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MEASUREMENTS AND COMPUTATIONS ON AIR MOVEMENT AND TEMPERATURE DISTRIBUTION IN A CLIMATE ROOM

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1. SUMMARY

RESUME: Le problème de la convection mixte forcée et naturelle dans un local a été étudié expérimentalement et théoriquement. Dans une chambre on a procédé à des mesures de champs de vitesse et de températures d'air. Les équations bi ou tridimensionelles de conservations de quantité de mouvement, d'énergie, d'énergie turbulent et du taux de dissipation ont été résolues par différences finies avec les programmes CHAMPION SGE (2D) et PHOENICS (3D). Les 2 cas étudiés sont:

- un local refroidi (par de l'air froid) et chauffé par le rayonnement solaire pénétrant à travers les fenêtres;
- un local chauffé et présentant une paroi vitrée (froide).

Les résultats obtenus montrent que les simulations sont en bon accord avec les mesures. Même un maillage assez grossier donne des résultats satisfaisants. Notons cépendant qu'en régime transitoire, le champs de température simulée évolue plus rapidement que le champs des vitesses. D'autre part, il faut souligner que des problèmes tridimensionels coûtent cher à résoudre (en temps de calcul) et que, malheureusement, des modèles 2D ne peuvent pas, en toute généralité, être employés pour résoudre des problèmes 3D.

2. INTRODUCTION

A suitable distribution of air velocity, temperature and humidity are important in such a way that helps to control the comfort and well being of individuals. Adequate mixing of inlet air with the room air is required to obtain a uniform temperature and fresh air distribution. At the same time the velocity should not be so intense that occupants feel draught. A lot of numerical studies of buoyancy affected flows have been reported in the last decade, but only a few of them were compared with experiments. Nielson et al [1], Jedrzejewska-Sibak et al [2] and Hjertager et al [3] gave comparisons between computation and measurement of two and three dimensional problems under steady boundary conditions. In the present study, the air movement and heat transfer in an air conditioned room is studied numerically and experimentally in order to reveal the steady and transient nature of mixing convection.

3. PROBLEM FORMULATION AND MEASUREMENT PROCEDURE

The computer code PHOENICS (3D) and CHAMPION SGE (2D) were used for the computations [4], [5]. The continuity, momentum and energy equations, in time averaged form, governing a three dimensional turbulent flow affected by buoyancy, which co-operate with k-ε turbulence model, were used in the computer codes in order to predict the air movement and heat transfer in a room. All differential equations are reducible to the general form of

$$\frac{\partial}{\partial t}(\rho\phi)+\mathrm{div}(\rho\vec{V}\phi-\Gamma_\phi \mathrm{grad}\ \phi)=S_\phi+S_{Buoyancy}$$

where ϕ, Γ_ϕ, S_ϕ and $S_{Buoyancy}$ are given in Table 1.

Table 1. Values of ϕ, Γ_ϕ, S_ϕ and $S_{Buoyancy}$ Terms

ϕ	Γ_ϕ	S_ϕ	$S_{Buoyancy}$
1	0	0 (continuity)	0
V_i	μ_{eff}	$-\partial p/\partial x_i+\partial(\mu_{eff}(\partial V_j/\partial x_i+\partial V_i/\partial x_j))/\partial x_i$	$-\rho\beta g_i\theta$
h	μ_{eff}/σ_h	0	0
k	μ_{eff}/σ_k	$G-\rho\varepsilon$	G_B
ε	$\mu_{eff}/\sigma_\varepsilon$	$\varepsilon(C_1G-C_2\rho\varepsilon)/k$	$C_3\varepsilon G_B/k$

V_i = u, v, w; velocity components in x, y and z direction respectively.

$\mu_{eff}=\rho(\nu+\nu_t)$; ν and ν_t are laminar and turbulent viscosity respectively.

$\mu_t=C_D k^2/\varepsilon$.

$\theta=T-T_0$; excess temperature, where T_0 is reference value.

$G=\mu_t(\partial V_i/\partial x_j+\partial V_j/\partial x_i)\partial V_j/\partial x_i$

$G_B=\rho\beta g_i \frac{\nu_t}{\sigma_h}\frac{\partial\theta}{\partial x_i}$

$C_1=1.44$, $C_2=1.92$, $C_D=0.09$, $C_3=1.44$, $\sigma_k=1.0$, $\sigma_\varepsilon=1.3$, $\sigma_h=0.9$

The wall function method of Launder et al [6] has been used for the near wall regions. The partial differential eauations were solved by finite-domain method in the algorithm reported in [4]. A detailed description of the mathematic basis and the computer code manual is available.

