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# SIMULATION OF AIR FLOW IN NATURALLY VENTILATED BUILDINGS

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ABSTRACT

The air flow pattern and temperature distribution in a naturally ventilated classroom were simulated using CFD techniques. The simulation model consists of equations for the conservation of mass, momentum and thermal energy, taking account of the effects of buoyancy and obstacles in the room. The well known k-$\epsilon$ turbulence model was used to simulate the effect of air turbulence. Close to the inside surface of the room and the obstacle boundary, the wall-function equations were used for momentum and heat flux. Heat sources existing in the classroom were included in the simulation. The equations were solved using the finite volume method and the SIMPLE algorithm. The predicted velocity and temperature distributions are in agreement with experimental data.

## NOMENCLATURE

- A cross-sectional area of supply outlet
- $C_p$ specific heat of fluid at constant pressure
- $C_\mu$ constant
- E logarithmic law constant (=9.793)
- g gravitational acceleration
- $I_u$ turbulence intensity of the U component of air velocity at supply outlet
- k kinetic energy of turbulence
- p pressure
- q heat production (W/m$^2$)
- T mean temperature
- $T^+$ dimensionless heat flux temperature
- U,V,W mean velocity components in x,y,z directions
- $U_i$ mean velocity component in $x_i$ direction
- $u^+$ dimensionless velocity
- $x_i$ coordinate in tensor velocity
- $y^+$ local Reynolds number

### Greek letters

- $\beta$ volumetric expansion coefficient
- $\Gamma$ diffusion coefficient
- $\epsilon$ turbulence dissipation rate
- $\kappa$ Karman's constant (=0.4187)
- $\mu,\mu_t$ laminar and turbulent viscosities
- $\rho$ fluid density
- $\sigma,\sigma_t$ laminar and turbulent Prandtl numbers

### Subscripts

- e effective
- i vector dimension
- k kinetic energy
- o supply outlet
- t turbulent
- $\epsilon$ kinetic energy dissipation rate

## INTRODUCTION

Buildings today tend to be designed for low energy consumption and better comfort. This often means tighter insulation for building shells. On the other hand, most buildings in the U.K. are naturally ventilated. The air flow in such buildings is largely dependent on the arrangement of doors and windows for given external conditions. Air infiltration design is usually based on the local climate (e.g. the prevailing wind direction) but rarely takes into account the air flow characteristics across and through the buildings. Croome (1989) stresses the need to consider details of the air flow patterns especially at head and foot levels when designing an effective ventilation system. Field tests on natural ventilation systems are difficult due to the uncertainty of various factors influencing their performance. The development of fast computers and accurate computer algorithms has presented a potential alternative to physical testing on the performance of ventilation systems.

In the last decade or so, there have been a lot of applications of computational fluid dynamics (CFD) in the simulation of room ventilation. Many of them are concerned with airconditioned spaces (see, e.g., Awbi 1989; Awbi and Setrak 1987; Holmes et al. 1990). The simulation of air flow in spaces or cavities dominated by buoyancy has also been carried out by several investigators such as Ideriah (1980b) and Markatos and Pericleous (1984). Tsutsumi et al. (1988) investigated the air flows in a model room caused by cross-ventilation under isothermal conditions using the k-$\epsilon$ and LES (large eddy simulation (Deardorff 1973)) models. However, the measurements were carried out only when the wind direction was perpendicular to the openings. They found that the k-$\epsilon$ turbulence model produced better prediction than the large eddy simulation model.

These investigations were largely performed for reduced-scale models or numerical experiments except the one by Holmes and his colleagues (1990) who simulated the air flow in a full-scale perimeter office space using a 2-D CFD code coupled with a thermal dynamic model. Verifications of CFD programs using field measurements seem to be lacking, especially in naturally ventilated buildings. Zainal and Croome (1990a, 1990b and 1990c) have conducted a series of investigation into ventilation characteristics of a naturally ventilated class room by various combinations and positions of door-window openings. This work deals with the validation of a computer model for the simulation of naturally ventilated buildings.

