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Thermal decay in underfloor air distribution (UFAD) systems: Fundamentals and influence on system performance

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ABSTRACT
Underfloor air distribution (UFAD) is a mechanical ventilation strategy in which the conditioned air is primarily delivered to the zone from a pressurized plenum through floor mounted diffusers. Compared to conventional overhead (OH) mixing systems, UFAD has several potential advantages, such as improved thermal comfort and indoor air quality (IAQ), layout flexibility, reduced life cycle costs and improved energy efficiency in suitable climates. In ducted OH systems designers have reasonably accurate control of the diffuser supply temperature, while in UFAD this temperature is difficult to predict due to the heat gain of the conditioned air in the supply plenum. The increase in temperature between the air entering the plenum and air leaving through a diffuser is known as thermal decay. In this study, the detailed whole-building energy simulation program, EnergyPlus, was used to explain the fundamentals of thermal decay, to investigate its influence on energy consumption and to study the parameters that affect thermal decay. It turns out that the temperature rise is considerable (annual median=3.7 K, with 50% of the values between 2.4 and 4.7 K based on annual simulations). Compared to an idealized simulated UFAD case with no thermal decay, elevated diffuser air temperatures can lead to higher supply airflow rate and increased fan and chiller energy consumption. The thermal decay in summer is higher than in winter and it also depends on the climate. The ground floor with a slab on grade has less temperature rise compared to middle and top floors. An increase of the supply air temperature causes a decrease in thermal decay. The temperature rise is not significantly affected by the perimeter zone orientation, the internal heat gain and the window-to-wall ratio.

1. INTRODUCTION

Underfloor air distribution (UFAD) is a method of providing space conditioning and ventilation to offices and commercial buildings. UFAD systems use an underfloor supply plenum located between the structural concrete slab and a raised access floor system to supply conditioned air through floor diffusers directly into the occupied zone [1]. UFAD systems have several potential advantages over traditional overhead systems, such as layout flexibility, improved energy efficiency in suitable climates and reduced life cycle costs [1]. Their performance has been investigated through field study investigations [2-4], full and bench-scale laboratory testing [5-7], computational fluid dynamics (CFD) and analytical modeling [5-8], and whole-building energy simulation [5, 9-15].

In UFAD systems, two distinguishing characteristics combine to change the dominant heat transfer dynamics related to energy balance in the conditioned space under cooling operation. These are: (1) room air stratification, in which comfortable air temperatures are maintained in the occupied zone near the floor but warmer air exists near the ceiling; and (2) underfloor air supply plenums, through which cool supply air is distributed to floor diffusers. The underfloor plenum creates a relatively cool reservoir of air extending across the entire building

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floorplate and establishes large-area pathways for heat to enter the supply plenum (1) through the slab in multi-story buildings from the warm return plenum below, and (2) through the floor panels from the room above and incident radiant heat loads. Thermal decay, defined as the temperature rise of the conditioned air due to convective heat gain as it travels through the underfloor supply plenum, is the result of this heat transfer process and is the subject of this paper.

Bauman et al. [16] used steady-state modelling to demonstrate that the magnitude of heat transfer into an underfloor plenum was larger than previously thought, resulting in a significant amount (35-45%) of the total zone cooling load entering the plenum. This heat gain to the plenum warms up the plenum air, resulting in higher diffuser discharge temperatures. To maintain space temperature control, room supply airflow rates will increase due to these higher diffuser supply temperatures, and thus will influence room air stratification and heat balances in the building. Schiavon et al. [14] showed that, according to whole-building energy simulations, on average (median value), 22% and 37% of the total zone peak UFAD cooling load goes to the supply plenum in the perimeter and interior zones, respectively. The heat transfer to the plenum manifests itself as thermal decay. Woods and Novosel [17] measured and compared the heat gain to the room and the actual heat extraction rates for both conventional overhead (OH) and UFAD systems and showed that the room heat extraction rate in UFAD systems is only 35-64% of the instantaneous heat gain to the room compared to 66-93% in OH systems. The authors indicate that the significant amount of heat that entered the supply plenum caused these differences. Schiavon et al. [18] showed that the mere presence of raised floor (not a UFAD system) in an OH system affects the heat exchange and thus the cooling load profiles compared to a system without the raised floor.

The objectives of this paper are to describe and quantify the heat transfer paths that cause thermal decay and how the thermal decay affects diffuser discharge temperatures, cooling load profiles, supply airflow rate, and energy consumption. In addition, through a sensitivity analysis, the influence of floor level, zone orientation, central air handler supply air temperature (SAT), climate, window-to-wall ratio, internal heat load and plenum configuration on the magnitude of the thermal decay is also investigated.

1.1 Example of field test

Results from field measurements conducted in May 2008 in a large open plan office building with UFAD provides evidence of the magnitude of plenum heat gain [3]. In Figure 1, each data point represents an individual diffuser supply temperature measured in the interior or perimeter zone on the 7th floor (2,700 m^2) of the building served by a single open plenum. Measurements were recorded in each sector (N, S, E, W) simultaneously using wireless sensing. Based on the recorded supply air temperature leaving the air handler (16.7 °C [62°F]), thermal decay as high as 5.3 °C (9.5°F) is observed at some diffusers, with the average for the entire plenum around 2.8°C (5°F). Additional details of this study, including plenum configuration, can be found in the reference.

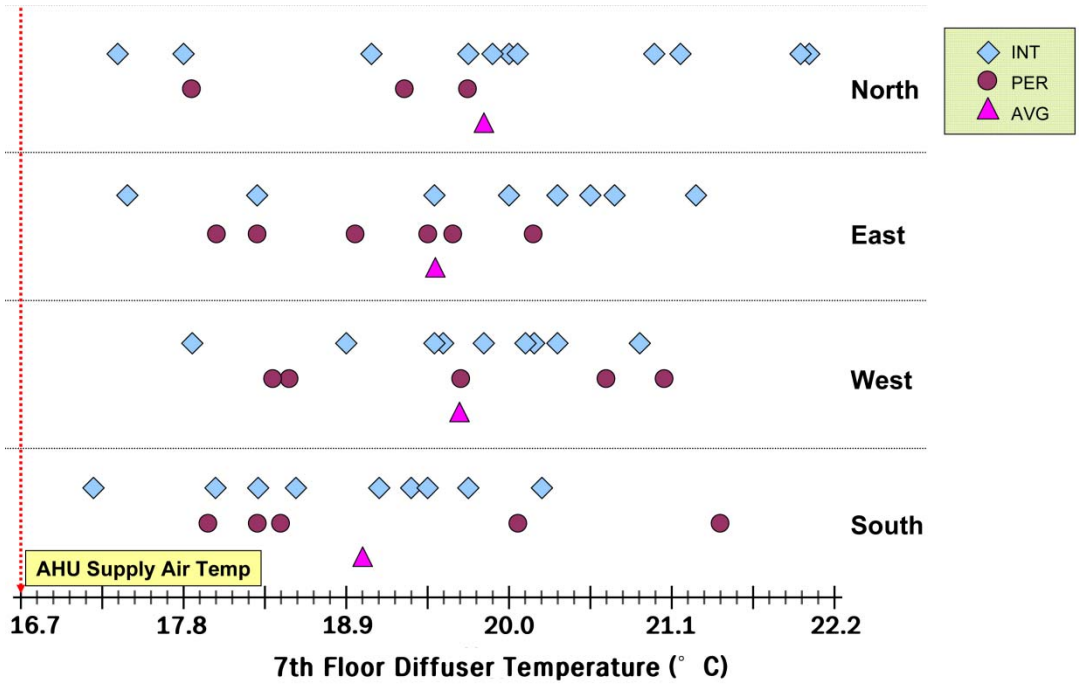


Figure 1. Supply plenum temperature distribution, 7th floor (INT = interior zone, PER = perimeter zone, AVG = overall average (perimeter + interior))

