Table .3. Bulk temperature at $H/2$ = 2.5 m for q = 800 W

Case	Inclination angle						
	20°	30°	40°	50°	60°	70°	90°
Heated sections S1 and S2	31.7	30.1	29.7	28.5	27.8	27.3	27.3
Heated sections S3 and S4	33.1	31.6	30.0	29.4	28.9	28.5	28.2

distance from the heated side behaves differently from the previous case. First the air temperature decreases rapidly with a decreasing angle and then the changes are slower.

RESULTS

Theoretically the relation between the total heat input, q, and the flowrate, Q, in the air gap is a power law relation $Q \sim q^{\gamma}$ with γ equal to 1/2 (laminar flow) or 1/3 (turbulent flow). To great extent the recorded values lie within this interval.

When changing the inclination angle, ϕ, the change in flow rate follows the theoretical, $Sin\ \phi$, relation. The effect of a shift in position of the solar cell module along the facade or roof is close to the theoretical shape factor and therefore an important parameter. For an aspect ratio in the range 20 to 110 the relation between a change in the aspect ratio and the change in flow rate follows a power law relation with an exponent starting at 0.44, for the lower heat fluxes, and decreases to 0.25.

For a roof at different inclination angles, shifting the position of the module from the upper side to the lower side of the air gap effects the temperature of the module and the bulk temperature of the air, but not much the generated flow rate. When positioned on the lower side, the side-to side convection becomes more important for the heat transfer and therefore the temperature of the module decreases rapidly with a decreasing angle and then the changes are slower while for module positioned on the upper side an almost linear relation between the air temperature and the angle is observed, the air temperature increases by a decreased angle.

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ENERGY EFFICIENT ROOM AIR DISTRIBUTION

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ABSTRACT

An environmental chamber has been used to compare the effectiveness of mixing and displacement ventilation in terms of heat and contaminant removal. Results are presented for CFD simulations of the air movement in the chamber and for measurements using a heated mannequin with displacement ventilation. The CFD simulations and the measurements suggest that displacement ventilation is more energy efficient than a mixing system.

KEYWORDS

Ventilation; room air movement; indoor air quality; ventilation effectiveness; thermal comfort.

INTRODUCTION

As the standard of building insulation has been increasing in the last two decades, the energy requirement for ventilation is becoming, in most cases, larger than the building fabric's heat loss/gain and internal load gains. Another factor which has contributed to the increase in ventilation rate is the increase in indoor pollutants which has resulted into the specification of larger outdoor air flow rates to maintain acceptable indoor pollution concentrations. Although there are various heat recovery methods which can be applied to reduce the impact of high ventilation rates on the energy consumption for a building, e.g. AIVC TN 45 (1994), the method of distributing the air also has some effect not only on the energy consumption but also on the indoor air quality, see Awbi and Gan (1993).

In room air distribution, there are usually two methods of supplying the air: either mixing ventilation (dilution) or displacement ventilation. In mixing ventilation, air is normally supplied at high level over the ceiling which is then deflected down into the occupied zone by the opposite walls thus causing a mixing of the air jet with room air. In displacement ventilation, the air is supplied at low level, usually over the floor, and then rises up due to buoyancy before it is extracted at high level.

In this paper the effect of the method of distributing the air in the space on the air quality, thermal comfort in the form of PMV/PPD and other thermal and air quality parameters is discussed. The impact of these parameters on the energy requirement for room ventilation is also discussed. The study is based on simulating the air movement, heat and contaminant distribution in an environmental chamber

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resembling a small office room using computational fluid dynamics and also measurements in the environmental chamber with a heated mannequin.

VENTILATION PARAMETERS

The purpose of ventilation is the provision of uncontaminated air and the removal of internally produced contaminants from the building. Very often the process also involves the distribution of heat or coolth to the building. It is, therefore, a process which involves both diffusion and convection. The pattern of air flow in the room being ventilated will largely determine the diffusion and convection processes. This is influenced by the characteristics of the air supply and the room characteristics. Such characteristics will involve air jet velocity and momentum, temperature difference between the air jet and room air, position and type of air supply device, distribution of room heat and contaminant sources, etc.

There are many parameters which can be quantified by measurement or CFD simulation which can be used to assess the effectiveness of a ventilation system. The most commonly used parameters are:

Ventilation Effectiveness for Heat Distribution or Removal (ε_t)- This is similar to a heat exchanger effectiveness and is defined by:

$$\varepsilon_t = \frac{T_o - T_i}{\bar{T} - T_i} \tag{1}$$

Ventilation Effectiveness for Contaminant Removal (ε_c)- This is a measure of how effective the ventilation system is in removing internally produced contamination. It is defined by:

$$\varepsilon_c = \frac{C_o - C_i}{\bar{C} - C_i} \tag{2}$$

In equations (1) and (2), T is temperature (°C), C is the contaminant concentration in parts per million (ppm), the subscripts i and o refer to inlet and outlet respectively and ($\bar{\ }$) represents the mean value for the occupied zone (to a height of 1.8 m). The values of ε_t and ε_c is dependent on the method of room air distribution, room characteristics, heat and contaminant sources, etc.