## MODEL EQUATIONS AND SOLUTION

### Model equations

The simulation model consists of a set of governing equations which are used for solving for the conservation of mass, momentum and energy. They are represented by the continuity equation, Navier-Stokes equation and thermal energy equation respectively. Coupled with these are the well known $k$-$\epsilon$ turbulence model equations to represent turbulence parameters for air. For a steady incompressible flow the time-average model equations can be generalised in the following form:

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

where $S_\phi$ = source terms of the dependent variable $\phi$ ($\phi$ = 1, U, V, W, T, k, $\epsilon$) (see Table 1)

### Boundary conditions

The model equations are solved with the following assumptions imposed on the room boundaries.

Supply conditions. The velocity and temperature of supply air are considered to be the known values from measurement. The kinetic energy of turbulence is calculated from

Table 1. Source terms of the model equations

| Equation | $\phi$ | $\Gamma_\phi$ | $S_\phi$ |
|---|---|---|---|
| Continuity | 1 | 0 | 0 |
| U-momentum | U | $\mu_e$ | $-\frac{\partial p}{\partial x} + \frac{\partial}{\partial x}(\mu_e \frac{\partial U}{\partial x}) + \frac{\partial}{\partial y}(\mu_e \frac{\partial V}{\partial x}) + \frac{\partial}{\partial z}(\mu_e \frac{\partial W}{\partial x}) - \frac{2}{3}\frac{\partial}{\partial x}(\rho k)$ |
| V-momentum | V | $\mu_e$ | $-\frac{\partial p}{\partial y} + \frac{\partial}{\partial x}(\mu_e \frac{\partial U}{\partial y}) + \frac{\partial}{\partial y}(\mu_e \frac{\partial V}{\partial y}) + \frac{\partial}{\partial z}(\mu_e \frac{\partial W}{\partial y}) - \frac{2}{3}\frac{\partial}{\partial y}(\rho k) - g(\rho - \rho_0)$ |
| W-momentum | W | $\mu_e$ | $-\frac{\partial p}{\partial z} + \frac{\partial}{\partial x}(\mu_e \frac{\partial U}{\partial z}) + \frac{\partial}{\partial y}(\mu_e \frac{\partial V}{\partial z}) + \frac{\partial}{\partial z}(\mu_e \frac{\partial W}{\partial z}) - \frac{2}{3}\frac{\partial}{\partial y}(\rho k)$ |
| Temperature | T | $\Gamma_e$ | $q/C_p$ |
| Kinetic energy | k | $\Gamma_k$ | $G_s - C_D \rho \epsilon + G_B$ |
| Dissipation rate | $\epsilon$ | $\Gamma_\epsilon$ | $C_1 \frac{\epsilon}{k}(G_s + G_B) - C_2 \rho \frac{\epsilon^2}{k}$ |

Notes: $\mu_e = \mu + \mu_t$ $\qquad \mu_t = C_\mu \rho k^2/\epsilon$

$\Gamma_e = \mu_t/\sigma_t + \mu/\sigma$ $\qquad \Gamma_k = \mu_e/\sigma_k$ $\qquad \Gamma_\epsilon = \mu_e/\sigma_\epsilon$

$$\sigma_\epsilon = \frac{\kappa^2}{(C_2 - C_1)C_\mu^{1/2}} \qquad G_B = \beta g \frac{\mu_t}{\sigma_t}\frac{\partial T}{\partial y}$$

$$G_s = \mu_t \left\{2\left[\left(\frac{\partial U}{\partial x}\right)^2 + \left(\frac{\partial V}{\partial y}\right)^2 + \left(\frac{\partial W}{\partial z}\right)^2\right] + \left(\frac{\partial U}{\partial y} + \frac{\partial V}{\partial x}\right)^2 + \left(\frac{\partial U}{\partial z} + \frac{\partial W}{\partial x}\right)^2 + \left(\frac{\partial V}{\partial z} + \frac{\partial W}{\partial y}\right)^2\right\}$$

