AMERICAN UNIVERSITY OF BEIRUT

OPTIMIZED PERFORMANCE OF DISPLACEMENT VENTILATION AIDED WITH CHAIR FANS FOR COMFORT AND INDOOR AIR QUALITY

by
WALID MOHAMAD JAMAL ABOU HWEIJ

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AN ABSTRACT OF THE THESIS OF

Walid Mohamad Jamal Abou Hweij for Master of Engineering
Major: Mechanical Engineering

Title: Optimized Performance of Displacement Ventilation Aided with Chair Fans for Comfort and Indoor Air Quality

This study optimizes the performance of displacement ventilation (DV) system aided with chair fans (CF) for providing acceptable thermal comfort and indoor air quality (IAQ) in office space. A 3-D computational fluid dynamics (CFD) model is used to predict the airflow, temperature, and CO_2 concentrations fields. The (CFD) model coupled with a bio-heat model was used to predict the airflow, temperature, and CO2 concentrations fields as well as the occupant segmental skin temperatures for local and overall thermal sensation and comfort evaluation. Occupant. The CFD model results were validated experimentally in a DV conditioned space using a thermal manikin seated on a chair equipped with fans

Simulations were performed using the validated CFD model to determine the fans optimal height from the floor. For a source located at 1.0 m from the floor, it is recommended to place the fans at 30.9 cm above the floor at fans total flow rate of 12 L/s while for a source located at 0.3 m from the floor, the optimal height for fans is at 30.1 cm above the floor at fans total flow rate of 12 L/s. Compared to stand alone system, energy savings were 20.6% and 11.6% for fans' optimal heights of the sources placed at 1.0 m and 0.3 m, respectively.

CONTENTS

NOMENCLATURE

C	: Carbon dioxide concentration
CF	: Chair Fans
CFD	: Computational Fluid Dynamics
DV	: Displacement Ventilation
H	: Height of fans relative to floor
IAQ	: Indoor air Quality
LTC	: Local thermal comfort
LTS	: Local thermal sensation
MV	: Mixed Ventilation
PMV	: Predicted mean Vote
PPD	: Percentage of people dissatisfied
PPM	: Parts per million
Q	: Flow rate
RH	: Relative humidity
TC	: Overall thermal Comfort
TS	: Overall thermal Sensation
Y+	: Dimensionless wall thickness

Greek Symbols:

ε	: Effectiveness

Subscripts

B	: Breathing
E	: Exhaust
Fr	: Fresh air
fs	: Fans
v	: Ventilation
s	: Source
su	: Supply

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CHAPTER I

DISPLACEMENT VENTILATION COMBINED WITH CHAIR FANS

A. Introduction

Acceptable thermal comfort and good indoor air quality (IAQ) are the main requirements to be provided by ventilation systems in open space office buildings. Displacement ventilation (DV) system supplies 100% fresh and cool air at low velocity near the floor level from a wall mounted diffuser relying on heat sources that push the resulting warm and polluted air by buoyancy to the upper room level to be exhausted; thus creating temperature stratification within the space (X.Yuan; Q. Chen and L. R. Glicksman, 1998). It was well documented in the literature that DV system provides acceptable thermal comfort and good IAQ compared to other conventional ventilation systems (Z. Lin; T. T. Chow; K. F. Fong; Q. Wang and Y. Li, 2005 a) (Z. Lin; T. T. Chow; K. F. Fong; C. F. Tsang and Q. Wang, 2005 b) (H. Xing; A. Hatton and H. B. Awbi, 2001). However, this system requires the conditioning of the 100% fresh air supply stream which is energy intensive compared to mixed streams used in conventional. Additionally, the restrictions on the supply air temperature (not to be less than 18 ^{o}C) and flow velocities (not to be more than 0.2 m/s) (ASHARAE Handbook, 2009), hindered the DV system from removing high internal loads. Thus, DV system might not be cost effective in hot and humid climate conditions. Furthermore, Brohus et al. (H. Brohus and P. V. Nielsen, 1996) reported poor performance when a passive contaminated source existed at low level.

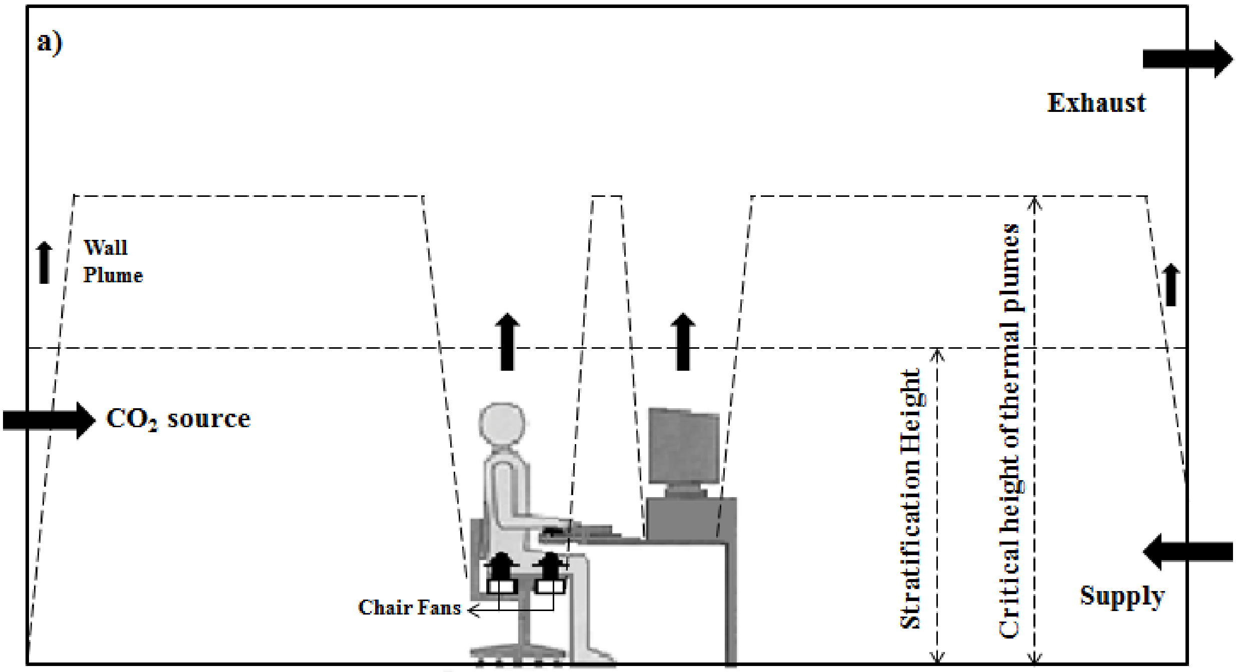
Due to the increasing popularity of DV system in Europe and USA as reported by Mundt (E. Mundt, 1995) and its superiority over other mixed ventilation systems, several studies proposed different approaches to improve the performance of DV system. One approach was to use chair-mounted fans of an office occupant. Sun et al. (W. Sun; K. W. D. Cheong and A. K. Melikov, 2012) proposed the use of four computer fans placed at the corners of the chair. They surveyed participants about their TC while changing speed of the fans for different room air temperatures. Their study suggested the appropriate fans' speed that provided TC for occupants for the various room temperatures. Moreover, Sun et al. (WEIMENG, S. 2010) investigated the effect of chair fans (CF) on an ambient passive contaminant represented by CO_2 tracer gas. The study showed that for the proposed configuration of fans (at the level of thighs), the fans increased the concentration of CO_2 at the occupant breathing zone. Additionally, they found that the use of this novel system resulted in a reduction in energy consumption up to 19 %. Habchi et al. (C. Habchi; K. Ghali; N. Ghaddar and A. Shihadeh, 2015) investigated the performance of DV system assisted by chair fans (DV + CF) in reducing particles transmission. They reported that the use of fans caused a reduction in inhalation effectiveness when compared to a standalone DV system. In addition, they reported that the novel system provided TC when assessed using predicted mean vote (PMV) model. Evidently, using CF in aiding DV system had noticeable effects on TC and particles transmission while reducing energy consumption.

All the previous studies focused on the use of CF at a fixed level without consideration of the effect of the height while varying fan flow rate on thermal comfort and IAQ. Recent study done by El-Fil et al. (B. El-Fil; K. Ghali and N. Ghaddar, 2015) showed noticeable improvements in TC and IAQ as a result of variations in height and flow rate of CF while aiding ceiling personalized ventilation system (CPV). In DV system, varying the height of the fans above the floor while varying their flow rate may affect the thermal comfort. Moreover, the use of CF at different heights and flow rates may affect the breathing air quality when passive contaminants are placed at different heights above the floor. To our knowledge, the effect of fans height above the floor and their flow rate on breathing air quality and comfort at different passive contaminant height above the floor has not been investigated in the literature.

