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Ventilation for good indoor air quality and energy efficiency

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Abstract

As a result of new energy efficiency directives and legislations in Europe and elsewhere, the ventilation component of energy usage in buildings has increased relative to the total energy consumption. At the same time, the air quality in some buildings has in recent years deteriorated as building designers and managers have been aiming to design more air-tight buildings and reduce the energy consumption. This article gives a brief overview of the types of mechanical ventilation and air distribution systems that are used for buildings. It briefly describes the performance of traditional and some new types of ventilation systems in terms of indoor air quality (IAQ) provision. A method for evaluating the performance of air distribution systems that uses the ventilation and energy effectiveness is then introduced. This is based on the Air Distribution Index which has two different expressions, depending on the nature of the room environment in which the air distribution system is used. One method is for use in uniform environment and the other for both uniform and non-uniform conditions. The two methods are then applied to different types of room air distribution to compare their performances in terms of IAQ provision for occupants and energy efficiency.

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*Keywords:*Room air distribution; ventilation systems, indoor air quality

1. Introduction

The energy consumption for heating, cooling and ventilating buildings often accounts for the largest part of a country's energy usage which is still mainly based on fossil fuels.

There is a great global emphasis on reducing the reliance of buildings on fossil fuel energy and a move towards Nearly Zero Carbon Buildings (NZCB).

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Achieving this goal will require a rethink of the traditional designs of and the types of systems currently in use. The proportion of ventilation energy in comparison with the total energy use in a building is expected to increase as the building fabric energy performance improves and ventilation standards recommend higher ventilation rates for improving indoor air quality (IAQ). At the same time, new building regulations [1, 2] are imposing air-tight construction which could inevitably impact on indoor air quality, health (e.g. sick building syndrome) and human productivity in future buildings [3].

Despite recent advances in building ventilation and air distribution systems [4, 5, 6, 7], it is evident that complaints about poor IAQ have increased in recent years [8, 9, 10, 11, 12]. There is a need therefore for assessing current methods of building ventilation and developing ventilation systems that are capable of providing good IAQ and energy performance to satisfy building occupants and, at the same time, meet new building energy regulations.

This article gives a brief overview of the various types of mechanical ventilation and air distribution systems that are used for buildings and introduces methods based on the Air Distribution Index (ADI) for evaluating and comparing their performances in terms of IAQ provision and energy performance. Although comparison is only given for limited cases, the aim is to inform and encourage the research community to continue research for developing new room air distribution methods and apply the ADI concepts to evaluate their performances.

Nomenclature

ACH	air change rate per hour
ADI	air distribution index
ADI_{New}	modified (new) air distribution index
c	contaminant concentration (ppm)
c(0)	initial contaminant concentration at $t = 0$ (ppm)
$c_p(t)$	contaminant concentration at a certain point in the room at time t(ppm)
E	fan power ($E \alpha \Delta pq$), W
N_{AQ}	modified (new) air quality number
N_c	air quality number
N_t,	comfort number
N_{TC}	modified (new) thermal comfort number
PD_{aq}	Percentage of Dissatisfied with the indoor air quality
PPD	Predicted Percentage of Dissatisfied
q	airflow rate (m^3s^{-1}),
$\lvert S \rvert$	absolute overall thermal sensation over the exposure time based on the Bedford 7-point scale
t	time (s) or air temperature (°C)
t_x	local air temperature (°C)
t_c	average (control) air temperature (°C)
v_x	local air velocity (ms^{-1})

Greek

Δp	pressure difference across fan ($\Delta p \propto q^2$), Pa
ε_c	ventilation effectiveness for contaminant removal
ε_t	ventilation effectiveness for heat removal
θ	effective draught temperature (K)
τ_n	room time constant (s)
$\bar{\tau}_p$	local mean age of air (s)

Subscripts

$_{i, m, o}$	inlet, mean and exhaust values for the occupied zone (e.g. to a height of 1.8m)

2. Types of ventilation systems

Ventilation is the process of mixing or replacing contaminated indoor air with fresh air from outside the building to reduce the level of indoor contaminants. The main types of ventilation methods used for general room ventilation are listed below, some of which are classical and some are less common. More information of these and other types of systems in use can be found in [13].

