Numerical Modelling of Large Air-Conditioned Space: Comparison of Two Ventilation Systems

Ali Alzaid*[1], Maria Kolokotroni*[1], Hazim Awbi[2]

1 College of Engineering, Design and Physical Science, Brunel University London
Uxbridge, Middlesex UB8 3PH, UK
AliMohammadA.Alzaid@brunel.ac.uk
Maria.kolokotroni@brunel.ac.uk

2 School of Built Environment, University of Reading
Reading RG6 6AY, UK

ABSTRACT

This paper presents a comparative study based on CFD simulation between the performance of Impinging Jet Ventilation (IJV) and Mixing Ventilation (MV) systems in providing indoor air quality and thermal comfort for a mechanically ventilated occupied large open plan office (floor-to-ceiling height > 5m). Large spaces differ from spaces with standard heights because of the significant upward stratification. The evaluation was carried out using the Air Distribution Index (ADI) which combines several parameters, such as overall ventilation effectiveness for removing pollutants and for temperature distribution ($\bar{\epsilon_c}$,$\bar{\epsilon_t}$), percentage of dissatisfied (PD) and predicted percentage of dissatisfied (PPD). ADI has been used to characterize ventilation systems in spaces with typical floor-to-ceiling heights, but it has not been studied for large spaces to-date. In this study, different test conditions were considered for two cooling loads (48 W/m^2, 68 W/m^2) with full and half occupancy respectively. The model was validated via spot measurements of air temperature, air speed and CO_2 concentration in a large office located in south east England which, is supplied by a mechanical cooling with overhead diffusers providing mixing ventilation. The predicted results were generally in good agreement with measured values of air temperature, air velocity and CO_2 concentration.
The calculated (ADI) for (IJV) and (MV) systems using the validated CFD model showed that the (IJV) system is more effective than (MV) system in removing both pollutants and heat from the occupied zone. It is concluded that using (ADI) to assess the effectiveness of different ventilation systems in large enclosures can provide useful information that combines both indoor air quality and thermal comfort.

KEYWORDS

Large space, Ventilation system, Air Distribution Index (ADI), Numerical model

1. INTRODUCTION

Large single cell spaces have become a considerable characteristic of modern buildings. Examples include shopping malls, lecture theatres, airports terminals, gymnasiums and large open plan offices. They are considered to be large when the floor-to-ceiling height is more than 5 meters (Li *et al.*, 2009). Ventilation of such spaces needs more careful attention by design engineers than spaces with a small volume, i.e. those with a ceiling height of 3 meters or less. This is because the volume of the occupied zone is generally small compared to the entire volume. Therefore specific ventilation strategies which direct the ventilated air and thermal conditioning to the occupied zone are needed to achieve energy efficiency (International Energy Agency (IEA), 1998). Heating, ventilation and air conditioning systems (HVAC) are used to supply the spaces with acceptable levels of air temperature, humidity, air velocity and acceptable indoor air quality by removing the contaminants from the spaces. Therefore, the implementation of a suitable air distribution system will have a significant impact on the air flow patterns and indoor air quality within the ventilated space. To characterise the combined thermal environment and indoor air quality in buildings, Awbi and Gan (1993) and Awbi (1998) developed the concept of the ventilation performance (VP) which was re-named later the Air Distribution Index (ADI) (Awbi, 2003; Karimipanah, Awbi and Moshfegh, 2008). This index is used to assess the thermal comfort, indoor air quality and energy performance of an air distribution system. In 2017, an extensive literature survey was carried out on studies of HVAC systems' performance in large rooms, which found that only three types of room air distribution strategies are used in large rooms. These are displacement ventilation (DV), mixing ventilation (MV) and underfloor air distribution systems (UFAD) (Mateus and Carrilho da Graça, 2017). Also, a fourth system, impinging jet ventilation (IJV) which has been evaluated only in two recent studies by Ye *et al.* (2016) and (Ye *et al.*, 2019) when it was used in the ventilation of a large space. It is traditionally used in typical spaces and described as having promising potential for large spaces.

This paper aims to evaluate the performance of two different ventilation systems: Impinging Jet Ventilation (IJV) and Mixing Ventilation (MV) systems in providing indoor air quality and thermal comfort and consequently energy performance for a mechanically ventilated occupied large open plan office (floor-to-ceiling height > 5m). The Air Distribution Index (ADI) will be used to characterize ventilation systems in large spaces for the first time since it has been studied only in spaces with typical floor-to-ceiling heights.

