward through the raised floor and upward through the slab into the supply plenum.

Case 1
(No Slab Insulation, Ceiling ε_c = 0.1)

The major findings are as follows:

- As shown in Tables 4 and 5 and Figures 3 and 4 for Case 1, 35%–37% of the total room cooling load leaving the space is transferred to the supply plenum and 63%–65% is accounted for by the return air extraction rate.
- The most obvious impacts of the low-e ceiling are a reduction in the radiant exchange between the ceiling and the floor and a resulting increase in the return air temperature.
- The total energy entering the supply plenum is only slightly reduced compared to the baseline (4%–13%) because the higher return temperature produces a higher rate of heat conduction through the slab, offsetting the decrease in heat entering through the raised floor (Tables 2 and 3). Due to this modest reduction of heat gain to the plenum, the plenum inlet temperature required to maintain the 18.3°C (65.0°F) supply air temperature increases by a relatively small amount (0.2°C–0.7°C [0.4°F –1.3°F]).
- The small increase in return air temperature (0.2°C–0.7°C [0.4°F–1.3°F]) produces only a modest increase in the return air extraction rate (2%–9%).

Case 2
(R-10 Slab Insulation, Ceiling ε_c = 0.9)

The major findings are as follows:

- In this example, approximately 25% of the total room cooling load is transferred into the supply plenum and 75% is accounted for by the return air extraction rate, representing an improvement in the efficiency of space heat removal by room airflow (Figures 3 and 4).
- Placing an insulation layer on the bottom of the slab proves to be an effective strategy for reducing the total

energy entering the supply plenum. It reduces conduction through the slab substantially, and although it causes higher radiant exchange between the ceiling and floor (due to the higher return temperature), it still reduces the total energy entering the supply plenum by 34%–38% (Tables 2 and 3).

- Due to the higher rate of radiative heat transfer between the ceiling and the floor, the top surface temperature of the carpet is warmer and in fact, for the hung ceiling configuration (Table 2), produces a net convective heat transfer from the floor to the room air.
- For Case 2, the plenum inlet temperature required to maintain the 18.3°C (65.0°F) supply air temperature is increased by 1.4°C–2.0°C (2.5°F–3.6°F) compared to the baseline.
- The higher room delta-T produces an increase in the return air extraction rate in the range of 20%–28% compared to the baseline, with the higher rate associated with the suspended ceiling (Tables 2 and 3).

Case 3 (R-10 Slab Insulation, Ceiling $\varepsilon_c = 0.1$)

The major findings are as follows:

- As shown in Tables 4 and 5 and Figures 3 and 4 for Case 3, less than 20% of the total room cooling load is transferred into the supply plenum and more than 80% is accounted for by the return air extraction rate.
- When slab insulation is combined with a low-e ceiling, the best distribution of energy flows is achieved. The total energy entering the supply plenum is reduced to its minimum level for all cases considered, representing a reduction compared to the baseline of more than 50%.
- The amount of energy entering the supply plenum through the slab is approximately equal to that entering through the floor panels.
- This example produces the highest return air temperature, representing an increase compared to the baseline of 2.1°C–3.0°C (3.8°F–5.4°F).
- This example also produces the highest plenum inlet temperature required to maintain the 18.3°C (65°F) supply air temperature, representing an increase compared to the baseline of 2.1°C–3.0°C (3.8°F–5.4°F). On average, the air entering the plenum increases in temperature by 2.1°C–2.3°C (3.8°F–4.1°F) before exiting the plenum.
- Case 3 produces the highest return air extraction rate, representing an increase compared to the baseline of about 30%–42%.

DISCUSSION

The present study was undertaken to improve our understanding of how heat is removed from a stratified room produced by a UFAD system under cooling operation. The results indicate that heat transfer into the cool underfloor supply plenum makes up a significant portion of the total room cooling load, particularly in a multi-story building. While many designers will point out that heat entering the supply plenum will simply reenter the conditioned space through the floor diffusers, these findings have important implications for proper design and operation of UFAD systems.

