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Thermal Comfort Assessment of Underfloor vs. Overhead Air Distribution System

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Abstract: This study presents numerical simulations of thermal comfort and contaminant removal in an air conditioned office room with two typical air distribution systems: under-floor and overhead. Well-known finite volume method was employed in 3D and 2nd order upwind scheme was used to discretize governing equations. SIMPLE algorithm was used for coupling velocity and pressure fields. Then, set of algebraic equations were solved in an iterative manner using ADI approach. After evaluating the utilized numerical approach by comparison of its results with the experimental data taken from relative literature, the study was expanded to reveal the effect of inlet position (Under-Floor Air Distribution (UFAD) system, angle of air inlet (Over Head Air Distribution (OHAD) system and comparison between two systems. Velocity, temperature, relative humidity, contaminant concentration, thermal sensation, Predicted Mean Vote (PMV) and contaminant removal factor was computed and used for assessing thermal comfort and contaminant removal characteristics of the office room in each under-investigation case. Results suggest that better thermal sensation and contaminant removal effectiveness may be obtained if typical air inlet diffuser setup used in both air distribution systems. Both simulated air distribution system, showed comparable thermal sensation. PMV was close to comfort zone in both configurations. The UFAD system had some advantages to OHAD system, especially in contaminant removal effectiveness.

Key words: UFAD, OHAD, Thermal comfort, contaminant removal effectiveness, thermal assessment

INTRODUCTION

Appropriate design of air conditioning systems to provide thermal comfort and contaminant removal requires comprehensive understanding of air flow pattern, thermal parameters, relative humidity and contaminant particles distribution in ventilated spaces. Sensibility of the subject becomes more significant in some special cases such as operating rooms and industrial clean ones where preparation of a healthy and comfortable environment for patients, workers and visitors. Due to the importance of the subject, there are fairly large amount of studies in the literature that focus on thermal comfort and contaminant control in closed spaces such as clean rooms (Wang *et al*., 2008), tropical buildings (Kwong *et al.,* 2009), tropical enclosed giant freshwater prawn (Zolkefli *et al*., 2011) and paint shop (Ismail *et al*., 2011). Also, air conditioning and ventilation has vital role in livestock production industry and there are some investigations in this issue (Almuhanna *et al*., 2011; Purswell *et al.,* 2008).

The rapid developments of Computational Fluid Dynamics (CFD) in the recent years have opened the possibilities of low-cost yet effective method for improving HVAC systems in design phase with less experiment required. This effective method has been utilized in some HVAC studies (Memeledje *et al*., 2011; Bhaskoro and Gilani, 2011; Serir *et al*., 2011; Wen *et al*., 2008; Wang *et al*., 2009).

Within the last few years, Under-Floor Air Distribution (UFAD) systems have become popular design alternatives to Conventional Air Distribution (CAD) such as Over Head Air Distribution systems (OHAD) for thermal and ventilation control (Webster *et al*., 2002). In comparison to classic overhead systems that deliver air at low velocities, typical UFAD systems deliver air through floor diffusers with higher supply air velocities (Webster *et al*., 2002). The UFAD systems can have significant impacts on room air stratification and thermal comfort in occupied zone.

Halza (2003) introduced the advantages of UFAD system improved air quality, lower life-cycle costs, as well as overhead system: better comfort, lower capital cost.

Webster *et al*. (2002) presented a series of full-scale laboratory experiment to determine room air stratification for a variety of design and operating parameters. Fukao (2002) carried a comprehensive experimental study on field

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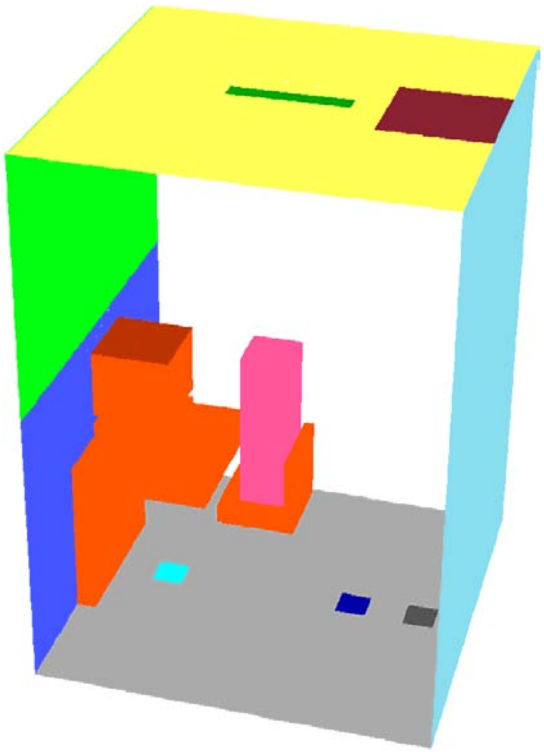

Fig. 1: 3D model of office cubicle with UFAD system

variables in an actual large-scale office building. Bauman (1999) offered a work presenting a discussion about several advantages shown by the UFAD systems.