An efficient ventilation system is one that can achieve good thermal comfort and air quality in the occupied zone. Although high values of ε_t and ε_c represent a high performance in terms of heat distribution and contaminant removal, these quantities alone do not give a good indication of the thermal comfort and air quality in the occupied zone. Fanger has developed expressions for the predicted percentage of dissatisfied (PPD) with the thermal environment (1972) and the percentage of dissatisfied (PD) with the quality of indoor air (1988). Other numbers for thermal comfort and air quality which are more suited to room air movement may be expressed as (see Awbi and Gan, 1993):

$$N_t = \frac{\varepsilon_t}{PPD} \tag{3}$$

$$N_c = \frac{\varepsilon_c}{PD} \tag{4}$$

Expressions for PPD and PD are found in Fanger (1972, 1988).

CFD SIMULATIONS

The CFD simulations were carried out using the program VORTEX (Gan & Awbi, 1994). This program has been extensively used for simulating the air flow in mechanically and naturally ventilated enclosures. The program has the capability of calculating the thermal comfort indices PMV and PPD, the air quality index PD as well as the parameters ε_t, ε_c, N_t, N_c and the local mean age of air distribution.

ROOM DESCRIPTION

The room which has been used for this study is the Environmental Chamber at the University of Reading. This chamber consists of a small environmental control compartment and a working compartment, Fig. 1. The working compartment has internal dimensions of 2.78 m x 2.78 m x 2.3 m ceiling height. This compartment has been used for the CFD simulations and for carrying out measurements of the "age of air" using a tracer gas (SF_6) in a displacement ventilation mode. The simulations were carried out for both high-level air supply representing mixing ventilation and low-level displacement ventilation.

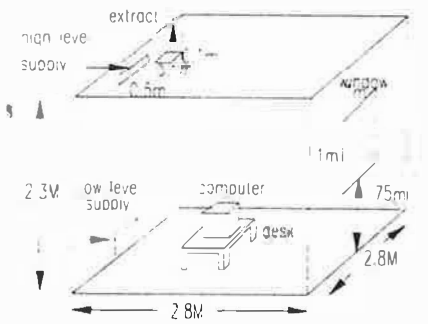

Fig. 1 Schematic of the working chamber

The air was supplied to the chamber from a flat surface displacement ventilation (DV) unit with about 4,000 perforated holes, 2 mm diameter each. The face area of the unit was 0.5 m wide x 0.5 m high, thus giving a free area of about 5%. A mannequin made of 1 mm thick aluminium plate and heated internally using resistance wires in the legs, the torso and the head to produce a skin temperature resembling that of a human was used in the measurements. The mannequin was also provided with a small hole in the nose to sample the tracer gas concentration from the nose. In addition to the heat input by the mannequin, there were heat loads from a box simulating a computer and electric lights. The ventilation load for the room varied between 13 to 48 Wm^{-2} floor area. The higher cooling load represents the upper limit of displacement ventilation. Further description of the chamber, the mannequin and experimental measurements can be found in Hatton and Awbi (1998).

The CFD simulations were carried out for cooling and heating of the room. For simplicity, the air supply terminals were chosen as rectangular openings. In the case of high-level mixing ventilation the air was supplied at the top of a wall and in the low-level displacement ventilation, it was supplied from the same wall at floor level. A contaminant source producing 0.1 ls^{-1} was placed in the middle of the room at a height of 1.2 m in each of the cases simulated. In all these simulations the air supply rate is 20 ls^{-1}. The loads used in the simulations are given in Table 1.

Table 1 - Room loads used in the CFD simulations

Load	Surface Area (m^2)	Cooling Load	Heating Load
Window gain (loss)	1.00	120	(-60)
Curtain wall gain (loss)	5.40	55	(-55)
Computer	0.87	100	100
Total (W)		274	(-15)
Total (Wm^{-2})		35	(-2)

RESULTS AND DISCUSSION

Results of CFD Simulations

A total of 10 simulations were carried out using VORTEX covering high-level mixing ventilation and low-level displacement ventilation. Apart from two cases, cooling conditions were used in the simulations. Table 2 summarises the conditions of the 10 simulations. The results are plotted in Figs. 2~6 with the supply jet momentum as the abscissa. It is known in room air distribution research, e.g. Awbi (1998), that the jet momentum is the main parameter which affects the room air movement. If the buoyancy force is significant then this too will influence the air movement.

