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Assessing the performance of the airflow window for ventilation and thermal comfort in office rooms

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Abstract In the present study performance of an airflow window in removing contaminants as well as providing thermal comfort for the occupants was investigated. Both natural/mixed ventilation methods were studied and the full heating load as well as contaminant sources in the office rooms considered. Then, the local and average temperature, relative humidity, velocity as well as CO_2 and dust concentration were extracted from simulation results and compared to criteria in international ventilation standards. It was found that except in the big room having 8 m $\times$ 6 m flooring, natural ventilation from the airflow window can satisfy the thermal and relative humidity conditions in the international ventilation standard except for the American Society of Heating, Refrigerating and Air-Conditioning Engineers. However, the thermal comfort in the room which was measured by extended predicted mean vote could not be achieved when the window operates in the natural ventilation mode, even with a 0.4 m height opening in the small (3 m $\times$ 4 m) room. Finally, results indicated that the airflow ventilation system installed in small and medium offices operation can provide indoor condition in the ventilation standard either in natural/mixed operation mode consuming less energy than the traditional heating, ventilation, and air conditioning. Besides, the airflow system not only was not able to provide thermal comfort condition in the big office but also its application was not economically feasible.

Keywords: HVAC; Natural ventilation; Airflow window; Thermal comfort; Contaminant removal

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Acronyms

ASHRAE	–	American Society of Heating, Refrigerating and Air-Conditioning Engineers
CFD	–	computational fluid dynamics
HVAC	–	heating, ventilation, and air conditioning
PMV	–	mean predicted vote
RH	–	relative humidity
UFAD	–	underfloor air distribution

1 Introduction

Green-house emissions are the primary source of climate change which is believed to be the most pressing challenge of the world at present. At the moment only two different ways are possible to reduce energy consumption and carbon emission, that is by reducing energy consumption or getting benefits of renewable energy sources such as wind or hydropower. As the statistics indicate, more than 30% of the world's energy consumption belongs to the office and residential section [1]. It also, predicted that that the energy consumption in buildings is growing and is expected to grow dynamically beside the emission production. To overcome this issue, many researchers attempted to optimize energy consumption in heating, ventilation, and air conditioning (HVAC) systems utilizing natural and renewable energy resources with almost zero carbon footprints [2–5].

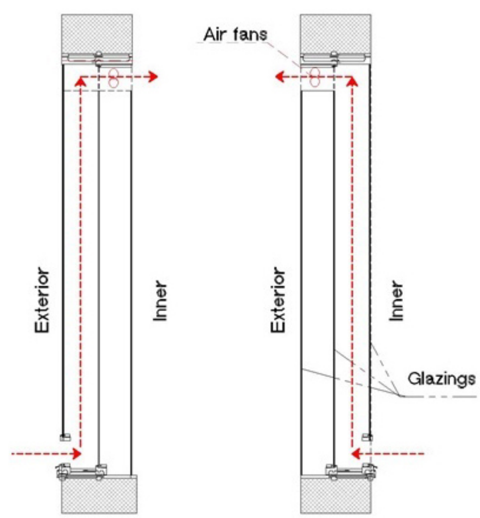
A multivariable generalized predictive controller for regulating temperature and humidity in HVAC system, by considering the cost function has been proposed in [2], and an intelligent approach to optimize the energy consumption of buildings with neural network system in [3]. Their model confirmed the accurate prediction of the energy consumption of the building and show that the optimization of the system can reduce energy by 34% in dried conditions. Awbi investigated the air movement and CO_2 concentration in an office room and atrium [4]. According to the results of their simulations, using natural ventilation, it is possible to reduce CO_2 concentration to an acceptable level. They also reported that ventilation through openings is capable of satisfying an adequate comfort level in both studied building types. Stavrakakis *et al.* [5] evaluated the potential capability of natural cross-ventilation with openings at non-symmetrical locations for providing thermal comfort in a full-scale test chamber. Their results showed that the non-symmetrical location of opening leads to a smaller temperature difference in the occupied zone and minimizes the local drought.

Bangalee *et al.* [6] examined wind-driven single-sided natural ventilation through multiple windows. In their 3D simulation, the Reynolds-averaged Navier–Stokes (RANS) equation was solved numerically along with the renormalisation group (RNG) k-ε turbulence model. It has been found that if the ambient condition, weather and surroundings are satisfactory, the performance of multiple windows would the better. Dascalaki *et al.* [7] investigated the accuracy of predictions from the various ventilation models with experimental results.

The airflow characteristics and pollutants in a cross-ventilated building dispersion were investigated in previous studies. Liu *et al.* [8] conducted an experimental study in a scaled multi-room chamber. Both mean velocities and root-mean square (RMS) velocity fluctuations were quantitatively studied to reveal the flow characteristics under different cross-ventilation paths. In this paper, the concentration distributions under different indoor source locations were quantitatively presented. Hu *et al.* [9] studied the performance of phase change material (PCM) enhanced ventilated window for cooling and heating. They concluded that this type of window is more effective in buildings in a climate with high daily outdoor air temperature differences and higher solar radiation level. Effect of control system and level of automation on the comfort level in natural ventilation were investigated by Chen *et al.* [10]. They analyzed increasing levels of automation in natural ventilation and concluded that a fully automatic window/HVAC control system with the model predictive control (MPC) shows the best performance both in terms of thermal comfort and energy consumption.

The impact of outdoor air quality on the natural ventilation usage of commercial buildings in the US was investigated by Chen *et al.* [11]. According to their study, the natural ventilation reduction varies from 5% to 70% across the US and most cities have less than 20% reduction except for four cities in the US. They also developed an advanced indoor air pollutant control with emulators. Numerical investigation for the indoor flow structure, crossflow ventilation characteristics, and thermal comfort of a room with various outlet opening has been proposed in [12]. Their results indicated that the room outlet mass flow rate increased up to 26% *via* reducing the room outlet elevation. The pros and cons of a smart window-integrated ventilation with a centralized HVAC one for thermal comfort and energy consumption in a typical office for various environmental temperatures has been compared in [13]. The performance of a naturally ventilated double-skin facade in buildings has been assessed in Tao *et al.* [14]. They concluded that window dimensions and size are the most important parameters in the thermal performance of the window.

Airflow window is among the most efficient methods of natural ventilation used. A schematic of the airflow window is shown in Fig. 1. In this ventilation method, the window acts as both inlet and exhaust of the room. In the summer condition (Fig. 1a) air enters from the top opening of the window and the lower opening of the window is used as an exhaust for the room. The inlet and outlet of the window are exchanged in the winter mode (Fig. 1b). Natural ventilation plays a key role in reducing energy consumption in indoor HVAC systems and can bring about a lot of environmental and economic benefits. When operating in the natural ventilation mode, both wind velocity and buoyancy contribute to the room ventilation. In most cases, the flow induced by the wind at the level of the window is prevailing, and the effect of buoyancy is negligible. Nevertheless, in airflow windows with a height of several meters, natural ventilation derived from the density difference of the air between the two height levels of the building is equally important. As mentioned by other researchers, the main drawback of natural ventilation resources is that, unlike the forced ventilation method, natural resources are not reliable and controllable to the same degree [3]. Though, to solve the controllability problem of the natural



(a) (b)

Figure 1: Schematic of airflow window: a) supply mode (summer), b) exhaust mode (winter).

ventilation as well as getting the advantages of natural ventilation systems such as energy-saving, mixed ventilation methods are proposed. All mixed ventilation techniques rely on the fact that in order to ensure providing a safe and comfortable environment for the occupants forced ventilation should be superimposed on top of natural ventilation. For instance, in the airflow system, when natural ventilation forces (wind and buoyancy) are not strong enough to provide a pleasant indoor condition for applicant mixed ventilation techniques should be employed.

Computational fluid dynamics (CFD) offers a cost-effective and fast alternative for full-scale testing in which the parameters of the ventilation system can be changed quickly without having to worry about the cost and time needed for experimental studies. Having these advantages many researchers used CFD for the simulation of HVAC systems [8, 15]. In the present study, CFD was employed to investigate the ventilation characteristics of the office room with airflow windows.

