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PAPER NO. 34

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ABSTRACT

It is the purpose of this paper to investigate the ability of a k-ϵ turbulence model to predict air flow and comfort conditions in a displacement-ventilated room.

Stratification effects, re-laminarization and heat flux at walls cause difficulties when CFD is applied and different approaches to overcome these problems are presented. Comparative studies between ordinary versus extended turbulence models are carried out and the effect of applying different types of temperature boundary conditions is shown. The exchange of radiative energy between the surfaces of the room and the heat flux at the walls are included to complete the description.

The results show promising features and it is shown how the distribution of percentage dissatisfied occupants in the room can be obtained by combining criteria of human comfort and CFD results. This offers the opportunity to point out critical areas.

INTRODUCTION

Displacement ventilation for comfort purposes was introduced in Norway and Sweden ten to twenty years ago (*Skåret 1985*) and in spite of a relatively short history it has grown popular in Scandinavia and in other parts of Europe.

During this period of time the progress in the calculations of room air flow by CFD has made it interesting to investigate whether it is possible today to simulate air flow in a displacement-ventilated room.

The success of applying CFD depends on the mathematical model of the flow, the numerical method and the boundary conditions. In case of 3-D non-isothermal low velocity flow in rooms it is necessary to make a compromise between detailed methods, which improve the accuracy but often are expensive in terms of consumption of computing time, and a simpler approach which constitutes an acceptable approximation.

METHODS

The CFD method is based on numerical solution of the time averaged Navier-Stokes equations for, in this case, momentum, mass conservation and temperature. To obtain a closed system of equations which governs the fluctuating motion as well as the mean flow quantities the turbulent viscosity is introduced. The turbulent viscosity depends on the local turbulence level and is defined as:

$$\mu_t = \rho C_\mu \frac{k^2}{\epsilon} \tag{1}$$

in a k-ϵ model where C_μ is an empirical constant, ρ is density, k is turbulent kinetic energy and ϵ is dissipation rate. The turbulent quantities k and ϵ are calculated from :

$$\rho U_i \frac{\partial k}{\partial x_i} = \left(\frac{\mu_t + \mu_l}{\sigma_k}\right)\frac{\partial^2 k}{\partial x_i^2} + \underbrace{\mu_t \frac{\partial U_i}{\partial x_j}\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right)}_{P_k} + \underbrace{\beta g_i \frac{\mu_t}{\sigma_t}\frac{\partial T}{\partial x_i}}_{G_k} - \rho\epsilon \tag{2}$$

$$\rho U_i \frac{\partial \epsilon}{\partial x_i} = \left(\frac{\mu_t + \mu_l}{\sigma_\epsilon}\right)\frac{\partial^2 \epsilon}{\partial x_i^2} + \frac{\epsilon}{k}(c_1 P_k + c_3 G_k - c_2 \rho \epsilon) \tag{3}$$

The buoyancy term, G_k appearing in eq. (2) and eq. (3) decreases the turbulent viscosity in areas with positive vertical temperature gradients and increases it under unstable conditions corresponding to a negative gradient. It has been discussed earlier whether G_k plays an important role in a buoyancy affected flow and as the following results show it is in fact important in modelling a displacement-ventilated room.

Turbulent fluctuations are reduced in stagnant areas of the room due to stratification and re-laminarization effect. The k-ϵ model assumes fully turbulent flow and is not able to predict transition to laminar conditions correctly. This problem has been addressed by a number of authors e.g. *Davidsson 1989* who applies a low Reynolds number model.

Low Reynolds' number models are originally designed for boundary layers and a refined grid is required close to the wall in order to achieve an improved description of the near wall behaviour. This causes a drastic increment of the CPU time used to obtain a solution - especially in the 3-D case. Still it is desirable to apply a low Reynolds number expression in the interior of the room where stratification plays an important role as described in the following.

$$\mu_t = f_{R_t} f_b \, \rho C_\mu \frac{k^2}{\epsilon}$$

$$f_{R_t} = \exp\left(\frac{-3.4}{(1+R_t/50)^2}\right) \qquad f_b = \begin{cases} 0.0 \text{ for } b \le -10 \\ 1 + b/10 \text{ for } -10 < b < 0 \\ 1.0 \text{ for } b \ge 0 \end{cases} \qquad R_t = \frac{\rho k^2}{\mu_l \epsilon} \qquad b = \frac{G_k}{\epsilon} \tag{4}$$

R_t is the turbulent Reynolds number and f_{Rt} the corresponding damping function for re-laminarization (*Jones & Launder 1972*). Furthermore, a buoyancy damping function f_b, suggested by *Chikamoto, Murakami & Kato 1992*, is introduced to strengthen the stability of the stratification. The model was originally designed for the flow in atria where large temperature gradients and stagnant zones occur. In a displacement-ventilated room the same features are dominating and the model modifications of eq. (4) are accordingly relevant to apply.

Three different types of temperature boundary conditions are applied:

a) Adiabatic boundaries.