Although there was some researchers who tried to simulate thermal performance and Indoor Air quality (IAQ) of office rooms, their results fell far from experimental data because of 2D nature of their simulation (Ho, 2004). In this study, results of numerical simulation for a full 3D office cubicle with actual heat loads are presented. Simulation includes UFAD and OHAD systems with various arrangement and angle of air inlet diffuser. Comparison between two system thermal performance and IAQ and characteristic of each individual system is presented and discussed in detail.

CFD model setup and geometry: A cubicle in a large office floor was modeled as a cubical region with dimensions of 2×2.7×2 m. Two air distribution systems were considered in the present investigation: under-floor air distribution and overhead air distribution. Office cubicle includes a desk, a personal computer (CPU and monitor) placed on it, a person sitting on a chair and lights on ceiling. Right symmetry boundary lays on backside of the person at half of the walkway. The air return outlet is

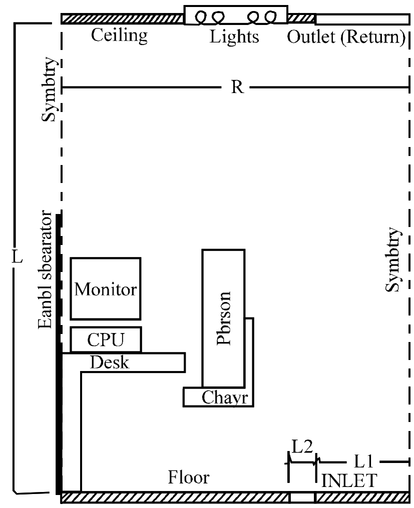

Fig. 2: 2D section of office cubicle with UFAD system

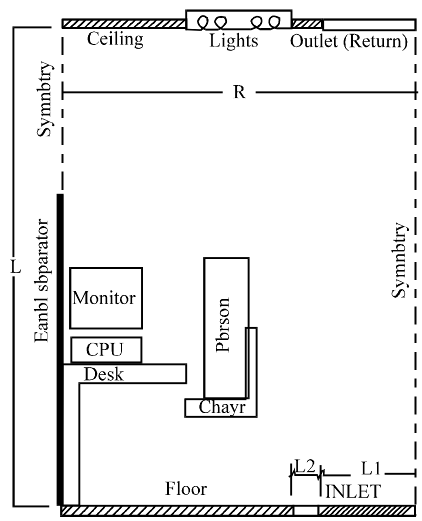

Fig. 3: 2D section of office cubicle with OHAD system

located on the ceiling above that region for both cases. Due to symmetric geometry of the office cubicle ,only one half of its modeled. This reduces required computer resources and computational time. In Fig. 1, a 3D model of simulated office cubicle is illustrated. Figure. 2, 3 show 2D section of modeled office cubicle for UFAD and OHAD systems, respectively

Two configurations are identical expect than the location of the inlet diffuser and angle of air through it.

In the UFAD system air enters the office cubicle normal to the floor plane. In this situation three , different location of inlet diffuser are modeled. In the OHAD system, only angel of inlet diffuser is varied (Table 1).

Table 1: Inlet setup and boundary conditions

Air distribution system	Under-floor			Overhead		
Inlet width (L2)	0.16 m			0.2 m		
	Inlet location (L1)			Inlet angle (A1)		
	0.54 m	1.54 m	0.18 m	30°	45°	60°
Varying parameter	Typical	Under desk	Face outlet	Typical	-	-

Mesh generation: A suitable structured hexahedron mesh was generated for each under-investigation case. To achieve this, office cubicle is divided to several cubical sub-regions. Mesh element are smaller next to high load objects like CPU and lights. This helps better capturing of thermal boundary layer in such areas. Also, to ensure that solution is mesh independent, some mesh independence studies has performed by using several finer and coarser grids. Finally, it has been concluded that a mesh with 225300 cubical elements is the best choice for this investigation.

