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Analysis on the impact of mean radiant temperature for the thermal comfort of underfloor air distribution systems

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ABSTRACT

Despite the potentially significant advantages of underfloor air distribution (UFAD) systems, the shortcomings in fundamental understanding have impeded the use of UFAD systems. A study has been carried out on the thermal stratification which is crucial to system design, energy efficient operation and comfort performance of UFAD systems with an aim of examining impact of mean radiant temperature (MRT) on thermal comfort. Clear elucidation of the benefit of UFAD systems has been shown by comparing it to the traditional overhead air distribution systems. Keeping the same level of comfortable environment in the occupied zone, UFAD systems require much higher temperature of supply air, which represents significant energy savings. The benefit of UFAD systems is more pronounced at the condition of high ceiling height building. Considerable discrepancies in thermal comfort are found on the assumption that air temperature rather than MRT is used for the evaluation of PMV. However, more rigorous analysis including the full radiation simulation does not show any significant difference in PMV distribution. The result of the full radiation simulations requires much longer simulation time but gives similar air temperature distribution and only slightly higher averaged temperature than present approaches.

1. Introduction

The design of heating, ventilating and air-conditioning (HVAC) systems for thermal comfort requires increasing attention, especially in matters arising from recent regulations and standards on ventilation [1,2]. Underfloor air distribution (UFAD) systems are innovative methods for air-conditioning that use the underfloor plenum below a raised (access) floor system to supply conditioned air directly into the occupied zone of the building, typically through floor diffusers. The United States Green Building Council (USGBC) has identified this type of HVAC system as a way to improve indoor air quality through their Leadership Energy and Environmental Design (LEED) program.

The potential advantages that UFAD systems have over traditional overhead air distribution (OH) systems are: Improved thermal comfort, improved ventilation efficiency and indoor air quality, reduced energy use, reduced life-cycle building costs, reduced floor-to-floor height in new construction, improved productivity and health and so forth [3]. Although the use of UFAD systems is still outside the experience of most of HVAC designers, the benefits attributed to UFAD systems suggest that they will continue to gain in popularity. UFAD systems have achieved considerable acceptance in Europe, South Africa, and Japan. In the late 1990s growth for raised floor installations in the US was dramatic and manufacturers predicted that 35% of new offices would use raised floors by 2004 with 50% of those using UFAD systems [4], although this rate of increase has slowed down due to the economic downturn and much reduced office construction.

Various aspects of UFAD systems have been intensively investigated by many researchers. The reported works are related to diffusers [5], energy performance [6], IAQ [7,8], design methods [9], air stratification [5,10,11], and thermal comfort [12–14]. Currently, there exists a strong need to improve the fundamental understanding of several key issues related to energy and comfort performance of UFAD systems. The control and optimization of temperatures in the occupied zone and the amount of thermal stratification is crucial to system design, energy efficient operation and comfort performance of UFAD systems. In most design situations, only air temperature is used for the evaluation of comfort performance, while the mean radiant temperature (MRT) is ignored. We expect this would lead to a considerable underestimate of the thermal comfort index, which would result in an overestimated design of higher supply air temperature (SAT). Consequently, the goal of this study is to examine the effect of MRT on the thermal comfort of UFAD systems. Also a comparison of thermal stratification between UFAD and OH systems was conducted.

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Nomenclature

A_p	projected area (m^2)
F	view factor
I	intensity of radiation
MRT	mean radiant temperature
N	total surface number
OH	overhead air distribution
PMV	predicted mean vote
Q	heat rate (W)
SAT	supply air temperature
T	temperature (K) UFAD nderfloor air distribution
Ω	solid angle (sr)

Subscripts

i,j	index
MRT	mean radiant temperature
w	wall

2. Methods

2.1. Model description

Traditionally, conditioned supply air is delivered into an occupied space through ductwork and diffusers spaced evenly in the ceiling overhead. Prior to reaching individuals within the space, the supply air is mechanically mixed, making it uniform in both temperature and pollutant distribution. Because the mixed air is uniform in temperature, there is usually no opportunity for user adjustment or control. This results in the single most common occupant complaint that the air temperature is either too hot or too cold. Compared to OH systems, UFAD systems supply conditioned air into the user's zone through diffusers located strategically in the floor. As the air moves through the room, it gains heat from occupants, computers, equipment, and lighting. It continues to move upward until it is exhausted out of the space through the return air plenum in the ceiling. This supports an overall floor-to-ceiling air flow pattern that takes advantage of the natural buoyancy produced by heat sources in the office and more efficiently removes heat loads and contaminants from the space. A more detailed description of UFAD systems is provided by Bauman [2]

A room of length 5.5 m, width 4.5 m and height 2.6 m has been considered as shown in Fig. 1. The height of the conditioned space was set at 2.6 m, which is the typical ceiling height of normal office environments. To see the clear advantage of UFAD system, we also examined the case of ceiling height 2.9 m. The conditioned cool air was discharged at two sets of diffusers of size 0.15 m by 0.15 m, located at floor (at ceiling in case of OH systems). Different combinations of supply (A–D in Fig. 1) and return diffusers (A–D in Fig. 1) were tested but showed no significant difference.

