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Numerical assessment of thermal comfort and air quality in an office with displacement ventilation

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SYNOPSIS

Computational fluid dynamics has been used for assessing the thermal comfort and air quality in an office ventilated with a displacement system for a range of supply air conditions. Thermal comfort is predicted by incorporating Fanger's comfort equations in the airflow model. Indoor air quality is assessed according to the predicted contaminant concentration and local mean age of air. The performance of the displacement ventilation system is then evaluated based on the predicted thermal comfort and indoor air quality. It is shown that discomfort in offices with displacement ventilation results more often from unsatisfactory thermal sensation than from draught. It is also shown that optimal supply air conditions of a displacement system depend on the relative position between the air diffuser and occupant. It has been found that increasing the air flow rate improves indoor air quality but may result in local thermal discomfort.

1. INTRODUCTION

The technique of positive displacement has been widely used in Scandinavia in particular for ventilation of industrial buildings. The system has now gradually been employed for office ventilation despite the concern over draught near feet. Numerical predictions [1, 2] show that displacement ventilation generally creates a better thermal environment and average air quality in the occupied zone than traditional mixing systems.

Recently computational fluid dynamics (CFD) has been applied in predicting the thermal environment of displacement-ventilated offices based on the draught risk [3, 4, 5]. In this paper the performance of the displacement ventilation system is assessed according to the predicted thermal comfort and air quality in an office using the CFD technique.

2. METHODOLOGY

Numerical evaluation of the indoor environment is based on the airflow model in conjunction with comfort models.

2.1 Air Flow Model

The airflow model consists of the continuity equation, momentum equation, enthalpy equation, concentration equation and the equation for the age of air together with the k-ε turbulence model equations. For an incompressible steady-state flow the model is represented by the following time-averaged equation:

$$\frac{\partial}{\partial x_i}(\rho U_i \phi) = \frac{\partial}{\partial x_i}\left(\Gamma_\phi \frac{\partial \phi}{\partial x_i}\right) + S_\phi \qquad (1)$$

where ϕ stands for mean velocity component U_i in x_i direction, mean enthalpy H, mean concentration C, local mean age $\bar{\tau}$, turbulent kinetic energy k or dissipation rate of turbulent kinetic energy ε; ρ is the air density; Γ_ϕ is the diffusion coefficient for ϕ; S_ϕ is the source term for ϕ.

A complete description of the model equations and the solution method is given elsewhere [6, 7].

2.2 Thermal Comfort

Assessment of thermal comfort includes overall thermal sensation and draught risk as well as local thermal discomfort from either or both of them.

2.2.1 Thermal sensation

Thermal sensation is evaluated in terms of the predicted mean vote (PMV) and the predicted percentage of dissatisfied (PPD) [8]. These indices are functions of air velocity, air temperature, mean radiant temperature, water vapour pressure, clothing thermal resistance and occupant's metabolic rate.

The air velocity, temperature and water vapour pressure distributions in a room are calculated from the air flow equations. The distribution of mean radiant temperature is attained with the help of a radiation heat exchange model. The procedure to calculate the mean radiant temperature at each grid point is as follows [6]:

i) calculate room surface temperature from the heat balance equations for conduction, convection and radiation;
ii) calculate room surface radiosity based on the room surface temperature and radiation shape factors;
iii) calculate six plane radiant temperatures for each rectangular parallelepiped grid cell;
iv) calculate the mean radiant temperature for the grid cell; it is taken as a weighted mean of the plane radiant temperatures.

2.2.2 Draught risk

Draught risk is assessed according to the draught model [9]. In this model, the percentage of dissatisfied due to draught (PD in %) is associated with air temperature (T in °C), velocity (V in m/s) and turbulence intensity (Tu in %) in the following form:

$$PD = (3.143 + 0.3696\, V\, Tu)\,(34 - T)\,(V - 0.05)^{0.6223} \tag{2}$$

2.2.3 Local thermal discomfort

In displacement-ventilated offices, local thermal discomfort particularly around occupants' legs and feet presents a potential problem for ventilation system designers and building users. In this study, the local thermal discomfort is assessed according to the levels of discomfort from cold thermal sensation or draught which are expressed in terms of the numbers of grid points around the legs and feet where the predicted comfort levels exceed 10% for PPD [10] and 15% for PD [11] and are denoted as N_{PPD10} and N_{PD15} respectively.

2.3 Air Quality

Indoor air quality is dependent on the quality and quantity of supply air and the distribution of fresh air in the space. It is assessed according to the concentration of CO_2 and the local mean age of air predicted from the airflow equations. The local mean age of air is defined as the average time for air to travel from a supply outlet to any point in a room.

