Journal of Building Engineering xxx (2016) xxx-xxx

Contents lists available at ScienceDirect

Journal of Building Engineering

journal homepage: www.elsevier.com

CFD investigation of airflow pattern, temperature distribution and thermal comfort of UFAD system for theater buildings applications

S.A. Nada,* H.M. El-Batsh, H.F. Elattar, N.M. Ali

Department of Mechanical Engineering, Benha Faculty of Engineering, Benha University, Benha, 13511 Qalyubia, Egypt

ARTICLE INFO

Article history:
Received 16 January 2016
Received in revised form 20 April 2016
Accepted 22 April 2016
Available online xxx

Keywords:
CFD
Air conditioning
UFAD
Parametric study
Thermal comfort
Energy saving

ABSTRACT

A 3D-CFD investigation of airflow, temperature distribution and thermal comfort in high rise ceiling theaters air conditioned with underfloor air distribution (UFAD) system is presented for different operating and geometric conditions. Numerical simulations are implemented, using a commercial CFD package (Fluent 6.3), to understand the effects of supply air temperature, supply air velocity, space height and number of supply air diffusers on the performance of the air conditioning system and thermal comfort. For UFAD system evaluation, the traditional overhead mixing air distribution (OHAD) system are also modelled and compared with the UFAD system. The results showed that (i) the used numerical technique could accurately predict the airflow and temperature distribution in the high rise conditioned space, (ii) UFAD system is capable of creating smaller vertical variations of air temperature and a more comfortable environment and energy saving than OHAD system, (iii) the supply air velocity and temperature, number of diffusers and height of the space have a significant impact on thermal comfort, (iv) the optimum system performance and thermal comfort obtained at 18 °C supply air temperature, 0.8 m/s supply air velocity and proper numbers and distributions of supply diffusers, (v) the percentage of energy saving due to using UFAD system increases with increasing the theater height. The simulation results are validated with the available experimental data and good agreement are obtained.

Nomenclature

UFAD	Underfloor air distribution system
3D	Three dimensional
CFD	Computational fluid dynamics
OHAD	Overhead air distribution system
PMV	Predicted mean vote
PPD	percent persons dissatisfied
μ	Dynamic viscosity (kg.m/s)
μ_t	Eddy viscosity (turbulence viscosity)
p	Pressure (Pa)
T	Air temperature (°C)
ρ	Air density (kg/m^3)
u	Velocity vector in x-direction
V	Velocity vector in y-direction
W	Velocity vector in z-direction
ΔT	Temperature ratio, $\Delta T=(T-T_s)/(T_{Exhaust}-T_s)$
Φ	phase function

1. Introduction

The importance of reducing building energy consumption has increased ever since global warming became a serious issue. For space heating and cooling, air distribution strategies in HVAC have a strong influence not only on indoor environmental thermal comfort but also on energy costs. Air distribution system also has a direct impact on space organization, floor height planning, interior layout and construction cost [1]. A relatively new approach of air distribution, the underfloor air distribution (UFAD) system, has been widely used in new commercial buildings. UFAD systems are mechanical air distribution systems that delivers conditioned air through grilles mounted in the floor. The air is directly supplied to the occupants' area causing occupants' plumes and zone heat load to stratify to the upper layer of the zone and then is typically exhausted from the ceiling. The space is divided into two zones, occupied zone where cold air and unoccupied zone in the upper layer where air is warm. The floor-mounted grilles are positioned so that each occupant receives his own flow of air which causes temperature stratification from the lower to the upper layer of the zone.

UFAD system is more effective when the zone height increases as in the case of theaters, hotel lobbies, showrooms, worship buildings, etc. Rapid economic growth the desire of a higher quality life has currently led a boom in construction of gymnasiums, concert halls, and theaters. Since most of these cultural facilities are composed of large spaces, they generally require a high level of dependency on mechanical ventilation with conditioned air. Furthermore, since these large cultural facilities have high ceilings, a great amount of energy could be required by traditional air distribution to maintain the optimal indoor temperature for a comfortable environment. For such facilities with high ceilings, an UFAD system would be more appropriate to enhance thermal comfort with energy saving, and would also allow both individual control of ventilation volume and distribution of air only to occupied zones. Using computational fluid dynamics (CFD) is capable for analyzing the flow pattern, temperature distri-

* Corresponding author.
Email address: samehnadar@yahoo.com (S.A. Nada)

http://dx.doi.org/10.1016/j.jobe.2016.04.008
2352-7102/

bution and thermal comfort of the air conditioning system in short span of time, which was previously impossible from experimental and theoretical methods [2]. Moreover, CFD gives virtual distribution of airflow, temperature, etc. in entire domain which is highly difficult to get from experiments because of time and cost involved. Unfortunately, there is no universal flow model to represent the entire flow pattern for the air conditioning system [3].

On introducing the UFAD systems, Bauman [4] presented discussion about several advantages shown by the UFAD system; Halza [5] compared the advantages of UFAD system and overhead ductless system; Webster [6] presented an overview of the principles, features, benefits, and limitations of the underfloor air distribution system. Woods [7] did a review by literature searching and field investigations to assess the actual performance of UFAD system in real world, showing that there are gaps in available data, and remarked on evaluation and selection approach. Webster et al. [8] presented a study about a building that operated with a UFAD system. They showed little troubleshooting with the system operation, pointing out the positive aspects of using well-designed UFAD systems. Alajmi and El-Amer [9] investigated the effectiveness of UFAD systems in commercial buildings for various types of application and different air supply temperatures in a hot climate (The State of Kuwait). The results showed that the saving of energy in using UFAD was not prejudicing on occupant comfort. It was found that the UFAD system can save up to 30% energy compared to OHAD. Xu and Niu [10] proposed a numerical procedure, based on coupling two types of modeling, CFD simulation and dynamic cooling load simulation, to predict annual energy consumption for UFAD systems. It was found that the dimensionless temperature coefficient was almost a constant, when the locations of heat sources were fixed. As compared with the mixing system, it was found that the UFAD system derives its energy saving potential from the following three factors: an extended free cooling time, a reduced ventilation load, and increased coefficients of performance (COP) for chillers. Chung et al. [11] clarified details of the thermal stratification due to UFAD, which is crucial to system design, energy efficient operation, and comfort performance, with an aim of examining the impact of mean radiant temperature (MRT) on thermal comfort.

Lin et al. [12] investigated using a numerical simulation, the effect of the air supply location on the design and performance of the displacement ventilation (DV) system. The study focused on a typical Hong Kong office under local thermal and boundary conditions. The results indicated that the supply should be located near the center of the room rather than to one side of the room. This will provide a more uniform thermal condition in the office. He also stated that the exhaust was found to have minimal effect on the thermal comfort. Lin and Linden [13] presented a simplified model of an underfloor air distribution (UFAD) system consisting of a single source of heat and a single cooling diffuser in a ventilated space. The model was based on plume theory for the heat source and a fountain model for the diffuser flow, and predicts steady-state two-layer stratification in the room. The results showed that the control parameters that affect the flow pattern are the buoyancy flux of the heat source, the volume flux and the momentum flux of the cooling diffuser. The results suggested ways to optimize UFAD design and operation. Ito and Nakahara [14] developed a simplified model to calculate the vertical space air temperature distribution in a ventilated space with an UFAD system.

Chung et al. [15] examined the effect of mean radiant temperature on the thermal comfort of UFAD systems. Also a comparison of thermal stratification between UFAD and OHAD systems was conducted. The study showed that (i) a full radiation simulation requires much longer simulation time but gives similar air temperature distribution and only slightly higher averaged temperature than present approaches, and (ii) UFAD systems require much higher temperature of supply air, which represents significant energy savings. Lin et al. [16] presented a case study to investigate the effect of partitions in an office on the performance of under floor air supply ventilation system via computational fluid dynamics. The assessment was in terms of thermal comfort and indoor air quality with the use of a validated computer model. The results indicated that the partitions may significantly affect airflow and performance of under floor air supply ventilation system and improve thermal comfort. Recently, physical and CFD modeling of data centers room air conditioned by UFAD system are presented by Nada et al. [17–20] to study the effect of the operating and geometric parameters on the cooling systems performance. The studies showed that the opening ratio of the supply diffusers, air supply velocity and location of the air conditioning unit and the supply diffusers have a significant effect on the system performance.

The above literature revealed that numerical studies succeed in predicting indoor airflow pattern and temperature distribution with good accuracy and less difficulty, and consumed time. Moreover, the numerical solutions can give more details in full study field and complete parametric study. Detailed comprehensive study on the effects of the different operating and design parameters on the airflow pattern, temperature distribution and thermal comfort in spaces of high ceiling and air conditioned with UFAD systems are not completely studied and understood. Therefore, the present work investigates numerically the effect of supply air temperature, supply air velocity, number of supply air diffusers and the ceiling height on the temperature distribution, air flow pattern and thermal comfort in a theater of high ceiling and air conditioned by using UFAD.

2. Mathematical formulation and numerical methods

2.1. Governing equations

The governing equations of fluid flow represent mathematical statements of the conservation laws of physics. The air flow considered in this study has very low velocity and the change in pressure is very small as well. Therefore, the flow is assumed to be incompressible. The three-dimensional steady flows with heat transfer continuity, momentum and energy equations can be written in Cartesian tensor notation as follows [21,22].

$$\frac{\partial U_i}{\partial x_i} = 0 \tag{1}$$

$$\rho U_i \frac{\partial U_i}{\partial x_i} = -\frac{\partial P}{\partial x_j} + \frac{\partial}{\partial x_i}\left[\mu\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right) - \rho\overline{u_i' u_j'}\right] \tag{2}$$

$$\rho c_p U_i \frac{\partial T}{\partial x_i} = \frac{\partial}{\partial x_i}\left[\lambda \frac{\partial T}{\partial x_j} - \rho c_p \overline{u_i' T'}\right] \tag{3}$$

$$p = \rho R T \tag{4}$$

where U_i and T are the time-averaged velocity and temperature. ρ, λ, p, μ and R are the density, thermal diffusivity, static pressure, viscosity and gas constant respectively. $u_i', u_j', T', -\rho\overline{u_i' u_j'}, \rho c_p \overline{u_i' T'}$ are the fluctuating velocities, temperature, the average Reynolds stresses and the turbulent heat fluxes, respectively.

For closure problem, a turbulence model is required and the choice of turbulence model will depend on many issues (i.e. flow physics, type of flow problem, and needed accuracy level, the available computational resources and the available simulation time) [23]. The Reynolds stresses, τ_{ij} in Eq. (2) must be modelled in order to close this equation, therefore Boussinesq hypothesis [24] is used as follows:

$$\tau_{ij} = -\rho\overline{u_i^{\prime}u_j^{\prime}} = \mu_t\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right) - \frac{2}{3}\rho k\delta_{ij} \tag{5}$$

The turbulent heat flux and turbulent viscosity can be found by:

$$-\rho\overline{u_i^{\prime}T^{\prime}} = \frac{\mu_t}{Pr_t}\frac{\partial T}{\partial x_i} \tag{6}$$

$$\mu_t = \rho C_\mu\frac{k^2}{\epsilon} \tag{7}$$

2.2. Realizable k-ε turbulence model

k-ε turbulence model is the most widely used model because of its applicability to wide – ranging flow problems and its lower computational demand than more complex models that are available [25]. In case of the *k-ε* turbulence models, two additional transport equations (for *k* and *ε*) are needed to solve the turbulent viscosity as a function of *k* and *ε*. In the present study, the Realizable *k-ε* turbulence model is used because its accuracy for predicting the spreading rate for both planar and round jets if it is compared with other *k-ε* models [26–28]. The modelled transport equations for *k* and *ε* are given as follows [24]:

$$\rho\frac{\partial}{\partial x_j}(ku_j) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k + G_b - \rho\epsilon - Y_M + \ldots$$

$$\rho\frac{\partial}{\partial x_j}(\epsilon u_j) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\epsilon}\right)\frac{\partial \epsilon}{\partial x_j}\right] + \rho C_1 S\epsilon - \rho C_2\frac{\epsilon^2}{k + \sqrt{\nu\epsilon}}$$

The model coefficients are $C_{1\epsilon}$=1.44, C_2=1.9, σ_κ=1.0 and σ_ϵ=1.2.

2.3. Computational methodology

CFD using FLUENT 6.3, commercial package is used in the present study to simulate three-dimensional turbulence flow of indoor air distribution in theater by using the realizable k-ε turbulence model at steady-state condition [12]. The governing equations are discretized by using the finite volume method [29]. The near-wall region was solved by using enhanced wall treatment. The pressure-velocity coupling was achieved by SIMPLE algorithm. First-order upwind discretization scheme was considered for pressure, momentum, turbulent kinetic energy, turbulent dissipation energy, and energy equations because it provides fast convergence and better results in the flow field and temperature distribution. The solution is considered converged when the normalized residual of continuity, momentum, turbulence and energy are less than 10^{-4}, 10^{-4}, 10^{-3} and 10^{-8}, respectively for determining accurate results. The Realizable k-ε turbulence model with enhanced wall functions is used in the present simulation, where y+<5 in most of the theater and with average value of 0.239. Therefore, it is concluded that the near-wall mesh resolution is acceptable in our problem to use enhanced wall function model.

2.4. Thermal comfort and comfort measuring parameters (PMV and PPD Scales)

An important role of air distribution provision is to create a comfortable thermal environment with the proper combination of comfort variables. At the design stage, an engineer needs to predict the performance of an HVAC system. The thermal comfort level is one of the most important aspects in assessing the system performance. There are six primary factors that must be addressed when defining conditions for thermal comfort [30]. The six primary factors are: metabolic rate (met), clothing insulation (Icl), air temperature, mean radiant temperature, air speed and relative humidity. All of these six factors may vary with time. However, this standard only addresses thermal comfort in steady state.

While a number of thermal comfort measuring parameters are currently available, the most common and probably best-understood parameters are the predicted mean vote (PMV) for thermal comfort and the associated percent persons dissatisfied (PPD). The PMV-index predicts the mean value of the subjective ratings of a group of people in a given environment. The PMV scale is a seven-point thermal-sensation scale. The ASHRAE thermal sensation scale, which was developed for use in quantifying people's thermal sensation, is defined as follows: +3 hot, +2 warm, +1 slightly warm, 0 neutral, −1 slightly cool, −2 cool, and −3 cold [30]. The formulae for calculating PMV can also be found in ISO 7730: 1994 [31] and was developed by Fanger [32] as given by Eq. (10):

$$\begin{aligned}\mathrm{PMV} = &\,(0.028 + 0.3033\exp(-0.036\mathrm{M}))\\ &\times \left\{\begin{array}{c}(M - W) - 0.42\{(M - W) - 58.15\} - 3.05\{5.733 - \ldots\\ -0.0173 \times M(5.867 - P_{\mathrm{v}}) - 0.0014 \times M \ldots\\ -3.96 \times 10^8 \times F_{\mathrm{cl}}\{(T_{\mathrm{cl}} + 273)^4 - (T_{\mathrm{r}} + 273)^4\} - \ldots\end{array}\right.\end{aligned}$$

where M is metabolism (W/m^2), W is external work, equals to zero for most activity (W/m^2), F_{cl} is ratio of body's surface area when fully clothed to body's surface area when nude, P_v is partial water vapor pressure (Pa), T_{cl} is clothing temperature (°C), T_r is mean radiant temperature (°C), T_a is air temperature (°C), h_c is convection heat transfer coefficient (W/m^2 K).

To predict how many people are dissatisfied in a given thermal environment, the PPD-index (Predicted Percentage of Dissatisfied) has been introduced. In the PPD-index people who vote −3, −2, +2, +3 on the PMV scale are regarded as thermally dissatisfied. The PPD is now widely used as criteria to evaluate thermal comfort.

The PPD formula can be given by the following equation [31]:

$$\mathrm{PPD} = 100 \times 95 \times \exp\left(-0.03353 \times \mathrm{PMV}^4 - 0.2179 \times \mathrm{PMV}^2\right)$$

ISO Standard 7730 recommends a PPD limit of 10% corresponding to $-0.5 \leq$ PMV$\leq +0.5$.

In practice, all these performance parameters are affected by the thermal and flow boundary conditions, including the size and geometry of the space, rates and temperatures of airflows and heat sources.

Thermal comfort degree index of body depends on the above mentioned six factors. But these six factors could not be full measured in practice. Especially on-line measurement of some of them is very difficult. So some variables can be hypothesized on the basis of practice and the basic assumptions considered in this study are:

- The people indoor usually sit or do some lighter physical labor. The metabolic rate of sitting is 58.15 W/m^2. So, this value was selected in this study (58.15 W/m^2=1 met (1% metabolic rate)) [32].
- For clothing, 0.5 clo (0.5% clothing) is the value light summer clothing, 1 clo is the value heavy summer clothing (business suit). Hot resistance of clothing of 0.6 clo (0.093 m^2 °C/W) is used [32].
- Mechanical work of body is zero.
- Average radiation temperature of indoor surface equals temperature of indoor.
- Relative humidity is 50%.

In the present study, the thermal comfort measuring parameters (PMV and PPD) are calculated from the above equations under the considered basic simplified assumptions by developing a MATLAB program. PMV and PPD are calculated as the average values based on the average calculated variables around the persons for presenting comparison studies of PMV & PPD for all ranges of the studied parameters with respect the recommended ranges of PPM and PPD.

2.5. Physical model, computational domain and boundary conditions

The studied physical model has been selected to have typical operating conditions and geometrical configuration to those found in the theaters. The selected section (i.e. repeated portion) from the total theater was chosen as a computational domain to minimize the simulation time. The section dimension is 5 m length×3 m width. According to the international building codes, the theater sections accommodate five rows of five seats. The conditioned air is delivered to the theater from supply diffusers and returned through the ceiling return diffuser. Twenty-five simulators to simulate 25 seated human bodies were placed on the seats with total heat generated of 95 W per each person according to ASHRAE standard [30]. Human bodies were simplified as rectangular cuboids of dimensions (0.4 m×0.4 m×0.9 m) as shown in Fig. 1(a). The supply air diffusers are defined as inlet uniform velocity and the return diffusers are set as pressure outlet boundary condition. All the vertical walls are defined as symmetry and adiabatic wall conditions are applied for the floor, ceiling and seats.

