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A numerical study of air movement in rooms with solar radiations gains

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A NUMERICAL STUDY OF AIR MOVEMENT IN ROOMS WITH SOLAR RADIATIONS GAINS

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ABSTRACT

The design of air distribution systems for air conditioning requires reasonably detailed knowlegde of the flow. Essentially, experimental tools have been in use for years to study such flows, but the rapidly decresing cost of powerful computer facility is now bringing a new design tool based upon the numerical solution of the general flow equations. The paper describes an application of this type. The air distribution arrangement adopted corresponds to a supply/return box placed below a window, with air blown upwards close to the window surface and removed through a slot near floor level.

INTRODUCTION

When determining the dimensions of a supply opening it is normally presupposed that the injected jet will form a free jet or a wall jet. In this jet it is easy to predict entrainment, velocity and temperature profiles, and the diffuser is often designed in a way the velocity falls to a given value (i.e. 25 cm/s) when jet has reached 3/4 of the lenght of the room, (1).

In a ventilated room there are other parameters such as the dimensions and geometry of the room, the distribution of load and the temperature difference between return and supply will have a strong influence on the flow (2). If the calculation of temperature and velocity profiles in the occupied zone is wanted, all these parameters must be taken into consideration.

This paper will describe results which can predict the flow in all regions of the room, including the occupied zone, taking all the mentioned parameters into consideration.

The procedure is based on solution of the equation of motion, in time-average form and with modeled turbulent stresses.

The present paper describes an application of a two-dimensional procedure based on the SIMPLE algoritm used to solve conservation equations for momentum (two components), mass and thermal energy. Turbulent versions are employed, (K-ε) model. In addition to this, radiative heat transfer between room surfaces is computed.

The following section introduce the procedure; in section 3 calculated results and its discussin are presented. The relevant conclusions appear in section 4.

EQUATIONS AND NUMERICAL PROCEDURE

The equations solved are the continuity, momentum and energy equations, written in time-average form and expressing the turbulent stresses and heat fluxes in termes of turbulent diffusivities for momentum ν and heat Γ . The diffusivities are realted throught the turbulent Prandtl number σ and the former is calculated from the turbulent Kinetic energy K and its dissipation rate ε.

The differential equations are:

mass:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0$$

momentum:

$$\rho \left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) = - \frac{\partial p}{\partial x} + \mu_{ef} \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right)$$

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$$\rho \left(u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} \right) = - \frac{\partial p}{\partial y} + \mu_{ef} \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial x^2} \right)$$

energy:

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \rho \left(\frac{\nu}{\sigma} + \frac{\nu_t}{\sigma_t} \right) \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right)$$

Kinetic energy:

$$u \frac{\partial k}{\partial x} + v \frac{\partial k}{\partial y} = \frac{\partial}{\partial x} \left(\frac{\nu_t}{\sigma_k} \frac{\partial k}{\partial x} \right) + \frac{\partial}{\partial y} \left(\frac{\nu_t}{\sigma_k} \frac{\partial k}{\partial y} \right) + \nu_t \left\{ 2 \left[\left(\frac{\partial u}{\partial x} \right)^2 + \left(\frac{\partial v}{\partial y} \right)^2 \right] + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 \right\} + \beta g \frac{\nu_t}{\sigma_t} \frac{\partial T}{\partial y} - C_D \frac{k^{3/2}}{l}$$

Dissipation energy:

$$u\frac{\partial\varepsilon}{\partial x}+v\frac{\partial\varepsilon}{\partial y}=\frac{\partial}{\partial x}\left(\frac{\nu_t}{\sigma_\varepsilon}\frac{\partial\varepsilon}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\nu_t}{\sigma_\varepsilon}\frac{\partial\varepsilon}{\partial y}\right)+C_{1\varepsilon}\frac{\varepsilon}{k}\left\{\nu_t\left\{2\left[\left(\frac{\partial u}{\partial y}\right)^2+\left(\frac{\partial v}{\partial x}\right)^2\right]+\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)^2\right\}+\beta g\frac{\nu_t}{\sigma_t}\frac{\partial T}{\partial y}\right\}*$$

$$*(1+C_{3\varepsilon}R_f)-C_{2\varepsilon}\frac{\varepsilon^2}{k}$$

where:

$$\mu_{ef}=\mu+\mu_t$$

The coefficient of volumetric expansions β is assumed as constant and the values of constants used are shown in table 1.

