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AIRFLOW AND THERMAL COMFORT IN NATURALLY VENTILATED CLASSROOMS

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SYNOPSIS

The airflow pattern and thermal comfort in a naturally ventilated classroom were predicted using CFD techniques. The CFD model for turbulent flow consists of equations for the conservation of mass, momentum and thermal energy and the equations for the $k-\epsilon$ turbulence model, taking account of the effects of buoyancy and obstacles in the room. The thermal comfort was assessed according to the predicted mean vote (PMV) and predicted percentage of dissatisfied (PPD). The mean radiant temperature for each grid point was calculated from the radiant heat balance for the classroom and the shape factor of the point. The predicted velocity and temperature distributions were compared with experimental results and reasonable agreement was achieved. It was found that occupants in the room have great influence on the airflow pattern and temperature distribution.

1. INTRODUCTION

Modern technology and high living standards make it possible for people to have better comfort whether at home or in offices. Indoor thermal comfort is influenced by air velocity, air temperature, mean radiant temperature and air humidity in addition to the personal parameters such as metabolic rate and clothing. Natural ventilation, in combination with heating when required, is still the main means to achieve adequate thermal comfort for most buildings in the U.K. The airflow in naturally ventilated buildings depends much on the arrangement of doors and windows for given external conditions. In order to achieve a comfortable thermal environment there is a need for defining details of the airflow patterns especially at head and foot levels when designing an effective ventilation system[1]. Difficulties may be encountered however in field tests of naturally ventilated systems because of the uncertainty of various factors that affect their performance. Mathematical modelling as a potential alternative can be used for predicting the performance of these systems.

Computational fluid dynamics (CFD) has for many years been applied in the simulation of room air movement in airconditioned spaces[2]. The simulation of airflow in spaces or cavities dominated by buoyancy has been carried out by a number of investigators[3,4]. These and other investigations were largely performed using reduced-scale models or numerical experiments. In contrast, there is not enough validation of CFD programs using field measurements, especially in naturally ventilated buildings.

Moreover, in most of the available CFD programs the radiant heat transfer is simplified or not accounted for when dealing with thermal comfort. Fanger[5] demonstrated how to evaluate thermal comfort by taking into account the radiant heat transfer process between the person and the internal surfaces of a room. Kaizuka and Iwamoto[6] calculated the distribution of thermal comfort index caused by radiation interaction in a heated room under the assumptions of uniform air temperature and given air velocity. In this paper the air movement and thermal comfort in a naturally ventilated classroom are predicted using the CFD program ARIA-R. The predictions are compared with measurements in an occupied and unoccupied classroom.

2. MODEL EQUATIONS AND SOLUTION

2.1 Flow equations

The airflow model is based on the continuity equation, Navier-Stokes equation and thermal energy equation together with the k-ϵ turbulence model equations. For a steady incompressible flow the time-average equations are represented by

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

where S_ϕ = source terms of dependent variable ϕ
U_i = mean velocity component in x_i direction
Γ_ϕ = diffusion coefficient for dependent variable ϕ
ρ = fluid density

The flow equations are solved for the 3-D cartesian system using the SIMPLE algorithm[7] with the boundary conditions described elsewhere[8].

2.2 Thermal comfort equations

Thermal comfort is evaluated in terms of predicted mean vote (PMV) and predicted percentage of dissatisfied (PPD) proposed by Fanger[5]. These comfort indices take account of the combined effect of environmental conditions such as air velocity, air temperature, mean radiant temperature and partial water vapour pressure of air and occupant conditions such as clothing and activity levels. In the present evaluation the air velocity and temperature are given by the flow equations. The mean radiant temperature, T_{mrt}, is a function of several parameters such as the temperature and thermal properties of room surfaces and the shape factors between the surfaces. Other parameters, e.g. vapour pressure of air, are given a single value for the whole field.

In the calculation of mean radiant temperature at a grid point in the field, the grid cell is considered as a rectangular parallelepiped. The plane radiant temperature, T_{prt} (K), at each face of the cell is obtained from[5]:

$$T_{prt}^4 = \frac{1}{\sigma} \sum_{i}^{N} F_{pi} \left[\epsilon_i \sigma T_i^4 + (1 - \epsilon_i) \sum_{j}^{N} F_{ij} J_j\right] \tag{2}$$

where σ is the Stefan-Boltzmann constant ($\sigma = 5.669 \times 10^{-8}$ W/m^2K^4); N is the number of room surfaces; F_{pi} is the shape factor for radiation from face p of the grid cell to the visible room surface i (i = 1 to N); F_{ij} is the shape factor for room surface i and surface j; T_i is the absolute temperature of room surface i; ϵ_i is the emissivity of surface i and J_j is the radiosity for surface j, which is defined as the total radiation that leaves a surface per unit time and per unit area (W/m^2).

The radiosity J_i for surface i is calculated using the following equation, for a specified surface temperature[5]:

$$J_i = \epsilon_i \sigma T_i^4 + (1 - \epsilon_i) \sum_{j}^{N} F_{ij} J_j \tag{3}$$

The mean radiant temperature, T_{mrt}, for the grid cell is then taken as the weighted average of the six plane radiant temperatures for each face of the rectangular parallelepiped based on the face areas.

The predicted mean vote, PMV, and predicted percentage of dissatisfied, PPD (%), are given by[5,9]

$$\begin{aligned} PMV = {} & [0.303\exp(-0.036M) + 0.028] \{(M - W) - 3.05 \times 10^{-3} \\ & \times [5733 - 6.99(M - W) - p_a] - 0.42 [(M - W) - 58.15] \\ & - 1.7 \times 10^{-5}M (5867 - p_a) - 0.0014 M (34 - T_a) \\ & - 3.96 \times 10^{-8} f_{cl} [(T_{cl} + 273)^4 - (T_{mrt} + 273)^4] \\ & - f_{cl} h_c (T_{cl} - T_a)\} \end{aligned} \tag{4}$$