In the base line case study (i) the finished ceiling height is varying from 4 m at the front and 3 m at the rear, (ii) five supply diffusers are used and distributed on the floor such that one diffuser is located in the middle of each row at front of seat, (iii) one return diffuser is placed in the center of the ceiling of the studied section, (iv) circular supply diffusers of cross sectional area of 0.01 m^2 and circular return diffuser of 0.4 m diameter are considered, and (v) air is supplied to the theater at 18 °C with a velocity of 0.8 m/s. The grid generation of computational domain is described in Fig. 1(b), where unstructured tetrahedral cells are chosen [33].

The studied parameters and their ranges that considered in the current work are proposed according to the actual operating and construction ranges of the real UFAD systems for theaters application and are presented in Table 1.

3. Grid independence study and model validation

3.1. Grid independence study

Models are built for the scale model with the same dimensions mentioned in Fig. 1(a). Four different number of cells (330694, 546258, 1054085 and 1551161) are used to check the model solution convergence. The cells were increased mainly in the region of high flow and temperature gradients. The temperatures and velocity distributions inside the room were obtained along Line 1 (at x=1.5 m and z=0.2 m) as shown in Fig. 2. The total temperature and velocity magnitude distributions along Line1 for different four cell numbers are presented in Fig. 3. As shown in the figure the solutions above 546258 cells have approximately the same values, therefore the solution was considered grid independent and the grid size of 546258 cells was recommended to carry out the present study for accurate simulations and less time consumptions.

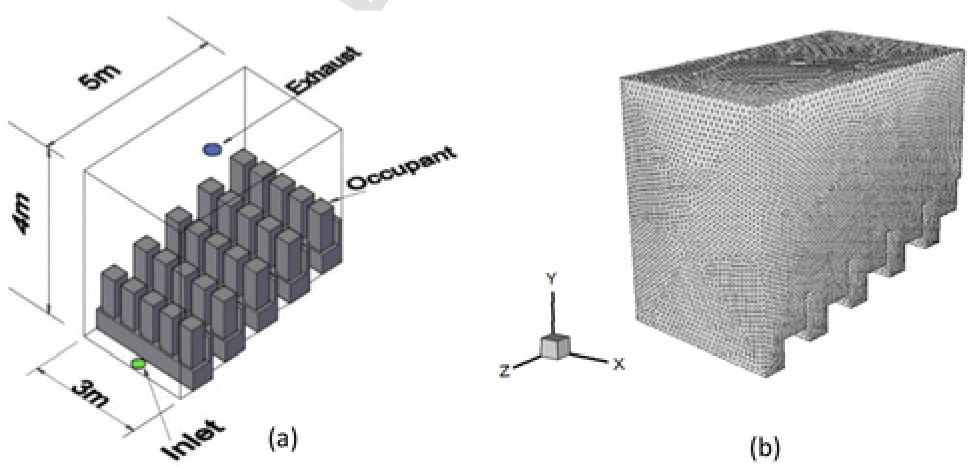


Fig. 1. (a) Computational domain, (b) Mesh with boundary conditions.

Table 1
Studied parameters and their ranges.

Studied parameter	Studied ranges
Supply air temperature, t_s	16,18, 20, 22 °C
Supply air velocity, V_s	0.4, 0.8, 1.2, 1.6 m/s
Space height, H	4, 7, 10 m.
Number of supply air diffuser, N_d	1, 2, 3.

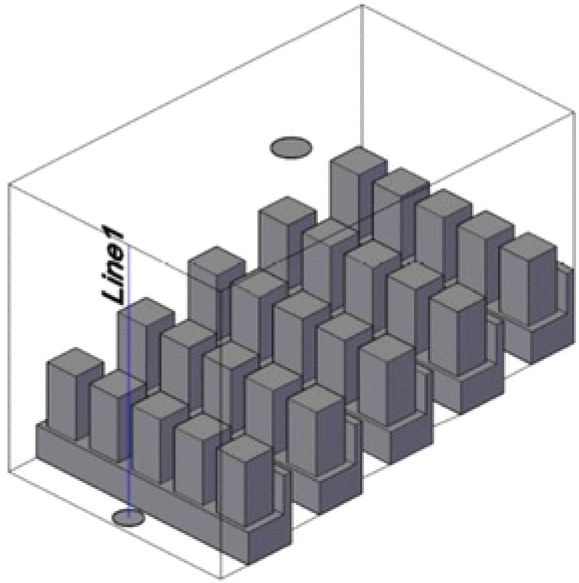


Fig. 2. Line1 location for grid dependency study.

3.2. Model validation

As per the literature review and according to the author's knowledge, no experimental measurements are available in the literature for air velocity and temperature distribution inside a stepped ground theater. Therefore, the present CFD work is validated using the experimental and numerical data reported by Zhang and Chen [34] for UFAD system inside a room.

Firstly, in order to choose the appropriate turbulent model in the present study, different numerical simulations were carried out using different turbulent models and then these results were compared to available experimental data [34]. The velocity and temperature distribution comparisons between present simulation and available experimental results on Line v5 are shown in Fig. 4(a) and (b), respectively. As shown in the figures, the realizable k-ε turbulent model provides fairly agreement model to the published experimental data in comparison to other turbulent models applied. In conclusion, the realizable k-ε turbulence model reveals sufficient agreement with the experimental data so, it is recommended to implement the present simulation.

Model validation is accomplished by comparing the velocity and temperature distributions between experimental measurements [34] and simulated computations using realizable k-ε turbulent model along four vertical lines (Line v4, Line v5, line v6 and Line v7) with different locations inside the room [34] and illustrated in Figs. 5 and 6, respectively.

As shown in the figures the present numerical results obtained from the CFD model gives good agreements with experimental and numerical results [34]. The small discrepancies between experimental and simulated results can be attributed to the simplifying assumptions in the turbulence model and also due to experimental uncertainty.

4. Results and discussion

The effects of supply air temperature, supply air velocity, number of supply diffusers and theater height on the temperature distribution, flow pattern, system performance and thermal comfort measuring parameters are firstly investigated. Then the performance of the UFAD system is compared with that of the overhead air distribution (OHAD) system.

Two middle vertical planes (Plane 1 and Plane 2) are selected as shown in Fig. 7 to show and present the results of temperature and velocity distributions along the length and width of the theater section. Also three lines (Line1, Line 2 and Line 3) along Plane2 are located at three horizontal sections at levels 0.86 m,1.26 m and 1.75 m, respectively measured from reference ground level are selected to investigate in details the temperature and velocity distributions at different levels of the occupied zone. Other addational vertical sections may also be selected at a specific location to capture a specific phenomenon during studing the effect of the different parameters on the UFAD performance. It is believed that information given by these selected sections and lines can capture all the information inside the theater.

4.1. Effect of supply air temperature

In UFAD system, air is thrown at ground level and gained heat due to thermal loads inside the room rising its temperature and moves it upwards under the action of the supplying velocity and buoyancy affects. The problem is that if the air is thrown at low temperature, it will cause a sense of coldness on people's feet which is an incon-

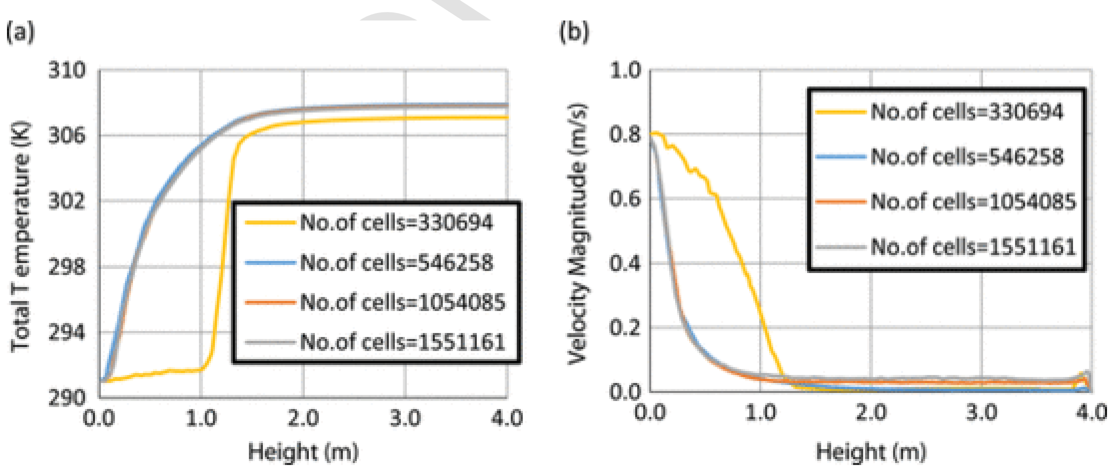


Fig. 3. Grade independence study: (a) total temperature, (b) velocity magnitude.

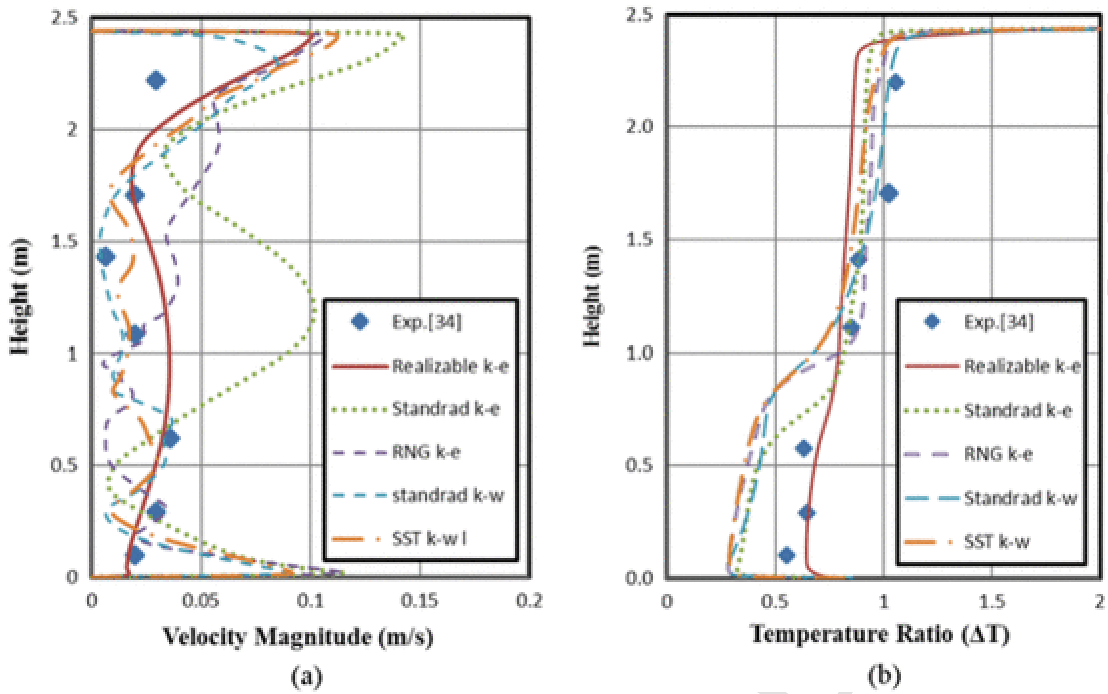


Fig. 4. Comparisons of different turbulence models with experimental. results [34] on Line v5: (a) velocity distribution, (b) temperature distribution.

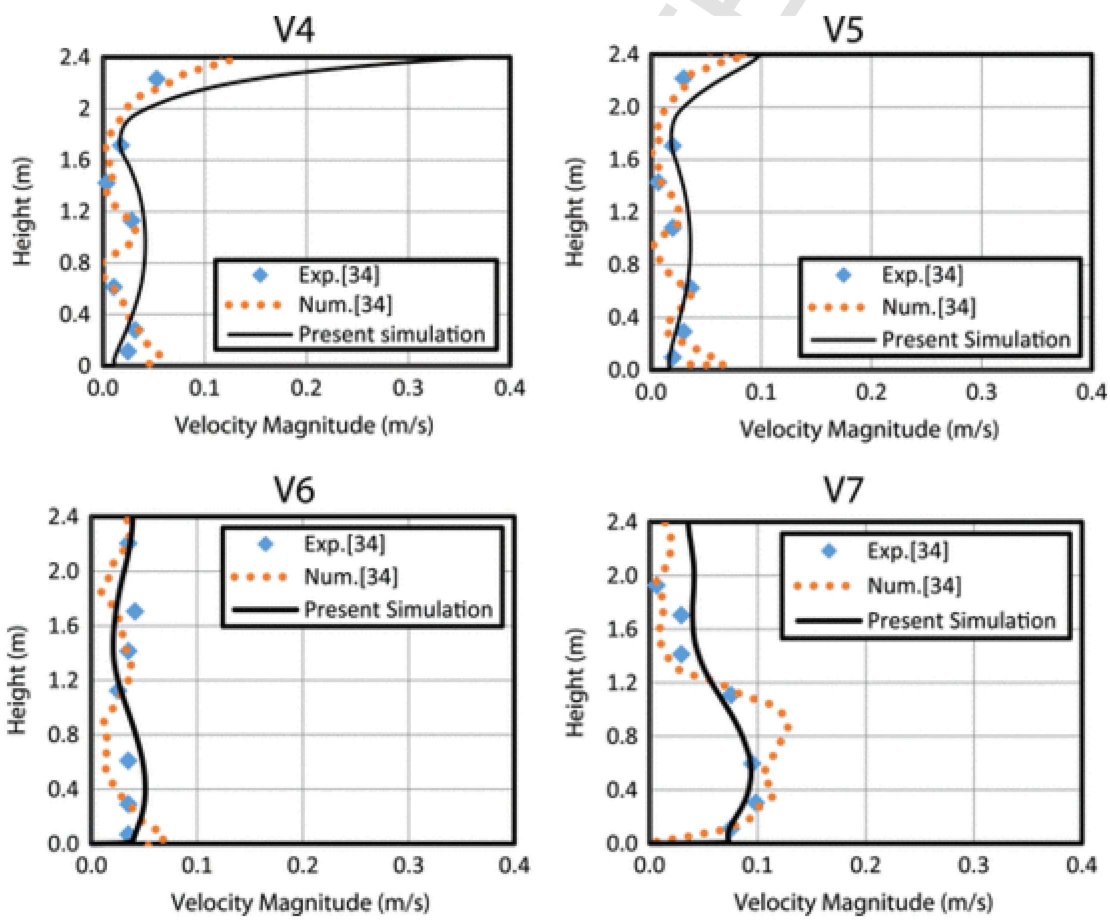


Fig. 5. Comparison of the vertical velocity profiles between simulated data and experimental results of Zhang and Chen [34].

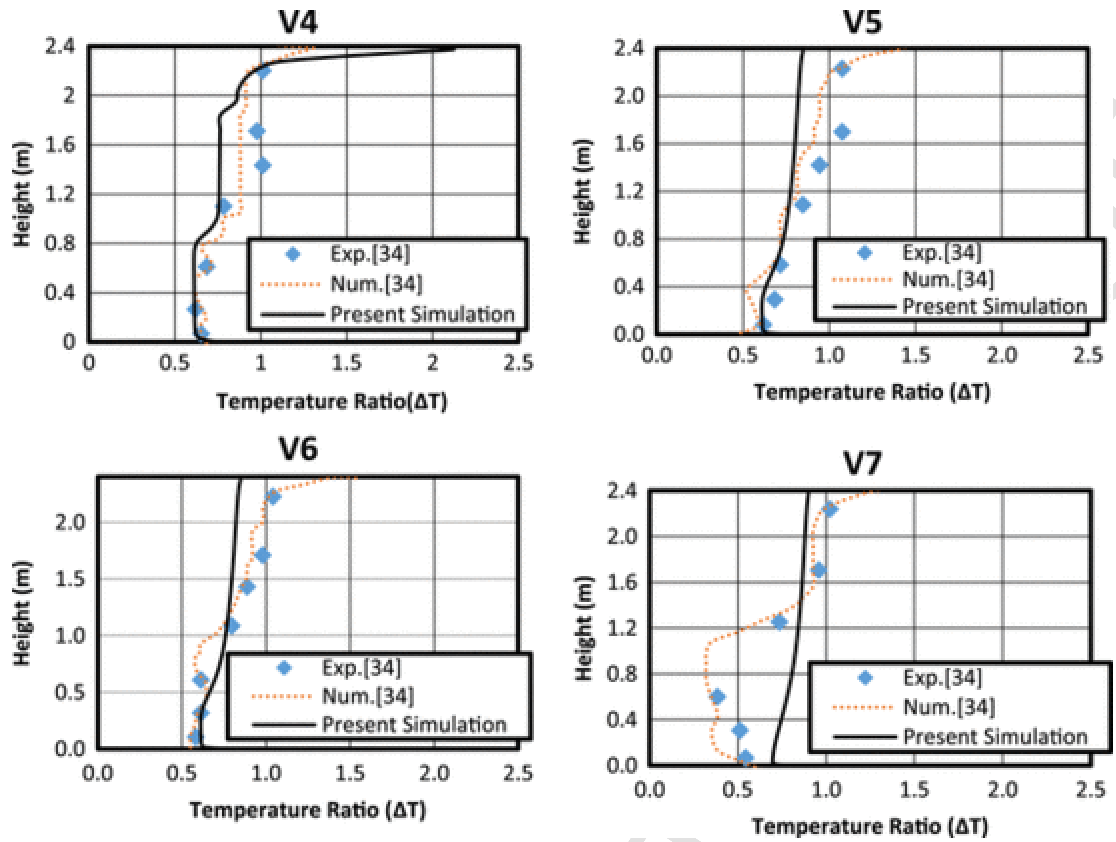


Fig. 6. Comparison of the vertical temperature profiles between simulated data and experimental results of Zhang and Chen [34].

