#5335

ANISOTHERMAL STEADY STATE MODELING OF HEATED DWELLING-CELL WITH ZONAL MODELS. APPLICATION TO RADIATOR AND HEATED FLOOR.

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Summary

This study deals with predicting thermal behavior of heated dwelling-cells in steady state using zonal models. Two kinds of heating systems are studied: radiator and heated floor. From experimental and numerical results, we show how zonal models can be developed. Comparisons between calculated and experimental heat powers and air temperature profiles show good agreement. At last, we compare results obtained by zonal models and simplified isothermal model with experimental results as reference. In the case of the heated floor, the usefulness of zonal models is not clear but zonal models are useful to take into account specific thermal couplings between radiators and the dwelling-cell.

List of symbols:

A : area (m^2)
a : air thermal diffusivity (m^2/s)
Cp : air specific heat capacity (J/KgK)
F_{ij} : view factor between wall i and wall j
Fr : Froude number
g : acceleration of gravity (m/s^2)
H : height of the cell (m)
Hra : height of the radiator (m)
hc : convective heat transfer coefficient (W/m^2K)
J : radiosity (W/m^2)
Lra : length of the radiator (m)
Nu : Nusselt number
P : total heat power (W)
Pconv : convective heat power (W)
Q(z) : linear heat flux of the plume at altitude z (W/m)
Ra : Rayleigh number
St : Stanton number
T_i : air temperature of zone i (°C)
t : time (s)
Tm(z) : maximum air temperature of the plume at altitude z (°C)
Tout : outside air temperature (°C)
Tra : radiator surface temperature (°C)
Tror : radiant room temperature (°C)
Tw : surface temperature (°C)
$T\infty$: air temperature outside of the plume (°C)
Um(z) : air maximum velocity in the plume at altitude z (m/s)
V_{ij} : air mass flow rate between zone i and zone j (Kg/s)
Vbl(z) : linear air mass flow rate of the boundary layer at altitude z (Kg/ms)

Ve : exhaust air mass flow rate (Kg/s)
Vp(z) : linear air mass flow rate of the plume at the altitude z (Kg/ms)
Vs : supply air mass flow rate (Kg/s)
v : velocity (m/s)
z : altitude (m)
z_o : virtual origin of plume heat source (m)

Greek symbols:
β : air volumetric expansion coefficient (K^{-1})
ϵ : LW emissivity
ϵra : radiator LW emissivity
ρ : air density (Kg/m^3)
σ_o : Stefan-Boltzmann constant (W/m^2K^4)
$\Phi conv$: convective heat flux (W)
$\phi conv$: convective heat flux density (W/m^2)
ϕrad : radiant heat flux density (W/m^2)

Subscripts:
b : backwall
c : ceiling
f : floor
t : trail
v : vertical walls

1. INTRODUCTION

The thermo-convective field of air conditionned dwelling-cells is induced by several components: isothermal or anisothermal supplied air, thermal plume from internal heat sources, boundary layers along walls, secondary flows...
All these flows are generally turbulent ones.
Predicting air distribution in ventilated rooms allows:
- a better evaluation of convective heat fluxes
- an approach of local thermal confort
- a better dynamic analysis of inside air behavior

The thermo-convective field analysis can be performed thanks to:
- experiments
- mathematical modelling

Experimental methods can be managed with several means such as flow visualization, tracer gaz technique, measurements of temperature and/or velocity lines... Obviously, the experiments represent physic reality, but they display some difficulties:
- expensive and tiresome setting up
- measurements related to sensors characteristics (response time, location...)
- difficulties to extend the results to others configurations

This can be avoided by developing predicting models.
Unfortunately, if we can write the set of equations of the problem, the computers don't allow until now to solve the complete coupled pattern essentialy for two reasons:
- the heat transfer time scales of the different phenomena are very different from each others (conduction/convection..)
- the convective flows are generally turbulent which increases the complexity of the problem.

For these reasons, since several years, anisothermal models

predicting the thermal behavior of dwelling-cells coupled with heating systems are developed.
These models are usually called "zonal models" because the indoor air volume is split into several zones coupled to each others by air mass flows.

