A reduction in the floor surface temperature, by means of a cooled floor, contributed towards better comfort conditions. The temperature gradient was made smaller and the turbulence eliminated.

The Laser-Doppler-Anemometry (LDA) measuring technique enabled these phenomena to be studied. The results would not have been feasible using traditional hot-wire anemometry.

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MODELING THE THERMAL AND INDOOR AIR QUALITY PERFORMANCE OF VERTICAL DISPLACEMENT VENTILATION SYSTEMS

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ABSTRACT

To predict the thermal and indoor air quality performance of vertical displacement ventilation systems using two-zone modeling, it is necessary to account for the different nature of the air flow due to thermal and contaminant mixing within these zones. Two zone modeling of vertical displacement ventilation was performed assuming piston flow in the clean zone, uniform mixing in the dirty zone, and no recirculation between the zones except via heat source plumes. The resulting equations for air change effectiveness and contaminant removal effectiveness more closely match observed vertical displacement ventilation performance than previous studies, especially under non-isothermal conditions with heat sources.

INTRODUCTION

Previous experiments with a vertical displacement ventilation system performed in a 2.8m x 2.6m x 3m test room during early summer and winter show that a room with heat loads (30-170 W/m^2) is divided into two zones: a lower, "clean" (unmixed), zone with temperature stratification connected by heat source plumes to an upper, "dirty" (mixed) zone with an almost uniform temperature distribution (1). Contaminants added to the upper zone mix with the turbulent air flow in that zone while contaminants added to the lower zone do not mix but, ascend until entrained by heat source plumes. Past attempts to mathematically model vertical displacement ventilation systems have not fully accounted for the different nature of the air flow and temperature distributions within the upper and lower zones (2, 3). These models have typically assumed uniform temperatures and contaminant mixing within each zone, and a recirculation factor applied between the two zones. For these models, air change effectiveness and contaminant removal effectiveness can only be evaluated for a specified recirculation factor and clean zone height, both of which are not easily quantified for a given room condition. The effects of internal heat loads and temperature stratification on air flow patterns and the resulting air quality have not been accounted for in these models.

Results described herein, indicate that modeling a vertical displacement ventilation system is improved by assuming: 1) piston flow in the clean zone and uniform mixing in the dirty zone; 2) no recirculation between zones except via heat source plumes; 3) that the clean zone height can be expressed as a function of heat loading and supply air flow conditions.

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METHOD

Figure 1 shows streaklines calculated by a computer simulation (1) (heater input of $100 W/m^2$, supply air flow rate $100 m^3/h$, and supply air temperature 19°C). The heat source is located at half of the ceiling height. Figure 1(a) shows the paths traveled by ten marker particles from their respective starting points located 0.5m above the floor. Initially, the particles primarily move upward, indicating a piston-like or displacement flow in this zone until they are entrained by the heat source plume. In Fig.1(b), the particles released at locations higher than, and in front of, the heater recirculate with the turbulent air flow in the upper zone. Both of these observations were verified experimentally(1). Thus it is reasonable to assume piston flow in the lower zone and uniform mixing in the upper zone. Particles are transported from the lower zone to the upper zone via heat source plumes only; thus in this model it was assumed that no recirculation occurs between the zones except via heat sources.

In vertical displacement ventilation, as shown in Fig. 2, a warm air zone is assumed to form in the upper part of the room by the thermal plumes generated by heat sources, e.g. office equipment, lamps, people, warm walls, etc. The warmer air will not descend below a certain level because of its lower density. As each plume entrains ambient air, it will grow, and therefore, the flow rate of the plumes ($\dot{V}_{Pi}$, i=1,2,3, ...) is a function of their height above the heat sources, Z_{Pi}. As there is no recirculation between the zones, except via heat source plumes, in steady conditions the warm zone height, Z, is the point at which the sum of all the plume flow rates is equal to the supply air flow rate, $\dot{V}$. Therefore:

$$\dot{V}_{P1} + \dot{V}_{P2} + \dot{V}_{P3} + \cdots = \dot{V} \qquad (1)$$

Air change effectiveness

Assuming a room is dosed with a tracer gas in the supply duct until equilibrium is obtained and then dosing is turned off (tracer decay test), the mass balance equations for the dirty zone 1 are:

$$V_1(dC_1/dt) = \dot{V}C_2(0) - \dot{V}C_1(t); \qquad (0 < t \le V_2/\dot{V}) \qquad (2)$$
$$V_1(dC_1/dt) = -\dot{V}C_1(t); \qquad (V_2/\dot{V} < t < \infty) \qquad (3)$$