We can assume that the total room cooling load (the sum of all design space heat gains) is the same for both UFAD and conventional overhead (OH) systems. For the well-mixed OH system, 100% of the cooling load is removed by the return air extraction rate. However, based on the results of our simplified analysis, the total cooling load for a UFAD system should be split into two components, one to the supply plenum and one to the room (defined as the combination of room and return plenum). Although no UFAD cooling airflow design tool is currently available, when one is developed, it will be important for it to take into account this fundamentally different distribution of room loads. In addition, to ensure comfortable cooling operation, it is critical that (1) control and operating strategies be used, such as reducing the plenum inlet temperature, increasing the room airflow rate, and actively controlling of room air stratification or (2) other design decisions be employed, such as slab insulation, low-e ceilings, or other approaches for reducing the impact of heat gain into the supply plenum.

The model was used to simulate two ceiling configurations, with or without a hung ceiling. For the exposed slab case (Tables 3 and 5), the "room/return plenum" control volume represented a reasonably accurate model of the physical space. For the case with a hung ceiling (Tables 2 and 4), the model predicts a slightly higher percentage of the total room cooling load entering the underfloor plenum compared to the no-hung-ceiling case. This result is consistent with expectations. All things being equal, when the hung ceiling is present, it acts as a higher temperature-radiating element that will increase, if anything, the amount of energy reaching both the underside of the slab and the top of the raised floor and, therefore, the amount of energy entering the plenum. When the suspended ceiling is not present (exposed slab), the floor and underside of the slab exchange energy via radiation, but both are at lower temperatures than the hung ceiling.

Although the simplified model considered only heat transfer in the vertical direction, the results should be quite representative of interior open-plan office spaces, where UFAD is often used and heat transfer out of the room through walls will be at a minimum. In addition, it is likely that in perimeter zones under direct sunlight, some amount of the raised floor surface (carpet) will be illuminated, thereby adding to the heat gain into the plenum. To investigate the sensitivity of the simplified model predictions to some of these assumptions, additional simulations were conducted as briefly summarized in the following.

Sensitivity of Model Assumptions

Room Airflow Rate. The previously described modeling results assumed a room airflow rate of 3.1 L/s·m^2 (0.6 cfm/ft^2), producing a well-stratified room. Temperature measurements from many completed UFAD installations, however, do not always demonstrate such a high level of stratification, indicating that these buildings are being over-aired for the actual cooling load in the building. This is not a surprising situation due to the lack of a validated cooling airflow design tool; designers will tend to be conservative in their airflow design calculations. To investigate the impact of higher airflow rates, two additional model simulations were made with airflow rates of 4.1 and 5.1 L/s·m^2 (0.8 and 1.0 cfm/ft^2). The runs were made for the room with a suspended ceiling and were compared with the baseline simulation for this configuration. Table 6 presents the calculated temperatures and heat fluxes for these three cases. Table 7 presents a summary of the room cooling load distribution, and Figure 5 shows the same results graphically. The room airflow rate is the only parameter that is changed between the three cases. All other assumptions are the same as the baseline, including the total room cooling load (46.0 W/m^2 [14.57 Btu/h·ft^2]).

The detailed results shown in Table 6 demonstrate that, as expected, when holding the total room cooling load constant, increasing the airflow rate decreases the amount of room air stratification and also decreases the magnitude of the temperature gain in the underfloor plenum. The average rise in supply air temperature as it passes through the plenum is only 2.3°C (4.2°F) for the highest airflow rate of 5.1 L/s·m^2 (1.0 cfm/ft^2). This result may help to explain why anecdotal accounts from some actual installations indicate that excessive thermal decay in underfloor plenums has not been observed. We hypothesize that it is very likely that these projects have lower stratification due to greater airflow, diffuser characteristics, or other design and operating issues.