2. MODELING PRINCIPLES OF ZONAL MODELS IN STEADY STATE

Assuming that indoor air volume is split into N zones, the convective flow pattern between two zones, respectively at temperature T_i and T_j, and the walls can be represented by the figure 1.

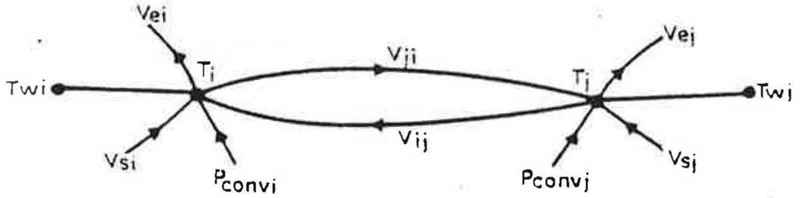


Fig. 1 : Convective flow pattern between two zones and the walls.

For each zone, we write the mass and energy balances:

$$\sum_j V_{ij} + Vs_i = \sum_i V_{ji} + Ve_i$$

$$\sum_j CpV_{ij}(T_i - T_j) + CpVs_i(T_i - Tw_i) + \Phi conv_i = Pconv_i$$

If we assume that air temperatures and air mass flow rates between each zone are the variables, we have N^2 unknowns and only (2N-1) available equations.
So, in order to solve the problem we must introduce additional hypothesises.
The first step can be to define the air flow scenario with, unfortunately, a loss of generality of the model. The scenario must represent as well as possible the physics and can be identified from flow visualizations, experimental measurements (isotherms and/or velocity lines) and also from numerical analyses. So, it is clear that each configuration must be linked with its own air flow pattern.
As an example, figure 2 shows one of the first air flow pattern proposed by Lebrun (1) in the case of a dwelling-cell heated by a localized heat source (radiator or convector).
The second step of the analysis consists in evaluating some of the air mass flow rates. Several ways are possible. Laret (2) used literature results linked to free line heat source plumes and natural convection boundary layers along a vertical plate in order to build his analytical model. From experimental results (air temperatures and convective heat fluxes), Ngendakumana (3) identified air mass flows and related them to the convective power of the heat source. The experimental data were obtained from tests carried out on a climatic chamber uniformly cooled and

the model was extended to the dynamic thermal behavior of the dwelling-cell. Another solution is to make specific temperature and velocity measurements for a better knowledge of specific air flows like boundary layers or thermal plumes. A first attempt was made by Howarth (4). The experimental results allowed the author to predict air temperature gradient in a dwelling-cell heated with a radiator, using a two zones model.

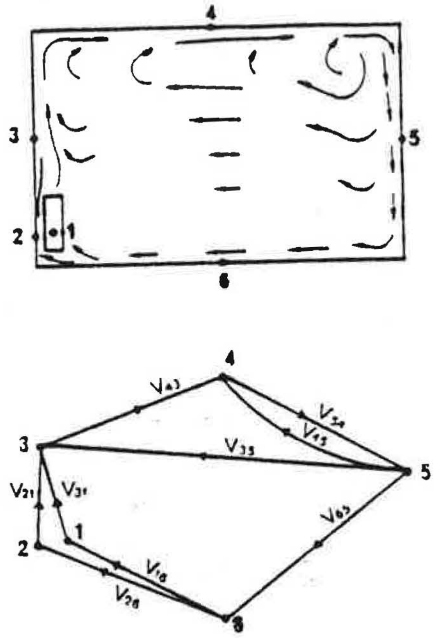


Fig. 2 : Airflow pattern proposed by Lebrun.

We applied the zonal principle to two kind of heating systems :
- radiator
- heated floor

Concerning the radiators, the analysis is based essentialy on experimental results whereas for heated floor we used both numerical and experimental results.

3. APPLICATION TO A DWELLING-CELL HEATED WITH A RADIATOR

In order to be able to define an anisothermal model, we developed an experimental study. The tests were performed in a climatic housing (5) on different radiators and on one electric linear heat source. Two kinds of measurements were performed:

- surface temperatures inside and outside the walls to evaluate conductive rates and compute radiative heat exchanges between walls and with the radiator

- air temperature and air velocity in the plume, which were integrated to compute the air mass flow rates and the heat flow inside the plume.