2. NUMERICAL METHOD

2.1 Physical Models

The large space under consideration is a large open plan office used by research staff and students which has floor-to-ceiling height of 6m is the CSEF Building at Brunel University London. The enclosure has dimensions of 15.5m x14m x 6m and a floor area of 201 m^2 with brick external walls and metallic roof which includes two large skylights. Two big rectangle windows are located on the south side wall of the building with dimensions 3.5m x 1.1m and 4.2m x 1.1m. There is one door on each end wall of the building. For the MV simulation, the external air is delivered into the building interior through a 13m long cylindrical supply duct with a 0.7m diameter. This duct has eight air diffusers located at a height of 3.7 m above the floor with a dimension of 0.8m x 0.15m. Air exhaust is via two return grills located at a height of 3.7m with dimensions 1.0m x 0.5m each, see Figure 1. For the IJV simulation, the supply air is delivered into the building interior through 21 IJV ducts each with a supply opening area of 0.03 m^2 and a distance of 0.7 m above the floor were uniformly distributed inside the office, as recommended by Awbi (2003). 17 IJ diffusers were attached to the walls while the remaining 4 were attached to a column at the centre of the office, as shown in Figure 2.

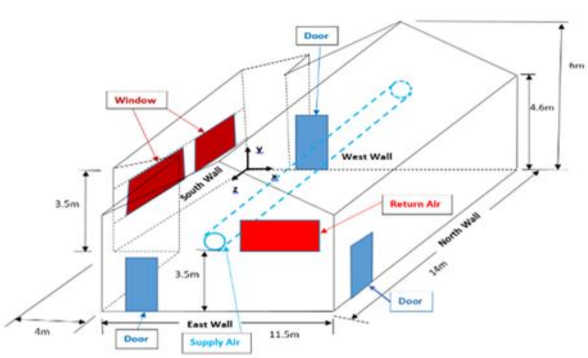

Figure 1: Sketch of the researchers' office with the MV

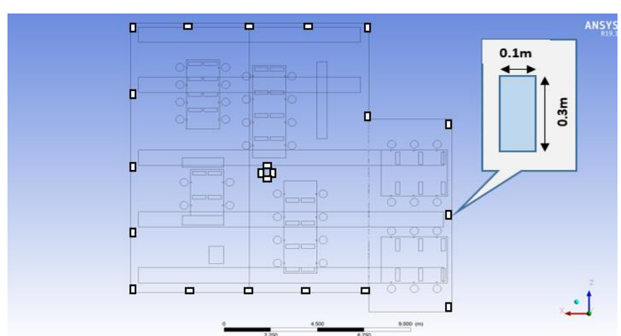

Figure 2: Plan sketch of the researchers' office with the IJV

In this study two different test cases were considered for two occupancy capacity loads; full occupancy (Case F) and half occupancy (Case H) for summer conditions. On this basis, the office was modelled to have 40 and 20 researchers as a full and half occupancy loads along with their computers respectively.
ANSYS Workbench Design Modeller 17.1 with Fluent 17.1 (ANSYS Fluent, 2016) was used to simulate the large space case-study (the researchers' office) at CSEF building. Some simplification of the geometrical model was made to save computing time and power while still preserving the most relevant physical aspects of the geometry. For instance, The body of the occupant was presented as a cylinder of height 1.4m and diameter of 0.4m giving a body surface area of approximately 1.8 m^2 according to (Pinkel, 1958).

2.2 Numerical Model

The finite-volume solver Fluent 17.1 was used to simulate the flow field of the ventilated enclosure, the governing equations were solved with a segregated scheme, and the SIMPLE algorithm was used for coupling the pressure and velocity. In the discretization scheme, the non-linear and the viscous terms were calculated with the second order upwind scheme while the BODY FORCE WEIGHTED scheme was used to reveal convergence when two consecutive iterations for the local airflow variable was less than 10^{-3} whereas for energy when it was less than 10^{-6}. Besides that, the net heat flow rate imbalance was less than 0.003% of the total flux through the system, and the net heat imbalance was less than 0.3% of the total energy flux through the system too.
Several turbulence models can be used for the simulation of turbulent flow. Reynolds-averaged Navier-Stokes (RANS) equation simulation using one or two-equation turbulence models such as RNG k-ε models, SST k-ω model and Reynolds Stress model (RSM) are often used. In fact, the performance of both RNG k-ε and SST k-ω models were compared in the large space by utilising the temperature and air velocity measurements in our previous study (Alzaid, Kolokotroni and Awbi, 2017). The prediction from SST k-ω shows better agreement with measurements in comparison with RNG k-ε model. Hence, the SST k-ω model was used in the present study to predict the turbulent features of air flow within the researchers' office.
Since CO_2 cannot be absorbed or filtered, it can be utilised as a good index for indoor air quality (Oke *et al.*, 2008). The CFD model was developed to predict the metabolic CO_2 distribution within the case-study building. The rate of production of carbon dioxide CO_2 by the human's respiratory system is associated with the metabolic rate as given by the following equation (MacIntyre, 1980):