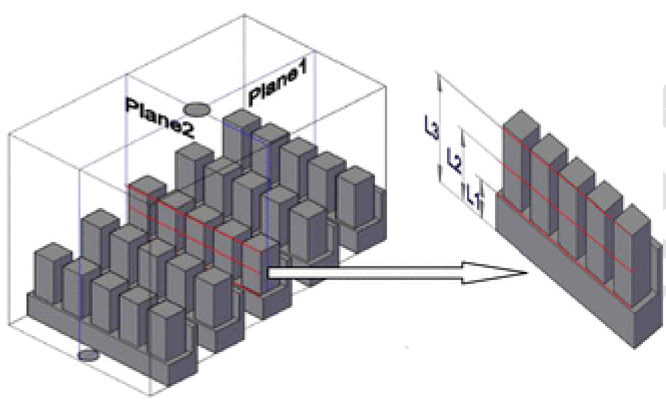


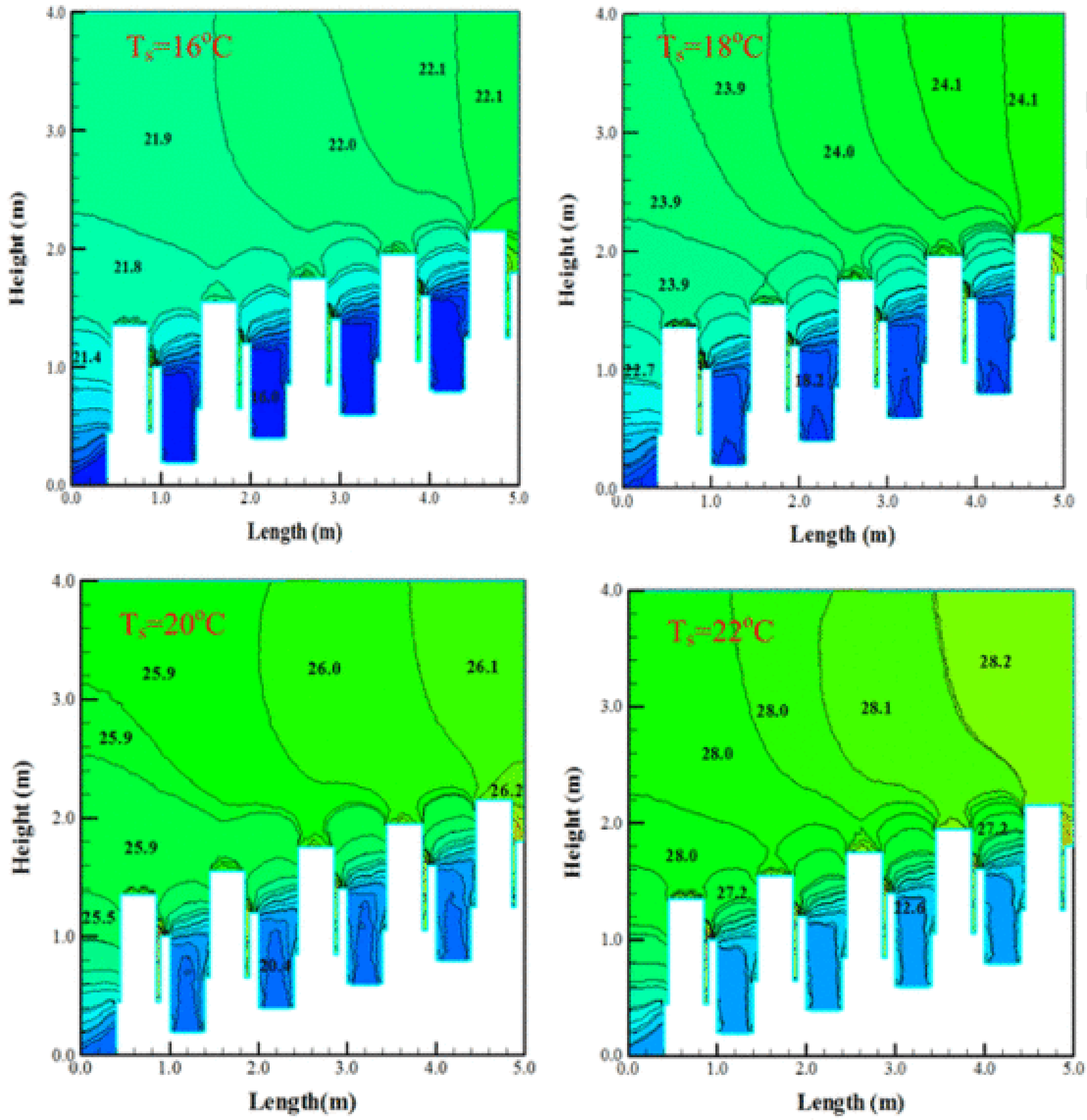
Fig. 7. Location of selected planes and lines.

venient issue. If the supply air is thrown at high temperature, it reaches people's occupied levels at high uncomfortable temperature which is also inconvenient. According to ASHRAE Standard 2000 [35], the large vertical air temperature difference between the head (1.1 m and 1.7 m above the floor at setting and stand conditions) and ankles (0.1 m above the floor) may cause discomfort. Therefore, the supply temperature should be precisely chosen to satisfy the required thermal conditions.

In order to investigate the effect of supply temperature, the numerical runs and calculations were repeated with increasing the inlet supply temperature in the range 16–22 °C by step 2 °C. The supply air velocity, number of supply diffusers and theaters height are maintained constants and equal the reference case values (V_s=0.8 m/s, H=4 m, and N_d=1). Temperature and velocity distributions over the different selected planes and lines are represented in figures and discussed. Thermal indices parameters (PMV and PPD) are also calculated from the computational results for the different supply temperatures.

Temperature distributions along the two midplanes (Plane 1 and Plane 2, see Fig. 7) are shown in Fig. 8 for different supply air temperatures. The figure shows that the temperature increases slowly with increasing the height measured from the supply level. The gradient of the temperature increase with height at the lower level where the heat source (occupants) exist is strongly higher than the gradient in the upper part of the theater. Moreover, the figure shows that the optimum supply temperature is 18 °C where the temperature in the occupied level (about 23 °C) lies within ASHRAE standard recommendation. It is also noticed from the figure that if the supply temperature increase over 18 °C, the temperature in the occupied zone increases (reaches above 25 °C) and may exceed the comfort level which recommended by ASHRAE standard. Furthermore, the figure shows that decreasing the supply temperature below 18 °C, decrease the temperature in the occupied zone below the reccomonded level by ASHRAE standard where the temperature in the occupied zone reaches to 21.5 °C when the supply temperature drops to 16 °C. Decreasing the supply temperature increases the energy consumption of the cooling system. The figure also shows that the temperature variation along the occupancy level (1.1<H<1.8) is lies within the 3 °C variation that reccomonded by previous investigations and standards [35,36].

Fig. 9 shows the velocity contours at the two midplanes (plane 1 and plane 2) for different supply air temperatures. As shown in figure, for all air supply temperatures, at the lower levels of the the-



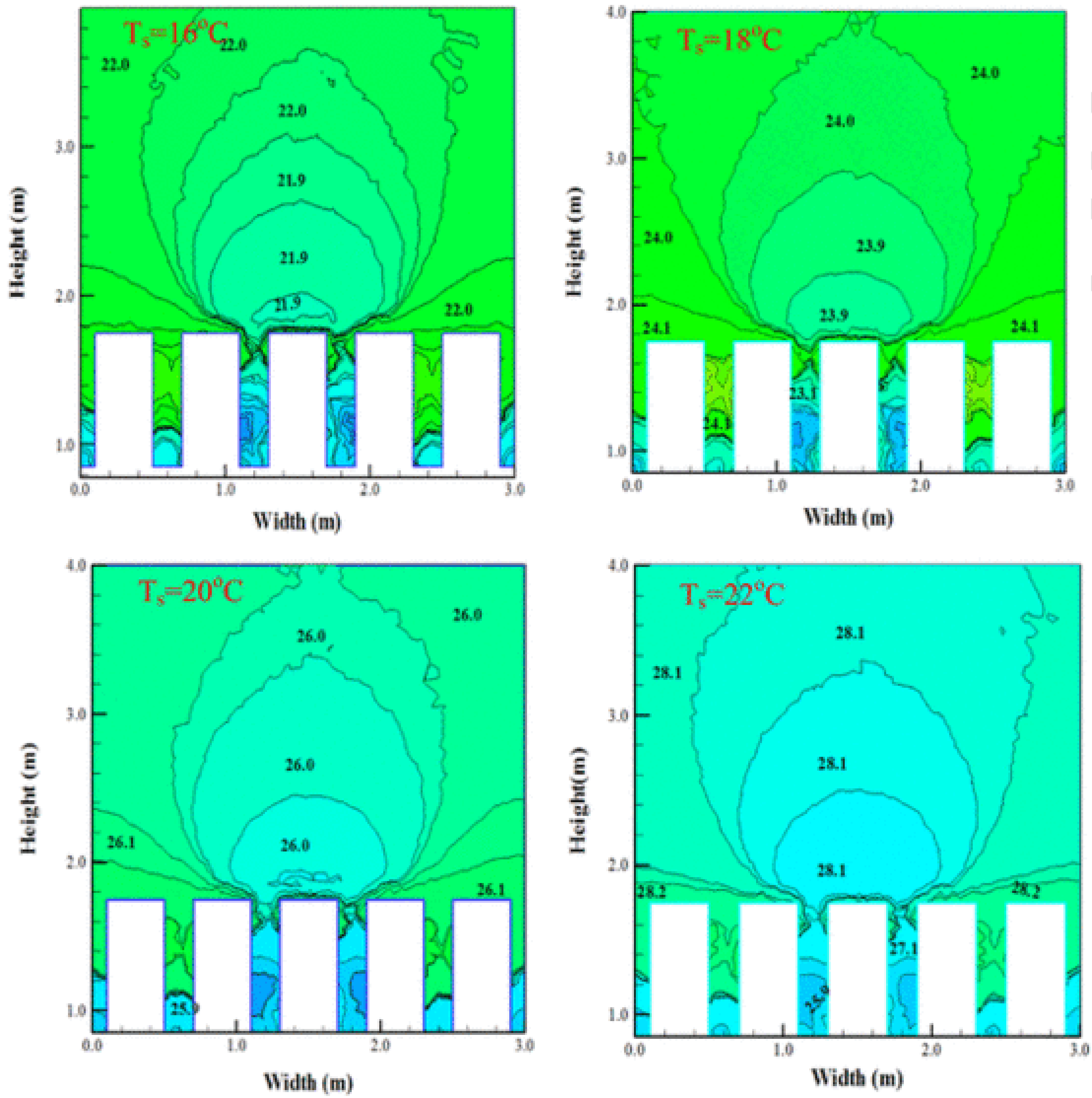
(a) plane 1

Fig. 8. Temperature contours on planes 1 and 2 at different supply temperatures.

ater height, the air velocity decreases as the height increases but at higher levels of the theater the air velocity increases as the height increase. This is attributed to that at lower levels, the flow is affected by the injection velocity. The effect of the injection velocity on the air velocity decreases with increasing the height until its effect is vanishes. At higher levels, the increase of the velocity with height can be attributed to the increase of the air temperature with height which increases the buoyancy force that increases the upward air velocity. The velocity of this upward flow increases with increasing the temperature gradient which increases with height. Also the extraction of the exhaust air from the ceiling cause extra increase of the air velocity at higher level. At higher levels (H>3 m) the air velocity dramatically increase due to the suction effect of the return diffusers.

Fig. 9 also shows that the velocity at any height increases with increasing the supply air temperature. This is attributed to the increases of the buoyancy force with increasing the supply temperature. The figure also shows that the velocity of the flow in the occupied zone is within 0.012–0.016 m/s which is within the permitted level and do not exceed the recommended comfort level given by ASHRAE (0.25 m/s) [32]. Fig. 9-a shows that the air velocity (0.7–0.75 m/s) at the lower levels (levels of feet) of plane 1 which passes by the supply air diffusers is very close to the air supply velocity (0.8 m/s) that lies outside the comfort limit. The velocity decreases as we move away from the diffuser (see Fig. 9-b). To avoid this discomfort, it is reccomonded to place the supply air diffusers in a dead zone away from persons feet.

The temperature and velocity distributions along the three lines: Line1, Line 2 and Line 3 (see Fig. 7) are shown in Fig. 10 for different supply air temperatures. As shown in the figure, at line L1 there is a strong temperature variation along the width where the temperature is minimum at width 1.5 m (location of supply air diffusers) and the temperature increases along the width as we move away from the location of the supply air diffuser. This variation of temperature along the width decreases along L2 and vanishes at higher levels along L3 where the effect of the air supply injection vanishes. Fig. 10 also shows that the temperature at any width increases with in-



(b) plane 2

Fig. 8. (*Continued*)

creasing the height from the ground where the temperature at L3 is higher than that of L2 which in turn is higher than that of L1. The figure also shows that at supply temperature of 18 °C, the temperature at L2 and L3 (occupancy levels) lies in the range 23–24 °C which is within the comfort zone while at lower (16 °C) and higher (20, 22 °C) supply temperatures, the temperature at L2 and L3 become below and above the recommended levels, respectively.

The velocity distribution in Fig. 10 shows that the supply air temperature has a negligible effect on the air velocity where the air velocity at line L1 and L2 does not affected by the supply temperature while the air velocity at L3 slightly increases with increasing the supply air temperature. The figure also shows that the air velocity decreases with increasing the height where the velocity at L1 is higher than that of L2 which in turn is higher than that of L3. This can be attributed to the vanishing of the effect of air supplying velocity on the flow field with increasing the height.

The effect of the supply air temperature on the thermal comfort indices PMV and PPD are shown in Fig. 11. As shown in the figure, the PMV and PPD for supply air temperature of 18 and 20 °C lies within the recommended limits ($-0.5 \leq$ PMV$\leq +0.5$ and PPD$<10\%$) as stipulated by ISO 7730 [32] while the PMV and PPD for supply air temperature of 16 and 22 °C lies outside the recommended limits.

Based on the figures and discussion presented in this section air supply temperature of 18°C can be considered as the best supply air temperature for UFAD system as it gives air temperature and air velocity in the occupied zone levels within the ASHRAE comfort level and at the same time gives the best values of PMV and PPD which lie within the recommended comfort zone.

4.2. Effect of supply air velocity

In ASHRAE comfort standard, the air velocity is an important factor in the calculation of thermal comfort indices. Moreover, the standard also restricts the extent to which air velocity can be used to achieve comfort by limiting it to a specified maximum level of 0.8 m/s in summer as some previous researches concluded that high supply