3.1. EXPERIMENTAL RESULTS

3.1.1. RADIATOR CONVECTIVE HEATING POWER

From the experiments, it has been possible to give the radiator convective power as a function of the difference between the radiator surface temperature and the air temperature measured at the center of the cell and close to the floor:

$$Pconv = A\ (Tra - T\ )^{n} \qquad (W)$$

As an example, Figure 3 shows the experimental results obtained from four different single panel radiators and various mean water temperatures.

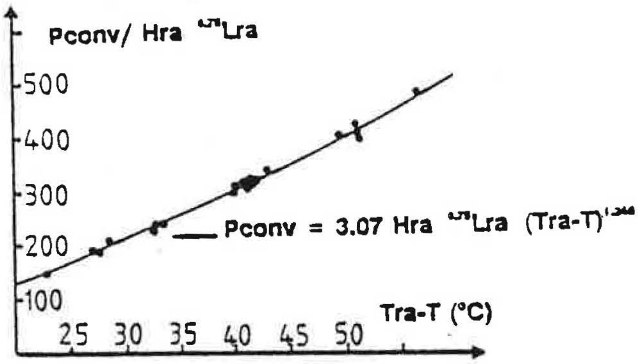


Fig. 3 : Relationship between Pconv and ΔT for single panel radiators

The identified value of n close to 1.25 is characteristic of laminar flow and compatible with the maximum value of Rayleigh number, based on radiator height, in the range of 2.10^{9}.

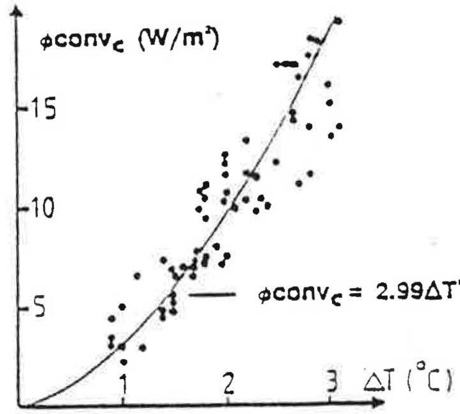
3.1.2. CONVECTIVE HEAT TRANSFER TO THE WALLS

Except for the wall in contact with the plume, the convective heat flux can be written :

$$\Phi conv = hcA(T - Tw) \qquad (W)$$

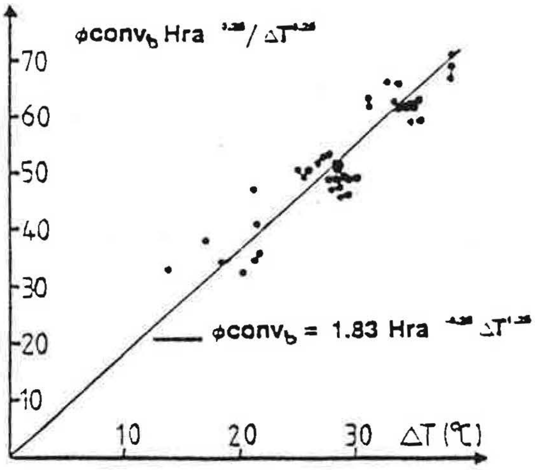
For the backwall of the radiators and the ceiling, the experimental values of hc are shown on Figures 4 and 5.

For the backwall, the temperature difference ΔT represents the difference between the average of the surface temperature of the wall and of the surface temperature of the radiator, and the air temperature measured close to the floor, at the center of the cell. In the case of the ceiling, ΔT represents the difference between the mean surface temperature of the ceiling and the air temperature measured close to this surface and at the center of the cell.



$\Delta T = T - T_c$

Fig. 4: Convective heat flux density at the ceiling.



$\Delta T = (Tra+Tw_b)/2 - T$

Fig. 5: Convective heat flux density at the backwall of the radiator.

3.1.3. PLUME CHARACTERISTICS

The thermal balance inside the plume gives:

$$\frac{dQ(z)}{dz} = -\phi conv_i(z) - Cp\ Vp(z)\ \frac{dT\infty}{dz} \qquad (1)$$

On the one hand the experimental results show, local similarity of velocity and temperature profiles, on the other hand that air mass flow rate in the plume of a radiator Vp(z) is similar to that of linear heat source (Figures 6 and 7).