Empirical constants in the $k$-$\epsilon$ turbulence model equations

| $C_\mu$ | $C_D$ | $C_1$ | $C_2$ | $\sigma_k$ | $\sigma_t$ |
|---|---|---|---|---|---|
| 0.09 | 1.0 | 1.44 | 1.92 | 1.0 | 0.9 |

$$k_0 = \frac{3}{2} I_u^2 U_0^2$$

The dissipation rate is:

$$\epsilon_0 = \frac{k_0^{1.5}}{\sqrt{A}\ C_\mu^{0.25}}$$

The pressure at the supply outlet is set to zero for computational purposes.

Exit conditions. The air velocity and temperature at the exit are obtained from the continuity equation and thermal energy balance equation respectively and assumed to be uniformly distributed across the exit area. The kinetic energy and its dissipation rate are not required due to the upwind computational scheme to be mentioned. Other quantities such as the pressure and the gradients of k and $\epsilon$ are taken as zero.

Wall boundary. Since the k-$\epsilon$ model is applicable only for high Reynolds number flows, the wall-function equations (Launder and Spalding, 1974) are used for the calculation of the velocity parallel to the boundary and heat flux through the wall, i.e.,

for $y^+ \leq 11.63$: $u^+ = y^+$

and $T^+ = \sigma y^+$

for $y^+ > 11.63$: $u^+ = \frac{1}{\kappa} \ln(Ey^+)$

and $T^+ = \sigma_t \, [u^+ + f(\frac{\sigma}{\sigma_t})]$

where $f(\sigma/\sigma_t)$ is a function of the ratio $\sigma/\sigma_t$. For smooth surfaces, Jayatillaka (1969) recommended the following correlation:

$$f(\frac{\sigma}{\sigma_t}) = 9.24 \, [(\frac{\sigma}{\sigma_t})^{0.75} - 1]$$

$$\{1 + 0.28 \exp[-0.007(\frac{\sigma}{\sigma_t})]\}$$

The boundary temperature is assumed to be the measured room surface temperature whenever available. The velocity components perpendicular to the boundary are taken to be zero.

The treatment of the wall boundary is also applied to the surface of obstacles in the room.

## Solution method

The model equations are solved for the 3-D cartesian system using the SIMPLE algorithm (Patankar 1980). In this method, the partial differential equations are discretised by means of a finite volume technique, i.e., by integration of the equations over a control volume on a staggered grid to yield finite difference equations. A hybrid (upwind/central) differencing scheme is used for the integration.

Due to the non-isothermal flow of air in naturally ventilated rooms, use is made of an inertial relaxation method for the vertical velocity component to increase the numerical stability (Ideriah 1980a), in conjunction with under relaxation factors which are applied to all the field equations.

Convergence was considered to have been reached when the sum of the residuals for each of the model equations for the whole field was less than 30% of the corresponding fluxes at the supply opening. This would take about 200 iterations. For a grid size of 40 x 24 x 20, the CPU time for the calculation is about 33 minutes using an Amdahl 580 mainframe.

## RESULTS AND DISCUSSION

The model was applied for the prediction of air flow in a naturally ventilated classroom for summer season. The room investigated has a dimension of 10.9 x 11 x 3.05 m (length x width x height), with east-west orientation. A schematic diagram of the room is shown in Fig. 1. 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. 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 carbon dioxide as the tracer gas. An infra-red gas analyser was used for the measurement of the carbon dioxide levels. For details of these tests reference should be made to Zainal and Croome (1990c).

In the computer prediction, the solar heat gain through the glazed area facing south was accounted for by assuming that it was distributed uniformly over the floor. Similar treatment was made for heat generation due to instruments in the room. Heat production by the artificial lights near the ceiling was considered to be a uniformly-distributed heat source over the ceiling. The air velocity at the supply openings was calculated from the measured air flow rate. Due to the difficulties in determining the velocity direction, the supply velocity was assumed normal and uniformly distributed across the opening in order to simplify the simulation.

