VELOCITY AND TEMPERATURE PREDICTION IN AN OFFICE ROOM WITH FURNITURE

P. Stankov[1], J. Denev[1] and S. Stamov[2]

[1] Department of Hydroaerodynamics and Hydraulic Machines, Technical University of Sofia, 8 Kliment Ohridski St., Sofia 1000, Bulgaria
[2] Department of Heat and Mass Transfer, Technical University of Sofia, 8 Kliment Ohridski St., Sofia 1000, Bulgaria

ABSTRACT

The paper presents results from the numerical modelling of the flow field in an existing ventilated office room. The numerical procedure is based on the 3D Reynolds equations closed by the k-ε turbulence model and an equation for temperature solved by the finite volume method. The boundary conditions are set in accordance to detailed measurements of the velocity distribution in the air supply diffuser. The established complex flow conditions in the room, which are due to the presence of furniture and buoyancy forces, are presented.

Field measurements were carried out for the particular investigation. An analysis is made of the temperature field based on a comparison between the experimental and numerical results. Conclusions are drawn about the organization of ventilation in the investigated room.

INTRODUCTION

The paper presents results from the comparison between numerical simulation and field experiments in a real office room with furniture. One purpose is the verification of the numerical results. However, when the comparison is discussed, many stages of the combined numerical-experimental investigation have to be addressed. Therefore, another purpose of the paper is to give details of those stages which presented the main difficulties in the investigation and to discuss how they influenced the accuracy of the results. These details consist of the following:

- demonstration of the complexity of the measured velocity field in ceiling-mounted air diffusers as well as of the subsequent procedure of feeding these data to the simulation;
- visualization of the complex flow structure in the room as a consequence of the velocity distribution in such diffusers;
- discussion of the difficulties in determining correct heat fluxes of the surrounding walls from measured data.

THE ROOM AND THE VENTILATION SYSTEM

The real office room under study is situated in an administrative building in the town of Plovdiv, Bulgaria. Its dimensions are: length - 4.35 m, width - 3.3 m and height - 3.2 m. The computer model of the radiator, the window and the furniture is shown in Fig. 1.

A central multizone low-velocity air-conditioning system has been envisaged to maintain the parameters of the indoor climate in the building. The air exchange in the room is effected according to the up-down scheme. The processed air enters the room through the ceiling air-supply diffuser having a square shape and dimensions of 300/300 mm (for details see Fig. 2). The design flowrate is 150 m^3/h with variable air temperature. The air goes out of the room through a grill installed on the entrance door. Its dimensions are 400/200 mm.

EXPERIMENTAL MEASUREMENTS

Detailed measurements of the temperature field in the room are performed under winter operating conditions. A multi-functional measuring instrument TESTO 454 is used. It has a thermistor with accuracy of $\pm 0.2^oC$ within the temperature range $-20^oC \div +50^oC$ and a resolution of 0.1 ^oC. Measurements at 100 points in the room were made. The points were distributed evenly in 5 vertical and 4 horizontal planes.

For the needs of the computer simulation the surface temperature of the radiator, the outer wall and the window is measured at different points. For the same purpose, the velocity magnitude of the air-supply diffuser is measured at several points.

NUMERICAL METHOD

The numerical method used is based on the three-dimensional Reynolds-averaged Navier-Stokes equations closed by the standard version of the k-ε turbulence model. The discretisation is made by the finite volumes method on a collocated numerical grid. The SIMPLE-algorithm was utilized for coupling the velocity and pressure fields. The numerical code FASTEST/3D, developed at LSTM (University of Erlangen-Nuernberg) and adapted by the authors for the purposes of room ventilation, was used. More details about the method and its algorithmic details can be found, e.g. in Ferziger and Peric (1996). The rather complex geometry of the room was covered by a numerical grid consisting of 53 x 40 x 46 = 97 520 numerical points. Fig. 1 gives details of the numerical grid mapped upon the obstacles in the room (i.e. the furniture) and the window.