2. METHODS

2.1 Simulation software

The energy simulation program, EnergyPlus, was used for this study [19]. EnergyPlus is a relatively new building energy simulation program, which has greater capabilities than other programs [20]. EnergyPlus was selected because its capabilities were recently upgraded to model UFAD systems, allowing it to do the following: model each underfloor plenum as a completely separate zone; perform a full heat balance on the underfloor plenum which accounts for the heat gain into the plenum from both the room above and the return plenum from the floor below [9]; calculate the surface temperatures in each time step by conducting a detailed heat balance on each surface that includes the radiant heat exchange between surfaces and internal loads [9]; take into account the thermal mass effect; and model the temperature stratification generated by the UFAD system [5]. The EnergyPlus UFAD model has not been independently verified yet. For more information about validation of the EnergyPlus program, see [21].

2.2 Description of office building

A three-story prototype office building with a rectangular shape (75 m x 51 m) and aspect ratio of 1.5 was chosen for this study. The floor plate size is 3,720 m^2 (total floor area is 11,200 m^2) and each floor is composed of 4 perimeter zones, an interior zone and a service core, which represent approximately 28%, 56% and 16% of the floor area, respectively (see Figure 2). The floor to floor height is 3.96 m and the return plenum height is 0.6 m. The raised floor height is 0.4 m. Strip windows are evenly distributed (i.e., a “ribbon” window) in the walls and the baseline window-to-wall ratio (WWR) is 40%. Different WWRs are achieved by varying the window height only. The constructions and the thermal properties of windows change based on each climate and they comply with table 5.5 of ASHRAE 90.1-2004 [22]. When doing the design day simulation, ASHRAE 0.4% summer and 99.6% winter design conditions were assumed [23].An internal or external shading system was not used in the simulations.

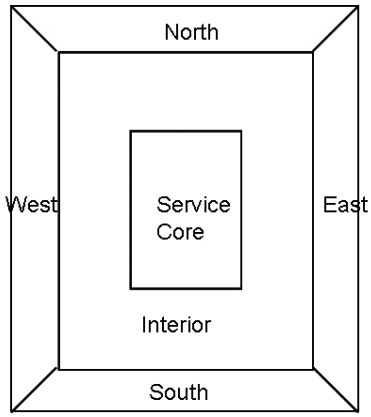


Figure 2. Floor plan of the simulated building.

2.3 Internal temperature, ventilation and infiltration rate, and HVAC system

From 5:00 till19:00 the system controls the internal air temperature to a cooling and heating temperature setpoint of 23.9°C and 21.1°C, respectively. During the nighttime the system is switched off. The infiltration was assumed equal to 0.000333 $m^3/(s\ m^2)$ (flow per exterior surface area). The minimum outdoor air flow rate was set to be 0.762 $L/(s\ m^2)$ (flow per gross area) and was provided from 5:00 until 19:00. Regarding the airflow outlets of each zone, the air is distributed through swirl diffusers in interior zones and linear bar grilles in the perimeter zones. Variable-speed fan coil units (FCU) supply air to the bar grilles in the perimeter zones only. The FCU shuts off when zone temperatures are in the dead-band between 21.1 and 23.9°C. In cooling mode the fan is on and the heating coil is off, and in heating mode, the fan and the heating coil are on. The building is served by a single variable-speed central station air handling unit (AHU) including an economizer, chilled water cooling coil, hot water heating coil and supply fan. The AHU fan is controlled with a static pressure reset strategy. The central plant consists of a centrifugal chiller with variable-speed pumps and a two-speed cooling tower. A gas fired boiler provides hot water to all heating coils. Table 1 shows details of system and plant inputs.

Table 1. Summary of HVAC system configuration.

HVAC	
AHU supply temperature	13.9°C, 15.6°C, 17.2°C
AHU fan design static pressure	1025 Pa (Ducted) 750 Pa (Series and parallel)
AHU fan efficiency	75%
AHU part load shutoff[1]	125 Pa
Minimum outside air rate	7.62 E-04 $m^3/s/m^2$
System cycles at night	No
Zone minimum airflow	7.62 E-04 $m^3/s/m^2$
Interior zone reheat	No
FCU design static pressure	125 Pa
FCU design efficiency	15%
Plant	
Chiller design COP	5.0
Boiler design efficiency	80%

[1]Designates low shutoff pressure associated with static pressure reset

2.4 Internal heat gains and occupancy

In this paper and in Figure 3, the fraction of the design value is defined as the ratio of the actual load to the peak or maximum load. The occupants' presence in the building varied according to Table 2 and Figure 3. Three levels of internal cooling load (people, equipment and lighting) were simulated, and they are summarized in Table 2. The equipment and lighting loads follow the schedules shown in Figure 3.

Table 2. Internal load level specifications.

Hear load category	Internal heat load 1	Internal heat load 2	Internal heat load 3
Overhead lighting, W/m^2	10.8	11.8	12.9
Peak occupancy, m^2/person	22.3	11.2	8.4
Equipment W/m^2	8.6	22.6	35.5
Total, W/m^2 (W/ft^2)	22.6 (2.1)	40.9 (3.8)	58.1 (5.4)