A total of five predictions were performed for the room unoccupied and two of them are discussed here. The predicted and measured average air velocities and temperatures on the level 0.9 m above the floor are shown in Table 2. It can be seen from the table that the program gives good predictions of the average values of the velocity and temperature.

Figures 2 and 3 show a comparison of predicted velocities and temperatures with the measured values for the two cases to be discussed here. The predicted velocity and temperature distributions for the corresponding two cases are shown in Figures 4 and 5. The air flow rates for case 1 and case 2 are 8.76 and 7.73 air changes per hour respectively. In case 1, six windows were half open. Air flowed into the room through four windows in the south face and flowed out from two windows in the north face. In case 2, there were six windows in fully open position in the south wall, with air flowing into the room via two windows close to the east wall and out from the other four. It can be seen from Figures 2 and 3 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 may be mainly attributed to the assumption of 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 two experiments, the wind direction was south-westerly. 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 formed at the opening. Another important factor which might affect the accuracy of prediction is the assumption made for accounting for solar radiation. According to Andrade (1990) air flow patterns were influenced by the position of the heat source due to solar radiation. In the present calculations the solar heat gain was however simplified to be uniformly distributed over the floor. A possible source for the deviation is the air infiltration through cracks and doors which would have contributed to the total air flow rate used in the calculation of the supply velocity. Nevertheless, it should be pointed out that the prediction has been compared with field measurements which are much more complex than measurements in scale or laboratory models both physically and thermodynamically. In view of this, the predictions can be considered satisfactory.

From the predictions it appears that there is little air movement in the regions far from the air jet. In these stagnant regions the air temperature is higher than that within the jet flow region and may be a cause of discomfort to those occupants using them, particularly in a hot climate. The effectiveness of ventilation is usually assessed by the comfort and air quality in the occupied zone. However, in the current predictions such a comparison of the effectiveness is difficult due to the variations in the internal and external conditions for different experiments. The effect of air supply location on the air flow and ultimately the ventilation effectiveness, which takes account of the thermal comfort, energy efficiency and air quality are the subject of further investigations.

## CONCLUSIONS

The validation of the ventilation model has shown that CFD is a useful tool for simulating natural ventilation. The 3-D CFD program that has been developed can be used to predict the air flow in naturally ventilated rooms provided that accurate measurements are made for dominant parameters such as supply air conditions and thermal load distribution.

This work is to be extended for analysing the stochastic influence of occupants and their distribution in the space on indoor air quality and comfort and assessing the ventilation effectiveness by appropriate arrangements of window or door opening.

## REFERENCES


Andrade, J.A. 1990. "A numerical study of air movement in rooms with solar radiation gains" Roomvent '90. Oslo, Norway.

Awbi, H.B. 1989. "Application of computational fluid dynamics in room ventilation". Building and Environment, 24(1): 73-84.

Awbi, H.B. and A.A. Setrak. 1987. "Air jet interference due to ceiling-mounted obstacles". Roomvent '87. Stockholm.


Table 2. Predicted and measured average air velocities and temperatures in a class room 0.9 m above the floor

| No. | Air change rate (1/hr) | Velocity (m/s) | | Temperature (deg. C) | |
|---|---|---|---|---|---|
| | | Prediction | Measurement | Prediction | Measurement |
| 1 | 8.76 | 0.133 | 0.114 | 25.29 | 25.31 * |
| 2 | 7.73 | 0.113 | 0.092 | 21.89 | 21.98 * |
| 3 | 10.20 | 0.123 | 0.110 | 23.69 | 24.43 |
| 4 | 14.34 | 0.161 | 0.107 | 27.86 | 27.52 |
| 5 | 2.98 | 0.079 | 0.094 | 23.07 | 20.88 |
| Mean | | 0.122 | 0.103 | 24.36 | 24.02 |

Note: * discussed in the text.