Governing equations: Governing equations of the present investigation are:

$$\nabla u = 0 \qquad (1)$$

$$u.\nabla u = -\nabla p+\mu\nabla^2 u+\rho g\beta(T-T_{ref}) \qquad (2)$$

$$\rho c_p u.\nabla T = k\nabla^2 T \qquad (3)$$

$$u.\nabla m_1 = D_1\nabla^2 m_1 \qquad (4)$$

$$u.\nabla m_2 = D_1\nabla^2 m_2 \qquad (5)$$

Above equations represent continuity, momentum, energy, water vapor transport and contaminant particles transport equations, respectively. In these equations, buoyancy force term arising from density variation is included by means of the Boussinesq approximation based on the assumptions that variation in fluid density affects only the buoyancy term and fluid density is a function of temperature and concentration only (last term in [2]). For a typical office room, the value of Rayleigh dimensionless number is of order of 10 to 100. Therefore, the effect of natural convection cannot neglected but flow is not buoyancy dominated and natural convection has a little contribution in room ventilation.

For most HVAC applications, species concentrations are very small such that the dependency of buoyancy term on them can be neglected. It's also noticeable that thermal diffusion effect (Soret effect) is negligible in the scalar transport Eq. 4 and 5.

Because of Reynolds number in the office cubicle is about 10000, it is necessary to employ an appropriate turbulence model. For that, RNG turbulence model is used in this simulation and its equations read:

$$\frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_i}(\alpha_k k \mu_{eff}\frac{\partial k}{\partial x_i}) + G_k + G_b - \rho\varepsilon \qquad (6)$$

$$\frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_i}(\alpha_\varepsilon \varepsilon \mu_{eff}\frac{\partial \varepsilon}{\partial x_i}) + C_{1\varepsilon}\frac{\varepsilon}{k}(G_k + C_{3\varepsilon}G_b) - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} - R_\varepsilon \qquad (7)$$

where, G_k is generation of turbulence kinetic energy due to the mean velocity gradients, G_b shows generation of turbulence kinetic energy due to buoyancy and the quantities $\alpha\kappa$ and $\alpha\varepsilon$ are the inverse effective Prandtl numbers for κ and ε, respectively.

Solution approach and Boundary conditions: Employing appropriate boundary condition is a very important procedure in simulating HVAC problem. In this, study all boundary conditions are taken from experimental study of Fukao (2002). Boundary conditions can be divided into three categories: Hydrodynamic, Thermal and scalar. For hydrodynamic boundary condition, no slip boundary condition is applied to all solid walls. Drichlet boundary condition is set at inlet diffusers while pressure outlet boundary condition is used for outlet.

Thermal boundary conditions have an important influence on ventilation and thermal comfort and must be set carefully to reflect actual thermal loadings of the room. There are only two surfaces in the room that release heat at constant flux: Lights and CPU. Person is modeled as a constant temperature. Air enters the room through diffusers with a constant temperature (20°C for UFAD system and 18°C for OHAD system). Air enters the room with relative humidity of 65%. The only humidity resource in the room is the person. It has been assumed that contaminant vapors are releasing from the rug with a small constant flux of 10^{-6}.

Due to presence of buoyancy terms in momentum equation, momentum and energy equation are coupled and must be solved simultaneously. For that, well-known Finite volume method employed to discretize the governing equations and SIMPLE algorithm was used for pressure-velocity coupling. Then, set of obtained algebraic system of equations was solved in an iterative manner. In order to achieve better results, an adaptive grid approach based on local gradients of velocity and temperature was employed. The convergence criteria for all solution variables were set to at least 10^{-6}.

Table 2: Comparison between numerical results and experimental data

Parameter	System UFAD	System OHAD	Experimental results UFAD	Experimental results OHAD	Maximum relative error (%)
Air speed (m sec^{-1})	0.139	0.201	0.12	0.19	13.60
Temperature (°C)	21.97	22.300	24.40	24.90	9.95
Relative humidity (%)	67.73	72.000	60.00	65.00	9.70

Numerical results validation: To evaluate the numerical simulation validity, numerical simulation results for mean value of velocity, temperature and RH of the office cubicle were compared with experimental data of Fukao (2002). Comparison between numerical and experimental data is given in Table 2.