Table 1 - Constants

C_D	$C_{1\varepsilon}$	$C_{2\varepsilon}$	$C_{3\varepsilon}$	σ_k	σ_ε	σ_T
0,09	1,44	1,92	1,44	1,0	1,3	0,7

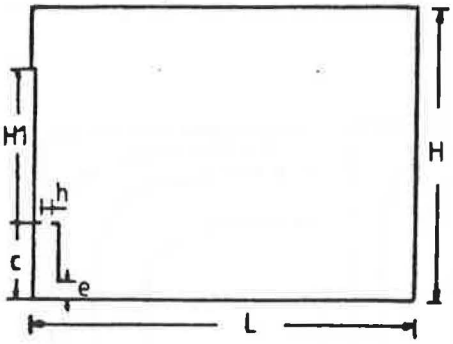
The boundary conditions are imposed through wall functions and for the energy equation heat flux distributions over the room surfaces are prescribed.

The equations were discretized and solved with the SIMPLE algoritm, described in (3), adapted to include the heat transfer by radiation at the solid surfaces.

The calculations were performed in a Data General MV 8000, on the Mechanical Department of the University of Porto.

RESULTS

The geometric configuration studied is showed in figure 1.

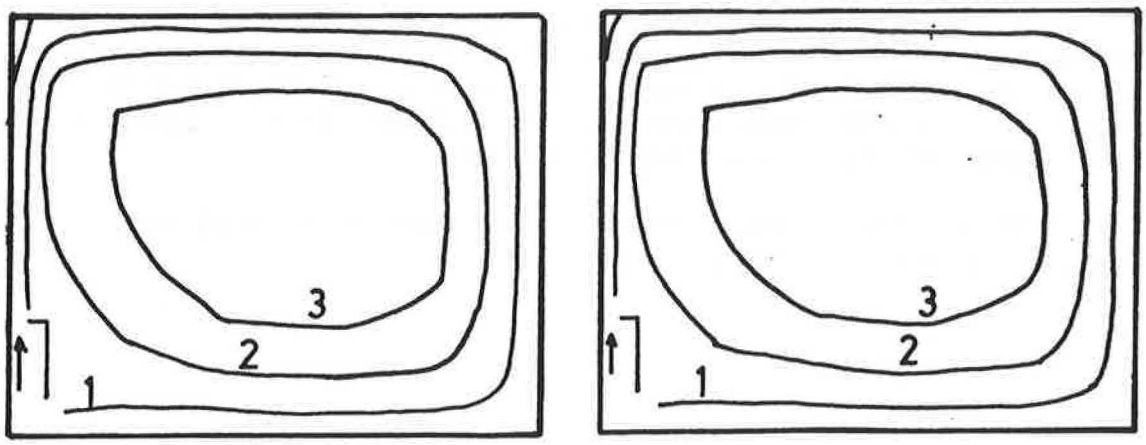


L	3,375
H	2,7
H1	1,3
h	0,04
c	0,8
e	0,09

Fig. 1 - Geometry

Air is injected upwards into the ventilated space and returns at floor level.

Preliminary simulations were performed and allowed to confirm that the flow patterns were essentially similar when the Archimeds number, for different conditions, is the same, as it is plotted on fig.2. This fact confirms the general equation analyses, (4,5), that concludes the little dependence for that parameter.