$$G = 4 \times 10^{-5}\, MA \tag{1}$$

where G is the CO_2 production in (L/s), M is the metabolic rate in (W/m^2) and A is the body surface in (m^2). In this study, an average sedentary adult produces around 0.005 L/s of CO_2 by the respiratory system.
To represent the CO_2 production per person in the CFD model, a separate cell zone shaped as a cube (5 x 5 x 5 cm) was used for modelling species transport. One mass CO_2 source term was added to that cells zone. Therefore, for G = 0.005 L/s ≈ 9.9×10^{-6} kg/s and as it was presented by a cube with a volume of 1.25×10^{-4} m^3, then the CO_2 production per person in the CFD model is 9.9×10^{-6} kg/s divided by 1.25×10^{-4} m^3 which leads to 0.0792 kg/m^3s. Consequently, the CO_2 source term is added in the conservation of the CO_2 mass fraction equation.

2.3 Boundary Conditions

The required supply air flow rates for the modelled office of full and half occupancy were calculated from the following equation (Tymkow *et al.*, 2013; BS EN 16798-3, 2017):

$$\dot{m} = \frac{\dot{q}}{c_p(t_e - t_\infty)} \quad (2)$$

where $\dot{q}$ is the thermal load in (W), $\dot{m}$ is the mass rate of air flow (kg/s), c_p is the specific heat of air (J/kg.K), t_e is the air temperature of air exhaust (°C) and t_∞ is the air temperature of air supply (°C).

To use the above equation, the thermal load $\dot{q}$ can be calculated using HAP 4.9 (hourly analysis program) which is a microcomputer program developed by Carrier Corporation (Carrier, 2010). Therefore, from HAP the cooling loads for the modelled office were calculated to be 68 W/m^2 and 48 W/m^2 for full and half occupancy respectively. Consequently, the calculated supply air flow rates using equation (2) are 2.6 m^3 /s (65 l/s) and 1.7 m^3 /s (85 l/s) for full and half occupancy, respectively.Thus, for the mixing ventilation system, the corresponding diffuser supply velocities were 2.7 m/s for full occupancy and 1.8 m/s for half occupancy. For the impinging jet ventilation system, the corresponding supply velocities were 4.1 m/s for full occupancy and 2.7 m/s for half occupancy.

Due to the high demand of energy needed to cool and dehumidify the outdoor air, recirculating part of the conditioned air is recommended as an essential aspect of energy conservation strategies in many parts of the world (Fadeyi et al. 2009). Many HVAC designers use the recirculated air after mixing it with fresh air then supply it to space after it has been filtered and cooled. In this study, the recirculating air that was contaminated with CO_2 was considered in both cases presented in this section. To find the CO_2 concentration for recirculating air, the supply flow rate (2.6 m^3/s) was used for this study after adding CO_2 source terms in the model to represent the CO_2 production per person. At the start, only fresh air (10 L/s) per person as recommended by ASHRAE standard 62.1 (American Society of Heating, 2016) with 400 ppm CO_2 concentration was used as a supply air flow rate. Then, the CO_2 concentration of the air at the exhaust was determined from the CFD output, which was 808 ppm. Consequently, the CO_2 concentration of the new supply air, which is a mixture of fresh and recirculated air was calculated to be 745 ppm, as shown in Figure 3.

A similar procedure was used to find the CO_2 concentration for recirculated air for the modelled office with half occupancy. The CO_2 concentration of the recirculated air was 686 ppm; while it was 652 ppm for the new supply air, which is a mixture of fresh and recirculated air.

Table 1 summarises the two cases which were used to compare the performance of the MV and IJV systems. It is worth mentioning that according to BS EN 16798-1 (BSI, 2019), the recommended range of the mean design operative temperature for energy saving in open plan offices is (20 – 24 °C) and (22 – 27 °C) for heating and cooling seasons respectively. Moreover, a study by (Seppanen, Fisk and Lei, 2006) recommended that the highest productivity in the office environment is at a temperature of around 22°C. Therefore, in this study, the air supply temperatures presented in the table below were set by an iterative procedure to achieve a rather steady temperature condition of 22.3°C ± 0.4°C in the occupied zone of the office building for providing accurate comparisons.

Therefore, the boundary conditions of Case F and Case H for both MV and IJV systems as flowing:

a. The condition of the air supply is set in the CFD model at values, as shown in Table 1.
b. The boundary conditions for the surfaces temperature of all the office walls were (28 °C), ceiling (35°C), windows (39°C), occupants clothing (33.7°C), personal computers (40°C),

lighting (40°C) and photocopier (40°C) which have been derived from the measurements by (Alzaid, Kolokotroni and Awbi, 2017).