Room airflow was varied over the range of 0.8–1.6 m/s (0.144–0.288 m^3/s, which gives air flow rate as 8–16 ACH) and SATs over 14.0–18.5 °C for constant interior heat gains. Internal gains were composed of 2 occupants (157 W for each), 4 lights (24 W for each), and 2 internal electric equipments (100 W for each). Lighting was fixed on the ceiling, as represented by two small surfaces and occupants and electronic equipment were simply represented by cubes of 0.3 m × 0.3 m × 1.8 m and 0.3 m × 0.3 m × 0.3 m, respectively.

The diffusers considered in this study were rectangular jet type. Swirl diffusers which are very common in UFAD systems were excluded in this study since other parameters such as swirl angle and strength would give additional complexity to compare the thermal stratification of OH and UFAD systems, especially for the thermal comfort.

2.2. Numerical simulation

A commercial code, STAR-CD, has been employed for the present simulations. Because symmetry prevails in the vertical plane at the center of the x-axis, one-half of the elements are chosen as a computational domain. In most of the simulated cases, the computational grids of 75 × 90 × 52 appear to be reasonable for resolving complicated flow patterns. The grid independence has proven to be valid within a tolerable limit. The convergence is assumed achieved

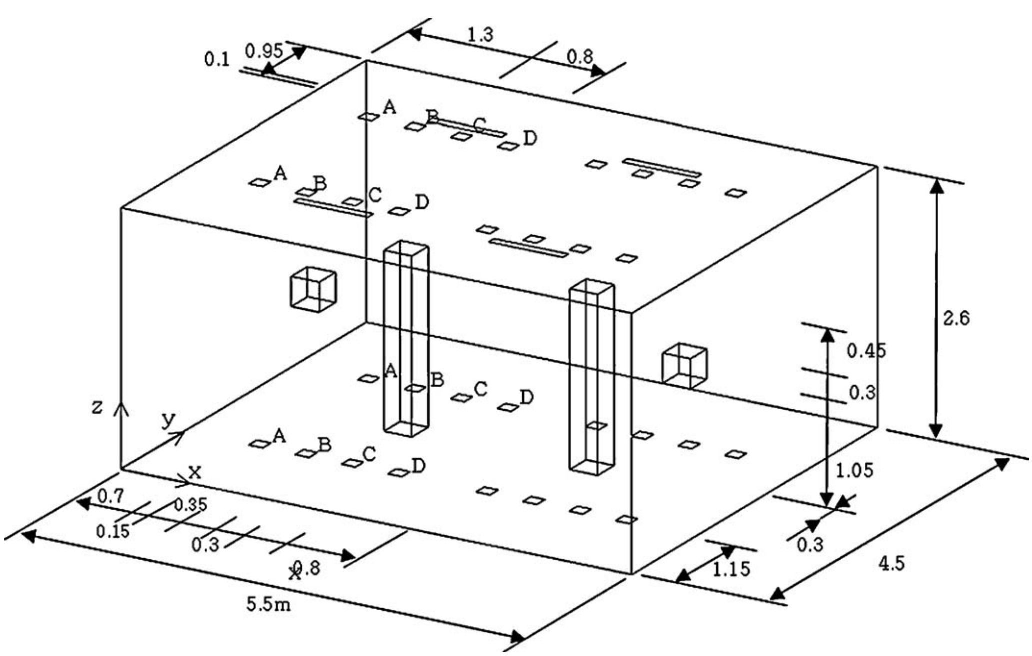

Fig. 1. Schematics of the present simulation.

