AIR MOVEMENT & VENTILATION CONTROL WITHIN BUILDINGS

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POSTER 50

INFLUENCE OF RADIATIVE PARTICIPATION OF INSIDE AIR ON NATURAL CONVECTION IN A ROOM

SYNOPSIS: The bases of this study are experimental results obtained on a real scale cell in controlled climatic conditions which are used to show the potential influence of radiative participation of inside air on natural convection in a room.

In a second part, a numerical analysis of flow patterns and heat tranfer in a two dimensionnal thermally driven cavity containing a participating fluid is presented. The results obtained show the influence of the radiative coupling between the walls and the fluid, and the influence of the radiative transfer inside the fluid itself on the thermal field and flow patterns obtained in the cavity.

F. ALLARD - C.INARD - A. DRAOUI

CETHIL, URA CNRS 1372
INSA de Lyon, bât. 307
F-69621 Villeurbanne cedex, France.

Nomenclature

A = aspect ratio ($A = Y_{max}/X_{max}$)
F_{ij} = view factor
$F\tau_{ij}$ = view function
d_{ij} = distance
G = radiation intensity
J = radiosity
L^0 = black body intensity
N_{CR} = interaction parameter
P = pressure
Pr = Prandtl number
Ra = Rayleigh number
S = Surface area
T = temperature
t = time
T_C = temperature of the cold wall
T_H = temperature of the hot wall
T_{ref} = reference temperature (mean temperature = $(T_H + T_C)/2.$)
X_{max} = width of cavity
Y_{max} = height of cavity
x, y = coordinates

U= x - component of the velocity
V= y - component of the velocity

Greek symbols
ß= mean extinction coefficient
ε = radiative emissivity
Δ= Laplace's operator
λ = thermal conductivity of the fluid
σ_o = Stephan-Boltzmann constant
τ_o = optical thickness of the fluid ($\tau_o = ß.X_{max}$)
Ω_o= scattering albedo
ν = frequency
φ = radiative flux
θ = angle

Superscripts
* dimensional quantity
e emitted
t transmited

1. - INTRODUCTION

A common assumption in building physics is to consider air as a transparent medium for long wave radiation heat transfer. In fact, looking more carefuly at the absorption spectra of air components such as water vapor or carbon dioxide, we can expect in certain cases a reasonnable radiative participation of inside air. In order to improve our knowledge about this particular point, we first carried out a serie of experiments in our MINIBAT test cell facility, then we introduced in a CFD code the radiative transfer inside the fluid itself to evaluate theoretically the influence of radiative participation of the fluid itself about flow patterns and heat transfers in a thermally driven cavity.

2. - BASIC METHOD FOR EVALUATION OF RADIATIVE PARTICIPATION OF AIR

Figure 1 shows the spectral emmisivities of water vapor and carbon dioxide

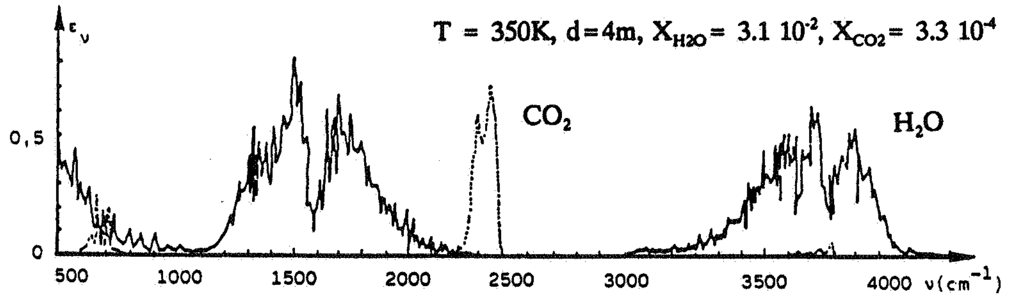

Figure 1 : Spectral emmissivities of H_2O an CO_2[1]

Even if the spectra given on Figure 1 show a significative emmissivity of water vapor, in buiding physics we used to neglect this radiative contribution of air to the energy balances considering that the overall emission of the air volume is compensated by the absorption of this same volume of the radiations coming from the surfaces of the room. That could be completely true if the surface temperatures and the air temperature were exactly similar. In fact all the experiments [2] carried out show clearly air stratification and surface temperature heterogeneousness.