Table 6. Calculated Temperatures and Heat Fluxes in Room with UFAD and Hung Ceiling for Different Airflow Rates

Conditions	Baseline Airflow = 3.1 L/s·m^2 (0.6 cfm/ft^2)	Airflow = 4.1 L/s·m^2 (0.8 cfm/ft^2)	Airflow= 5.1 L/s·m^2 (1.0 cfm/ft^2)
T_{sb}, C (°F)	23.1 (73.6)	22.3 (72.1)	21.8 (71.2)
T_r , °C (°F)	25.6 (78.0)*	24.4 (76.0)	23.5 (74.3)
T_c, °C (°F)	23.8 (74.8)	22.9 (73.2)	22.3 (72.1)
Ceiling-floor radiation,† W/m^2 (Btu/h·ft^2)	5.7 (1.80)	4.5 (1.43)	3.7 (1.17)
Slab-supply plenum conduction, W/m^2 (Btu/h·ft^2)	13.1 (4.16)	11.0 (3.49)	9.4 (2.98)
T_{ft}, °C (°F)	21.4 (70.5)	21.0 (69.8)	20.7 (69.3)
T_{nf},‡ °C (°F)	22.2 (72.0)	22.2 (72.0)	22.2 (72.0)
Floor-supply plenum conduction, W/m^2 (Btu/h·ft^2)	6.4 (2.03)	5.6 (1.77)	5.0 (1.58)
Floor-room convection, W/m^2 (Btu/h·ft^2)	0.8 (0.24)	1.1 (0.35)	1.4 (0.44)
T_p,** °C (°F)	18.3 (65.0)	18.3 (65.0)	18.3 (65.0)
Total heat transfer into supply plenum, W/m^2 (Btu/h·ft^2)	19.5 (6.19)	16.6 (5.26)	14.5 (4.60)
% change from baseline	N/A	–14.9	–25.6
Return air extraction rate, W/m^2 (Btu/h·ft^2)	26.4 (8.38)	29.5 (9.35)	31.6 (10.02)
% change from baseline	N/A	11.7	19.7
Room airflow rate, L/s·m^2 (cfm/ft^2)	3.1 (0.6)	4.1 (0.8)	5.2 (1.0)
T_{pi}, °C (°F)	13.0 (55.4)	14.9 (58.8)	16.0 (60.8)

* Assumed to be 25.6°C (78.0°F) for the baseline.
† Calculated based on a radiation view factor of 0.5.
‡ Assumed to be 22.2°C (72.0°F) for the baseline and held constant for all other cases.
** Assumed to be 18.3°C (65.0°F) for the baseline and held constant for all other cases.

Table 7. Summary of Room Cooling Load Distribution with UFAD and Hung Ceiling for Different Airflow Rates

Energy Flow	Baseline Airflow = 3.1 L/s·m^2 (0.6 cfm/ft^2) W/m^2 (Btu/h·ft^2)	% of Total	Airflow = 4.1 L/s·m^2 (0.8 cfm/ft^2) W/m^2 (Btu/h·ft^2)	% of Total	Airflow = 5.1 L/s·m^2 (1.0 cfm/ft^2) W/m^2 (Btu/h·ft^2)	% of Total
Total room cooling load (= total heat transfer leaving room)	46.0 (14.57)	100	46.0 (14.57)	100	46.0 (14.57)	100
Heat transfer through slab into plenum	13.1 (4.16)	28.5	11.0 (3.49)	23.9	9.4 (2.98)	20.4
Heat transfer through floor into plenum	6.4 (2.03)	14.0	5.6 (1.77)	12.1	5.0 (1.58)	10.9
Total heat transfer into supply plenum	19.5 (6.19)	42.5	16.6 (5.26)	36.0	14.5 (4.60)	31.4
Return air extraction rate	26.4 (8.38)	57.5	29.5 (9.35)	64.0	31.6 (10.02)	68.6

As shown in Table 7 and Figure 5, the increased airflow rate also reduces the percentage of total room cooling load entering the underfloor plenum from about 43% for the baseline to 31% for the highest airflow rate. Nevertheless, even at this elevated airflow rate, representing an increase of 67% over the baseline rate of 3.1 L/s·m^2 (0.6 cfm/ft^2), these findings indicate that approximately one-third of the total room cooling load is still transferred into the underfloor supply plenum. Due to the increased airflow rate, the plenum inlet temperature is increased from 13°C (55.4°F) at baseline to 16°C (60.8°F) for the highest airflow. Future research is needed to address some of the control strategy trade-offs suggested by these sensitivity simulations, when a whole-building energy simulation program capable of modeling UFAD systems becomes available (CEC 2006). For example, optimizing performance may involve trading off fan energy savings from reduced airflow vs. economizer savings (in suitable climates) from increased airflow and greater coil leaving air temperatures.

