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The stratification study of public buildings microclimate parameters with supply and exhaust ventilation

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Abstract. The paper presents the results of the UFAD system numerical modeling where the air is supplied through diffusers installed in a floor of a room. Warm air is removed from the upper zone of the room through the exhaust duct. The purpose of this study is to determine the magnitude of the temperature stratification in various areas of an office premise in the conditions of the under floor air distribution. Enclosed volume with local heat sources has been chosen as the computational domain: people, computers, and lighting. The problem is solved in a stationary setting using the k-omega SST turbulence model. As a result of the study the temperature distribution throughout the room has been obtained and the influence dependence of wall temperature on an average air temperature in the room has been defined. The numerical data received are in good agreement with the results of the borrowed experimental study that demonstrate the adequacy of the developed numerical model as well as the possibility of its further use for the optimization of the considered ventilation systems.

1 Introduction

Nowadays under floor air distribution (UFAD) is not widely used in residential and office buildings. However, such systems provide effective removal of contaminants from the ventilated space, especially in working areas [1]. The UFAD system provides ventilation design with minimum air flow in comparison to conventional ventilation systems [1, 2]. These systems are rather difficult to implement in converted premises [3, 4], but it increases the cost of newly constructed buildings and structures insignificantly [3]. In UFAD systems the air supply is made from the ventilation ducts under the floor. Polluted air is removed from the upper zone of the room through the exhaust channel [2]. It is more effective to use the UFAD system in summer when the heated air is removed from the upper zone, and the cooled air is fed directly into the working zone. Since the UFAD system implies zoning of

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the ventilated room with a clear separation of working areas, they are the most effective in office premises, classrooms and workshops. [4] The aim of this work was to develop a numerical model that can adequately describe the temperature stratification in the enclosed space under conditions of forced convection in the presence of local heat sources.

2 Materials and methods

To study the effectiveness of the ventilation systems we use experimental measurements and computer modeling. The experimental data provide the most reliable information about the measured system parameters: humidity, pressure, temperature, the concentration of pollutants, etc. On the basis of these data, we determine the performance of the ventilation system as a whole and its individual components. However, the experimental studies are associated with a number of factors that make it difficult to obtain reliable experimental data and their interpretation.

Thus, in some cases, it is difficult or impossible to measure certain parameters of the monitored volume. For example, a reliable determination of air flow at speeds less than 0.1 m/s is not a trivial task even with the use of modern measuring equipment [5]. It is also complicated to measure simultaneously a number of environmental parameters in the entire volume because it requires synchronous operation of a large array of sensors. Multiple reproductions of experimental conditions can also cause a number of difficulties, in particular taking into account the external weather conditions. Thus, experimental studies involve significant financial and time costs.

There is a lack of such disadvantages in numerical modeling and in most cases, it is an effective addition to designing ventilation systems [6]. The primary experimental research is an integral part of the process since the model developed must undergo verification by comparison with experimental data [7]. Numerical simulation makes it possible to develop new projects based on existing systems, opening new perspectives for research.

Free software Salome [8, 9] and Code Saturne [9, 10] were used in the work. The packages are designed to generate computational grids, develop numerical models and process the results. Code Saturne is mainly applicable for solving the problems of the continuum hydrodynamics, and it has the extended functionality in relation to convective transfer under subsonic flows.

In this paper, we used the experimental research of UFAD systems [2]. An experimental study was conducted in a confined space, simulating full-size office space (fig. 1). The room is 2.43 m height, 4.2 m length, and 2.4 m width. People (H) and computers (E) were reproduced using boxes with the built-in heat source (incandescent lamp). The room was equipped with linear fluorescent lamps (L, 1550x140 mm). Tables (T 1000x600 mm) and closet (F, 1200x400x1200 mm) were used as furniture. The cooled air supply was carried out through diffusers (D, 600x600 mm) placed on the floor. The air was removed from the room through the exhaust port (R, 300x300 mm) on the ceiling.

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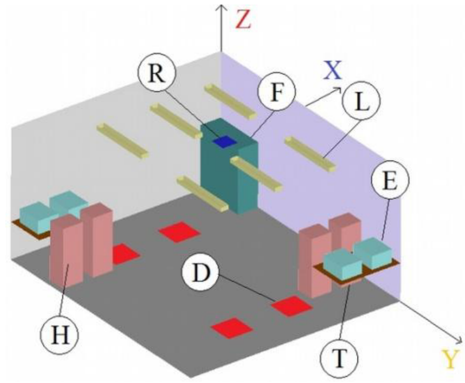

Fig. 1. Office space layout [2].