2 Significance of the study

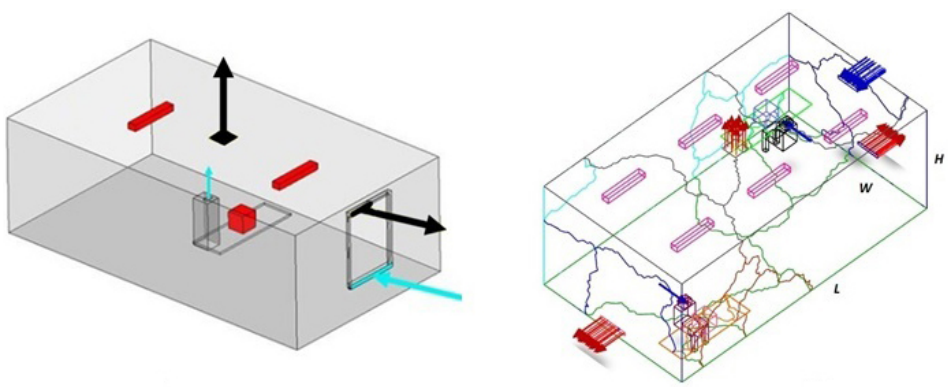
Due to the shortage of renewable energy sources and environment-related issues such as global warming, the tendency to replace traditional building ventilation resources is surging rapidly in recent years. However, the potential of employing natural resources to ventilate residential and commercial buildings is not fully investigated in the literature, especially researches concerning forced and natural and forced ventilation from airflow window are rare. For that, the capability of airflow window to provide acceptable indoor air quality and thermal comfort condition in office rooms of different size is investigated in the present study. Moreover, for the first time, the data obtained for different operating conditions of the airflow window was compared with the ventilation standard in different countries.

Unlike other studies conducted previously, there is no auxiliary room ventilation other than airflow window operating either in natural or mixed ventilation mode in the offices studied in the current research. For the location of the office, it is assumed that the studied office rooms are located in Istanbul city in the mid of the hottest month of the year. To include weather condition in the study, weather data was directly adapted from the region and used in simulations. The main objective of this study is to help understand the potentials, applications and limitations of office rooms with different sizes utilizing the airflow window system alone.

3 Mathematical model

3.1 Problem description

An outline of the studied office room is illustrated in Fig. 2a. The office is ventilated using an airflow window installed 1.1 m from the floor. The airflow window consists of two parts. On half of the windows is located indoor while the other part installed outdoor (respectively called interior and exterior windows hereinafter). It was assumed that the window is operating in the summer mode (Fig. 1a) so that the fresh enters the space from the top inlet of the interior windows (inlet opening) and the return air is exhausted to the environment through the bottom opening (window outlet opening). Moreover, to facilitate ventilation of the office, an outlet opening is located at the center of the ceiling whose dimensions is 0.16 m × 0.16 m whereas on the other side of the room the door is installed which has an opening that can be opened and closed when necessary. The overall height of the airflow windows and width of inlet and outlet openings are fixed but the height of the opening is varied from 0.1 m to 0.4 m and its effect on indoor air quality and thermal comfort of the office is investigated. Three different office types are studied in the present study called: small, medium, and big office room. Dimensions and characteristics of each room are given in Table 1.



(a) (b)

Figure 2: Outline of different sizes of office rooms studied: a) schematic view, b) sizes.

It is also worth mentioning that all dimensions and all boundary conditions of the studied offices are assumed to be identical to those used in [16].

Table 1: Parameters of the studied offices (in m).

Name	Size (in m)			No. of occupants	No. of lamps	No. of personal computers and tables
	Length	Width	Height			
Small	3	4	3.2	1	2	1
Medium	6	4	3.2	1	4	1
Big	8	4.5	3.2	2	6	2

Location, dimensions and boundary conditions for human and objects in the room are given in Table 2.

Table 2: Location and dimensions of objects in the room.

Item	Length (m)	Width (m)	Height (m)	Location (m)			Heat (W)
	Δx	Δy	Δz	x	y	z	Q
Occupant 1	0.4	0.35	1.1	0.81	1.115	0.4	75
Occupant 2	0.4	0.35	1.1	5.77	2.88	0.4	75
Computer 1	0.4	0.4	0.4	0.4	2.55	0.55	108.5
Computer 2	0.4	0.4	0.4	5.77	2.55	0.55	173.5
Table 1	2.33	0.75	0.01	2.23	2.88	0.74	0.0
Table 2	2.33	0.75	0.01	5.77	2.88	0.74	0.0
Lamp 1	0.2	1.2	0.15	1.03	0.16	2.18	34
Lamp 2	0.2	1.2	0.15	2.33	0.16	2.18	34
Lamp 3	0.2	1.2	0.15	3.61	0.16	2.18	34
Lamp 4	0.2	1.2	0.15	1.03	2.29	2.18	34
Lamp 5	0.2	1.2	0.15	2.33	2.29	2.18	34
Lamp 6	0.2	1.2	0.15	3.61	2.29	2.18	34

Please be noted that the dimensions mentioned here are for the big office. In the small and medium offices the length, width, and height of the objects as well thermal loads are similar with the big room, but the number of objects (see Table 1) and their location differ. In these rooms, all objects including occupant, computer, and table are assumed to be disturbed in the room symmetrically. A realistic render of the airflow window is shown in Fig. 3.

Figure 3: Enlarged view of the airflow window.

3.2 Notation of convection cases

To get an insight into the windows' performance and find the optimum working condition for a combination of studied parameters, more than 100 CFD instances of the studied offices are created, simulated and analyzed. So, in order to simplify the process of categorizing utilization of a naming system for referring to each case seems to be necessary. For that, a system of convection notation involving one letter plus three numbers and a number after the dash is used. The letters S, M, and B denote the small, medium and big offices, respectively. The next three numbers show the air condition at the window inlet (Table 3). The first number indicates the range of inlet temperature, the second one denotes the range of inlet velocity and the third number is for the relative humidity (RH) of the inlet. The number after the dash is reserved for the height of inlet and outlet of the windows which is 1, 2, 3, and 4 for 0.1, 0.2, 0.3, and 0.4 m openings respectively and was removed where the window height is mentioned explicitly.

In addition to the naming convections mentioned that are used above for mixed ventilation, a simpler naming method is used for natural convection cases. When the natural ventilating is intended, the name begins

Table 3: Abbreviations of naming convention.

Inlet temperature		Inlet velocity		Inlet relative humidity	
Value (°C)	Digital designation	Value (m/s)	Digital designation	Value (%)	Digital designation
24	0	0.2459	0	65	0
20	1	0.5000	1	60	1
22	2	1.0000	2	55	2
–	–	1.5000	3	50	3

with N followed by a letter that stands for the size of the room. The next number denotes the windows opening height and is the same as above. It is also worth mentioning that this naming convention only employed for referring to natural and mixed ventilation cases, and when required, forced ventilation cases are mentioned explicitly.

3.3 Method of study

In order to study HVAC and thermal comfort in the office rooms, computational fluid dynamics were used which, is the most common tool in simulation flow and heat transfer problem including those encountered in buildings. To simulate weather conditions on the ventilation in the building the weather data are taken from the Istanbul weather station and used for calculation of boundary conditions. Besides, to ensure the validity of the numerical solution results obtained from CFD code were compared with the experimental data available in the literature showing a good agreement.

3.3.1 Governing equations

The flow and heat transfer model in the simulations is assumed to be steady, three-dimensional and incompressible. The fluid is Newtonian having constant properties. Since the studied problem is turbulent, the governing equations for the problem include Reynolds averaged Navier–Stokes equations and energy conversation equation. The effect of buoyancy forces on the velocity field is inserted into the momentum conservation equation through the Boussinesq approximation which is based on the assumption that fluid density is independent of temperature expect in the buoyancy force term. Moreover, vapor and contaminant transport in the air is modelled using the convection-diffusion equation with the diffusion coefficients adapted from the literature. Due to the nature of the flow in the room, all mentioned equations should be coupled with equations for turbulence transport.

3.3.2 Mesh generation and independency analysis

Meshing is one the most important parts of a numerical study and the accuracy of the final results depend on the quality of meshes. In the present study, a multiblock structured mesh was generated for studied office rooms. This was achieved by dividing the room into rectangular zones. An example of mesh generated for the medium office is shown in Fig. 4.

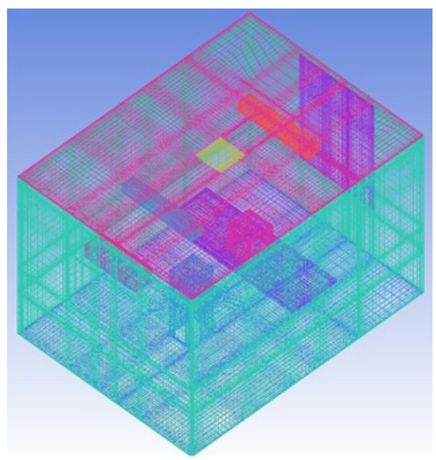


Figure 4: Mesh generated.