and

$$PPD = 100 - 95 \exp(-(0.03353 PMV^4 + 0.2179 PMV^2)) \tag{5}$$

where

$$\begin{aligned} T_{cl} = {} & 35.7 - 0.028(M - W) - I_{cl} \{3.96 \times 10^{-8} f_{cl} \\ & \times [(T_{cl} + 273)^4 - (T_{mrt} + 273)^4] + f_{cl} h_c (T_{cl} - T_a)\} \end{aligned} \tag{6}$$

$$h_c = 2.38(T_{cl} - T_a)^{0.25} \quad \text{for } 2.38(T_{cl} - T_a)^{0.25} > 12.1\sqrt{V_r} \tag{7}$$

$$h_c = 12.1\sqrt{V_r} \quad \text{for } 2.38(T_{cl} - T_a)^{0.25} < 12.1\sqrt{V_r} \tag{8}$$

$$f_{cl} = 1.00 + 1.290 I_{cl} \quad \text{for } I_{cl} \leq 0.078 \ m^2K/W \tag{9}$$

$$f_{cl} = 1.05 + 0.645 I_{cl} \quad \text{for } I_{cl} > 0.078 \ m^2K/W \tag{10}$$

M is the metabolic rate; W is the external work (taken zero here); p_a is the partial water vapour pressure of air; T_a is the mean air temperature; f_{cl} is the ratio of man's surface area while clothed to the area while nude; T_{cl} is the surface temperature of clothing; h_c is the convective heat transfer coefficient; I_{cl} is the thermal resistance of clothing and V_r is the relative air velocity.

3. RESULTS AND DISCUSSION

The predictions of airflow and thermal comfort were carried out for a naturally ventilated classroom for the summer season. Figure 1 shows a schematic diagram of the classroom at the University of Reading. The room has the dimensions 10.9 x 11 x 3.05 m (length x width x height), with east-west direction along the length. The west side of the room is linked to a main entrance door via a corridor. Part of the south and north faces of the room are glazed, each with six openable windows. Various combinations and positions of window openings have been used during the experiments to investigate the variation in the ventilation characteristics of the room. Air velocities and temperatures were measured at one level, 0.9 m above the floor (head level when seated), with omnidirectional hot wire anemometers. Air flow rates were determined using the concentration decay method with iso-butane as the tracer gas. Besides, indoor air quality was assessed on the basis of the carbon dioxide levels during the occupancy periods and a subjective survey of the occupants was undertaken by means of vote on their sensations of thermal environment and impressions of odour. Tests were conducted both with and without occupancy[10-12].

The following assumptions were made in the CFD prediction:

1. **Heat sources.** The solar heat gain through the glazed area facing south and the heat generation due to equipment in the room were considered to be distributed uniformly over the floor. Heat production by artificial lights near the ceiling was taken as a uniformly-distributed heat source over the ceiling.

2. **Obstacles.** Tables and columns in the room were simulated as obstacles. Occupants were treated as obstacles with heat production (each person 100 W).

3. **Supply air.** The air velocity at the supply openings was calculated from the measured air flow rate and was assumed normal and uniformly distributed across the openings.

A total of ten computer simulations, of which five with occupancy, were performed. The predicted and measured average air velocities and temperatures at the level 0.9 m above the floor are shown in the table below.

Predicted and measured average air velocities and air temperatures in the classroom 0.9 m above the floor

No.	Air change rate (1/hr)	Velocity (m/s)		Temperature (°C)	
		Predicted	Measured	Predicted	Measured
1	3.62	0.090	0.112	23.85	23.03*
2	5.64	0.135	0.160	29.41	28.39
3	5.88	0.124	0.110	26.13	24.80
4	1.68	0.062	0.096	25.23	23.21
5	5.76	0.122	0.097	24.23	23.03
Mean of 1 to 5		**0.107**	**0.117**	**25.77**	**24.49**
6	7.73	0.113	0.092	21.89	21.98*
7	10.20	0.123	0.110	23.69	24.43
8	14.34	0.161	0.107	27.86	27.52
9	2.98	0.079	0.094	23.07	20.88
10	8.76	0.133	0.114	25.29	25.31
Mean of 6 to 10		**0.122**	**0.103**	**24.36**	**24.02**
Mean of 1 to 10		**0.115**	**0.110**	**25.07**	**24.26**

Notes: number 1 - 5 with occupancy and number 6 - 10 without occupancy; the combination and position of window openings varied from one to another.
* discussed in the text.

It can be seen from the table that the simulation gives fairly good predictions of the average values of the velocity and temperature. The mean difference between the predictions and measurements is less than 5% for velocity and 1 K in temperature.

Figures 2 and 3 show the predicted velocity and temperature distributions for the two cases to be discussed hereafter, one with occupancy (case 1), and the other without (case 6). A comparison of predicted velocities and temperatures with the measured values for the corresponding two cases are shown in Figures 4 and 5 whereas the predicted comfort indices are shown in Figures 6 and 7. The air flow rates for case 1 and case 6 are 3.62 and 7.73 air changes per hour respectively (The corresponding supply velocities are 0.356 and 0.76 m/s). In case 1, six windows were half open. Air flowed into the room through four windows in the north face and flowed out from two windows in the south face. In case 6, there were six windows in fully open position in the south wall. However, the direction of wind flow into or out of these windows was not known and it has been assumed that the wind was flowing into the room via the two windows close to the east wall and out from the other four. It can be seen from Figures 4 and 5 that, in general, the predictions are in reasonable agreement with the measurements except that the predicted values near the supply openings are somewhat higher for velocity and lower for temperature in the air jets. This is mainly attributed to the assumption about the velocity direction at the supply opening. The actual supply velocity might not be perpendicular to the opening area as assumed in the prediction. In fact, during the time of the experiments, the wind directions were north-westerly for case 1 and south-westerly for case 6. Due to the channelling effect of windows open outwards, the incoming air could have been inclined and there might have been vortices formed, which would have interfered with the air jet at the opening.