NUMERICAL INVESTIGATION

Real-life office rooms present a challenge for numerical simulations. At first sight, it seems that the major problems are related to the complex geometry of such rooms. However, this is not valid for the modern computer codes (including the present one) which are capable of handling any kind of irregular geometry. What the reader should have in mind is that the real difficulty arises in the process of imposing the boundary conditions for the momentum and temperature equations. As is shown below, these difficulties are basic for such combined experimental-numerical investigations and hence do not depend on the particular software code used. The present investigation gives a very suitable example of typical problems arising when numerical simulations follow field measurements for their verification together with the corresponding examples for a (reasonable) solution of such problems.

Boundary conditions. The centre of the ceiling-mounted square diffuser is located at x = 2.10m, y = 2.35m (the coordinate system is shown in Fig. 1). Fig.2 gives details of its geometry; the lamellas are 1.5 mm thick. The previous experience of the authors has shown that the air velocity at different symmetrical points of similar diffusers may vary considerably, the differences reaching up to 50% ! Such a velocity distribution deviates largely from the idealized flow conditions usually assumed in numerical computations. Since the velocity distribution in the diffuser can have a strong influence on the overall flow structure and temperature fields in the room, a detailed experimental investigation of the air velocity distribution in the diffuser was made. The measured values of the velocity magnitude in the diffuser are also presented in Fig.2.

The boundary conditions for the simulation were prepared by means of interpolation of the measured values and by additional assumptions about the direction of the velocity vector. The interpolation was necessary due to the fact that the control volumes positioned in the air-supply device do not coincide in position and number with the locations where the measurements were made. As the values in Fig.2 are measured with an omnidirectional probe, no information about the direction of the velocity vector was obtained. Therefore, with the view of setting properly the velocity components in the inlet, the velocity vector at each point was assumed to pass through the "centre" of the diffuser. The direction of the supply air is set (assumed) to be at an angle of 30° from the ceiling.

The temperature (32.1 °C) measured in the diffuser was used as an inflow boundary condition. For reasons explained below, the measured values of the surface temperature of the outer wall and the window were not directly used in the simulation. Instead, the measured values were averaged so that one surface temperature was obtained for the entire wall and one for the window. From these average values and the measured average room temperature, the heat flux was estimated utilizing equations for the assessment of the heat flux [e.g. Stamov 1990]. The other walls of the room are assumed to be adiabatic. Heat flux boundary conditions were also used for the radiator.

RESULTS AND ANALYSIS

The flow structure in the room. Fig. 3 presents the velocity vectors in a plane below the ceiling and in two other vertical planes. The horizontal plane below the ceiling exhibits many recirculation zones which, according to the non-symmetry of the experimental data, create a non-symmetric flow near the ceiling. The figure shows how the air coming from the ceiling diffuser is redirected down the vertical wall with the window and accelerated due to the cold downdraught. The complexity of the three-dimensional flow becomes also obvious from the tracelines plotted in the figure.

The temperature distribution. The numerical results on the temperature distribution along vertical lines in the room are plotted in Fig. 4. The same figure presents for comparison the experimental data from the measurements. The figure shows that the calculated temperature level is several degrees (between 1 and 3 °C) higher than the measured one. The latter is due to the use of the method described above for assessing the heat flux through the outer wall and the window, which because of its nature, leads to a relatively low accuracy. However, this method gives two important advantages when compared to the Dirichlet-type boundary conditions for the temperature equation. Firstly, it allows to impose such boundary conditions for the temperature equation that will satisfy integrally the room heat balance. This corresponds to the conservative nature of the control volumes method used. Secondly, which is even more important, it allows to avoid the use of either the wall-functions type of boundary conditions (where the calculated heat flux depends strongly on the numerical grid and is therefore of very low accuracy) or their alternatives like the low-Re number turbulence models (which for such investigations would drastically increase the computer resources and decrease their applicability in practice).

Despite the difference in the temperature level, a good correlation is observed between the experiment and the simulation in view of the vertical temperature gradient in the room. Both the numerical results and the experimental data show a stable stratified temperature field resulting from the fresh air supplied from the ceiling at higher temperature.