where t denotes time $\dot{V}$ the supply air flow rate C_1 and C_2 are the contaminant concentrations in zone 1 and 2, and, V_1 and V_2 are the volumes in zone 1 and 2. Solving the above equations for C_1 allows a determination of the room mean age of air, $<\tau>$, and air change effectiveness, ε_a:

$$C_1(t) = C_1(0) = C_2(0); \qquad (0 < t \le V_2/\dot{V}) \qquad (4)$$
$$C_1(t) = C_1(0)\exp[-(\dot{V}/V_1)(t-V_2/\dot{V})]; \qquad (V_2/\dot{V} < t < \infty) \qquad (5)$$
$$<\tau> = (V^2+V_1^2) / (2\dot{V}V) \qquad (6)$$
$$\varepsilon_a = \tau_n/<\tau> = 2/[1+(V_1/V)^2] \qquad (7)$$

where $V=V_1+V_2$, and $\tau_n=V/\dot{V}$. Thus, the air change effectiveness can be expressed as a function of the dirty zone volume ratio V_1/V, where the dirty zone volume ratio is a function only of the heat load and room ventilation conditions.

Contaminant removal effectiveness

Figure 3 diagrams a single room ventilation system including recirculation and an air cleaner located in the supply air system. Contaminant generation can occur in the system at the rate of N_0, in zone 1 at the rate of N_1 and in zone 2 at the rate of N_2. Although in reality there will be some lateral diffusion in the lower zone, for the purposes of this model it is adequate to assume that N_2 is immediately entrained into thermal plumes.

Mass balance equations for zone 1 and zone 2 are:

$$\dot{V}C_2 + N_1 + \dot{N}_2 = \dot{V}C_1 \qquad (8)$$
$$\dot{V}C_{in} + N_2 = \dot{V}C_2 + N_2 \qquad (9)$$

with the concentration of contaminant in the supply air given by:

$$C_{in} = (1-\varepsilon_0)(1-\sigma)C_1 + (1-\varepsilon_0)\sigma C_0 + (N_0/\dot{V}) \qquad (10)$$

and C_0 is the concentration of contaminant in outside air; N_0, N_1, and N_2 are the contaminant generation rates in the supply air system, zone 1, and zone 2, respectively; ε_0 is the air cleaner efficiency; and σ is the outside air ratio. For the simple case where $C_0=0$, $N_0=0$, $\varepsilon_0=0$ and $\sigma=1$:

$$C_1 = (N_1+N_2)/\dot{V} \qquad (11)$$
$$C_2 = 0 \qquad (12)$$
$$\varepsilon^C = Ce(\infty)/<C(\infty)> = C_1/((C_1V_1+C_2V_2)/V) = 1/(V_1/V) \qquad (13)$$

with $C_e(\infty)$ as the concentration of contaminant at exhaust and $<C(\infty)>$ as the room mean concentration of contaminant. The contaminant removal effectiveness can be expressed as a function of the dirty zone volume ratio, V_1/V, which is only a function of the heat load, and room ventilation conditions. The dirty zone volume ratio can be estimated by equations proposed by Koganei (1) which are:

$$H(1-V_1/V) = 0.64\,(bl)^{0.5}Ar^{-0.2} + h \qquad (14)$$
$$\text{with:} \quad Ar = g\beta\,(T_a - T_{in})\,b/u_{in}^2 \qquad (15)$$
$$T_a - T_{in} = P/(2\dot{V}\rho C_p) \qquad (16)$$

where b is the height of the diffuser opening (m), l is the length of the diffuser opening (m), H is the room height (m), g is the gravitational acceleration (m/s^2), P is the heat load (W), $\dot{V}$ is the supply air flow rate (m^3/s), T_a is the mean room air temperature (°K), T_{in} is the supply air temperature (°K), u_{in} is the diffuser inlet air velocity (m/s), C_p is the specific heat of air (J/kg °K), β is the coefficient of volumetric expansion (1/°K) , and ρ is the density of air (kg/m^3).

An experiment performed in a vertical displacement ventilation test room (4.2m x 4.4m x 3.2m) provides data to check the validity of this model (4). The tests were conducted at two different flow rates (510 and $1700 m^3/h$) and two heat load conditions (13 and $44 W/m^2$) using SF_6 and tobacco smoke as contaminants.

