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The role of computational fluid dynamics in the building energy and environmental analysis

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Abstract

The goal of this paper is to introduce a Computational Fluid Dynamics model to study the steady three-dimensional turbulent airflow pattern, with thermal buoyant effects, the heat transfer, the gas contaminant transport and the moist air transport into mechanical or natural ventilated spaces, with furniture and people. The calculations made use of a numerical procedure with solves the three-dimensional steady state equations for the conservation of mass, momentum, thermal energy and air contaminant transport, tanking account of the effects of buoyancy and airflow turbulence. In order to demonstrate its applicability, the code has been used to predict the thermal, the airflow pattern and the environment in an office ventilated room. Based on these predictions and on thermal comfort models it was possible to evaluate the thermal comfort.

INTRODUCTION

An accurate understanding of indoor air motion is crucial to the design of building heating, ventilating and air-conditioning (HVAC) systems in providing thermal comfort and indoor air quality, as well as in increasing the energy efficiency of mechanical and electrical systems.

The purpose of an air distribution system in a ventilated room is to supply fresh air, remove heat load or supply heating and create a pleasant and uniform climate in the occupied zone. In this context, a pleasant climate is defined as a fairly low air velocity, small velocity and temperature gradients throughout the room, and a low concentration of pollutants, if any.

Contaminants are essentially controlled by the way that supply air is circulated within the occupied zone. The pattern of air circulation is in turn influenced by the location of air inlets and outlets, windows, room geometry, and interior furnishings, as well as by the heating and cooling systems of the building.

Measurements of room air motion, distribution of temperature and gaseous components are very time consuming and expensive, requiring sophisticated sensors and instrumentation techniques.

With the increasing computer power and the fluid dynamics research, several methods have been developed based on numerical calculation of fundamental field equations for the analysis of fluid phenomena (CFD – Computational Fluid Dynamics).

Since the first application to room air distribution in 1973 [1], there has been great interest in developing CFD computer codes for predicting the airflow in ventilated rooms (see [2]; [3]; [4]; [5]; [6]; [7]). The majority of these CFD programs are based on the solution of Navier-Stokes equations, the energy equation, the mass and concentration equations as well as the transport equations for turbulent velocity and its scale.

The numerical solution of these equations in two an three dimensions has been applied to flow problems ranging from the diffusion of jets to the prediction of smoke and fire spread in buildings.

NOMENCLATURE

C_p = specific heat of air (J/KgK)
g_i = gravitational acceleration in x_i direction (m/s^2)
k = turbulent kinetic energy (J/Kg)
P = pressure
q = volumetric heat generation rate (W/m^3)
U_i = mean velocity component in x_i direction (m/s)
x_i = cartesian coordinate (m)
T = local air temperature (C)
v = mean air velocity (m/s)
v′ = air velocity fluctuation (m/s)
I = turbulence intensity (%)
PMV = Predicted Mean Vote
PPD = Predicted Percentage of Dissatisfied (%)
DR = draught risk (%)
Γ = diffusivity (m^2/s)
ϕ = variable
α = thermal diffusivity (m^2/s)
β = coefficient of thermal expansion (1/k)
ε = dissipation rate of turbulent kinetic energy (J/Kgs)
ν = kinematic viscosity (m^2/s)
ν_t = eddy viscosity (m^2/s)
ρ = air density
σ = Prandtl number
$C_\mu, C_1, C_2, \sigma_k, \sigma_\varepsilon$ = empirical constants of the k-ε model

GOVERNING EQUATIONS

To simulate the turbulent incompressible flow the well-known k-ε two-equation turbulence model [8] is used. Room airflow simulation based on the k- ε model was first conducted by Nielsen [1] and has been widely applied in predicting many types of turbulent flow phenomena with sufficient [4]; [9].

The governing equations for the averaged steady flow can be written in the general elliptic form for an incompressible fluid as

$$\frac{\partial (U_j \phi)}{\partial x_j} - \partial\left(\Gamma_\phi \frac{\partial \phi}{\partial x_j}\right) / \partial x_j = S_\phi \qquad (1)$$

Where the parameters ϕ, Γ_ϕ and S_ϕ are identified for each equation in Table 1.

For the empirical constants of the k-ε model (C_μ, C_1, C_2, σ_k, σ_ε) were assigned the standard value proposed by Launder and Spalding [8]. For the Prandtl number σ_ϕ a value of 0,9 was used.