Furtheremore, the heat fluxes exchanged along the walls are weak, and thus the heat balance can be very sensitive to any additionnal phenomenon.

In order to show out what could be the influence of radiative participation of air on the thermal balance of a room, we introduced in a classical spectral radiosity radiative model the effects of emission and absorption of the gas. As a first approximation we consider also that the air temperature differences inside the room will not change the absorption and emission spectra. We select as reference temperature the temperature of convective equilibrium of the air volume, this temperature is usually very close to the air temperature measured at the center of the room. In this case, the flux radiated by a surface element dS_i in a frequency domain $\Delta\nu$ centered on ν_0 and arriving on an other surface element dS_j can be written

$$d\phi^{t}_{i,j,\Delta\nu} = J_{i,\Delta\nu}\frac{\cos\theta_i \cos\theta_j dS_i dS_j e^{-K\nu_0 d_{ij}}}{\pi d_{ij}^2} \qquad (1)$$

Where K represents the monochromatic extinction coefficient. If we assume the reflection and emission of each surface to be diffuse, and integrating on each surface S_i and S_j, we will obtain the total flux radiated by S_i and arriving on S_j as:

$$\phi_{t_{i,j\Delta\nu}} = J_{i,\Delta\nu}\, F\tau_{ij}\, S_i \qquad (2)$$

We define here a new exchange function $F\tau_{ij}$ which includes in its definition both geometry characteristics and radiation transmission through the gas column. Following the same concepts, we define the proper emission of the gas column arriving on S_j.

$$\phi^{e}_{G,i,j,\Delta\nu} = \pi L^{0}_{G,\Delta\nu}\iint_{S_i S_j}\frac{\epsilon_{G,\Delta\nu}\cos\theta_i \cos\theta_j\, dS_i dS_j}{\pi d_{ij}^2} \qquad (3)$$

Or using again the above mentionned symbols:

$$\phi^{e}_{G,i,j,\Delta\nu} = \pi L^{0}_{G,\Delta\nu}(F_{ij}-F\tau_{ij})\, S_i \qquad (4)$$

Then, calling upon all the directions in the space and therefore all the elementary surfaces defined on the walls forming the room, we obtain the total monochromatic flux radiated by the gas to surface S_i.

$$\phi^{e}_{G,j,\Delta\nu} = \pi L^{0}_{G,\Delta\nu}\sum_{i=1}^{n}(F_{ij}-F\tau_{ij})\, S_i \qquad (5)$$

Writing the same expression for each surface forming the room, we finally obtain for each spectral interval a system of equations defining the spectral radiosity of each surface.

$$J_{j,\Delta\nu}-\rho_j\sum_{i=1}^{n}J_{i,\Delta\nu}F\tau_{ij}\frac{S_i}{S_j} = \epsilon_{j,\Delta\nu}M^{0}_{j,\Delta\nu}+\rho_j\pi L^{0}_{G,\Delta\nu}\sum_{i=1}^{n}F_{ij}-F\tau_{ij}\frac{S_i}{S_j} \qquad (6)$$

Integrating over all the considered spectrum (practically wavelenght from 2.5 to 70 μm), we then obtain the net fluxes of each surface and the total fluxes emitted and absorbed by the gas volume.The difference between these two quantities called net flux of the gas characterizes the overall radiative participation of air.

3. - INFLUENCE OF AIR RELATIVE HUMIDITY ON ITS RADIATIVE PARTICIPATION.

The basis of this study is a serie of experiments carried out in our Minibat Test cell [3]. In order to raise the potential effect of radiative participation of air we study the case of a convective heating of the cell. The convective power of the source is 2158W, the final air temperature obtained at the center of the room is 46.5 °C. As the extinction spectral coefficient depends directly on the relative pressure of water vapor, we can look at the influence of relative humidity of air varying this parameter in our model. Figure 2 shows this influence on the net radiative flux of the overall mass of air. We use here the surface temperature measured during the experiment as input data for our model.

The radiative deficit of air rapidly reaches 10% of the total convective power injected in the room. (.015 represents 23% of humidity at the conditions of the experiment) It is important to mention here than very low relative humidity values have a significant influence on the radiative participation of air.