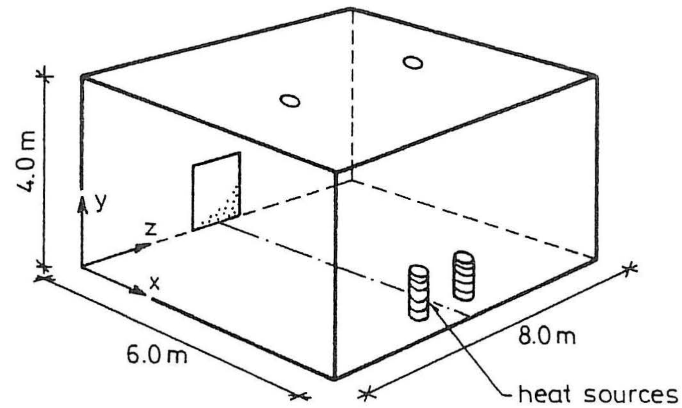
b) Prescription of a linear temperature profile at boundaries based on measurements. Calculation of heat flux by logarithmic wall function.

c) Adiabatic boundaries and radiative energy transfer model.

The temperature wall functions applied in case b) are of the standard type developed by (*Patankar & Spalding 1967*). The radiation model of case c) is a discrete transfer model (*Lockwood & Shah 1981*).

RESULTS

Measurements and CFD simulation are made with the set-up in the full-scale room shown in fig. 1. The symmetry makes it possible to perform the calculations for only half the room.



Inlet flow, Q_i : 300 m^3/h
Inlet temperature, T_i : 15.3 °C
Supplied heat : 600 W

Figure 1. The displacement-ventilated room employed for the measurements and for the model.

Air movement in a displacement-ventilated room is characterized by being mainly buoyancy driven which implies that the temperature gradient in the room is decisive. The shape of the resulting temperature gradients is consequently a suitable criterion for the comparison and evalution of the different turbulent models.

The vertical temperature profiles in fig. 2 are located at a point of the room where no direct influences of heat sources and inlet device are present (x=4.0m, z=7.0m). The velocity profiles to the right in fig. 2 are located at (x=2.0m, z=4.0m) which is in the symmetry plane of the room. The inlet air entering the room is accelerated downwards due to the acting gravity forces, and at a distance of approximately 1.0 m from the inlet