The finite difference form of the three-dimensional governing equations is obtained using a finite volume method. The pressure field (which does not have an explicit differential equation to be discretized) was iteratively treated with the SIMPLE algorithm [10].

Table 1. Values of ϕ, Γ_ϕ and S_ϕ associated with Equation 1

ϕ	Γ_ϕ	S_ϕ
1	0	0 (continuity)
U_i	$\nu + \nu_t$	$\left(-\frac{1}{\rho}\right)\frac{\partial P}{\partial x_i} - \beta g_i \theta$
θ	$\nu + \nu_t / \sigma_\theta$	$q / \rho c_p$
k	$\nu + \nu_t / \sigma_k$	$\nu_t + G - \varepsilon$
ε	$\nu + \nu_t / \sigma_\varepsilon$	$(C_1 \nu_t S - C_2\ \varepsilon + C_3 G)\ \varepsilon / k$

with:

$$\nu_t = C_\mu k^2 / \varepsilon$$

$$S = \left(\frac{\partial U_i}{\partial x_j} - \frac{\partial U_j}{\partial x_i}\right)\frac{\partial U_i}{\partial x_j}$$

$$G = \beta g_i \frac{\partial \theta}{\partial x_i} \nu_t / \sigma_\theta$$

A hybrid (upwind/central). differencing scheme is used for the integration.

OUTLINE OF CALCULATION PROCEDURE

The room model used in this study is shown in Figure 1. It is a typical 18 m^2 office room ventilated by mixing flow. The room sizes are 3 m * 6 m * 3,5 m, the walls heat transmission coefficients are 2 W m^{-2} C^{-1} and the floor and the ceiling are adopted adiabatic.

Table 2. Room operating conditions

Supply air velocity	2 m/s
Inlet air temperature	14 C
Volumetric flow	28 l/s
Heat source	340 W
Inlet area	0,014 m^2
Outlet area	0,16 m^2

The office worker is operating a computer terminal . The room operating conditions are shown in Table 2.

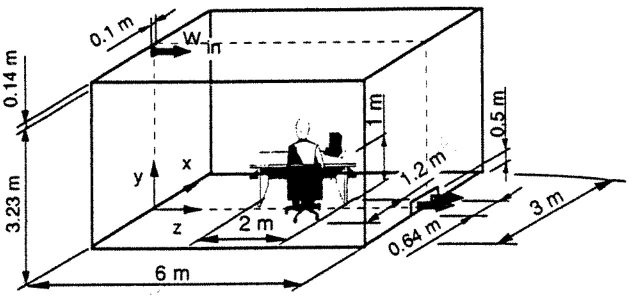

Figure 1. Office room model

The thermal effect of the occupant and the computer are lumped together, with 340 W convective heat.

In order to simulate the furniture aerodynamic blockage, a 2 m * 1,2 m * 1 m obstacle was introduced at the centre of the room.

All variables are given at the supply inlet, including the turbulence quantities. The air velocity at the exit is obtained from the continuity equation and the air temperature from energy balance equation.

Since the k-ε model is applicable only for high Reynolds number flows, the "wall-functions" equations [8] are used for the calculations of the velocity parallel to the boundary and the heat flux through the wall.

The velocity components perpendicular to the boundary are taken to be zero. The treatment of the wall boundary is also applied to the obstacle surfaces in the room.

MODELS FOR THE ANALYSIS OF THERMAL COMFORT IN ROOMS

Thermal comfort is defined as "that condition of mind in which satisfaction is expressed with the thermal environment". Therefore, both thermal environment and personal variables influence thermal response and comfort.

The physical quantities that affect the thermal comfort of the human body are air velocity, air temperature, humidity, mean radiant temperature, metabolic rate of the body, and thermal resistance of clothing. These values are combined to express the level of thermal sensation in various thermal comfort indices.

ISO 7730 [11] standardizes the PMV-"Predicted Mean Vote" and PPD-"Predicted Percentage of Dissatisfied" index [12] as the method for evaluation of moderate thermal environments. To quantify the degree of comfort, the PMV index gives a value on the seven-point ASHRAE thermal sensation scale: +3 = hot, +2 = warm, +1 = slightly warm, 0 = neutral, -1 = slightly cool, -2 = cool and –3 = cold. An equation in the standard calculates the PMV index.

Even if a PMV value of 0 is obtained, at least 5 % of the occupants will be dissatisfied with the thermal environment.