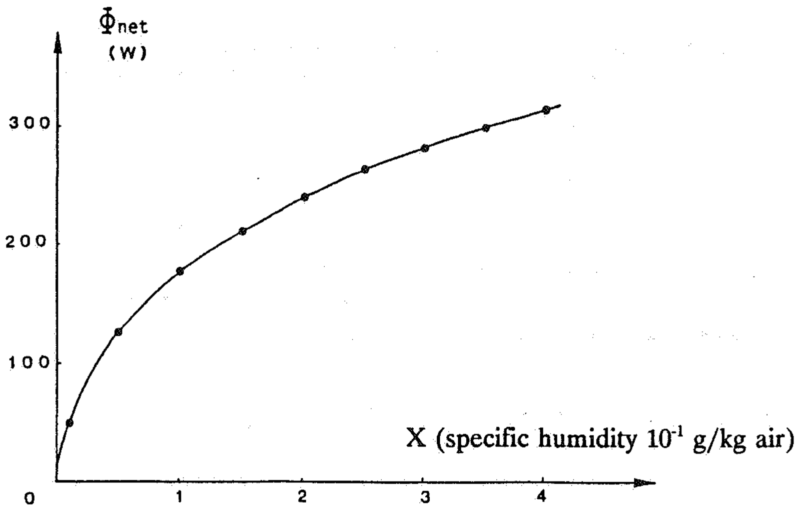

Figure 2 : Influence of specific humidity on the air radiative net flux.

If we now consider the influence of water vapor content on the heat balance of each wall surface, we see on Figure 3 that the deviation in estimating the convective heat flux at the surface of the walls is significative, it represents 25 percent of the weakest convective heat fluxes in our case.

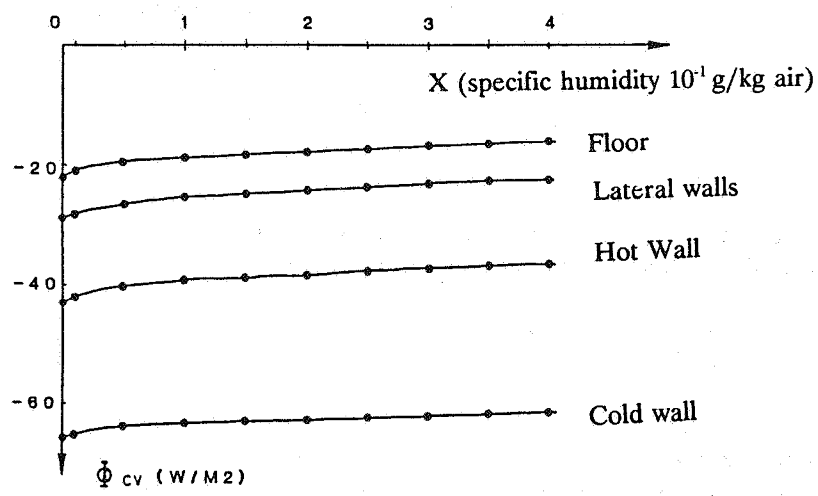

Figure 3 : Influence of the radiative participation of air on the evaluation of convective fluxes along the walls.

This first experimental study enables to show that radiative participation of air can be a significant phenomenon perturbating the energy balance of a room. Nevertheless it is limited to a convective heating situation and does not allow us to unterstand how mass and heat transfer patterns can be modify. To answer this question, we used the CONCAV code developped in our team and we coupled the Navier Stokes equations with the radiative transfer equation.

4. - NUMERICAL STUDY OF NATURAL CONVECTION IN SEMI-TRANSPARENT FLUIDS.

4.1. - MATHEMATICAL FORMULATION

The flow and heat transfer within a two-dimensional enclosure filled with a gas (absorbing, emitting, and isotropically scattering), are described by the Navier-Stokes equations in their Boussinesq-Obberbeck approximation, as the energy equation, as well as an additional equation for the radiative transfer. In our case, the P1 differential approximation [4] of the radiative transfer equation has been preferred, due to its compatibility with the governing equation system which gives the flow and heat transfer. The complete system (nondimensional) of governing equations is therefore expressed as :