Table 1: CFD cases used for the MV and IJV system performance comparison

Case name		Case F		Case H	
Occupancy		Full		Half	
Distribution system		MV	IJV	MV	IJV
Flow rate (m^3/s)		2.6	2.6	1.7	1.7
Diffusers	Number	8	21	8	21
	Area (m^2)	0.12	0.03	0.12	0.03
	Velocity (m/s)	2.7	4.1	1.8	2.7
	Temperature (°C)	18	19	19	20
	CO_2 (ppm)	745	745	652	652

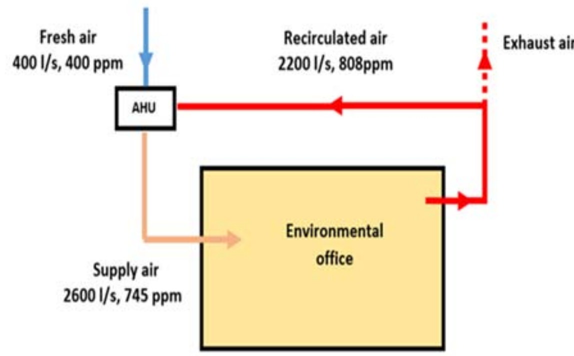

Figure 3: Recirculation system of the modelled office for full occupancy

2.4 Validation of the CFD Model and Computational Grid

The current CFD model for predicting the thermal and airflow fields in a large space is validated with experimental data in our previous study (Alzaid, Kolokotroni and Awbi, 2017). The ANSYS code was used to construct the three-dimensional geometry and generate the mesh. The present study adopted the same mesh strategy as the one used in the validation study, see (Alzaid, Kolokotroni and Awbi, 2017), i.e., non-uniform grid strategy was utilised to cover the whole computational domain for the room. The finer grid was used close to air inlets, outlets and walls, and also in the areas that were anticipated to have steep velocity gradients. Based on that, the total grid size generated for case F and Case H was (16,853,380) and (15,824,322), respectively.

2.5 Air distribution Index (ADI)

The overall ventilation effectiveness for the removal of pollutants ($\bar{\varepsilon_c}$) and the removal of heat ($\bar{\varepsilon_t}$) are used to evaluate the effectiveness of an air distribution system and are defined as (BS EN 16798-3, 2017):

$$\bar{\varepsilon_c} = \frac{C_e - C_\infty}{C_m - C_\infty} \quad (3) \qquad \bar{\varepsilon_t} = \frac{t_e - t_\infty}{t_m - t_\infty} \quad (4)$$

where C is contaminant concentration (p.p.m), t is the temperature (°C), subscripts e, ∞ and m denote exhaust, supply and mean value in the occupied zone. Even though high values of both the overall ventilation effectiveness for the removal of pollutants ($\bar{\varepsilon_c}$) and for heat removal ($\bar{\varepsilon_t}$) state a high efficiency of the ventilation system, they do not alone offer a strong indication of the air quality and thermal comfort in the occupied zone (Awbi, 2003). Awbi and Gan (1993) combined ($\bar{\varepsilon_c}$) with Fanger's Percentage of Dissatisfied (PD) for air quality and ($\bar{\varepsilon_t}$) with Fanger's Predicted Percentage of Dissatisfied (PPD) to define two new numbers that include the percentage of dissatisfied for air quality and thermal comfort, thus

$$N_c = \frac{\bar{\varepsilon_c}}{PD} \quad (5) \qquad N_t = \frac{\bar{\varepsilon_t}}{PPD} \quad (6)$$

where N_c and N_t are air distribution number for air quality and thermal comfort respectively.

The expression for PD and PPD are as follow (Fanger 1988; BSI 2007)

$$PD = 395 \times EXP(-1.83\dot{v}^{0.25}) \quad (7)$$

$$PPD = 100 - 95 \times EXP - \{0.03353(PMV)^4 + 0.2179(PMV)^2\} \quad (8)$$

Where $\dot{v}$ is the outdoor air flow rate (L/s) per standard person and PMV is the Predicted Mean Vote as defined in ISO 7730 (BSI, 2005). The two numbers N_c and N_t have been combined to produce what they called the Air Distribution Index (ADI)(Awbi and Gan, 1993):

$$ADI = \sqrt{(N_c \times N_t)} \quad (9)$$

Achieving a value of $ADI \geq 10$ by a ventilation system designed for an acceptable value of 10% for both PD and PPD can be assumed to be a good air distribution system. However, a large value of ADI would not ensure the existence of a good air distribution system as unwanted local conditions might be still present. Consequently, ADI should be applied along with room temperature distribution and air movement measurements or predictions to evaluate the global and the local conditions particularly the presence of high or low temperature or local draughts regions within the occupied region (Awbi & Gan 1993; Awbi 1998).

3. RESULTS AND DISCUSSION

3.1 Comparison between the Performance of the MV and IJV Systems for Case F

Figures 4 to 6 illustrate the air velocity vectors, air temperature contours, CO_2 concentration contours on a lateral plane located in the middle of the office at Z=6.6m for the MV and Z=7.2 m for the IJV systems for Case F.