PS1: Single panel
LAM: Lamellar
PS2: Single panel
PAD: Double panel
CEC: Hot water convector

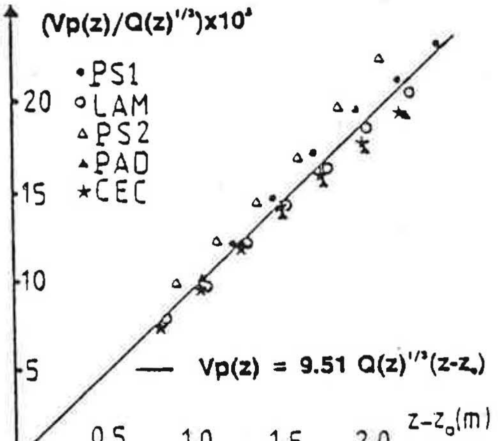


Fig. 6: Air mass flow rate in the plume of radiators.

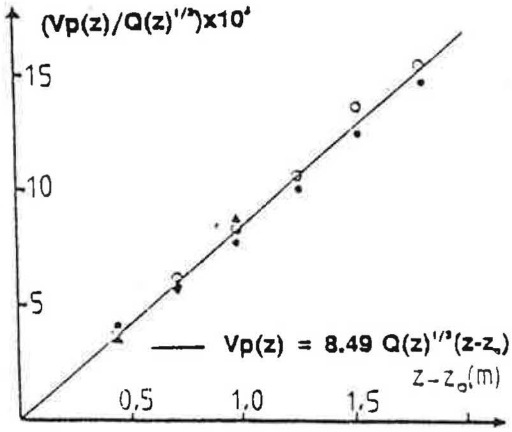


Fig. 7: Air mass flow rate in the plume of linear heat source.

The general equation for Vp(z) is:

$$Vp(z) = K\, Q^{1/3}(z)\ (z - z_o) \qquad \text{(Kg/ms)} \qquad (2)$$

From an integral analysis of the plumes, the maximum velocity and temperature can be described by eq. 3 and 4:

$$Um(z) = [\frac{g\beta}{\rho Cp} Q(z)]^{1/3} Fr^{2/3} \tag{3}$$

$$Tm(z) - T\infty(z) = [\frac{g\beta}{\rho Cp} Q(z)]^{2/3} k Fr^{2/3} / g\beta(z - z_o) \tag{4}$$

k is calculated with dimensionless integrals of the plumes. Moreover, the convective heat flux along the trail is written as a function of the dimensionless Stanton number:

$$\phi conv_1(z) = St\,\rho Cp[Tm(z) - Tw_1(z)]\,Um(z) \tag{5}$$

From equations (1) to (5) and assuming that the temperature gradient δ in the central zone independant of z, we obtain:

$$\frac{dQ(z)}{dz} = -[\alpha_1[T\infty_o - Tw_1(z)] + \alpha_3(z - z_o)]\,Q(z)^{1/3} - \frac{\alpha_2}{z - z_o} Q(z) \tag{6}$$

where : $\alpha_1 = St\,\rho Cp\,Fr^{2/3}(\frac{g\beta}{\rho Cp})^{1/3}$

$\alpha_2 = k\,St$

$\alpha_3 = (\alpha_1 + CpK)\,\delta$

Fr Froude number

Experimental results have shown that Froude number, Stanton number and k reach constant values from around 0.4m above the top of the radiators. Moreover, these values are independant of the type of radiator. Thus, assuming that the wall in contact with the plume is isothermal ($Tw_1(z)$=cte), integration of equation (6) gives:

$$Q(z) = Q(Hra)\,(\frac{Hra - z_o}{z - z_o})^{\alpha_2}(1 - Q\infty - Q\delta)^{3/2} \quad (W/m)$$

Where Q(Hra) is the convective heat flow in the plume at altitude z = Hra (sum of the radiator convective power and of the radiator backwall convective flux).
From Q(z) we can estimate Vp(z) (eq.2) and $\phi conv_1(z)$ (eq. 3, 4 and 5).