In order to capture large velocity and temperature gradients near walls and heating loads, the grid is refined in this region. Furthermore, to ensure that the solution is independent of mesh resolution, mesh independence analysis was carried out. For that, average temperature and relative humidity were monitored during the mesh refinement process. As the number of elements in the mesh increase until no change in the monitored parameters observed. The results of the mesh independence study are shown in Fig. 5. In addition, for successful implementing of the turbulence model controlling y^+ (nondimensionalized wall normal distance) near walls is very important. Since in this study k-ε turbulence model was used, wall y^+ values are kept between 50 and 100. Due to accuracy and flexibility, scalable wall function wall used for viscous sublayer modelling.

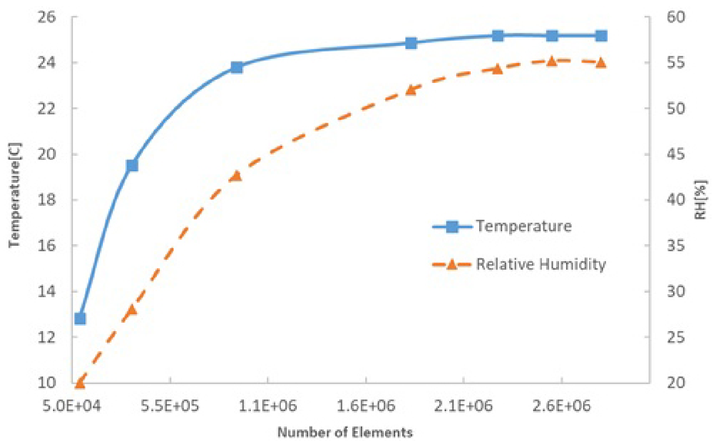


Figure 5: Grid independence study for a number of mesh elements.

It could be concluded from the figure that when using 2 500 000 elements the solution is independent of the mesh resolution. Hence this number of mesh elements was chosen for simulations.

3.3.3 Turbulence and radiation modelling

In ventilated rooms, turbulence is generated for either forced flow or buoyancy force. Therefore, choosing an appropriate turbulence model is of prime importance for the success of the CFD approach, however, resource cost requirements must be borne in mind. As the other researchers [16, 17] already mentioned, RNG k-ε or realizable k-ε model delivers acceptable results for HVAC simulation while consuming a reasonable amount of system resources. Hence, in the the present study, to account for turbulent fluctuations RNG model is chosen. Below are the constants for RNG k-ε turbulence model $C_{\varepsilon 2} = 1.68$ derived from experiments:

$$C_{\varepsilon 1} = 1.42; \qquad C_{\mu} = 0.0845; \qquad \mu_0 = 4.38; \qquad \beta = 0.012;$$
$$\sigma_{\varepsilon} = \sigma_k = 0.7194; \qquad C_{\varepsilon 2} = 1.68;$$

Since surface-to-surface (S2S) radiation modelling is widely used for the prediction of hot and cold surfaces in the literature [18], this model was used for radiation modelling in the present study. S2S models consider radiation heat exchange between grey-diffuse surfaces in the enclosure.

3.3.3.1 Boundary conditions

In order to obtain accurate results, specifying appropriate boundary conditions are an inevitable part of any CFD simulation. So, as mentioned before, both buoyancy and wind contribute to the flow generated by an airflow window. Therefore, to calculate velocity and flow rate of inlet air from window opening to the office, a semi-empirical equation developed by Phaff *et al.*, [19] that considers the effect of mean, fluctuating flow and the effect of buoyancy is employed

$$u_{eff} = \sqrt{c_1 u_w^2 + c_2 H \Delta T + c_3}\,, \tag{1}$$

where u_w is the wind velocity at the window level which is used for the evaluation of flowrate, c_1 and c_2 are the wind speed constant and buoyancy constant, respectively, c_3 is the turbulence constant, H is the height between the center of the openings, and ΔT is the temperature difference between inside and outside. A reasonable correspondence between measured and calculated values was obtained for the $c_1 = 0.001$, $c_2 = 0.035$, and $c_3 = 0.01$.

In this study, it is assumed that the offices are in Istanbul (Turkey) urban area and the aim of the ventilation system is to provide thermal comfort condition in the area. The outdoor condition including wind velocity temperature, relative humidity is extracted from meteorological station data and used to assign appropriate boundary conditions for the room inlet. For the natural ventilation case, based on the mean monthly temperature, relative humidity and wind speed in July at Istanbul, the inlet temperature, and relative humidity of the flow from windows to the office is chosen to be 24°C and 65%, respectively. The flow velocity is calculated from Eq. (1) and fixed at the inlet. It is assumed that air enters the room with 5% turbulent intensity. The inlet kinetic energy is calculated based on the turbulent intensity and hydraulic diameter of the opening as follow:

$$\kappa = \frac{3}{2}\left(u_{avg} I\right)^2 \,, \tag{2}$$

where k is the turbulent kinetic energy and u_{avg}, and I are mean flow velocity and turbulent intensity, respectively. In addition, pressure boundary condition (the outlet pressure is fixed and backflow suppressed) is used for all outlet openings including ceiling, window and door outlets. Moreover, a no-slip boundary condition is applied to all walls and solid objects. Moreover, all interior walls of the office, floor and ceiling are kept at 25°C.

The thermal boundary condition of the hot objects in the room is similar to those of [20] (see Table 2). It is assumed that the thermal load of occupant, lamp, and personal computer (PC) is 75 W, 34 W, and 140 W, respectively. Thermal fluxes are calculated by dividing the thermal load by the area of the corresponding object and assigned as heat flux in the CFD code. Tables and other furniture are assigned thermal insulation boundary conditions (zero heat flux). To evaluate the capability of the ventilation system purposed to eliminate pollutant and dust from the room, the transport of CO_2 species in the office is simulated.

The only source of the CO_2 in the office is occupant exhalation which is modelled as mass flow input with a fixed CO_2 concentration located at the nose of each occupant. The mass flow rate of exhaling air in an average person and concentration of CO_2 in exhaling is calculated from the literature [21–23] and applied as the boundary condition in the occupant nose location. For that, a code in C++ (Fluent UDF user guide) developed and linked to the main CFD code. In addition, the ability of the proposed system to disperse dust originated from the floor area and furniture is evaluated. It is supposed that the inlet air is dust-free and the dust is released from the floor area at a constant rate (constant flux boundary condition).

3.3.4 Solution method

All convection terms were discretized using the second-order upwind scheme and the central difference was used for diffusion terms. Due to the nature of equation, the coupled method was used for calculating the pressure-velocity relation. Solutions are not considered converged until the residual of the governing equations is less than 10^{-6}. Besides, to ensure that the accuracy of the solution average temperature, RH and heat and mass balance of the domain were monitored during the solution process.

Fluent commercial code was employed for solutions of governing equations [24]. Convection terms in the governing equations were discretization by the second-order upwind method whereas the central differencing method was employed for diffusion terms. Obtained system of algebraic equation system was solved by a pre-conditioned conjugate iterative solver (PCGI) algorithm capable of solving large matrix systems in parallel. The details of the discretization and solution methods are given in [24].

In order to assess thermal and indoor air conditions in the office rooms, temperature, water vapor concentration, and pressure, relative humidity computed using the recommended method in [25]. The relative humidity is defined as

$$\varphi = \frac{p_w}{p_{ws}} . \tag{3}$$

The partial pressure of water vapor is calculated from

$$p_w = \frac{(101325 + p)m_1}{0.62198 + 0.37802 m_1} , \tag{4}$$

where m_1 is the mass fraction of water vapor in the air, and p is the partial pressure of water wapor in air. The saturation pressure water in the air is calculated from

$$p_{ws} = 1000 \exp \left[\begin{array}{c} \dfrac{5.800 \times 10^2}{T + 273.15} - 5.516 \\ -4.864 \times 10^{-2} (T + 273.15) \\ +4.176 \times 10^{-5} (T + 273.15)^2 \\ -1.445 \times 10^{-8} (T + 273.15)^3 \\ +6.546 \ln (T + 273.15) \end{array} \right] , \tag{5}$$

where T is the temperature of air in degrees Celsius (°C).

Due to the three-dimensional nature of the problem, all simulations are carried out with an HP workstation featuring 32 GB RAM and 24-cores. On this system, the time required for a single case was about 48 hours, not considering the time needed for post-processing and analyzing the results. The entire research involved about 100 studied cases requiring more than 300 computational days to complete.

3.4 Thermal comfort, indoor air quality and ventilation standards

In order to assess the performance of the airflow window in different office sizes and operating conditions, the results obtained in the present study are assessed against the criteria set by the international ventilation standards. To do so, the conditions specified in different ventilation standard were extracted from the standards' text and summarized in Table 4. As can be seen in Table 3, in ventilation standards the accepted range of indoor temperature, relative humidity and velocity are specified but also the limits for contaminants in the room are mentioned (indoor air quality criteria).