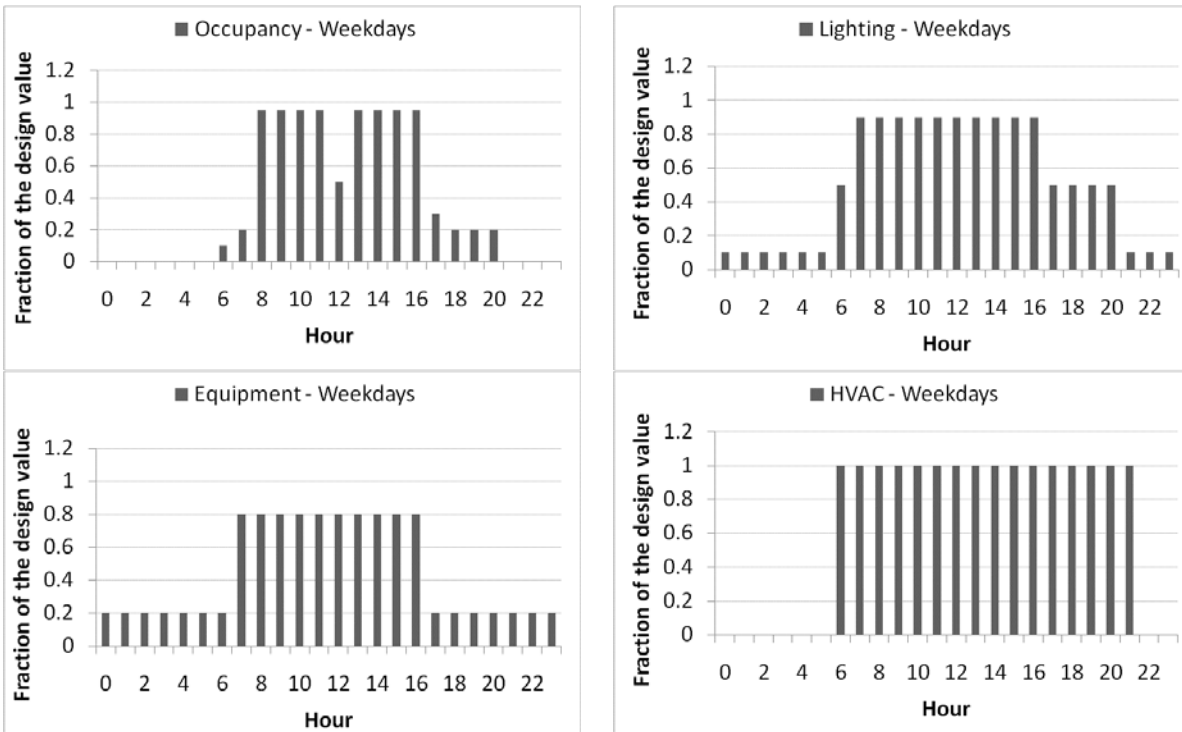


Figure 3. Occupancy, lighting, equipment and HVAC schedules.

2.5 Simulated cases

The purpose of the study is to investigate the influence of thermal decay on the UFAD thermal behavior. Some important parameters affecting the thermal decay that were studied are: floor level, zone orientation, central air handler supply air temperature (SAT), climate, window-to-wall ratio, internal load, and plenum configuration. The simulated cases are listed in Table 3. Regarding the plenum configuration, the "series" option indicates that all the conditioned air leaving the air handler first enters the interior plenum and, after gaining heat due to thermal decay, the warmer air leaving the interior plenum then enters each perimeter plenum. In the "parallel" plenum configuration, the conditioned air from the AHU independently enters each plenum in parallel (equivalent to ducting supply air out to a separate (subdivided) perimeter plenum. "Ducted" option is an idealized configuration in which the conditioned air is ducted all the way to the diffusers and thus, no thermal decay exists. As shown in Table 1, AHU fan design static pressure for series and parallel configurations use 750 Pa, as

opposed to 1025 Pa for the ducted configuration (equivalent to a fully ducted overhead air distribution system). In addition, three different levels of internal load are summarized in Table 2.

Table 3. Simulated cases.

Case	Plenum configuration	AHU SAT *	WWR **	Internal load ***	Climate
1 (reference)	Series	15.6	40	#1	Baltimore, MD
2	Parallel	15.6	40	#1	Baltimore, MD
3	Ducted (no thermal decay)	15.6	40	#1	Baltimore, MD
4	Series	13.9	40	#1	Baltimore, MD
5	Series	17.2	40	#1	Baltimore, MD
6	Series	15.6	20	#1	Baltimore, MD
7	Series	15.6	60	#1	Baltimore, MD
8	Series	15.6	40	#2	Baltimore, MD
9	Series	15.6	40	#3	Baltimore, MD
10	Series	15.6	40	#1	Minneapolis, MN
11	Series	15.6	40	#1	Miami, FL
12	Series	15.6	40	#1	San Francisco, CA

* Air handling unit supply air temperature (ºC)
** Window-to-wall ratio (%)
*** Details summarized in Table 2

3. RESULTS AND DISCUSSION

3.1 Heat transfer fundamentals of thermal decay

EnergyPlus performs a fundamental energy balance on each building surface by explicitly calculating all three heat transfer components (radiation, convection, conduction) for each hourly time-step of the simulation. Figure 4 shows a schematic diagram identifying the important horizontal building surfaces in a UFAD system. Thermal decay is caused by heat transfer into the underfloor plenum through the concrete slab (from the return plenum below) and through the raised floor panels (from the room above). Figure 5 illustrates a typical daily pattern of hourly surface heat fluxes, as predicted by EnergyPlus, for the key surfaces that influence thermal decay: top and bottom of the raised floor, and top and bottom of the concrete slab. Results are shown for the hours of 8:00-20:00 for a west perimeter zone during a peak design day simulation. To assist in interpreting the direction of heat flows at the surfaces in question, positive heat flux refers to radiation, convection, or conduction that is flowing upward.

Beginning with the top of the carpeted raised floor (top surface shown in Figure 5), it is seen that the incident radiant heat transfer is the largest heat flux shown for all four surfaces. This heat flux includes the radiant portion of internal heat sources in the room, radiant exchange with other room surfaces, and solar radiation striking the floor in the afternoon (peaking at nearly 40 W/m^2). The radiant heat flux absorbed by the top surface of the raised floor (downward heat flow) is equal to the sum of convection and conduction heat fluxes leaving the surface. In this case, the relatively warmer floor conducts heat downward into the floor panel and convects heat upward into the room, contributing to the room portion of the total cooling load. At the bottom surface of the raised floor (second surface shown in Figure 5), the direction of all three heat transfer components is downward into the underfloor plenum. The conduction arrives at the bottom surface after having passed through the raised floor panel and is equal in magnitude to the sum of radiant and convective heat flux leaving the underside of the floor panel. Convective heat transfer from the raised floor bottom contributes directly to thermal decay and varies over the range of 7-10 W/m^2 during the day. Note that for this example the AHU supply temperature entering the plenum each hour is 15.6°C.

Turning our attention to the concrete slab (bottom two surfaces shown in Figure 5), it is seen that again the dominant mechanism for heat transfer entering the bottom of the slab (heat flow upward) is radiant exchange with the top of the drop ceiling (Figure 4), which has been warmed by stratification in the room. The magnitude of this heat flux (peaking out at about 6 W/m^2 in the late afternoon) is significantly less than that of the incident radiant flux on top of the raised floor identified above (peaking at nearly 40 W/m^2). The amount of conduction which flows upward into the concrete slab is nearly the same as the radiant heat flux, leaving a convective heat flux between the slab and return plenum air that is surprisingly close to zero for most of the day. At the top

surface of the concrete slab, the results indicate that there is more conduction occurring at the top of the slab in the morning hours than in the afternoon hours due to thermal storage effects and the operating schedule of the HVAC system. When the UFAD system is turned on in the morning, heat that has been stored in the slab overnight is released into the cool plenum by convection. The time delay in heat conducting through the slab is also demonstrated by the fact that the pulse of heat entering the bottom of the slab during the late afternoon hours (peak loads) never reaches the top of the slab during normal operating hours. The storage of this afternoon heat in the slab clearly contributes to the higher conduction values at the top of the slab in the morning. The magnitude of the convective heat flux from the top of the slab into the plenum air (contributing to thermal decay) varies between about 6-8 W/m^2 over the day. The slight increase in the late afternoon is actually due to radiant heat transfer down from the underside of the floor panels.

