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Intermittent personalized ventilation coupled with mixing ventilation for occupant protection against active and passive contaminants and energy savings

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Abstract. An intermittent personalized ventilation PVMmodule (PVMM) coupled with a conventional mixing ventilation system (MV) was studied to investigate its ability in protecting occupants from indoor contaminants. This study examines the intermittent PVM operating frequency range that minimizes active and passive contaminant concentration near the occupant and nearby surfaces. A 3D CFD model of a typical office space was developed for this study and validated through experiments done in a climatic chamber. Simulations were performed under different typical indoor frequencies (0.3 Hz, 0.5 Hz and 1 Hz) and a typical personalized ventilation average flow rate (7.5 L/s). The space was considered to be infected with CO_2 and particles of diameter 1 µm. Results showed that under a frequency range of [0.78 Hz – 0.94 Hz], the intermittent PVM was able to provide good occupant protection with a satisfactory ventilation efficiency of around 77%, intake fraction range of [11.8 %, 26.8 %] and a deposition fraction range of [7.5 %, 8.75 %]. Moreover, the system was able to reduce energy costs by 16.1 % compared to steady state personalized ventilator.

Key words personalized ventilation, mixing ventilation, intermittent, contaminants

1. Introduction

With the increasing time that occupants spend indoors and especially in office spaces, providing thermal comfort as well as good indoor air quality (IAQ) is of upmost importance [1]. The indoor space can be polluted by several types of contaminants [2]. Some contaminants are passive and considered as a continuum with the airflow field in the space. Passive contaminants include species such as CO_2, CO, NO_2, toluene… and can be issued from human respiration, combustion devices or office products. Other pollutants include active contaminants that are heavier than species pollutants and can settle and deposit on surfaces (tables, chairs, computer…) due to their weight. Active contaminants result from human respiratory activities (sneezing coughing…) or office activities (painting, smoking) [3]. While passive and active contaminants infect occupants through direct inhalation, active contaminants can also be transmitted indirectly from indoor surfaces through hand to mouth contact [4]. Therefore, it is crucial for the heating ventilation and air conditioning (HVAC) system to provide occupants with fresh air for the sake of their wellbeing as well as their level of productivity in the office space.

To achieve that purpose, many applications use hybrid HVAC systems that couple conventional background ventilation system with add-on systems [5, 6]. A hybrid HVAC system is the conventional mixing ventilation system (MV) coupled with personalized ventilation (PVM) [7]. MV systems supply a mixture of fresh and recirculated air into the space while PVM conditions the occupant microclimate by supplying cool fresh air directly into the occupant breathing zone (BZ). There has been many field and lab studies on different types of PVM (ceiling integrated, desk mounted…) [8, 9]. Through varying PVM supply temperature and flow rate, it was found to provide good thermal comfort and breathable air quality [10]. For enhanced thermal comfort, recent PVM applications suggest operating the PVM under intermittent conditions of flow rate. Huang et al. [11] conducted human subject experiments and varied intermittent PVM frequency to record human thermal response. An operating range of 0.5 Hz to 1 Hz was found to provide comfort in warm conditions. A frequency of 1 Hz was found to provide comfort in a similar study by Al-Assaad et al. [12] who used computational fluid dynamics (CFD).

All studies found in literature considered the ability of intermittent PVM in enhancing comfort. However, the critical issue of IAQ was not tackled. In fact, while intermittent PVM can enhance comfort, varying the flow rate between a high and a low can enhance turbulence and flow agitation especially with an increasing frequency. This could increase the concentration of active and passive contaminants transmitted to the occupant compared to steady state PVM and compromise IAQ. Therefore it is important to investigate the efficiency of intermittent PVM in protecting occupant from passive and active contaminants in indoor spaces while still ensuring thermal comfort.

In this study, an intermittent MV+PVM system is considered. The purpose of this work is to investigate the efficiency of intermittent PVM in protecting the occupant from active and passive contaminants in the space while still assuring comfort. A parametric study was then conducted to find the optimal operational PVM frequency range for good IAQ and thermal comfort. An energy analysis was also performed to find the energy savings provided by the intermittent system compared to steady state personalized ventilation for equivalent comfort. To achieve these objectives, the work was conducted through 3D CFD modeling that was validated experimentally in a climatic chamber.