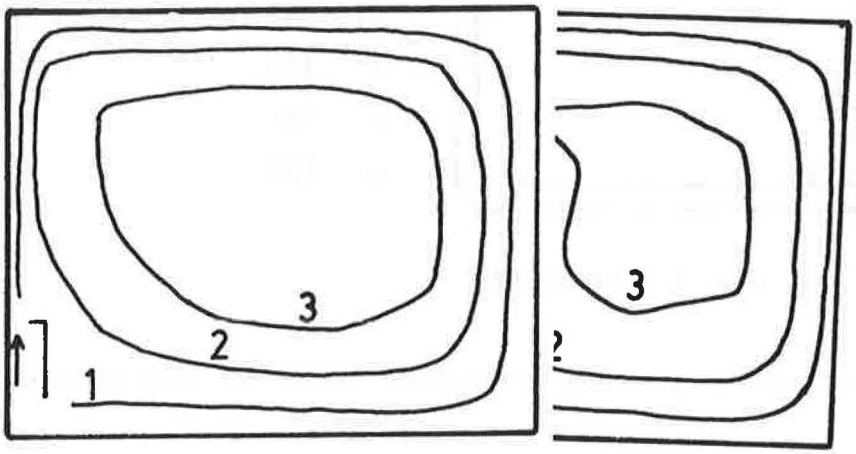


a) 9 ACH; ΔTo=-3 °C

b) 13 ACH; ΔTo=-6.2° C

Fig. 2 - Stream function

The inclusion of radiation heat exchanges may, in some cases, modify the flow patterns. It is expected that higher air temperatures, differences between the inlet and the outlet, will cause different temperatures of surfaces, and so induce natural convection that will change the flow patterns in a way, (6,7), as it is plotted on fig.3.



a) 13 ACH; ΔTo=-7.2°C without radiation

b) 13 ACH; ΔTo=-7.2°C with radiation

Fig. 3 - Stream function

In these figures it is presented the stream function for 13 air changes per hour and the temperature difference between inlet and outled is -7.2°C. In fig.3a the radiation is not included but in the fig.3b the radiation heat exchanges are included. These figures show some changes of the stream function contours.

Different emissivities for enclosures surfaces were assumed; 0.8 for glass surface and 0.6 for other ones.

The existence of a glass panel in a room, like the studied geometry, allows the solar radiation to strike the interior surfaces. The winter and summer conditions are different as it is plotted on fig.4.

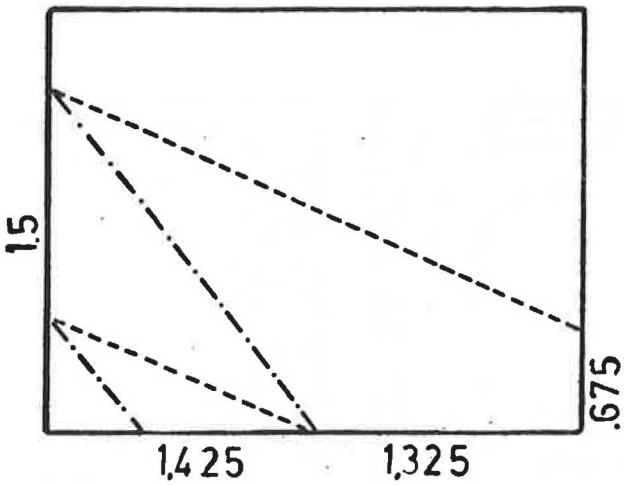


Fig. 4 - Winter and Summer conditions

In this figure are indicated the dimensions of surfaces that are striked by the solar radiation. For the summer condition it is assumed a dimension on floor over 1.425 m from 0.625 m of the glass panel. In winter period it covers a range of 1.325 m from 2.05 of glass panel and on the opposite wall it is 0.675 m height.

On the numerical simulation the heat fluxes on the different surfaces are considered as internal gains as is plotted on fig.5.