Hence, this study will investigate the performance of (DV + CF) system with the purpose of optimizing the height of fans above the floor for different passive contaminant height. A computational fluid dynamics (CFD) model of the DV-conditioned space with chair fans is developed. The CFD model will be validated experimentally using thermal manikin to compare the segmental surface temperature and CO_2 contaminant concentration in the breathing zone at a specified source height. A parametric study will follow to assess the system performance in providing acceptable thermal comfort and good IAQ for an office occupant at low energy consumption.

B. System Description

Figure 1(a) shows a schematic diagram of (DV + CF) system. The supply diffuser supplies fresh air into the room, the thermal manikin represents an office occupant, the chair mounted fans, placed on each side, blow air upwards, the CO_2 tracer gas represents passive contaminants in the room, and the exhaust diffuser extracts contaminated air from the space. Fig. 1(b)-(c) shows in details the configuration of chair mounted-fans. The fans can move vertically up and down resulting in a controlled height (H_{fs}) while supplying uniform flow rate$(\dot{Q})$.



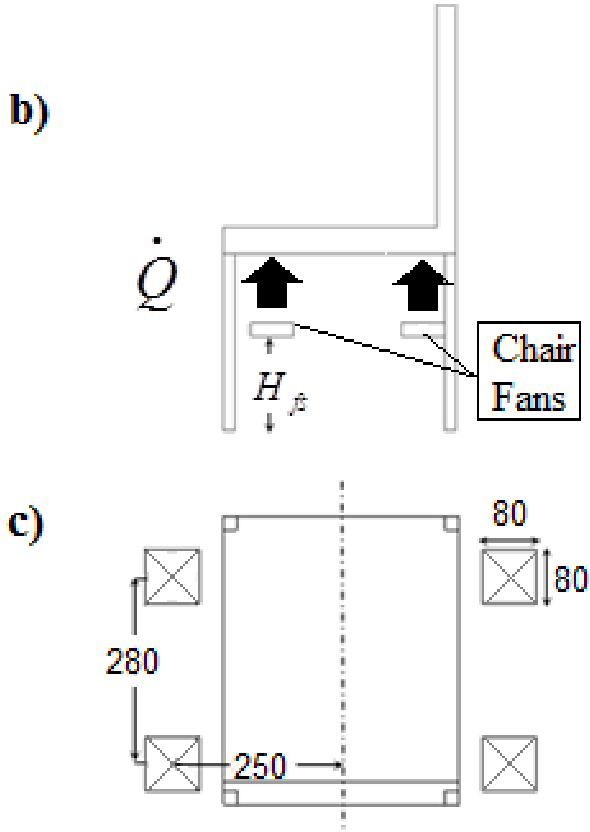


Figure 1: (a) Schematic diagram of the proposed (DV + CF) system and (B) Chair mounted fans (b) Chair side view and (c) chair top view. All dimensions on the drawing in millimeter.

CHAPTER II

DEVELOPMENT OF CFD MODEL AND ITS COUPLING WITH BIO-HEAT MODEL

The complex interactions resulting from the upward jets of fans and the rising thermal plumes should be accurately predicted to assess their behavior on the thermal and CO_2 concentration fields. Similar studies done by Makhoul et al. (A. Makhoul; K. Ghali and N. Ghaddar, 2013 a) resulted in accurate prediction of the airflow and thermal patterns along with the CO_2 concentration fields. Thus, CFD modeling would be used to investigate the effectiveness of the proposed (DV + CF) system under different configurations.

A. Air Flow Modeling

The commercial software ANSYS© Fluent was selected for assessing the flow fields of indoor environment as shown in Fig. 2. Accurate prediction of the airflow, thermal, and CO_2 concentration fields are based on proper modeling of flow physics including turbulence models, buoyancy effects, and boundary layers near the surfaces (A. Makhoul; K. Ghali and N. Ghaddar, 2013 b). Proper controlling of the grid resolution at some surfaces is crucial to capture perfectly the shear-layer entrainment process and the different thermal plumes as well as the fluid/thermal boundary layers around the occupant (A. Makhoul; K. Ghali and N. Ghaddar, 2013 a). The flow behavior at the boundary layer is accurately predicted using inflation layers. The selection of the inflation layers is based on first layer thickness y such that the dimensionless number y^+ ranges between 0.8 and 4, in order to a accurately resolve the viscous sub-layer as reported by ANSYS online manual (ANSYS Software). Accordingly, the first layer thickness based on a selected y^+ value is estimated as follows:

$$y = \frac{y^+ \mu}{\rho \mu_\tau} \tag{1a}$$

where ρ and μ are the air density and dynamic viscosity and μ_τ the friction velocity is given by:

$$\mu_\tau = \sqrt{\frac{\tau_w}{\rho}} \tag{1b}$$

where τ_w is the wall shear stress.

The boundary faces were set to different element sizes using tetrahedral unstructured grid. A grid independence test was performed. The mesh is considered independent when the maximum relative error between two consecutive grid sizes is less than 5%. Initially, a face size of 2 cm and 8 cm were assigned to the surfaces of the manikin and the walls respectively resulting in 766,144 elements and 184,707 grid nodes. The face size was increased where the temperature, velocity and concentration fields at certain locations were compared with the previous grid test. This was repeated until grid independence was assured. A total number of 3,516,946 elements and 876,576 grid nodes were selected to accurately predict the flow behavior with less computational cost as illustrated in Table 1. Fig.3 (a)-(b) represent the generated grid for the computational domain and the manikin.

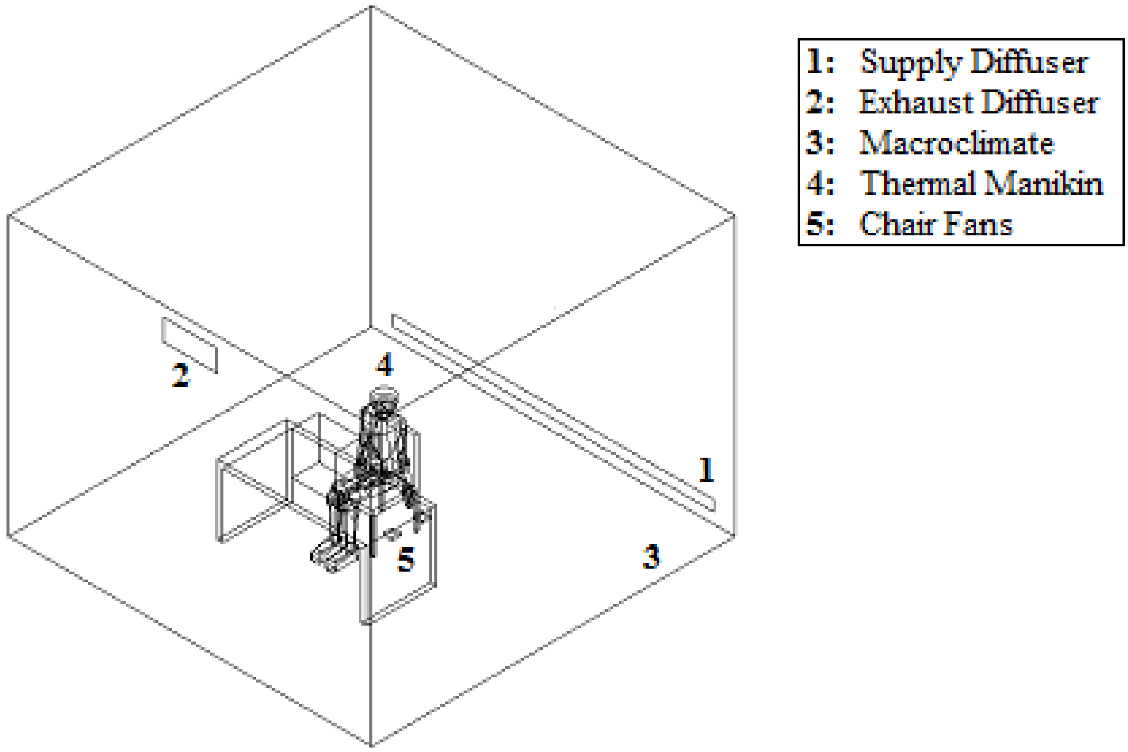


Figure 2: The Computational domain used in ANSYS (1) Supply diffuser, (2) Exhaust, (3) Macroclimate, (4) Thermal manikin, and (5) Chair fans.