Figure 4 also indicates that for the case with occupancy the predicted air temperatures vary from the lowest near the supply opening to the highest close to the occupants, whereas the measured values are nearly uniform. This is because of a relatively large difference between the supply air and room bulk temperatures, especially the body temperature, used in the prediction. The measurement was made at some discrete points and therefore could not reflect the full picture of the variations. The variation in the predicted air temperature for the unoccupied room is small (Figure 5), and gives better agreement with measurement in comparison with that for the occupied room. Similarly, the velocity decreases from the supply jet value to a negligible magnitude as the air jet diffuses into the room. Again the variation of predicted velocities is larger than that obtained from measurement. These discrepancies may be attributed to the uncertainties and assumptions in key parameters such as the magnitude and direction of supply air velocity and distribution of heat sources in the space. Nevertheless, in view of the fact that the prediction has been compared with field measurements which are much more complex and, most of all, less controllable than those in, for example, scale or laboratory models, the predictions may be considered satisfactory.

Figures 6 and 7 indicate that the predicted percentage of dissatisfied for the two cases is generally below 10%, well within the acceptable range of 20%[9]. In fact, the average PPDs in the occupied zone (from floor to 1.8 m height) for case 1 and case 6 are about 8% and 7% respectively, both being between neutral and slightly warm. These tests were, however, conducted under a mild environmental temperature (about 20°C). The inference is that the indoor environment may be beyond the acceptable level for thermal comfort when the outdoor temperature is much higher unless it is compensated for by a higher air flow rate. Therefore, in a hot climate some measure may have to be taken to provide sufficient and preferably cool air without causing excessive draught if discomfort in the space is to be avoided. In a cold season, heat needs to be provided to maintain indoor thermal comfort which conforms with normal practice.

To investigate the effect of occupants on the airflow and thermal comfort, an additional prediction was made for the conditions represented by case one but without occupancy. The air flow pattern and temperature distribution for this case are shown in Figure 8. By comparing Figure 2 with Figure 8 it can be seen that the occupants have a great influence on the air flow patterns and temperature distributions within a space. As pointed out above, the presence of an occupant increases the air temperature close to the body and in the wake of the air plume rising about the head due to body heat. The occupant, like other obstacles, influences and diverts air movement. The diversion occurs not only as a result of air separation when flowing over the body but also through the buoyancy force due to the temperature difference between the body and the surrounding air. Exposed skin surfaces are at about 33°C and clothing surfaces can be 24 - 28°C. Consequently, in the absence of the occupants, the velocity in the occupied zone was reduced by 8% and the temperature by 1 K compared to those with occupancy. The occupants naturally affect the distribution of comfort indices as they bring about the change of the indoor environment. However, because of the decrease in the velocity and temperature in the space simultaneously, the overall effect of occupancy on the predicted thermal comfort is small. For example, the average value of the predicted percentage of dissatisfied in the occupied zone decreased from 8% with occupancy to 7% without occupancy. This change, of course, may be quite substantial at certain spots, particularly in and around the area where occupants were originally situated.

4. CONCLUSIONS

The 3-D CFD program ARIA-R, that has been specifically developed to predict the indoor environment, can be used to predict the airflow and thermal comfort in naturally ventilated rooms given knowledge of key parameters such as supply air velocity and temperature. The program can also be used for the prediction of the performance of mechanical ventilation systems.

In a mild climate adequate thermal comfort can be achieved by appropriate arrangement of doors or windows. There may be some difficulties however in maintaining a comfortable environment in a hot climate through natural ventilation alone, depending on the layout of the spaces, the occupancy density and the supply-exit configuration. In this circumstance, cooling may also be required.

In a naturally ventilated room, the airflow patterns and temperature distribution are greatly influenced by the occupants and their distribution in the space. This is also true for mechanical ventilated or airconditioned spaces when the occupancy density is high such as in auditoria[13].