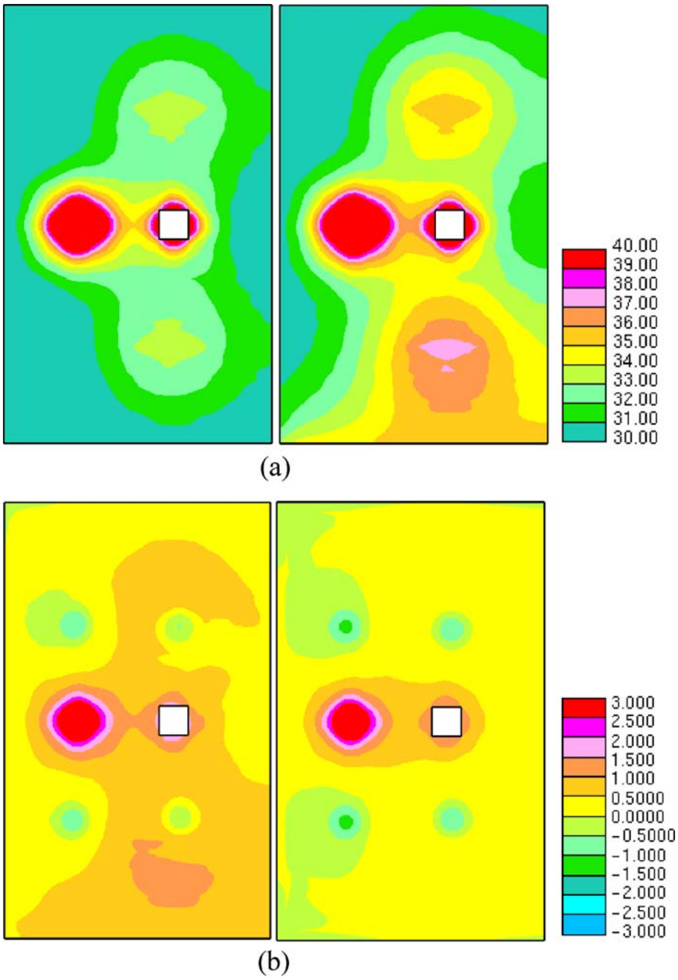

Fig. 2. Comparison of T_w = 40 °C (left) and T_w = 30 °C (right) at the breathing height of standing adults, 1.5 m on the condition of SAT = 18.5 °C, V_{in} = 1.2 m/s: (a) MRT and (b) PMV.

when the global residual is less than 10^{-3}. The computation time depends primarily on the case under consideration, but is typically about 40 h on Intel® Core™2 Quad 3 GHz.

The fluid is assumed to be Newtonian and incompressible. Since $Gr/Re^2 \sim 10$, natural convection is the dominant mode of heat transfer and the density variation with temperature, which gives rise to buoyancy force, is expressed as an ideal gas. The two-equation model κ–ε is employed for modeling of turbulence. The floor and ceiling are assumed to be adiabatic and all walls are of a fixed temperature of 30 °C, except for one wall of 40 °C due to solar radiation. Since building design nowadays would expect a sufficiently high level of insulation to prevent so much solar energy from being transmitted through the building façade, analysis including thermal conduction might be required. As shown in Fig. 2, the effect on MRT by lowering wall temperature of 40 °C to 30 °C is clearly shown near wall, but PMV is just slightly decreased with the section averaged PMV reducing from 0.57 to 0.25. In our model case, the high heat flux from the lighting and internal electric equipments is the main source of PMV distribution rather than the wall.

For the evaluation of MRT, post processing of the calculating view factor is required. A discrete transfer radiation model is adopted with 40,000 of the number of beams per patch. The satisfaction of the summation relation, $\sum_{j=1}^{N} F_{i-j} = 1$, is strictly checked. And then predicted mean vote (PMV), which is the most frequently used and best-understood model for quantitative thermal comfort analysis, is calculated according to ISO standard [15]. Here, metabolic rate is assumed to be 1.2 met (70 W/m^2) and this indicates an occupant who is quietly carrying out some clerical work in a sedentary position. The clothing value in this case is taken as 0.49 clo (0.076 m^2 °C/W), which we quite typical of office-goers in a hot and humid climate. Compared to this strict analysis, most of earlier works assumed that MRT is equal to air temperature [13,16]. We compared these two approaches and showed that MRT crucially impacts on thermal comfort.

In the above approach, radiation is indirectly considered by evaluating MRT in post processing. Full simulations of radiation coupled with fluid flow are also conducted. In these cases, the air in the room was assumed to be radiatively non-participating and the emissivities of the walls, occupants and electric equipments were set at 0.9, 0.6, and 0.6, respectively. Note that much more time resources, ~60 h, were required for these full simulations.

The validity and resolvability of the present numerical approaches were checked in the authors' earlier works [17,18]

2.3. Mean radiant temperature

Among the six primary variables to predict PMV, mean radiant temperature is of special concern in this analysis. Most of the earlier works assumed that MRT is equal to air temperature [13,16]. The assumption used in earlier works is compared to the present analysis by using MRT, which will show the impact of MRT on thermal comfort analysis.

The mean radiant temperature indicates the radiant energy exchange in a room, defined as "the uniform surface temperature of an imaginary black enclosure in which the radiation from the occupant equals the radiant heat transfer in the actual non-uniform enclosure" [19].

The net radiation on an area, A_p is described as

$$Q = \int I(\Omega)\, dA_p(\Omega)\, d\Omega \tag{1}$$

This equation is a continuous summation over all the directions represented by the solid angle Ω [20,21]. The intensity and projected area in the direction Ω are represented by $I(\Omega)$ and $A_p(\Omega)$, respectively.