From Figure 4 (a) it can be seen that the left supply jet for the MV system is spreading over the floor towards the south (external) wall after impinging on it. In addition, the hot sources and heat gain through the window generate thermal plumes causing considerable buoyancy forces which move the air up to the ceiling. This large and robust air circulation produces the least stratification level by the MV system compared to the IJV system. A cross-recirculation is created in different parts of the occupied zone for both the MV and IJV systems due to partitions and furniture, as shown in Figure 4 (a) and (b). The supplied air from two IJV diffusers spreads along the floor for about 3 m and decays as it moves onto the office floor, as can be seen in Figure 4 (b).

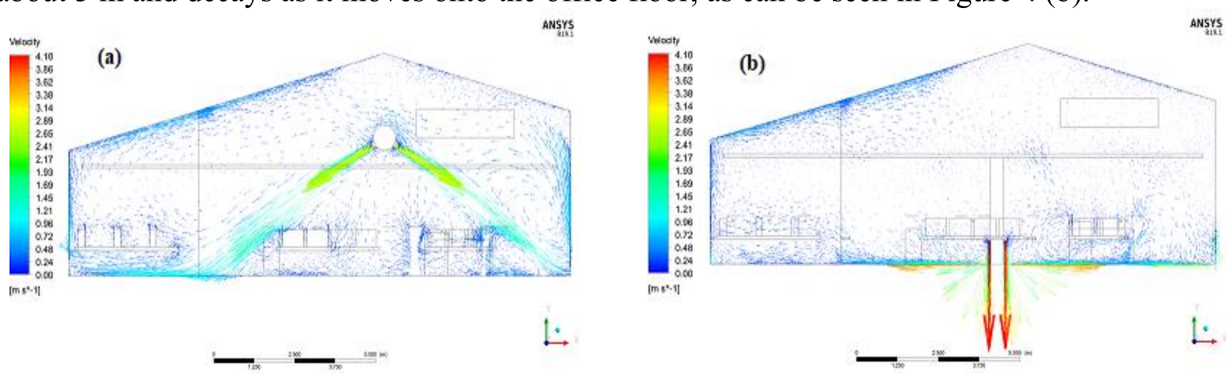

Figure 4: Velocity vector plots on a lateral plane located in the middle of the office for Case F: (a) MV system at Z=6.6 m and (b) IJV system at Z=7.2 m.

Figure 5 (a) and (b) show that with the different supply air temperatures used, 18°C for the MV and 19°C for the IJV, the temperature field in both the MV and IJV systems are somewhat similar when the average temperature in the occupied area was maintained at the same level (22°C). Also, temperature stratification for the office is observed in both contours, which was very significant for IJV supplies, as shown in Figure 5 (b).

The predicted CO_2 concentration on a vertical plane located in the middle of the office for the MV and IJV systems are shown in Figure 6 (a) and (b), respectively. In Figure 6 (a), the concentration of CO_2 in a part of the occupied zone is higher than its mean value in that zone. This is because the supplied air is displaced downward and mixes with the office air apart from that areas covered by the obstructions such as tables. However, for IJV system, the concentration of CO_2 is generally uniform in the occupied zone within the range 750 and 800 ppm. Therefore the IJV system was more efficient in providing fresh air to the breathing zone than the MV system.

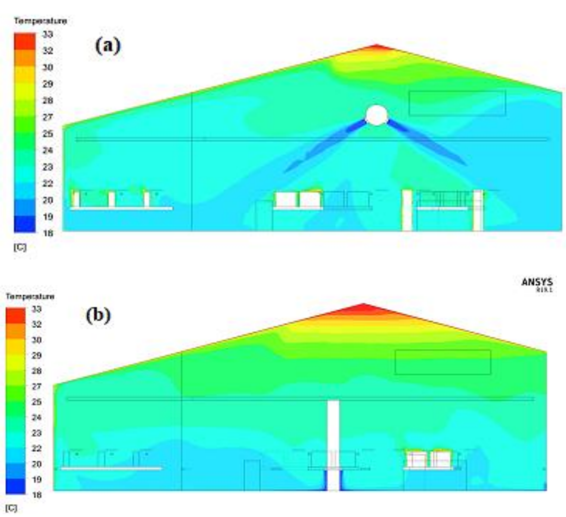

Figure 5: Temperature contour plots on a plane located in the middle of the office for Case F: (a) MV system at Z=6.6 m and (b) IJV system at Z=7.2 m