Table 1: Grid independence testing using five mesh cases

Grid Type	Face Sizing [cm] Manikin/Walls	Number of Elements	Number of Grid Nodes	Maximum relative difference in predicted values of temperature velocity and concentration with the previous mesh values %
Mesh 1	2 / 8	766,144	184,707	-
Mesh 2	2 / 5	1,217,543	301,264	43.52
Mesh 3	1.5 / 3	2,559,155	638,040	10.32
Mesh 4	1.5 / 2.5	3,516,946	876,576	5.5
Mesh 5	1.5 /2	5,652,622	1,384,638	4.4

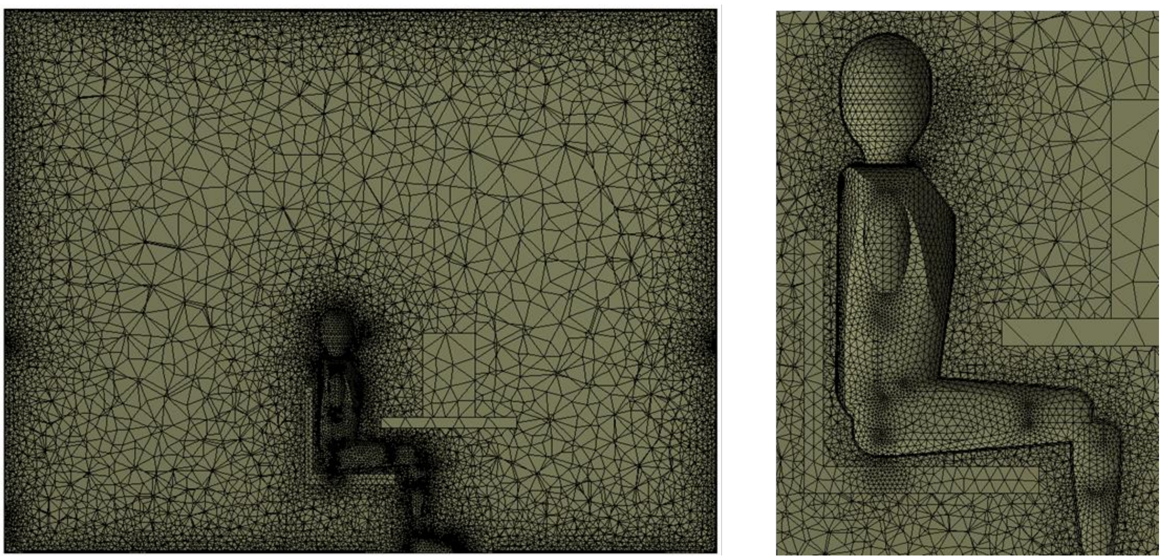


Figure 3: (Left) Mesh generation of the computational domain. (Right) Mesh generation of the manikin.

Numerical modeling established by the commercial software ANSYS Fluent© is used to solve for the velocity, thermal, and species concentration fields within the space. The standard k-ε turbulence model is selected based on its ability to predict accurate results that were validated experimentally in published literature (P. Zelenský; M. Barták; J. L. M. Hensen and R. Vavřička, 2013) (M. P. Wan and C. Y. Chao, 2005) (M. Kanaan; N. Ghaddar; K. Ghali and G. Araj, 2014). The enhanced wall treatment option is selected to accurately predict the flow fields that involved separation and recirculation (A. Makhoul; K. Ghali and N. Ghaddar, 2013 a) (ANSYS Software). The Boussinesq approximation is used in the density modeling to account for buoyancy effects (ANSYS Software) (M. Kanaan; N. Ghaddar; K. Ghali and G. Araj, 2014). On the other hand, the species transport equation is used to assess the CO_2 concentration field (ANSYS Software). The momentum, k, ε, energy and even the species transport differential equations are discretized using the second order upwind scheme (ANSYS Software). The "Standard" staggered scheme is used for the pressure (M. Kanaan; N. Ghaddar; K. Ghali and G. Araj, 2014) where the latter is coupled with velocity fields using the simple algorithm (ANSYS Software). The numerical convergence criteria was based on assigning scaled residual of minimum value of 10^{-5}, monitoring the actual values at certain points, and having a net heat flux of the computational domain below 1% of the total heat gain (A. Makhoul; K. Ghali and N. Ghaddar, 2013 a).

B. Air Quality Modeling

The mixing between the supplied fresh air and the passive contaminants affect the level of air quality inside the room. Different studies in the literature used the CO_2 tracer gas as an indication of IAQ. In this work, this was done by measuring the concentration of CO_2 at the breathing zone of occupant relative to that at the exhaust and supply diffusers (A. Makhoul; K. Ghali and N. Ghaddar, 2013 a). Hence, the ventilation effectiveness ε_v was expressed as follows:

$$\varepsilon_v = \frac{C_R - C_B}{C_R - C_{Fr}} \tag{2}$$

where C_R is the CO_2 concentration of the recirculated air, C_B is the CO_2 concentration at the breathing zone, and C_{Fr} is the CO_2 concentration at the supplied fresh air. Therefore, the higher the ventilation effectiveness the better will be the air quality.

Passive contaminant source represented by CO_2 tracer gas was placed at a certain height above the floor where it supplied CO_2 at a specified flow rate. Real fresh air in the range of 300-500 ppm was introduced via DV supply diffuser as recommended by (ANSI/ASHRAE standard 62.1-2013). The concentration of inhaled air is measured as a volume average concentration in the spherical volume of 2 cm diameter placed at 2.5 cm from the manikin's nose (A. Makhoul; K. Ghali and N. Ghaddar, 2013 a).

C. Boundary Conditions

Proper CFD modeling is based on the setting of boundary conditions to accurately predict airflow, thermal and CO_2 concentration fields. The supply diffuser and CO_2 source are set to a specified velocity inlet, temperature, turbulence intensity, CO_2 concentration, and hydraulic diameter. The exhaust diffuser is set to pressure outlet with zero gauge pressure at a specified turbulence intensity and hydraulic diameter. Fans' inlets are set to "fan" boundary condition with a predefined pressure drop depending on fan's flow rate and fans' outlets are set to "interior" boundary condition (A. Makhoul; K. Ghali and N. Ghaddar, 2013 a). Finally, the walls and ceiling are set to constant heat flux.

D. Bio-Heat Coupling

In order to assess the effect of fans on the thermal performance of (DV + CF) system, a validated bio-heat model proposed by Salloum et al. (M. Salloum; N. Ghaddar and K. Ghali, 2007) is coupled with the developed CFD model to accurately predict the occupant physiological response to the thermal environment. The prediction of occupant TC is evaluated using the models proposed by Zhang et al. (H. Zhang; E. Arens; C. Huizenga and T. Han, 2010). The TC ranged from -4 (very uncomfortable) to +4 (very comfortable) where 0 referred to just comfortable. The thermal sensation scaled from -4 (very cold) to +4 (very hot) where -3 referred to cold, -2 to cool, -1, to slightly cool, +1 to slightly warm, +2 to warm, and +3 to hot (H. Zhang; E. Arens; C. Huizenga and T. Han, 2010). The coupling between the CFD and bio-heat models is shown in Fig. 4.

Initially, in the CFD model, the manikin was set to conventional skin temperatures along with the proper settings for the energy, turbulence models, CO_2 transport equations, and boundary conditions for different system components. Appropriate solution controls along with proper discretization schemes were assigned for the modeling equations. The bio-heat model divides the body into 11 segments (head, chest, back, abdomen, buttocks, U-arm, L-arm, hands, thighs, calves and feet). The prediction of physiological thermal responses of each body segment was based on the air temperature and the convective heat transfer coefficients at the vicinity of each body segment. The latter components were well predicted by the CFD model. The resulting segmental skin temperatures predicted by the bio-heat model were introduced as boundary condition for the occupant in the CFD model, where the coupling procedure would be repeated such that the difference between two consecutive iterations is less than 10-3. Upon convergence, the predicted segmental skin temperatures were integrated with Zhang models to predict the local and overall thermal sensation and comfort.