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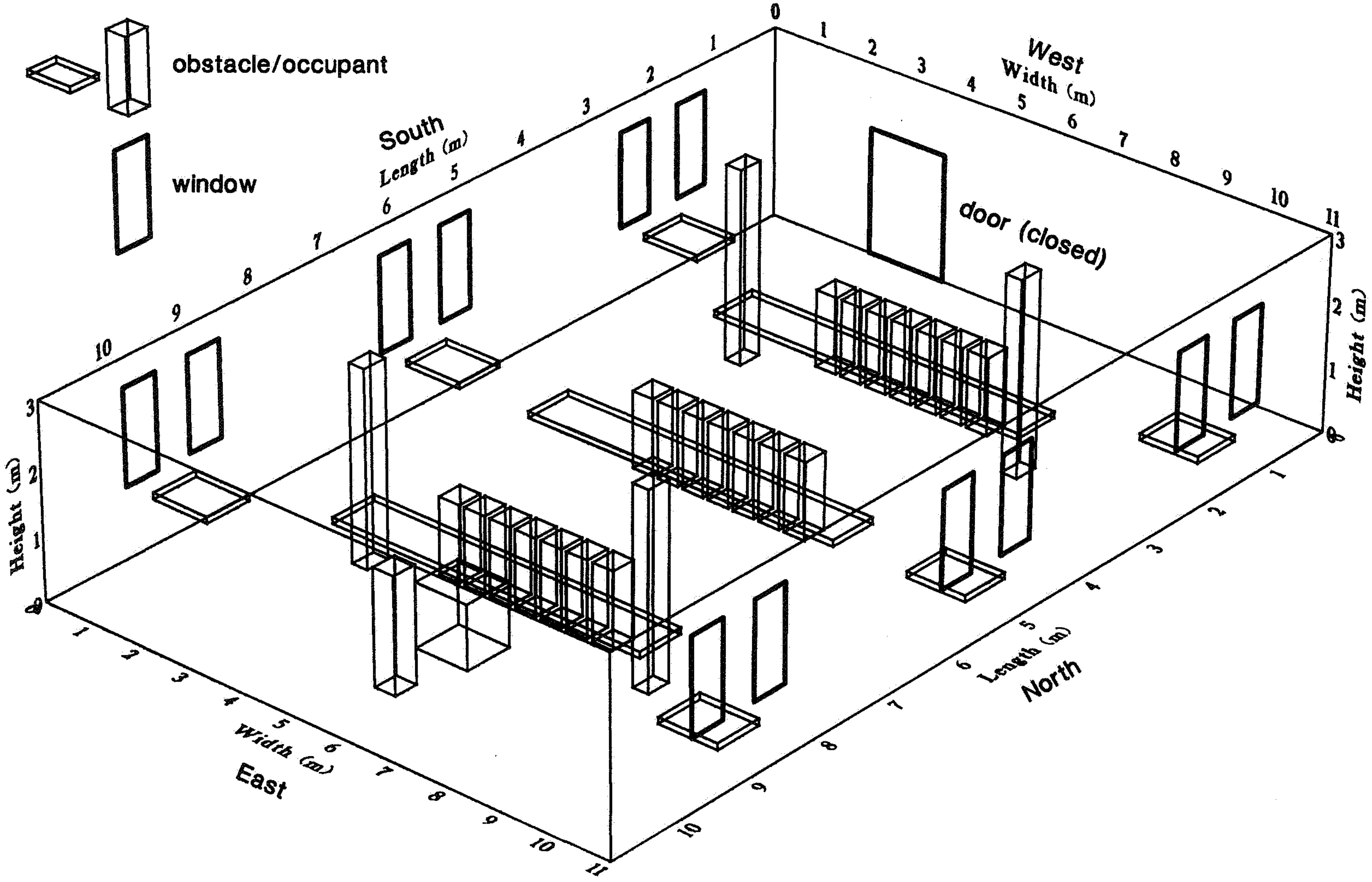
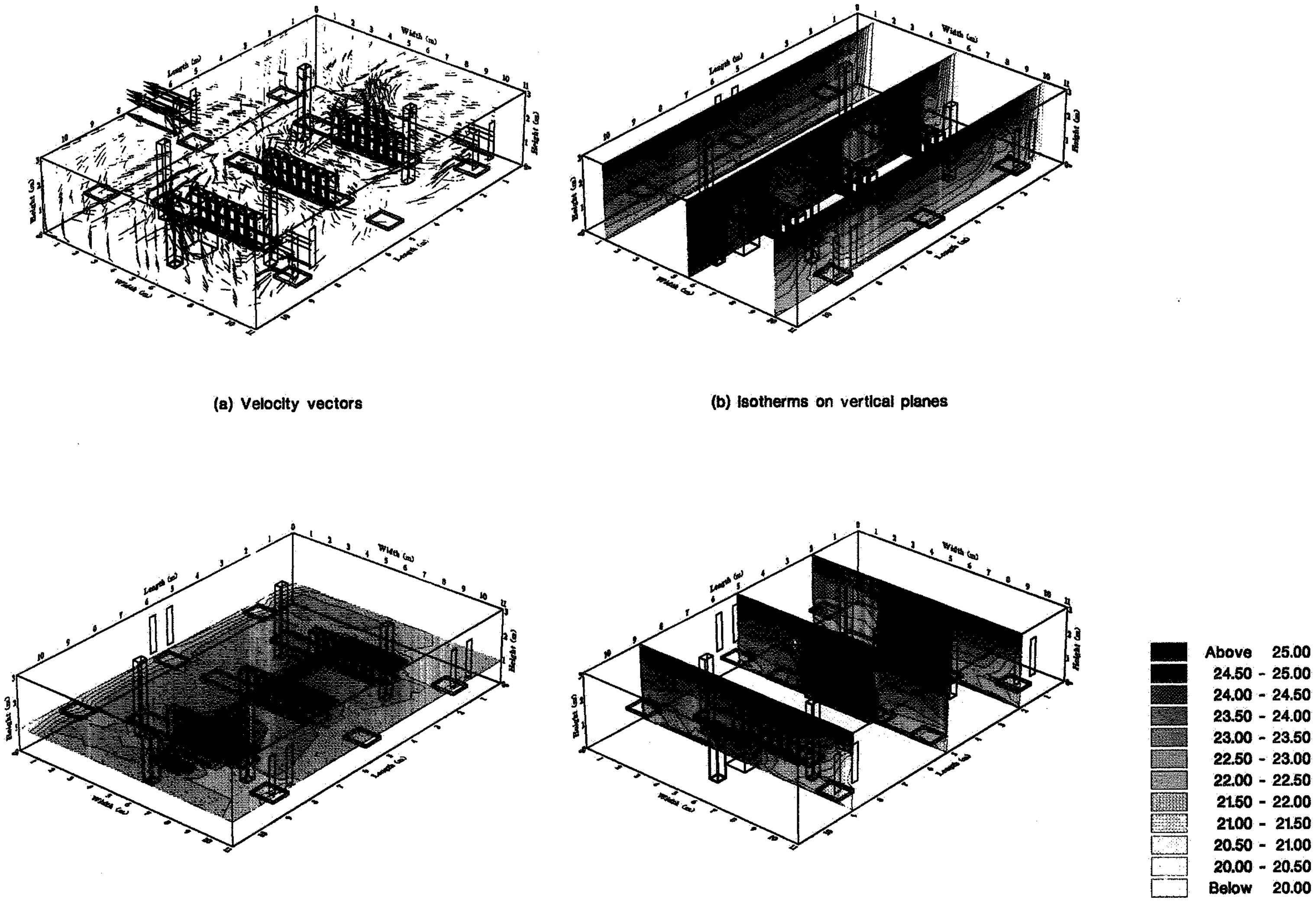