The temperature fields were measured through thermo-couples by means of a data logging system and converted into printed results by a micro-computer. The velocities were observed by means of smoke.

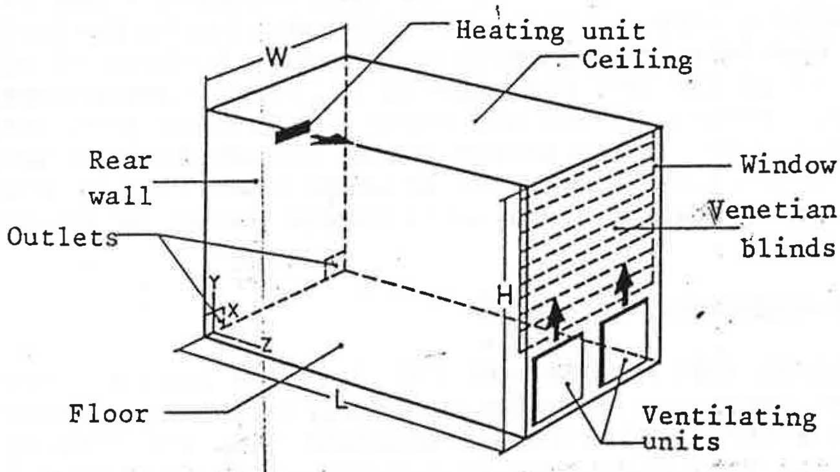


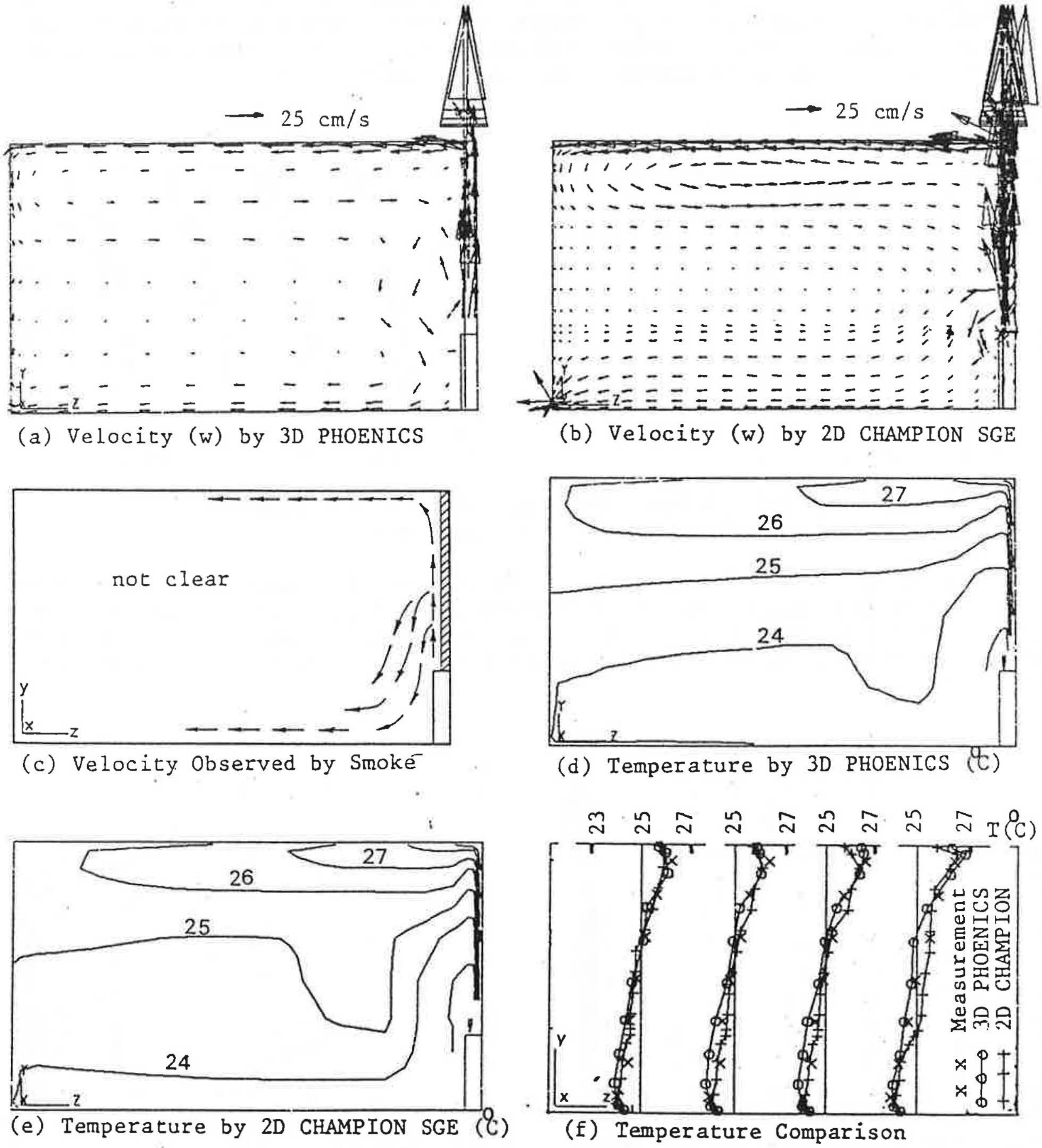
Fig.1 Geometry of the room and notation

4. RESULTS

COOLING

The above model was applied to the case described in Fig.1. It consists of a rectangular room 3.2m high, 3m wide and 5.6m long. The two inlet units of the cooling system (100cm*0.8cm each) are located near the window. The inlet mass flow for the room was 0.075m³/s (that is, ventilation rate five times/hour) which corresponds to a Reynolds number 2400. A concentrated heat (a step fuction of 950W) was put on the venetian blinds. The outlet temperature was controlled at 23.0 °C. The heating unit did not function in this case.