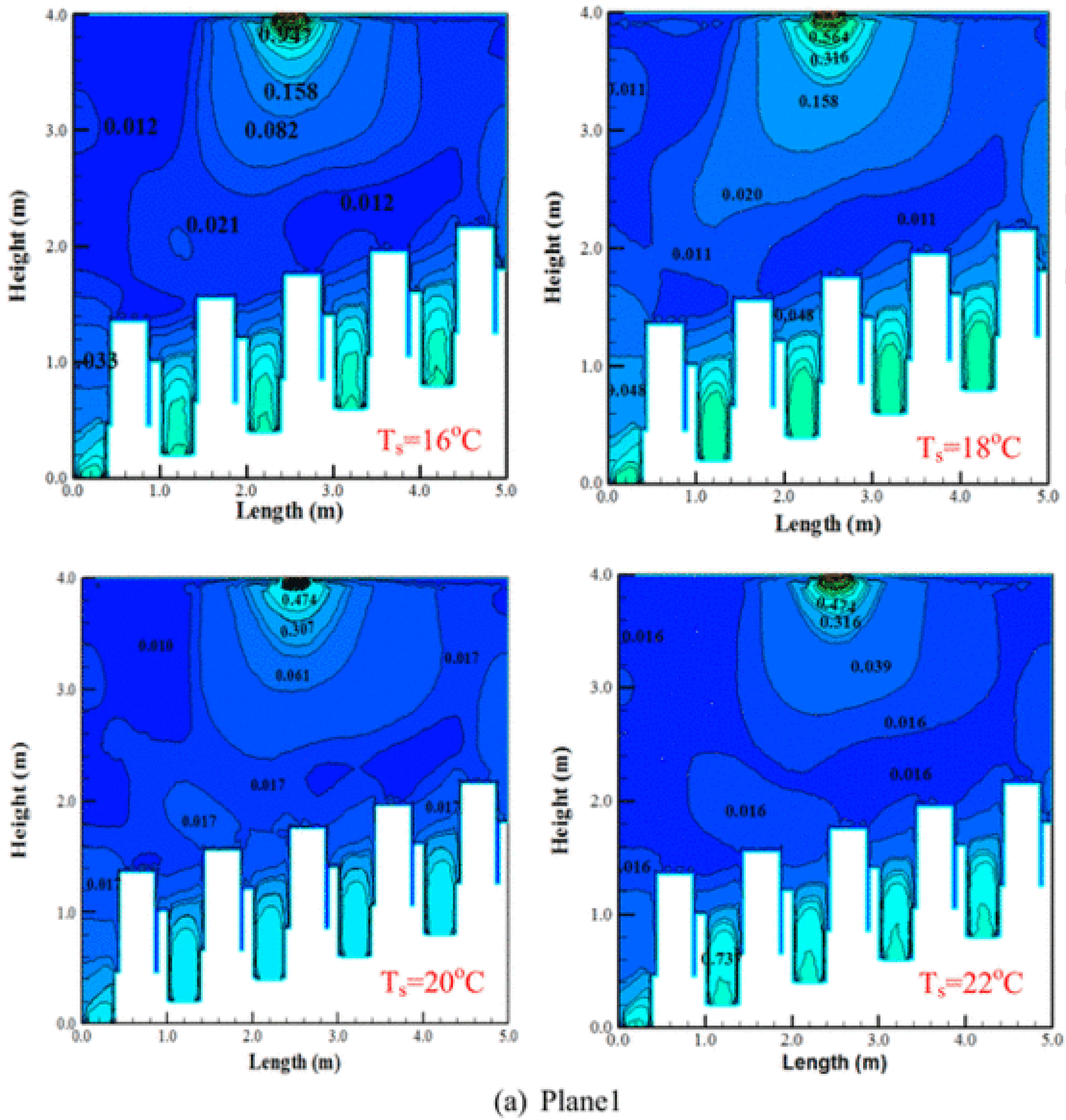


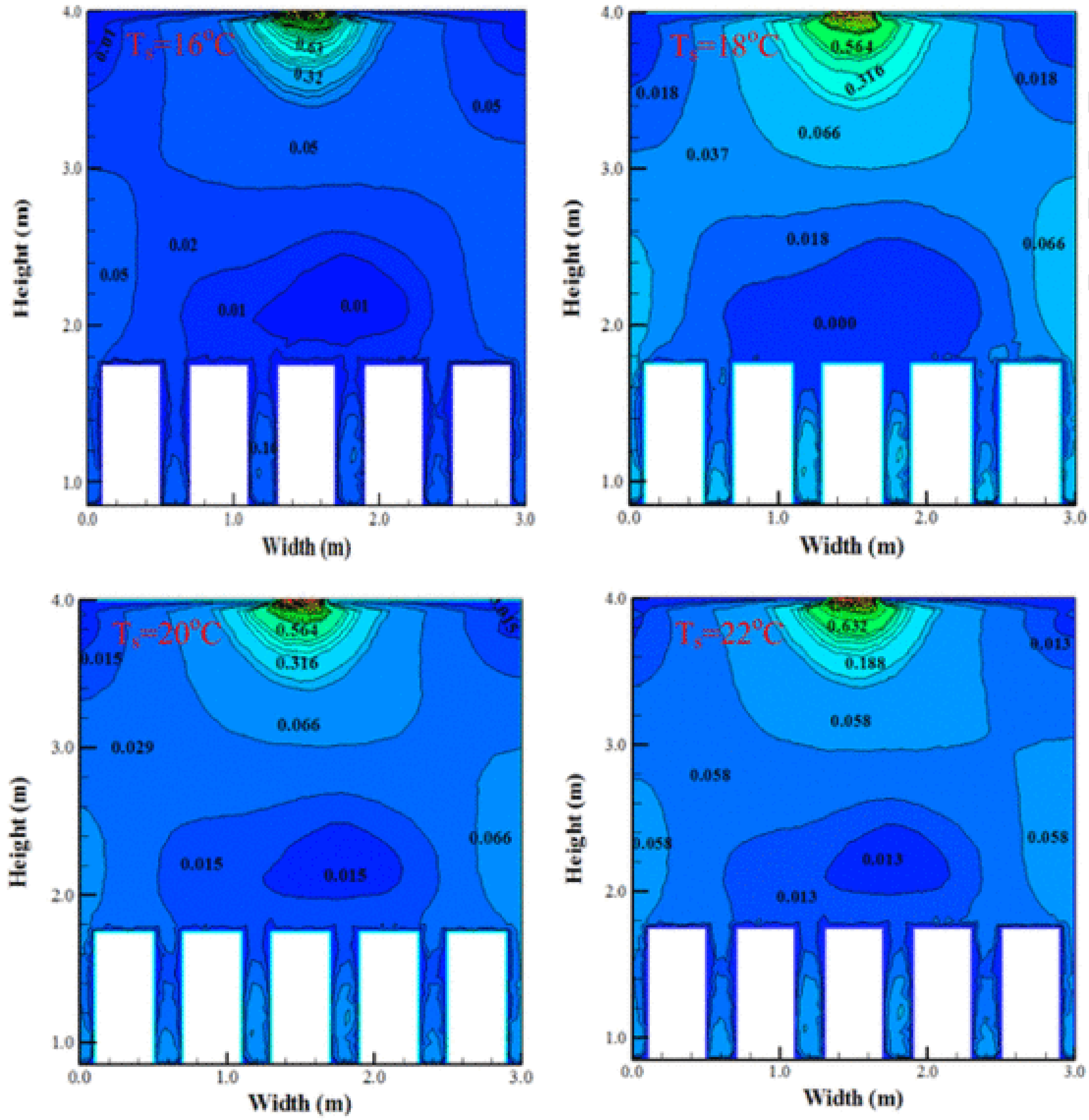
Fig. 9. Velocity contour on planes 1 and 2 at different supply temperatures.

air velocity is needed for more mixing in the occupied zone for UFAD system.

In order to study and investigate the effect of low and high supply air velocity on the performance of the UFAD system, numerical runs and calculations were conducted and repeated with increasing the air supply velocity in the range 0.4–1.6 m/s by step 0.4 m/s. The supply air temperature, number of supply diffusers and theater height are maintained constants at the reference case values (t_s=18 °C, H=4 m, and N_d=1). The temperature distributions at the two planes (see Fig. 7) are shown in Fig. 12 for different supply air velocities. The figure shows that the temperature contours at the different supply air velocities have approximately the same shape but differ in magnitude. The figure shows that the optimum supply velocity is 0.8 m/s where the temperature in the occupied level (about 23.7 °C) lies within ASHRAE standard recommendation. It is also noticed from the figure that if the supply velocity decreased or increased from 0.8 m/s, the temperature in the occupied zone increases (reaches above 25.9 °C) exceeding the comfort level. The figure also shows that the temperature variation along the occupancy level (1<H<1.8 m) lies within the 3 °C variation that recommended by previous investigations and standards [35].

Fig. 13 shows the velocity contours at the two midplanes (plane 1 and plane 2) for different supply air velocities. As shown in the figure (i) for all supply air velocities, the air velocity decreases as the height increases, (ii) the air velocity at any height increases as the supply air velocity increases, and (iii) the air velocity at the occupied height lies within the recommended values whatever the supply air velocity. Fig. 13 also shows that the air velocity at the lower levels (levels of feet) of plane 1 which passes by the supply air diffusers is very close to the air supply velocity and lies outside the comfort limit. The velocity decreases as we move away from the diffuser (see Fig. 13-b). To eliminate this discomfort, it is recommended to decrease the supply air velocity and place the supply air diffusers in a dead zone away from the persons' feet.

The temperature and velocity distributions along the three lines: Line1, Line 2 and Line 3 (see Fig. 7) are shown in Fig. 14 for dif-



(b) Plane (2)

Fig. 9. (*Continued*)

ferent supply air velocities. As shown in the figure, at line L1 there is a strong temperature variation along the width where the temperature is minimum at width 1.5 m (location of supply air diffusers) and the temperature increases along the width as we move away from the location of the supply air diffuser. This variation of temperature along the width decreases along L2 and vanishes at higher levels along L3 where the effect of the air supply injection vanishes. The trend is the same for all supply velocities. Fig. 14 also shows that at any supply air velocity, the temperature at any width increases with increasing the height from the ground where the temperature at L3 is higher than that of L2 which in turn is higher than that of L1. The figure also shows that at supply velocity of 0.8 m/s, the temperature at L2 and L3 (occupancy levels) lies in the range 22.5–24 °C which is within the comfort zone while at lower (0.4 m/s) and higher (1.2, 1.6 m/s) supply air velocities, the temperatures at L2 and L3 increases and reaches to 26 °C which is higher than the recommended levels. The velocity distribution in Fig. 14 shows that the maximum air velocity in the occupied zones (L1 and L2) for any supply air velocity does not exceed 0.2 m/s which lies in the comfort recommendations.

The effect of the supply air velocity on the thermal comfort indices PMV and PPD are shown in Fig. 15. As shown in the figure, the cases of high supply velocity are not suitable for the occupied zones with UFAD system as it has a higher values of PMV and PPD as compared to the cases of lower velocities. At high supply air velocities (over 1.2 m/s) the values of PPD become higher than the permitted level while at lower supply air velocities the values do not exceed the comfort level as they are lower than the 10% recommended by ASHRAE. Fig. 15 also shows that the values of PMV and PPD for the case of V_s=0.8 m/s are the closest values to the permitted level of comfort ($-0.5\leq$ PMV$\leq+0.5$ and PPD<10%) as stipulated by ISO 7730 [32] while the PMV and PPD for supply air velocity over 1.2 m/s lie outside the recommended limits.

Based on Figs. 12–15 and the discussion presented in this section, air supply velocity of 0.8 m/s gives air temperature and air velocity in

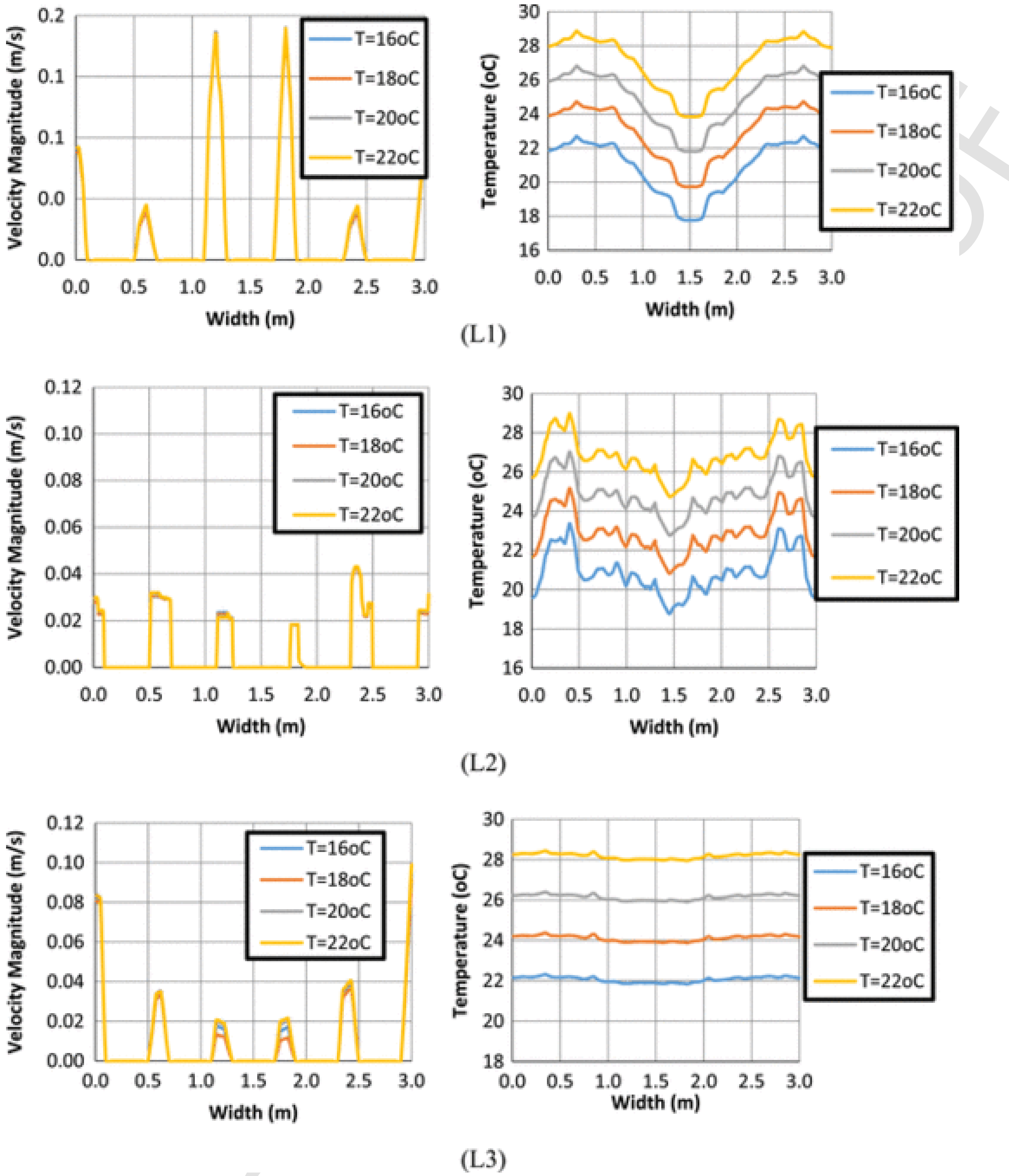


Fig. 10. Effect of supply air temperature on velocity and temperature distributions along L1, L2 and L3.

the occupancy zones levels within the ASHRAE comfort level and at the same time gives the best values of PMV and PPD which lies within the recommended comfort zone. Increasing the supply velocity above this values moves the conditions away from the comfort condition. So a supply air velocity of 0.8 m/scan be considered as the best supply air velocity for UFAD system.

4.3. Effects of number of supply air diffusers

Among the design and operating parameters, the number of supply air diffusers has a significant impact on the overall performance of UFAD system. Normally the supply air velocity and temperature are set to the recommended values as discussed in the previous sections. Increasing the number of air diffuser must be companies with decreasing the diffusers opening areas to maintain the recommended supply air velocity. In order to study and investigate the effect of the number of supply air diffusers on the performance of the UFAD system, numerical runs and calculations were conducted and repeated for different numbers of diffusers per row, namely 1, 2 and 3 diffusers. The supply air temperature and velocity and theater height are maintained constants and equal to the reference case values (t_s=18 °C, V_s=0.8 m/s and H=4 m). The temperatures and velocity

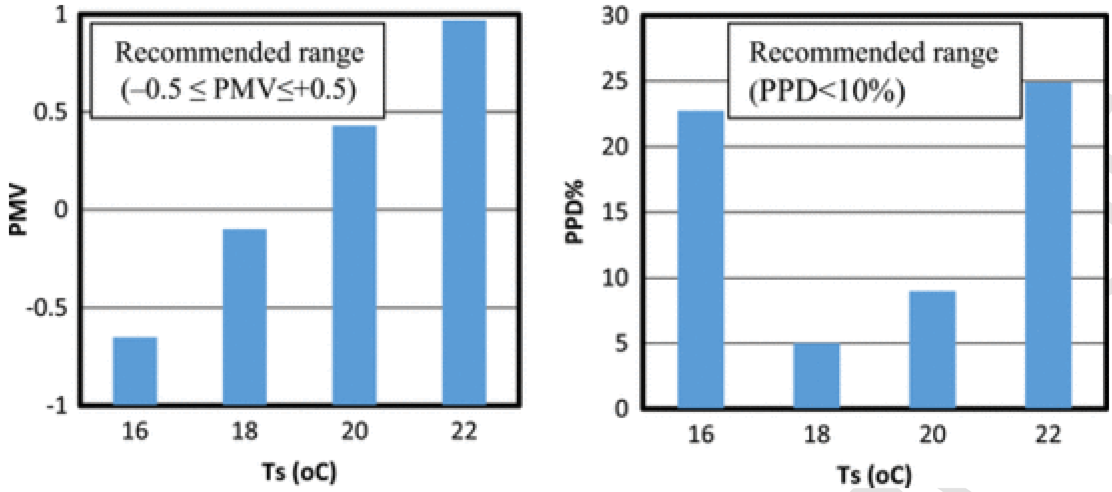
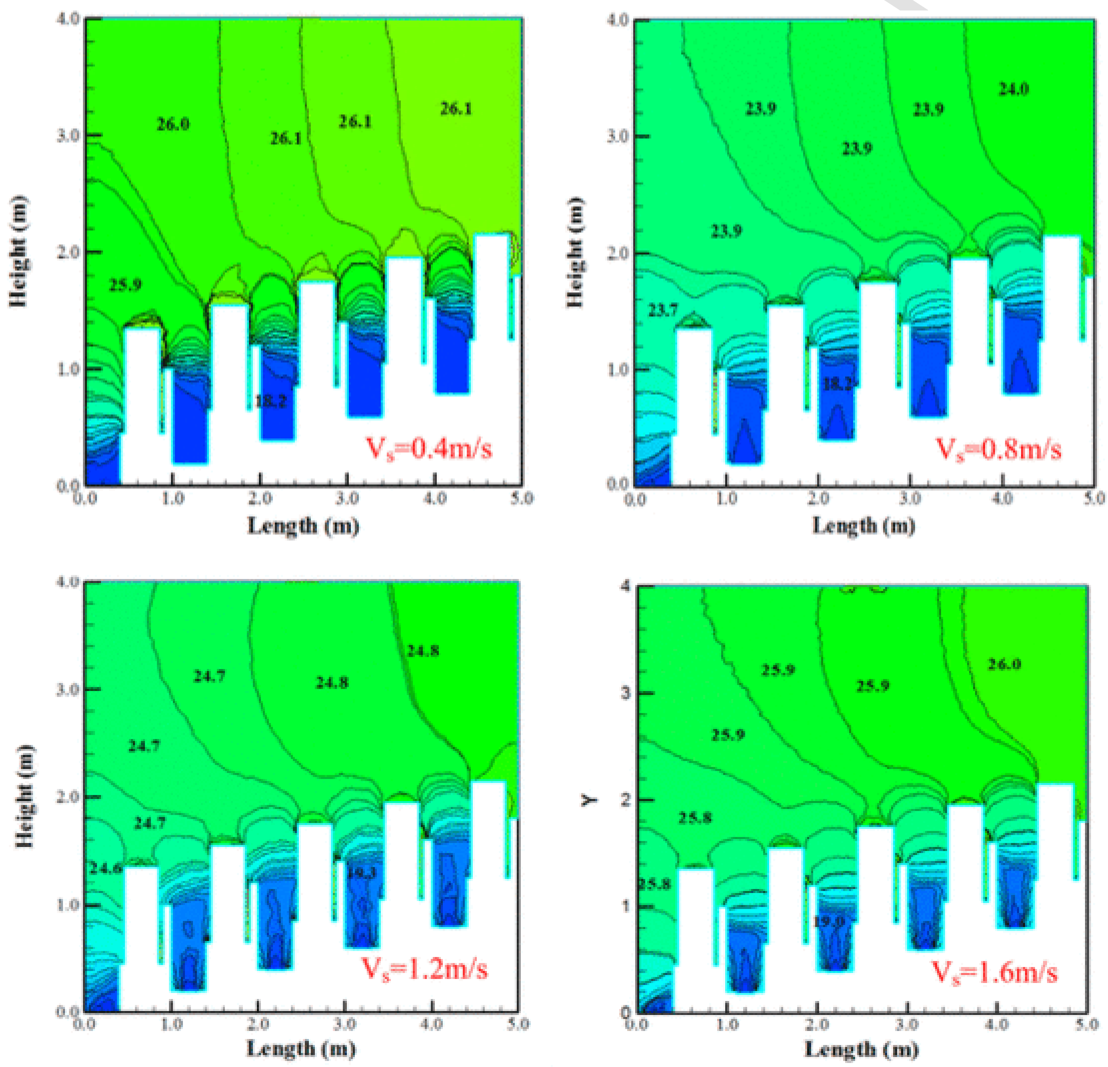
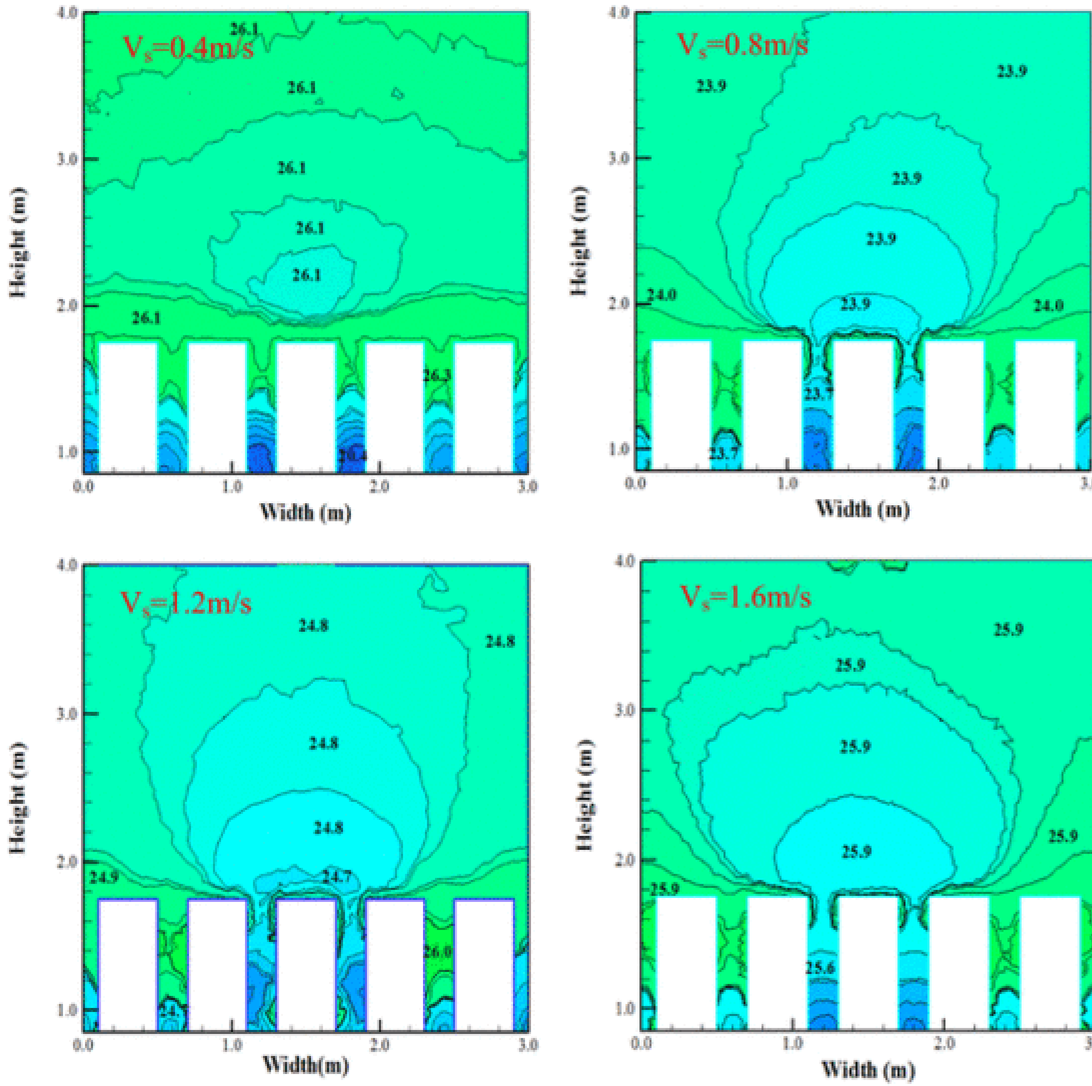