The percentage of dissatisfied (PPD) index can be determined from the following equation

$$\mathrm{PPD} = 100 - 95\exp - \left(0{,}03353\mathrm{PMV}^4 + 0{,}2179\mathrm{PMV}^2\right) \qquad (2)$$

The standard recommendation for an acceptable environment is $-0{,}5 < \mathrm{PMV} < 0{,}5$, PPD < 10 %.

Besides the general thermal state of the body, a person may find the thermal environment unacceptable or intolerable if local influences on the body from asymmetric radiation, high air velocities, vertical air temperature differences or contact with hot or cold surfaces are experienced.

It was found that persons with lower activity levels (sedentary or standing) are sensitive to draughts, an undesired local cooling of the human body caused by air movement [13]; [14]. A draft is probably the most common reason for complaints in air-conditioned spaces.

Occupants who are subjected to draughts in winter tend to elevate the room temperature to counteract the cooling sensation thereby increasing the energy consumption. In extreme cases ventilation systems are shut off or air supply outlets are blocked off with a consequent deterioration of the indoor air quality.

Fanger *et al* [15] developed a mathematical model to quantify the draught risk in terms of the percentage of dissatisfied people. In this model, the percentage of dissatisfied people due to draughts, DR (%), is calculated from:

$$DR = (34 - T)(v - 0{,}05)^{0{,}62}\,(3{,}14 + 0{,}37vI)$$

for $v < 0{,}05$ m/s insert $v = 0{,}05$ m/s, and for $PD > 100$ % let $PD = 100$ %

(3)

Where T is the local air temperature (C), v is the mean velocity (m/s) and I is the turbulence intensity (%), which is defined as the velocity fluctuation over the mean velocity.

The impact of turbulence intensity on sensation of draft was investigated by [15], [16], and [17] for different methods of air distribution. These studies have shown that a turbulent air velocity is less comfortable than a laminar air velocity of the same average. In occupied spaces the air movement is rather random and not well defined, which is a characteristic of turbulent flow. The turbulence intensity shows the magnitude of the velocity fluctuations v′ in comparison with the mean velocity v:

$$I = \frac{\sqrt{v'^2}}{v} \qquad (4)$$

For the ISO 7730 [11] more than 15 % dissatisfied due to draughts among the occupants will result if the turbulence intensity in the room exceeds 7 % in the summer and 25 % in the winter. These values can be easily exceeded in mechanically ventilated rooms where turbulence intensities can be as high as 50 %. Although ISO do not specify limits for turbulence intensity for the purpose of providing a comfortable environment.

The values of T, v and I can be obtained from the airflow and the heat transfer prediction and, therefore, the PMV, the PPD and the DR indices distributions can be calculated.

NUMERICAL PREDICTION

The results presented here are for a non uniform mesh grid size of 12 * 20 * 15 and non-isothermal airflow conditions [18]. The maximum residual in the mass conservation was < 5 *10^{-3}.

Figures 2 and 3 presents computed results of mean velocity and temperature of room air along the symmetry plane.

The air discharged by the supply air device gives rise to a high-velocity jet which entrains air from the occupied zone, inducing a recirculating air movement. The velocity, in the strongly negatively bouyant jet, accordingly decreases as it reaches the occupied zone at high difference temperature. A flow of this type is undesirable, because the jet have a high velocity (> 0,2 m/s) when it reaches the occupied zone.

Figure 2. Predicted mean air velocity

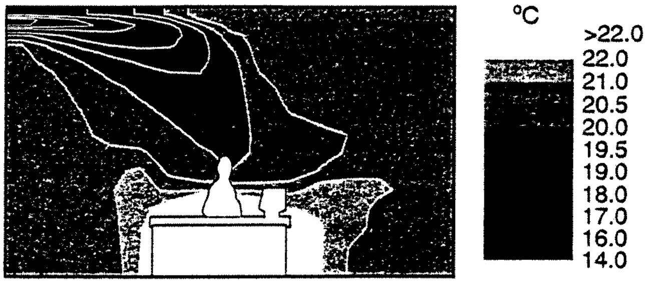

Figure 3. Predicted air temperature contours

Due to the mixing action, temperature is distributed uniformly, outside the jet.

Figure 4 shows the distribution of the percentage of thermal comfort index (PPD). The room air in the low level room area, outside the jet, is thermally comfortable (0< PMV < -0,3; 5% < PPD < 7%) according to the prediction. However, it is slightly cool in the area near the supply jet (PMV<-0,7; PPD > 15 %).