RESULTS

Figure 4 presents the predicted air change effectiveness calculated from equation (7) compared to earlier model predictions by Mundt (2) and Mathisen (3) (note that the recirculation from zone 1-to- zone 2 is set to zero for references (2, 3)). As shown in the figure, when the dirty zone does not exist ($V_1/V = 0$), this model predicts $\varepsilon_a=2$, which agrees with the expected piston flow condition. When the clean zone does not exist ($V_1/V = 1$), ε_a of 1 is predicted - again in agreement with the observed fully mixed flow conditions in the dirty zone. Thus, these results are more intuitively correct than the predictions of references (2,3).

Figure 5(a) shows the contaminant removal effectiveness, ε^C, as calculated in equation(13). Experimental results for several kinds of contaminants indicate values of ε^C which vary from 1.3 to 3.6 at $V_1/V=0.55$ and 0.31.

Figure 5(b) shows the contaminant removal effectiveness, ε^C, as a function of air cleaner efficiency ε_0, and outside air ratio, σ, with outside air having $C_0=0$ and zero contaminant generation rates in the system, $N_0=0$, $V_1/V=0.4$ (this is the occupied zone height of 1.8m assuming a room ceiling height of 3m). As σ approaches 0 (no outside air), higher efficiency air cleaning is required to achieve the equivalent contaminant removal effectiveness values.

Figure 6 shows an example of the predicted dirty zone volume ratio calculated from equations(14), (15) and (16) as a function of flow rate and heat load. For these particular curves, b=0.25m, l=1.6m, h=1.6m, H=3.5m, and the floor area was 15m^2.

DISCUSSION

This model calculates air change effectiveness based on the room mean age of air. The calculations can also be based on the average age of air in the occupied space or at breathing level by using the appropriate local contaminant concentration value in this model.

Contaminant removal effectiveness is evaluated by assuming that contaminants added to the lower zone are immediately transported to the upper zone via heat source plumes and thus have no effect on the contaminant concentrations in the lower zone. Experimental results show that the lower zone does not have pure piston flow since low levels of contaminants were measured in this zone with a clean air supply to the test room. The variance in the data, shown in Fig. 5(a), may be due to different diffusion and recirculation rates for each of the contaminants. This may be due to their different densities and the varying temperature of the contaminant sources. A more exact simulation should include diffusion and recirculation terms in the lower zone, however, the fundamental characteristics of displacement ventilation phenomena are represented well in this model.

This modeling is based on the existence of thermal plumes within the room. When there are no heat sources, the upper warm dirty zone is not formed, and thus the dirty zone volume ratio becomes 0, corresponding to piston flow conditions over the entire room. As shown in Fig. 6, the dirty zone volume ratio V_1/V becomes small when the heat load is small. However, in the actual phenomena with only small or nonexistent heat loads, the piston flow condition may be very unstable or sensitive to small disturbances, so, there may be some diffusion and not perfect piston flow.

CONCLUSIONS

Modeling of vertical displacement ventilation has been improved by assuming that:
1) piston flow in the lower "clean" zone and uniform mixing in the upper "dirty" zone; and
2) recirculation between the zones does not occur except via heat source plumes. The primary strengths of this model are:

1) Fundamental characteristics of displacement ventilation systems are more realistically simulated by including consideration of heat load, room ventilation, and system conditions (e.g. outside air ratio, air cleaner location and efficiency, total heat load, supply air flow rate, etc).

2) It is unnecessary to know the details of the plume flow conditions. These are complicated functions of various conditions (e.g. surface temperature and the shape of the heat load, and temperature around the heat loads, etc.).

ACKNOWLEDGEMENT

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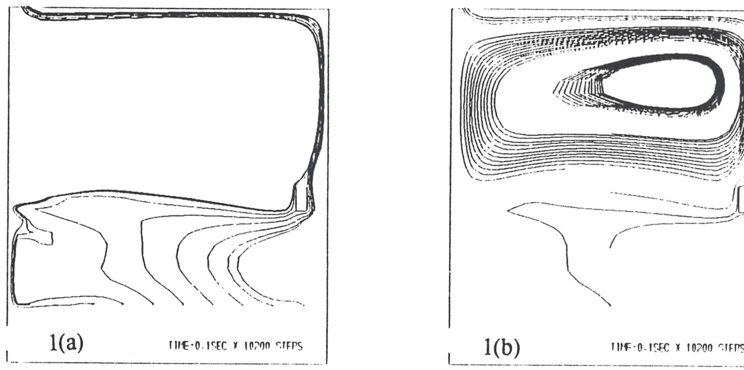


Figure 1. Streaklines of marker particles in computer flow simulation.