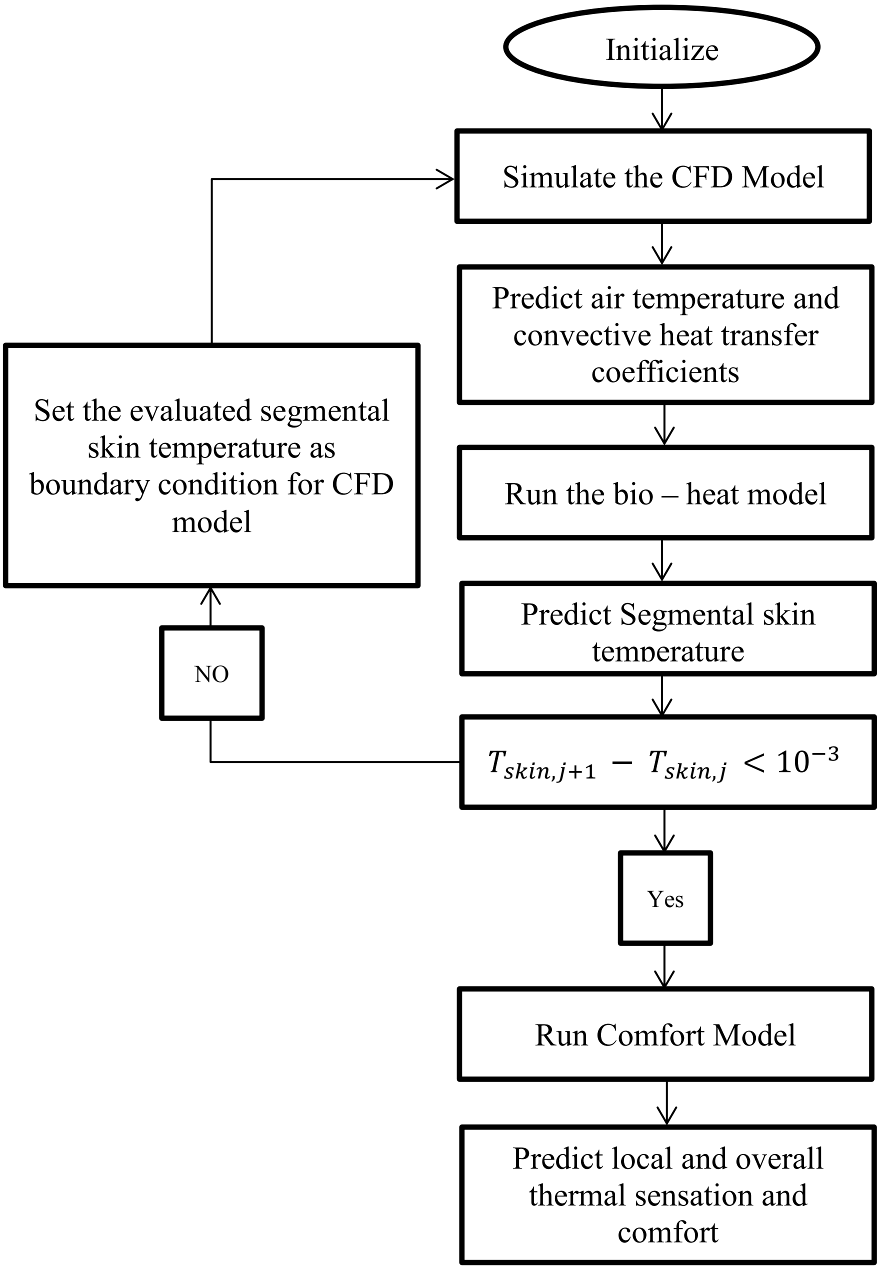


Figure 4: Flow chart for the coupling of the CFD model with the Bio-Heat and Comfort model

CHAPTER III

EXPERIMENTATAL VALIDATION OF (DV + CF) SYSTEM

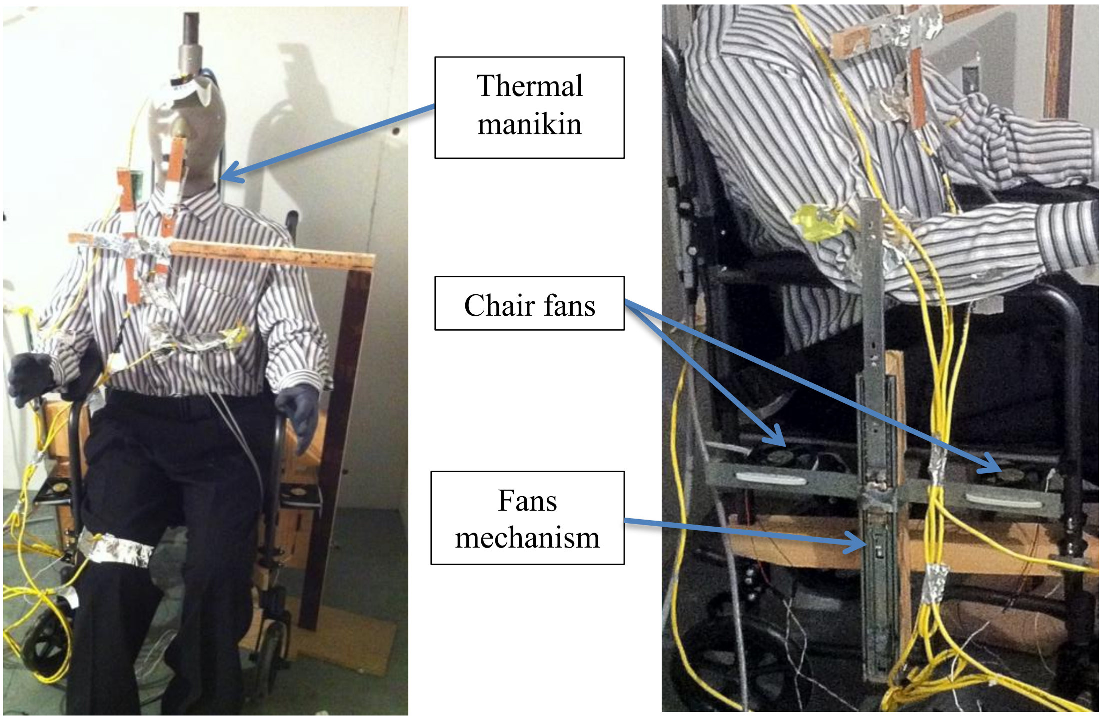
A. Experimental Setup and Measurements

In this study, an experimental validation is crucial to reveal the ability of the CFD model in predicting the performance of (DV + CF) system. The validation would be based on detailed comparison between the predicted and experimental data in the name of segmental surface temperature of occupant and ventilation effectiveness. Accordingly, three different sets of experiments will be conducted. The fans will be placed at two different levels and will supply constant flow rates, where their performance will be compared to a standalone DV system. The DV supply diffuser will be set to constant flow rate and temperature. The CO_2 source will be located near the occupant breathing level.

The experimental set-up consisted of a controlled climatic chamber ventilated by a DV system. The chamber was of inner dimensions of 2.5 × 2.75 × 2.8 m with supply and return grills cross-sectional area of 0.582 m (width) × 0.24 m (height) and 0.44 m (width) × 0.19 m (height) respectively. In this chamber, the thermal manikin "Newton" (see Fig. 5) manufactured by the Northwestern measured technology (Epoxy Thermal Manikin). This manikin is characterized by high performance of $\pm$ 0.1^{o}C temperature measurement and set-point control with a maximum heat flux of 700 W/m^2. "Newton" is subdivided into twenty different control zones where each zone could report the segmental surface temperature based on assigned constant heat flux. "Newton" is controlled through "ThermDAC" control software that is user-friendly Windows-based application providing all possibilities of control. On the manikin's chair, four fans were placed, two at each side, having a single degree of freedom allowing them to move vertically up and down leading to a height ranging between 20 and 50 cm above the floor as shown in Fig. 5b. The dimensions of the fans were 80 mm (length) × 80 mm (Width) × 25.4 mm (height). These fans have a maximum DC voltage of 12V and maximum power of 6.25 Watt. ABK precision 731A thermo anemometer was installed downstream of the fans. It is characterized by an accuracy of $\pm$ 2% of full scale for velocity measurements. The airflow through each fan was adjusted by varying the supply voltage using a 12V DC – Dimmer resulting in a flow rate ranging between 0 and 10 L/s/fan.

An internal load of 30 W/m^2 was generated in the room due to the presence of lighting and heat flux through the walls (no perfect insulated walls) resulting in 135 W, and thermal manikin generating 70 Watt.

For air quality assessment, a constant CO_2 dose was used representing the ambient passive contaminants. Sampling tests for CO_2 were conducted by measuring the CO_2 concentration using FIGARO CDM4161- CO_2 sensors having an accuracy of $\pm$20 ppm. The CO_2 sensors were connected to OMEGA DacPro data logger to store data and check stabilized residuals. The CO_2 sensors were located at three different positions, two at the breathing level, two at the exhaust, and one at the supply diffuser.



a) b)