In addition to the full radiation analysis, post processing of the calculating view factor has also been conducted for the evaluation of MRT using Eq. (2). And then PMV is calculated according to ISO standard [15].

$$T_{j,\mathrm{MRT}} = \left[\sum_{i=1}^{N} F_{ij} T_i^4 \right]^{1/4} \tag{2}$$

2.4. Thermal comfort index

Thermal comfort is essentially a subjective response, or state of mind, where a person expresses satisfaction with the thermal environment. While it may be partially influenced by a variety of contextual and cultural factors, a person's sense of thermal comfort is primarily a result of the body's heat exchange with the environment. This is influenced by four parameters that constitute the thermal environment (air temperature, radiant temperature, humidity and air speed), and two personal parameters (clothing and activity level, or metabolic rate).

PMV is the most frequently used and best-understood model for quantitative thermal comfort analysis. It is an index that expresses the quality of the thermal environment as a mean value of the votes of a large group of persons on the ASHRAE seven-point thermal sensation scale (+3 hot, +2 warm, +1 slightly warm, 0 neutral, −1 slightly cool, −2 cool, −3 cold). PPD (Predicted Percentage Dissatisfied) is an index expressing the thermal comfort level as a percentage of thermally dissatisfied people, and is directly determined from PMV. Among the six primary variables used for PMV, the activity level and clothing value are determined by the room usage for most design situations. Additionally, the air velocity and humidity depend on the thermal distribution system for the entire building. In an individual room, the air temperature and mean radiant temperature are the only two variables over which the design engineer may have control.

ASHRAE Standard 55 [22] specifies a maximum allowable vertical air temperature difference of 3 °C between heights of 1.7 m and 0.1 m. In the case of dissatisfaction as a function of the vertical air temperature difference between head and ankles, it has been observed that PPD varies exponentially, with 2% dissatisfied at 2 °C and rises to as high as 60% at 8 °C [22]. According to the ISO comfort standard, PPD should be lower than 10% for thermal comfort. It was also suggested by ASHRAE standard 55 that the local air speed near an office worker should be controlled at or below 0.25 m/s to avoid annoyance and distraction.

3. Results

3.1. UFAD vs. OH systems

To compare UFAD systems with OH systems, we choose 3 cases of typical systems according to the locations of supply and return diffusers, i.e. supply at floor and return at ceiling level (UF_OH), supply at ceiling and return at floor level (OH_UF), both supply and return at ceiling level (OH_OH). Note that OH_UF system is not common but gives clear elucidation of the benefit of UFAD systems by comparing to UF_OH system.

Since controlling stratification is critical to maintain thermal comfort, the average temperature gradients as a function of room height are compared and depicted in Fig. 3. The base conditions are SAT = 18.5 °C and V_{in} = 1.2 m/s. In contrast to the UFAD system showing overall floor-to-ceiling airflow pattern, the cases of OH systems (OH_OH or OH_UF) require more energy to push cooler air down into the user zone because warmed air naturally rises. Thus, air needs to be cooler than the UFAD system to overcome the hot air barrier so that it reaches users at the necessary comfort level. Keeping the same level of comfortable environment in the occupied zone, the OH_UF system requires SAT = 14.2 °C and the OH_OH system requires SAT = 15.0 °C, which are much lower temperature compared to the UFAD system (SAT = 18.5 °C). Because

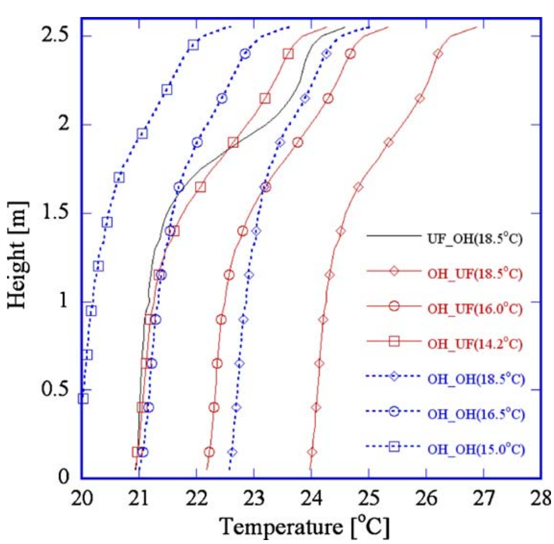

Fig. 3. Vertical section averaged temperature profiles for the three typical air distribution systems according to supply air temperature.