Table 2. Data for CFD simulations

Sim. No.	A_{in} (m^2)	V_{in} (ms^{-1})	M_{in} (m^4s^{-2})	T_{in}	τ_{oz}	V_{oz} (ms^{-1})	T_{oz} (°C)	PMV_{oz}	ε_t	ε_c	Type of air supply
1	0.005	4.0	0.08	15.0	1.118	0.081	23.4	-0.042	134.9	96.6	High level
2	0.01	2.0	0.04	15.0	1.051	0.053	22.9	-0.076	142.9	96.0	High level
3	0.015	1.33	0.027	15.0	1.058	0.053	22.42	-0.132	152.5	101.6	High level (dumping)
4	0.02	1.0	0.02	15.0	1.017	0.043	22.4	-0.123	153.2	101.9	High level (dumping)
5	0.05	0.4	0.008	18.0	0.914	0.024	22.31	-0.170	268.3	104.0	Displacement vent
6	0.10	0.2	0.004	18.0	0.893	0.020	22.17	-0.207	277.6	102.9	Displacement vent
7	0.25	0.08	0.0016	18.0	0.884	0.019	22.07	-0.232	280.8	102.0	Displacement vent
8	0.5	0.04	0.0008	18.0	0.882	0.019	22.07	-0.234	283.7	103.2	Displacement vent
9	0.05	0.4	0.008	27.0	1.004	0.029	22.32	-0.967	8.47	98.8	Displacement vent (heating)
10	0.005	4.0	0.08	29.0	1.17	0.036	22.43	-0.87	8.08	90.14	High level (heating)

A_{in}=inlet area. V_{in}=inlet velocity. M_{in}=jet momentum. T_{in}=inlet temperature. subscript oz refer to occupied zone

Figure 2 gives the mean air velocity in the occupied zone for the cooling cases. As would be expected, the velocity in the case of displacement ventilation is much lower than that for mixing ventilation because the jet momentum is low. The kink in the line for mixing ventilation represents the drop of the cold jet into the occupied zone (dumping) because of the lower jet momentum in the case of simulations 3 and 4. If these two points are excluded, one can see almost a linear relationship between the mean velocity in the in occupied zone and jet momentum for the two types of ventilation systems.

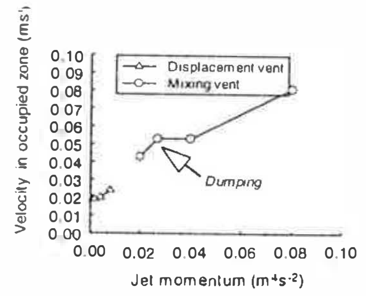

Fig. 2 Mean velocity in occupied zone

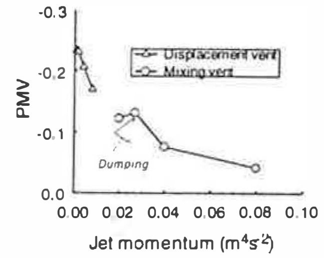

Fig. 3 Predicted Mean Vote

The *Predicted Mean Vote* (PMV) results are shown in Fig. 3. The PMV in the case of displacement ventilation is lower than that for mixing ventilation since in the former the air is supplied directly into the occupied zone. However, for all the simulations with cooling, the PMV > -0.5 which is required for thermal comfort, Fanger (1972). The PMV increases with increase in jet momentum (except for cases 3 and 4) presumably because the air movement, and hence the thermal distribution, is better for a higher jet momentum.

The *Effectiveness of the Heat and Contaminant Removal* (ε_t and ε_c) are plotted in Fig. 4. Both ventilation systems give a value of ε_c = 96~104% which is an indication of good contaminant removal effectiveness. However, there is a large difference in ε_t between the two ventilation systems. The displacement system is much more efficient in the removal of heat than the mixing system. Furthermore, the mixing system has produced unacceptable room environment in the case of simulations 3 and 4 as result of jet 'dumping'. The *Thermal Comfort and Air Quality Numbers* (N_t and N_c) are plotted in Fig. 5. The trend is similar to that of Fig. 4 for the ventilation effectiveness.

The *Mean Age of Air* $\langle\tau_{oz}\rangle$ in the occupied zone is shown in Fig. 6. This is the average of the local mean age of all the points in the room. The local mean age for a point is defined as the average time it takes for air to travel from the inlet to that point. The results in Fig. 6 have been normalised with respect to the time constant (τ) for the room, i.e. τ = Room Volume / Ventilation Rate. The value of $\langle\tau_{oz}\rangle$ for the displacement system is considerably lower than that for the mixing system and is also lower than the room constant. As in the case of the air velocity in the occupied zone (Fig. 2), here also there is a linear correlation with the jet momentum, if cases 3 and 4 are excluded (dumping).