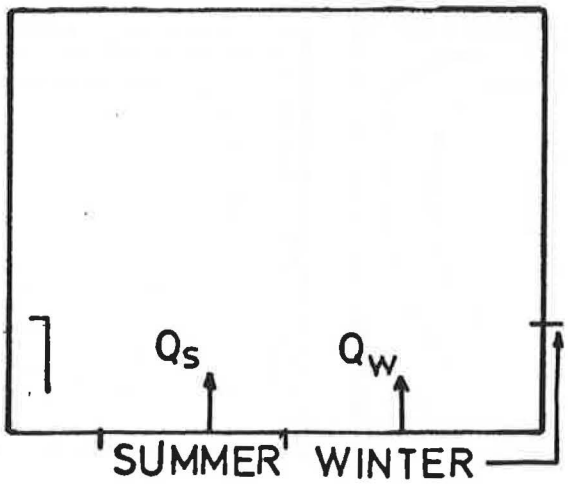
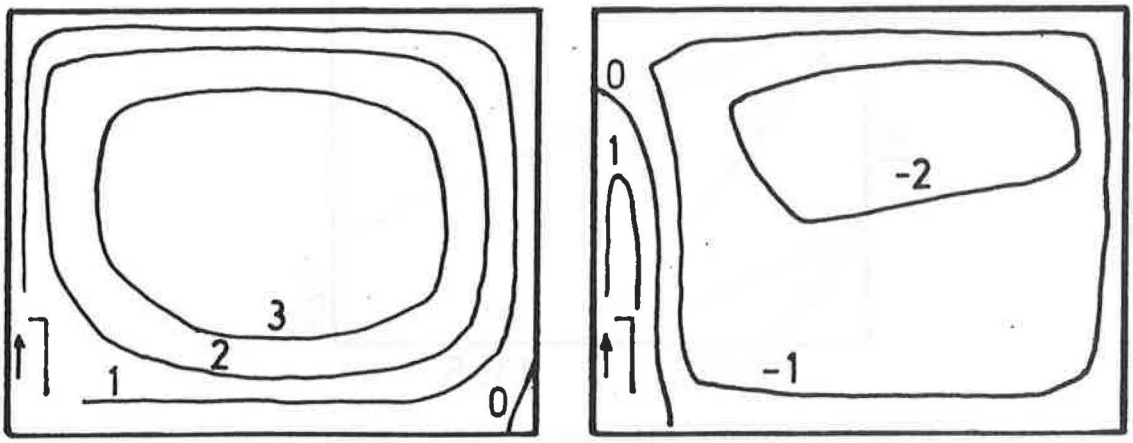


Fig. 5 - Position of internal gains

The winter situation are plotted on fig. 6. The heat flux of 60 W/m^2 on the floor and on the opposite wall, correspondent to the winter situation, produces a temperature difference of -0.2°C for 13 air changes per hour, fig.6a.

For the same air change rate, the temperature difference between inlet and outlet is -1.4°C if the heat flux on the floor is 90 W/m², fig.6b. The heat flux conditions for the glass panel are variable. These one are the balance between convective and radiation heat fluxes.