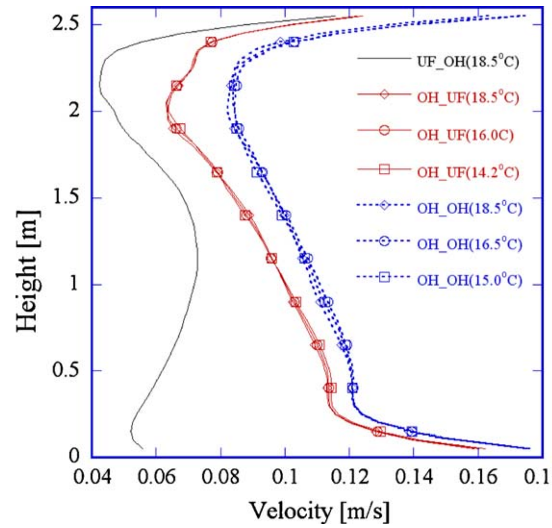

Fig. 4. Vertical section averaged velocity profiles for the three typical air distribution systems according to supply air temperature.

UFAD systems supply higher temperature air than conventional HVAC systems, UFAD systems increase the opportunity to use outdoor air for free cooling via economizer systems under suitable climatic conditions. Example calculations for a building in San Francisco reported that there are 2200 additional hours of free cooling per year via an economizer with UFAD system vs. conventional air distribution system [23]. The increased supply air temperature can also improve the coefficient of performance of air-conditioning systems. Akimoto et al. [24] pointed out that UFAD systems required 34% of energy less than OH systems, both with outdoor air-cooling. Also, Matsunawa et al. [25] have performed a case study on a “Smart” building in Tokyo and reported the benefit of the UFAD systems in the point of energy saving.

Another striking feature of Fig. 3 is that the temperature profiles increase or decrease with the change in supply air temperature, but retain approximately the same shape, i.e. stratification, for each system of OH_UF and OH_OH [5]. The UFAD system shows the same trend. Since there is little effect of varying SAT on the stratification, the key determinant of stratification level would be the airflow. After determining the airflow for the required stratification level, the only variable left that needs to be determined is the SAT required to meet the thermal load in the space.

Vertical averaged velocities for these three cases are shown in Fig. 4. For all cases considered, ASHRAE standard 55 is satisfied but the UFAD system is the quietest compared to the others. Note that SAT does not give any discernible effect on the velocity distribution.

Overall room air stratification is primarily driven by the balance of room airflow rate in relation to the room cooling load. As room airflow is reduced for constant heat input, stratification will increase, creating uncomfortable conditions. On the other hand, if too much air is delivered to the space, stratification will be reduced, approaching a well-mixed room at the upper limit. Thus a minimum ventilation rate should be supplied to keep stratification within acceptable comfort ranges. Fig. 5 shows the impact of airflow in a simulated interior space of UFAD system with the base condition. At the highest flow rate of 1.6 m/s, the temperature profile exhibits only a small amount of stratification with a head–ankles temperature difference of 0.4 °C. This would represent a case where the space is being “over-aired”. On the other hand, at the lowest flow rate of 0.8 m/s, the head–ankles temperature difference has increased to 3.5 °C, exceeding the limit of 3 °C specified in ASHRAE Standard 55. For the base conditions of 1.2 m/s, a 40% reduction in airflow which represents significant energy savings keeps the

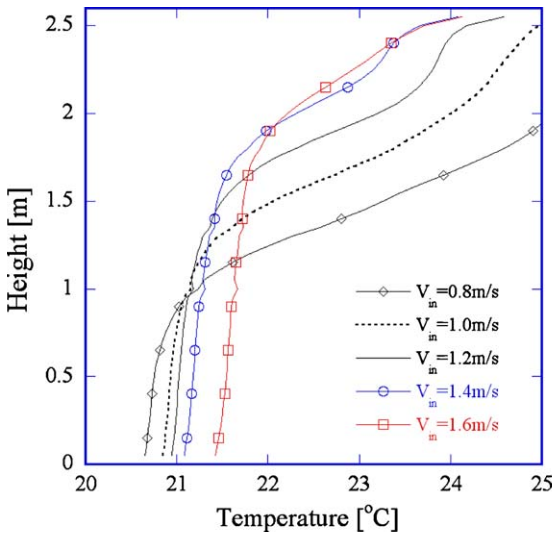

Fig. 5. Vertical section averaged temperature profiles for the UFAD system according to supply airflow.

temperature in the occupied zone within a reasonable range for comfort.

Localized distribution of conditioned air, particularly when occupants are given individual control of the incoming air, is a key component of the more flexible office arrangements required for the office of the future. Fig. 6 represents the air temperature distributions at the breathing height of standing adults, 1.5 m. In Fig. 6(a), the UFAD system shows localized distribution of conditioned air near the supply diffusers, which gives more freedom for the control near the supply diffuser. However, the OH_UF system with the same SAT of the UFAD system (Fig. 6(b)) has well-mixed distribution and this trend is conspicuous even for the Fig. 6(c) although the averaged air temperatures are the same with Fig. 6(a). The result of full radiation simulations (Fig. 6(d)) gives similar air temperature distribution of Fig. 6(a) but slightly higher averaged temperature (~0.48 °C). The prominent impact of radiation is not shown in air temperature distribution except for this temperature profile shift.