3. RESULTS

Numerical predictions of thermal comfort and air quality are carried out for an office room in summer conditions. The office has dimensions of 4.7 m long, 3.65 m wide and 2.5 m ceiling height. It consists of one external wall and five internal walls including the floor and ceiling. The internal walls are assumed adiabatic. The external wall is insulated to a U-value of 0.22 W/m^{2}K. It has a double-glazed window of width 2.95 m and height 1.3 m with a U-value of 2.9 W/m^{2}K and with external shading. It is assumed that the window is closed and the room is ventilated by a displacement system. This is achieved by introducing cool air horizontally from a diffuser installed in the rear or curtain wall. The outdoor air temperature is assumed 30°C and the wind speed 3 m/s normal to the external wall.

The office is occupied by one person, seated by a desk. The simulated occupant generates metabolic heat of 70 W/m^2 of which 30% is considered to be latent heat and produces CO_2 of 4.72 x 10^{-3} l/s. The moisture production rate by the occupant is estimated from the amount of latent heat. The occupant wears clothes equivalent to a clothing level of 0.6 clo (1.0 clo = 0.155 m^{2}K/W).

Simulations were performed for 12 sets of supply air conditions. The conditions for the simulations are presented in Table 1. Table 2 shows the predicted thermal comfort, local thermal discomfort and air quality indices.

Figures 1 and 2 show the distributions of the predicted room environmental parameters, thermal comfort and air quality indices in the office for the first and third cases studied. These two cases represent (i) the supply air diffuser in the real wall and the occupant 1.2 m away from the window and (ii) the diffuser installed under the window for the same location of the occupant. As indicated by the velocity vectors, the supply air spreads over the

Table 1 Conditions of supply air and locations of diffuser and occupant for simulations

Case	Supply air					Location	
	Velocity (m/s)	Flow rate (l/s)	Temp. (°C)	RH (%)	CO_2 (ppm)	Diffuser	Occupant
1	0.2	24	20	71	350	D1	O1
2	0.2	24	20	71	350	D2	O1
3	0.1	12	20	71	350	D2	O1
4	0.1	24	20	71	350	D2	O1
5	0.1	24	22	62	350	D2	O1
6	0.1	12	20	71	350	D2	O1
7	0.2	24	20	71	350	D2	O2
8	0.1	12	18	80	350	D2	O1
9	0.15	18	18	80	350	D2	O1
10	0.2	24	18	80	350	D2	O1
11	0.25	30	18	80	350	D2	O1
12	0.3	36	18	80	350	D2	O1

Notes:

Diffuser location:
D1 — rear wall
D2 — curtain wall

Occupant location:
O1 — 1.2 m from window
O2 — mid room length

Table 2 Predicted thermal comfort and air quality in the occupied zone and local thermal discomfort around occupant's legs and feet

Case	V (m/s)	T (°C)	PMV	PPD (%)	PD (%)	N_{PPD10}*	N_{PD15}*	CO_2 (ppm)	$\overline{\tau}/\overline{\tau}_n$#
1	0.028	24.4	0.05	5.29	3.9	0	0	518.1	0.91
2	0.021	24.5	0.07	5.37	3.6	34	10	522.2	0.90
3	0.019	25.9	0.45	9.44	2.3	0	0	694.0	0.95
4	0.021	24.5	0.07	5.35	3.9	21	5	522.0	0.89
5	0.019	25.2	0.25	6.51	2.9	1	0	520.3	0.90
6	0.032	24.9	0.10	5.34	3.5	2	0	694.0	0.96
7	0.024	24.4	0.05	5.27	3.8	0	0	521.4	0.92
8	0.021	25.4	0.30	7.11	2.8	4	0	692.4	0.94
9	0.021	24.5	0.04	5.36	3.2	36	7	581.4	0.92
10	0.021	24.0	-0.10	5.62	3.8	59	17	523.9	0.87
11	0.022	23.5	-0.22	6.46	4.6	83	27	489.4	0.86
12	0.023	23.2	-0.30	7.48	5.7	104	42	466.0	0.85

* Total number of grid points around the legs and feet = 144.
\# $\overline{\tau}$ is the local mean age of air and $\overline{\tau}_n$ is the nominal time constant = net room volume divided by air flow rate.

floor and after reaching the occupant air then rises due to thermal buoyancy. The vertical temperature stratification can be observed in Figures 1b and 2b but the stratification in the occupied zone is not excessive (<2°C) because of small internal heat production. The variation of water vapour pressure with space is also small except for the area around the moisture generation source where the vapour pressure is high. The mean radiant temperature is higher near the 'hot' window than that in the area about one meter away from the window. For both cases, the area near the window is warm whereas the area along the supply air stream is cool (seen from the PMV contours in Figures 1e and 2e); the potential draught risk also exists along the air jet (Figures 1f and 2f). The CO_2 concentration is generally between 400 ppm and 800 ppm and relatively high near the source of generation. The local mean age of air is low near the floor and increases with the height as air flows upwards.

4. DISCUSSION

As seen from Table 2, in terms of predicted comfort indices, both the overall thermal sensation and draught risk are acceptable (PPD < 10%; PD < 15%) under all the supply air conditions investigated. However, local thermal discomfort is also predicted for most of the cases. Besides, the predicted local discomfort often arises simultaneously as a result of cold thermal sensation and draught and the majority of this is caused by dissatisfaction with thermal sensation rather than draught as commonly reported at low temperatures. It is also seen that optimum supply air conditions vary with the relative position between the air diffuser and occupant; Cases 1, 3 and 7 give the least thermal discomfort for three different combinations of air diffuser and occupant locations.