As regards the temperature field, it can be concluded that the level of the temperature in the occupation zone is much higher than the level at which comfort is achieved [see, e.g. Awbi 1991 and EN ISO 7730-1995 Standard]. Therefore, a decreased temperature in the ceiling diffuser is recommended on the basis of the numerical results obtained. The results from a subsequent numerical simulation, where the temperature in the ceiling diffuser is decreased from 32.1 °C to 27.0 °C, show that satisfaction of the room comfort conditions has been achieved according to the EN ISO 7730-1995 Standard.

CONCLUSIONS

The paper presents a combined numerical-experimental investigation of the complex airflow and temperature field in a real office room with furniture. All principal difficulties connected with the combined investigation of complex real-life flows are demonstrated and discussed. The difficulties are shown to be related mainly to different kinds of boundary conditions. To obtain their values for the simulation, it was proved that we often need to rearrange, interpret and assess appropriately the available experimental data.

A comparison of the numerical and experimental data on the real office room has been made. The comparison shows a good agreement between the numerical and experimental vertical temperature gradients. It also shows slight differences in the measured and simulated temperature levels: the numerically obtained temperatures are somewhat higher (1 to 3 °C) than the measured ones.

The analysis of the temperature field shows that people would feel the environment unacceptably warm. Based on the numerical results, actions for decreasing the temperature are prescribed. A subsequent numerical investigation was conducted which proved that the prescribed actions would lead to a comfortable office environment.

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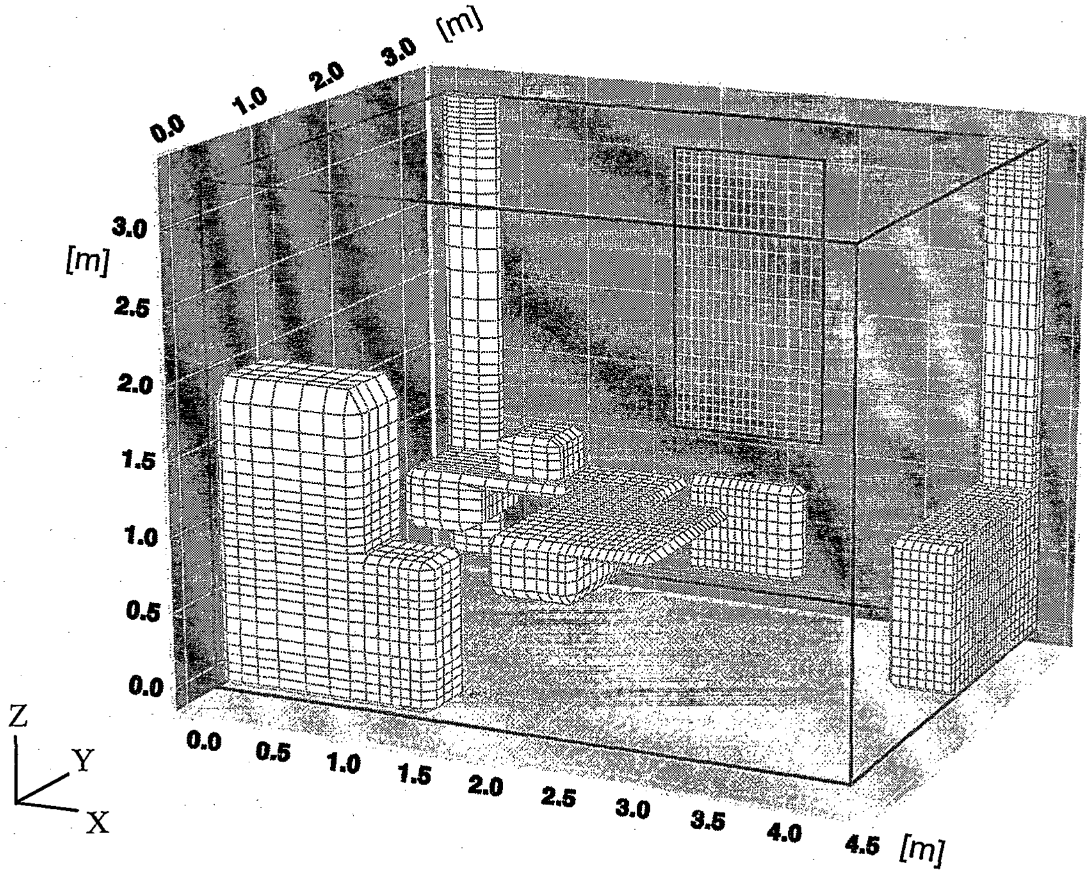