Table 4: Recommended limit values in standards related to indoor air quality.

Standard	CO_2 (ppm)	Particulate matter (PM) (μg/m^3)	Relative humidity (%)	Temperature (°)
ABD ASHRAE	1000	< 75	30–60	20–25
Germany	5000 9000 (15 min)	–	30–70	20–26
OSHA	Administration) notes the time-weighted average value of 5000 ppm as the transitional limit, and the time-weighted average value of 10000 ppm and the short-term exposure limit of 30000 ppm as the final rule limits.	–	–	–
Canada	3500	< 40 (8 hours) < 100 (1 hours)	30–80 (summer) 30–55 (winter)	–
China	–	< 150	–	26
WHO	–	< 20	–	–
England	–	< 50	–	–
Norway	–	< 20	–	–
European Union	–	< 35	–	–
Hong Kong	800 (1. level) 1000 (2. level)	< 20 < 180	40–70	20–25.5

In order to assess thermal and indoor air condition in the office rooms, temperature, water vapor concentration, and pressure, relative humidity computed using the recommended method in [26]. Another commonly used criteria used in the evaluation of thermal comfort is the mean predicted vote (PMV), a thermal comfort model proposed by Patankar [27] and is based on steady-state energy balance. This model was originally developed to predict human thermal comfort in office-like environments and has gained popularity in the evaluation of HVAC systems due to its simplicity and acceptable predictions. The scores obtained from this model varies from -3 for extremely cool ambient to $+3$ for extremely hot. The PMV score of 0 is deemed to be neutral which is called the thermal comfort zone. In ASHRAE standard 55 the comfort zone is defined as $-0.5 \leq \text{PMV} \leq +0.5$ [24].

According to ASHRAE standard 55, the PMV model of the Fanger is given by [28]

$$\begin{aligned}
\mathrm{PMV} = \{&0.303 \exp\left[-0.036(M-W)\right] + 0.028\}\{(M-W). \\
&- 3.05 \times 10^{-3}\left[5733 - 6.99(M-W) - p_w\right] \\
&- 0.42\left[(M-W) - 58.15\right] - 1.7 \times 10^{-3} M\left(5867 - p_w\right) \\
&- 1.4 \times 10^{-3} M\left(34 - T_a\right) \\
&- 3.96 \times 10^{-8} f_{cl}\left[(T_{cl}+273)^4 - (T_a+273)^4\right] \\
&- f_{cl} h_c\left(T_{cl} - T_a\right)\}, \qquad (6)
\end{aligned}$$

where clothing temperature

$$\begin{aligned}
T_{cl} = {}&35.7 - 0.028(M-W) \\
&- I_{cl}\Big\{3.96 \times 10^{-8} f_{cl}\left[(T_{cl}+273)^4 - (T_a+273)^4\right] \\
&+ f_{cl} h_c\left(T_{cl} - T_a\right)\Big\}, \qquad (7)
\end{aligned}$$

$$h_c = \begin{cases} 32.38\left(T_{cl} - T_0\right)^{0.25}, \\ 12.1 v^{0.5}, \end{cases} \qquad (8)$$

depending on which value is greater, and

$$f_{cl} = \begin{cases} 1.00 + 1.29 I_{cl} & \text{for} \quad I_{cl} \le 0.078 \dfrac{\mathrm{m^2K}}{\mathrm{W}}, \\ 1.05 + 0.645 I_{cl} & \text{for} \quad I_{cl} \ge 0.078 \dfrac{\mathrm{m^2K}}{\mathrm{W}}, \end{cases} \qquad (9)$$

where M, f_{cl}, T_{cl}, and T_0 are metabolic rate, external work, clothing insulation level, and mean radiant temperature (MRT), respectively, W is the effective mechanical power, T_a is the air temperature, I_{cl} is the clothing insulation, and $v$is the relative air velocity.

In calculating PMV, it's assumed that staff in the office room are typing in the sedentary position while wearing summer official suits. Therefore, metabolic rate, cloth insulation and MRT are set to 1.1, 0.85, and 25°C, respectively. According to HVAC standards, the external work for typing is 0. Because the clothing insulation level is initially unknown, a trial and error process should be used to solve the PMV equation in ASHRAE standard 55. For that, a code in the C++ programming language is developed (Fluent UDF) and its results are verified against CBE Thermal Comfort Tool [29–31] which is an online tool for the calculation of thermal comfort condition based on ASHRAE standard 55 and EN-16798. Furthermore, to facilitate visualization of thermal conditions in the room and agreement of studied cases with international ventilation standards, limitations mentioned in Table 4 are depicted graphically in Fig. 6. The comfort zone is also illustrated in this figure by black dashed lines.

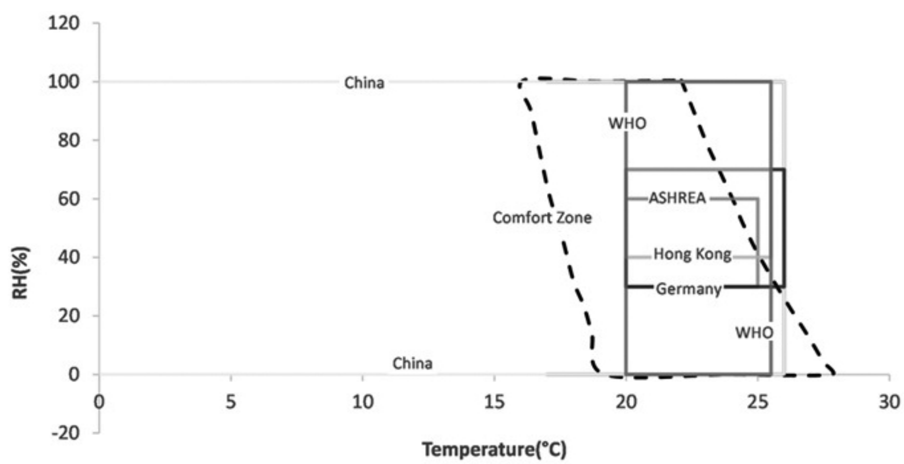


Figure 6: Ventilation standard and comfort zone on the temperature – relative humidity chart.

4 Model validation

In order to assure the validity of the numerical code, the result obtained should be validated against experimental data in the literature. However, in the case of the airflow window, reliable experimental data lacks in the literature. For that, to validate the solution procedure in the present study, the experimental measurements of Fuliotto *et al.* [32] were used. An outline of the room studied by them is shown in Fig. 7.

The reason for choosing this study for validation is that the dimensions and heating loads of the room are identical to those modelled in the present study, except that ventilation is provided through a diffuser rather than the airflow window. In the figure, velocity and temperature in the center-line of the room are compared with the numerical results from this study. In Fig. 8

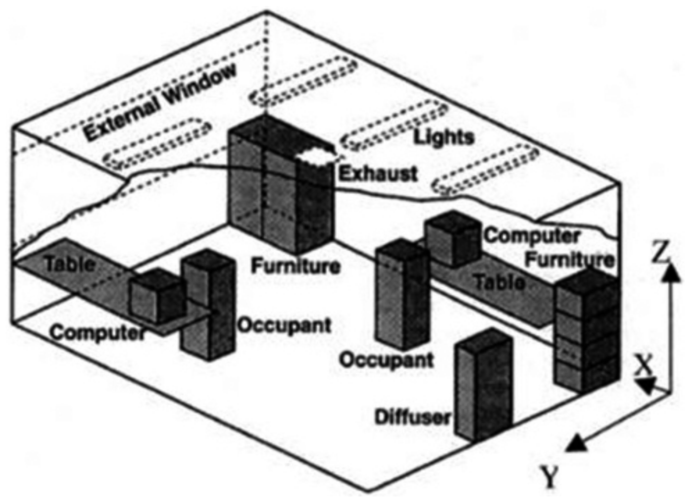


Figure 7: Room outlines used in [32].

center-room vertical line temperature and velocity profile are compared between numerical and experimental measurement and numerical data. As figure indicates, there is good agreement between experimental data and the results of numerical code.