*** Continuity :**

$$\frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} = 0 \qquad (7)$$

*** Momentum :**

$$U\frac{\partial U}{\partial x} + V\frac{\partial U}{\partial y} = - \frac{dP}{dx} + Pr.\Delta U \qquad (8)$$

$$U\frac{\partial V}{\partial x} + V\frac{\partial V}{\partial y} = - \frac{dP}{dy} + Pr.\Delta V + Ra.Pr.(T - 0.5) \qquad (9)$$

*** Energy :**

$$U\frac{\partial T}{\partial x} + V\frac{\partial T}{\partial y} = \Delta T + \frac{1}{3N_{CR}}.\Delta G \qquad (10)$$

*** Radiative transfer :**

$$\frac{\partial^2 G}{\partial x^2} + \frac{\partial^2 G}{\partial y^2} - 3.\tau_o^2.A^2.(1 - \Omega_o).G = \qquad (11)$$

$$- 12.A^2.\tau_o^2.(1 - \Omega_o).\frac{(T_{ref} + T - 0.5)^4}{T_{ref}^4}$$

where the non-dimensional variables are defined as :
$x = x^*/Y_{max}$, $y = y^*/Y_{max}$, $A = Y_{max}/X_{max}$, $U = U^*.Y_{max}/a$, $V = V^*.Y_{max}/a$,
$T = (T^* - T^*_H)/(T^*_C - T^*_C)$, $T^*_{ref} = (T^*_H + T^*_C)/2$, $T'_{ref} = T^*_{ref}/(T^*_H - T^*_C)$
$G = G^*/(\sigma_o.T^{*4}_{ref})$, $\tau_o = \beta.X_{max}$, $N_{CR} = \lambda.\beta(T^*_H - T^*_C)/(\sigma_o.T^{*4}_{ref})$

The boundary conditions in a dimensionless form are :
U = V = 0 at all the solid surfaces; T(0,y) = 1. and T(1./A,y) = 0

$$\frac{\partial T}{\partial y} + \frac{1}{3N_{CR}} \cdot \frac{\partial G}{\partial y} = 0 \qquad at\,y = 0 \;\vee\; y = 1$$

and

$$\frac{\partial G}{\partial n_i} = \pm\frac{3.\tau_o.A.\epsilon_i}{4 - 2\epsilon_i}.[G - 4.(T_i + T_{ref} - 0.5)^4/T_{ref}^4]$$

Where n_i denotes the coordinates normal to the wall, "i" denotes quantities at the wall i (i = 1, 2, 3, 4). T_i, ϵ_i are the dimensionless temperature and the emissivity of the wall i. The positive sign is taken at x = 0 or y = 0, and the negative sign at x = 1/A or y = 1. The governing equations (7)-(11) are solved by integration on elementary control volumes of all the set of differential equations including radiation transfer, and finite difference formulation. The transport-diffusion equations are expressed using a hybrid scheme. The **SIMPLER** algorithm [5], is used to obtain iteratively the solutions and by the use of a staggered grid ensures a conservative formulation.

4.3. - NUMERICAL RESULTS AND DISCUSSION

While the values of the parameters concerning natural convection are assumed to be constant, the effects of radiation on the natural convection in a square cavity are examined by decreasing the Planck number N_{CR} or by increasing the optical thickness τ_o. In the first computations, the **"Window Problem"** is treated with a Planck number decreasing from 10 to 0.05. Figure 4 shows the isotherms, streamlines and heatlines for two different Planck numbers with the same values of optical thickness (τ_o = 1) and of Rayleigh number (Ra = 10^8). It indicates that when N_{CR} decreases, the isotherms become closer near the top of the hot wall, and near the bottom of the cold wall. This phenomenon is coupled with a decrease in vertical gradient of the temperature. Furthermore, when the radiative participation becomes more important. The effect of the Planck number on the mean Nusselt number at the hot wall is presented in table 1. The effect of radiation on natural convection clearly appears with the decrease in N_{CR} [6-7]. The mean Nusselt number for convection decreases because the medium near the hot wall absorbs radiation and so makes the temperature gradient at the lower lower. However, the radiative heat transfer is so intense near the hot wall that the radiative Nusselt number increases, whereas, for the window problem, the mean Nusselt number is independent of coordinate x in the all vertical plan, and is equal to the value at the hot wall .