When air is supplied from the rear wall near the floor at a velocity of 0.2 m/s and a temperature of 20°C and the occupant is over three meters away from the air diffuser (Case 1 and Figure 1), there is no risk of local thermal discomfort for the occupant either due to cold thermal sensation or due to draught. This is the result of decreasing momentum and increasing temperature as the supply air diffuses along the floor such that the fresh air reaches a thermally acceptable level near the occupant. As the distance between the diffuser and occupant is reduced, the warmth and movement of fresh air experienced by the occupant deviate from the optimum conditions. Hence, when the diffuser is moved to the curtain wall and the air is supplied at 0.2 m/s and 20°C (Case 2), there exists thermal discomfort around the legs and feet due to draught as well as cold thermal sensation.

For this distance between the diffuser and occupant (i.e. 1.2 m), when the supply air velocity is reduced to 0.1 m/s (Case 3 and Figure 2), the thermal discomfort around the legs and feet disappears but there is some discomfort due to warm thermal sensation at head level because of the reduced air flow rate and the thermal stratification. When the air flow rate is doubled by increasing the diffuser opening area (Case 4), the warm thermal discomfort at the head level is avoided but the cold thermal discomfort around the legs and feet reappears. When the supply air temperature is increased to 22°C (Case 5), the cold thermal discomfort at foot level is almost eliminated but again the head will experience slightly warm feeling. It is clear that at this distance between the air diffuser and occupant a very delicate adjustment of supply air conditions is required to achieve a satisfactory indoor thermal environment and that even so this may not be a good solution to maintain the comfort level because of unavoidable disturbance to the surroundings in practice. One way to alleviate the local thermal discomfort problem is to make use of a chilled ceiling system. The chilled ceiling can be achieved, for example, by installing cooling panels onto the ceiling [12].

Figure 3 shows the predicted indoor thermal environment for the same supply air conditions as Case 3 but with a chilled ceiling system covering 50% ceiling area (Case 6). In this simulation, it is assumed that the area with the chilled ceiling is at a temperature of 20°C while the rest of the ceiling is adiabatic. It is seen that warm discomfort at head level is avoided although slight cool discomfort at foot level occurs. The PPD level around feet slightly exceeds 10% at two out of 144 grid points, 10.8% at one point and 13.9% at another point. This improvement of thermal comfort, compared with Case 3, results from reduced air temperature and mean radiant temperature at head level. The air temperature and mean radiant temperature in the occupied zone are reduced from 25.9°C and 25.9°C without the chilled ceiling to 24.9°C and 24.4°C with the chilled ceiling, respectively. The mean radiant temperature decreases with the distance from the floor whereas air temperature increases; one stratification offsets the other. As a result, the distribution of the comfort indices in the space is fairly uniform apart from the areas near the air diffuser and hot window. Hence the occupant will not have the feeling of warm head.

For the same location of the diffuser, the discomfort can also be relieved or eliminated if the occupant moves away from the window. For example, when the occupant is supposed to sit at the mid room length, the local thermal discomfort is avoided (Case 7 and Figure 4) and the occupant will feel as comfortable as in Case 1, even though the air is supplied at 0.2 m/s and 20°C.

The predicted CO_2 concentration and local mean age of air in the occupied zone decrease with the increase of air flow rate as seen from Cases 8 through 12. For example, when the air flow rate increases from 12 l/s (Case 8) to 24 l/s (Case 10), the CO_2 concentration is reduced from 692.4 ppm to 523.9 ppm and the normalised age of air from 0.94 to 0.87. Hence, indoor air quality can be improved by increasing the air flow rate for displacement ventilation. This is not always true for conventional ventilation systems where short circuiting of supply air could sometimes occur so that contaminants could be trapped in the breathing zone. However, increasing the amount of supply air requires more energy use and for displacement ventilation increases the possibility of local thermal discomfort. Therefore, a proper balance is needed between the requirement for good indoor air quality and that for acceptable thermal comfort when setting supply air conditions.

5. CONCLUSIONS

This numerical investigation shows that in displacement ventilation optimal supply air conditions vary with the distance between the air diffuser and occupant other than the cooling load and load distribution. The distance between the diffuser and occupant should preferably be greater than two meters for maintaining a comfortable indoor thermal environment.

A better indoor environment is achievable through fine tuning of the conditions of displacement ventilation in combination with a chilled ceiling system than is provided by conventional ventilation systems.

When designing a displacement ventilation system and determining supply air conditions the need for good air quality and the need for acceptable thermal comfort should be carefully considered in order to minimise the energy requirement for ventilation.