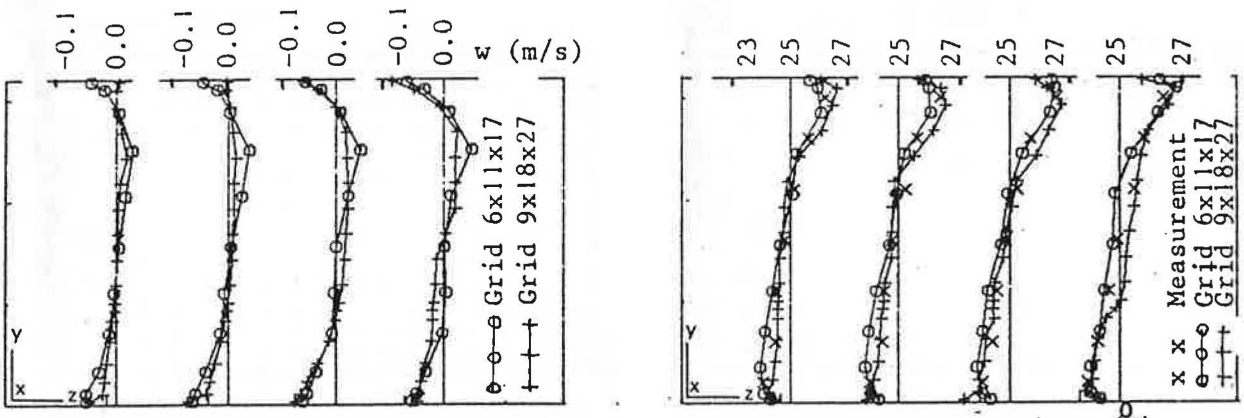
The computed and measured velocity and temperature fields (3D by PHOENICS and 2D by CHAMPION SGE) in the section x=0.25W, are shown in Fig.2(a-f). The agreement between the 2D, 3D computations and measurements is rather good. The



(a) Velocity (w) by 3D PHOENICS
(b) Velocity (w) by 2D CHAMPION SGE
(c) Velocity Observed by Smoke
(d) Temperature by 3D PHOENICS (°C)
(e) Temperature by 2D CHAMPION SGE (°C)
(f) Temperature Comparison

Fig.2 Velocity and temperature distributions of cooling system by 2D and 3D computations and measurements air temperature difference between the ceiling and the floor is about 3.5K (see Fig.2 (f)). It seems too big to be used in practical engineering, but the temperature differences in the occupied zone are less than 2K. The highest velocity in the occupied zone is less than 25cm/s. Therefore the occupants will not feel draught.