Croome, D.J. 1989. "A comparison of upward and downward air distribution systems". Ventilation '88. London, 433-438.

Deardorff, J.W. 1973. "The use of subgrid transport equations in a three-dimensional mode of atmospheric turbulence". J. Fluids Engineering, 95: 429-438.

Holmes, M.J.; J.K.W. Lam; K.G. Ruddick; and G.E. Whittle. 1990. "Computation of conduction, convection and radiation in the perimeter zone of an office space". Roomvent '90. Oslo, Norway.

Ideriah, F.J.K. 1980a. "An inertial relaxation method of curing numerical instability in prediction of flows influenced by severe body forces". J. Mech. Engng Sci., 22: 153-156.

Ideriah, F.J.K. 1980b. "Prediction of turbulent cavity flow driven by buoyancy and shear". J. Mech. Engng Sci., 22: 287-295.

Jayatillaka, C.L.V. 1969. "The influence of Prandtl number and surface roughness on the resistance of the laminar sub-layer to momentum and heat transfer". Progress in Heat and Mass Transfer, 1: 193-329.

Launder, B.E. and D.B. Spalding. 1974. "The numerical computation of turbulent flows". Comput. Methods Appl. Mech. Engng, 3: 269-289.

Markatos, N.C. and K.A. Pericleous. 1984. "Laminar and turbulent natural convection in an enclosed cavity". Int. J. Heat and Mass Transfer, 27: 755-772.

Patankar, S.V. 1980. Numerical Heat Transfer and Fluid Flow. Hemisphere Publishing Co., Washington.

Tsutsumi, J.; T. Katayama; T. Hayashi; Q. Zhang; and H. Yoshimizu. 1988. "Numerical simulation of indoor turbulent air flows caused by cross-ventilation and its model experiments". Proceedings of the 9th AIVC Conference. Belgium.

Zainal, M. and D.J. Croome. 1990a. "Building planning and ventilation: The effects of natural ventilation via windows and doors linked by corridor/passage-way". Roomvent '90. Oslo, Norway.

Zainal, M. and D.J. Croome. 1990b. "Ventilation characteristics for buildings". Indoor Air '90. Toranto, Canada.

Zainal, M. and D.J. Croome. 1990c. "Ventilation characteristics of selected type of buildings and indoor climate". 11th AIVC Conference. Belgirate, Italy.