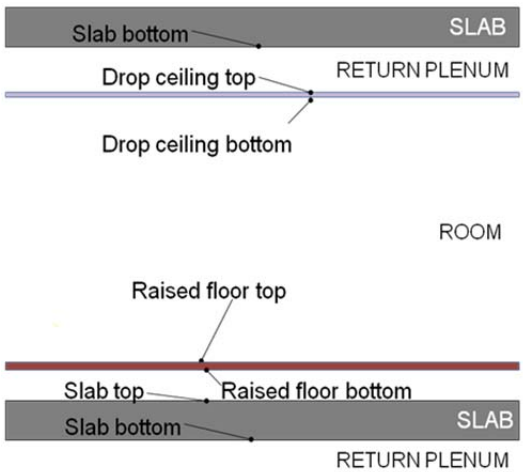
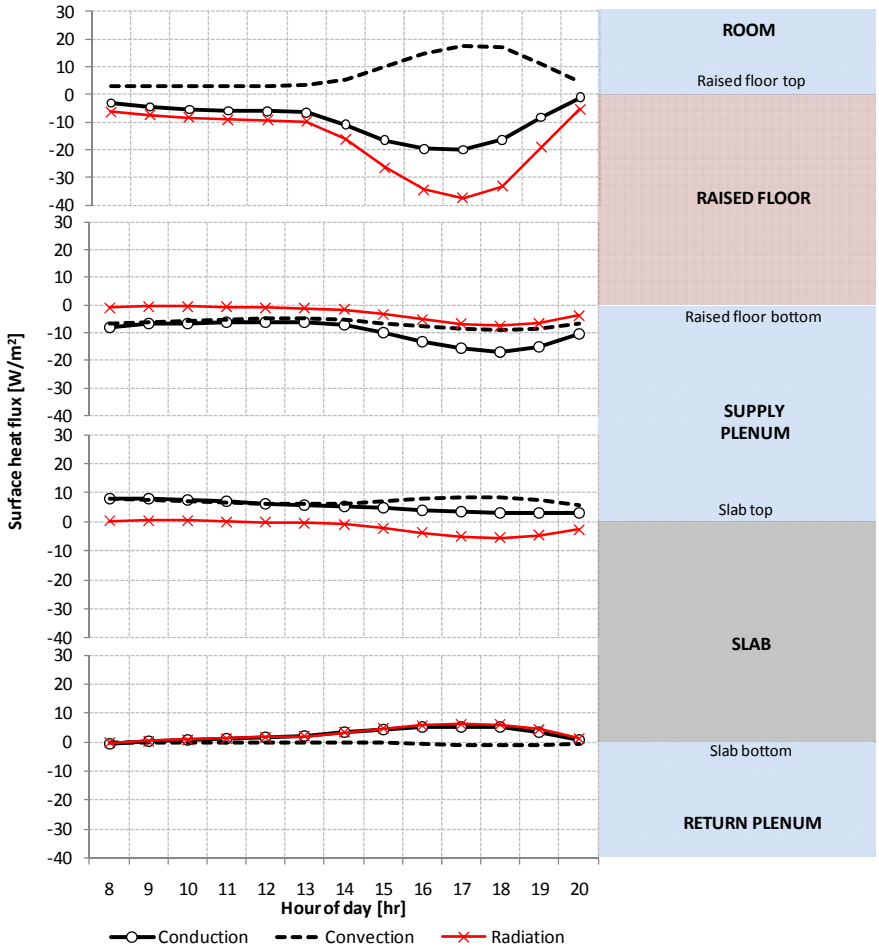


Figure 4. Schematic of room, slab and plenums for middle floor of UFAD system.



Notes: 1) Positive heat flux = upward heat flux
2) The dimensions of the UFAD scheme (raised floor, supply plenum, slab) are not in scale in order to keep the same surface heat flux scale

Figure 5. Surface heat transfer breakdown for underfloor supply plenum (middle floor, west).

3.2 UFAD airflow rate and diffuser discharge temperature

Figure 6 illustrates the hourly variations of diffuser discharge temperature for each zone in the middle floor during the summer design day period for Cases 1, 2 and 3 (see Table 3). By comparing Cases 1 and 3, the influence of the thermal decay can be clearly observed. As shown, the series plenum configuration (Case 1) always shows higher discharge temperatures than the air handler supply air temperature (SAT) setpoint due to heat gain in the underfloor plenum. On the other hand, in Case 3 it shows a constant discharge temperature of 15.6°C, which is the same as the air handler SAT (duct heat gain is assumed to be zero for Case 3). In Case 1 it can be observed that the average temperature rise during occupied hours is no lower than 4.6 K for north, 4.8 K for east, 4.6 for south, 4.2 K for west and 2.6 K for the interior zone. The temperature rise in Case 1 is always higher in the perimeter zones than in the interior (core) zone. This is due to the "series" plenum configuration where all the conditioned air leaving the air handler first passes through the interior plenum and the warmer air leaving the interior plenum then enters each perimeter plenum. Therefore, the discharge temperatures in the perimeter zones are the result of two temperature rises: first in the interior and then in the perimeter plenum, making the diffuser discharge temperature of the perimeter always higher than that of the interior. The sharp peaks in the early morning are due to the start-up condition. During the night (the system operation is from 6 am to 10 pm), the system is off, no conditioned air is supplied into the plenum and thus the heat is accumulated in

the plenum. When the system starts to operate in the early morning, the accumulated heat is removed first, producing a high plenum temperature rise in the early warm-up period.

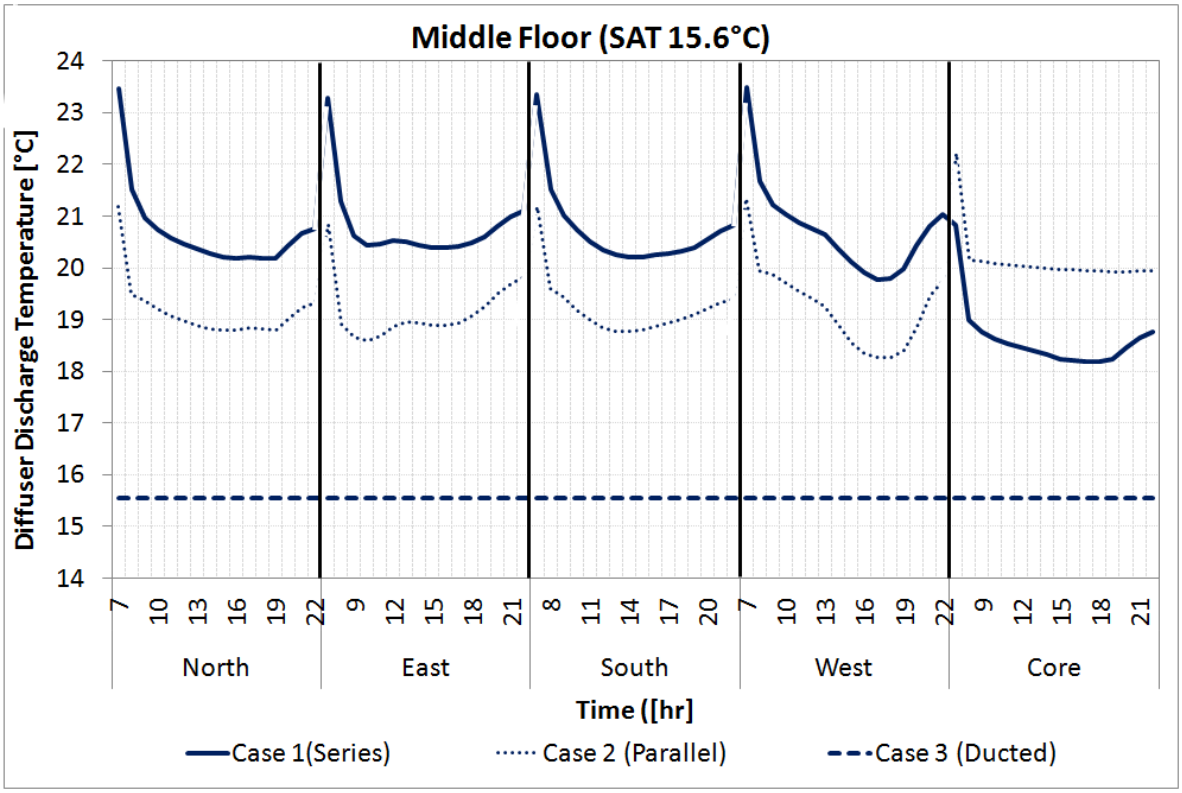


Figure 6. Average diffuser discharge temperature (Middle floor).