Fig. 11. Effect of supply air temperature on PMV and PPD.



(a) Plane 1

Fig. 12. Temperature contours on planes 1 and 2 at different supply air velocities.



(b) Plane 2

Fig. 12. (*Continued*)

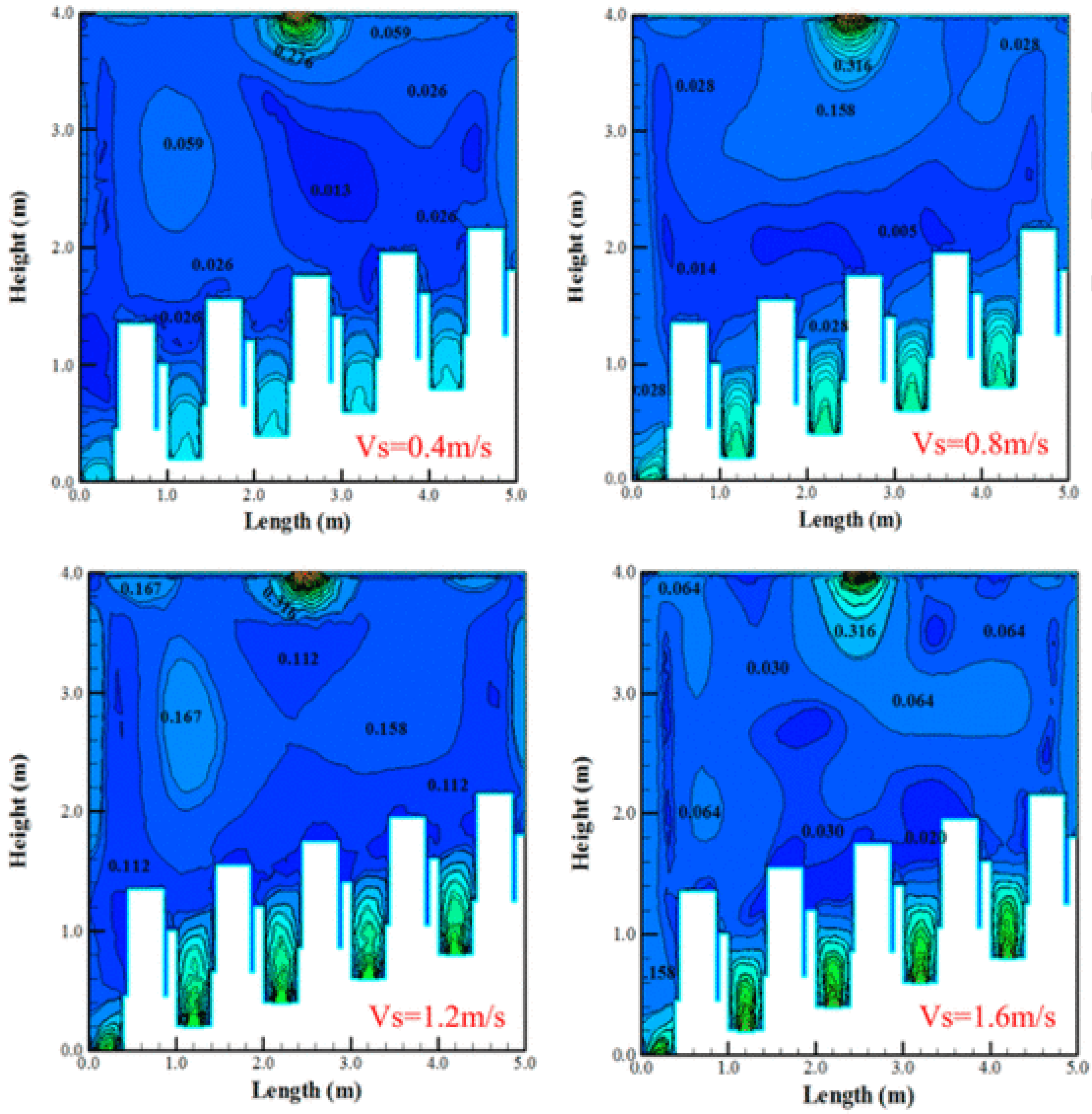
distributions at different three plans (plane 1, plane 2 and plane 3 passing by the diffusers, see Fig. 16) are presented to show the effect of the number of diffusers.

Fig. 17 shows the temperature contours over the three different planes (plane 1, plane 2 and plane 3), respectively for the different numbers of diffusers. The figure shows that the flow is uniform and stratified in the occupied zones for the different numbers of diffusers. The uniformity of the flow velocity and flow temperature along the width of the model increases with increasing the numbers of diffusers. Fig. 17-a shows that the temperature of plane 1 in case of using one diffusers is lower than that of case of using three diffusers (in both cases, plane 1 passes by a diffuser). This can be attributed to that the air flow thrown at plane 1 in case of using one diffuser is three times the flow thrown in case of using three diffusers and this make the temperatures at lower heights (before stratification) in case of using one diffuser is lower than that of three diffusers. The temperature levels of plane 1 in case of using one and three diffusers are lower than the temperature level of plane 1 in case of using two diffusers, especially at lower heights (before the occurrence of stratifications) and this can be attributed to that plane 1 does not pass by diffuser in case of using two diffusers. Fig. 17-c shows that the flow becomes more uniform and more stratified at lower heights as the number of diffusers increases.

Fig. 18 shows the velocity distributions at the different three planes for different number of diffusers. The figures of the three planes show that better velocity distribution and velocity comfort levels can be obtained with increasing the number of diffusers. This can be attributed to that increasing the number of diffusers creates more uniform distribution along the horizontal plane, faster flow uniformity along the height, and low velocity levels in the occupied zone.

The temperature and velocity distributions along the occupancy level, L3 (see Fig. 7) are shown in Fig. 19 for different number of diffusers. The figure shows that the velocity and temperature distribution at becomes more uniform with increasing the number of diffusers.

The effect of the number of supply diffusers on the thermal comfort indices PMV and PPD are shown in Fig. 20. As shown in the



(a) Plane 1

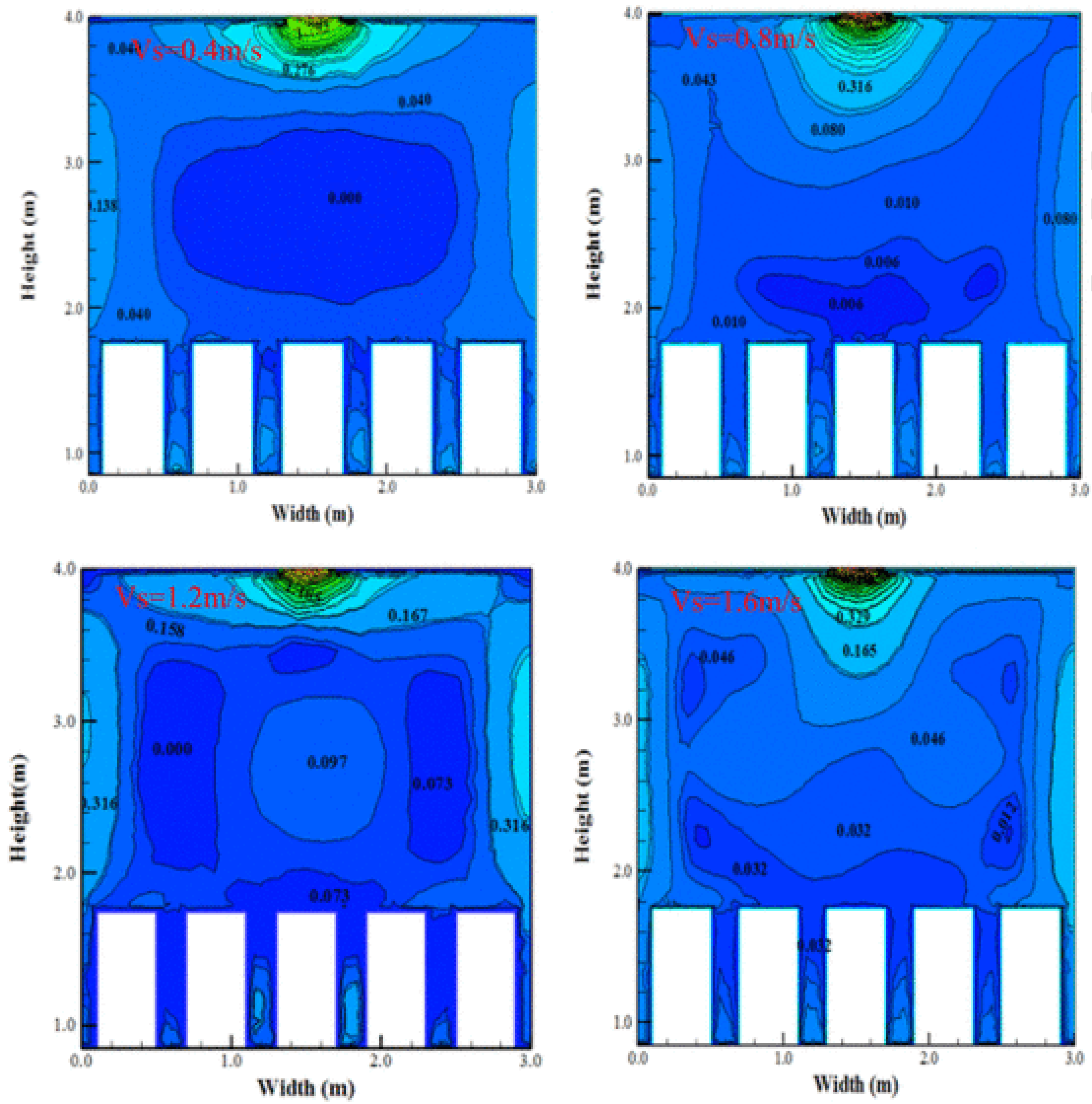
Fig. 13. Velocity contours on planes 1and 2 at different supply air velocities.

figure, the case of using one supply diffuser is not suitable for the occupied zones with UFAD system as it has a higher values of PMV and PPD as compared to the cases of using two and three diffusers. The figure shows that the PMV and PPD decrease with increasing the number of diffuser. The improvement of the thermal comfort with increasing the number of diffusers can be attributed to the poor air distribution (low temperature with high velocity) in the entire theater obtained in case of using one diffusers, while with more diffusers, cases of using two and three supply diffusers, create more comfortable environment in the theater. The figure also shows that there is small difference in values of PMV and PPD for the cases of N_d=2 and N_d=3.

Based on Figs. 17–20 and the discussion presented in this section, using three diffusers gives more uniform temperature and velocity distribution in the theater and also gives temperature and velocity levels in the occupied zones closest to ASHRAE comfort level. At the same time they gives the best values of PMV and PPD which lies within the recommended comfort zone. So increasing the number of supply diffusers or using a slot diffuser along the entire width of the theater sections can be considered as the best supply air distribution for the UFAD system.

4.4. Effects of building height

The potential energy benefit of using a UFAD system would be expected to be greater for large spaces with high ceiling. More investigations need to be done on building heights to optimize the utilization of thermal stratification at design and operation stages. Therefore, the effect of theater height is examined in the present study. In order to study and investigate the effect of the theater height on the performance of the UFAD system and energy consumptions, numerical runs and calculations were conducted and repeated for three different theater heights, namely 4, 7 and 10 m. The supply air temperature and velocity and numbers of supply diffusers are maintained constants as those of the reference case (t_s=18 °C, V_s=0.8 m/s and N_d=1).



(b) Plane 2

Fig. 13. (*Continued*)

Fig. 21a–c shows the temperature contours on the vertical planes 1, 2 and 3, respectively for different theater heights. The figures show that for the three different heights, the airflow is characterized by stable thermal stratifications. It is obvious that by increasing height of theater, temperature variations in the occupied zone are very small. So it can be concluded that for higher ceiling, the area above the occupied zone height can be neglected in the calculations of cooling load. As a result, less sensible zone load and volume flow rate are required, as only the occupied zone is conditioned and the remaining air is allowed to be stratified to the unoccupied zone. This air is extracted or returned to be part of the coil load leading to significant energy consumption saving.

Fig. 22a–c show the velocity contours on plane 1, plane 2 and plane 3 for different theater heights. It is shown that the air velocity in the theater, especially in the occupied zone, increases as the theater height decreases. This can be attributed to the extraction effect of the exhaust grille where its effect increases with decreasing the theater height.

The effect of the theater height on the temperature and velocity distribution along the occupancy level (L3) are shown in Fig. 23. The figure shows that the height has no effect on the temperature however the air velocity at these heights slightly increases with decreasing the theater heights due to the increase of the extraction effect of the return diffusers.

The effect of the theater height on the thermal comfort indices PMV and PPD are shown in Fig. 24. As shown in the figure, there is no effect of the theater height on the thermal comfort where the PMV and PPD for all the studied theater heights are within the range of 5.11–5.5% which lies within the ISO comfort standard suggested by ISO 7730 (less than 10%).

Based on Figs. 21–24 and the discussion presented in this section, UFAD system is suitable for all the studied theaters height where the comfort conditions are satisfied for the different heights. The advantage of using UFAD system increases with increasing the building heights where the zone above the occupied zone is excluding form the cooling load calculations which leads to huge energy saving.