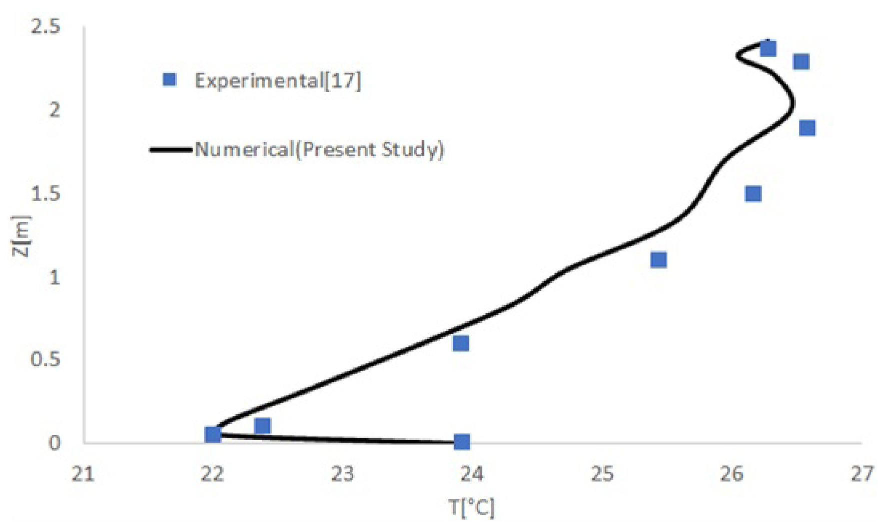


Figure 8: Numerical solution validation.

5 Results and discussion

The objective goal of the study is to assess the performance of the airflow window in providing an indoor air condition for occupants that meets the criteria in ventilation standard as well as a thermally comfortable environment harvesting the potential maximum benefits of the natural ventilation. For that, the air inlet conditions is varied and energy consumption in each mixed ventilation case is compared with the forced ventilation mode.

5.1 Natural ventilation

The effect of natural ventilation from the airflow window has been investigated. It is assumed that the office rooms are ventilated by the airflow window that is operating in the natural ventilation mode. The flow generated by the window is created by the combination of wind and buoyancy. In Figs. 9a–b the flow pattern of natural ventilation in the small room and 0.4 m height windows (Case N-S4) are depicted.

As expected, the temperature in the vicinity of heat sources such as PC lamps are higher and the maximum temperature is on the surface of the PC which is about 45°C. As air enters from natural ventilation in the office it creates a complex pattern in the room with recirculation zones. After heat transfer to room, the hot air leaves the office from either ceiling outlet or window outlet. Streamlines shown in Fig. 9c indicate that most of the air leaves the office form ceiling in the outlet while a smaller portion of air leaves the office through a lower part of windows. Contours of velocity, temperature and relative humidity in the midplane of the office are given in Figs. 10c–f. As it is obvious from Fig. 10c, the velocity of the flow in the vicinity of the occupant is of the order of magnitude of 0.01, which is well below the velocity limits mentioned in ventilation standards. A region with low velocity formed in front of the occupant near to the windows outlet which is not effectively cooled by the ventilation system. When considering the temperature distribution around the occupant, the air temperature around the ankles is around 24°C while the rest of the body is exposed to about 10°C warmer airflow. The maximum temperature can be found near the hip and chest. The trend of relative velocity is reversed. The relative humidity is higher at feet and lower at the upper body and chest. All of these parameters influence the thermal comfort of the occupant. Figure 9e shows the extended predicted mean vote (PMV_e) for NS4. As it can be seen in this arrangement the feet will feel comfortable while the upper body is

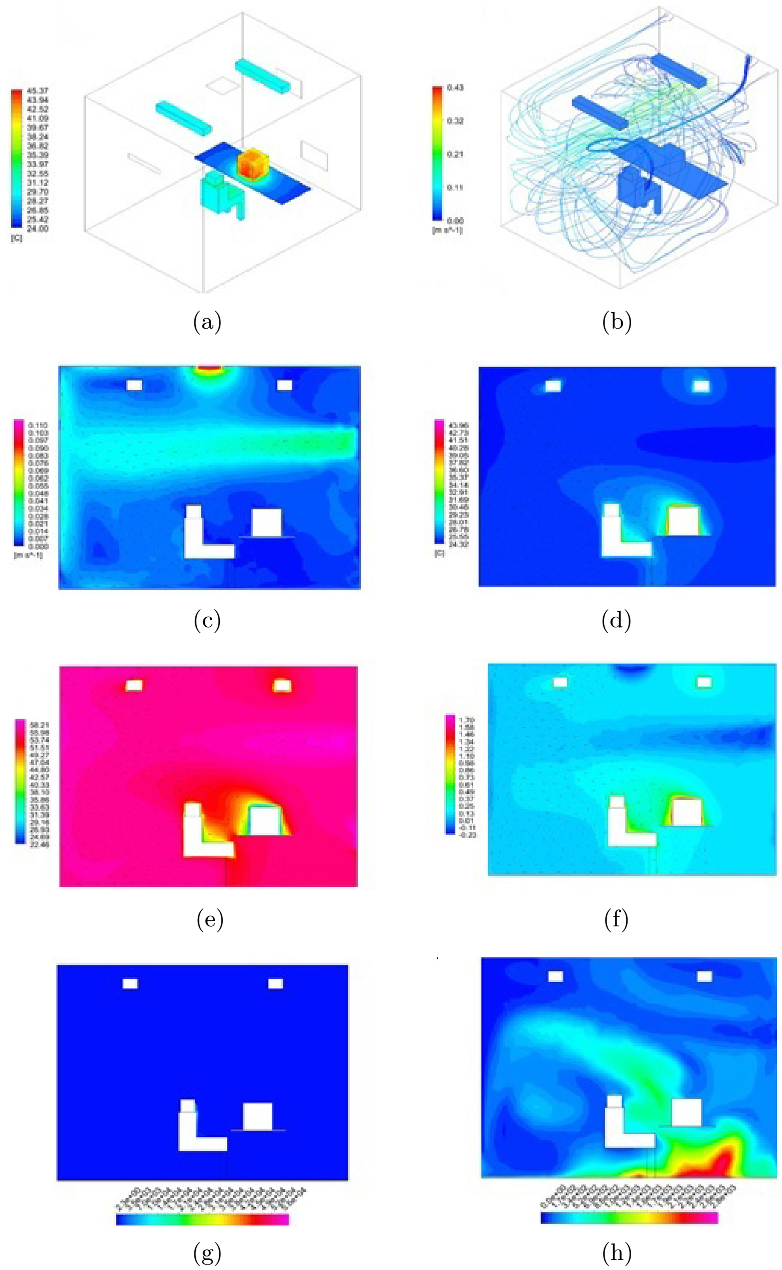


Figure 9: Effect of natural ventilation from the airflow window in the small room and 0.4 m height windows for case NS4: a) streamlines, b) temperature contours, c) velocity, d) temperature, e) RH, f) PMV_e, g) concentration of CO_2, h) dust distribution in the midplane.

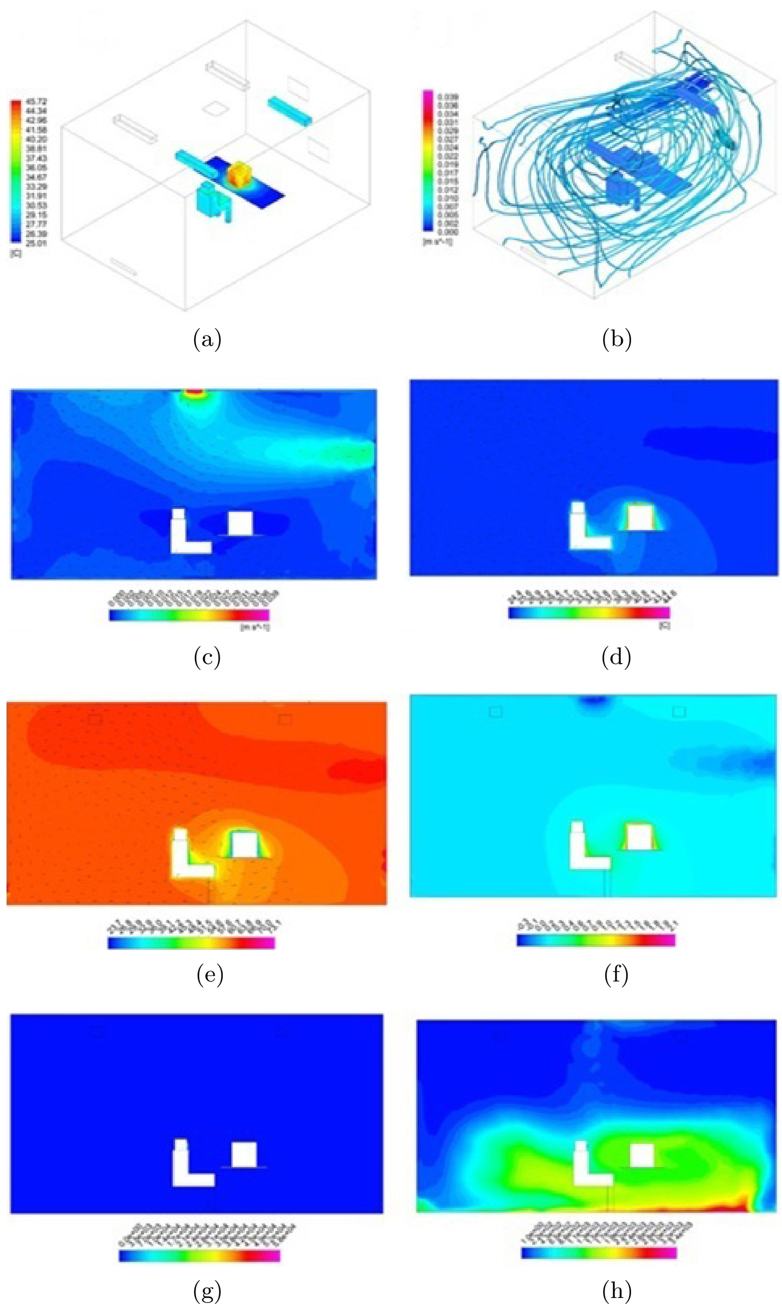
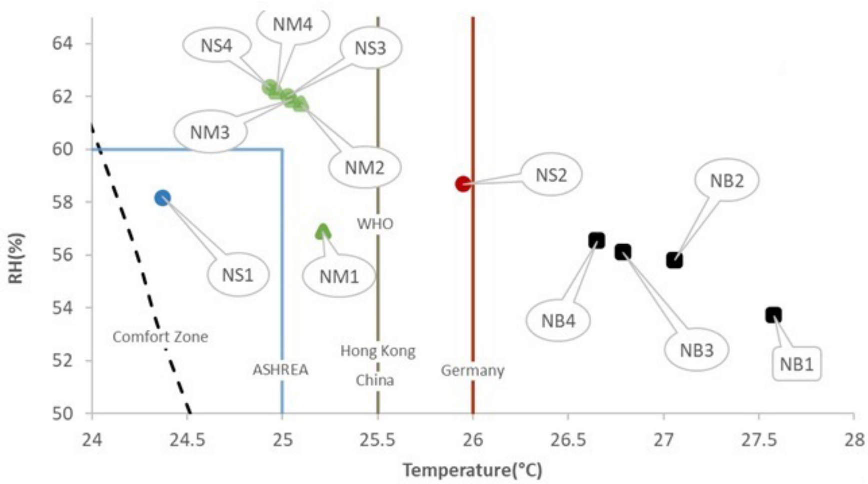