2.1. Mixing ventilation (MV)

The principle behind a MV system is to mix fresh air with contaminated room air to provide a fresh supply of air and lower the contaminant concentrations [4].An air jet is normally supplied in the upper parts of the room (ceiling or wall at high level) at a high velocity (typically > 2.0 $m.s^{-1}$) to provide air circulation around the room. With a properly designed system, the resulting temperature and contaminant concentration in the occupied zone should be quite uniform. However, MV is known to have lower ventilation effectiveness when compared to other air distribution systems, as will be shown later.

2.2. Displacement ventilation (DV)

A DV system is based on the principle of displacing contaminated room air with fresh air from outside [5]. Cool air is normally supplied at low velocity (typically < 0.5 m/s) at or near the floor to create an upward air movement (thermal plumes) as it warms up by heat sources in the room. This will normally create vertical gradients of air velocity, temperature and contaminant concentration. This is usually a more energy efficient air distribution system as it requires lower fan power and has higher ventilation effectiveness than mixing ventilation. However, it has limitation in terms of low penetration distances and low cooling capacity (< 40 W per m^2 of floor area) and it is unsuitable for heating [14].

2.3. Personalized ventilation (PeV)

The previously mentioned room air distribution systems are designed to provide fresh air to the whole of the occupied space. In certain types of occupancy it may be sufficient to supply the fresh air directly to the occupants, i.e. office desk or hospital bed. This is known as “Personalized Ventilation” (PeV), sometimes also referred to as Task/Ambient Conditioning (TAC), and theoretically this method of air supply should reduce the quantity of fresh air that is required for a ventilated space. PeV aims to provide fresh air directly to the breathing zone of every occupant in the space to improve the air quality and increase occupants' satisfaction [15]. However, this type of system is not for general application as it restricts occupants’ movement and sometimes causes excessive draft on the occupants’ faces.

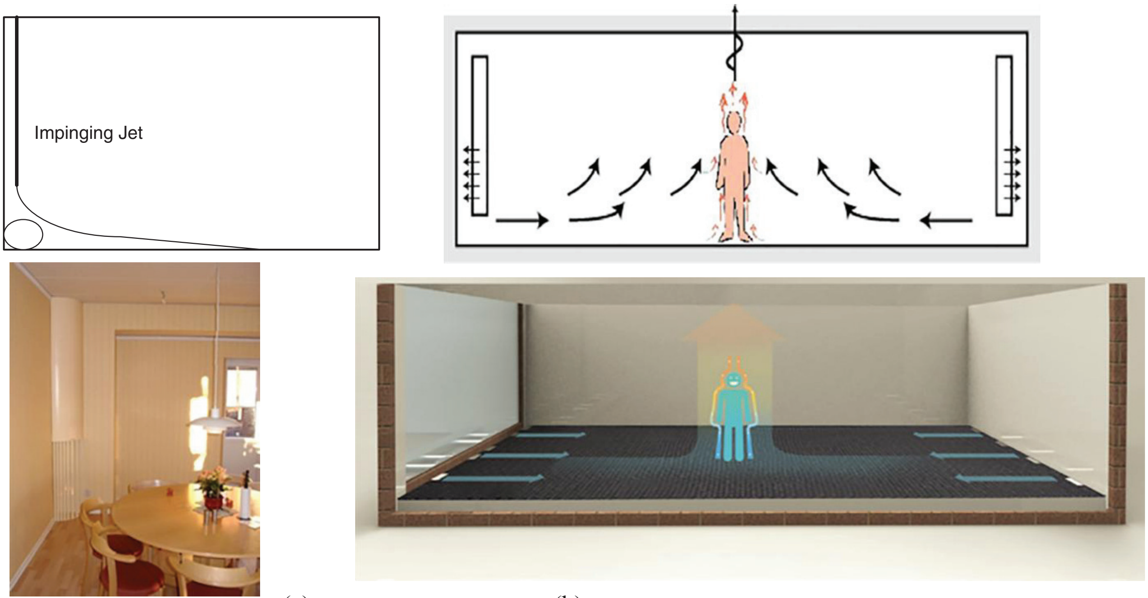
2.4. Hybrid air distribution (HAD)