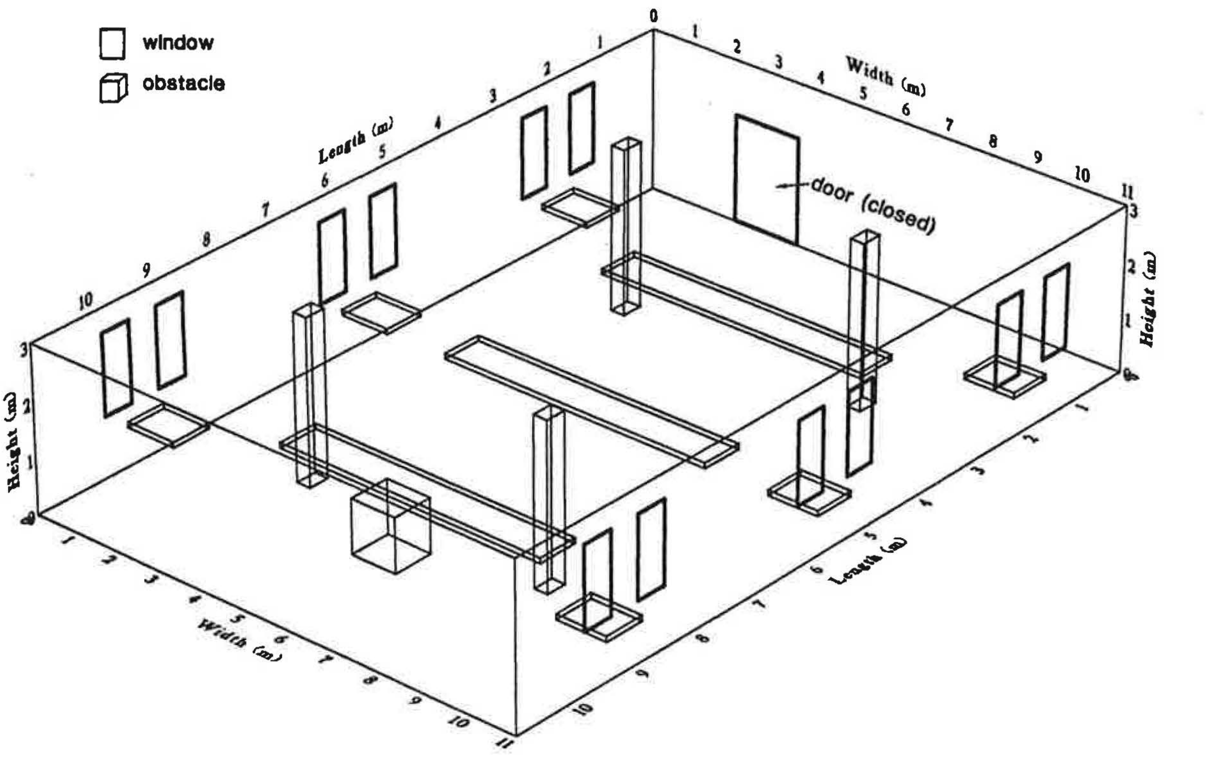
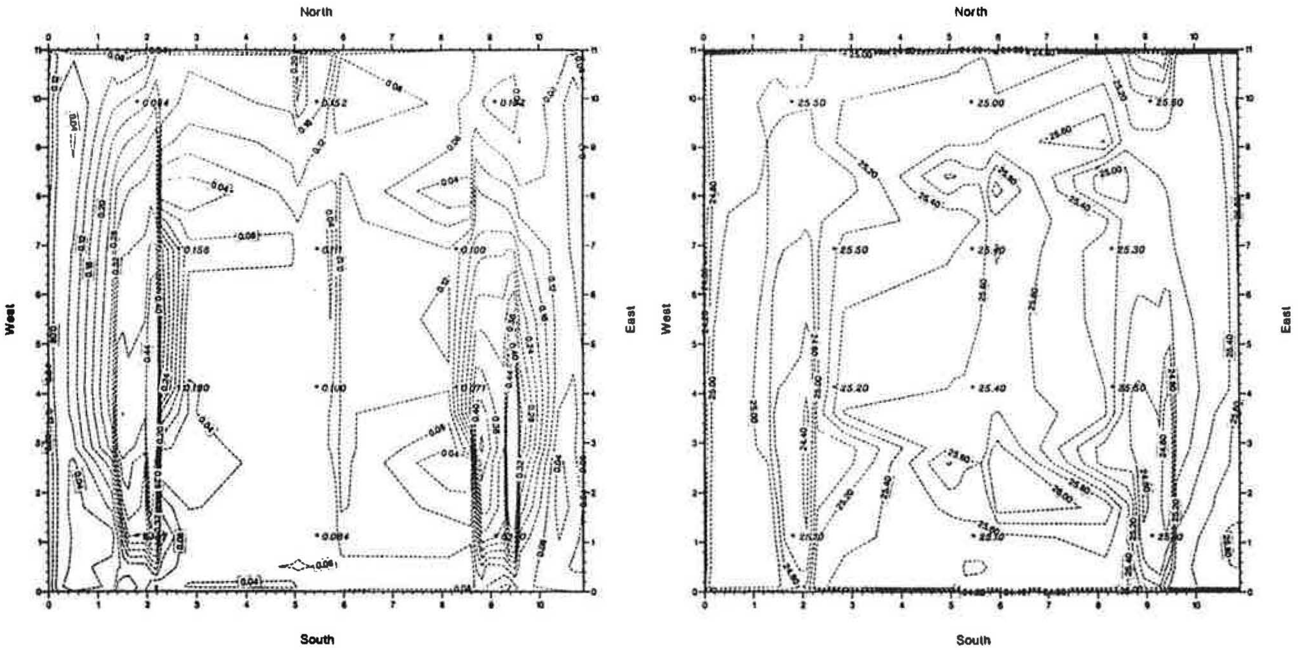