Therefore, we obtain a complete description of the plume over the radiator.

3.2. MODELING WITH ZONAL MODEL
3.2.1. MODEL DESCRIPTION

According to Lebrun analysis (1), in most cases the air flow consists in a main circulation along the way: radiator, trail ceiling, vertical walls, floor and radiator. This air flow pattern can also appears through experimental isotherm lines obtained in the mid plane of a cell heated with a convective heat source as showed in figure 8 (6).

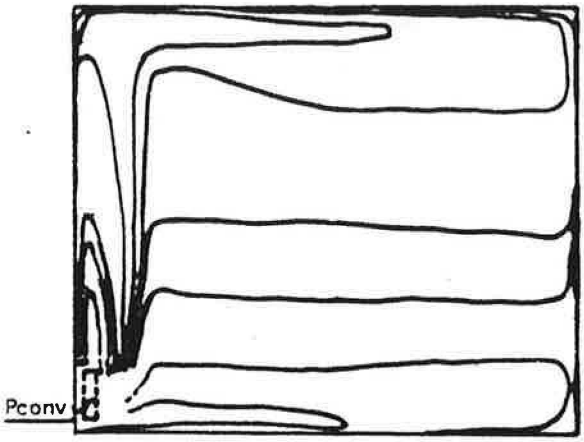


Fig. 8: Isotherm lines in the mid plane of a cell heated with a convective heat source (8).

So, the inside air volume is split into five zones representative of the circulation shown in Figure 9.

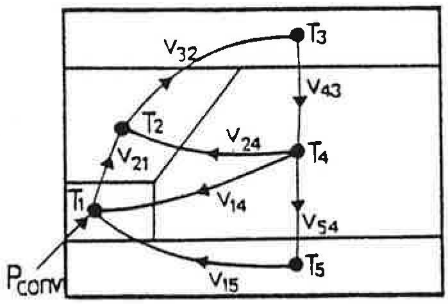


Fig. 9: Partition into five zones of a room heated with a radiator.

- zone 1 is representative of the heat source: the convective power is injected in this zone.
- zone 2 represents the heat source thermal plume.
- zone 3 is in contact with the ceiling.
- zone 4 represents the core of the cell, in contact with the vertical walls.

- zone 5 is in contact with the floor.

3.2.2. CONVECTIVE HEAT TRANSFER COEFFICIENTS

For the backwall, trail and ceiling heat transfer coefficients we use experimental results (see 3.1.2. and 3.1.3.). For the others surfaces, we use literature results:

Vertical walls (1) : $hc_v = 3.(T_4 - T_v)^{1/3}$ (W/m^2K)

Floor (7) : $hc_f = 1.$ (W/m^2K)

3.2.3. AIR MASS FLOW BETWEEN ZONES

From plume air mass flow rate expression (see 3.1.3.), we are able to calculate V_{21} and V_{32}. Mass balance equations allow us to calculate V_{43} and V_{24}.

Computing V_{14}, V_{54} and V_{15} is more difficult. We have three unknows and only two mass balance equations.

In a first approach, we identified the ratio $V_{15}/(V_{15}+V_{14})$ from experimental values of air temperature measured near the floor. We found a mean value of 0.05. This shows that in our experiments the air temperature near the floor and the floor surface temperature are close to each other. This can be explain by the weak cooling of the vertical walls and the no cooled floor. It is clear that the identified value can't be an universal one and more work is necessary in that way.

4. APPLICATION TO A DWELLING-CELL HEATED WITH HEATING FLOOR

The first step of the analysis is to define the air flow pattern. In that way, we carried out experiments with our experimental set up called Minibat with a heated floor and a vertical cold wall (8). Figure 10 shows isothermal lines measured in the vertical mid plane of the cavity.