As residents do not occupy the whole space but only up to a certain height level, a cool and well-mixed environment is only needed within this occupied region. From this point of view, UFAD systems are expected to promote the benefits of thermal stratification for higher ceiling buildings since air is supplied directly into the occupied zone near floor level. The results of higher ceilings, 2.9 m, support this argument. In Fig. 7, contrary to the UFAD systems, OH systems for the higher ceiling cases such as OH_UF and OH_OH systems show considerable temperature difference in the occupied zone compared to the corresponding lower ceiling cases. Also, the stratification becomes weak as ceilings get higher.

3.2. Effect of mean radiant temperature

Illustrated in Fig. 8 are (a) air temperature, (b) difference of MRT and air temperature, and (c) PMV at the breathing height of standing adults, 1.5 m. Three cases of UF_OH (left), OH_UF (center) and OH_OH (right) are compared on the base conditions, i.e. SAT = 18.5 °C and V_{in} = 1.2 m/s. The section averaged values, i.e. area-averaged, of air temperature and PMV are examined. Apart from the lower value of section averaged air temperature of the UFAD system (~21.42 °C) compared to the OH_UF (~24.54 °C) and the OH_OH (~23.05 °C) systems, the UFAD system results in localized distribution of conditioned air which gives more freedom for the control over the local thermal environment. The section averaged PMVs are 0.57, 1.29, 0.77, respectively. Unlike to the UFAD

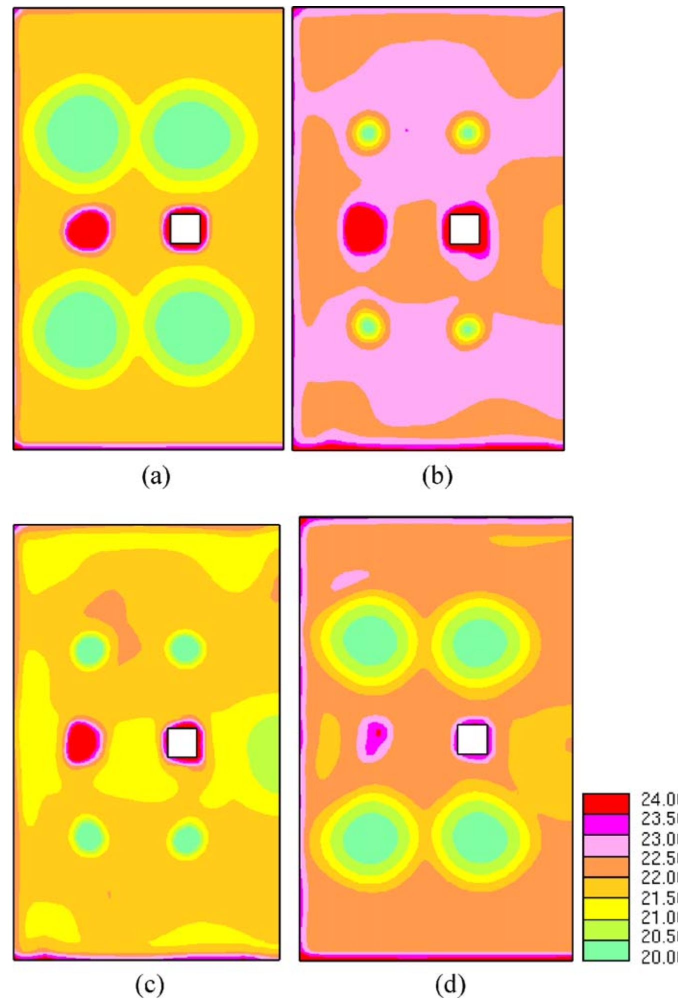

Fig. 6. Air temperature distribution at the breathing height of standing adults, 1.5 m for (a) UF_OH system (SAT = 18.5 °C), (b) OH_UF system (SAT = 18.5 °C), (c) OH_UF system (SAT = 14.2 °C), and (d) UF_OH system (SAT = 18.5 °C) with full radiation simulation.

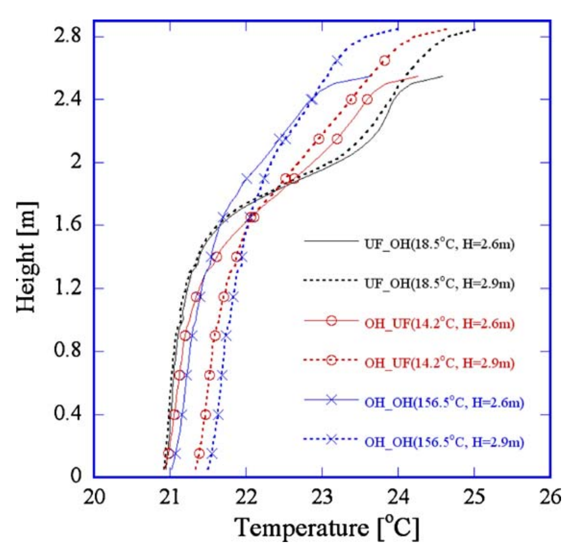

Fig. 7. Effect of ceiling height on the thermal stratification for the three typical air distribution systems.