2. Problem description

The study considers a typical office space of dimensions 3.4 m × 3.4 m × 2.8 m. The office is equipped with a workstation and an occupant represented by a thermal manikin. The room is equipped with a 2 m × 1 m wooden door and the office is considered to have small apertures (0.5 m × 0.6 m). The room walls are constituted of several layers (brick, concrete). The inner wall layer is constituted of a Gypsum board having a thermal conductivity of 0.161 W/m.K. The total load to be removed is a typical load of 60 W/m^2 due to computer, lighting, occupants [13]. The office is conditioned by a MV system having its own air handling unit (AHU) supplying 100 % filtered recirculated air into the space. The supply air has negligible concentration of active particulate matter, CO, NO_2, SO_2… etc. and typical CO_2 fresh air concentrations of 450 ppm. The MV system is coupled with an intermittent PVM system. The PVM nozzle is mounted on the desk, has a diameter outlet of 5 cm and located at 40 cm from the occupant. Figure 2 illustrates the considered space. The PVM withdraws clean 100 % filtered fresh air from an adjacent fresh air source. The PVM fresh air has also negligible concentration of active particles and passive contaminants such as CO, NO_2, SO_2 …etc and typical fresh air CO_2 concentrations (450 ppm). The horizontal jet is supplied towards the occupant breathing level at a characteristic average flow rate and frequency f. The instantaneous PVM velocity is given in equation (1):

$$V(t) = V_{ave} + \frac{A}{2}\sin(2\pi f t) \qquad (1)$$

Where $V(t)$ is the instantaneous velocity, A is the amplitude, V_{ave} is the average velocity, f is frequency and t is time. The occupant contamination with species is assessed with the ventilation efficiency index ε_v [10] presented in equation (1):

$$\varepsilon_v = \frac{C_R - C_{BZ}}{C_R - C_{Fr}} \qquad (2)$$

where C_R, C_{BZ}, C_{PVM} are the species concentrations in the return, BZ and PVM fresh air respectively.

The direct occupant contamination with active particles is assessed with the intake fraction IF and deposited fractions DFr. The intake fraction given in equation (3) is the concentration ratio of particles at a certain breathing location over generated particles at the source location. The deposited fraction DFr given in equation (4) is the number of particles deposited on a surface over the generated particles number at the source. The surface which were considered were the occupant, the workspace floor, table, chair and computer. It is worthy to note that the intake fraction IF was evaluated at the occupant surrounding microclimate at the extreme lefts and rights of the occupant head ($IF_{max,\ micro}$) which is more critical than the BZ since the PVM jet is more infected at the peripheries than the clean core region.

$$IF = \frac{\text{Particle concentration in occupant microclimate}}{\text{Particle concentration at source}} \qquad (3)$$

$$DFr = \frac{\text{Number of particles deposited on surface}}{\text{Number of Particles at the source}} \qquad (4)$$

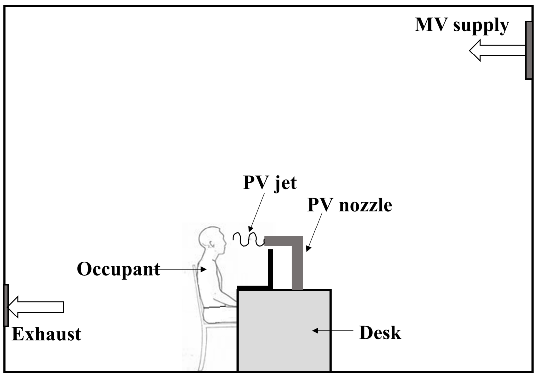

Fig. 1 Schematic of the office space considered with MV and intermittent PVM.

Since the objective of this study is to protect occupants from species and contaminants, the particle source is considered at the seated level of a human, 1 m away, generating particles of 1 µm towards the occupant constituting a critical scenario. As for the species, four species sources are considered at the walls generating in total 2 L/min of CO_2 [13] which the extreme case of four breathing occupants in the office.

3. CFD model

In the considered study, several physical phenomena take place such as the intermittency of the PVM jet, the resulting increase in turbulence, the different source heat fluxes and the rising thermal plumes, the recirculating air from the MV supply, species transport, as well as particle dispersion, escape and deposition. These phenomena affect the flow field inside the space such as velocity, temperature and concentration fields. This calls for the use of a 3D CFD modeling tool which is able to solve for the different unknowns present inside the space and used especially to assess the species and particle concentration in the occupant breathing zone and microclimate. The breathing zone (BZ) is defined as the sphere of diameter 2 cm located at 2.5 cm from the nose [10]. The occupant microclimate is defined as the circular area of diameter 16 cm surrounding the occupant's head at a height of 1.05 m.