Figure 5: Photos (a) Thermal manikin used in the experiment (b) Chair fan

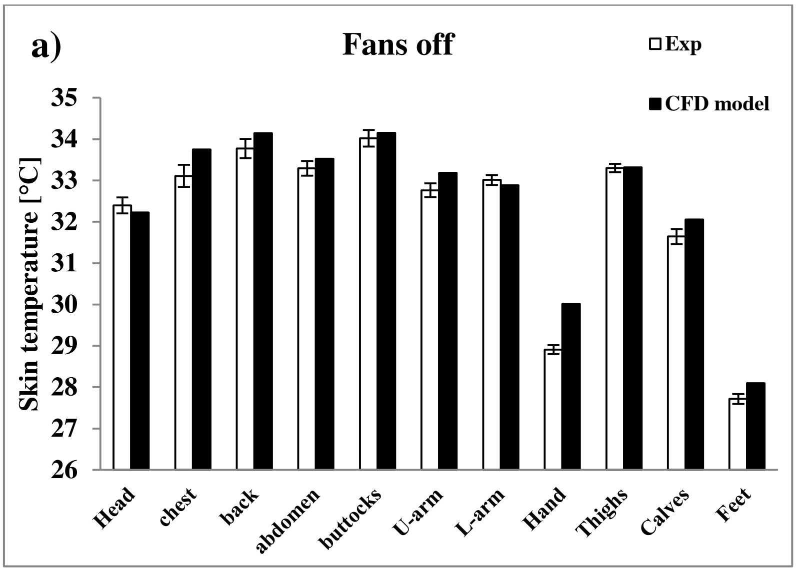
B. Experimental Protocol

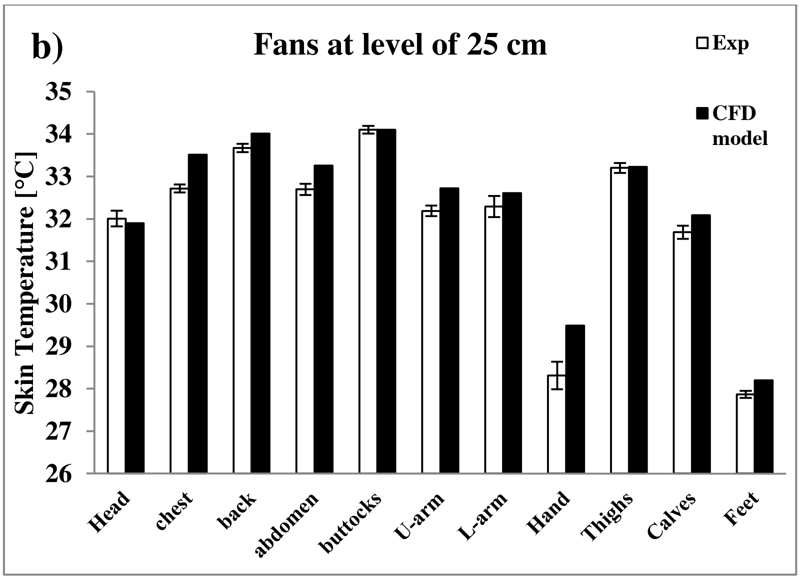
Prior to each set of measurements, the light was turned on; the thermal manikin was set to constant heat flux of 39 W/m^2 equally distributed over body segments. After attaining steady state conditions, the standalone DV system was turned on. The DV flow rate was set to 80 L/s (minimum available supply flow rate in the experimental chamber) with a supply temperature of 21.5^{o}C, corresponding to an air change rate of 14.96 per hour. The passive contaminant source supplying 2 L/min at room temperature and located at a level of 1 m above the floor was introduced into the room. The CO_2 concentration in the supplied fresh air was checked to be within the fresh air CO_2 limit ranging between 300 ppm and 500 ppm. The experiment was allowed to run for at least 1-2 hours where the readings of CO2 and segmental surface temperature fields were stabilized.

The performance of the standalone DV system was compared to (DV + CF) system. The fans were set to two different heights of 25 cm and 40 cm above the floor while supplying a total flow rate of 25 L/s. The same procedure was repeated until the readings of the CO_2 and segmental surface temperature fields were stabilized. Each experiment was repeated three times to ensure precision and accuracy. A total of nine experiments were conducted including the repeated ones.

C. Model Validation Results

Fig. 6 shows the predicted and experimental values of the segmental surface temperature for different fan configurations (a) fans off, (b) fans at level of 25 cm, and (c) fans at level of 40 cm. It is clear that the fans at the two heights reduced the segmental surface temperature mainly at the upper body parts (head, chest, abdomen, U-arm, L-arm, and hands). This is due to the improvement in the convective currents near the body segments. On the other hand, it is observed that the segmental surface temperature increased as the height of fans is increased. At 40 cm height compared to fans at a level of 25 cm, the surface temperature at the head and upper arm increased by 0.26oC and 0.11oC respectively. This could be justified by the principle of DV and fans operation. Due to buoyancy effects, the temperatures of air layers away from the floor level are hotter than those near the floor level. The fans placed at high level would draw air layers that are relatively warmer than those drawn by fans placed at low levels. This explains the increase in segmental surface temperature that was depicted as the position of fans was shifted from 25 cm to 40 cm above the floor. The results showed good agreement between predicted and measured segmental surface temperatures with a maximum relative error of 4.02% reported at the hands as shown in Fig. 6(a-c).





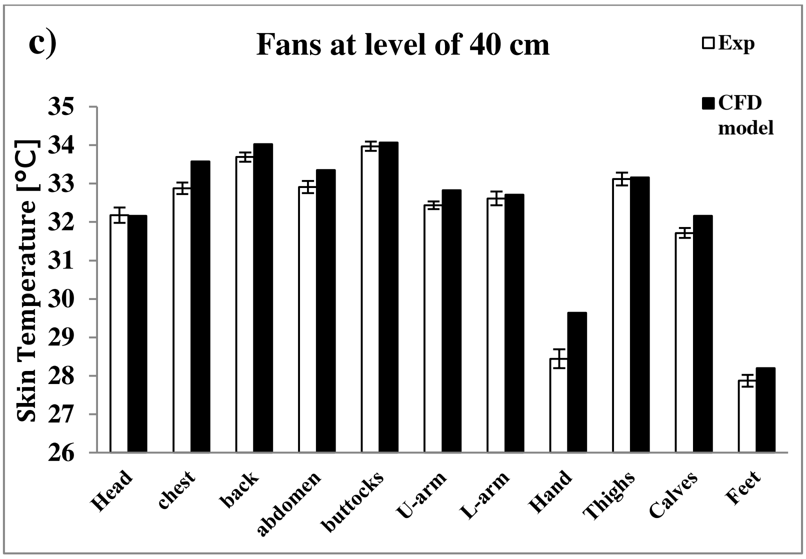


Figure 6: Segmental surface temperature validation: (a) Fans off (b) Fans at level of 25 cm, (c) Fans at level of 40 cm; Total fans flow rate of 25 L/s.

The air quality is assessed using the ventilation effectiveness(ε_v). Fig. 7 presents the predicted and experimental values of ventilation effectiveness for different fan configurations. The results reveal that the fans at the two heights improved the ventilation effectiveness to reach a maximum of 38% when fans are set at a level of 25 cm. This is due to the ability of fans in drawing fresh air from lower levels and blowing it upwards towards occupant's breathing zone. On the other hand, an increase in fans height above the floor resulted in a drop in ventilation effectiveness i.e. (38% versus 34%). The relatively contaminated air drawn by the high level fans and delivered to the breathing zone at high momentum could justify this decrease in ventilation effectiveness. The results showed good agreement between experimental and measured values with a relative error ranging from 3.54% to 5.84%.

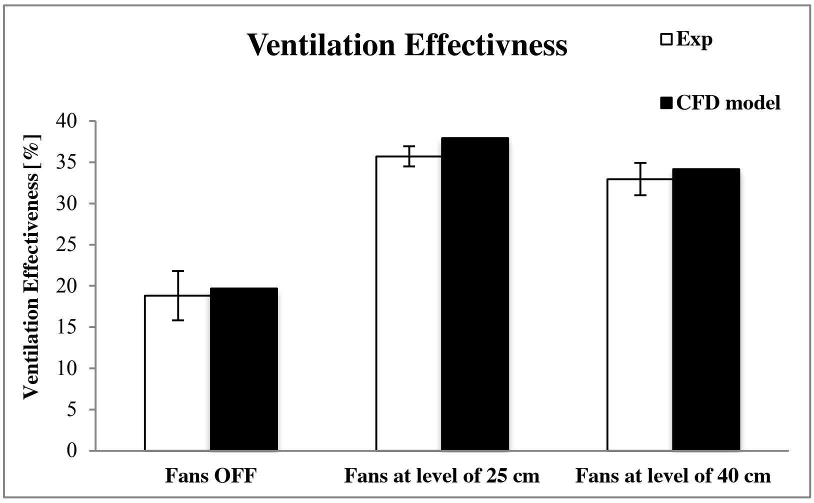


Figure 7: The measured and predicted ventilation effectiveness for different fan configurations

CHAPTER IV

PARAMATRIC STUDY

A. Parametric Study Description

The validated CFD model results showed a clear dependence of segmental skin temperature and IAQ on the variation of height of fans above the floor for the two considered heights. Accordingly, finding the optimum height is of interest. To this aim, a typical office space is studied at inner dimensions 3.4 m × 3.4 m × 2.6 m with supply and return grills cross-sectional area of 3 m (width) × 0.1 m (height) and 0.5 m (width) × 0.2 m (height) respectively as shown in Fig. 2. The room air temperature is set to 25^{o}C. The passive contaminant source is assumed to produce 2 L/min of CO2 at room temperature. The load in the room is set to 40 W/m^2 representing a typical maximum load removed by a DV system as reported by Yuan et al. (X.Yuan; Q. Chen and L. R. Glicksman, 1998) and Vivian et al. (L.Vivian; B. Rohini; M. Michelle; V. Edward and M. Mike, 2002). This load was generated as a result of typical lighting load of 10 W/m^2 (ASHRAE Handbook 2009); desktop computer of 93 W (ASHRAE Handbook 2009); an office occupant of 70 W under sedentary activity (ASHRAE Handbook 2009); and the remaining load is assumed generated by the walls. The operating conditions are selected such that they provide a stratification height of 1.1 m which is the minimum stratification height recommended by ASHRAE standard (ASHRAE Handbook 2009) and are capable of removing the load generated in the room.. Accordingly, the supplied fresh air was set to 21^{o}C and 47.5 L/s. The selection of the optimized height for the fans will be based on the assessment of three different parameters including total fans' flow rate$(\dot{Q})$, fans' height above the floor(H_{fs}), and passive contaminant height above the floor(H_s). Table 2 presents the simulated cases used in the parametric study. Furthermore, an energy analysis will be conducted to check the energy savings provided by the novel system when compared to a standalone DV system providing approximately same thermal comfort and IAQ levels.