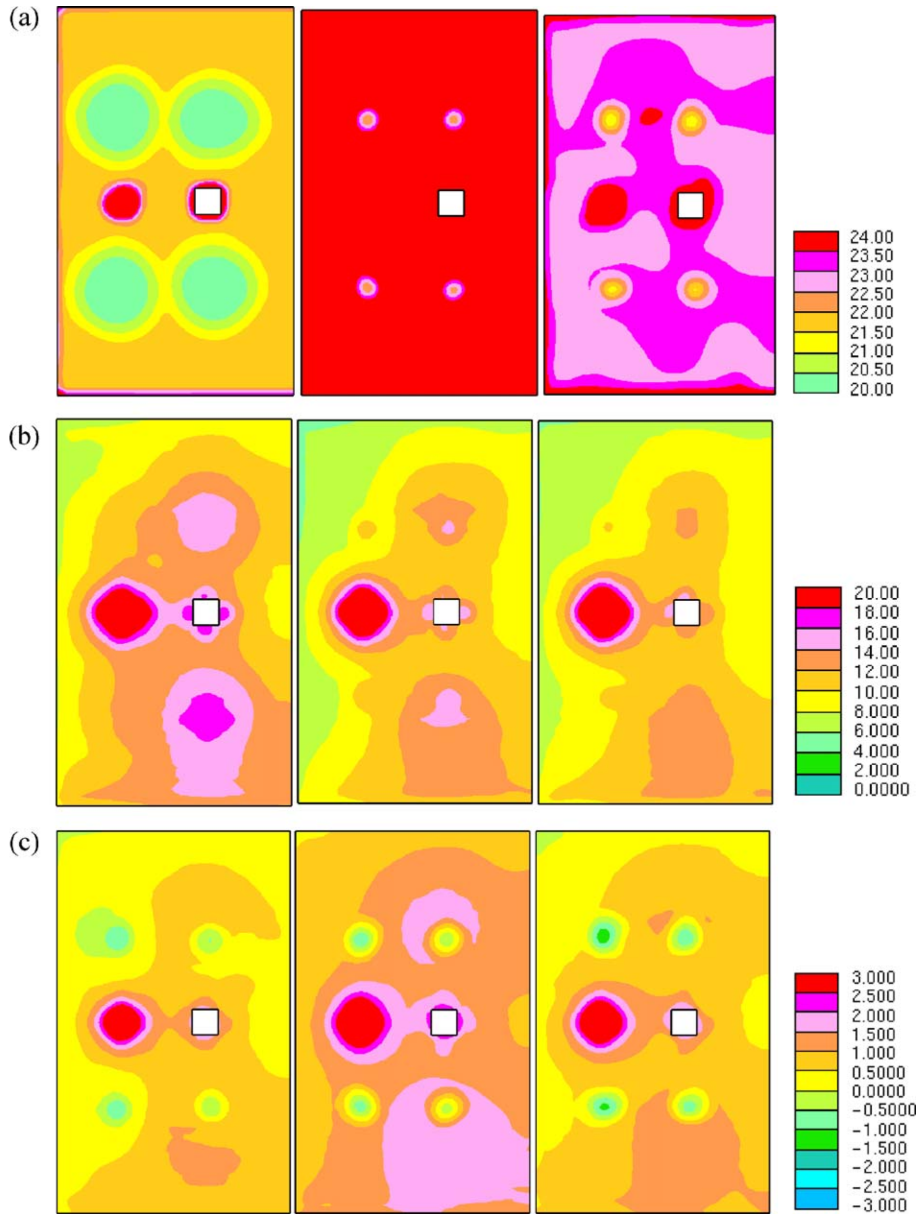

Fig. 8. Comparison of UFAD system (left) to OH systems such as OH_UF (center) and OH_OH (right) at the breathing height of standing adults, 1.5 m on the condition of SAT = 18.5 °C, V_{in} = 1.2 m/s: (a) air temperature, (b) difference of MRT and air temperature, and (c) PMV.