Although a DV system usually provides a more efficient method of air supply, it suffers from two main problems: (1) it cannot be used in the heating mode; (2) the air supply has a limited penetration depth into the room. A hybrid air supply system combines the characteristics of both MV and DV systems to overcome the shortcomings of the DV system. Some such systems have recently been developed, such as the impinging jet (IJ) system and the confluent jet (CJ) system, see Figure 1. The distribution of airborne contaminants is similar to that in a DV system because of the low-level fresh air supply in both the IJ and CJ methods of air distributions.

An impinging jet air supply consists of a vertical duct supplying air towards the floor which then spreads over the floor area as a thin jet. In confluent jets system (CJ), a number of jets issuing from closely spaced slots or circular apertures in the same flow directions merge together a short distance downstream to form a single jet normally close to a room surface such as a wall or floor. The combined jets are then directed towards the floor to create a similar effect to that from an impinging jet system, thus producing a greater horizontal spread over the floor than a displacement jet system [14, 16, 17, 18]. The characteristics of CJ are similar to the IJ in terms of being a higher momentum room air supply rather than a buoyancy-driven flow as it is in the case for a DV system.

Studies involving IJ and CJ systems have shown that these methods of room air supply are capable of providing considerably better air quality performance and at the same time require less energy than the MV system [19] and a higher cooling capacity than DV systems [20, 21].

In this article the performance of systems based on the hybrid method is compared with the mixing and displacement methods of air distribution, which are the most commonly used methods presently.



(a) (b)

Fig. 1. Hybrid air distribution systems; (a) impinging jet; (b) confluent jets

3. Performance of room air distribution systems

It is now well recognised that the air distribution system has a major influence on various indoor environment parameters, such thermal comfort, indoor air quality, energy efficiency, etc. One of the first methods for assessing the effectiveness of air distribution was the Air Distribution Performance Index (ADPI) [22]. Due to its limitations however, more advanced methods for assessing the real performance of the air distribution in a room, which include the indoor air quality and energy effectiveness, have been developed by the author and co-workers. These are known as the Air Distribution Index (ADI) and the New Air Distribution Index (ADI_{New}). These and *ADPI* are described in the following sections.

3.1. Air Diffusion Performance Index (ADPI)

The ADPI was primarily developed for understanding air jet distribution but does not address real thermal comfort parameters nor indoor air quality. In this method, the performance of an air distribution system is assessed by the level of draught created in the occupied zone of a room due to the supply air jet. Using the *Effective Draught Temperature* concept, an analytical procedure was established [23] that is based on the measured values of air temperature and velocity at uniformly spaced points throughout the occupied zone or in a vertical, centre-line plane through the air supply outlet. The number of points at which the draught temperature (θ), defined by Equation (1), satisfies a specified comfort limit, expressed as a percentage of the total number of points, was defined as the *Air Diffusion Performance Index* (*ADPI*).

$$\theta = (t_x - t_c) - 8\,(v_x - 0.15) \tag{1}$$

The *ADPI* has been found to be a function of the type of air terminal device, the room load, the supply air flow rate and the room geometry. The dependency of *ADPI* on a number of diffuser and room related factors excludes its use as a universal index and more significantly, it does not address the most important room parameters such as thermal comfort and air quality. Despite this drawback the index is extensively used in North America and some other countries [24] and is recommended by ASHRAE [22] for evaluating the performance of air terminal devices. Another disadvantage of this method is the uncertainty in defining an acceptable limit of *ADPI* for comfort. A value of 80% is usually considered to be the minimum acceptable [24].