Fig. 1 Schematic diagram of the classroom



(a) Velocity vectors

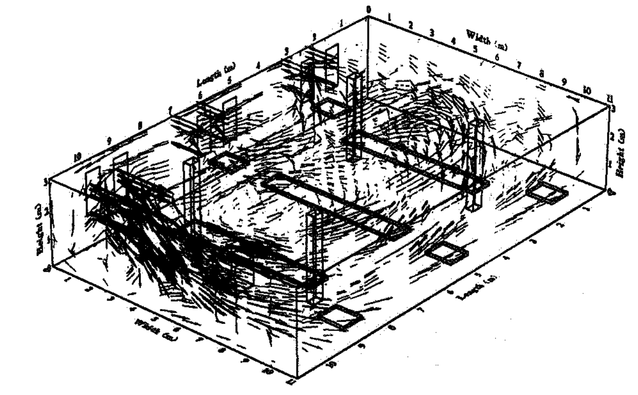
(b) Isotherms on vertical planes

(c) Isotherms on a horizontal plane

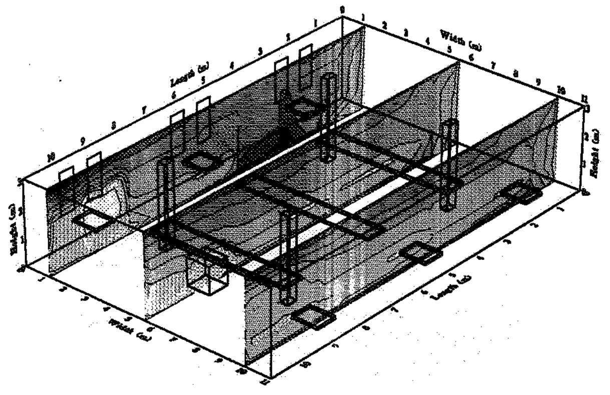
(d) Isotherms on vertical planes

Supply air: U = 0.356 m/s ; T = 19.9 deg.C

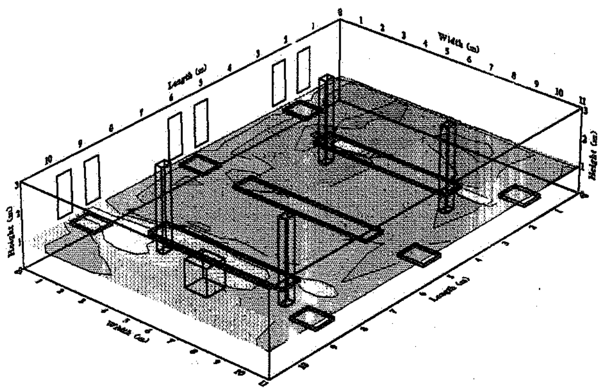
Fig. 2 Predicted velocity vectors & isotherms of air in a classroom with occupancy (case 1)