Table 2: Simulated cases used in parametric study

Simulation Cases	Passive contaminant source height [m]	Height of fans [cm]	Total flow rate of fans [L/s]
Set 1	1	25	7
			12
			16
		40	7
			12
			16
Set 2	0.3	25	7
			12
			16
		40	7
			12
			16

B. Results and Description

1. Effect of Fans Height and Flow Rate

a. Air Quality

Figure 8 shows a comparison between different configurations of fans on the basis of ventilation effectiveness. A standalone DV system (Fans off) resulted in a ventilation effectiveness of 49%. For fans at a level of 25 cm, as the flow rate increases, the ventilation effectiveness also increases to reach a maximum of 55% at 12 L/s and then decreases to 52% at 16 L/s. Unlike the case when fans are at a level of 40 cm, the ventilation effectiveness reached a value of 53.4% at low flow rate of 7 L/s which is higher than the value reached at a level of 25 cm for the same flow rate. This effectiveness is further reduced as flow rate increases to reach a value of 45.4% at 16 L/s indicating deteriorated IAQ compared to a standalone DV system.

The use of fans resulted in an increase in ventilation effectiveness since fans delivered fresh air from low levels towards the occupant's breathing zone. For low flow rate of fans, placing the fans at higher level increased the momentum of the drawn fresh air to reach the breathing zone resulting in good IAQ. On the other hand, placing the fans at low level while increasing their flow rate resulted in more fresh air delivered to the breathing zone at high momentum. However, excessive increase in flow rate would increase the tendency of mixing with more contaminated air resulting in decrease in ventilation effectiveness. For high level fans, the increase in flow rate of fans resulted in more contaminated air drawn from the middle zone, delivered to the breathing zone at high momentum, and mixed with more contaminated air resulting in low and degraded IAQ. Evidently, the use of fans improved ventilation effectiveness by 12.24% in the best scenario (fans at a level of 25 cm while supplying 12 L/s) and reduced effectiveness by 7.35% in the worst scenario (fans at a level of 40 cm while supplying 16 L/s).

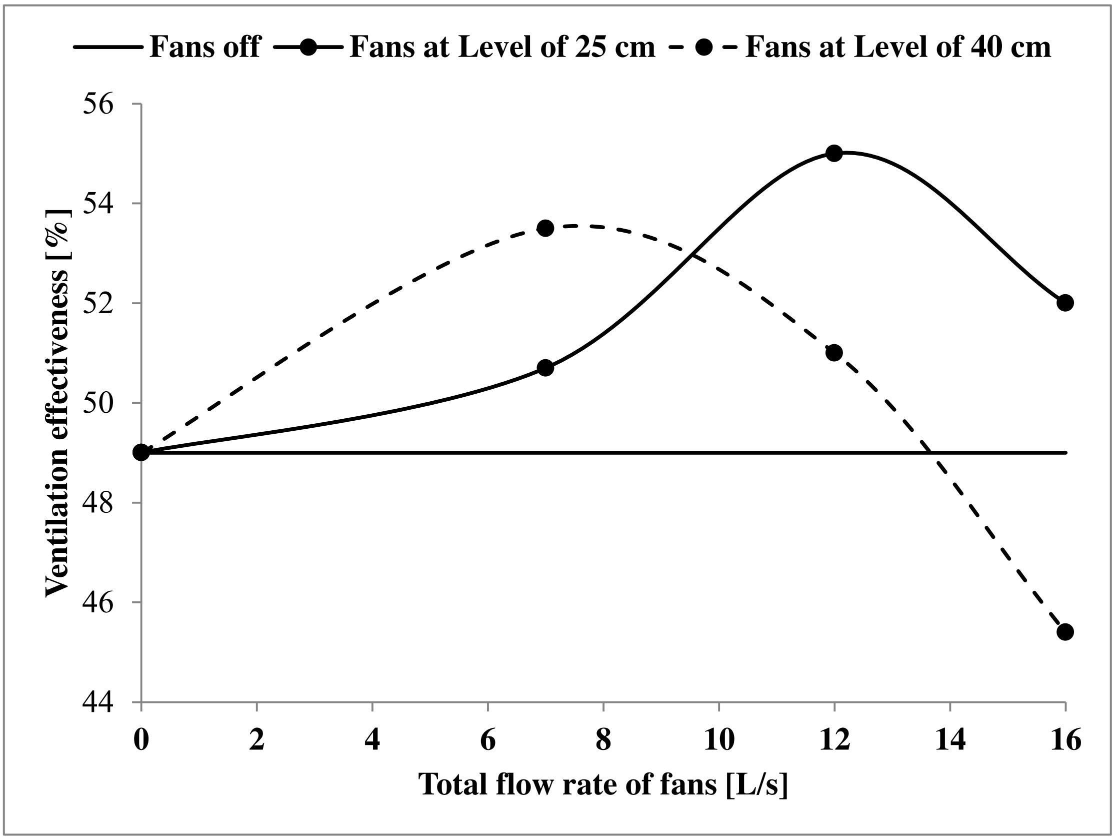
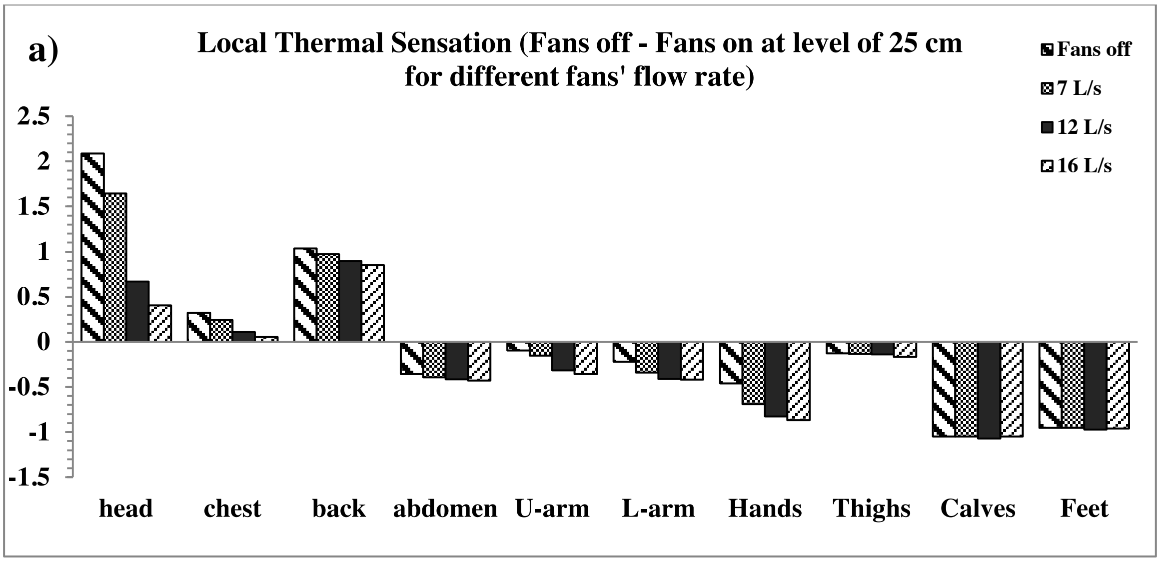
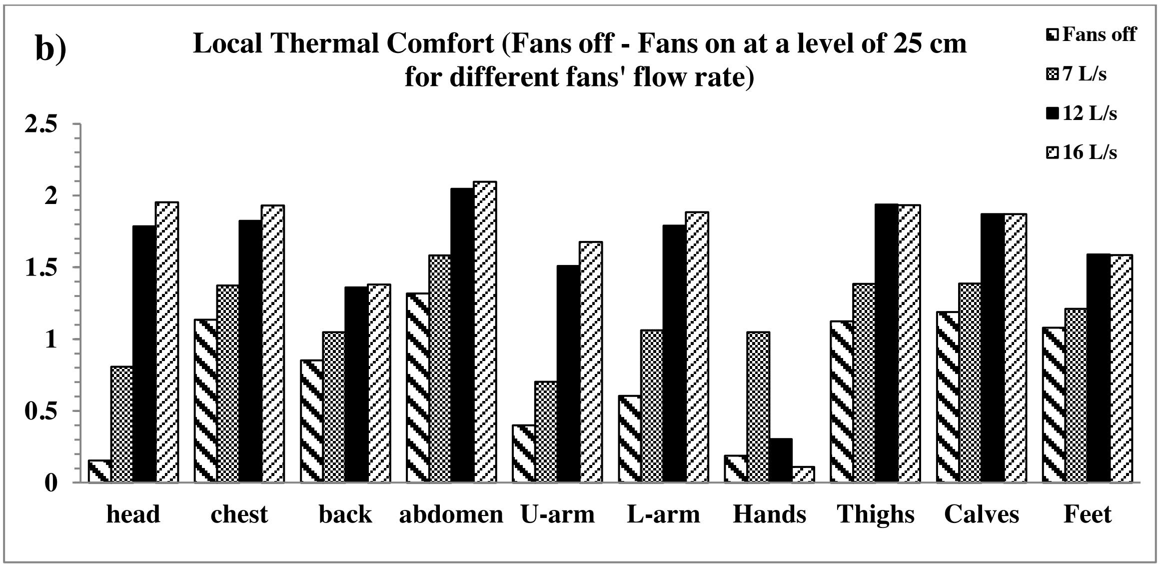