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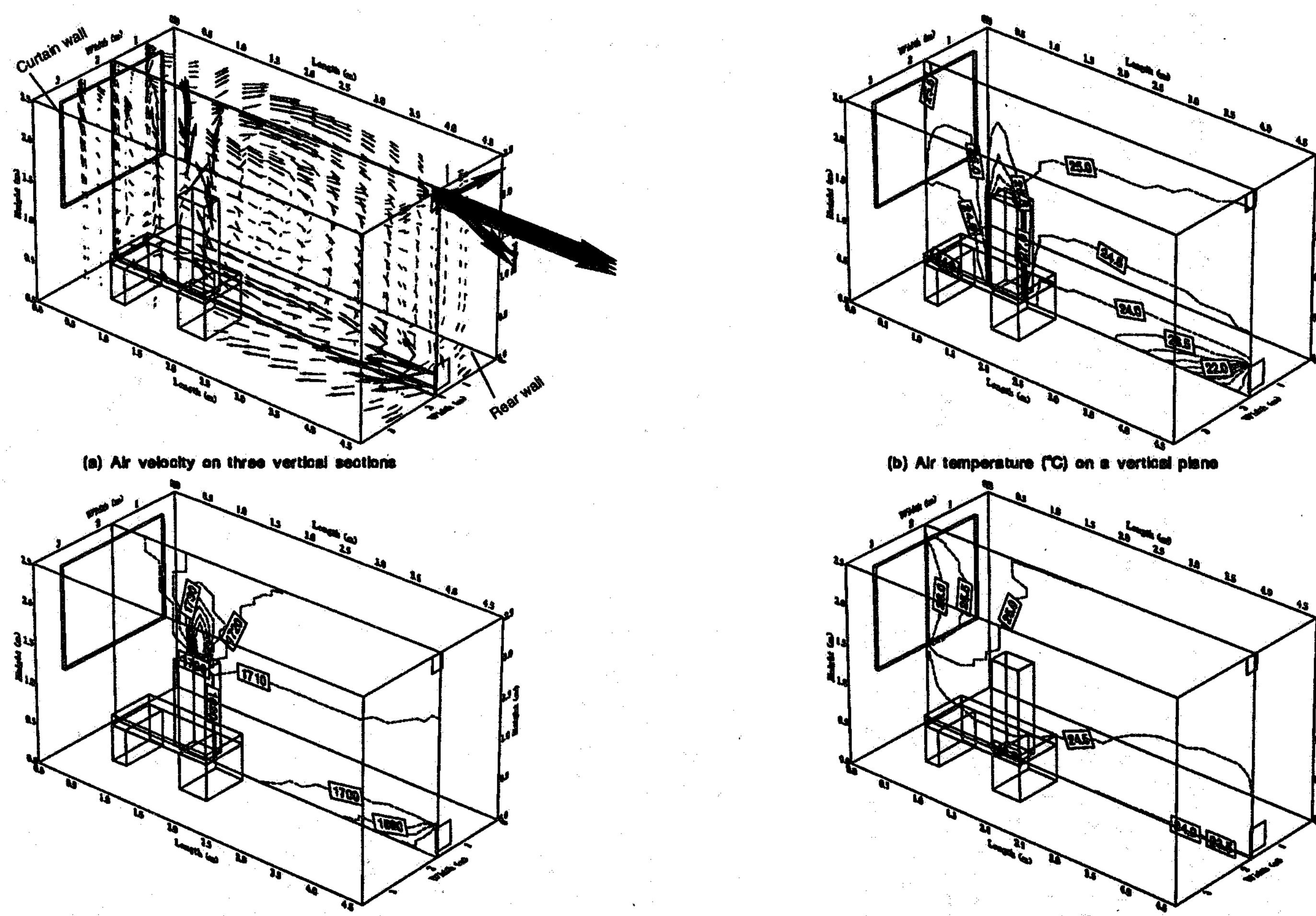