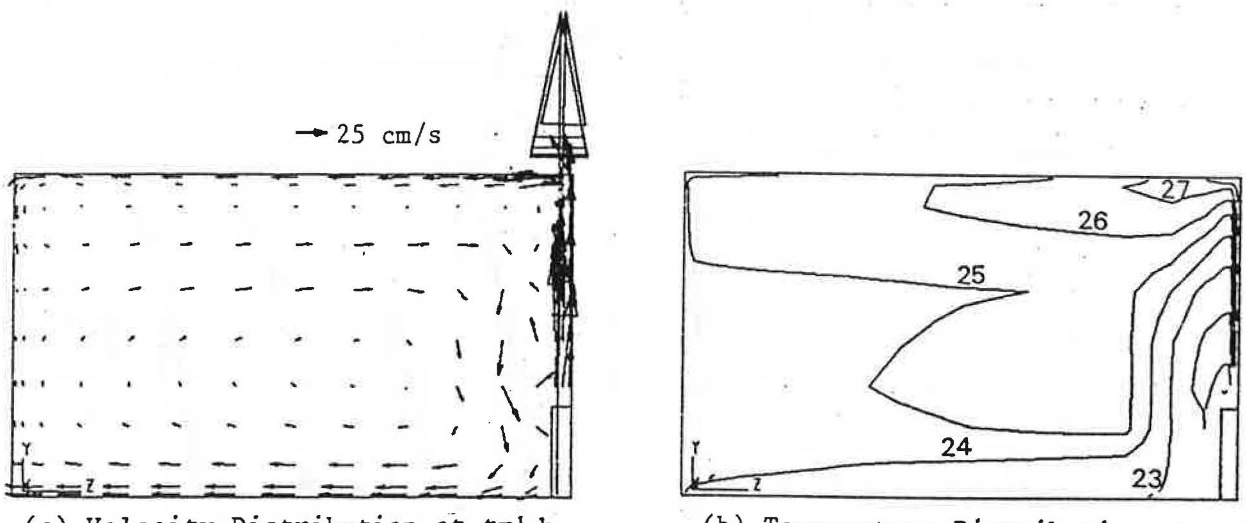
A comparison between a finer grid number (9*18*27) and a coarse grid number (6*11*17), which was used above, is shown in Fig.3(a÷b). The differences are very small. The biggest temperature difference is 1.0K and the biggest velocity difference 4.0cm/s. The computing time (CPU) for grid number 9*18*27 is 40 minutes and for grid number 6*11*17 8 minutes by PHOENICS and for grid number 18*27 2 minutes by CHAMPION SGE in an IBM 3083÷JX1 computer. For 2D computation, it is easier to get convergent results.



(a) Velocity Comparison (b) Temperature Comparison (C)

Fig.3 Velocity and temperature comparisons between two kinds of grid numbers

Fig.4(a÷b) gives the velocity and temperature distribution one hour after the concentrated heat was put in. Comparing these figures with those six hours after the concentrated heat was put in (see Fig.2(a) and (d)), it is clear that temperature distribution changes a lot while velocity remains as a constant.



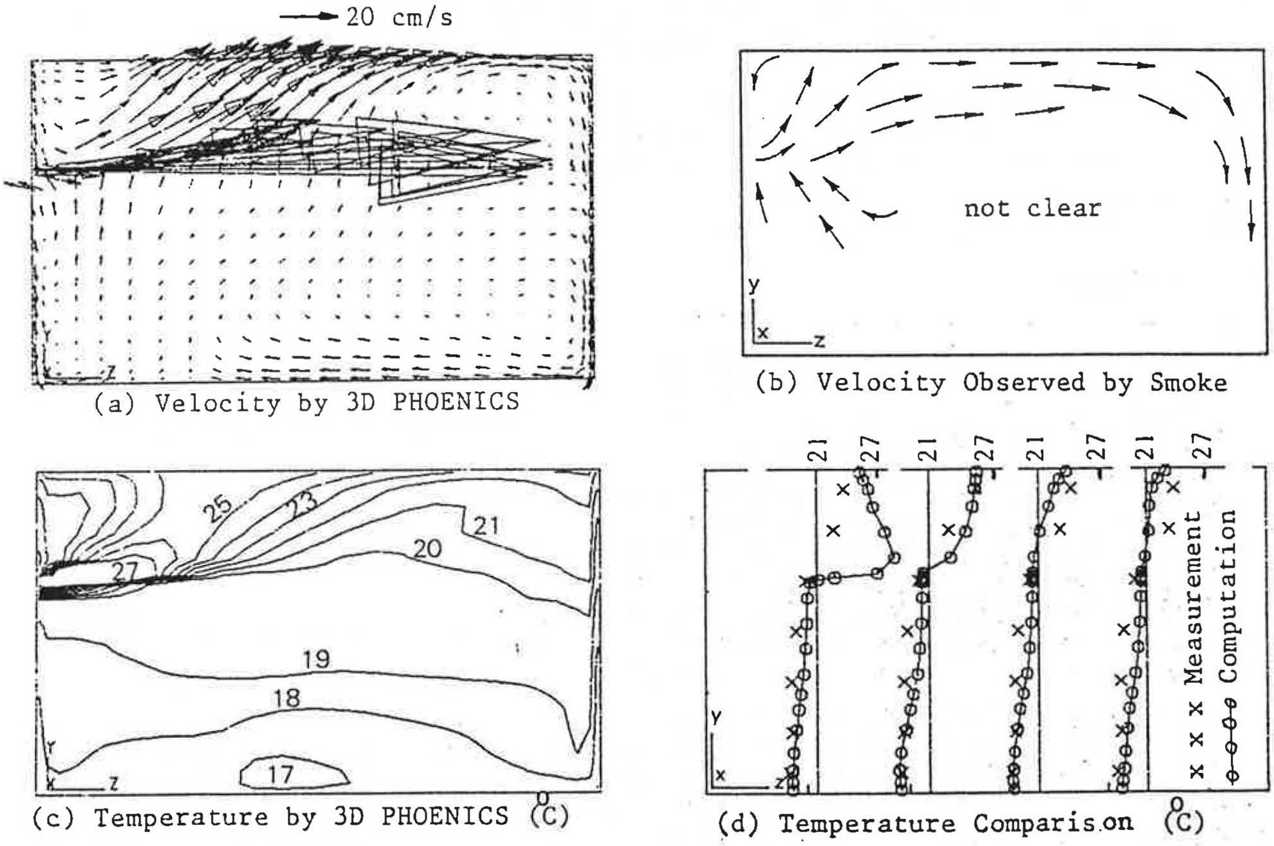
(a) Velocity Distribution at t=1 h (b) Temperature Distribution at t=1h