The commercial software ANSYS Fluent [14] was used to solve for the different equations in the space (momentum, energy, turbulence, species, particles…). Fig. 2(a) illustrates the computational domain used in ANSYS. For a precise solution, the domain should be properly meshed as seen in Fig. 2(b). To properly capture the flow behavior near surface boundaries, inflation layers are created around these boundaries such at the dimensionless wall number y^+ varies in the range of [0.8, 4] [13]. The mesh used consisted of tetrahedral element with element sizing of 1.5 cm and 2 cm respectively for the manikin and walls. This mesh sizing assured a grid independent solution with a maximum relative error of 5%. The final grid of the considered space has 6,501,719 elements and 1,661,932 nodes.

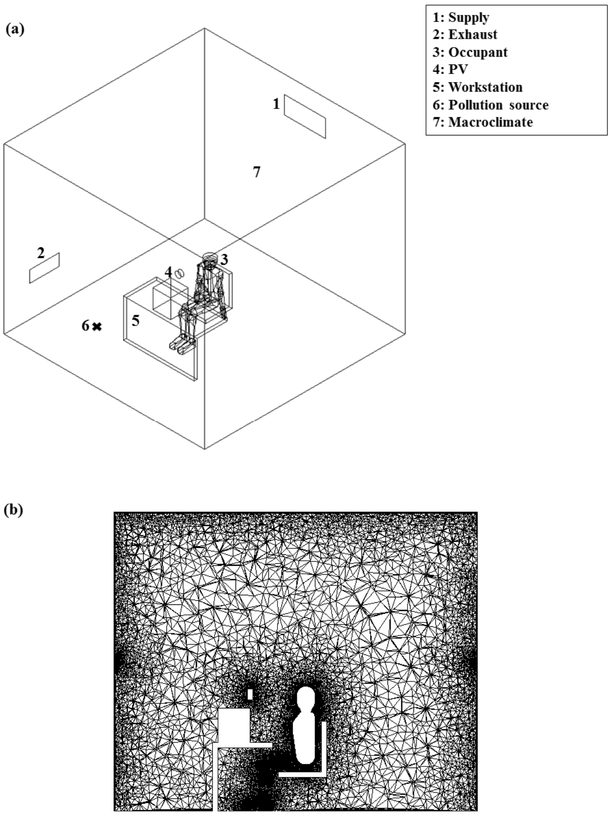

Fig. 2. Illustration of: a) the Fluent computational domain; b) the corresponding mesh.

Due to the transient fluctuations of the PVM jet and the mixing effect of the MV supply, increased turbulence is witnessed in the space. For this reason, turbulence needs to be accurately modeled. The RNG k-ε model was proven by Liu et al. [15] to be accurate in indoor spaces. Due to small density gradients in the space, the Boussinesq approximation is used to account for buoyancy effects. The discretization schemes for the different variables (momentum, energy, k, ε, species…) are presented in Table 1, including the transient term which accounts for the intermittent PVM variations. Numerical convergence is reached for scaled residuals of less than 10^{-5} for all quantities except energy where it should be less than 10^{-7}.

Table 1. Flow field variables and the different discretization schemes

Flow Field variables	Scheme
momentum, energy, k, ε and turbulence equations, species transport	second order upwind scheme
pressure equation	"PRESTO!" scheme
Transient term	second order implicit time stepping scheme, time step of 0.05 seconds
Pressure velocity coupling	PISO scheme

As for particle modeling, the discrete phase model (DPM) was used. The DPM model solves the second law of Newton and accounts for the different forces acting on the particle (drag, lift, gravity, thermophoretic forces…). The Lagrangian technique was used to track the particles' trajectories in the space along with the discrete random walk (DRW) which was used in the study of Makhoul et al. [16] and yielded good results.

In order to get physical results, the boundary conditions should be properly selected in the CFD model. The MV and PVM supply are set as velocity inlets with MV assigned a constant velocity (1 m/s) and PVM assigned a transient velocity (average flow rate of 7.5 L/s). The PVM sinusoidal velocity is taken as input into Fluent through a user defined (UDF) function as seen in equation (1). The PVM and MV inlets are also characterized with inlet temperatures (22°C and 16°C respectively), turbulence intensities and hydraulic diameters. The exhaust grills are chosen as a pressure outlet with zero gauge pressure. Solid surfaces (walls, table chair, occupant…) are set as walls with zero velocity with some of them emitting a constant heat flux such as the computer (93 W), the occupant (100 W) and the walls (10 W/m^2). CO_2 inlets are specified a constant flow rate and a constant velocity. As for discrete phase modeling (DPM) boundary conditions, all solid surfaces are considered as trap boundary condition to account for deposition. The exhaust is assigned an escape boundary condition since particles exit the space through the exhaust grills. Remaining trajectories are tracked in the space.