3.2. Air Distribution Index (ADI)

The efficiency of an air distribution system is based on its effectiveness for contaminant removal from the ventilated space as well as its effectiveness in distributing the thermal energy of the supply air. The ADPI concept, for example, does not provide information on these two parameters and is therefore not capable of determining the true performance of an air distribution system. Therefore, an effective ventilation index should be able to account for the effectiveness of contaminant removal as well as thermal energy distribution in the ventilated space. This requires the evaluation of two parameters; one is the ventilation effectiveness (ε_c) which is defined as:

$$\varepsilon_c = \frac{c_o - c_i}{c_m - c_i} \tag{2}$$

The second parameter that describes the energy performance of an air distribution system can be assessed using the effectiveness for heat removal (ε_t), defined as:

$$\varepsilon_t = \frac{t_o - t_i}{t_m - t_i} \tag{3}$$

The comfort number, N_t, and the air quality number, N_c, [25] which combine ε_t and ε_c with the Predicted Percentage of Dissatisfied(PPD) and the Percentage of Dissatisfied with the indoor air quality(PD_{aq}) respectively [26, 27], are considered to be useful indicators of the level of performance of a ventilation system. These are defined as:

$$N_t = \frac{\varepsilon_t}{PPD} \quad , \quad N_c = \frac{\varepsilon_c}{PD_{aq}} \tag{4}$$

These two numbers can be combined into a single parameter that determines the effectiveness of an air distribution system in providing air quality and thermal comfort in the form of a ventilation parameter which is defined as:

$$ADI = \sqrt{N_t \times N_c} \tag{5}$$

This parameter is called the Air Distribution Index (ADI) [25].

The Air Distribution Index (ADI) has been developed to take into consideration the overall performance of an air distribution system [19, 25]. As discussed earlier, this index incorporates the Predicted Percentage of Dissatisfied, *PPD* (which is derived from the thermal sensation index Predicted Mean Vote, PMV), the air quality index the Percentage of Dissatisfied, PD_{aq} [26, 27], in addition to the two expressions for effectiveness given by Equations (2) and (3). The PD_{aq} is related to the breathing zone (nose) of the occupant, however the PMV is based on providing a uniform distribution of the physical comfort parameters throughout the occupied space and therefore an acceptable PMV value for the occupants may not be achieved if there are significant local variations of these parameters within the occupied zone. Therefore, the *ADI* is suitable for "uniform" thermal environment in the occupied zone but may not be adequate in situations where, for example, a large degree of thermal stratification is present as in the case of displacement ventilation. For non-uniform environment, a modified(new) Air Distribution Index(ADI_{New}) has more recently been developed as described in the following section.

3.3. New Air Distribution Index (ADI_{New})

Since the thermal comfort part of the *ADI* index is based on Fanger's PMV/PPD model, the index can be effective in evaluate the performance of a ventilation system that is able to provide uniform thermal environment (e.g. a mixing system) but it could be less accurate in evaluating non-uniform environments, such as in the case of a displacement ventilation system. Hence, a modified air distribution index (ADI_{New})which can be used to assess the performance of a ventilation system that produces either uniform or non-uniform thermal environments has recently been developed [21, 28].

The New Air Distribution Index $(\mathrm{ADI})_{\mathrm{New}}$ combines the thermal comfort and air quality numbers as follows:

$$ADI_{New} = \underbrace{\left[\left(1 - \frac{|S|}{3}\right) . \varepsilon_t\right]}_{N_{TC}} + \underbrace{\left[\left(\frac{\tau_n}{\bar{\tau}_p}\right) . \varepsilon_c\right]}_{N_{AQ}} \tag{6}$$