We were solving the problem in a fixed setting, which helped to stabilize the solution in vortex flow areas (above diffusers) and to reduce the computational load. When constructing the geometry of the computational domain, some simplification in relation to desks and computers was made. This has greatly simplified computational grid without losing the essential characteristics of the designed room. Also, the symmetry condition was applied, which allowed dividing the room in half parallel to the plane XZ.

When modeling the unstructured computational grid consisting of 0.2 million pyramidal cells were used (fig. 2). The grid was constructed according to the NETGEN algorithm [11] in the Salome software package [8]. The maximum size of cells in the bulk of the estimated area reached 0.15 m at a rate of 0.1 cell growth. In addition, the grid was reduced to a maximum of 0.03 m on the heat transfer surfaces (mannequins, computers, lamps). The relative displacement of the cell centers was less than 2.14 (0.21 medians, 0.01 modes). The absolute non-orthogonality of cells was less than 72^0 (21.7^0 median, 25^0 mode) on surfaces and less than 78^0 (9^0 medians, 1.2^0) in volume. The proportions of the vast majority of the estimated elements were in the range that satisfies the tolerance of a solver [12]: up to 1.5 on the surfaces and up to 2 in the volume (fig 2.). This indicates a satisfactory quality of the computational grid taking into account the small number of elements.

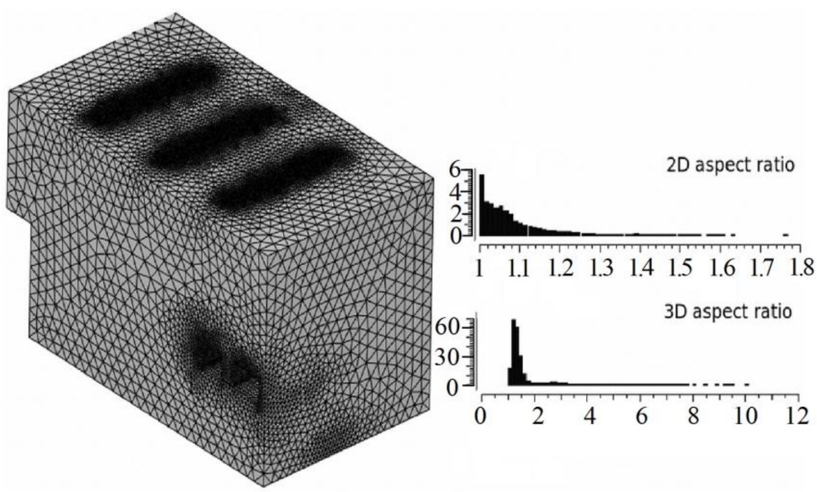

Fig. 2. The computational grid

As an initial condition, we adopted the temperature in the simulated room before the start of the experiment (T_0 = 26 ^{0}C). The boundary conditions for the diffusers surfaces were determined from the assumption about the uniform distribution of velocity fields (U_0= 0.1 m/s normally, the temperature is 19 ^{0}C). Turbulence intensity (I = 2%) was determined by the hydraulic diameter diffusers (d_h = 0.6 m). The boundary conditions for the heat transfer surfaces are fixed with values of heat fluxes: for mannequins (96.15 W/m^2), for computers (261.44 W/m^2) and lamps (30 W/m^2). Due to the absence of significant temperature gradients in the considered volume, the constant thermodynamic properties of the medium (air) were set at an initial temperature (T_0 = 26 ^{0}C). In order to account the effect of natural convection on the distribution of temperature fields in the room, the air density temperature dependence was set according to the ideal gas law [13].

Based on the experimental data the researchers [2] determined the coefficient of the enclosing structure of the premises thermal resistance which amounted to 0.45 kW/m^2. This coefficient was used to set boundary conditions in one of the calculations (see Fig. 3). In other cases, the enclosing structure was adiabatic.

Since the room has a turbulent air flow (Re > 2500) near the diffusers and it is necessary to reproduce the surface heat transfer, the turbulence k-omega SST model was used in the formulation of the first order [14, 15]. This model is an industry standard of numerical simulation and it allows obtaining a stable solution for free flows in a volume as well as for flows in a wall viscous layer. Taking into account the relative cell size of the computational grid, the vortex models (LES) were not considered.

Considering the satisfactory quality of the grid, gradients were calculated with an iterative method based on the non-orthogonalities. Due to the low possibility that abrupt pressure differential would occur, we didn't use its correction by combining momentum conservation equation and continuity equation. To stabilize the solution the gradient and divergence accounting algorithm was activated when solving momentum conservation equation. The solver relaxation in pressure was not used (R = 1). The stratification pressure was not considered since there are no high-temperature heat sources and high velocity flows in the computational domain. Pseudo-stationary formulation of the problem made it possible to use a modified semi-implicit method for solving the Navier-Stokes equations (SIMPLEC) [16, 17], with the increase of the Courant's limit number (Cr_{max} = 5).