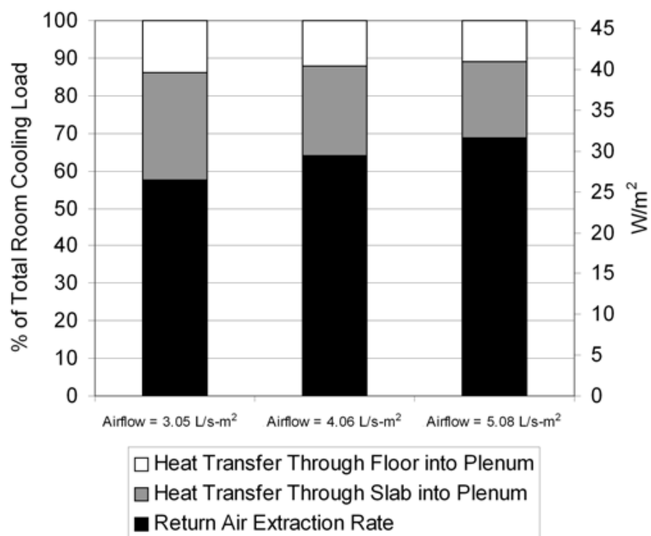

Figure 5 *Impact of room airflow rate on predicted percentage of total room cooling load and amount (W/m^2) of energy flows leaving a room with UFAD and hung ceiling.*

Slab Insulation. Preliminary model simulations investigated the choice of insulation layer on the underside of the 15 cm (6 in.) concrete slab. The thermal resistance of the slab plus the insulation layer was varied from no insulation (0.164 m^2·K/W [0.932 ft^2·h·°F/Btu] for the slab alone) to R-10 (1.924 m^2·K/W [10.9 ft^2·h·°F/Btu]), and up to R-30 (5.44 m^2·K/W [30.9 ft^2·h·°F/Btu]). R-10 slab insulation was selected for all simulations because it represented a more practical choice that reduced the steady-state slab heat transfer (compared to no insulation) by only 19% less than that of R-30 insulation.

Ceiling Emissivity. Although a ceiling emissivity level of 0.1 was selected for the model to investigate the maximum possible reduction in radiant exchange between the ceiling and floor surfaces, a quick review of the literature revealed that the lowest emissivities from commercially available low-e coatings were approximately 0.4. A set of simulations for the hung ceiling configuration (Figure 1) was conducted for a ceiling emissivity of 0.4. This only changed the model predictions for cases 1 and 3. Compared to results shown in Tables 2 and 4 for $\varepsilon_c = 0.1$, the major impact of the higher emissivity was to increase the radiative heat transfer between the ceiling and the floor (by more than a factor of 2) and to correspondingly increase the rate of heat transfer into the underfloor plenum through the raised floor panels. The other model predictions were quite similar.

Radiation View Factor. To account for the effect of furniture and other objects in the room that would block radiant exchange between the ceiling and the floor, a radiation view factor of 0.5 was used in the model. A set of simulations was conducted with this view factor equal to 0.3 and 0.7 to investigate variations in the density of furniture layout. These simulations were done for the hung ceiling configuration for the baseline and case 2, since radiation exchange between the ceiling and floor was at its highest level for these two cases ($\varepsilon_c = 0.9$). The results indicated a very small impact on the predicted distribution of room cooling load into the supply plenum or return air extraction rate. The maximum difference from values listed in Table 4 was 2%, with most falling within 1%.

Radiation Exchange with Room Furniture. For calculation purposes, the above described radiation view factor was modeled by assuming that 50% of a unit ceiling area radiated to the floor and the remaining 50% exchanged radiation with the furniture and other objects in the room. The modeling results presented in this paper have ignored the contribution of radiant exchange with the furniture. To investigate the sensitivity, a full set of simulations (with and without a hung ceiling) was conducted in which the radiative heat transfer between the ceiling and furniture was included. For these simulations, the furniture (and other objects) was assumed to be at a uniform temperature equal to the average of the room temperatures near the floor and ceiling. For example, in the baseline case, the room temperature near the floor was 22.2°C (72.0°F) and near the ceiling was 25.6°C (78.0°F), producing an assumed furniture temperature of 23.9°C (75.0°F). The results indicated that for almost all cases, the radiant exchange between the ceiling and furniture is quite small in comparison to the other major heat transfer pathways. The reason for this finding is that the ceiling temperature is always less than the return temperature and, in fact, is quite close to the assumed furniture temperature. Therefore, with a small temperature difference, there is little radiative heat transfer and little impact on the predicted distribution of room cooling load.