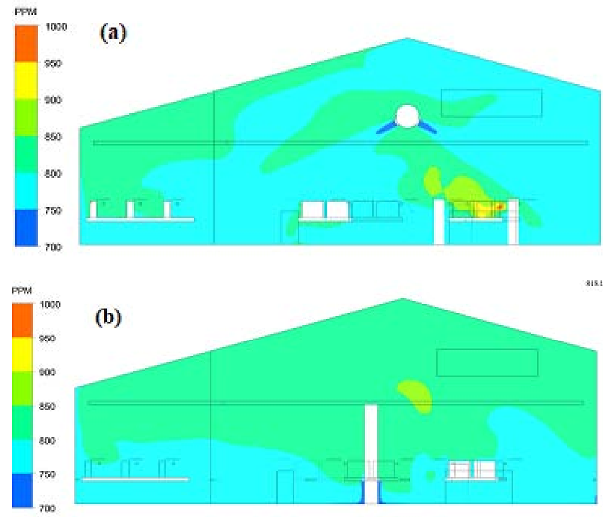

Figure 6: CO_2 concentrations contours on a plane located in the middle of the office for Case F: (a) MV system at Z=6.6 m and (b) IJV system at Z=7.2m

.

The predicted parameters were used to calculate the Air Distribution Index (ADI) for this case, as shown in Table 3. The IJV system was able to remove contaminants and heat ($\overline{\varepsilon_c}$,$\overline{\varepsilon_t}$) from the ventilated space more effectively than the MV system. Also, both air distribution numbers for air quality and for thermal comfort (N_c,N_t) were higher (better) for the IJV than for the MV system and consequently, it shows higher (better) ADI value. However, PPD values for both systems were too high because of dumping of cold and high velocity jet into the occupied zone.

Table 3: The predicted ADI for the MV and IJV systems for Case F

systems	Diffusers					$\overline{\varepsilon}_c$	PD (%)	N_c	$\overline{\varepsilon}_t$	PPD (%)	N_t	ADI
	No.	A (m^2)	V (m/s)	T (°C)	CO_2 (ppm)							
MV	8	0.12	2.7	18	745	1.2	15.2	7.7	1.5	40	3.8	5.4
IJV	21	0.03	4.1	19	745	1.8	15.2	11.7	2.2	31	7.2	9.2

3.2 Comparison between the Performance of the MV and IJV Systems for Case H

Figure 7 to 9 show the air velocity vectors, air temperature contours, CO_2 concentration contours on a lateral plane located in the middle of the office at Z=6.6 m for the MV system and Z=7.2 m for the IJV system for case H. As can be seen from Figure 7 (a), a large clockwise air circulation is present on the left side of the office. The stable thermal stratification near the ceiling tends to slow down this circulation. Also, on the right side of the office, another extensive air recirculation was created with counter clockwise direction. In Figure 7 (b) for the IJV case, the heat sources from the window and the left part of the ceiling created a strong buoyancy force which pushes the air up to the highest part of the office.

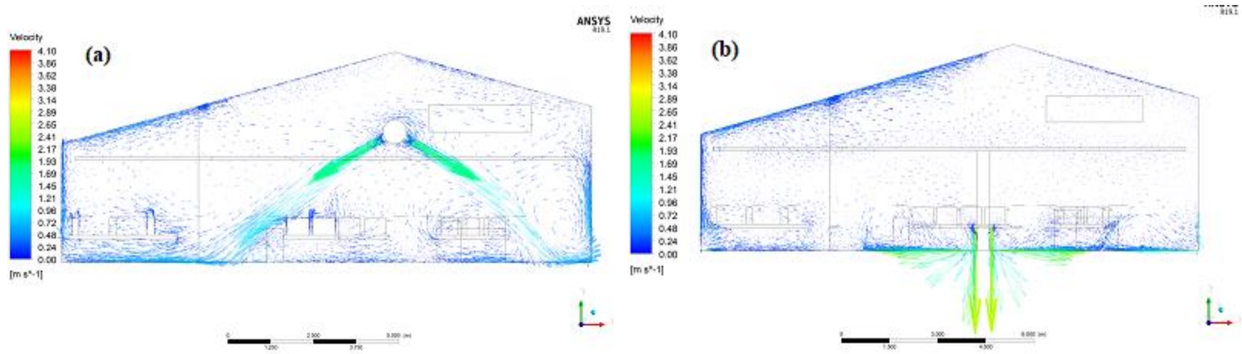

Figure 7: Velocity vectors on a lateral plane located in the middle of the office for Case H: (a) MV system at Z=6.6 m and (b) IJV system at Z=7.2 m.