The local mean age of air ($\bar{\tau}_p$) is calculated using the following equation:

$$\bar{\tau}_p = \frac{1}{C(0)} \int_0^\infty c_p(t)dt \tag{7}$$

The room time constant τ_n is defined as:

$$\tau_n = \frac{1}{ACH} \tag{8}$$

The logic behind $(\mathrm{ADI})_{\mathrm{New}}$ is that when the occupant's thermal sensation is neutral (i.e. $|S|= 0$), which is the ideal thermal condition, $\mathrm{N_{TC}}$ reaches its maximum value and when $|S|$ reaches its extreme values (i.e. -3: cold or +3: hot), $\mathrm{N_{TC}}$reaches its minimum value (zero). The value of $|S|$ can be obtained using a multi-node thermal sensation model, such as the CBE model [29]. A high value of ε_t implies that the ventilation system is more efficient in removing heat (or coolness) from the occupied zone. ε_c, $\bar{\tau}_p$ and ACH are factors that are used to determine the value of N_{AQ} that has an impact on the value of $\mathrm{ADI_{New}}$. A high value of ε_c and low values of both $\bar{\tau}_p$ and ACH mean that the ventilation system is efficient in removing contaminants as well as in providing fresh air to the occupied zone. Therefore, the $(\mathrm{ADI})_{\mathrm{New}}$ index developed in the form shown by equation (6) could be a useful tool for evaluating both thermal sensation and air quality, whether the thermal environment is uniform or non-uniform.

4. Applications of *ADI* and *ADI_{New}*

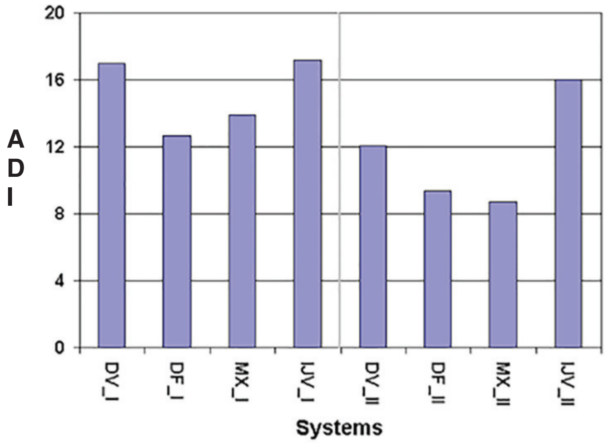
4.1. Evaluation of air distribution using ADI

Using ADI, the performance of four air distribution systems were compared using data from measurements in an environmental test chamber and computational fluid dynamics (CFD) simulations. Figures 2 shows the results of tests carried out in an environmental test chamber [6] comparing four types of air distribution systems for cooling loads of 36 and 60 W per $m^2$of floor area. In reference [6], the ADI has the former title Ventilation Parameter (VP) but it is exactly the same parameter as ADI. It can be seen from Figure 2 that the performance of a system can be measured by the value of ADI it achieves, i.e. the higher the value (e.g. > 10) the better the performance, providing that there are no adverse comfort conditions such as draught present. The results show that although mixing ventilation can perform reasonably well at the higher cooling load, it is out performed by the DV and IJ systems for both cooling loads (36 and 60 $W.m^{-2}$). Furthermore, although the IJ system showed similar airflow tendencies to the DV system, it was found from the test results to have a better balance between buoyancy and momentum forces and produced slightly better velocity distributions and ventilation performance than the DV system [6].

In a CFD study involving the *ADI* index, the air flow rates for four different types of air distribution systems that are required to achieve equal indoor environment were compared [19]. Using the equation for fan power (E):

$$E \propto q^3 \tag{9}$$

the air supply rate required to obtain an $ADI \sim 16$ was compared for four systems, see Table 1. The lowest flow rate was found to be for the confluent jets system (0.025 m^3 s^{-1}). For the impinging jet system the flow rate is 1.4 times higher and requires 2.74 times more fan power than a confluent jet system. The fan power consumption of the impinging jet system is about half that of the mixing system. The displacement system requires 1.1 times greater flow rate and 1.33 times higher power consumption compared to the confluent jets.