Figure 10: Effect of natural ventilation from the airflow window in the office for case NM4: a) streamlines, b) temperature contours, c) velocity, d) temperature, e) RH, f) PMV_e, g) concentration of CO_2, h) dust distribution in the midplane.

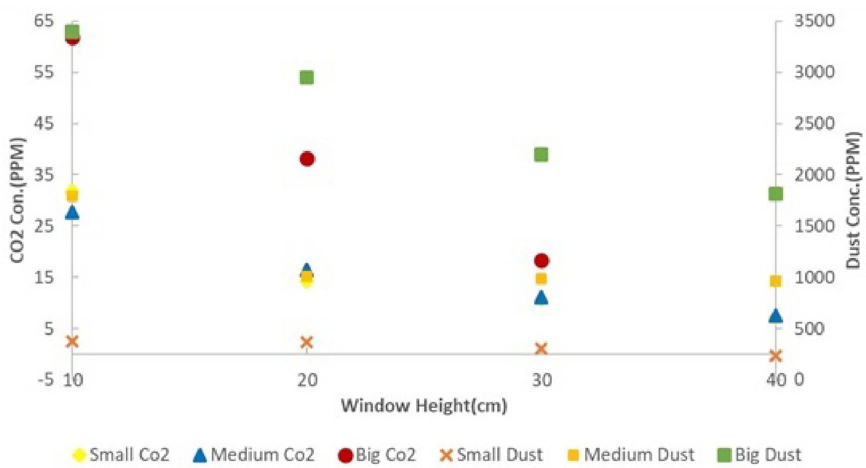
in a slightly light condition. So, the occupant will report a natural feeling at the foot level and a slightly warm thermal condition. Contaminants distribution in the including CO_2 and dust are shown in Figs. 10g–h. The CO_2 concentration is only significant in the close vicinity of the occupant's nose and can be neglected everywhere else. Also, dust released from the floor is disturbed by the flow and removed from outlet. The dust is trapped under the table and a moderate level of the dust can be found around the body. Flow pattern temperature contours on non-isothermal surfaces are illustrated in Figs. 10. The temperature distribution on the hot surface is similar to the small room but the average temperature is slightly higher, which can be contributed to more thermal sources (4 lamps) as well as a greater volume of air in the room.

The flow and temperature field are similar to the corresponding case in the small room, however, in this case, distribution of PVM_e around the occupant is more uniform and fall in the range designated as though, in this case, natural ventilation cannot provide the local thermal comfort condition. Like case NS4, the concentration of the CO_2 in the medium office is very high in the small region around the nose and negligible in the other regions. Moreover, the dust is accumulated under the floor in the right half of the room near the window outlet and is disappeared flow upward to include more than the height of the occupant. So, it is more likely that the occupant exposed to inhale the dust. The average temperature and relative humidity for all studied natural ventilation cases are computed and depicted in Fig. 11a.

None of the natural ventilation cases is inside the thermal comfort region. So, in this case, although natural ventilation can provide local thermal comfort at some locations in the office, it fails to satisfy thermal comfort in the office on average. When considering convection, in this circumstanses, NS1 is the best case which is in agreement with ASHRAE standard, and although it lays outside the thermal zone, it is very close to the thermal zone boundaries. In contrast, natural ventilation is unable to satisfy relative humidity and temperature conditions. All other cases belonging to small and medium offices doesn't meet the requirements set by ASHRAE but are inside the region for Austria/China/Hong Kong standards. Therefore, it can be speculated that natural ventilation is not effective in providing thermal comfort as well as satisfying thermal conditions in the ventilation standard in big offices. The average temperature and relative humidity in medium and small offices agree with the most ventilating standard exempt ASHRAE but lay outside (but very close) to the comfort zone. The av-



(a)



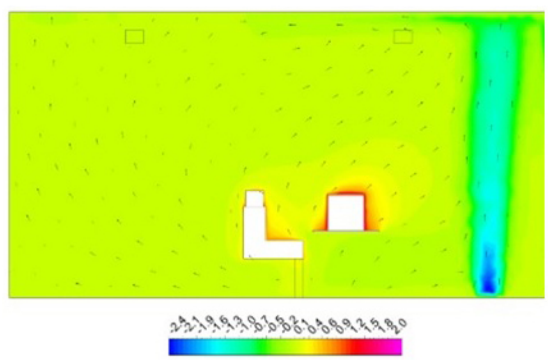
(b)

Figure 11: a) Evaluation of ventilation standard and thermal comfort. b) Contaminant(s) concentration for natural ventilation cases (small room 3 m × 4 m, medium room 6 m × 4 m, and, big room 8 m × 4.5 m.

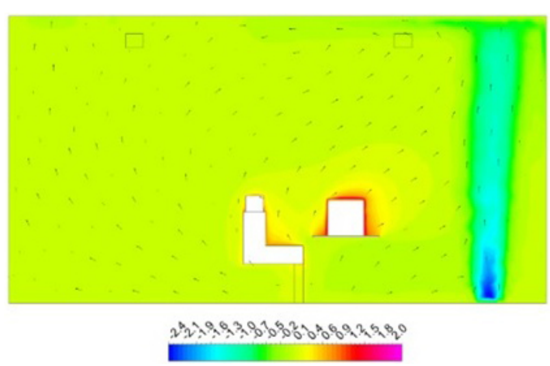
erage concentration of CO_2 and dust vapor under natural ventilation is given in Fig. 11b. The average concentration of CO_2 in offices are less than 10% of the limited mentioned in the ventilation standard and the average concentration decreases as the window opening height increased. This can be justified by the fact that increasing window opening will lead to an increase in the mass flow entering the room. This means that air in the office is changing more frequently and contaminants are removed better from the office. Therefore, it can be concluded that the airflow windows can be used for effectively removing dust and contaminant from offices studied. The big advantage is that this is achieved using natural forces in the region without consuming external energy.