(a) Air velocity on three vertical sections

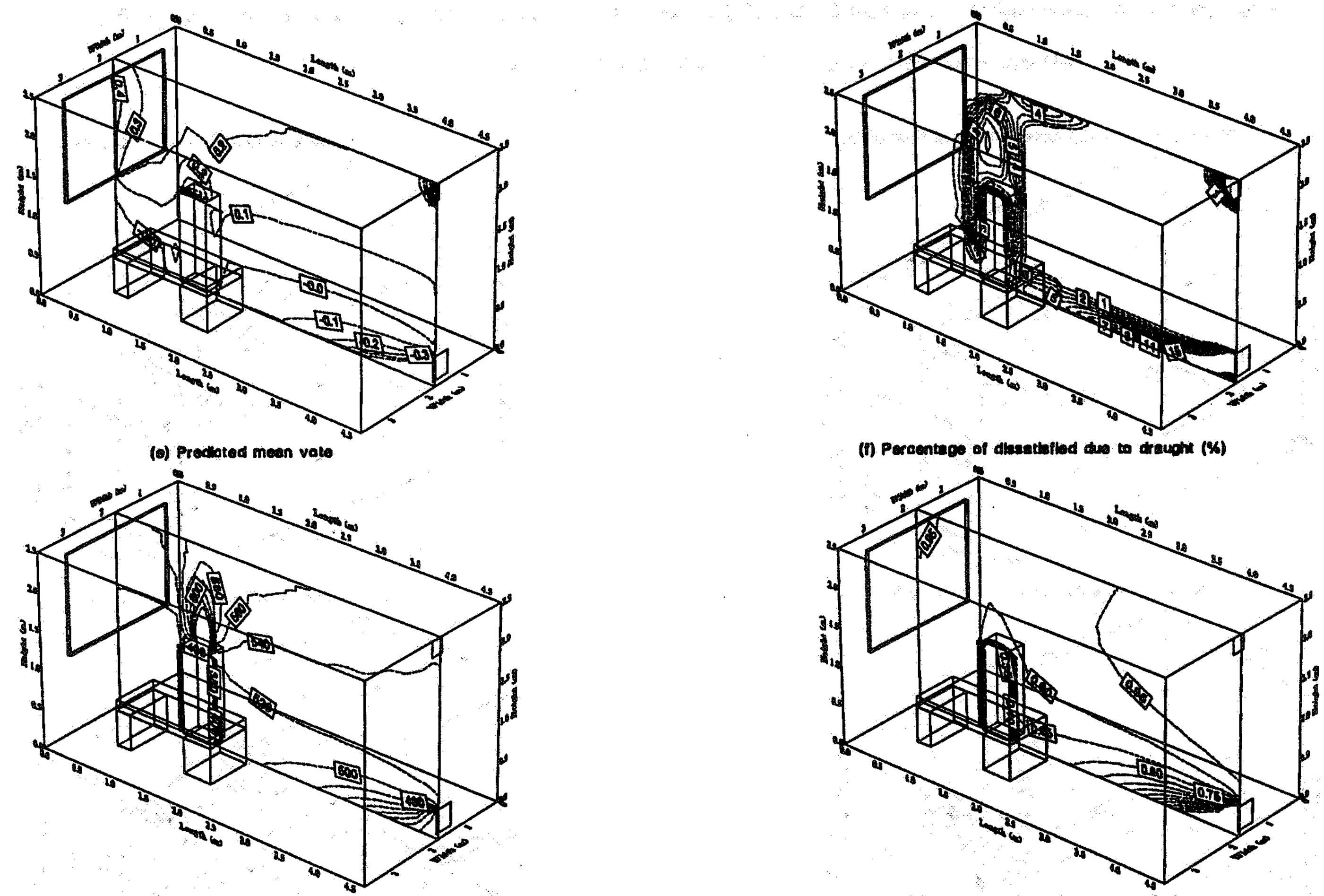
(b) Air temperature (°C) on a vertical plane

(c) Vapour pressure (Pa) on a vertical plane

(d) Mean radiant temperature (°C) on a vertical plane

Supply air: V = 0.2 m/s; T = 20.0°C; RH = 71%; CO_2 = 350 ppm

Fig. 1 Predicted environmental parameters, thermal comfort and air quality in the displacement-ventilated office