One can conclude from Table 2 that a very good agreement exists between them and their difference not exceeds than 15% in the worst case. Hence, developed numerical model can simulate the room correctly and accurately.

RESULTS AND DISCUSSION

Velocity, temperature, relative humidity and contaminant concentration obtained from CFD simulation was used to compute and evaluate thermal sensation, Predicted Mean Vote (PMV) and contaminant removal factor. Vector plot of velocity in the mid-plane section of typical setup of UFAD system is shown in Fig. 4.

Two circulation areas are observable in Fig. 4: Some areas above the person and CPU due to natural convection and another weaker one blow the desk. Contour plot of temperature in mid-plane section of above case is depicted in Fig 5. In this Fig. 5, warmer regions around thermal loading sources and thermal stratification in room are obvious.

From the primary parameters: temperature, water vapor concentration and pressure, relative humidity can be computed by using the procedure recommended in Ashrae (2010) which is summarized as follows:

$$\Phi = \frac{P_w}{P_{ws}} \tag{8}$$

$$P_w = \frac{(101325 + p)m_1}{0.62198 + 0.37802m_1}$$

$$P_{ws} = 1000 \ \exp\Big[-\frac{5.8\times10^3}{T+273.15} - 5.516 - 4.864\times10^{-2} (T+273.15) + 4.176\times10^{-5} \ (T+273.15)^2 - 1.445\times10^{-8} (T+273.15)^3 + 6.546 \ \ln(T+273.15)\Big] \tag{9}$$

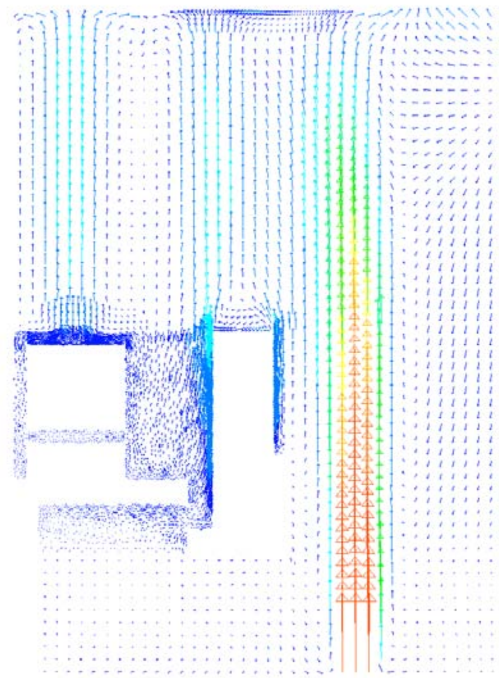

Fig. 4: Velocity vectors at mid-plane of the office cubicle (UFAD-typical)

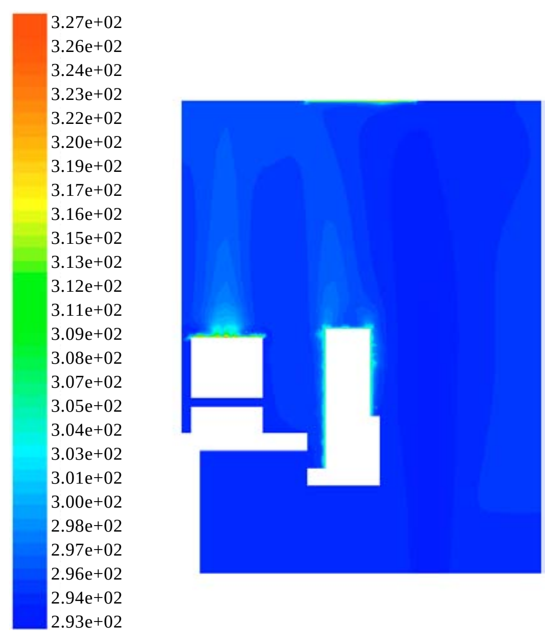

Fig. 5: Temperature contours at mid-plane of the office cubicle (UFAD-typical)

The contour plot for relative humidity in mid-plane section of the above case is given in Fig. 6.