Threshold values of accuracy (10^{-6}) and sub iteration number (10000) were introduced when solving linear equations to reduce the computational load.

Momentum and energy equations were solved by the sampling technique of the second order (SOLU) without degradation. Turbulence equations (k, omega) were solved by the first-order sampling technique (Upwind). Since the nonorthogonal grid had been used in the calculation, the balance adjustment was carried out. To stabilize the solution of the energy equation the range of acceptable temperatures was limited: 19... 26.7 ^{0}C.

The duration of the numerical experiment was determined based on the air exchange rate and was amounted to $\tau = 50$ s. The time step was set dynamically from conditions ensuring the Courant and Fourier numbers maxima ($Cr < 5$, $F < 1000$), however, the time step dynamic decrease was not required. The total number of iterations was 500 in both cases: with adiabatic walls and fixed thermal resistance of the enclosing structure.

3 Results

Due to the purpose of the work, the evaluation of the adequacy of the developed numerical model was carried out by comparing the results of the experimental study. The evaluation was conducted for two cases of boundary conditions formulations for heat exchange with enclosing structures (fig 3.): θ_1 – when there is no heat exchange with the walls, θ_2 – for the fixed heat transfer coefficients from the enclosing structures.

The dimensionless temperature (θ) was determined by the actual temperature at the point (T, ^{0}C) and was dependent on the supply (T_s = 19 ^{0}C) and exhaust (T_e = 26.7 ^{0}C) air temperature:

$$\theta = (T - T_s)/(T_e - T_s) \quad (1)$$

Simulation results (fig. 3) are presented in a form of medium temperature distribution in a longitudinal section throughout the height of the computational domain. The cutting plane position (parallel to the XZ plane) was determined in accordance with field measurement conditions [2] and it coincided with the symmetry plane used in the model. The average of the actual temperature values was carried out taking into account the relative size of computational grid cells located on the surface of symmetry plane.

The results obtained after using the developed model (fig. 3) show that the reproduction of the caloric temperature at the height distribution is satisfactory.

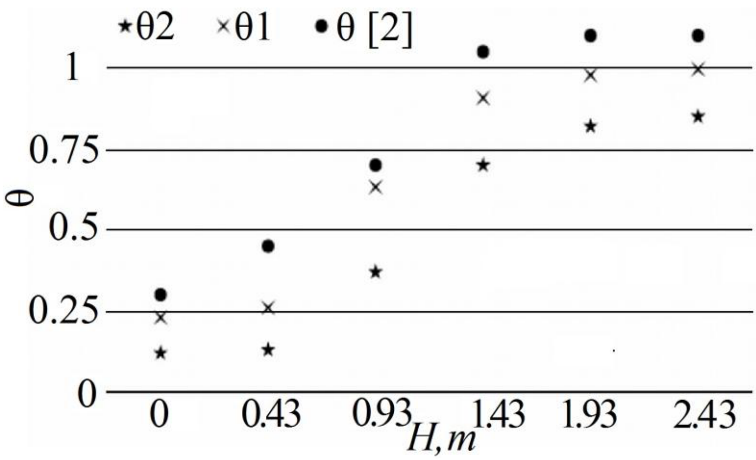

Fig. 3. The numerical modeling results.

4 Discussion

A slight deviation was detected at a level of 0.5 m which was probably caused by the idealized reproduction of honeycomb diffusers. Also, this deviation may be caused by the reverse currents of conditioned air from the mannequins the shape of which was somewhat simplified.

The introduction of heat transfer condition from the enclosing structures has not given the desired result and the obtained temperature distribution in the control section reproduces the experimental results much worse. The relative temperature reduction (θ) is a direct consequence of the average temperature decrease in this section ($\theta \sim T$), which was probably caused by the improper reproduction of the experimental conditions.

Conclusions

The obtained results indicate on achieving the goal to develop the numerical model that adequately describes the temperature stratification in the enclosed space in conditions of the forced convection with local heat sources. The adequacy of the model is confirmed in the working environment temperature range from 19 ... 27 0C in the presence of the medium flow (Re < 3000). The model takes into account the influence of the localized low-grade heat sources (q < 262 W/m2), and also the influence of the gravitational component on the process of natural convection.

The results do not allow us to analyze the effect of boundary conditions on the solution accuracy within the framework of the developed model. At present we have only defined the qualitative influence of heat transfer conditions with the enclosing structures on the distribution of temperature fields in the volume. Further studies will allow us to summarize the efficiency dependence of the UFAD ventilation system from the enclosing structure parameters.

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