(e) Predicted mean vote

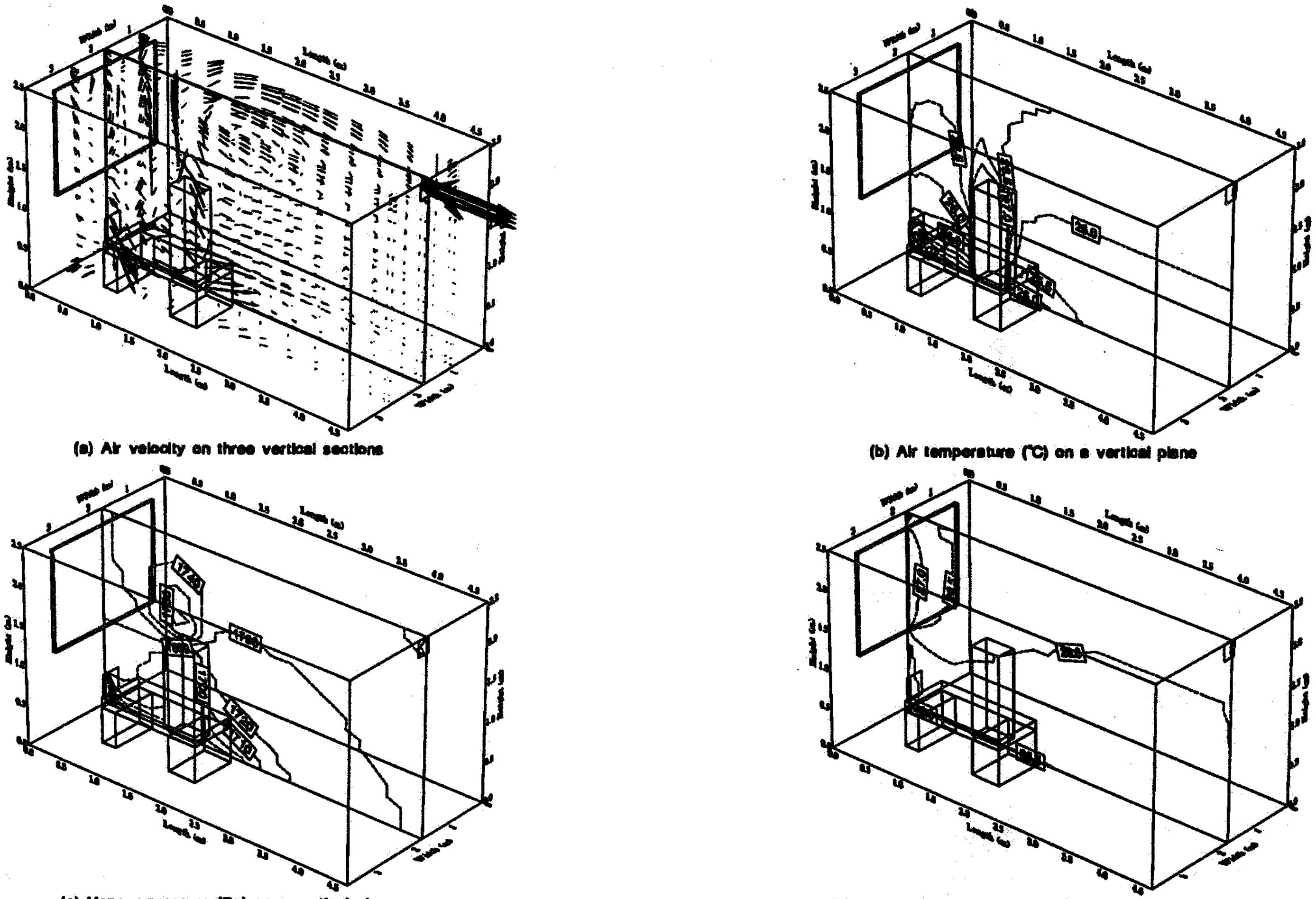
(f) Percentage of dissatisfied due to draught (%)

(g) CO2 concentration (ppm)

(h) Normalised local mean age of air

Supply air: V = 0.2 m/s; T = 20.0°C; RH = 71%; CO2 = 350 ppm

Fig. 1 Continued



(a) Air velocity on three vertical sections

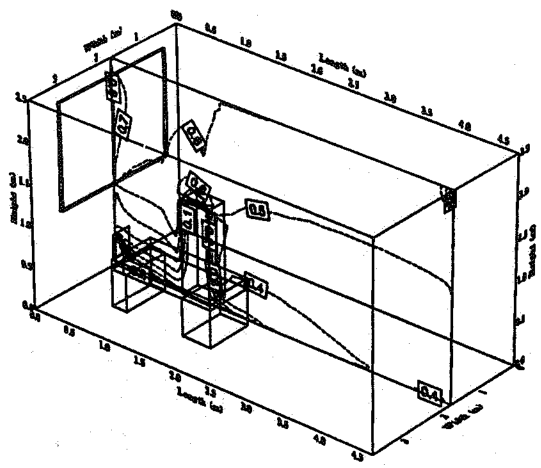
(b) Air temperature (°C) on a vertical plane

(c) Vapour pressure (Pa) on a vertical plane

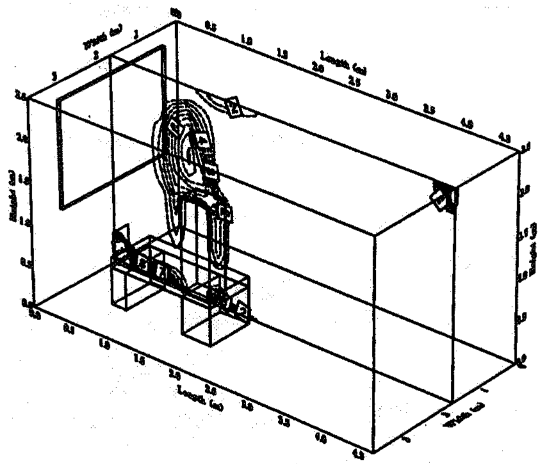
(d) Mean radiant temperature (°C) on a vertical plane

Supply air: V = 0.1 m/s; T = 20.0°C; RH = 71%; CO_2 = 350 ppm

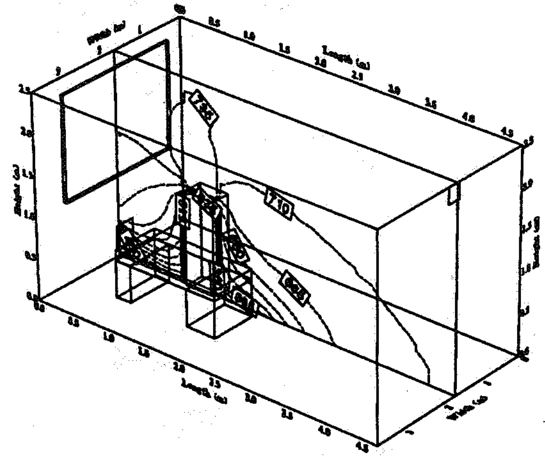
Fig. 2 Predicted environmental parameters, thermal comfort and air quality in the displacement-ventilated office