Figure 8: Comparison of ventilation effectiveness at different fan levels and flow rates at 47.5 L/s DV supply flow rate

b. Thermal Comfort

Figure 9 illustrates (a) local thermal sensation (LTS) and (b) local thermal comfort (LTC) for Fans off – Fans on at level of 25 cm for different flow rate of fans and (c) LTS and (d) LTC for Fans off – Fans on at level of 40 cm for different flow rate of fans. Compared to a standalone DV system (Fans off), the upper body parts (head, chest, abdomen, u-arm, l-arm, and hands) are the segments mostly affected by fans operation. This is clearly illustrated by the drop in LTS accompanied by an increase in LTC as shown in Fig. 9 (a)-(d). However, the use of fans at different heights and flow rates resulted in variation in occupant thermal response.





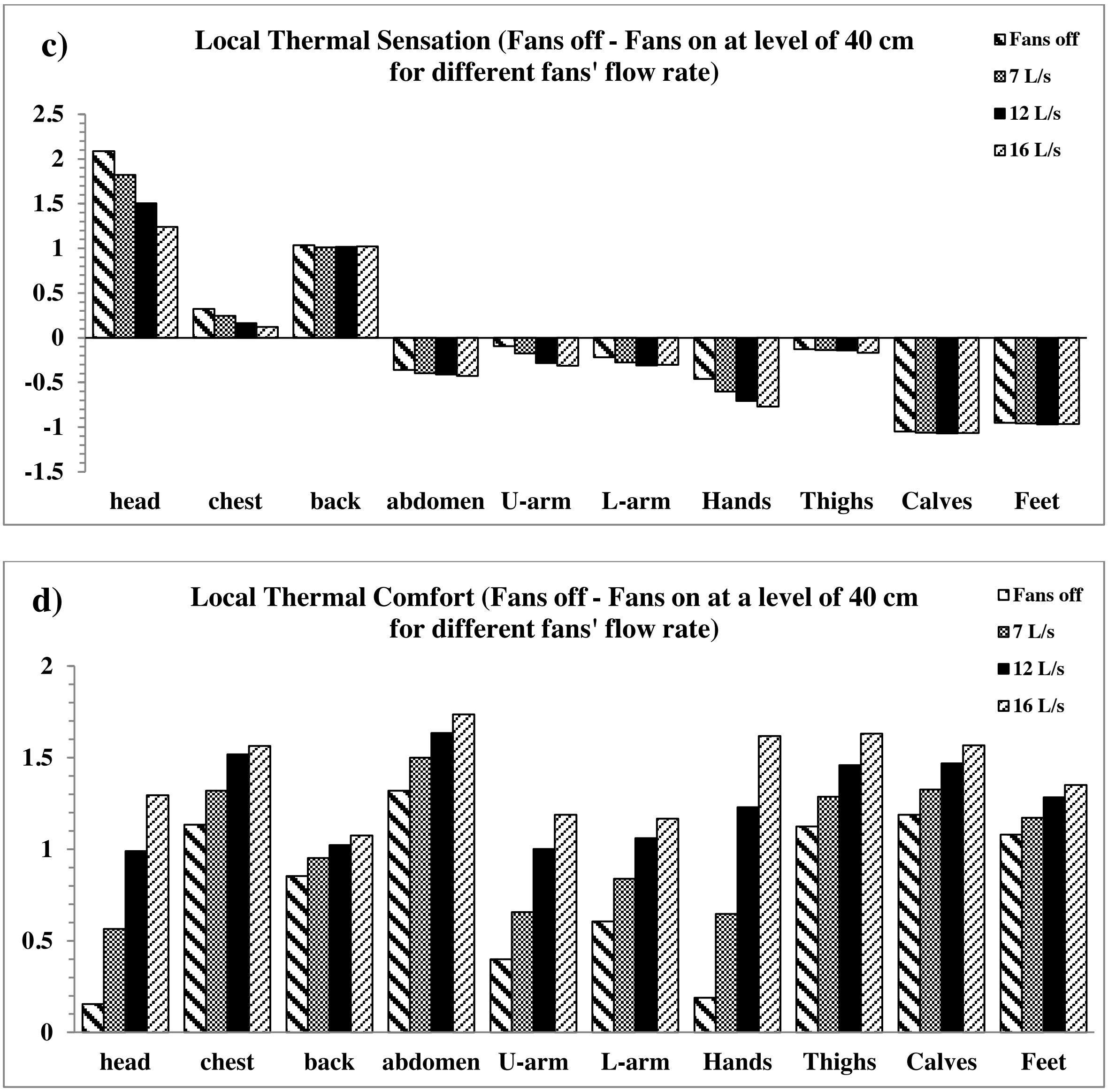


Figure 9: Local thermal a) and c) sensation and b) and d) comfort for different fans configurations.

Table 3: Comparison of TS and TC at different fans configuration at 47.5 L/s DV supply flow rate

Fans off		Total flow rate of fans [L/s]	Fans at level of 25 cm		Fans at level of 40 cm	
TS	TC		TS	TC	TS	TC
0.82	0.8	7	0.59	1.19	0.68	1.08
		12	0.17	1.24	0.53	1.32
		16	0.14	1.18	0.43	1.4

Table 3 presents the overall thermal sensation (TS) and overall thermal comfort (TC) for different fan configurations that are simulated. For the case of fans off, the TC of an office occupant reached 0.8. For the case of 25 cm, as the flow rate is increased, TC increases to reach a maximum of 1.24 at 12 L/s and then decreases to reach a value of 1.18 at 16 L/s. However, for the case of 40 cm, as the flow rate is increased, the TC continues increasing to reach a value of 1.4 at 16 L/s. In both locations of fans, the increase in TC attributed to the enhancement of the convective currents near the occupant. Compared to a standalone DV system, at low flow rate of 7 L/s, the enhancement in TC is at a lower rate when fans are shifted from a level of 25 cm to 40 cm. This could be justified by the ability of low level fans to draw cooler air at favorable thermal sensation (0.59 versus 0.68). Conversely, an improvement in TC is observed at high flow rates as the fans are shifted from 25 cm to 40 cm. The low values of TC reported for low level fans could be attributed to fans drawing cooler air and blowing it at high velocity resulting in thermal draft sensation mainly at the hands (see Fig. 9 (a)-(b)). However, the increase in fans velocity overcame the hot air layers drawn by high level fans providing comfortable thermal conditions. Evidently, compared to a standalone DV system, the fans provided the occupant with comfortable environment resulting in low improvement in TC of 35% (fans at a level of 40 cm while supplying 7 L/s) and high improvement in TC of 75% (fans at a level of 40 cm while supplying 16 L/s).

c. Height Optimization

As stated in the previous sections, the interrelation between height and flow rate of fans affected the performance of the novel system. Therefore, an optimum height that could provide the best combination of TC and IAQ was highly recommended. Accordingly, the two different indices (TC and ventilation effectiveness) are plotted for three different flow rates where the compromised height is selected based on the intersection between the two indices. Fig. 10 shows the criterion for height selection when fans are supplying 16 L/s. Table 4 illustrates the selected heights and their corresponding TC, enhancements in TC, ventilation effectiveness, and enhancements in ventilation effectiveness when compared to the case when fans are off.