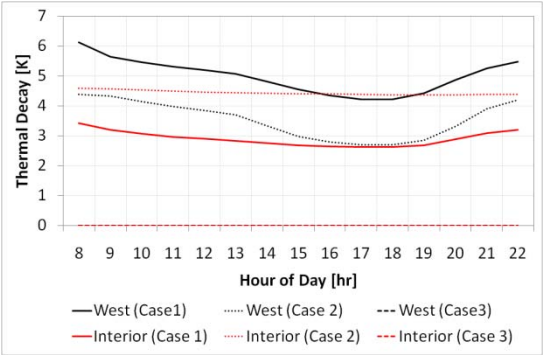


Figure 7. Thermal decay (middle floor) for the west and interior zone for Case 1, 2 and 3 (for Case 3 the thermal decay is always zero).

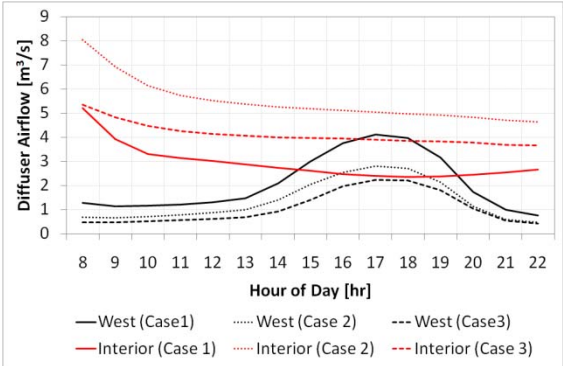


Figure 8. Zone supply airflow rate (middle floor) for the west and interior zone for Case 1, 2 and 3.

Figure 7 illustrates the thermal decay for the middle floor interior and west perimeter zones. For the west perimeter zone, Case 1 (series plenum) shows higher thermal decay than Case 2 (parallel plenum) due to the fact that the plenum air gains heat as it passes through the interior plenum zone on its way out to the perimeter in the "series" plenum configuration, as discussed earlier. On the other hand, for the interior zone, Case 2 shows higher thermal decay then Case 1 due to the lower airflow rate passing through the interior supply plenum in parallel

plenum configuration. Unlike the series plenum configuration, the supply airflows for the perimeter zones do not pass through the interior plenum in Case 2. In Case 3, thermal decay is assumed to be zero for all zones.

Figure 8 presents the hourly supply airflows of the west and interior zones in Cases 1, 2 and 3 during the design day period. For the west perimeter zone, Case 1 (with thermal decay) shows higher airflow rates than Case 3 (no thermal decay), as expected, because for Case 1 the larger supply airflow rate results from the increased diffuser discharge temperature. The increase in airflow during the afternoon is due to the solar radiation reaching the west zone. However, in the interior zone, the supply airflow turns out to be higher in Case 3 without thermal decay than in Case 1 with thermal decay. This is due to the fact that there is no conditioned air supplied into the plenum in Case 3 and thus the heat is continuously accumulated in the plenum, increasing the surface temperatures of the adjacent raised floor and the concrete slab. Therefore, as can be seen in Figure 9 for the interior zone, the convective heat gain to the room from both raised floor and suspended ceiling in Case 3 (without thermal decay) is much higher than those in Case 1 (with thermal decay), therefore raising the supply airflow rate despite the low diffuser temperature in Case 3. Although Case 3 does not represent a system configuration that occurs in practice, the results demonstrate that EnergyPlus is properly accounting for the heat transfer processes.

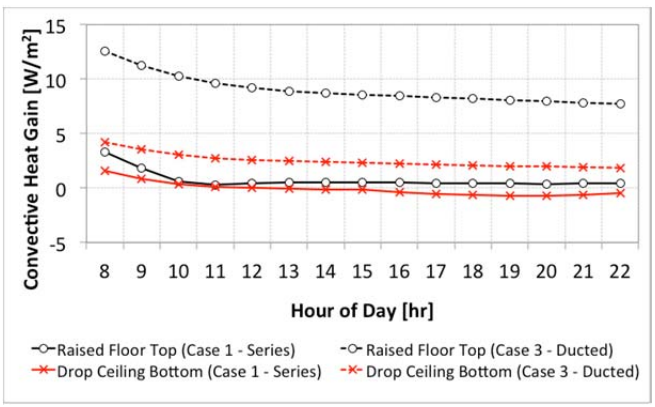


Figure 9. Convective heat gains in the room from the raised floor and suspended ceiling (middle floor, interior) in Case 1 (continuous lines) and Case 3 (dotted lines).

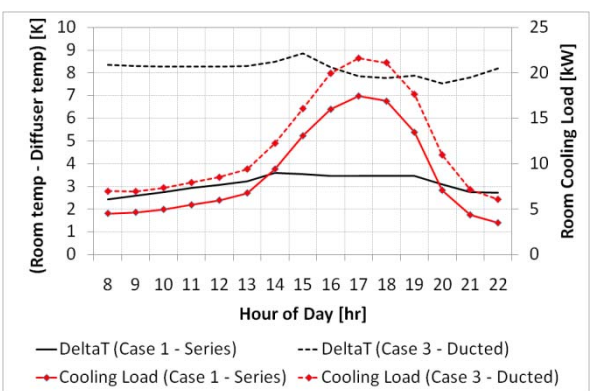


Figure 10. *Room* cooling load vs. DeltaT$_{rm}$ between room and diffuser temperatures (middle floor, west).

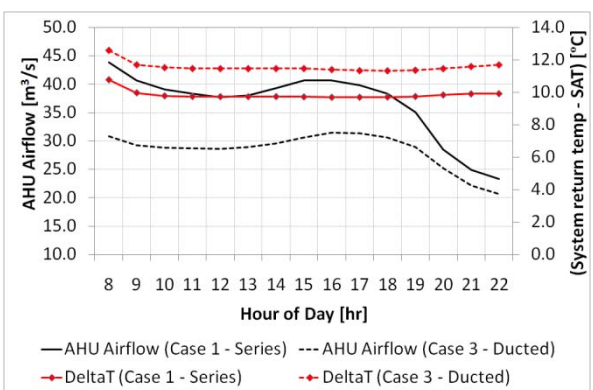


Figure 11. System (AHU) airflow vs. DeltaT$_{sys}$ between return and supply air.