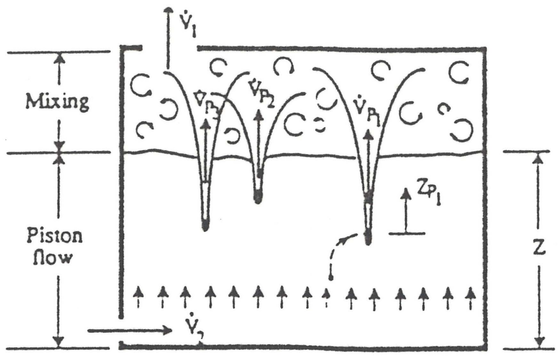


Figure 2. Schematic of two-zone model.

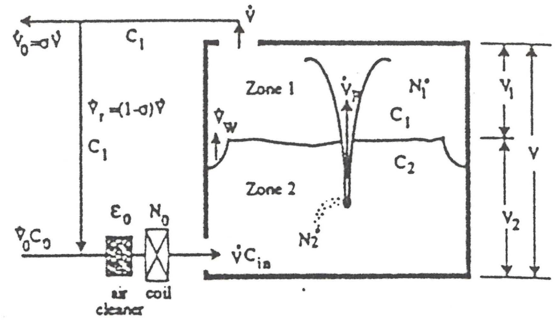


Figure 3. Diagram of test room and HVAC system model.

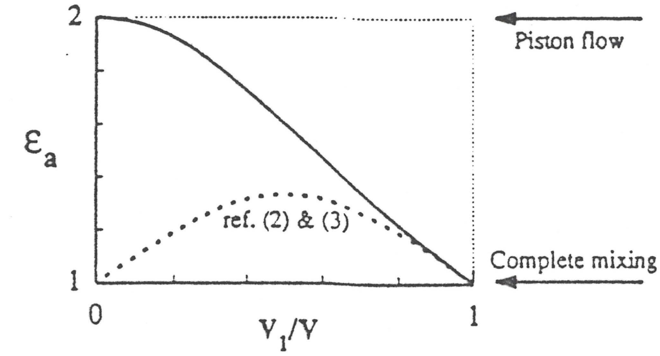
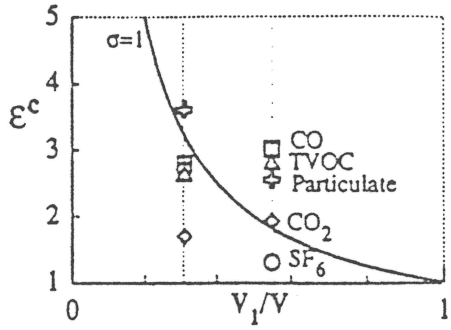


Figure 4. Predicted air change effectiveness.



(a) Comparison with measurements at $V_1/V = 0.31$ & 0.55, for $\sigma=1$.

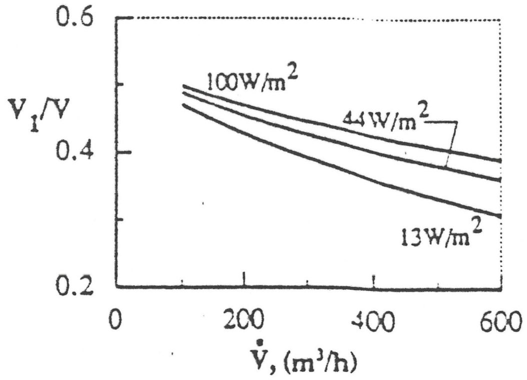
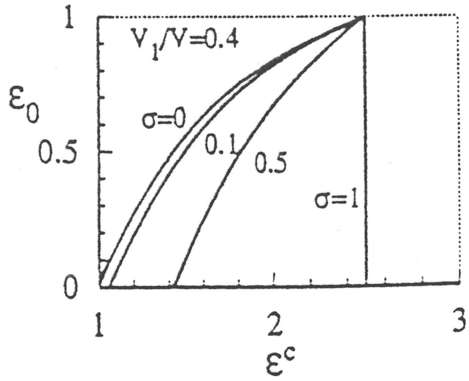


Figure 6. Predicted values of V_1/V as a function of the supply flow rate.



(b) Filter efficiency required for a specified σ and ε^c.

Figure 5. Predicted contaminant removal effectiveness.