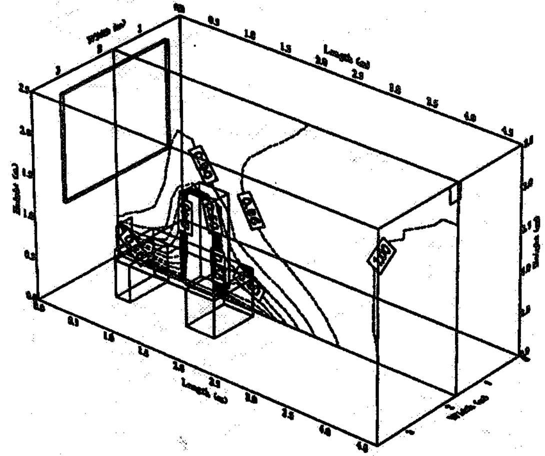
(e) Predicted mean vote



(f) Percentage of dissatisfied due to draught (%)



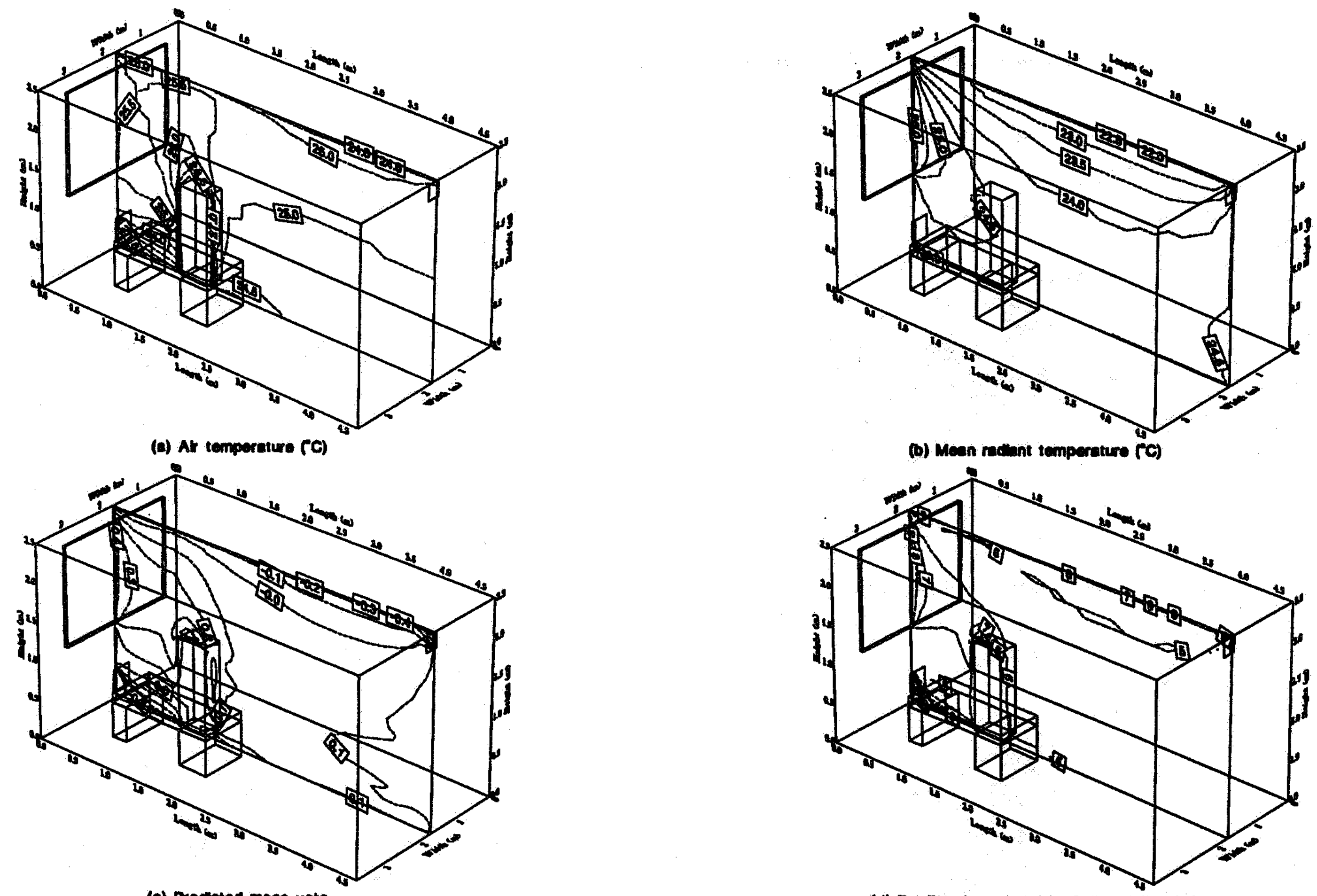
(g) CO2 concentration (ppm)