Figure 8 (a) and (b) shows the temperature contours in the middle of the office and the temperature stratification in the case of MV and IJV systems. The mean air temperature in the occupied zone for the MV system is higher than that for the IJV system. This is because, in the IJV system, the cold jet over the floor has sufficient momentum to continue along the floor despite the heat sources due to the occupants. As for the MV system case, the cold jet momentum was not strong enough to produce good air mixing inside the office. Thus, $\overline{\varepsilon_t}$ value was 1.75 for the MV system compared to 3.5 for IJV system (see table 4).
It is noticed that the occupied zone has a low and uniform CO_2 concentration for IJV system compared to that for the MV system, as shown in Figure 9 (a) and (b). As evidence of that, the overall ventilation effectiveness for contaminant removal ($\overline{\varepsilon_c}$) value is 1.16 for the MV system compared to 2.35 for the IJV system (see table 4).

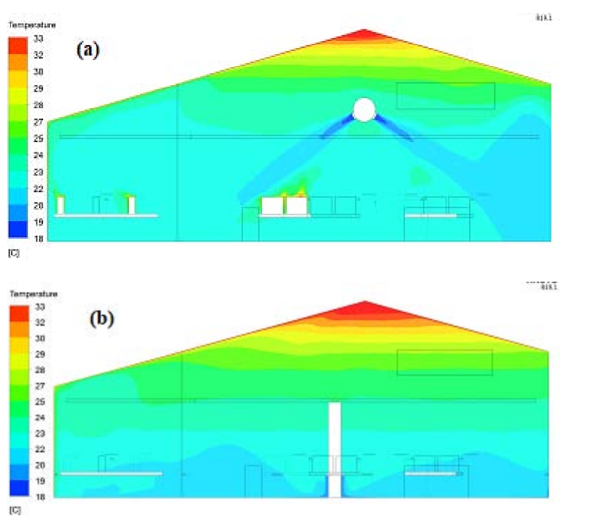

Figure 8: Temperature contour plots on a plane located in the middle of the office for Case H: (a) MV system at Z=6.6 m and (b) IJV system at Z=7.2 m

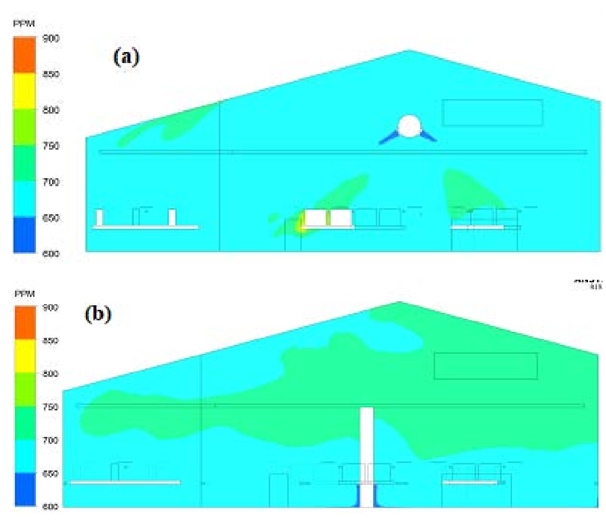

Figure 9: CO_2 concentrations contour on a plane located in the middle of the office for Case H: (a) MV system at Z=6.6 m and (b) IJV system at Z=7.2 m.

Table 4 shows the predicted ADI parameters for the MV and IJV systems for this case. From the table, it can be seen that the air distribution numbers for air quality, N_c, and thermal comfort, N_t, for the MV system were 7.7 and 6.7 respectively while they were double these values for the IJV system. Hence, the results reveal that the highest value of ADI was produced by the IJV system which was 15.

Table 4: The predicted ADI for the MV and IJV systems for Case H

systems	Diffusers					$\bar{\varepsilon}_c$	PD (%)	N_c	$\bar{\varepsilon}_t$	PPD (%)	N_t	ADI
	No.	A (m^2)	V (m/s)	T (°C)	CO_2 (ppm)							
MV	8	0.12	1.8	19	652	1.2	15.2	7.7	1.8	26	6.7	7.2
IJV	21	0.03	2.7	20	652	2.4	15.2	15.4	3.5	24	14.6	15.0

4. CONCLUSION

The ventilation performance of the two air distribution systems used (MV and IJV) were examined by applying the Air Distribution Index (ADI) for the first time in large spaces. Based on the results obtained, the IJV system performed better than the MV system for all the conditions considered as it provided higher ADI values. Since the occupied zone temperature for the two systems was maintained at 22°C, the IJV system performed better than the MV system where the air supply temperature for the IJV system was higher by 1 °C than that for the MV system for both thermal load cases. Thus, the IJV system uses less power than the MV system and also performs better. In general, the ADI concept presented in this work could be a beneficial tool for evaluating a ventilation system performance in large spaces as it provides assessments for both air quality and thermal comfort.

5. REFERENCE

Alzaid, A., Kolokotroni, M. and Awbi, H. (2017) ‘Experimental and Numerical Investigation of Air Distribution in a Large Space’, in. 38th AIVC International Conference.

American Society of Heating, R. and A.-C. E. (2016) *Standard 62.1 User’s Manual: Based on ANSI/ASHRAE Standard 62.1-2016, Ventilation for Acceptable Indoor Air Quality*. ASHRAE.