5.2 Forced ventilation

In order to provide a base of comparison for natural and mixed, in this section underfloor air distribution (UFAD) system in the office room were simulated. The simulation results for the forced ventilation small, medium, and big office rooms were presented in Figs. 12. As shows the concentration in the office room, it can be found that the UFAD system removes con-

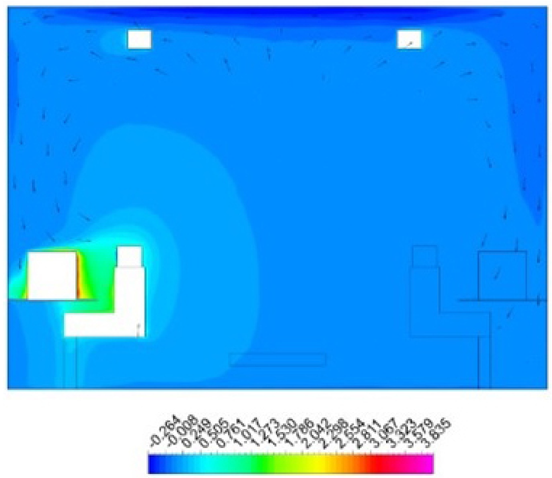


(a)



(b)

Figure 12.



(c)

Figure 12: Dust concentration (in ppm) for UFAD systems: a) small room, b) medium room, c) big office room.

taminants and particles from the occupied zone more effectively. Besides, the average and maximum of dust in the office room is lower meaning that UFAD system can a better indoor air quality for the occupants than the airflow window.

Dust concentration for forced ventilation is shown in Fig. 12. The results of simulations for UFAD system are shown in Fig. 13. The average conditions of air in the small room falls inside the thermal zone but it does

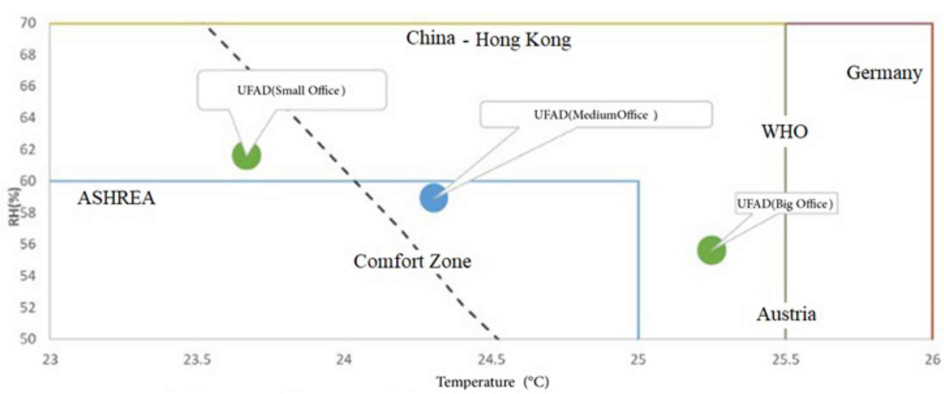


Figure 13: Evaluation of ventilation standard and thermal comfort for UFAD system.

not agree with the relative humidity limits in ASHRAE standard 55. So, to meet the requirements of ASHRAE standard, the relative humidity of the air at the inlet nozzle should be decreased. The point for the medium office is within the limits specified by ASHRAE but outside (and very close to) the thermal comfort zone. The results obtained for the big office agree with neither thermal zone nor ASHRAE standard but is within the limits of Austria/China and Germany standards.

5.3 Mixed ventilation

Natural ventilation through the airflow windows does not meet the requirements of ventilation standard and provide thermal comfort for the occupants except in some cases. Hence, to provide thermal comfort for the occupants a supplementary ventilation system should be included in the window system. For that, it has been assumed that a fan system is installed inside the airflow that is operated when more inlet flow in addition to flow generated by the airflow window is required for the office room ventilation. This system is also useful when there is no wind in the region. It is also assumed that the airflow window is equipped with a dehumidifier and cooler that can be used for the reduction of air humidity and temperature when necessary. Although requiring more energy (energy consumed by fans) the main advantages of the mixed or the combined ventilating mode over natural ventilation mode is its controllability and ability to provide thermal comfort condition in offices even there is no wind in the area or the wind speed is too low.

The average temperature and relative humidity obtained from CFD are drawn on the chart showing thermal comfort zone and ventilation standards (Figs. 14–16), moreover, the size of each point shows its relative energy consumption compared to the similar case in the UFAD. The relative energy consumption of each point is also given in the point name after a comma. In the case of natural ventillation (labelled with N) or when energy consumption is at the very low or negligible location if the case is marked with a small triangle and numbers after the comma is eliminated. In each office room the velocity, temperature and relative humidity for the inlet of the airflow window are varied. The goal of this research was to find the best case for each room that agrees with thermal comfort and ventilation standards while requiring the minimum amount of external energy. According to the thermal balance from the first law of thermodynamics for the airflow window, the energy required to cool humid form T_1 and RH_1 to T_2 and

RH_2 are

$$E = m\left(h_o + \frac{T_o^2}{2} + h_i + \frac{T_i^2}{2}\right), \tag{10}$$

where h and T stand for enthalphy and temperature, and the subscribes o and i denote thermal condition at the outside and inside of airflow window, respectively. In order to compute enthalpy in each case, the Mollier diagram [33] is used.

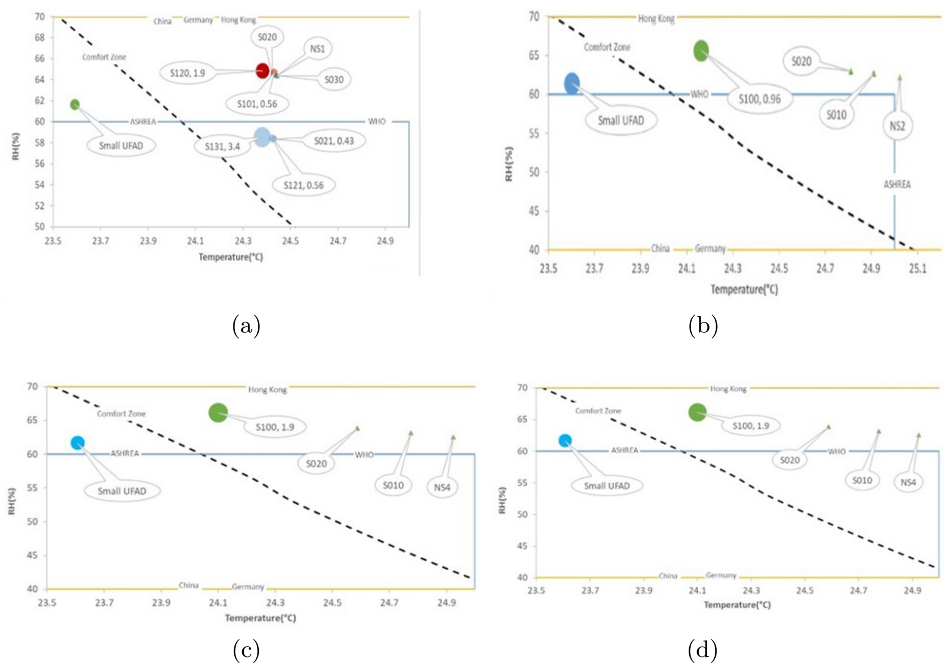


Figure 14: Evaluation of ventilation standard and thermal comfort of mixed ventilation in the small office and airflow window opening height: a) 0.1 m, b) 0.2 m, c) 0.33 m, d) 0.4 m.

Agreement with thermal comfort and ventilation standards for mixed ventilation in the small office is assessed in Figs. 14a–d. When the window height is 0.1 m, the ASHRAE standard can be satisfied by increasing the inlet velocity to 1 m/s and decreasing relative humidity to 60% (Case S021). In this case, the energy required to cool the system is only about half of that required for the UFAD system. In this configuration, despite the UFAD system, mixed ventilation can be configured to satisfy ASHRAE standard consuming less energy. Other cases are either not offering any advantage in the ventilation or require more energy than the corresponding UFAD system (S131) but all fall inside the borderline ventilation standard and outside of comfort zone. So S021 can be designated as the best operating case for the airflow window mixed ventilation.