The impact of high air velocity, low temperature and high turbulence intensity on sensation of draft can be seen in Figure 5. When the air jet reaches the occupied zone, at the neck level, a percentage of dissatisfied people due to draughts DR = 10 % was obtained.

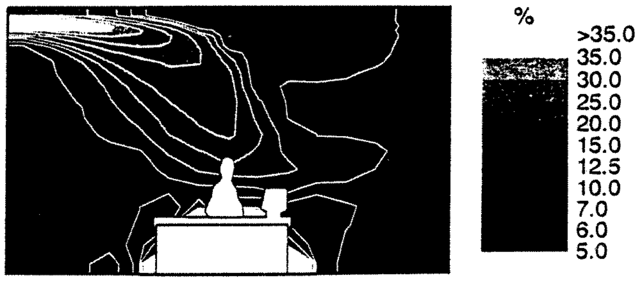

Figure 4. PPD index distribution

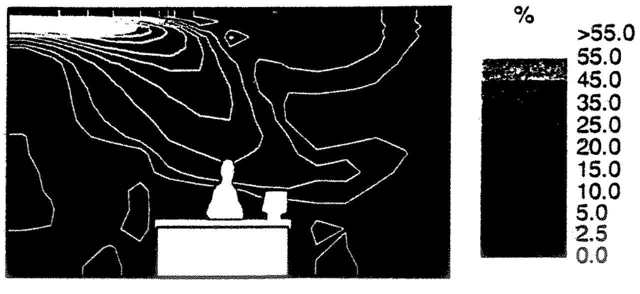

Figure 5. Draught risk distribution

Overall this and previous predictions (see [18]) have demonstrated the capability of the CFD program for obtaining detailed distributions of indoor airflow pattern and environment, giving accurate boundary conditions.

CONCLUSIONS

As confidence is gained in the ability of CFD techniques to accurately simulate room air flows, efforts can be directed toward practical uses of the methodology to support design.

This paper concentrates on the development of a methodology for indoor airflow computation of a room with a well-mixing ventilation system.

From the numerical solutions of the distributions of air velocity, air temperature and turbulence energy, in the room, the PPD index and the draught risk (DR) distributions can be calculated.

The impact of high air velocity, low temperature and high turbulence intensity on sensation of draft should be considered when air distribution systems are designed.

Further investigations are underway to evaluate the comfort in this and other offices in order to explore the effects of individuals, climate and room use on the comfort requirements.

The study of methods of air distribution to promote improved comfort distribution is such a use, and this work demonstrates the application.

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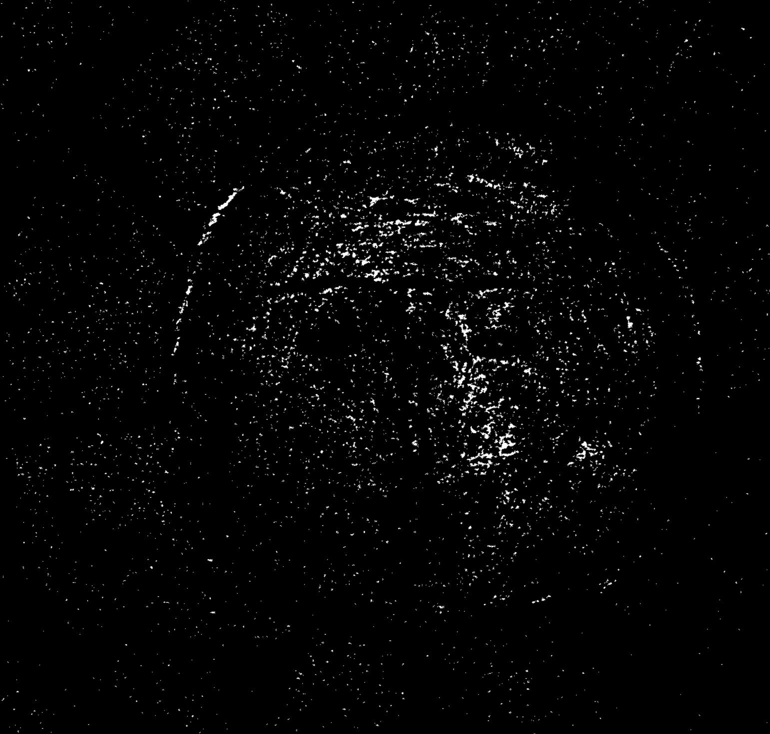

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