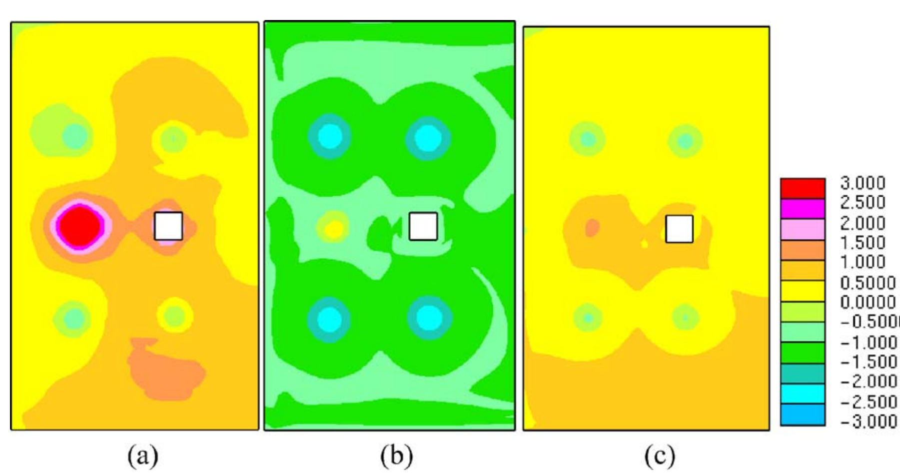

Fig. 9. Effect of mean radiant temperature on thermal comfort for the UFAD system on the condition of SAT = 18.5 °C, V_{in} = 1.2 m/s: (a) PMV distribution evaluated by using MRT, (b) PMV distribution evaluated by using air temperature, and (c) PMV distribution evaluated by using MRT in case of full radiation simulation.

system and the OH_OH system, the OH_UF system shows "slightly warm" condition of thermal comfort. Air temperature of the OH_OH system is much higher than that of the UFAD system but the difference of MRT and air temperature of the OH_OH system is much lower than that of the UFAD system. This combined effect results in the PMV of the OH_OH system becomes slightly higher than that of the UFAD system.

The MRT distribution is far from the air temperature distribution as shown in Fig. 8(b). The section averaged difference of MRT and air temperature is about 11 °C, thus considerable discrepancies of thermal comfort would be expected if air temperature rather than MRT is used for the evaluation of PMV. The PMV distribution which is evaluated by using MRT (Fig. 9(a)) shows substantial deviation from Fig. 9(b) which is based on the assumption that air temperature is used for the evaluation of PMV. Thus "neutral" condition of thermal comfort (section averaged PMV ∼ 0.57) is distorted to "slightly cool" condition of thermal comfort (section averaged PMV ∼ −1.11) if air temperature is used to evaluate PMV as done in earlier works. However, more rigorous analysis including the full radiation simulation does not show any significant difference in PMV distribution (Fig. 9(c), section averaged PMV ∼ 0.42). The full radiation simulation eliminates the unrealistic high temperature of the surface of electric equipment which is located just below the hot spot in Fig. 9(a). This results in flattening PMV distribution.

4. Conclusions

The UFAD systems have been compared to the OH systems by choosing 3 cases of typical systems according to the locations of supply and return diffusers, i.e. supply at floor and return at ceiling level (UF_OH), supply at ceiling and return at floor level (OH_UF), both supply and return at ceiling level (OH_OH). The lower value of section averaged air temperature of UFAD system (∼21.42 °C) compared to OH_UF (∼24.54 °C) and OH_OH (∼23.05 °C) systems is found at the breathing height of standing adults, 1.5 m. Also the UFAD system shows the benefit of the localized distribution of conditioned air which gives more freedom for the control over the local thermal environment. The section averaged PMVs of UF_OH, OH_UF and OH_OH systems are 0.57, 1.29, and 0.77, respectively. The OH_UF system shows a "slightly warm" condition of thermal comfort and the UFAD system shows the most comfortable thermal condition.

Keeping the same level of comfortable environment in the occupied zone, the OH_UF system requires SAT = 14.2 °C and the OH_OH system requires SAT = 15.0 °C, which are much lower temperatures compared to the UFAD system (SAT = 16.5 °C). This represents significant energy savings. Although the temperature profiles increase or decrease with the change in SAT, the shape, i.e. stratification, is found to remain approximately the same for each system of OH_UF and OH_OH.

The benefit of UFAD systems is more pronounced in buildings with high ceilings. Results reveal that contrary to the UFAD system, the OH systems such as OH_UF and OH_OH systems show considerable temperature difference in the occupied zone according to the ceiling height.

The MRT is far from the air temperature. The section averaged difference of MRT and air temperature is about 22 °C, thus considerable discrepancies of thermal comfort would be expected if air temperature rather than MRT was used for the evaluation of PMV. The "neutral" condition of thermal comfort (section averaged PMV ∼ 0.57) is distorted to "slightly cool" condition of thermal comfort (section averaged PMV ∼ −1.11) if air temperature is used to evaluate PMV as done in earlier works. However, more rigorous analysis including the full radiation simulation does not show any significant difference in PMV distribution (section averaged PMV ∼ 0.42). The result of full radiation simulation requires much longer simulation time but gives similar air temperature distribution and only slightly higher averaged temperature (∼0.48 °C).

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