Figure 10 shows a comparison of room cooling load and supply air to room temperature difference ($DeltaT_{rm}$) for the West perimeter zone for Cases 1 and 3. While the cooling load is greater for Case 3 as previously discussed, the $DeltaT_{rm}$ is proportionally much greater causing the room airflow to be less for Case 3. This is reflected at the system level as shown in Figure 11 where air handler (AHU) airflow and supply return temperature difference ($DeltaT_{sys}$) are shown. However, there is only a small difference in the product of these two factors (i.e., system demand) for the two cases. On an annual basis (not shown) Case 1 demand is about 10% greater than Case 3, apparently due to heat transfer and thermal storage differences.

Figure 6 through Figure 11 show that the thermal decay can have a large impact on the supply air flow rate and temperature. These affect fan and chiller energy consumption, indicating that the thermal decay should be carefully considered to assess UFAD performance. Although not reported here, the cooling load split between the room, supply plenum and return plenum was analyzed. Results similar to those reported in [14] were obtained.

3.3 Parallel configuration

In addition to the series plenum configuration, another possible option is the parallel configuration where the conditioned air from the AHU independently enters each plenum in parallel. Since the plenum configuration can have significant impact on the thermal decay, it is worth looking into this configuration in detail. As opposed to the series configuration where AHU supply air first enters the interior plenum and then the perimeter plenums in series, the parallel configuration can reduce the perimeter diffuser discharge air temperature. The diffuser temperature difference between the series and parallel configurations are illustrated in Figure 6. As shown, the parallel configuration (Case 2) has lower perimeter and higher interior discharge temperatures compared to the series configuration (Case 1). The lower perimeter discharge temperature is due to the fact that there is only one temperature rise in the parallel configuration as opposed to the series configuration where there are two temperature rises, first in the interior and then in the perimeter plenum. The zone supply airflows for each zone in Cases 1 (series) and 2 (parallel) are also presented in Figure 8. As shown, the parallel configuration has lower perimeter but higher interior supply airflows compared to the series configuration. The lower temperature rise in the perimeter plenum (Figure 6) reduces the perimeter supply airflow, while the higher plenum temperature rise in the interior plenum increases the interior supply airflow.

3.4 Influence of thermal decay on energy consumption

In this section is reported a simple evaluation of the impact of plenum configuration on energy consumption. The annual HVAC primary "source" energy usage of the three different plenum configurations under Baltimore climatic conditions is summarized in Figure 12. The secondary to primary conversion factors of 3.167 and 1.084 are assumed for the electricity and natural gas, respectively [19]. Comparing Case 1 to Case 3, the chiller and fan energy use are increased by the effects associated with thermal decay by 19% and 8.5%, respectively.
As expected, Case 3 without any thermal decay shows the lowest energy consumption compared to Case 1 and Case 2. The annual total source HVAC energy of Case 3 is 12.7% less compared to baseline Case 1 with the series configuration. Comparing Case 1 to Case 2, the parallel plenum configuration (Case 2) has less thermal decay which reduces the chiller and fan energy in much the same way as Case 3 does. However, the heating energy for Case 2 is increased (due to increased reheat in the perimeter) compared to Case 1, and thus there is a tradeoff between cooling and heating energy between these two different configurations, making the overall HVAC energy difference between Case 1 and Case 2 only 2.2%. From this simple analysis, it appears that reducing the thermal decay may lead to energy savings.

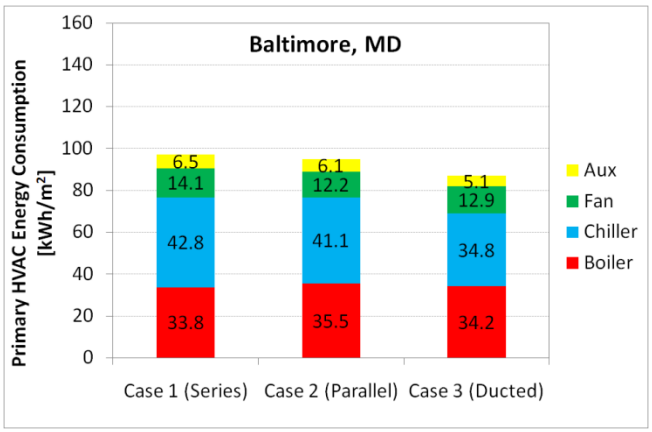


Figure 12. Annual primary "source" energy usage (kWh/m^2).

3.5 Sensitivity analysis: Parameters affecting thermal decay

The impact of variations in key parameters on the plenum temperature rise is shown as box-plots in Figure 13, Figure 14 and Figure 15 derived from annual simulations of the model described in Section 2.2. These parameters include the floor level, zone orientation, central air handler SAT, climate, window-to-wall ratio, internal load and plenum configuration. A box-plot is a way of graphically summarizing a data distribution. In a box-plot the dark horizontal line in the box shows the median value. The bottom and top of the box show the 25th and 75th percentiles, respectively. The horizontal line joined to the box by the dashed line shows either the maximum or 1.5 times the interquartile range of the data, whichever is smaller. Points beyond those lines are outliers. The interquartile range is the difference between the 25th and 75th percentiles [24]. In Figure 13, Figure 14 and Figure 15 each zone of each floor contributed one data point. The relevance of each point has not been weighted by the area represented by the zone. The aim of this paper was not to develop a predictive model of the thermal decay (in this case weighting by floor area would be needed) but to investigate the impact of thermal decay on UFAD system behavior and to study which are the parameters that affect the most thermal decay.

For all the simulated cases, the median temperature rise is 3.7 K during the occupied hours with 50% of the values between 2.4 and 4.7 K. In Figure 13b the box-plots of the temperature rise for the three different plenum configurations are presented. The negative thermal decay indicates that there is a heat loss instead of heat gain, but it represents a very small frequency as shown in Figure 13a. The results indicate that the series configuration shows the highest temperature rise as expected. In the series configuration, the conditioned air should travel through both interior and perimeter plenums before entering the perimeter terminal units, while it needs to pass through only the perimeter plenums in the parallel configuration. Although the temperature rise in the interior zone is lower in series configuration compared to parallel configuration, the higher temperature rise in the perimeter zones in the series configuration turns out to increase the overall temperature rise as shown in the figure. The ducted options do not show any temperature rises due to the modeling assumption of no duct heat gain. By comparing (a) and (b) graphs of Figure 14 showing the results of each zone, the parallel configuration (Case 2) in Figure 14 shows higher temperature rise in the interior, but shows lower temperature rise in the perimeter zones, eventually decreasing the overall temperature rise compared to series configuration (Case 1). The temperature rise is not significantly affected by the perimeter zone orientations. The median was around 3-4 K for all the four perimeter zones, while the distribution slightly varies among each zone.