Benefits of Slab Insulation and Low-e Ceiling

Potential benefits of employing one or both of the ceiling thermal treatments investigated in this study are summarized below. Further testing and analysis would be needed to verify the magnitude of the estimated savings and improved operation of the system.

- *Reduced heat gain to supply plenum, thereby improving control of supply air temperature*. This benefit will allow design engineers greater confidence and freedom in the specification of their plenum designs, impacting such issues as supply temperature leaving the air handler, supply airflow rate, size of zone (distance to farthest diffuser), amount of ductwork, and number of plenum inlets.
- *Higher return air temperatures and higher resulting return air extraction rates.* The higher temperatures at the ceiling level increase the overall (return-supply) temperature difference for the room, thereby increasing the rate of heat removal via airflow from the room (return air extraction rate) for the same airflow quantity.
- *Potential to use higher supply air temperatures (compared to no insulation and normal ceilings) to achieve optimized performance of the system*. Due to the reduced heat gain into the underfloor supply plenum, higher plenum inlet temperatures may be used to achieve acceptable supply diffuser temperatures. Attention would need to be paid to the upper limit of acceptable room air stratification when making these control adjustments.
- *In a highly stratified environment with a suspended low-e ceiling, reduced radiant load from ceiling to occupied zone*. Furniture, people, and other objects in the occupied zone of the space will be slightly cooler, all other factors remaining the same. This reduced radiant load may allow higher room air setpoint temperatures to be used to maintain equivalent comfort conditions, producing additional energy savings.

CONCLUSIONS

A simplified heat balance model was used to estimate and compare the amount of energy being removed from a stratified room with UFAD under cooling operation 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 is shown 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 is transferred into the supply plenum and only about 60%–70% is accounted for by the return air extraction rate. This helps to explain why temperature gain in supply plenums can be a substantial problem in the control and operation of UFAD systems, particularly in well-stratified environments.

The model was used to investigate two strategies for reducing the magnitude of energy transferred into the underfloor plenum: (1) an insulation layer on the underside of the structural slab and (2) low-emissivity coating applied to the top and bottom surface of a hung ceiling (if present) or to the bottom surface of the slab or slab insulation (if no hung ceiling). The results indicate that a combination of R-10 (1.924 $m^2 \cdot K/W$ [10.9 $ft^2 \cdot h \cdot °F/Btu$]) slab insulation and low-e ceiling ($\varepsilon_c = 0.1$) reduces the total energy entering the supply plenum by 50% compared to the baseline. For this case, approximately 20% of the total room cooling load is transferred into the supply plenum and 80% is accounted for by the return air extraction rate. In addition, simulations at higher airflow rates also demonstrated a reduction in total energy entering the supply plenum and a corresponding decrease in the temperature gain within the plenum, albeit at the expense of higher fan energy use.

These findings have important implications for the design and operation of UFAD systems.

- In a stratified UFAD system, the return air extraction rate based on the temperature difference between the room return and diffuser supply air temperatures will always be significantly less than the total room cooling load based on the sum of all space heat gains.
- Cooling airflow design calculations must account for the distribution of room cooling load between the underfloor supply plenum and the room. At this time, such a design tool is sorely needed by the building design community.
- The amount and distribution of temperature variations in the underfloor supply plenum can have significant impacts on the cooling operation of a UFAD system. This is especially true in perimeter zones where increased diffuser supply temperatures can make it more difficult to satisfy peak cooling loads.
- The impact of higher airflow rates on stratification and temperature gain within the underfloor plenum suggests that more research and field studies are needed to determine optimized design and operating strategies for UFAD systems in terms of energy use, thermal comfort, and controllability.
- Slab insulation proved to be the most effective strategy among those investigated for reducing supply plenum heat gain in a multi-story building.
- The benefits of employing one or both of the ceiling thermal treatments investigated in this study include the potential for reduced energy use (under suitable climatic conditions) and significantly improved controllability of supply air temperature while also maintaining acceptable thermal comfort.

