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PREDICTION OF AIR FLOW AND THERMAL COMFORT IN OFFICES

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ABSTRACT

A computer program for predicting the distributions of airflow patterns and thermal comfort indices is developed by incorporating Fanger's comfort equations in an existing computational fluid dynamics model. The mean radiant temperature for each grid point, required for the comfort equations, was calculated from the radiation heat exchange. The importance of taking account of radiosities for calculating the mean radiant temperature was illustrated by comparing the results obtained using the method developed with those obtained by other approximations used in practice.

1. INTRODUCTION

People expect the office environment to be as comfortable as feasible. Consequently, more and more offices are being equipped with mechanical ventilation or air-conditioning systems. The need for accurate prediction of the air flow in occupied spaces has become even more crucial because, with mechanical ventilation, large quantities of air supplied at a few locations have to be evenly distributed in the space being ventilated. Without adequate air distribution, excessive air movement (draught) can occur in some zones whereas stagnant air may be present in other zones of the same room. Poor air distribution can affect the indoor climate and degrade the air quality.

Until recently the air movement in a space was predicted from air jet diffusion data and/or testing a physical model as described by Awbi (1991). Computational fluid dynamics (CFD) is now being applied in the simulation of room air movement in spaces. In most of the available CFD programs, however, the radiant heat transfer is simplified or not accounted for when dealing with thermal comfort. Kaizuka and Iwamoto (1987) 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. Awbi and Gan (1991) and Gan, et al. (1991) have developed a CFD program in which the radiation heat exchange is taken into consideration and which is used to predict the air movement and thermal comfort in both mechanically and naturally ventilated buildings. In this paper the predictions of the air movement and thermal comfort in a mechanically ventilated office module and a naturally ventilated office are presented.

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

Equation (1) is solved for the 3-D cartesian system using the SIMPLE algorithm (Patankar, 1980). Wall function expressions for turbulent boundary layers are used to specify the value of ϕ at the first node point from a solid boundary.

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 (1982). 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 this work the air velocity and temperature are calculated from the flow equations. The mean radiant temperature, T_{mrt}, is calculated with the help of a radiation heat exchange model for the room. Other parameters, e.g. vapour pressure of air, are given a single value for the whole field.

2.2.1 Room surface temperature. When a heat source or sink is present in a room the heat transfer to the room air takes place by convection and to the room surfaces by radiation. It is then required to calculate the internal temperatures of room surfaces and use these as boundary values in the CFD prediction. The temperature T_i (K) of a surface i is obtained from the radiosity J_i of the surface using

$$J_i = \epsilon_i \sigma T_i^4 + (1-\epsilon_i) \sum_{j=1}^{n-1} F_{ij} J_j \tag{2}$$

where σ is Stefan-Boltzmann's constant; ϵ_i is the emissivity of surface i; F_{ij} is the shape factor for surface i with respect to room surface j ($i \neq j$); n is the number of room surfaces and J_j is the radiosity of surface j which is the total radiant energy leaving a surface (W/m^2). If a radiant heat flux q_i is present on surface i then the radiosity is calculated using the equation:

$$J_i = q_i + \sum_{j=1}^{n-1} F_{ij} J_j \tag{3}$$

The emissive power from a black surface i is given by:

$$E_{bi} = J_i + \frac{1 - \epsilon_i}{\epsilon_i} q_i \tag{4}$$

Hence the temperature of surface i is:

$$T_i = \sqrt[4]{(E_{bi}/\sigma)} \tag{5}$$

2.2.2 Mean radiant temperature. 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:

$$T_{prt}^4 = \frac{1}{\sigma} \sum_{i-1}^{n} F_{pi} J_i \tag{6}$$

where F_{pi} is the shape factor for radiation from face p of the grid cell to the visible room surface i.

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.

2.2.3 Predicted mean vote and predicted percentage of dissatisfied. The predicted mean vote, PMV, and predicted percentage of dissatisfied, PPD (%), are calculated using Fanger's equations which include the environmental and personal parameters (Fanger, 1982).

3. APPLICATIONS AND DISCUSSION

The theories of indoor airflow and thermal comfort described earlier are applied to the prediction of the indoor environment in a mechanically ventilated office module and a naturally ventilated office.

3.1 Mechanically ventilated office

In recent years the air conditioning of office buildings in the U.K. has become very popular. Most air conditioning systems for offices employ ceiling supply air terminals to utilise the ceiling surface for diffusing the air jet taking advantage of the Coanda effect. A slot diffuser is often used for supplying the air over the ceiling. A typical ceiling supply for a perimeter office module has, therefore, been selected to demonstrate the potential of CFD in room air movement prediction.

The office module, which was subjected to detailed measurements by Jackman (1973) in a laboratory mock-up, is 4.9m long, 3.7m wide and 2.75m ceiling height. It has a full width single-glazed window of height 1.5m. A continuous slot diffuser of slot width 20mm (7.2mm effective width) was installed on the ceiling 150mm from the curtain wall discharging air away from the window. In the simulation carried out, the heat loss from the cold window in winter is 747W. The supply air temperature is 5.5K higher than the room air temperature and the air supply rate is 112 l/s.