Fig. 1 Schematic diagram of the class room

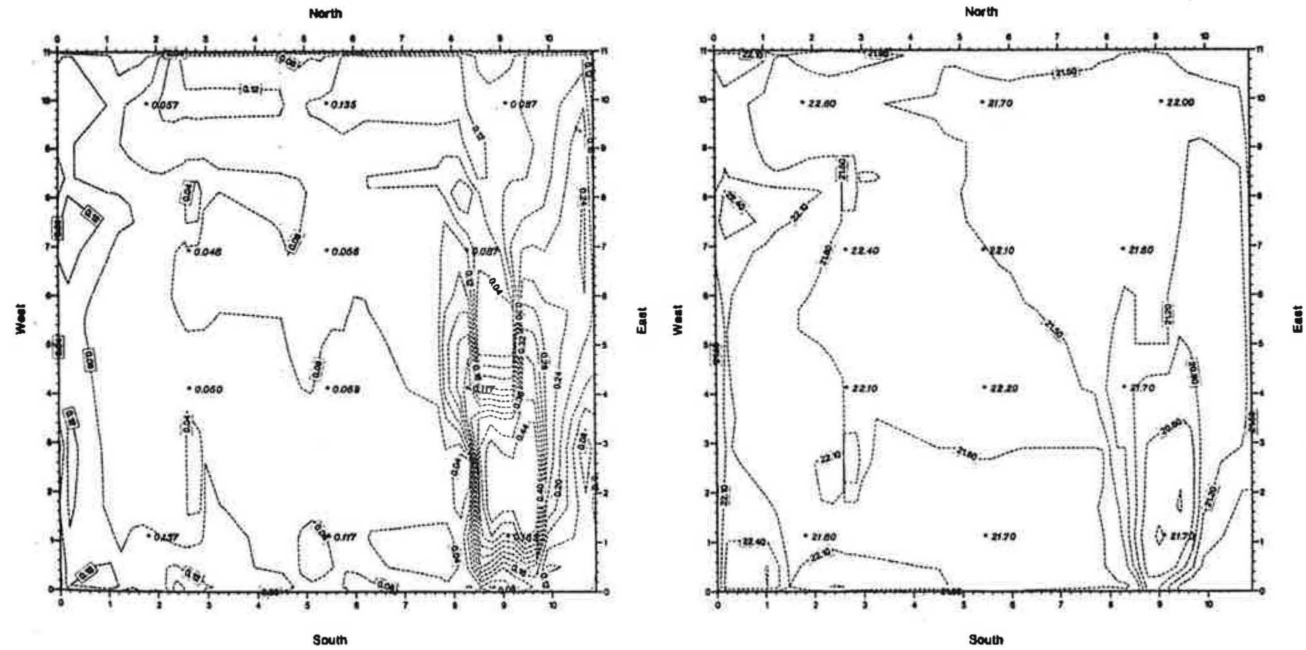



(a) Velocity (m/s)

(b) Temperature (deg.C)

Supply air: U = 0.86 m/s ; T = 24.1 deg.C

Fig. 2 Comparison of predicted isovels and isotherms with measured velocities and temperatures (•) on a horizontal plane (0.9 m above the floor) of a class room without occupancy (case 1)




(a) Velocity (m/s)

(b) Temperature (deg.C)

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

Fig. 3 Comparison of predicted isovels and isotherms with measured velocities and temperatures (•) on a horizontal plane (0.9 m above the floor) of a class room without occupancy (case 2)