(DV=Wall Displacement Ventilation; DF=Floor Displacement Ventilation; MX=Mixing Ventilation; IJV=Impinging Jet Ventilation. Cases I and II refer to cooling loads of 36 and 60 W m^{-2} respectively)

Fig. 2. The Air Distribution Index from tests on four air distribution systems

Table 1. Comparison between the fan power consumption of four air distribution systems for almost equal *ADI* values

Ventilation system	ADI	Total flow rate (m^3s^{-1})	Fan energy usage compared to CJ
Mixing ventilation (MV)	15.5	0.045	5.83
Impinging jet ventilation (IJ)	15.7	0.035	2.74
Displacement ventilation (DV)	15.9	0.0275	1.33
Confluent jets ventilation (CJ)	16.1	0.025	1.0

4.2. Evaluation of air distribution using ADI_{New}

The thermal comfort and air quality numbers required for calculating the ADI_{New} index were obtained for mixing and displacement ventilation systems from tests in an environmental test chamber (2.78m x 2.78m x 2.3 m high) using 4 males and 4 females (one subject used for each test) for providing the thermal sensation data [21].

The measured air temperature distribution in the occupied zone of the chamber was used to calculate ε_t whereas the measured CO_2 concentrations were used to calculate ε_c and local the mean age of air ($\bar{\tau}_p$) at the breathing zone. The occupants' overall thermal sensation $|S|$ was calculated from the CBE thermal comfort model [29] which was also checked for accuracy with the subjects' votes. The results for the two ventilation systems are given in Table 2. These tests were conducted with a supply air flow rate of 15 ls^{-1} and supply temperature of 18 ^oC with a total room cooling load of 21.2 Wm^{-2} (ventilation load of 9 Wm^{-2}) of floor area.

Although the occupants' thermal sensation value ($|S|$) for the mixing ventilation (MV) system was slightly better than that for the displacement ventilation (DV) system, the DV system produced better thermal comfort number (N_{TC}) and air quality number (N_{AQ}) and this is attributed to its better performance in removing heat and air pollutants from the occupied zone (higher ε_c and ε_t) compared with the MV system.

Table 2. Parameters for the $(ADI)_{New}$ index based on measurements with male and female subjects for two ventilation systems

Ventilation System	$\|S\|$ *	ε_t	N_{TC}	τ_n (hr)	$\bar{\tau}_p$(hr)	ε_c	N_{AQ}	ADI_{New}
MV	|0.146|	0.952	0.906	0.333	0.578	1.036	0.597	1.503
DV	|-0.431|	1.129	0.967	0.333	0.493	1.096	0.740	1.707

Average overall thermal sensation votes for male and female subjects

Almesri [30] performed CFD simulations for the test chamber mentioned above occupied by a breathing manikin and ventilated by DV, IJV and MV systems with the same heat load, see Figure 3. The ventilation effectiveness (ε_c) and the local mean age of air ($\bar{\tau}_p$) were calculated using the concentration of CO_2 at different locations in the chamber and similarly ε_t was obtained using the predicted temperature distribution in the chamber. The overall thermal sensation was also calculated using the CBE thermal comfort model [29]. The CFD simulations were coupled with the thermal comfort model to produce the thermal sensation values. As in the measured cases above, the CFD predictions were performed for 15 l s^{-1} air supply rate and supply air temperature of 18 ^oC but a higher ventilation load (cooling) of 17 W m^{-2}. The predicted results for the three methods of air distribution are given in Table 3.