The predicted air movement and temperature distributions in the perimeter office are shown in Fig. 1 as a velocity vector and temperature contours (isotherms) respectively. The predicted velocity and temperature distributions have been found to be in good agreement with the measured values (Awbi and Gan, 1991).

The downdraught along the curtain wall caused by the cold window is clearly evident from the predicted velocity vectors and isotherms shown in Fig. 1. The flow is dominated by two triangular circulation zones; an upper zone controlled by the warm air jet and a lower zone of opposite circulation controlled by the downward buoyancy due to the cold window. As a result, the airflow pattern is not satisfactory. The resulting vertical temperature stratification can be observed in Fig. 1 and the thermal discomfort in the lower part of the occupied zone is evident from the large predicted percentage of dissatisfied (PPD) shown in Fig. 2.

To compare the PMV and PPD from the above method with other methods in which the mean radiant temperature is simplified, two more predictions were made. In one of these predictions the mean radiant temperature for all the space is taken to be the area-weighted mean surface temperature and in the other prediction the mean radiant temperature at any point is taken to be the same as the air temperature for the corresponding grid point. The predicted contours of PMV and PPD are shown in Figs. 3 and 4. It can be seen from figure 3 that when the mean radiant temperature is calculated from the mean surface temperature a more uniform distribution of the thermal environment is obtained than that when the radiation heat exchange is taken into account. However, when the mean radiant temperature is assumed the same as air temperature the prediction gives a warmer environment for the whole office than that given by the prediction using the radiation model because in this case the air temperature is higher than the mean radiant temperature due to the warm air supply. These comparisons show that it is very important to take into account the radiation heat exchange for the prediction of thermal comfort when the difference in surface temperatures is large.

3.2 Naturally ventilated office

Despite the increasing use of air conditioning in modern offices, natural ventilation, in combination with heating when required, is still the principal means of providing thermal comfort in most buildings in the U.K. The airflow in naturally ventilated buildings depends much on the arrangement of doors and

Awbi, H.B. and Gan, G. 1991. Computational fluid dynamics in ventilation. Proc. CFD Seminar for Environmental and Building Services Engineer. Institution of Mechanical Engineer, London, 26 November, 1991, pp.67-79.

Croome, D.J., Gan, G. and Awbi, H.B. 1992. Air flow and thermal comfort in naturally ventilated offices. Roomvent '92. Aalborg, Denmark, September 1992.

Fanger , P.O. 1982. Thermal Comfort -- Analysis and Applications in Environmental Engineering. Robert E. Krieger Publishing Company, Florida.

Gan, G., Awbi, H.B. and Croome, D.J. 1991. Airflow and thermal comfort in naturally ventilated classrooms. Proc. 12th AIVC Conference. Ottawa, Canada, September 1991.

Jackman, P.J. 1973. Air movement in rooms with ceiling-mounted diffusers. HVRA (BSRIA) Laboratory Report No. 81. Bracknell, UK.

Kaizuka, M. and Iwamoto, S. 1987. A numerical calculation on the distribution of surface temperature and thermal comfort index caused by radiation interaction in a heated room. Trans. ASHRAE 33, pp.103-113.