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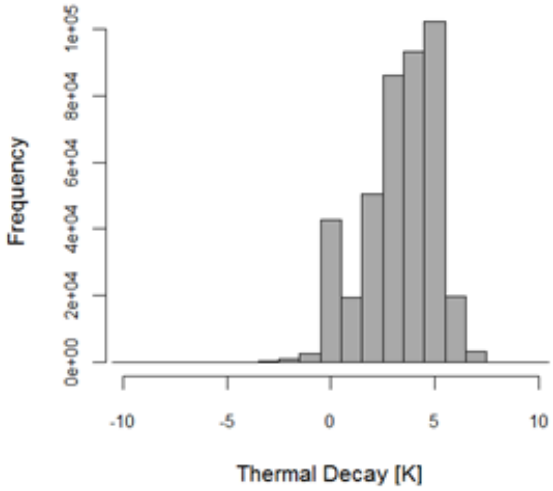
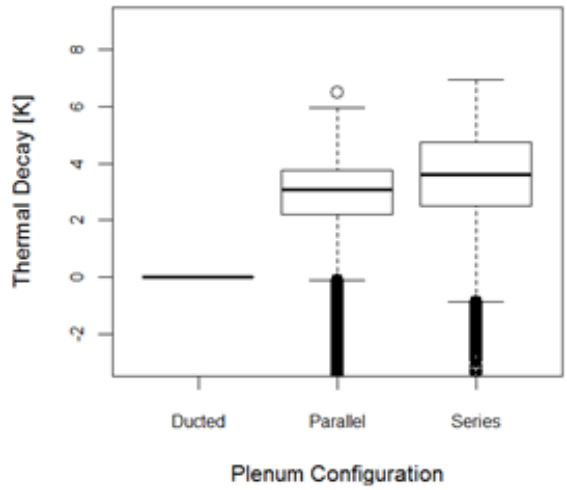
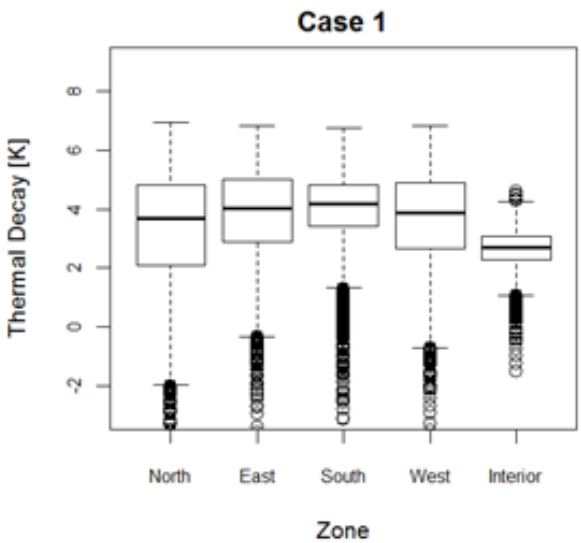


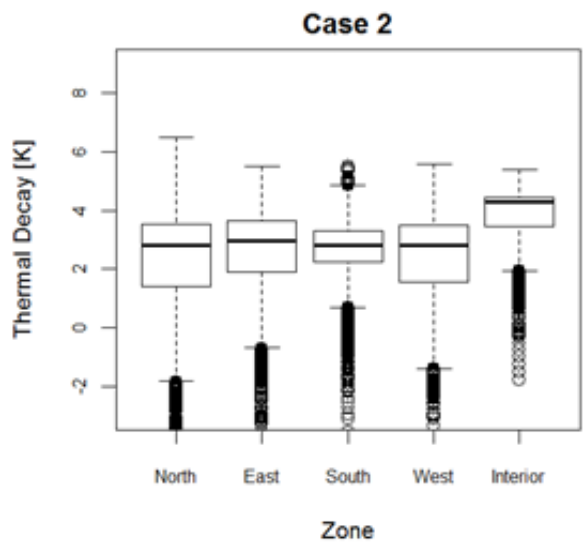
(a) Thermal decay frequency



(b) Plenum configuration vs. thermal decay



(c) Zone orientation vs. thermal decay (Case 1)



(d) Zone orientation vs. thermal decay (Case 2)

Figure 13. Box-plots of the thermal decay distribution calculated for all simulated cases versus different plenum configurations, and versus zones (Core (i.e., Interior), East, North, South and West).

The thermal decay for different floor levels (ground, middle and top floors) is illustrated in the first plot of Figure 14. The results indicate that the ground floor has less temperature rise compared to middle and top floors. This is due to the fact that the ground floor slab is in direct contact with the relatively cooler soil, while the bottom surfaces of middle and top floor slabs are facing higher temperature return plenums of the floor below, increasing the heat gain of the plenum air from the concrete slab. The second plot of Figure 14 presents the results for each month. The median value increases in the summer seasons and decreases in the winter seasons due to the higher heat gains in the summer compared to winter. Although there are some hours when the temperature rise goes

below zero in the winter, i.e., the plenum air is delivered into the room at a lower temperature than when it is supplied into the plenum, the median value in the winter is positive, indicating that temperature rise takes place even in the cold winter period. In addition, the number of hours when thermal decay is negative is very small, as indicated in Figure 13a.

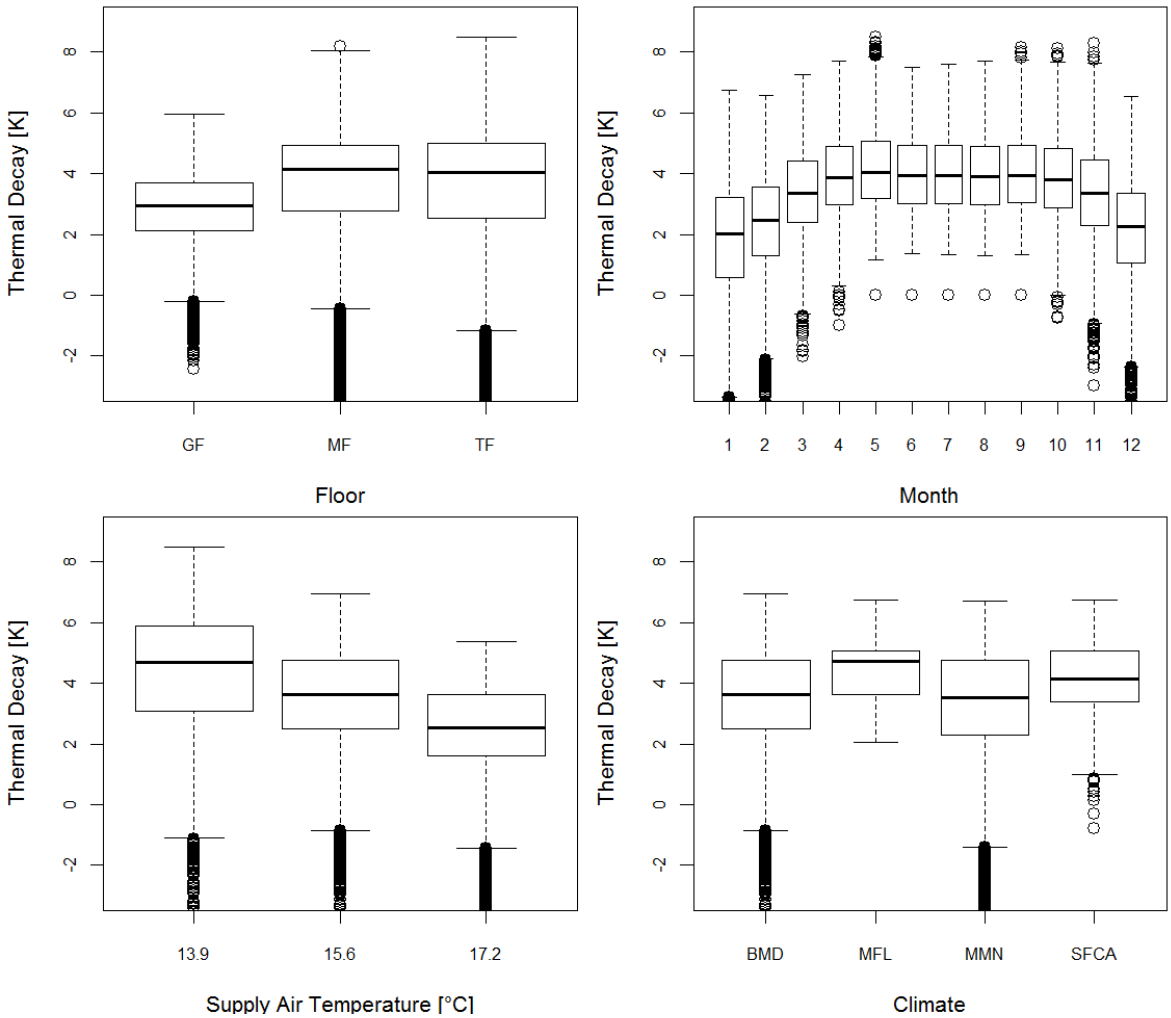


Figure 14. Box-plots of the thermal decay versus the floor level, versus each month, versus SAT and versus climates.