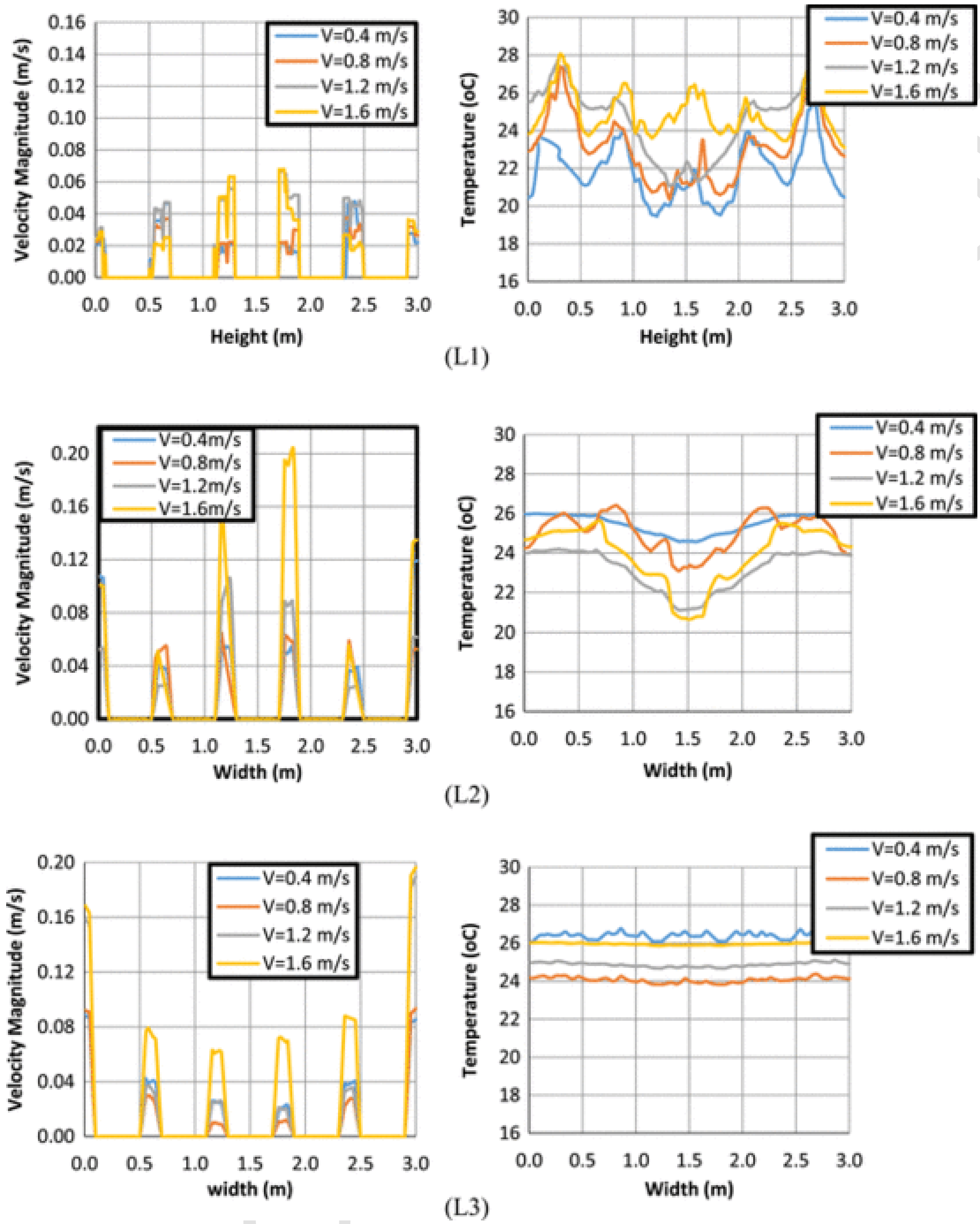


Fig. 14. Supply velocity effect along lines on: a) velocity distributions b) temperature distributions.

5. Comparisons between performance of UFAD and OHAD systems

It is interesting and necessary to quantitatively compare the temperature and velocity contours and the thermal comfort for the two air distribution mechanisms UFAD and OHAD. The key differences between the UFAD and OHAD systems arise with the location and the size (and thus the flow rate) of supply diffusers. The UFAD design uses diffusers installed on the floor for supplying conditioned air to the space and exhaust grilles at the ceiling, whereas the OHAD system has ceiling mounted diffusers and exhausts grilles. The two models are almost identical except for the air supply inlet. For the sake of this comparison study, two supply ceiling diffusers located at equal distances from the exhaust grille and the ends of the studied section are considered. Fig. 25 shows the physical model and computational domain of the OHAD system. Supply air flow rate and supply temperature are taken similar to the ones of the basic case of the UFAD system but higher supply air velocity of 1.5 m/s simulating the actual situation of OHAD is considered [36].

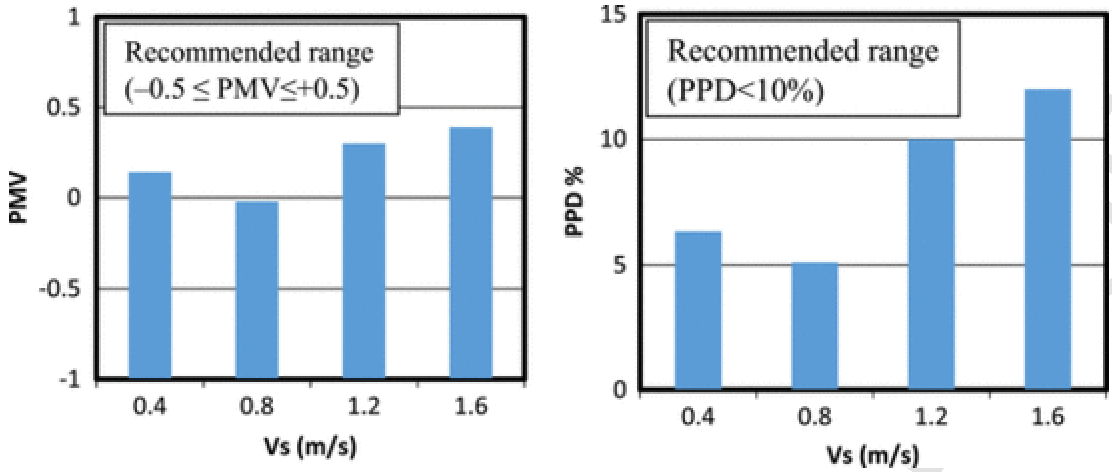


Fig. 15. Effect of supply air velocity on PMV and PPD.

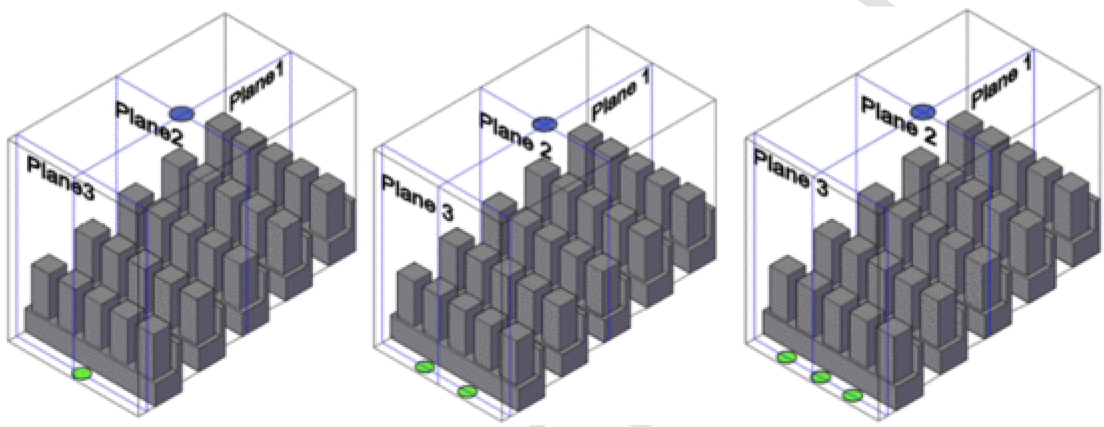


Fig. 16. Physical model with different numbers of diffusers.

Fig. 26 show the temperature contours for the UFAD and OHAD systems at the two mid planes (Plane 1and plane 2) for the sake of comparison. The figures show that for UFAD system the flow is uniform and stratified in the occupied zones. Also, the temperature levels of plane 1 in case of using UFAD system are higher than the temperature level of plane 1 in case of using OHAD system. As a result, for UFAD model the temperature variation along the occupancy level (1<H<1.8 m) lies within the 3 °C variation that recommended by previous investigations and standards [35]. While for OHAD model, air at the occupied zone is slightly cold and temperature gradient is higher than the recommended levels.

Fig. 27 shows the velocity contours at the two midplanes for the UFAD and OHAD systems for the sake of comparison. It is shown that for UFAD model, there is no discomfort level airflow velocity occurs in the whole space (ASHRAE recommends that the threshold limit for discomfort due to air velocity is 0.8 m/s). The figures of the two planes show that better temperature, velocity distributions and comfort levels can be obtained by using UFAD system.

The comparison of the temperature and velocity distribution along the three lines levels L1, L2 and L3 for the UFAD and OHAD systems is shown in Fig. 28. The figure shows that the temperature and velocity distribution in the occupancy levels L2, and L3 for UFAD system lies in the recommendation range, however those of OHAD systems lie outside the recommended range.

The comparison between UFAD and OHAD systems on the thermal comfort indices PMV and PPD are shown in Fig. 29. As shown in the figure, the PMV and PPD for UFAD system lies within the recommended limits (–0.5≤PMV≤+0.5 and PPD<10%) as stipulated by ISO 7730 [32] while the PMV and PPD for OHAD system lies outside the recommended limits. Finally, UFAD system has some advantages over overhead system as UFAD system shows more comfortable environment and smaller vertical variations of air temperature.

6. Conclusions

In the present paper airflow pattern, velocity and temperature distribution and thermal comfort inside an air conditioned theater were numerically investigated for different operating, geometric and design conditions and configurations. The effects of supply air temperature, supply air velocity, number of supply air diffusers and theater height on air flow pattern, temperature distribution, velocity distribution and thermal comfort are studied. Temperature distribution, velocity distribution and thermal comfort measuring parameters (PMV and PDD) are used as measurable overall performance of the air distribution system inside the theater. A comparison study between the under floor air distribution system and the overhead air distribution systems was also conducted to evaluate the suitability of the under floor air distribution systems for theater applications. The major find-

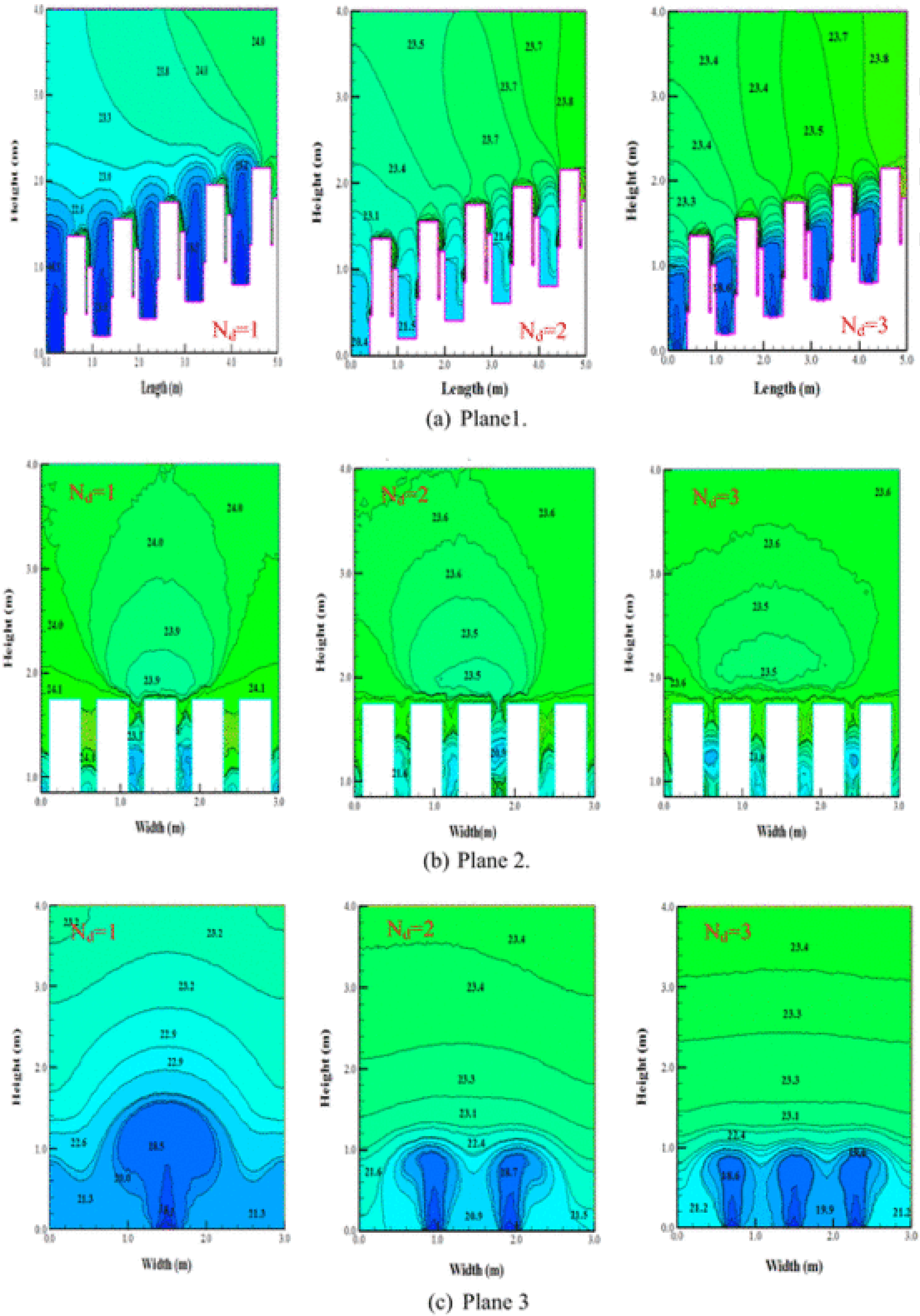


Fig. 17. Temperature contours on planes 1,2 and 3 at different numbers of diffuser.

ings and contributions of the present work are summarized as follows:

- The numerical technique used in the present study can predict air temperature and velocity distributions inside the conditioned space within reasonable accuracies.

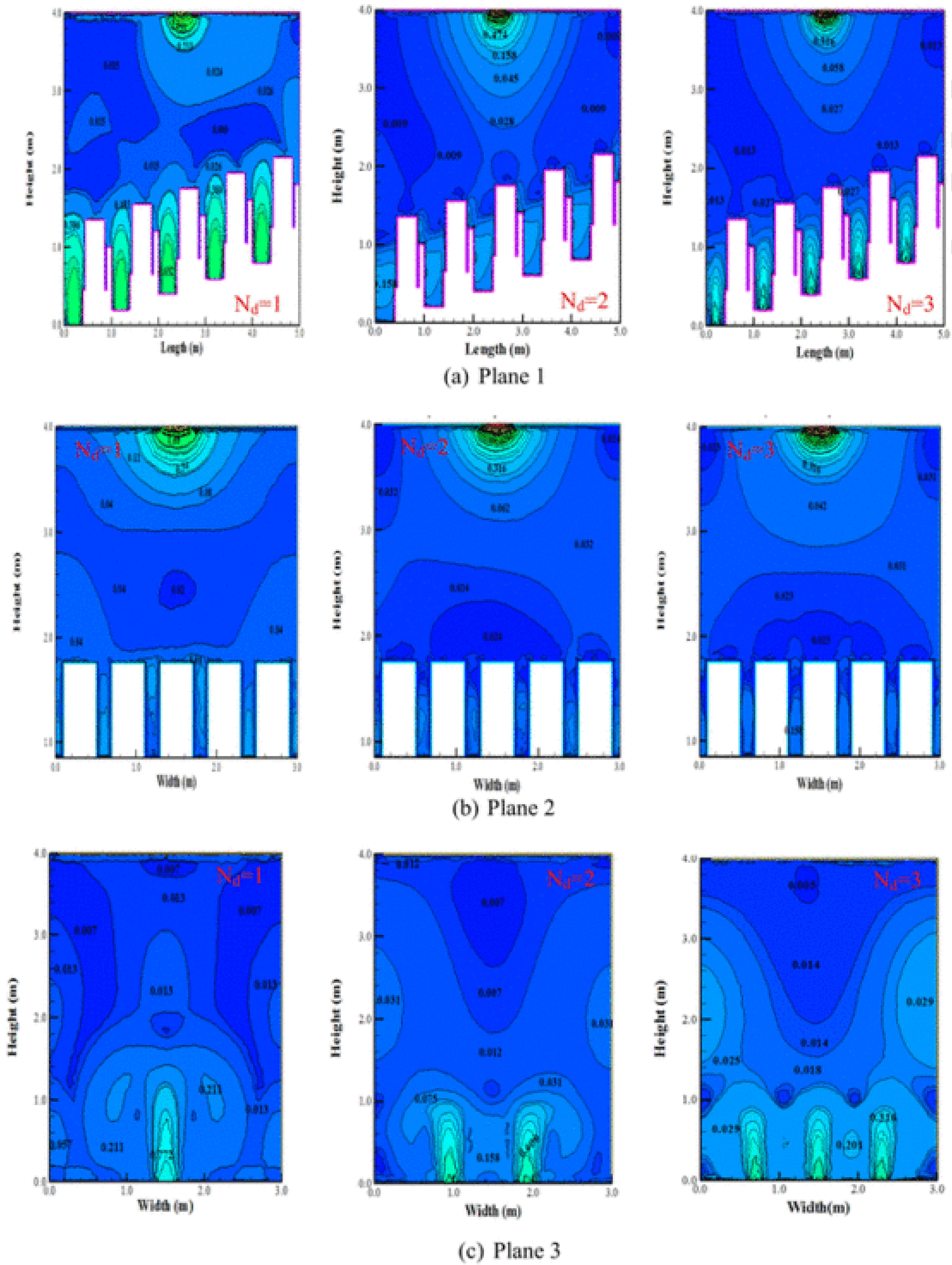


Fig. 18. Velocity contours on planes 1, 2 and 3 for different numbers of diffuser.

- The UFAD system has some advantages OHAD for theater applications as UFAD system shows more comfortable environment.
- The UFAD system leads to finer temperature distributions and lower mean temperature.
- Air temperature and velocity distribution of the UFAD system lie within the recommended range if the supply air temperature and velocity are properly selected.
- Supply temperature and velocity of 18 °C and 0.8 m/s are the best for UFAD system.
- The low air velocities of a UFAD system can provide the appropriate level of environmental comfort.