4. Experimental Work and Validation

In order to assess the effect of frequency on the species and particle concentrations in the space, the CFD model was validated with experiments in a climatic chamber. The experimental room has dimensions of 2.75 m × 2.5 m × 2.8 m. It is conditioned by a MV system supplying recirculated air and assisted by an intermittent PVM nozzle supplying fresh air withdrawn from an adjacent room. The room was equipped with a thermal manikin "Newton" [17] emitting a constant heat flux of 39 W/m^2. Four sources of CO_2 were placed at each wall, and particles were generated using an aerosol particle generator (TSI model 3475) in the critical scenario mentioned previously. The particle concentration was measured using an optical sizer (TSI model 3330) while the species concentration was measured using CO_2 sensors (FIGARO CDM 7160). The validation was done based on values of ventilation efficiency and intake fractions in the surrounding microclimate. It was performed for three frequencies (0.3 Hz, 0.5 Hz and 1 Hz), for a PVM temperature of 24°C and a background temperature of 28°C. The experimental and predicted values of ventilation efficiency and intake fractions for different PVM frequencies and a PVM average flow rate of 3.5 L/s are shown in Table 2.

Table 2. Experimental and predicted values of ventilation efficiency and intake fraction as a function of frequency.

	Frequency (Hz)	ε_v (%)	$IF_{max,\ micro}$ ($\times 10^{-4}$)
Exp	0.3 Hz	66.7±3.2	7.94±0.15
	0.5 Hz	71.23±1.6	7.65±0.2
	1 Hz	70.12±3.8	8.42±0.15
CFD	0.3 Hz	63.86	8.1
	0.5 Hz	72.76	7.72
	1 Hz	66.4	8.3

The results illustrated in Table 2, show that with the increase in frequency to 0.5 Hz, both ventilation effectiveness and intake fractions reach optimal values. However with a further increase in frequency, air quality declines. It can be concluded that there are two opposing effects witnessed with frequency increase: the turbulence effect and the fresh air supply. In the range of [0.3 – 0.5] Hz, the increasing rate of fresh air supply overcame the increasing turbulence (13 % increase in turbulence) while the opposite is true in the frequency range of [0.5 – 1] Hz.

Good agreement was found between experimental and predicted values with a maximum error of 5.3 %/ 1 Hz and 1.9%/0.3 Hz for ε_v and $IF_{max,\ micro}$ respectively.

5. Results and Discussion

The CFD model of the intermittent PVM+MV system is simulated for three typical frequencies found in indoor spaces (0.3 Hz, 0.5 Hz, 1 Hz) and a typical flow rate of 7.5 L/s. Figure 3 represents the contours and streamlines of the velocity and temperature fields for an average flow rate of 7.5 L/s and a frequency of 0.3 Hz.

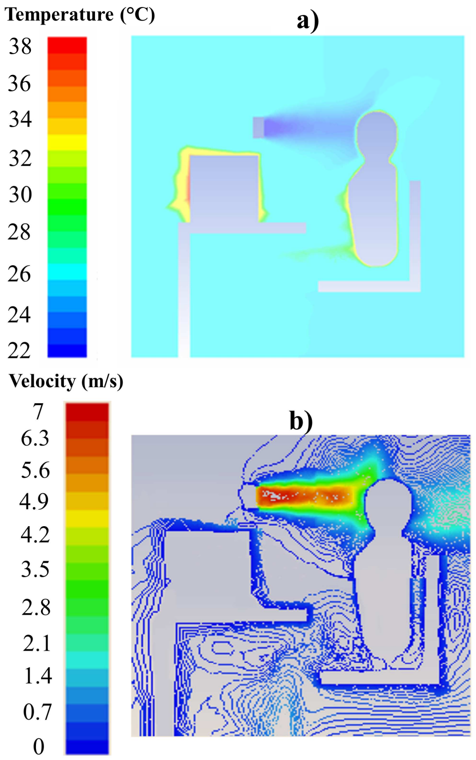

Fig. 3 Illustration of (a) Temperature contours and (b) velocity streamlines for an average PVM flow rate of 7.5 L/s and frequency of 0.3 Hz.

The MV system assures a background temperature of 26°C and the PVM supply temperature is set to 22°C.

Species and particles are considered to be present in the room. Passive and active particles are generated at a seated occupant height of 1.1 m from sources at the walls at a flow rate of 0.5 L/min and 8.4 L/min respectively. The species considered are CO_2 while the active contaminants are 1 µm particles belonging to the inhalable range found indoors [17].