The result for mixed ventilation in the medium office is presented in Fig. 15. Investigating Fig. 15b (0.1 m window height) shows that both M130, M131 cases are in the ASHRAE region but the energy consumption of them are more than the similar UFAD system. There is no best case for this configuration. The trend is the same for the 0.2 m window, but when it comes to the 0.3 m window opening ASHRAE requirements will be satisfied only by dehumidifying inlet air from 65% to 60% whose required energy is negligible in comparison with the UFAD system. It's interesting that when thermal comfort intended, the energy consumption is about 8.8 times of the UFAD system, making this case not economically feasible. So, the best case in this configuration is M001. For the 0.4 m windows fol-

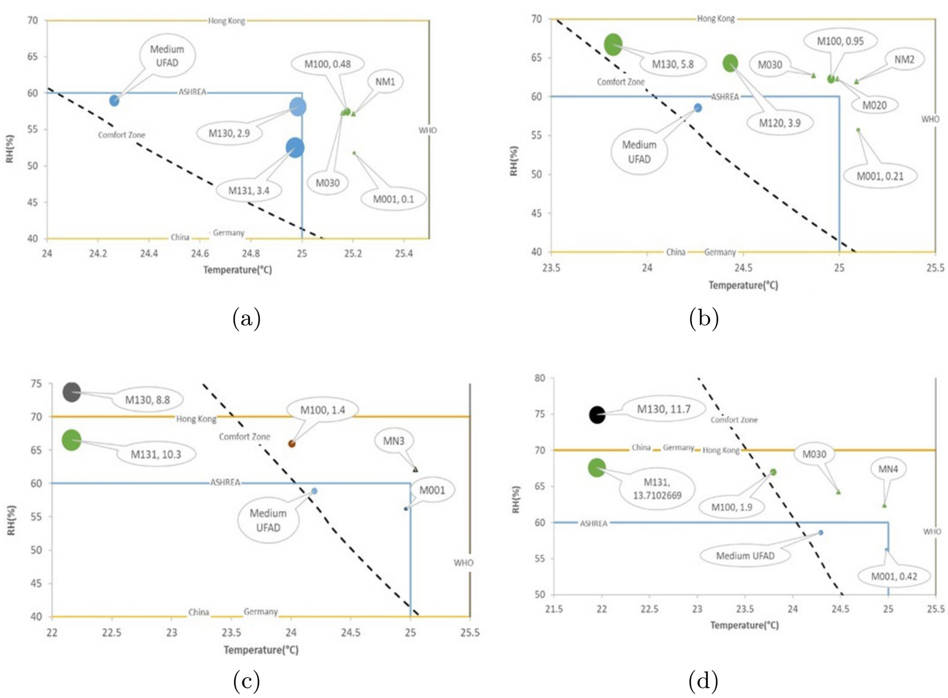
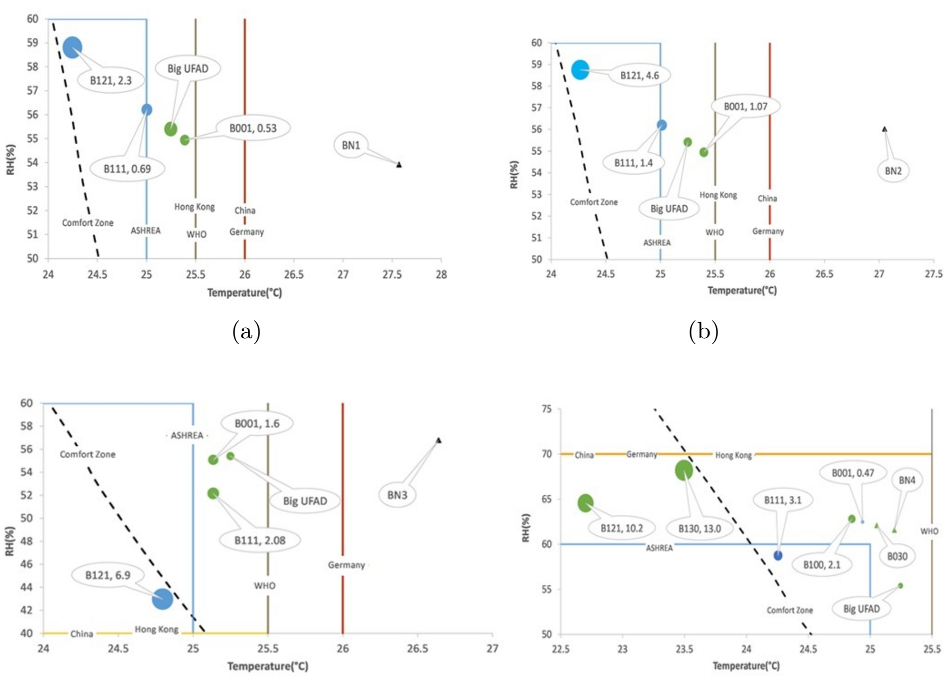


Figure 15: Evaluation of ventilation standard and thermal comfort of mixed ventilation in medium office and airflow window opening height: a) 0.1 m, b) 0.2 m, c) 0.3 m, d) 0.4 m.

low the same trend, in this configuration M001 is the best operating case too except that the energy consumption of the airflow window is about half of the corresponding UFAD case. The increase in energy consumption is due to an increase in the area of the inlet opening and though increase in mass flow rate of the airflow window.

As it can be seen in Fig. 16 the natural ventilation satisfies neither ventilation zone nor thermal comfort condition in the big office. Hence, when the studies offices are the ventilation system should be always operated in a mixed mode. Using mixed ventilation, the condition in the big office ventilated from a 0.1 m airflow window can be changed to satisfy ventilation standard. In this configuration, the best case is B111 in which the air cooled down to 20°C, the velocity increased to 0.5 m/s and the inlet relative humidity is 60%. It's also interesting that only by decreasing RH from 65% to 60%, most ventilation standards except ASHRAE are satisfied.



(a) (b)

(c) (d)

Figure 16: Evaluation of ventilation standard and thermal comfort of mixed ventilation in big office and airflow window opening height: a) 0.1 m, b) 0.2 m, c) 0.3 m, d) 0.4 m.

Varying the design parameters in the big windows with a 0.2 m opening will result in a similar behavior. However, the energy consumption ratio is greater than 1 in each case. Therefore, in this case, the application of airflow is not feasible in this configuration. The situation is worth the 0.4 m window. Although ventilation standards can be satisfied using 0.1 m windows, this configuration requires multiple times external energy which is not practical.

Hence, it can be concluded that the proposed airflow window (except with 0.1 m opening) cannot provide the indoor air condition compatible with ventilation standards and thermal condition with the same level of external energy equal with energy consumption less than the UFAD system, either in natural or mixed operation mode. In addition, in all windows, there is no case that lays inside the thermal comfort zone without demanding more external energy than of UFAD system. Therefore, it can be concluded when providing thermal comfort for the occupants intended airflow windows are not economically efficient, but in most cases, ventilation standard can be satisfied using the mixed ventilation and consuming only a portion of energy needed for the corresponding configurations.

6 Conclusions

In the presented study, the performance of airflow in respect to the conditions specified in ventilation standard and thermal comfort condition was assessed employing numerical simulation method. Both natural and mixed ventilation modes were considered for ventilation of an office room with three sizes called small, medium and big rooms, respectively. The windows opening height, inlet velocity, humidity and temperature on local average of indoor air condition were investigated.

The main findings of the research are:

1. Natural ventilation from the airflow window is able to effectively remove CO_2 and particles from offices of all sizes and reduce average contaminants level to comply with ASHRAE and other ventilation standards studied. Therefore, it can be concluded that the proposed airflow window is a useful ventilation system to improve indoor air quality. The maximum average CO_2 concentration is about 62 ppm which is observed in an 8 m $\times$ 4.5 m room with two occupants.

2. Local thermal comfort condition can be achieved by employing an airflow window. While the windows can provide thermal comfort for

almost all parts of the occupant body in the 3 m × 4 m office cubicle, the occupant(s) sitting in 6 m × 4 m and 8 m × 4.5 m offices will experience a slightly warm thermal sensation around occupant's body especially the upper part. The local comfort condition varies with the window height and improves as the window height increases.

3. In most cases (90% of studied cases), the thermal condition in the 3 m × 4 m and 6 m × 4 m offices comply with the thermal, velocity and contaminant limitation set by the ventilation standards except ASHRAE. The only case that is in agreement with the ASHRAE requirement is NS-1 (natural ventilation in 3 m × 4 m office with windows height of 0.1 m).
4. Natural ventilation in the 8 m × 4.5 m office results in indoor air being neither in agreement with the ventilation standard nor with thermal comfort conditions (mean predicted vote in range –0.5–0.5). Hence, natural ventilation from the airflow window could not be used for this office room.
5. It is found that in the mixed ventilation mode, in most cases, the indoor condition for 3 m × 4 m and 6 m × 4 m offices can satisfy the requirements of ASHRAE only consuming a fraction of external energy used in the underfloor air distribution system. The maximum ratio of the energy required for mixed ventilation to that underfloor air distribution system is around 50%.
6. Except for the 0.1 m height window, employing mixed ventilation for the 8 m × 4.5 m office with two occupants requires more energy than the underfloor air distribution system and is, though not economically feasible.
7. In general, utilizing the airflow windows either in natural or mixed is only recommended for 3 m × 4 m and 6 m × 4 m offices having smaller ventilation demands but not for the 8 m × 4.5 m office.
8. Despite that airflow window can provide local thermal comfort for 3 m × 4 m and 6 m × 4 m office cubicles, when the average thermal comfort in the room intended, the average condition of room ventilated by the airflow windows falls out of the comfort zone suggested by the Fanger's model.

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