Fig. 1. The layout of the simulated room geometry:
the window, the radiator, the two pillars and the furniture

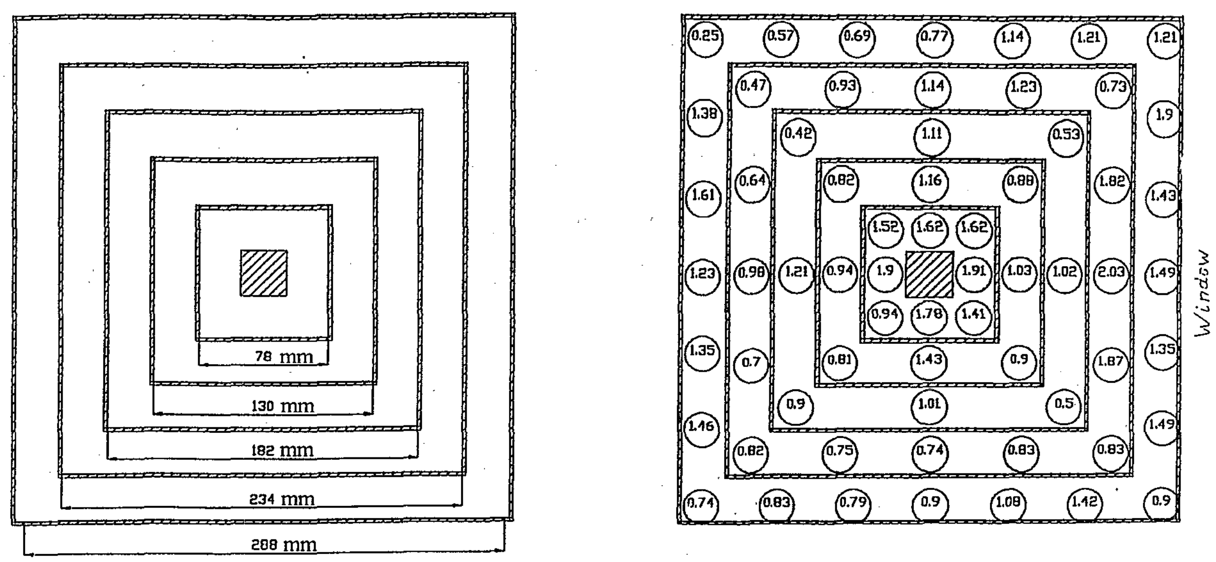

Fig.2 The geometry of the ceiling mounted diffuser and the measured velocities (in m/s) at different locations in it

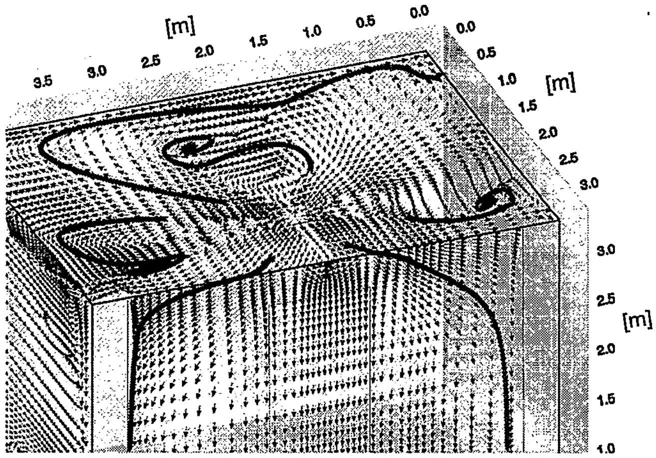

Fig.3. Velocity vectors in three planes of the room and streamtraces, originating in the vicinity of the diffuser