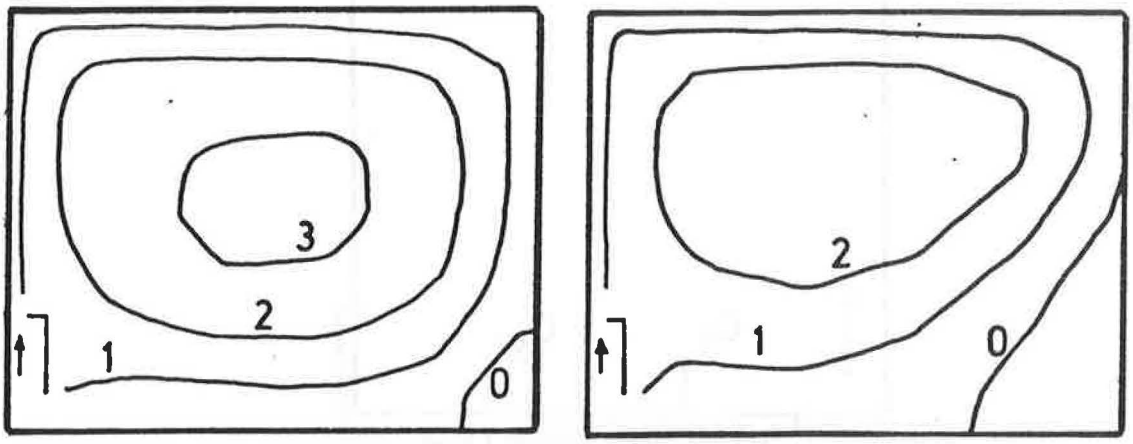


a) 13 ACH; ΔTo=-0.2°C
Winter; Qw= 60 W/m²

b) 13 ACH; ΔTo=-1.4°C
Winter; Qw= 90 W/m²

Fig. 6 - Stream function

The summer situations are plotted on fig.7. The calculations were carried out for 13 ACH. On the fig.7a the heat flux is Q_s = 10 W/m² and on the fig.7b the heat flux on the floor is Q_s = 40 W/m².

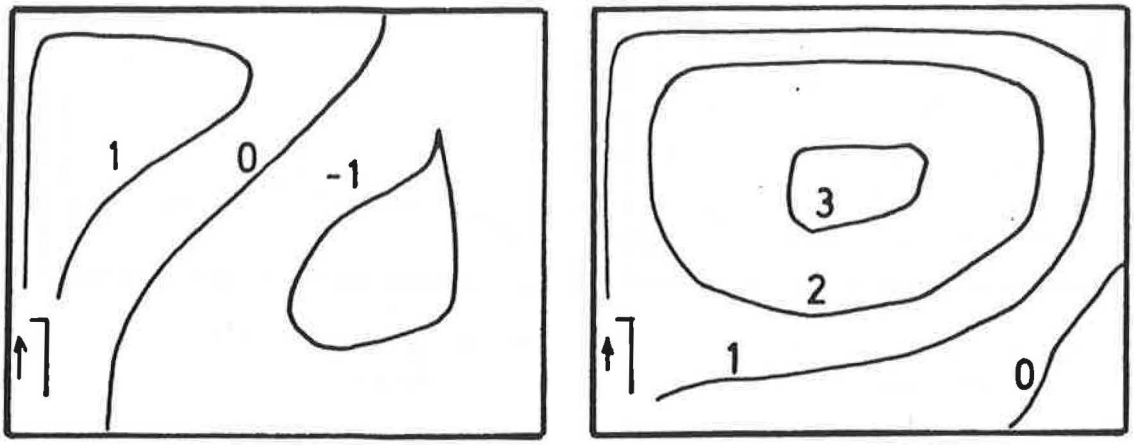


a) 13 ACH; ΔTo=+2.4°C
Summer; Qs= 10 W/m²

b) 13 ACH; ΔTo=+3.5°C
Summer; Qs= 40 W/m²

Fig. 7 - Stream function

On the figure 8 are plotted two situations. On fig.8a is considered the heat radiation exchanges between the surface and a heat flux on the glass panel. On figure 8b is for summer condition. The two simulations might be similar because they have the same Archimedes number, A_R=0,63x10^{-2}.



a) 9 ACH; ΔTo=+1.5°C
with radiation

b) 13 ACH; ΔTo=+2.4°C
Summer

Fig. 8 - Stream function

Although the Archimedes number is the same the flow patterns are different, because the heat sources are in different positions. On fig.8a the heat flux is on the glass panel and on fig.8b the heat fluxes are in the glass panel and on the floor.

The numerical simulation is important but it is necessary to make validation by experimental results. Sometimes the flow patterns are very differents.

On the simulation carried it is assumed that the opaque surfaces have not heat losses for the exterior. This condition is obtain on the practical by a double adiabatic room. In the absence of this apparatus there is a heat flux to outside. This fact may be simulated by the imposition of a heat flux of losses on the surfaces.