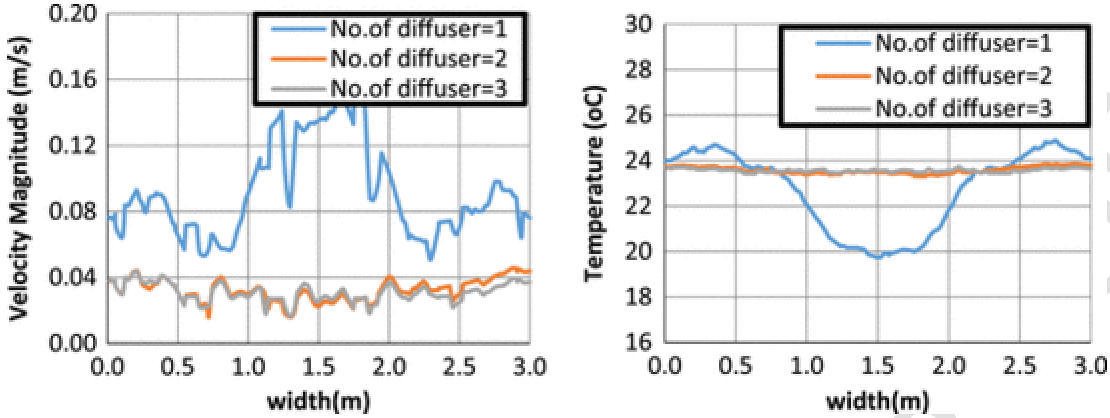


Fig. 19. Effect of numbers of diffusers on velocity and temperature distributions along the occupancy level (L3).

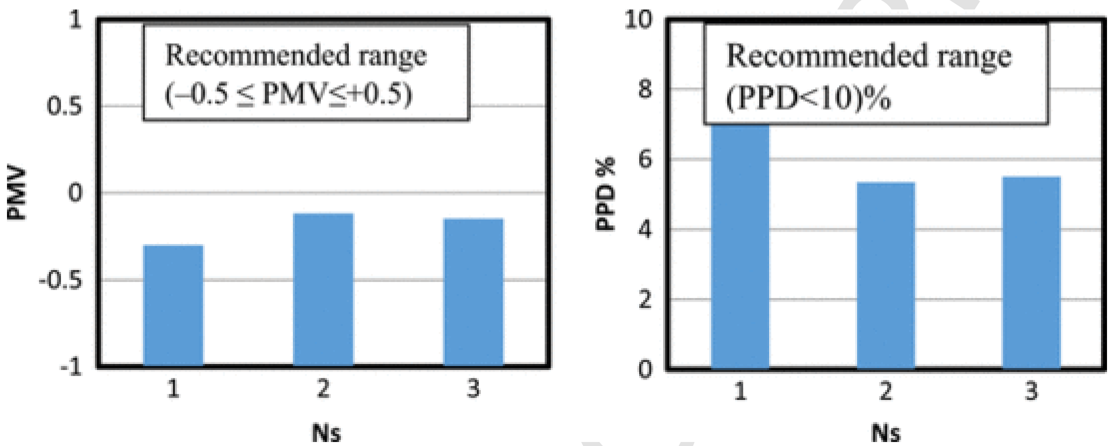


Fig. 20. Effect of supply air velocity on PMV and PPD.

- A smaller number of diffusers cannot obtain better comfort and higher numbers of diffusers are recommended.
- Energy savings of UFAD system increases as the theater height increases.

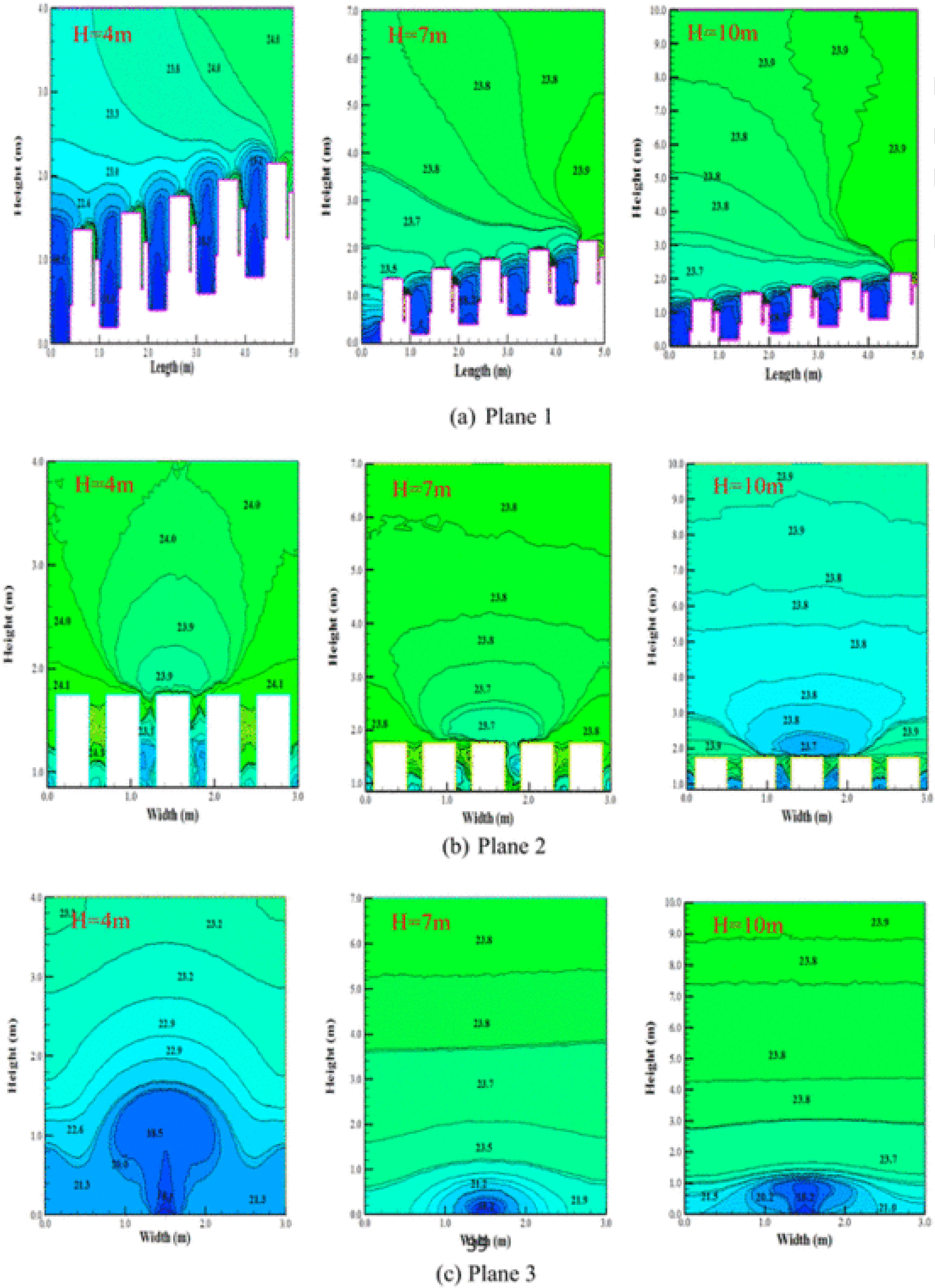


Fig. 21. Temperature contours on planes 1,2 and 3 for different theater heights.

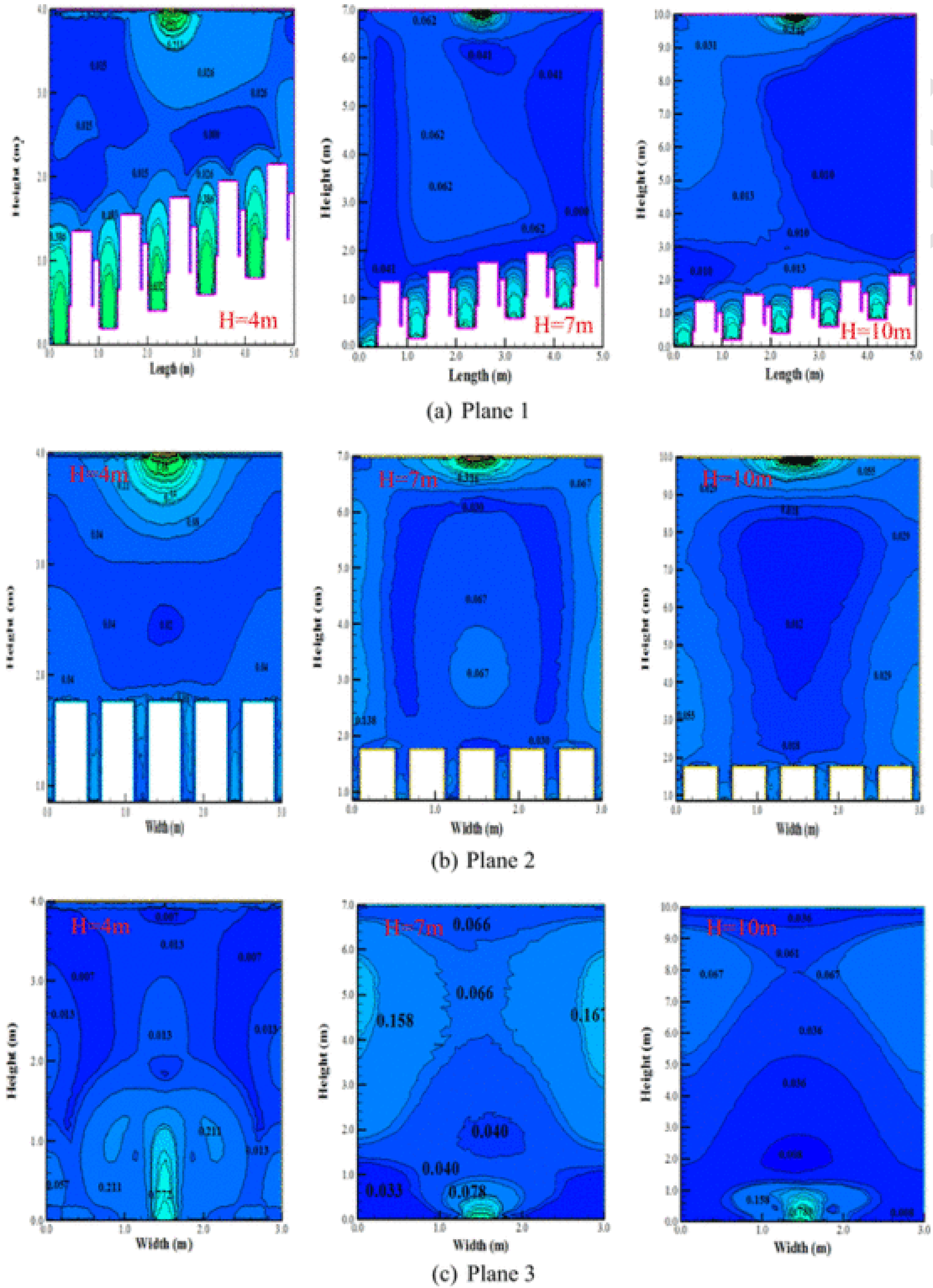


Fig. 22. Velocity contour on planes 1,2 and 3 at different heights.

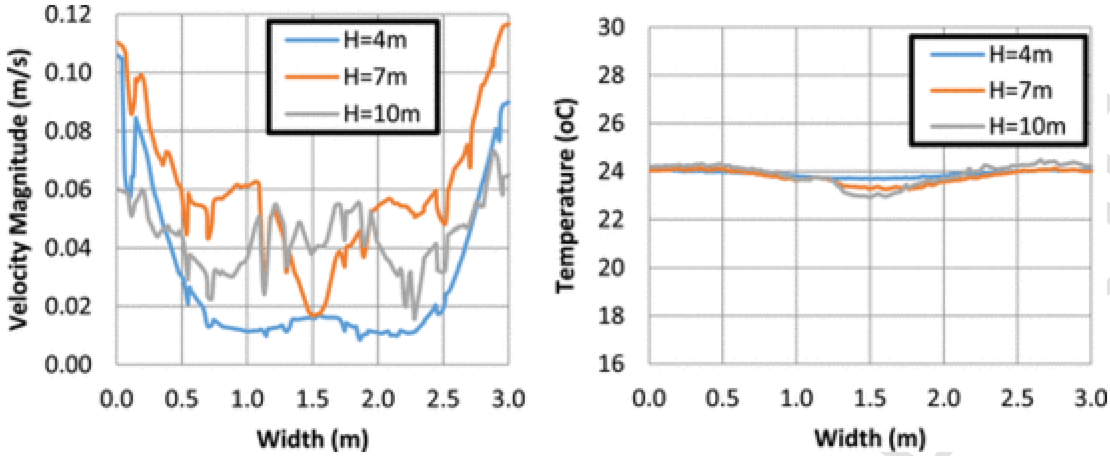


Fig. 23. Effect of theater height on velocity and temperature distributions along the middle row at occupancy level L3.

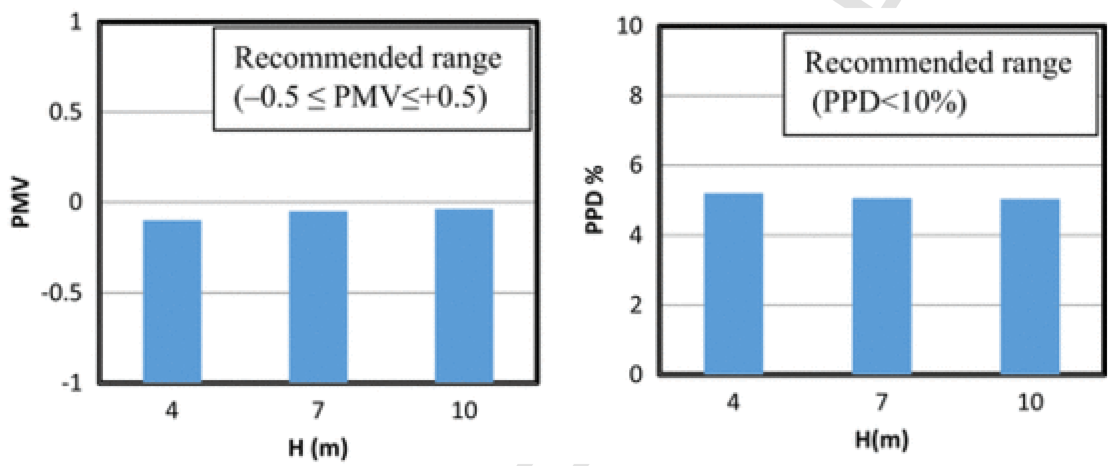


Fig. 24. Effect of theater height on PMV and PPD.

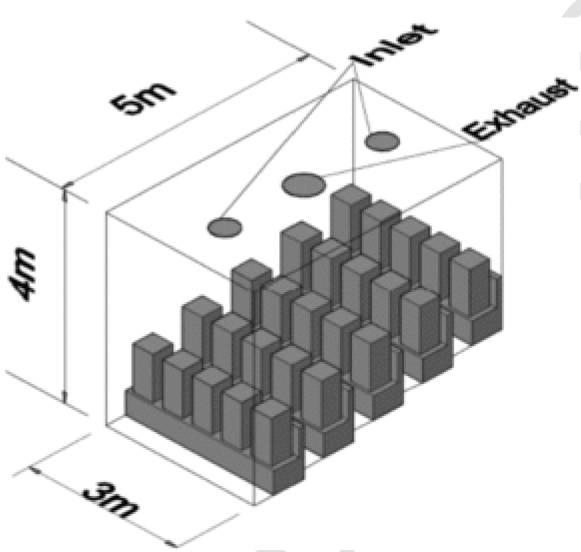


Fig. 25. Overhead air distribution mixing system model.

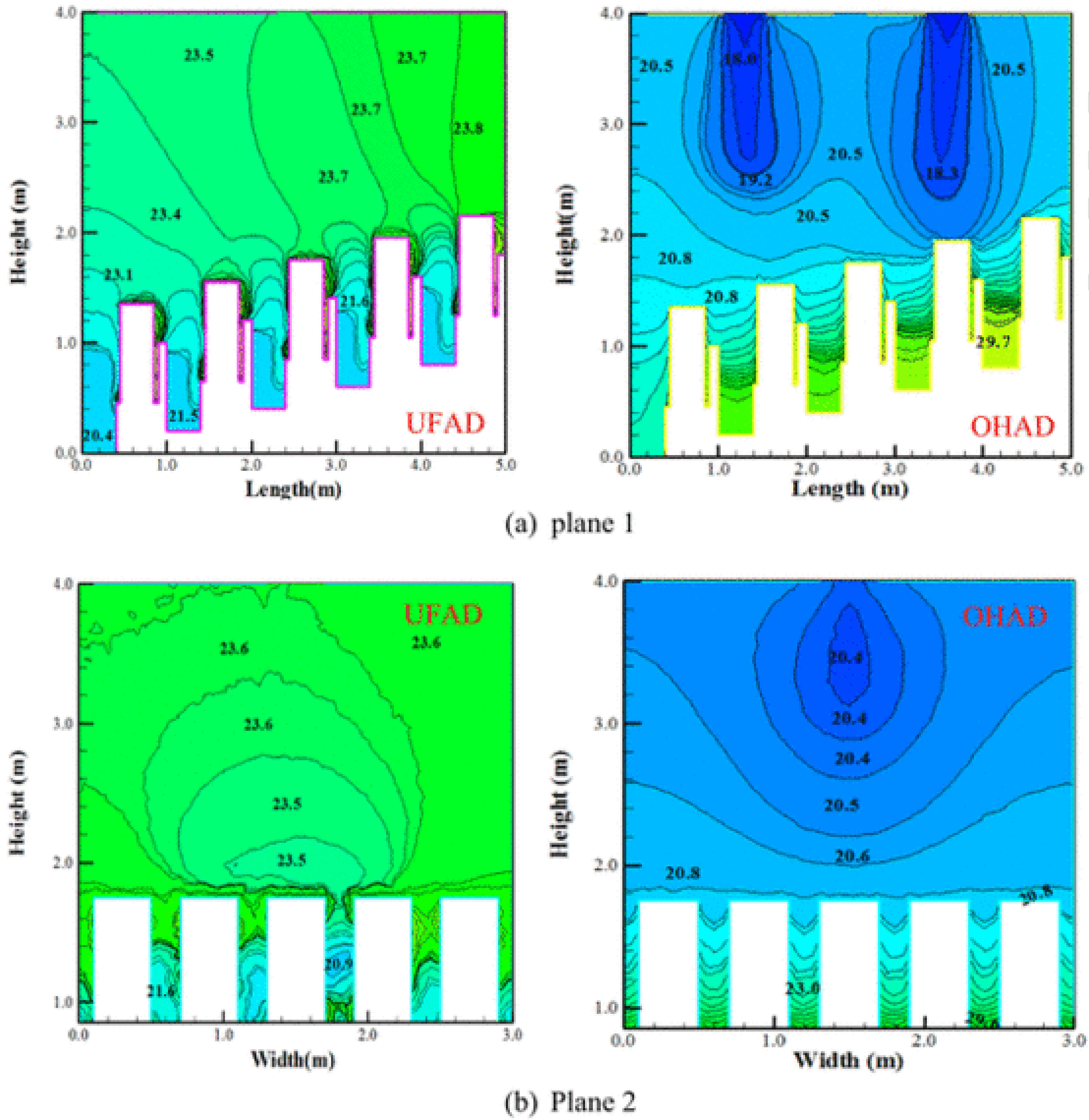


Fig. 26. Comparison of temperature contours on planes 1 and 2 for UFAD and OHAD systems.