Fig.4 Velocity and temperature distribution of 3D computation at t=1 hour

HEATING

The heating system concerned a mixture of natural convection with cold window surface and forced convection with a hot air supply (1000W) in the rear wall in 2/3 height of the room through the heating unit as shown in Fig.1. The cooling inlet units were closed for the system. The temperature on the window surface was 11 °C and on the other surfaces about 20 °C. The corresponding inlet Reynold number was about 5000.

The computations by PHOENICS and the measurement of the velocity and temperature distributions in the section x=0.5W are shown in Fig.5(a-d). The agreement of the velocity profiles between the measurements and computation is good but there are some discrepancies in the temperature distribution near the heating unit as shown in Fig.5(d). The cause of this discrepancy is difficult to find, maybe it was caused by the possible differences between the numerical and experimental boundary conditions with respect to inlet turbulence parameters. This system was also computed with the 2D computer code CHAMPION SGE. The results, however, were not in good agreement with the measurements. The reason is that for this typical 3D situation the temperature and velocity in x-plane change very much so that a 2D computation for this case can not result in good predictions.



(a) Velocity by 3D PHOENICS
(b) Velocity Observed by Smoke
(c) Temperature by 3D PHOENICS (°C)
(d) Temperature Comparison (°C)

Fig.5 Velocity and temperature distributions of heating system by 3D computations and measurements

5. DISCUSSION

The computations by the computer codes PHOENICS were in quite good agreement with the measurements both in temperature and velocity fields. A few discrepancies were found. For 2D computer code CHAMPION SGE, only the nearly

2D cooling system showed good results. A coarse grid mesh in the computation gave rather reasonable results. The differences between the two grid numbers were less than 1.0K for temperature and 4.0cm/s for the velocity. Therefore, a coarse mesh seems sufficient for practice. The prediction of the air flow in a room is very expensive. About 40 minutes CPU time of IBM 3083 computer is needed for a 3D steady computation with a total grid number 5000. It seems that this is the only way for a real 3D problem (like heating system above), because it is hard to get good results from 2D simulations.

Inlet sizes and locations seem to be very important to the temperature and velocity distributions. According to authors' experience, for cooling situation, an inlet located in the rear wall near the ceiling gave a much more uniform temperature distribution than the system presented in the former section (see [7]). The temperature difference of the occupied zone in the system in former section is less than 2K, this system meets the comfort requirement.

Under transient situation above, the temperature distribution may change very much but velocity remains as a constant. This means that the transient temperature distribution may be calculate easily through the 'unchanged' velocity distribution and the transient boundary conditions for a certain time interval instead of solving the governing equation each time step.

The heat exchange coefficients calculated from the computer codes are always too small compared with measurements. Therefore, the wall function used to compute the heat exchange coefficients has to be improved. In the computations above, all heat exchange coefficients were taken from measurements.

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