The third plot of Figure 14 illustrates the sensitivity to the central air handler SAT. As the SAT increases, the thermal decay decreases mainly due to the increased supply airflow with higher SAT and the smaller temperature difference between the plenum air and the heat transfer surfaces (slab and floor panels). The median value for SAT 13.9°C is 4.6 K as opposed to 2.6 K in SAT 17.2°C. Given the similar cooling load of the room, the supply airflow should increase as the SAT gets higher, which, in turn, reduces the temperature rise in the plenum. The annual fan energy consumption directly connected to the supply airflow explains this discussion. The fan energy of SAT 13.9°C case is 13.9 kWh/m^2, while it is 16.1 kWh/m^2 for SAT 17.2°C case, representing a 16% increase in average airflow for SAT 17.2°C. The last plot of Figure 14 shows the climate dependency of the thermal decay. As expected, the hot Miami climate (MFL) causes the greatest heat gain in the plenum compared to other mild and cold climates. The mean value of Miami is 4.7 K, while Baltimore, Minneapolis and San Francisco were 3.6

K, 3.5 K and 4.2 K, respectively. Baltimore and Minneapolis showed wider distribution than the other two climates.

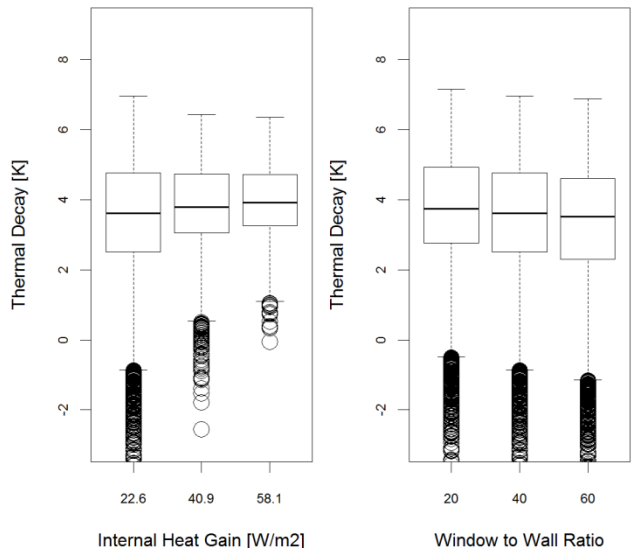


Figure 15. Box-plots of the thermal decay versus the internal heat load condition and versus the zones window-to-wall ratio.

The two plots shown in Figure 15 illustrate the thermal decay as a function of variations in internal heat gains and window-to-wall ratio, respectively. This shows that the median values are not very sensitive to either of these two parameters. However, the low internal load condition shows wider distribution than the high internal load condition. Although not shown, the number of hours when there is a heat loss instead of heat gain (i.e., negative thermal decay) turns out to be relatively large in the low internal load condition compared to the other cases.

3.6. Recommendations to control thermal decay

In UFAD buildings under cooling operation, one of the most commonly observed outcomes resulting directly from plenum thermal decay is the following scenario: building operators frequently decrease the supply air temperature entering the plenum (in the interior zone) to offset thermal decay and provide adequate cooling in the perimeter zone. However, if this supply temperature reset is not done carefully, overly cool diffuser discharge temperatures in the interior zone will increase the likelihood of occupant complaints of overcooling. Based on ongoing research and field experience, the following recommendations are provided to improve the control of thermal decay in UFAD systems.

1. Try to provide the coolest supply air into perimeter zones, allowing thermal decay to warm up the plenum air as it flows into the interior zone. Possible strategies to accomplish this include:
    a. Use ductwork (flexible, textile, or rigid) to deliver air to/towards the perimeter
    b. Direct plenum inlets with higher velocity toward the perimeter
    c. Consider placing primary inlet locations (shafts) in the perimeter, if possible
2. Increasing the overall airflow rate will reduce thermal decay, although there is a tradeoff with increased fan energy.
3. On larger floor plates (>2,300 m^2), consider adding plenum dividers to create more plenum control zones.

4. LIMITATIONS OF STUDY

The main limitations of this study are related to the selection of the cases to be simulated and to the variables investigated. We did not test the influence of the different types of slab and raised floor construction, room height,

raised floor height and building aspect ratio. The interaction with a secondary cooling system, e.g., radiant panels or TABS (thermally activated building systems), was not investigated. Validation through field measurements was not done. Since it is not coupled with CFD (computational fluid dynamics) analysis, the assumption of the uniform air temperature distribution in the underfloor plenum is made, i.e., the three-dimensional temperature distribution in the plenum is not taken into account. The thermal decay values reported in this paper are not weighted by the floor area that they represent. The values of the variables selected for the reference case (Case 1) have an unbalanced influence on the average thermal decay values. In order to have a balanced influence, all possible combinations of variables should be simulated. This was beyond the scope of the paper.

5. CONCLUSIONS

The main conclusions of the analysis of thermal decay in the supply plenum of underfloor air distribution systems are summarized below.

- The air temperature rise in the plenum, also named thermal decay, is considerable (annual median=3.7 K, and 50% of the values between 2.4 and 4.7 K during the occupied hours based on the annual simulations).
- In existing implementations of UFAD systems some degree of thermal decay is inevitable and has an impact on performance, increasing both the supply airflow rate and the fan energy. Simulation results from the idealized comparison discussed in this paper show how thermal decay results from a complex interaction of system thermal elements and how this affects performance. Although a ducted system as simulated is not one that would be implemented in practice, the comparison indicates the potential for improved performance if thermal decay could be completely eliminated and justifies efforts to minimize its impact (see Recommendations to Control Thermal Decay below).
- The primary "source" annual HVAC energy use for typical series plenum configurations with thermal decay) is greater than that of an idealized ducted system (no thermal decay); chiller and fan energy is increased by 23% and 10%, respectively. The parallel and series configurations have similar primary annual HVAC energy consumption; i.e., there is little difference in their performance.
- The energy savings associated with the elimination of thermal decay result from lowering airflow and the associated impact on cooling due to the operation of an airside economizer.
- The series configuration shows the highest median temperature rise among the three different plenum configurations investigated. Thermal decay in summer is higher than in winter and it depends on the climate. The ground floor has less temperature rise compared to middle and top floors. Decreasing the air handler supply air temperature causes thermal decay to increase, but also leads to reduced supply airflow rates.
- The temperature rise is not significantly affected by the perimeter zone orientations, the internal heat gain and the window-to-wall ratio.

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