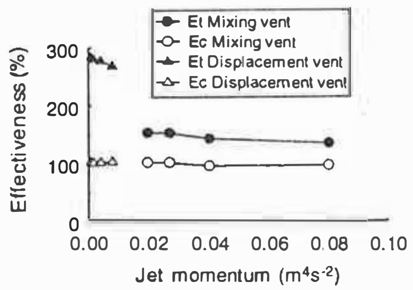

Fig. 4 Thermal and contaminant removal effectiveness

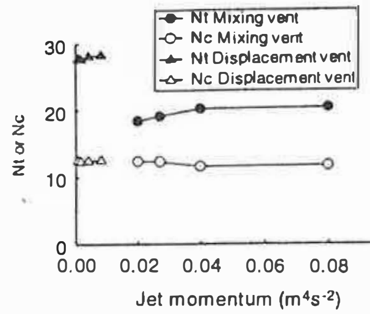

Fig. 5 Thermal comfort and air quality numbers

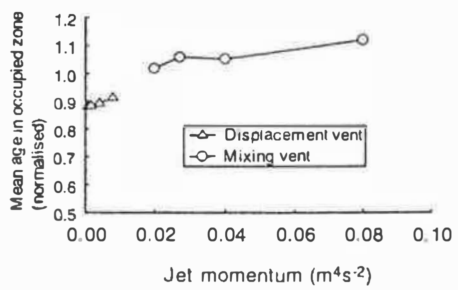

Fig. 6 Mean age of air in occupied zone

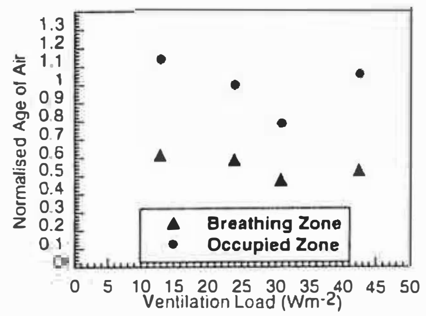

Fig. 7 Measured age of air

The air distribution for the two heating conditions was found to be unacceptable. In the case of displacement ventilation (Simulation 9) the low momentum, high temperature air supply jet at floor level rose upward soon after it entered the room and combined with the plume rising from the computer. This produced high temperature near the ceiling and low temperature in the occupied zone. In Simulation 10 the warm air jet supplied close to the ceiling did not have sufficient momentum (even though the jet has the highest momentum used in the simulations) to achieve an acceptable mixing. In both simulations, the thermal condition in the occupied zone was uncomfortable which is reflected by the low PMV value, see Table 2.

Although N_t and N_c have been found to give a good overall assessment of the thermal condition and the air quality in a ventilated room, they are not good indicators of local conditions. A typical example is the case of dumping of the cold jet into the occupied zone (Simulations 3 and 4) in which case high values of N_t and N_c were obtained but it was found from the air flow patterns in the room that local discomfort will occur as a result of dumping. These numbers should therefore be used with room air movement patterns to assess the room condition.

<u>Experimental Results</u>

Figure 7 shows the age of air at the breathing zone (1.2 m from floor) of the mannequin and the mean age for the occupied zone as a whole with displacement

ventilation. The age values have been normalised with respect to the value at the exhaust. It can be seen that the age at the breathing zone is about 40% lower than that for the occupied zone. This is due to the entrainment of the 'fresh air' supplied over the floor by the body plume which is then carried upward to the nose.

CONCLUSIONS

The results from the CFD simulations show that the *Ventilation Effectiveness for Contaminant Distribution* (ε_c) is almost the same for mixing and displacement ventilation. However, the *Ventilation Effectiveness for Heat Distribution* (ε_t) for displacement ventilation is almost twice the value for mixing ventilation. This means that displacement ventilation is more energy efficient for cooling a room.

The *Thermal and Air Quality Numbers* (N_t and N_c) can be good indicators of the thermal and air quality conditions in ventilated rooms. A high value of N_t or N_c suggests that the ventilation system is energy efficient. However, other qualitative or quantitative information will be required to obtain a comprehensive assessment of the room condition, such as room air movement patterns.

The experimental results with a heated mannequin show that the age of air at the breathing zone is about 40% lower than the mean value of the occupied zone with a displacement ventilation system. This is an indication that the air reaching the nose is less contaminated than that in the occupied zone and hence this can lead to a reduction in the energy requirement for cooling and dehumidifying the air supplied to a room.

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