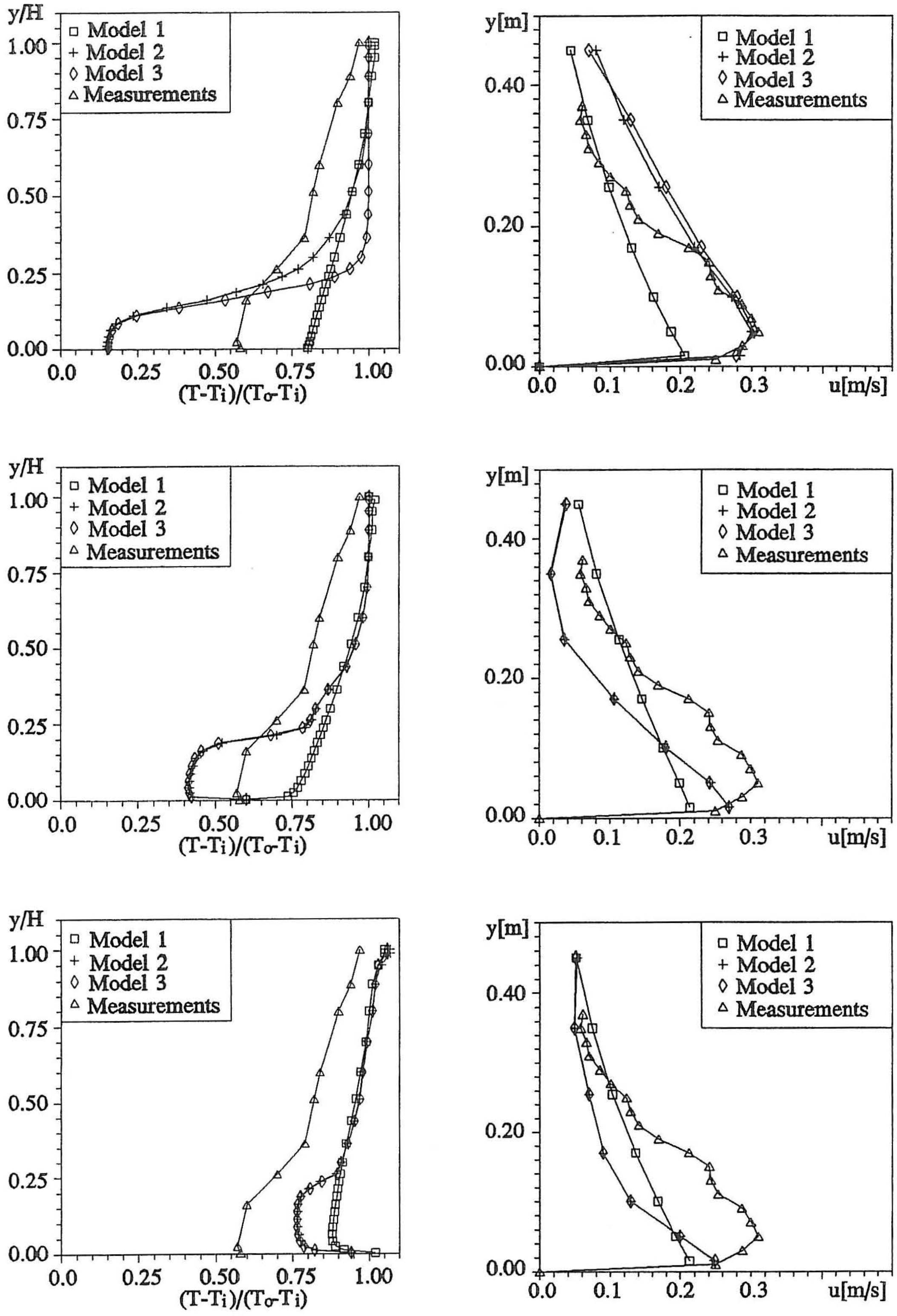


Figure 2. Temperature and velocity profiles for different turbulence models and boundary conditions. Model (1): eq. (2) + (3) without G_k-term. Model (2): eq. (2) + (3) with G_k-term. Model (3): eq. (2) + (3) + (4).

device self-similar velocity profiles arise. A closer discussion of the temperature and velocity field can be found in (*Jacobsen & Nielsen 1992*).

The uppermost profiles in fig. 2 correspond to a) boundary conditions mentioned above, the middle profiles to b) boundary conditions and the bottom profiles to c) boundary conditions.

DISCUSSION

The results presented in fig. 2 clearly illustrate the effect of including the buoyancy extensions in eq. (2) and eq. (3). The mixing between the cool supply air and the surrounding room air is decreased and the resulting temperature profile becomes strongly non-linear with a distinctly separated cool lower zone and a warm upper zone.

The damping functions of eq. (4) have only a minor influence except where extremely high temperature gradients occur. The interfacial mixing in the layer between cool and warm air is further reduced but the measurements indicate that the local temperature gradient in this layer is smaller than predicted.

Regarding the temperature boundary it is obviously essential to include the energy transfer through the walls and the radiative heat transfer from the upper part of the room to the lower part. The heat flux approach of boundary condition b) implicitly includes radiation because the measured temperature profile which is applied is a function of the radiative exchange between surfaces. Consequently the best agreement between modelled and measured temperature level should be obtained when the temperature profiles at the surfaces are prescribed by applying measured data. The problem is, however, that the performance of the temperature logarithmic law at the walls depends on the chosen grid.

The velocity profiles in fig. 2 show that the measured maximum velocities are generally higher than the computed maximum velocities. The observed discrepancy is of course related to the unsatisfactory accordance between measured and modelled temperature profiles. The local Archimedes number which is a key parameter differs from the measurements to the model. This means that the horizontal spreading of cool air close to the inlet device and the shape of the velocity profiles are not correctly modelled.
The model results presented here may be further improved by describing the boundary conditions better. In particular the inlet boundary conditions can be improved by applying measurements of the velocity and temperature field close to the inlet device.

A more detailed description of the thermal plume above the heat source may also contribute to a better prediction of the vertical temperature profile in the room. This will be the subject of future work.

Finally an example will demonstrate how the model results can be used in evaluation of indoor climate in terms of human comfort. The comfort can be quantified as percentage dissatisfied occupants (*Fanger, Melikov & Hanzawa 1988*).

$$PD = (34-T_a)(V-0.05)^{0.62}(3.14+0.37VT)$$

where PD, T_a, V, and I are percentage dissatisfied occupants, local air temperature, mean velocity and turbulent intensity, respectively. The turbulent intensity is defined as the fluctuating velocity component divided by the mean velocity. By using the computed values of turbulent kinetic energy I is calculated.

Fig. 3 clearly shows, what has also been the conclusion of experimental investigations, that the comfort conditions are critical close to the inlet device. The relatively high velocities combined with low air temperature constitute a problem and at the design stage it is valuable to be able to predict the human comfort consequences.

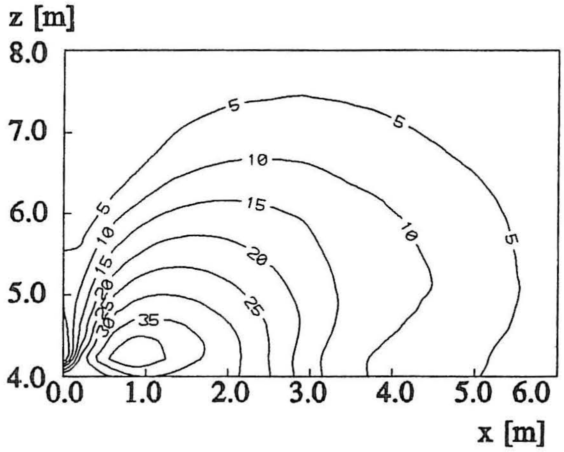


Figure 3. Example of a PD-map in a displacement-ventilated room. The iso-PD-curves are plotted at 0.05 m above floor level and model 2 is used in connection with boundary conditions b).

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