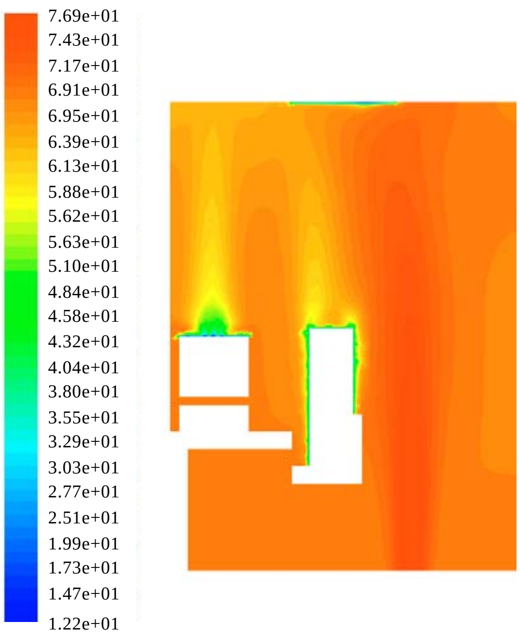

Fig. 6: Relative humidity contours at mid-plane of the office cubicle (UFAD-typical)

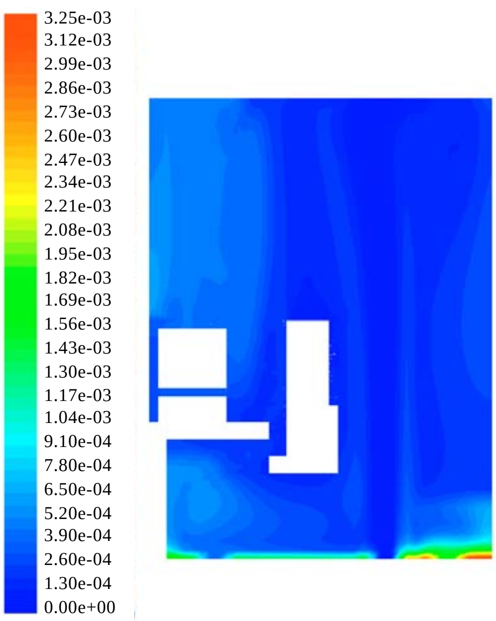

Fig. 7: Contaminant contours at mid-plane of the office cubicle (UFAD-typical)

It can be concluded from Fig. 6 that pattern of relative humidity contours is very similar to those of temperature and relative humidity is lower around the person because of high temperature in that areas.

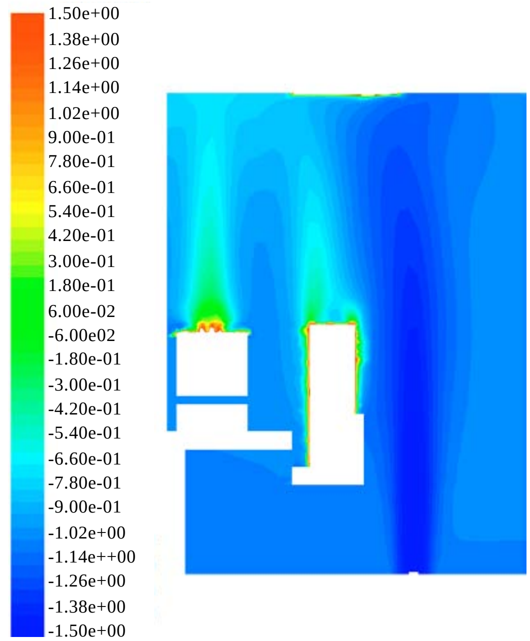

Fig. 8: Distribution of the thermal sensation index in UFAD system (typical configuration)

Contours of contaminant concentration are illustrated in Fig. 7. Although a portion of contaminants were sweep out of room by the supply air, contaminant vapors are mostly accumulated close to floor. It is interesting that by utilizing this configuration, areas around the person are almost contaminant free.

Thermal sensation index indicates the effect of environmental and personal variables on thermal response and comfort level, such as temperature, humidity, sex and length of exposure. Thermal sensation can be predicted using empirical equations from the work of Rohles and Nevins (1971). The empirical equation for men and women combined with exposure period of 3 h, conversed for SI units, is given by:

$$Y = 0.243T + 0000278P_W - 6.802 \tag{10}$$

Thermal sensation index distribution in the room cubicle is depicted in Fig. 8.

Through analyzing of the Fig. 8 one can conclude that in UFAD system, foots are cooler but in the despite, heads are slightly warmer. The reverse situation occurs with OHAD ventilation system. Fortunately all areas in vicinity of the person completely fall into the thermal comfort zone (Thermal sensations scale between -1 and 1).