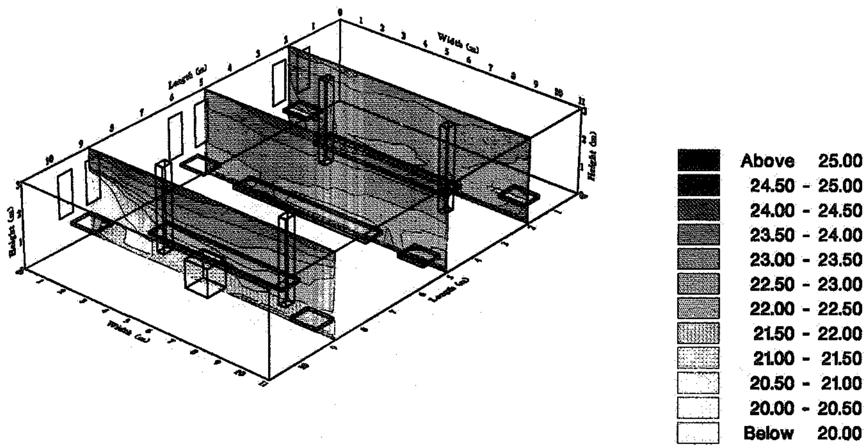
(a) Velocity vectors



(b) Isotherms on vertical planes



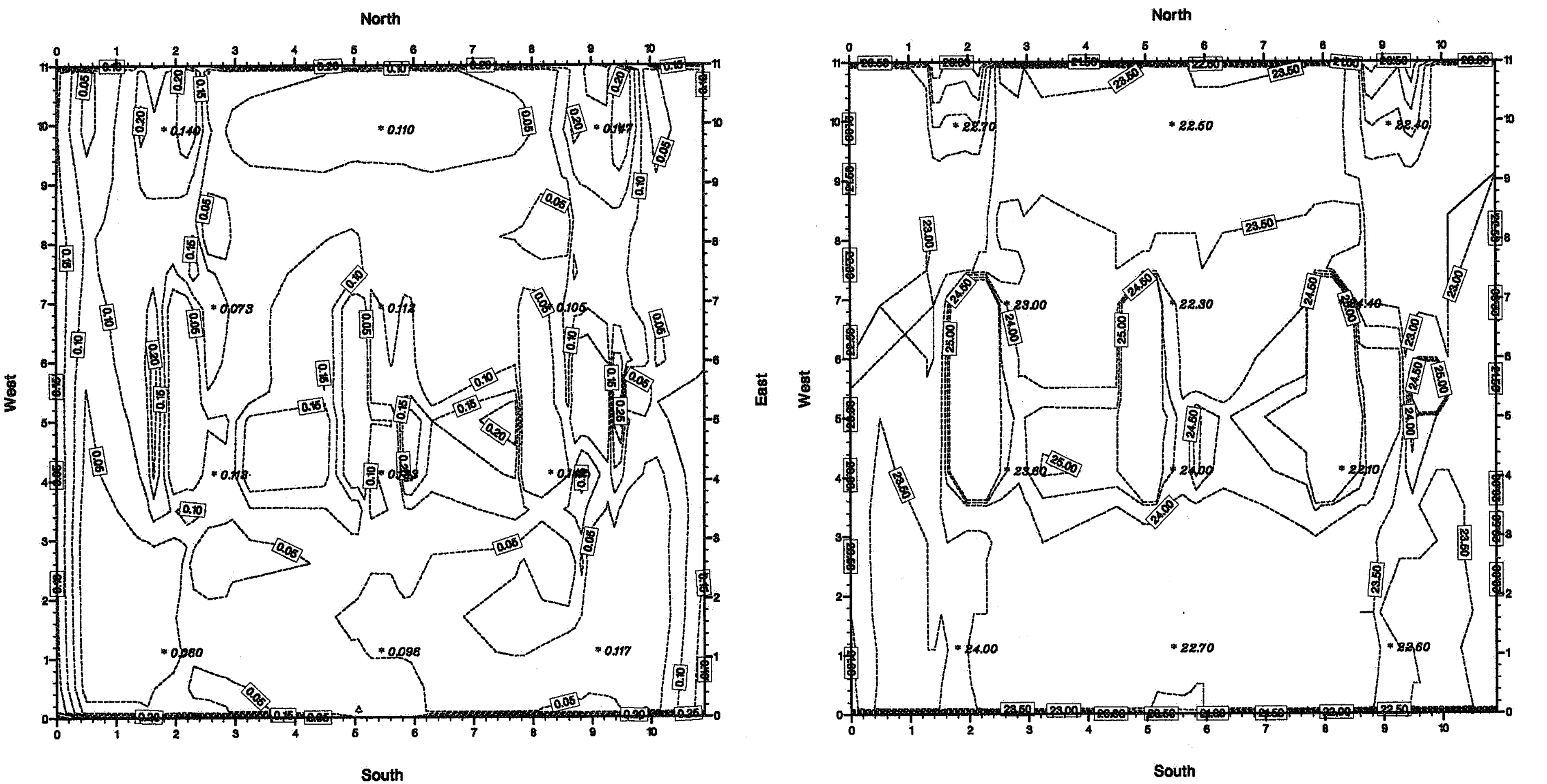
(c) Isotherms on a horizontal plane



(d) Isotherms on vertical planes

Supply air: U = 0.76 m/s ; T = 19.9 deg.C

Fig. 3 Predicted velocity vectors & isotherms of air in a classroom without occupancy (case 6)

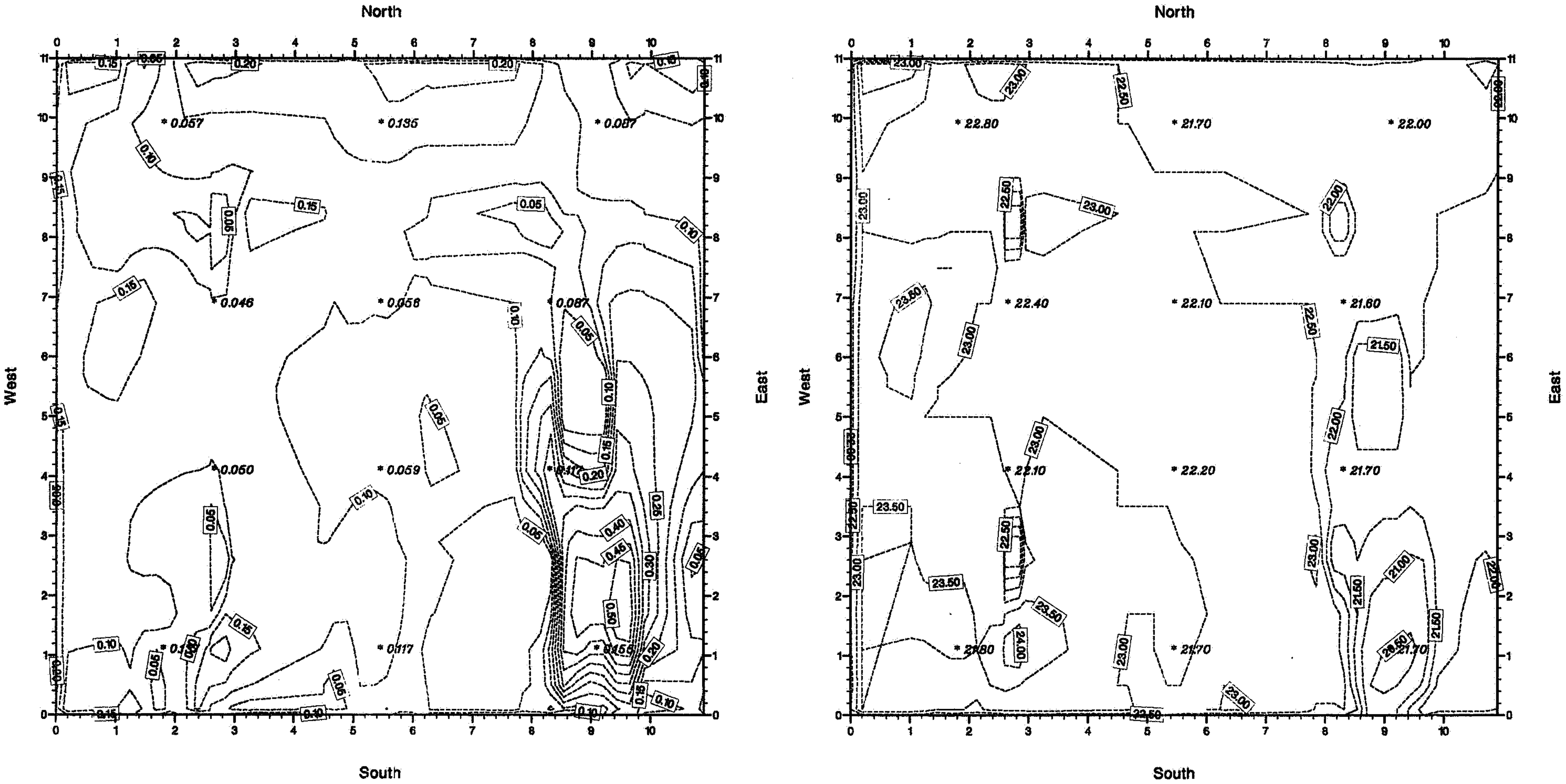


(a) Velocity (m/s)

(b) Temperature (deg.C)

Supply air: U = 0.356 m/s ; T = 19.9 deg.C

Fig. 4 Comparison of predicted isovels and isotherms with measured velocities and temperatures (*) on a horizontal plane (0.9 m above the floor) of a classroom with occupancy (case 1)