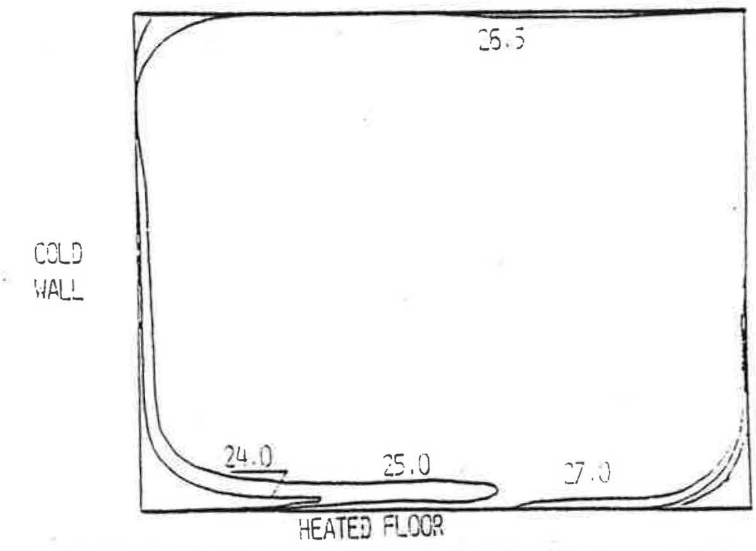


Fig. 10: Isothermal lines in the vertical mid plane of the cavity for a heated floor.

The global indoor air behavior can be described in a first approach by a large isothermal core connected to a cold vertical boundary layer flow along the facade and to a more disturbed region with higher gradients along the floor. If this experiment is useful to understand the thermal behavior of such configuration, we have no information about the dynamic behavior of the cavity. This is one of the reasons why we developed numerical simulations of natural convection flows in cavities in parallel with the experimental work.

4.1. NUMERICAL APPROACH

4.1.1. NUMERICAL MODEL

The governing equations are the Navier-Stokes equations written in their Boussinesq-Obberbeck approximation. The reference length is the height of the room, the reference time is obtained from the thermal diffusion inside the cavity ($t = H^2/a$), this gives also the scale of velocity ($v = a/H$).
These scales lead to a general writing of the equations in dimensionless form:

$$\mathrm{div}\ V = 0$$

$$\frac{dV_i}{dt} = - \frac{\partial P}{\partial x_i} + RaPr\delta_{i2}\,(T-0.5) + Pr\ \Delta V_i \quad (i=1,3)$$

$$\frac{dT}{dt} = \Delta T$$

The whole set of equations is then discretized using a finite difference scheme, and integrated over control volumes. The pressure is solved using SIMPLER procedure developed by Patankar (9). To ensure the conservation of all quantities in our model we used a staggered grid.
Even by using a sophisticated method, it is impossible to simulate by direct approach the behavior of a room at high Raleigh numbers, the time and space scales becomes too rich, and it is no more possible to have at the same time a good representation of the large structures of the flow, and a precise simulation of the small fluctuating eddies. The usual way is then to introduce a model representing the energy dissipation in the small structures and to look only at the general structures of the flow. The more common model is the so called k-ϵ model. The basic hypothesis of this model proposed by Boussinesq in 1877 is to represent the energy dissipation in the small structures of the flow by a turbulent viscosity which will be added to that of the fluid itself. This turbulent viscosity is then related to the turbulent kinetic energy k and its rate of dissipation ϵ.
Hence, we obtain a new set of transport-diffusion equations. In our turbulent model all the boundary conditions are defined at the very limit of the domain in order to avoid the use of wall functions. This solution requires of course a very precise description of the boundary layer, and we use a thiner grid with various nodes inside the laminar sublayer itself.

4.1.2. NUMERICAL RESULTS

In order to get better interpretation of our experimental results, we first simulate the steady state behavior of a two dimensionnal cell representing the vertical mid plane of our cavity. For this purpose we used the dimensionless temperature profiles obtained experimentally along the surfaces of our cavity as input in our numerical simulations. We show in figure 11 the isothermal lines and the streamlines obtained in simulating the behavior of the mid plane of floor-heated-room. The agreement with experimental data shown in figure 11 is quite satisfactory.