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

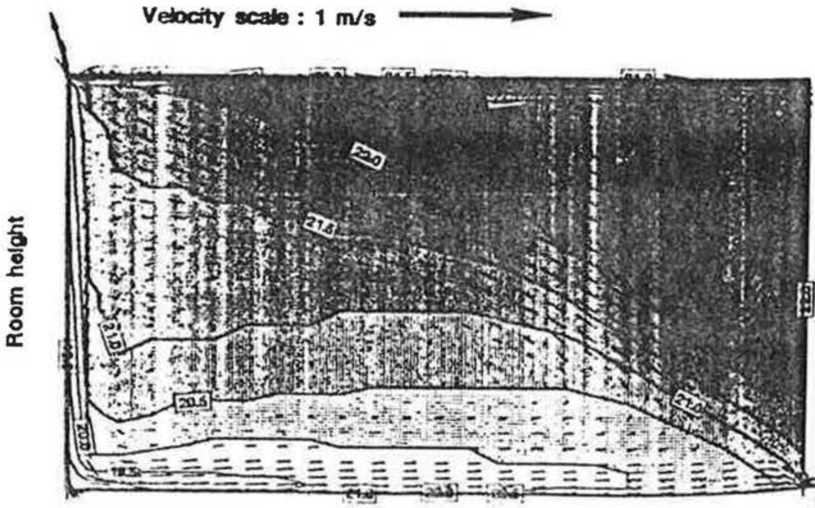
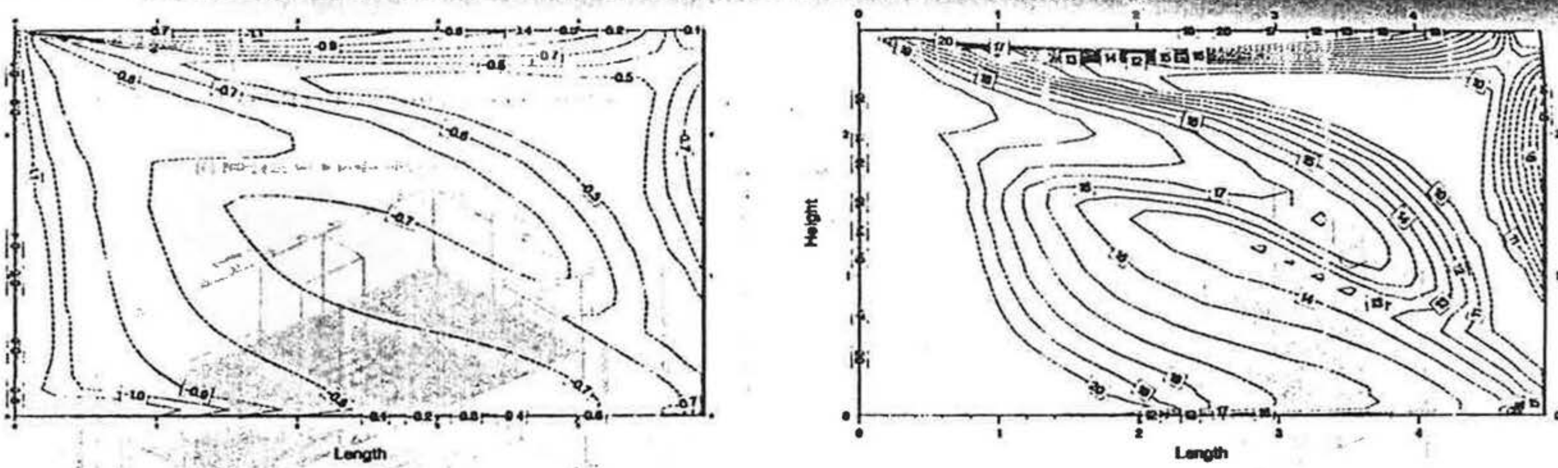


Fig. 1 Predicted velocity vectors and isotherms in the perimeter office



(a) Predicted mean vote

(a) Predicted percentage of dissatisfied (%)

Fig. 2 Predicted mean vote & predicted percentage of dissatisfied in the perimeter office with T_{mrt} calculated from radiosity

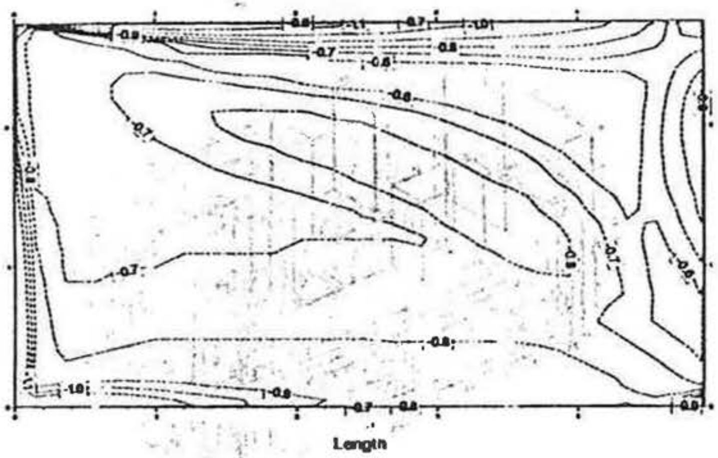


Fig. 3 Predicted mean vote in the perimeter office, taking T_{mrt} to be the mean surface temperature

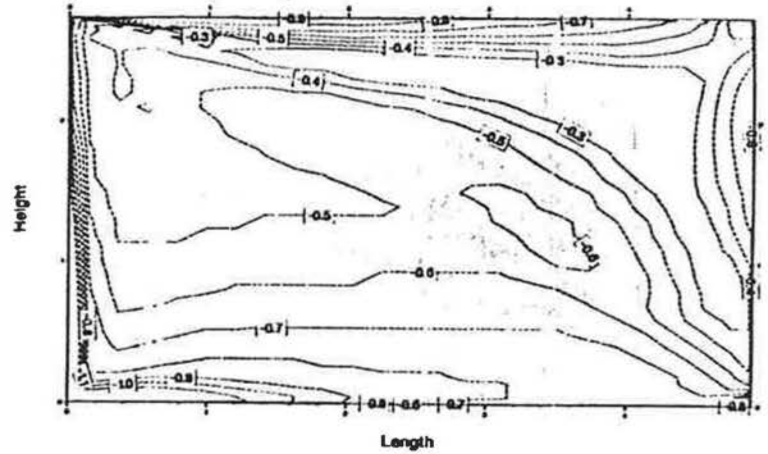


Fig. 4 Predicted mean vote in the perimeter office, taking T_{mrt} to be the air temperature