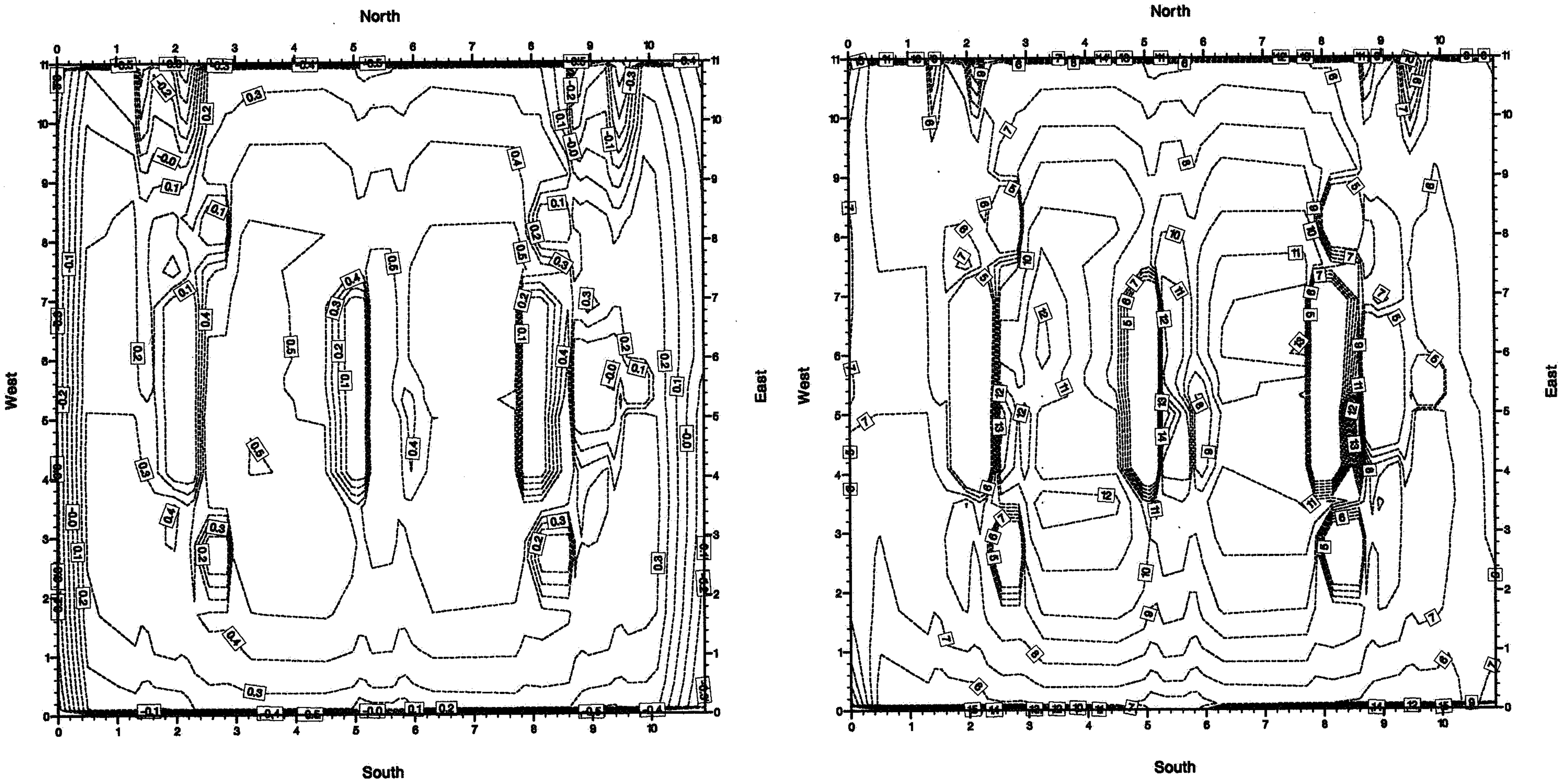


(a) Velocity (m/s)

(b) Temperature (deg.C)

Supply air: U = 0.76 m/s ; T = 19.9 deg.C

Fig. 5 Comparison of predicted isovels and isotherms with measured velocities and temperatures (*) on a horizontal plane (0.9 m above the floor) of a classroom without occupancy (case 6)