Figure 4 illustrates the ventilation efficiency variation as a function of frequency for an average flowrate of 7.5 L/s. The ventilation efficiency increases with frequency due to increased rate of fresh air supply. It reaches a maximum for 0.5 Hz before declining with a further increase of frequency to 1 Hz due to increased turbulence. Correlating with the results of Al-Assaad et al. [12] on thermal comfort. A compromising frequency between thermal comfort and ventilation efficiency is 0.94 Hz.

Figure 4 represents the effect of PVM varying frequency on intake fraction and deposition fractions for an average flow rate of 7.5 L/s and a particle size of 1µm. With increasing frequency, the intake fraction behaves similarly to the ventilation efficiency with an optimum at 0.5 Hz. As for the deposition fraction, it decreases with increasing frequency. This is due to the fact that with increasing frequency, more particles are entrained into the jet and less are being deposited. A compromise between the intake and deposition fraction is a frequency of 0.78 Hz.

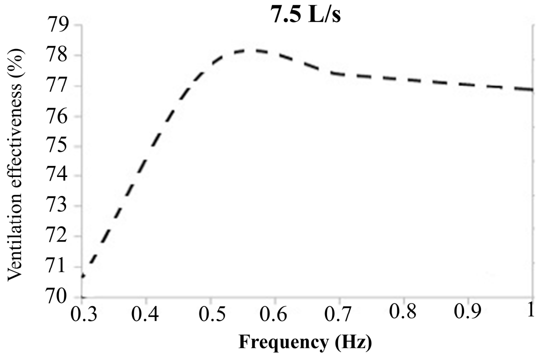

Fig. 3. Variation of ventilation efficiency as a function of frequency for an average flow rate of 7.5 L/s.

Therefore an operational frequency range of [0.78 Hz – 0.94 Hz] is recommended for occupant protection against passive and active particles. It is also noted that this range provided thermal comfort between 0.75 and 0.95 based on the [-4, +4] comfort scale of Zhang et al. [18 – 20] as obtained in the study of Al-Assaad et al. [12].

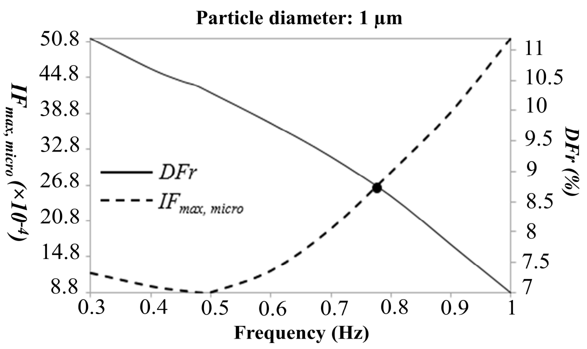

Fig. 4. Intake and deposition fractions as a function of frequency for a PVM average flow rate of 7.5 L/s.

In order to calculate the energy savings of the intermittent PVM+MV system, the fan power consumption as well as the cooling capacity of a steady and intermittent system were computed and compared. The reference steady state condition is chosen to provide the same thermal comfort as the intermittent case which corresponds to a steady PVM flow rate of 9 L/s.

The correlation of Keblawi et al. [21] was used to calculate the fan power consumption:

$$P_f = P_{ref}\left(\frac{\dot{m}_f}{\dot{m}_{f,ref}}\right)^3 \qquad (5)$$

Where $\dot{m}_f$ is the mass flow rate of the fan, P_f is the power consumption of the fan and, P_{ref} and $\dot{m}_{f,ref}$ are the nominal power consumption and mass flowrate respectively for steady state PVM operation. To account for time variation, the power was averaged over one period. The fan power consumption was calculated for transient operation for the optimal frequency range as well as the cooling capacity for both systems. It was found that a transient fan operation (2 W) was slightly higher than the steady fan operation (1.62 W). However, the decrease in cooling capacity was more significant than the increase in fan power consumption between transient and steady state PVM operation (92 W and 111 W respectively). Therefore a transient PVM system operating at an average flowrate of 7.5 L/s and the optimal frequency range resulted in average energy savings of 16.1 %.

6. Conclusion

A transient 3D CFD model was simulated to conduct a study on intermittent PVM assisting a conventional MV system. A parametric study was conuducted to investigate the ability of the system in providing good IAQ while still enhancing thermal comfort. Results show that an intermittent PVM system can provide occupant with protection against passive and active contaminants if operated at a frequency range of [0.78 Hz – 0.94 Hz]. Moreover, this range was also found to provide occupant with thermal comfort as well as reduce energy costs by a percentage of 16.1 %.

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