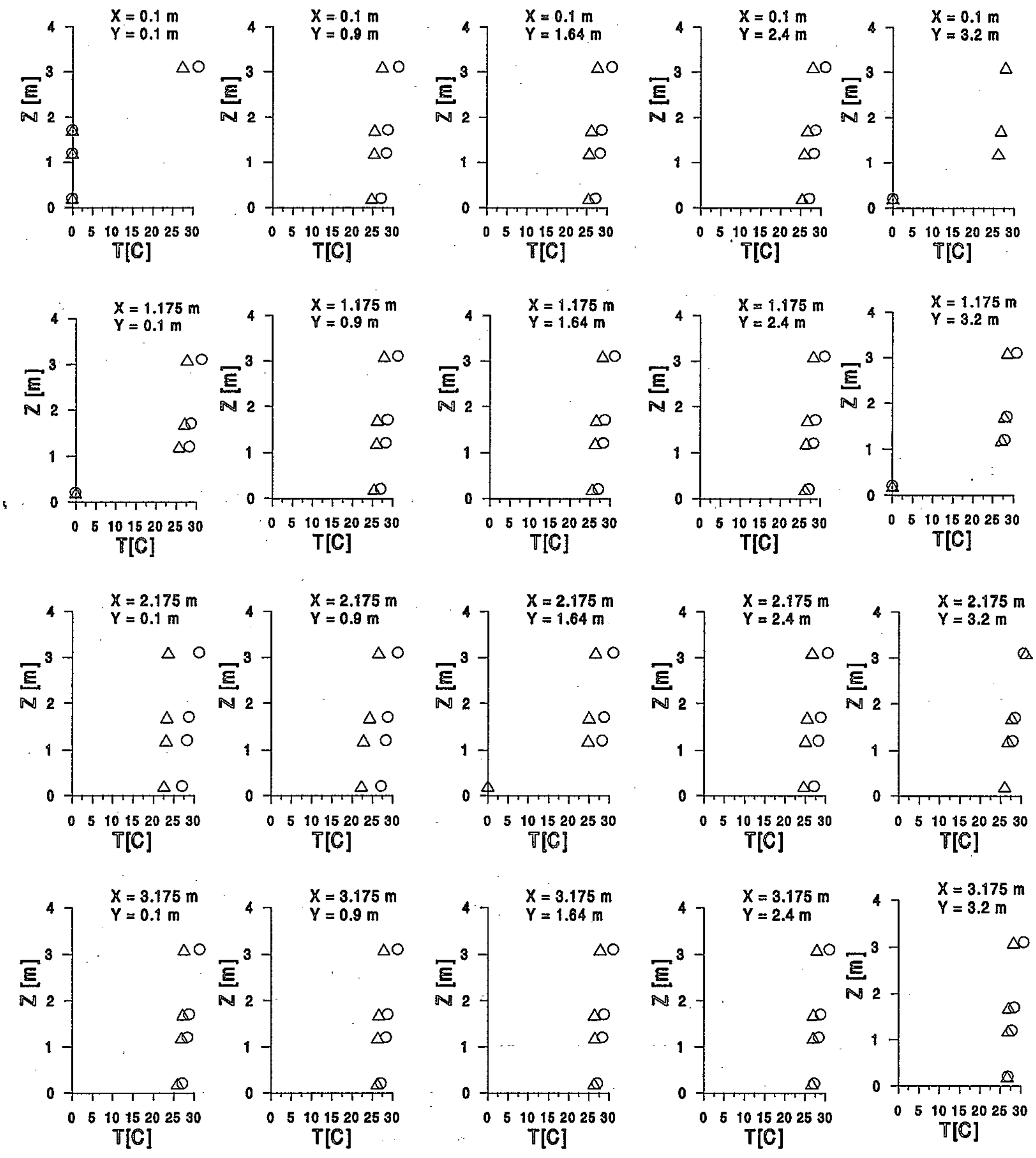

Fig.4. Temperature distribution along several vertical lines in the room:
Δ - experiments; o - present simulation