Table 3: Comparison of contaminant removal effectiveness

	UFAD system			OHAD system		
Simulation	Typical	Under desk	Face outlet	30°	45°	60°
CRE	0.325	0.385	0.395	0.285	0.295	0.4

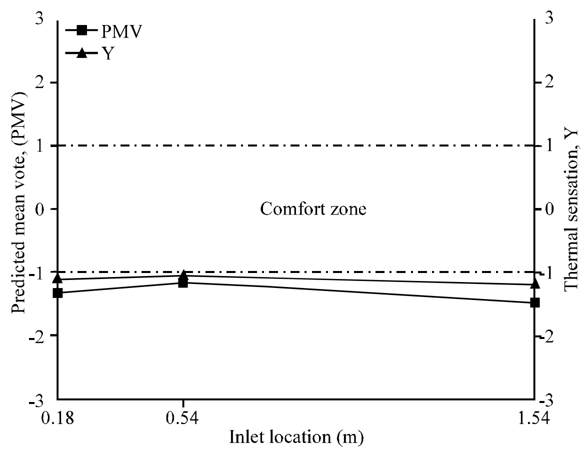

Fig. 9: Distribution of the thermal sensation index in UFAD system (typical configuration)

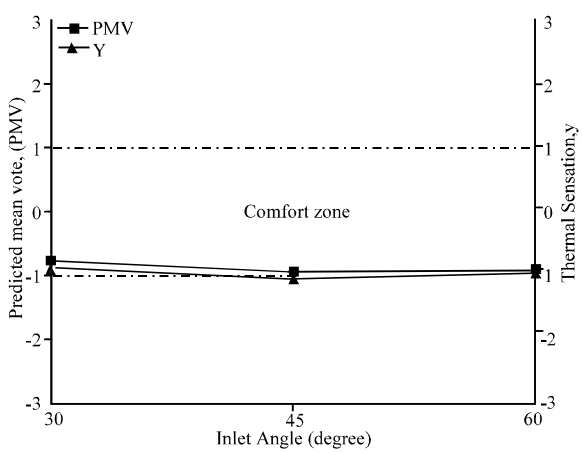

Fig. 10: Thermal comfort vs. Inlet location for OHAD system

Predicted Mean Vote (PMV) is a parameter for assessing thermal comfort in an occupied zone based on the conditions of metabolic rate, clothing, air speed besides temperature and humidity. For formulation and detailed discussion around PMV consult work of Fanger (1970).

Figure 9 depicts PMV and thermal sensation as inlet location changes for the under-floor system model. The higher values of PMV and thermal sensation appear the when the inlet is located under the seat.

Variation of PMV and thermal sensation as inlet angle changes for the overhead system model is illustrated in Fig. 10. In OHAD system, PMV is entirely inside the comfort zone but closer to lower limits. In an actual office cubicle there are some extra thermal loading sources which not taken into account in the numerical model. So it's predic Table 3 that in an actual office room, PMV and thermal sensation index will entirely lay inside the comfort zone.

It can be postponed, from Fig. 9 and 10, both of the UFAD an OHAD system satisfy thermal comfort condition and the typical setup is the best choice in each air distribution system.

For assessing the effectiveness of an occupied zone, the Contaminant Removal Effectiveness (CRE) is used. The CRE was determined based on the mean contaminant Concentration in the supply inlet, in the exhaust outlet and in the occupied zone (Gadgil *et al.*, 1999):

$$CRE = \frac{C_{E}-C_{S}}{C_{BZ}-C_{S}} \tag{11}$$

where, C_E indicates the mean concentration in exhaust;, C_s is the mean concentration in supply air and C_{bz} is the mean concentration in occupied zone (up to 1.8 m). Assuming that the supply airflow is contaminant-free, the contaminant removal effectiveness from Eq. 10 can be computed as:

$$CRE = \frac{C_{E}}{C_{BZ}} \tag{12}$$

Contaminant removal effectiveness of all under investigation cases is given in Table 3.

By comparing contaminant removal factor of two systems one can conclude that UFAD system possesses advantage of better contaminant removal effectiveness over the OHAD system. This is because in UFAD system air is supplied from the floor, it can better sweep contaminant vapors from the floor, where its concentration is very high.

CONCLUSION

Main findings of this paper are summarized as below:

- Both UFAD and OHAD systems satisfy thermal comfort condition and are identical in this respect
- For each air distribution system the typical inlet diffuser setup is the best choice

- UFAD system has some advantages over OHAD system, especially in contaminant removal factor

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