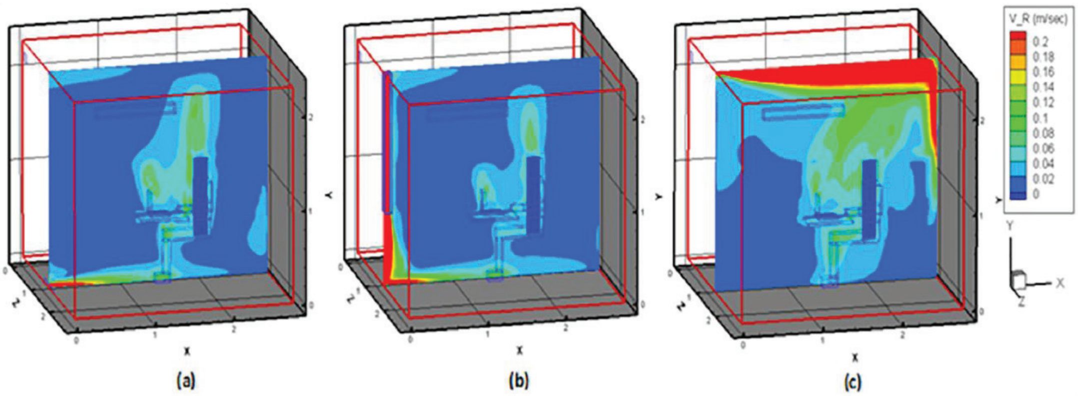


Fig. 3. CFD simulations of the performance of three air distribution systems (a-DV system, b- IJ system and c- MV system)

Table 3. Parameters for the $(ADI)_{New}$ index based on CFD simulations (Air flow rate = 15 l s^{-1} and air supply temperature = 18 ^oC in each case)

Ventilation System	$\|S\|$	ε_t	N_{TC}	τ_n (hr)	$\bar{\tau}_p$ (hr)	ε_c	N_{AQ}	$(ADI)_{New}$
DV	|0.17|	1.10	1.04	0.33	0.23	1.35	1.98	3.02
IJ	|0.34|	1.08	0.95	0.33	0.25	1.28	1.68	2.63
MV	|0.49|	0.97	0.81	0.33	0.40	0.93	0.76	1.57

The results in Table 3 show that under these air supply conditions (15 ls^{-1} - equivalent to 3.64 $ls^{-1}m^{-2}$ floor area - and 18^{o}C supply air temperature), the DV system produced higher thermal comfort and air quality numbers ($\boldsymbol{N_{TC}}$ and $\boldsymbol{N_{AQ}}$) than those for the IJ and MV and consequently it showed a better (higher) ADI_{New} value than the other two systems. A close second was the IJ system and far behind was the MV system. These simulations and others given in [30] confirm the findings from experimental measurements in an environmental chamber using human subjects as given in Table 2.

4. Conclusions

In this article the performances of different room air distribution systems were compared using the concepts of the Air Distribution Index (*ADI*) and the New Air Distribution Index (ADI_{New}). Although the older *ADI* method, which uses the *PPD* concept for thermal comfort assessment, can be applied when uniform room environmental conditions are present, it is not generally applicable as generally room environmental conditions are rarely uniform. However, for non-uniform thermal environment, the ADI_{New} index was found to be more appropriate for assessing such situations (often when buoyancy is dominant) as it is coupled with a multi-node thermal sensation model in additional to some of the parameters used in *ADI* such as $\varepsilon_c$and ε_t. Both indices can easily be determined using CFD simulation software coupled with a thermal sensation model or using data from measurements in test chambers.

Comparisons between the performances of some unconventional room air distribution systems, namely the imping jet and confluent jets systems, with the conventional mixing system have demonstrated that better indoor air quality at lower ventilation energy requirement can be achieved using these more recently developed systems. It was also shown that the conventional displacement ventilation (DV) system generally performed well. However, it is well-known that the DV system has limitations on the amount of cooling that can be delivered and the penetration depth of the DV jet. Furthermore, this system cannot be used for heating. On the other hand, the other two systems that were also shown to provide good performance, namely the impinging jet and confluent jets systems, are capable of providing higher cooling loads, penetrate deeper into the room, because of the larger supply jet momentum, and can be used for both cooling and heating.

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