ANSYS Fluent (2016) ‘Thory guide’. ANSYS Inc,.

Awbi, H. (2003) *Ventilation of buildings*, *Zhurnal Eksperimental’noi i Teoreticheskoi Fiziki*. doi: 10.1080/02786820490519234.

Awbi, Hazim B. (1998) ‘Calculation of convective heat transfer coefficients of room surfaces for natural convection’, *Energy and Buildings*, 28(2), pp. 219–227. doi: 10.1016/S0378-7788(98)00022-X.

Awbi, H B (1998) ‘Energy efficient room air distribution’, *Renewable Energy*. Elsevier, 15(1), pp. 293–299.

Awbi, H. B. and Gan, G. (1993) ‘Evaluation of the overall performance of room air distribution’, in *Proceedings of the 6th International Conference on Indoor Air Quality and Climate, Helsinki*, pp. 283–288.

BS EN 15251 (2007) ‘Indoor environmental input parameters for design and assessment of energy performance of buildings- addressing indoor air quality , thermal environment , lighting and acoustics Contents’, 3(BS EN 15251), pp. 1–50.

BS EN 16798-3 (2017) *Energy performance of buildings — Ventilation for buildings Part 3: For non-residential buildings - Performance requirements for ventilation and room-conditioning*

systems (Modules M5-1, M5-4).
BSI (2005) 'BS EN ISO 7730: 2005: Ergonomics of the thermal environment Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria', *Management*, 3, pp. 605–615. doi: 10.1016/j.soildyn.2004.11.005.
BSI (2019) 'BS EN 16798-1: 2019: Energy performance of buildings - Ventilation for buildings-Part 1:Indoor environmental input parameters for design and assessment of energy performance of buildings addressing indoor air quality, thermal environment, lighting and aco', (BS EN 16798-1).
Carrier (2010) 'hourly analysis program and system design load program'. NY: Syracuse.
Fadeyi, M. O., Weschler, C. J. and Tham, K. W. (2009) 'The impact of recirculation, ventilation and filters on secondary organic aerosols generated by indoor chemistry', *Atmospheric Environment*. Elsevier, 43(22–23), pp. 3538–3547.
Fanger, P. O. (1988) 'Introduction of the olf and the decipol units to quantify air pollution perceived by humans indoors and outdoors', *Energy and Buildings*, 12(1), pp. 1–6. doi: 10.1016/0378-7788(88)90051-5.
International Energy Agency (IEA) (1998) 'IEA Annex 26: Energy Efficient Ventilation of Large Enclosures - Technical Synthesis Report'.
Karimipanah, T., Awbi, H. B. and Moshfegh, B. (2008) 'The Air Distribution Index as an Indicator for Energy Consumption and Performance of Ventilation Systems', *Journal of the Human-Environment System*, 11(2), pp. 77–84. doi: 10.1618/jhes.11.77.
Li, Q. *et al.* (2009) 'CFD study of the thermal environment in an air-conditioned train station building', *Building and Environment*. Elsevier Ltd, 44(7), pp. 1452–1465. doi: 10.1016/j.buildenv.2008.08.010.
MacIntyre, F. (1980) 'The absorption of fossil-fuel CO 2 by the ocean', *Oceanologica Acta*, 3, pp. 505–516.
Mateus, N. M. and Carrilho da Graça, G. (2017) 'Simulated and measured performance of displacement ventilation systems in large rooms', *Building and Environment*. Elsevier Ltd, 114, pp. 470–482. doi: 10.1016/j.buildenv.2017.01.002.
Oke, S. A. *et al.* (2008) 'A mathematical modelling of the intensity of contaminants (C02) on occupancy level of a spacein continuous use'. Inderscience Enterprises Limited.
Pinkel, D. (1958) 'The use of body surface area as a criterion of drug dosage in cancer chemotherapy.', *Cancer research*, 18(3), pp. 853–856.
Seppanen, O., Fisk, W. J. and Lei, Q. H. (2006) *Effect of temperature on task performance in officeenvironment*. Ernest Orlando Lawrence Berkeley NationalLaboratory, Berkeley, CA (US).
Tymkow, P. *et al.* (2013) *Building services design for energy efficient buildings*. Routledge.
Ye, X. *et al.* (2016) 'Heating energy consumption of impinging jet ventilation and mixing ventilation in large-height spaces: A comparison study', *Energy and Buildings*. Elsevier B.V., 130, pp. 697–708. doi: 10.1016/j.enbuild.2016.08.055.
Ye, X. *et al.* (2019) 'Comparison study of contaminant distribution and indoor air quality in large-height spaces between impinging jet and mixing ventilation systems in heating mode', *Building and Environment*. Elsevier, p. 106159.