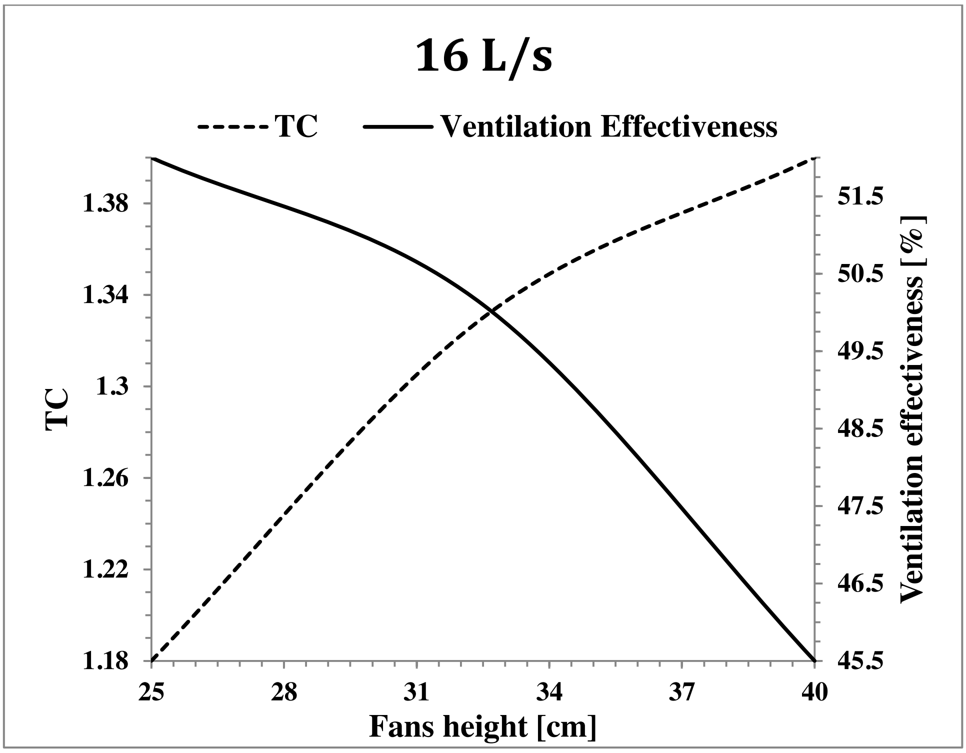


Figure 10: Selected Height for Fans Supplying 16 L/s

Table 4: The selected heights and their corresponding TC, enhancements in TC, ventilation effectiveness, and enhancements in ventilation effectiveness along with the energy savings for three different fans flow rate (7L/s, 12 L/s, and 16 L/s) for a source at a level of 1 m.

Cases with source at 1 m						
Total flow rate of fans [L/s]	Optimum height of fans [cm]	TC	Enhancement in TC [%]	ε_v [%]	Enhancement in ε_v [%]	Energy savings [%]
7	30.2	1.148	43.5	52.4	6.9	9.3
12	30.9	1.407	75.9	54.4	10.9	20.6
16	32.7	1.332	66.5	50	2.1	1.1

To check the energy savings of the proposed novel system, the energy consumed consumption by (DV + CF) system is compared to a standalone DV system providing approximately the same TC and air quality. Accordingly, a chiller with an average COP of 3.5 is selected. Fresh air at 32℃ and 70% RH supplied by a fan is cooled and dehumidified by exchanging heat with a cooling coil provided by the chiller where it is extracted from the room by an exhaust fan. The electrical power consumed by the chiller would be added to the electrical power consumed by the fans. The chair-mounted fans consumed a power of 25 W. The energy savings for the selected heights was evaluated and tabulated as shown in Table 4. Accordingly, for the current thermal environment, it is recommended to place the fans at 30.9 cm while supplying 12 L/s and providing 20.6% energy savings.

2. *Effect of source height*

The purpose of this study is to check the effect of fans when the passive contaminant source height is varied. For the current study, fan configurations along with the DV supply flow rate and temperature are held similar to those used in the previous sections; however, the passive contaminant source is shifted to a level of 0.3 m above the floor. As expected, changing the source height would only affect the air quality in the room. Fig. 11 presents a comparison based on ventilation effectiveness between different fan configurations for a source located at 0.3 m.

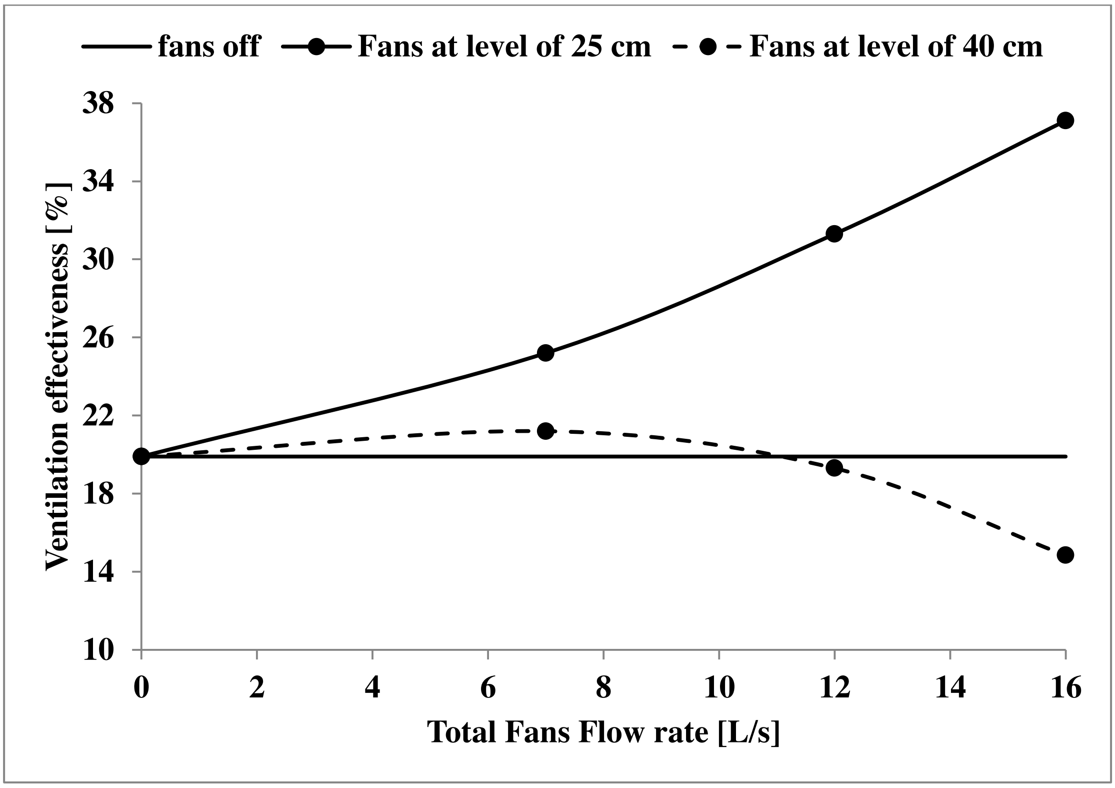


Figure 11: Comparison of ventilation effectiveness at different fans levels and fans flow rates at 0.3 m source location

Comparing Fig. 8 and Fig. 11; for the case of fans off, the ventilation effectiveness decreases to reach a value of 19.9%, when the source level is shifted from 1 m to 0.3 m above the floor. The layer near the floor level became highly contaminated when the source is located at 0.3 m, which in turn delivered higher contaminated air to the breathing zone resulting in degraded IAQ. On the other hand, for a source located at 0.3 m, when fans are positioned at 40 cm above the floor and their flow rate is increased, the ventilation effectiveness follows the same trend depicted when the source is located at 1 m. For fans at a level of 25 cm and, unlike the case of 1 m source, as the fans flow rate is increased, the ventilation effectiveness continues increasing to reach a value of 37.1% at 16 L/s. The observed difference is due to the different interactions between the drawn air by fans and the contaminants produced by the passive source.

Following the same procedure used in section 4.2.1.3, the selected heights and their energy savings for different flow rates of fans at a source level of 0.3 m is illustrated in Table 5. Accordingly, for the current thermal environment, it is recommended to place the fans at a level of 30.1 cm while supplying 12 L/s and providing 11.6% energy savings.

Table 5: The selected heights and their corresponding TC, enhancements in TC, ventilation effectiveness, and enhancements in ventilation effectiveness along with the energy savings for three different fans flow rate (7L/s, 12 L/s, and 16 L/s) for a source at a level of 0.3 m.

Cases with source at 0.3 m						
Total flow rate of fans [L/s]	Optimum height of fans [cm]	TC	Enhancement in TC [%]	ε_v [%]	Enhancement in ε_v [%]	Energy savings [%]
7	26.5	1.178	47.3	25.3	26.9	6.5
12	30.1	1.39	74	28.3	42.2	11.6
16	31.2	1.309	63.6	27.9	40.2	10.2

C. Conclusion

A validated CFD model is integrated with a validated bio-heat model to select the optimized height that can provide acceptable TC and good IAQ at the least energy consumption. This assessment is based on varying fans flow rate, fans height above floor, and contaminant source height above floor. It was found that the fans could provide the highest TC (1.4) when set at a level of 40 cm above the floor while supplying 16 L/s. On the other hand, the concentration of CO_2 at the breathing zone was shown to be affected by source location. For a source located at 1m, it was found that the fans positioned at 25 cm and supplying 12 L/s could provide the maximum improvement in ventilation effectiveness (55%). For a source located at 0.3 m, it was found that the fans positioned at 25 cm and supplying 16 L/s could provide maximum enhancement in ventilation effectiveness (37.1%). The interrelation between various parameters and their effect on TC and IAQ led to an evaluation of an optimized height that could provide a compromise between best TC and IAQ. Accordingly, for a source located at 1 m, it was found that the fans positioned at 30.9 cm and supplying 12 L/s could provide the best combination of TC and IAQ at an energy savings of 20.6%. Similarly, for a source located at 0.3 m, it was found that the fans positioned at 30.1 cm and supplying 12 L/s could provide the best combination of TC and IAQ at an energy savings of 11.6%.

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