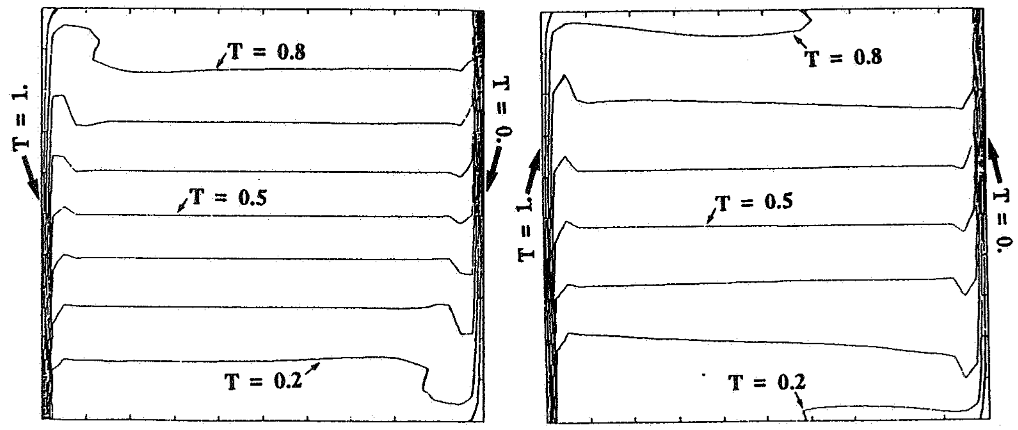

Figure 4 : Effect of parameter interaction N_{CR} on isotherms patterns, (Ra = 10^8, τ_o = 1., A = 1., Ω_o = 0., Pr = 0.71, ϵ_1 = ϵ_2 = 1., ϵ_3 = ϵ_4 = 0.).

Interaction parameter N_{CR}	Mean Nusselt number for convection	Mean Nusselt number for radiation	Mean total Nusselt number
∞	30.02	0.00	30.02
1	29.91	0.54	30.45
0.1	28.75	5.42	34.17
0.05	27.74	10.78	37.52

Table 1 : Effect of Planck number on the mean Nusselt number at the hot wall (τ_o = 1., A = 1., Ra = 10^8, Ω_o = 0., $\epsilon_1 = \epsilon_2$ = 1., $\epsilon_3 = \epsilon_4$ = 0.)

5. - CONCLUSION

The two examples presented here are only to illustrate what can be the effect of radiative heat transfer on flow patterns and heat transfer in natural convection. In some cases, radiative participation of water vapor in air may represent around of 10% of the total heat fluxes exchanged in a room, if this phenomenon can be neglected in most applications, it can significantly perturb the evaluation of convective heat transfer at the surfaces of the walls. Furtheremore as far as scale models using water or freon are used to study natural convection in rooms, the results obtained with such fluids should be carefuly analyse before their application to building physics.

REFERENCES

[1] J. TAINE
"Spectres d'absorption à haute résolution des mélanges H_2O, $C0_2$ gaz neutres", personal comunication, Ecole Centrale des Arts et Manufactures, Paris, 1985.

[2] F. ALLARD
"Contribution à l'étude des transferts de chaleur dans les cavités thermiquement entrainées à grand nombre de Rayleigh". Thèse de Doctorat ès Sciences, INSA Lyon 1987.

[3] F. ALLARD, J. BRAU, C. INARD and J. M. PALLIER
"Thermal experiments of full scale dwelling cells in artificial climatic conditions". Energy and Buildings, N° 10, P. 49-58, 1987.

[4] M. N. ÖZISIK
" Radiative Transfer and interactions with Conduction and Convection", Wiley-Interscience, New York, (1973).

[5] S. V. PATANKAR,
"Numerical Heat Transfer and Fluid Flow", Mac Graw Hill Edit., London, (1980).

[6] A. DRAOUI, J. P. SIMONEAU, F. ALLARD,
"Combined radiation and convection in gray fluids enclosed in 2D cavities", ASME Winter Meeting, Chicago, december, Vol. 3, pp.141-146, (1988).

[7] A. DRAOUI, F. ALLARD, C. BEGHEIN,
"Numerical Analysis of Heat Transfer by Natural Convection and Radiation in Participating Fluide Enclosed in Square Cavities", **Numerical Heat Transfer**, Part A, Vol. 20, pp. 001-009, 1991.