(h) Normalised local mean age of air

Supply air: V = 0.1 m/s; T = 20.0°C; RH = 71%; CO_2 = 350 ppm

Fig. 2 Continued



(a) Air temperature (°C)

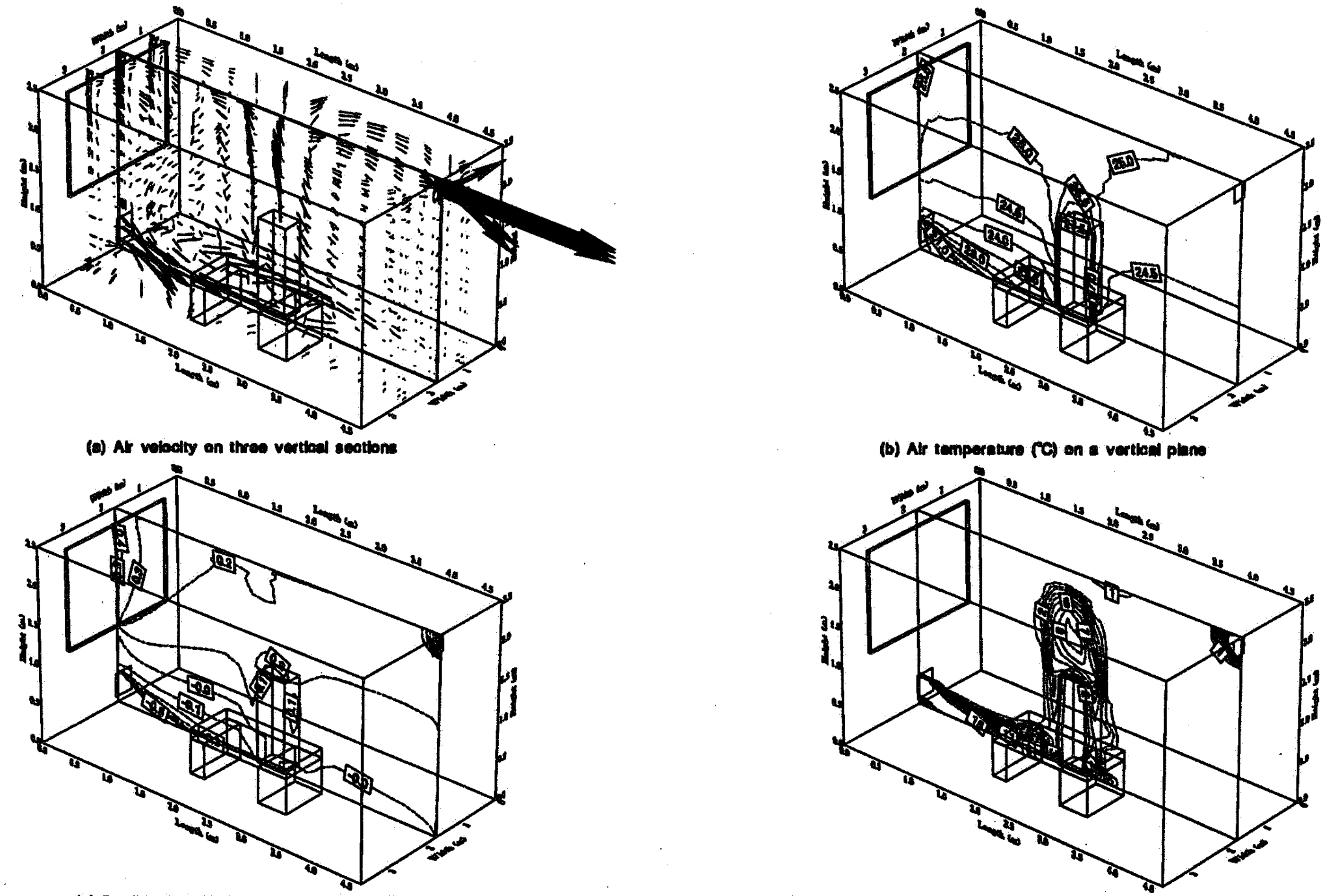
(b) Mean radiant temperature (°C)

(c) Predicted mean vote

(d) Predicted percentage of dissatisfied (%)

Supply air: V = 0.1 m/s; T = 20.0°C; RH = 71%

Fig. 3 Predicted thermal comfort on a vertical plane in the displacement-ventilated office with 50% chilled ceiling



(a) Air velocity on three vertical sections

(b) Air temperature (°C) on a vertical plane

(c) Predicted mean vote on a vertical plane

(d) Percentage of dissatisfied due to draught (%) on a vertical plane

Supply air: V = 0.2 m/s; T = 20.0°C; RH = 71%

Fig. 4 Predicted air flow and thermal comfort in the displacement-ventilated office