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NOMENCLATURE

T_{fb} = floor bottom temperature, °C (°F)
T_{ft} = floor top temperature, °C (°F)
T_{fta} = floor top absolute temperature, °R (K)
T_{sb} = slab bottom temperature, °C (°F)
T_{sba} = slab bottom absolute temperature, °R (K)
T_{st} = slab top temperature, °C (°F)
T_c = ceiling temperature, °C (°F)
T_{ca} = ceiling absolute temperature, °R (K)
T_r = return air temperature, °C (°F)
T_p = plenum air temperature, °C (°F)
T_{nf} = air temperature near floor, °C (°F)
T_{pi} = plenum inlet air temperature, °C (°F)
h_{cn} = natural convection coefficient, upward or downward, $W/(m^2 \cdot K)$ $(Btu/[h \cdot ft^2 \cdot °F])$
h_{cnu} = upward natural convection coefficient, $W/(m^2 \cdot K)$ $(Btu/[h \cdot ft^2 \cdot °F])$
h_{cnd} = downward natural convection coefficient, $W/(m^2 \cdot K)$ $(Btu/[h \cdot ft^2 \cdot °F])$
h_{cf} = forced convection coefficient, $W/(m^2 \cdot K)$ $(Btu/[h \cdot ft^2 \cdot °F])$
k_s = slab thermal conductivity, $W/(m^2 \cdot K)$ $(Btu/[h \cdot ft^2 \cdot °F])$
k_f = raised floor (including carpet) thermal conductivity, $W/(m^2 \cdot K)$ $(Btu \cdot in./[h \cdot ft^2 \cdot °F])$
t_s = slab thickness, m (in.)
t_f = floor thickness, m (in.)
R = insulation thermal resistance, $m^2 \cdot K/W$ $(ft^2 \cdot h \cdot °F/Btu)$
F = view factor
σ = Stefan-Boltzmann constant
ε_c = ceiling emissivity
ε_f = floor emissivity
ε_s = slab emissivity

REFERENCES

ASHRAE. 2004. *ANSI/ASHRAE Standard 55-2004, Thermal Environmental Conditions for Human Occupancy.* Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Bauman, F. 2003. *Underfloor Air Distribution (UFAD) Design Guide*. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

CBE. 2005. CBE Web site, www.cbe.berkeley.edu. Center for the Built Environment, University of California, Berkeley.

CEC. 2006. Energy performance of underfloor air distribution systems. http://www.energy.ca.gov/pier/buildings/projects/500-01-035-1.html. California Energy Commission (CEC) Public Interest Energy Research (PIER) Buildings Program, Sacramento, CA.

Jin, H., F. Bauman, and T. Webster. 2006. Testing and modeling of underfloor air supply plenums. *ASHRAE Transactions* 112(2).

Loudermilk, K. 1999. Underfloor air distribution solutions for open office applications. *ASHRAE Transactions* 105(1):605–613.

Nagase, O., K. Matsunawa, H. Izuka, S.-I. Tanabe, and T. Arai. 1995. Effects of internal load, supply air volume, and floor surface conditions on the thermal environment in the office space with underfloor HVAC. *Proceedings, Annual Conference of the Society of Heating, Air-Conditioning, and Sanitary Engineers of Japan, Hiroshima, Japan, October 2–4*.

Webster, T., F. Bauman, and J. Reese. 2002a. Underfloor air distribution: Thermal stratification. *ASHRAE Journal* 44(5):28–33.

Webster, T., F. Bauman, J. Reese, and M. Shi. 2002b. Thermal stratification performance of underfloor air distribution (UFAD) systems. *Proceedings of Indoor Air 2002, Monterey, CA, June 30–July 5*.

Yuan, X., Q. Chen, and L. Glicksman. 1999. Performance evaluation and design guidelines for displacement ventilation. *ASHRAE Transactions* 105(1):298–309.