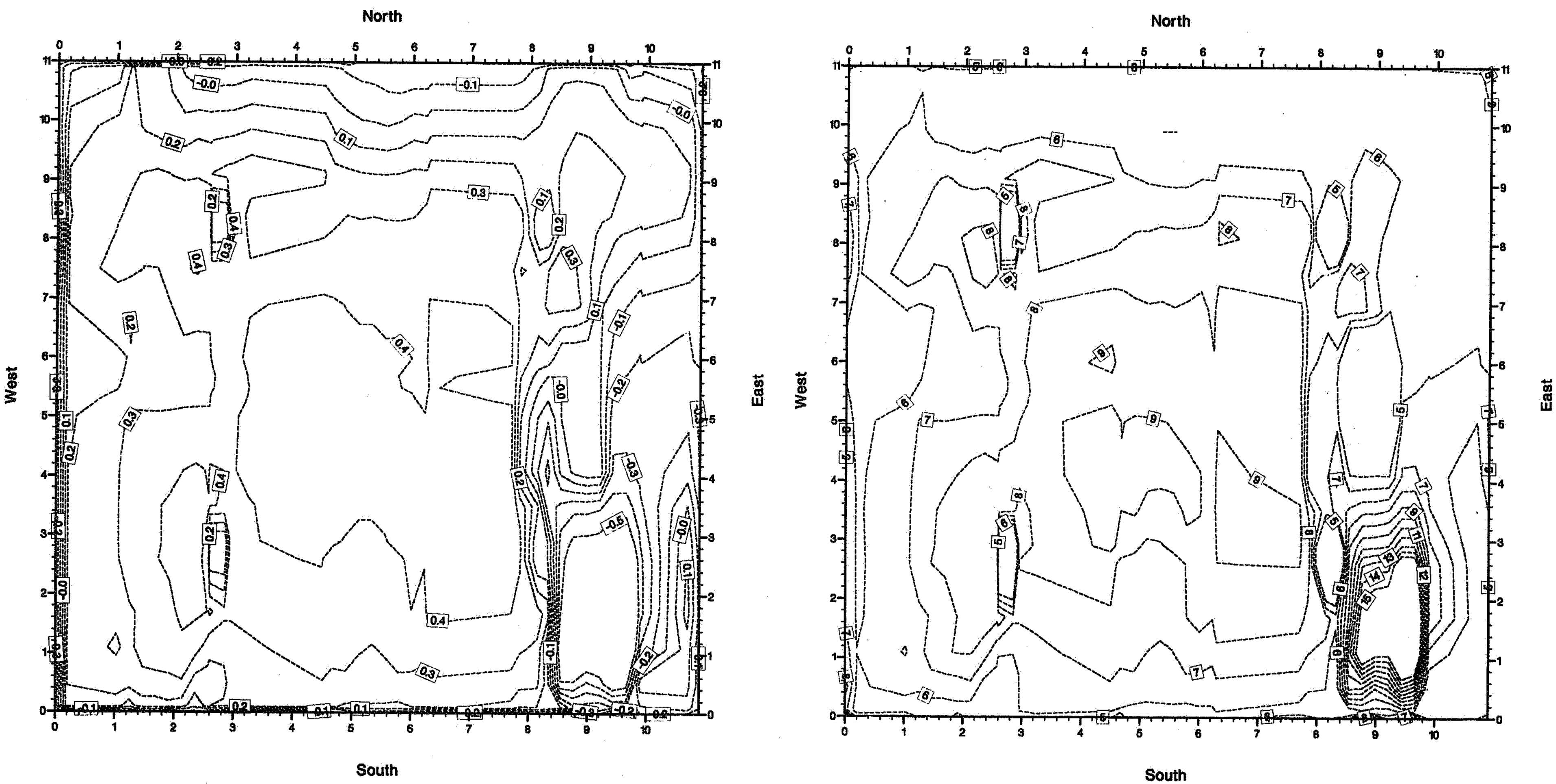


(a) Predicted mean vote

(b) Predicted percentage of dissatisfied (%)

Supply air: U = 0.356 m/s ; T = 19.9 deg.C

Fig. 6. Contours of predicted mean vote and predicted percentage of dissatisfied on a horizontal plane (0.9 m above floor) of a class room with occupancy (case 1)

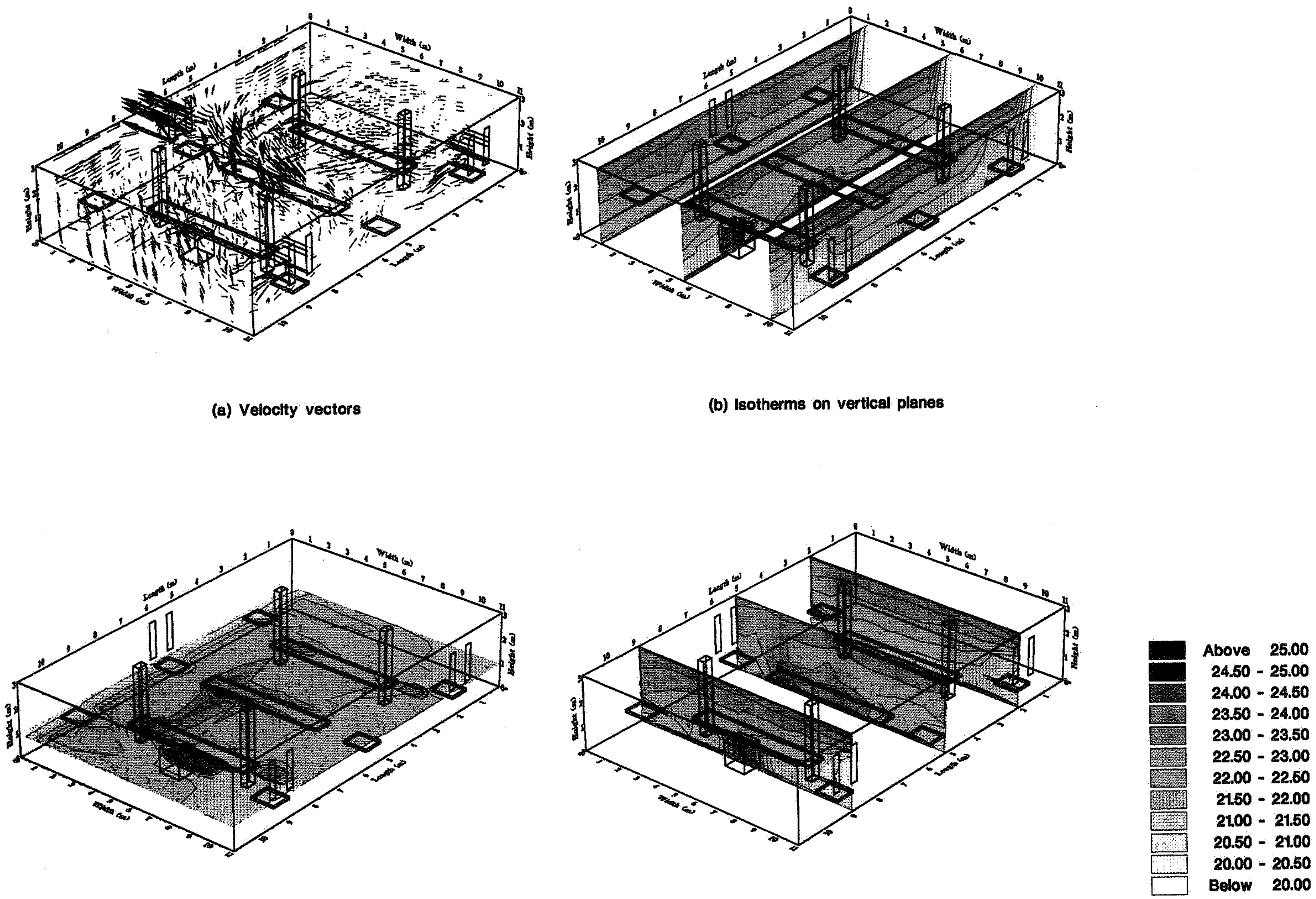


(a) Predicted mean vote

(b) Predicted percentage of dissatisfied (%)

Supply air: U = 0.76 m/s ; T = 19.9 deg.C

Fig. 7. Contours of predicted mean vote and predicted percentage of dissatisfied on a horizontal plane (0.9 m above floor) of a class room without occupancy (case 6)



(a) Velocity vectors

(b) Isotherms on vertical planes

(c) Isotherms on a horizontal plane

(d) Isotherms on vertical planes

Supply air: U = 0.356 m/s ; T = 19.9 deg.C

Fig. 8 Predicted velocity vectors & isotherms of air in a classroom without occupancy