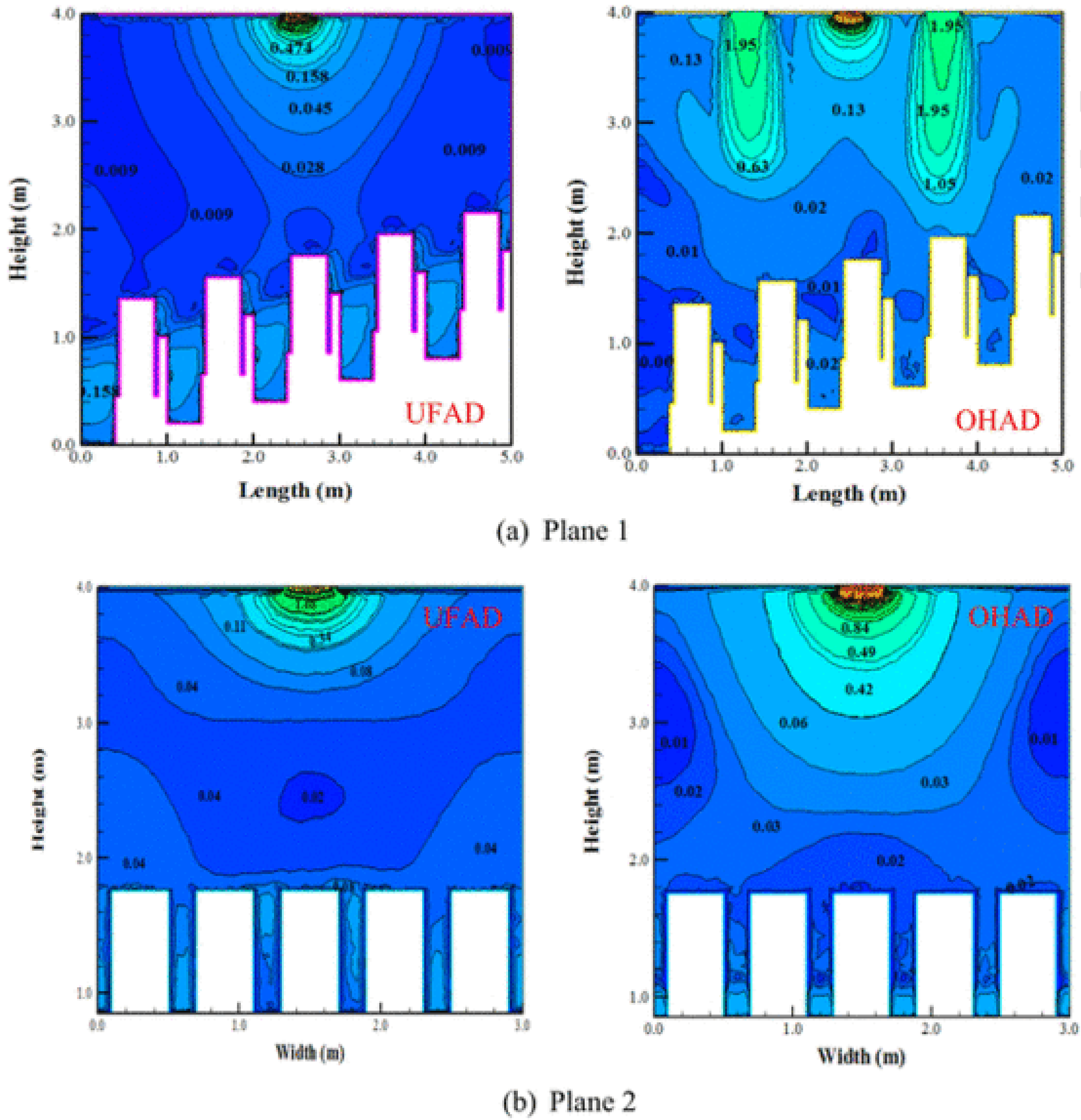
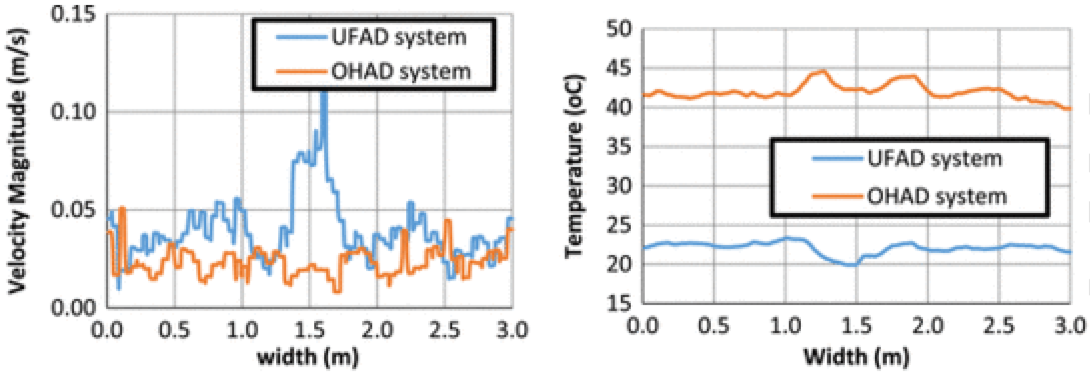
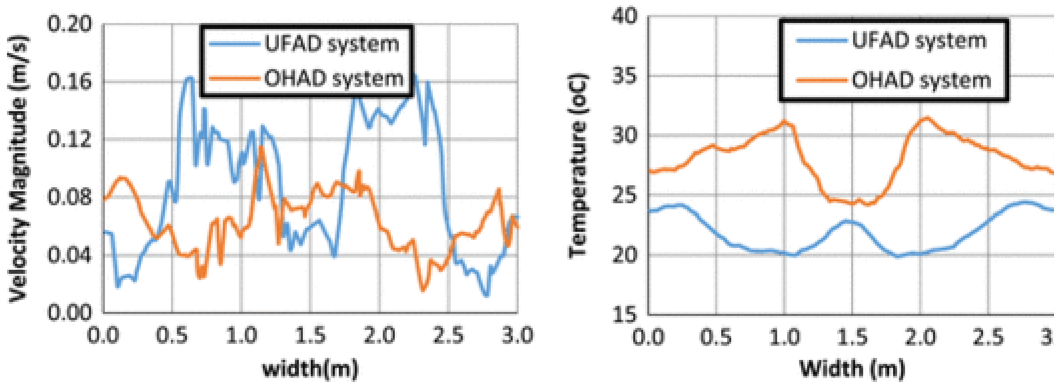


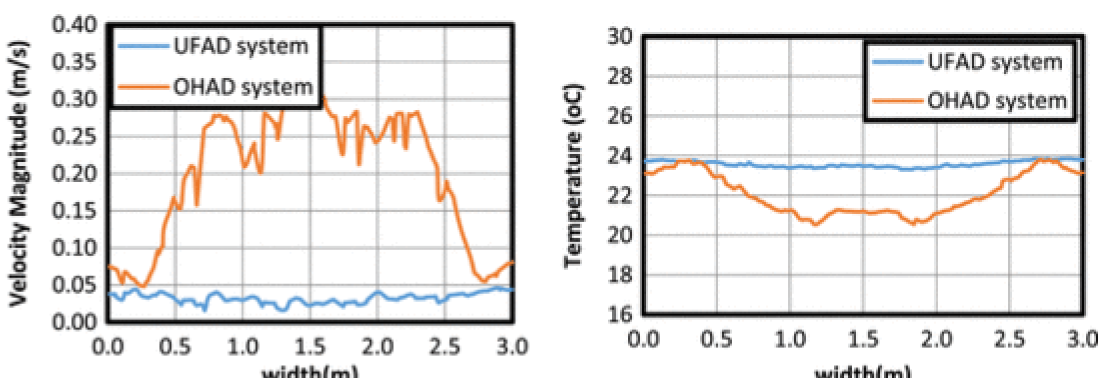
Fig. 27. Comparison of velocity contours on planes 1 and 2 for UFAD and OHAD systems.



(L1)



(L2)



(L3)

Fig. 28. Temperature and velocity distributions along L1, L2, and L3 for UFAD and OHAD systems.

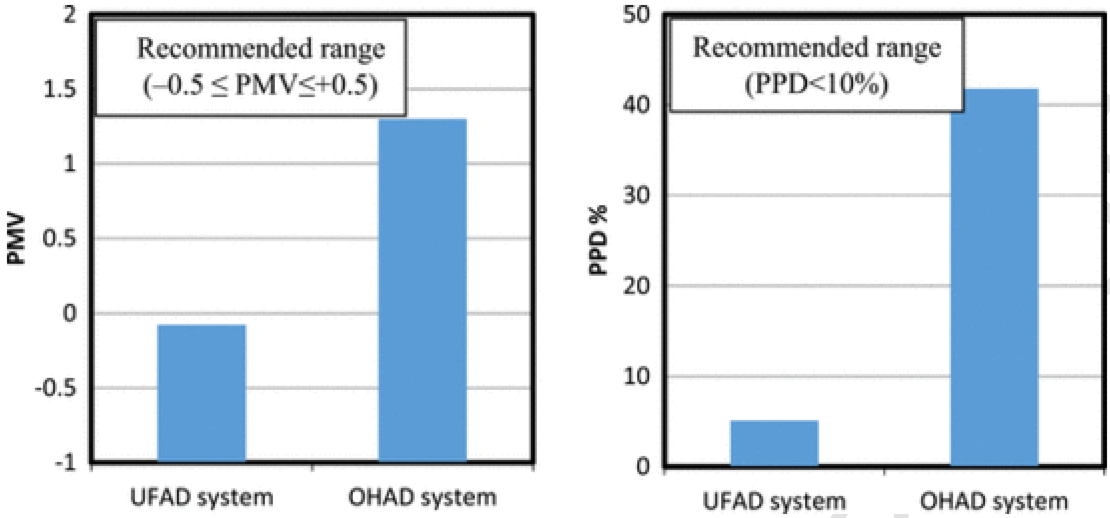


Fig. 29. Comparison between UFAD and OHAD system by PMV and PPD.

References

[1] R.W. Haines, Control System for Heating, Ventilating and Air Conditioning, 2nd editionVan Nostrand Reinhold, New York, 1977.
[2] J.D. Anderson, Computational Fluid Dynamics, Mcgraw-Hill, New York, 1995.
[3] A.J. Baker, M.K. Richard, B.G. Eliott, Roy Subrata G.S. Edward Computational fluid dynamics: a two-edged sword, ASHRAE J. 8 (1997) 51.
[4] F.S. Bauman, Giving occupants what they want: guidelines for implementing personal environmental control in your building, in: Proceedings at World Workplace 99, Los Angeles, CA, 1999.
[5] J.M. Halza, Underfloor and overhead ductless VAV systems, ASHRAE J. 45 (11) (2003) 43e8.
[6] T. Webster, Alternative air conditioning technologies: underfloor air distribution (UFAD), Energy Eng. 102 (6) (2005) 58–77.
[7] J.E. Woods, What real-world experience says about the UFAD alternative ASHRAE J. 46 (2) (2004) 3–15.
[8] T.W. Webster, F. Bauman, J. Reese, Underfloor air distribution: thermal stratification, ASHRAE J. 44 (5) (2002) 28–33.
[9] Ali Alajmi, Wid El-Amer, Saving energy by using underfloor-air-distribution (UFAD) system in commercial buildings, Energy Convers. Manag. 51 (2010) 1637–1642.
[10] Hongtao Xu, Jianlei Niu, Numerical procedure for predicting annual energy consumption of the under-floor air distribution system, Energy Build. 38 (2006) 641–647.
[11] J.D. Chung, H. Hong, H. Yoo, Analysis on the impact of mean radiant temperature for the thermal comfort of underfloor air distribution systems, Energy Build. 42 (2010) 2353–2359.
[12] Z. Lin, T.T. Chow, C.F. Tsang, K.F. Fong, L.S. Chan, CFD study on effect of the air supply location on the performance of the displacement ventilation system, Build. Environ. 40 (2005) 1051–1067.
[13] Y.J.P. Lin, P.F. Linden, A model for an under floor air distribution system, Energy Build. 37 (2005) 399–409.
[14] H. Ito, N. Nakahara, Simplified calculation model of room air temperature profiles in underfloor air-condition system, in: Proceedings of the International Symposium on Room Convection and Ventilation Effectiveness, ISRACVE, ASHRAE, 1993.
[15] Jae Dong Chung, Hiki Hong, Hoseon Yoo, Analysis on the impact of mean radiant temperature for the thermal comfort of underfloor air distribution systems, Energy Build. 42 (2010) 2353–2359.
[16] Zhang Lin, T.T. Chow, C.F. Tsang, K.F. Fong, L.S. Chan, W.S. Shum, Luther Tsai, Effect of internal partitions on the performance of under floor air supply ventilation in a typical office environment, Build. Environ. 44 (2009) 534–545.
[17] S.A. Nada, M.A. Said, M.A. Rady, Numerical investigation and parametric study for thermal and energy management enhancements in data centers buildings, Applied Thermal Engineering, Doi: 10.1016/j.applthermaleng.2015.12.020
[18] S.A. Nada, K.E. Elfeky, Experimental investigations of thermal managements solutions in data centers buildings for different arrangements of cold aisles containments, J. Build. Eng. 5 (2016) 41–49.
[19] S.A. Nada, K.E. Elfeky, A.M.A. Attia, Experimental investigations of air conditioning solutions in high power density data centers using a scaled physical model, International Journal of Refrigeration, DOI: 10.1016/j.ijrefrig.2015.10.027.
[20] S.A. Nada, K.E. Elfeky, Ali M.A. Attia, W.G. Alshaer, Thermal management of electronic servers under different power conditions, Int. J. Emerg. Trends Electr. Electron. 11 (4) (2015) 145–150.
[21] F. Bazdidi-Tehrani, A. Shahmir, A. Haghparast-Kashani, Numerical analysis of a single row of coolant jets injected into a heated crossflow, J. Comput. Appl. Math. 168 (2004) 53–63.
[22] S.J. Wang, A.S. Mujumdar, Flow and mixing characteristics of multiple and multi-set opposing jets, Chem. Eng. Process.: Process Intensif. 46 (8) (2007) 703–712.
[23] H.K. Versteeg, W. Malalasekera, An Introduction to Computational Fluid Dynamics: The Finite Volume Method, 2nd ed, Pearson Prentice Hall, Harlow, 1995.
[24] J.O. Hinze, Turbulence, Mcgraw-Hill Publishing Co, New York, 1975.
[25] H. Tennekes, J.L. Lumley, A First Course in Turbulence, MIT Press, Cambridge, MA, 1972.
[26] T.H. Shih, A. Shabbir, J. Zhu, A new k-ε eddy-viscosity model for high reynolds number turbulent flows-model development and validation, Comput. Fluids 24 (3) (1995) 227–238.
[27] H.F. Elattar, R. Stanev, E. Specht, A. Fouda, CFD simulation of confined non-premixed jet flames in rotary kilns for gaseous fuels, Comput. Fluids 102 (10) (2014) 62–73.
[28] S.A. Nada, A. Fouda, H.F. Elattar, Parametric study of flow field and mixing characteristics of outwardly injected jets into a crossflow in a cylindrical chamber, Int. J. Therm. Sci. 102 (2016)185–201.
[29] H.K. Versteeg, W. Malalasekera, An Introduction to Computational Fluid Dynamics: The Finite Volume Method, Longman Scientific and Technical, England, 1995.
[30] ASHRAE, ASHRAE Standard 55–2004, Thermal Environmental Conditions for Human Occupancy, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, 2004.
[31] ISO7730, Moderate Thermal Environments – Determination of the PMV and PPD Indices and Specification of the Conditions for Thermal Comfort, International Standard Organization, Geneva, 1994.
[32] P.O. Fanger, Thermal Comfort, Mcgraw-Hill Book CompanyNew York, 1972.
[33] F. Inc., FLUENT 6.3 User's Guide, FLUENT User Services Center, Lebanon (NH), Fluent Inc., 2006.
[34] Z. Zhang, Q. Chen, Experimental measurements and numerical simulations of particle transport and distribution in ventilated rooms, Atmos. Environ. 43 (2009) 319–328.
[35] ASHRAE, HVAC Systems and Equipment Handbook (SI), S6, Space Air Diffusion, 2000.
[36] ASHRAE STANDARD, BSR/ASHRAE Standard 55P, Proposed American National Standard Thermal Environmental Conditions for Human Occupancy, 3rd public review, February 2003.