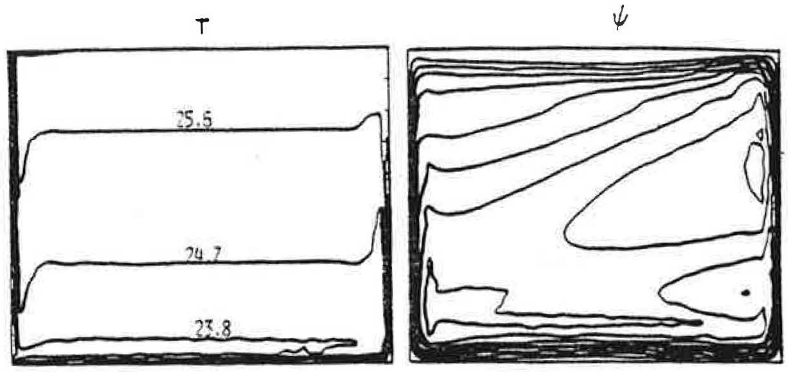


Fig. 11: Isotherms and streamlines (heated floor).

Concerning the streamlines, the cold air coming down from the cold boundary layer is heated by the floor as a forced convection flow transported upward by the main flow along the warm vertical wall and then entrained by a slow movement through the cavity. The higher velocities are located along the cold facade and along the floor. The boundary layer is alimented from the core of the cavity by a slow recirculation flow satisfying mass continuity.

4.2. MODELING WITH ZONAL MODEL

4.2.1. MODEL DESCRIPTION

From the experimental and numerical results, the indoor air volume is split into three isothermal zones, each one been representative of the flow pattern displayed:
- zone 1: cold boundary layer
- zone 2: flow along the floor
- zone 3: core of the cell

The Figure 12 gives the scheme of the flow pattern.
The air temperatures are the variables, we must evaluate convective heat transfer coefficients and the air mass flow between the zones.

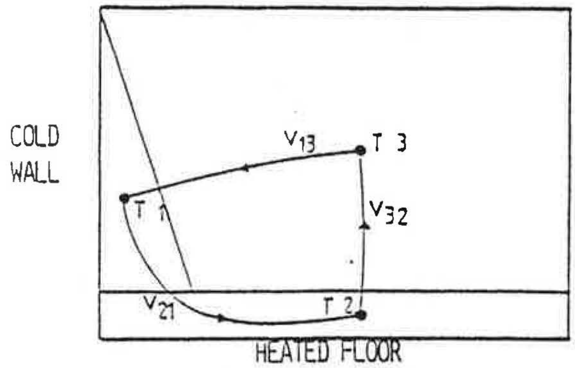


Fig. 12: Scheme of the flow pattern for a heated floor.

4.2.2. CONVECTIVE HEAT TRANSFER COEFFICIENTS

4.2.2.1. THE FLOOR

For an isolated horizontal plate, significant experimental results have been obtained by Fischenden and Saunders (10), and Fujii and Imura (11). The relationship between Nusselt and Rayleigh numbers is given by:

$$Nu = K\ Ra^{1/3},\ 10^{7} < Ra < 3.10^{10}$$

The 1/3 power law is characteristic of turbulent flow.
The K value is respectively 0.14 (10) and 0.13 (11). We can notice that this value is close to that obtained for a vertical plate (12). These results were confirmed by Bovy and Woelk (13) for larger plates (1 to 3 m length). Considering a K value of 0.14, we obtain:

$$hc_f = 1.65\ (Tw_f - T_3)^{1/3} \qquad (W/m^2K)$$

Experimental study closer to our configuration was carried out by Kirkpatrick and Bohn (14) on a scale model filled with water. Five faces were cooled and the floor was heated. The authors give the following relationship:

$$Nu = 0.346\ Ra^{0.285},\ 4.10^{9} < Ra < 5.10^{10}$$

which can be written as:

$$hc_f = 1.68\ (Tw_f - T_3)^{0.285} L^{-0.145} \qquad (W/m^2K)$$

L: length of the cell
Kast and Klan (15) reviewed several studies dealing with heating floor convective power, especially those carried out by Schlapmann (16) and Bach (17). Their conclusion was to recommend the following relationship:

$$hc_f = 1.5\ (Tw_f - T_3)^{1/3} \qquad (W/m^2K)$$

All these convective heat transfer coefficient expressions are close to each other. We use Kast and Klan (15) relationship because it deals with heated floor and it is gives a reasonable mean value of other expressions.

4.2.2.2. THE VERTICAL WALLS

We consider that the heated floor does not directly affect the flow along the vertical walls. So, we are dealing with natural convection boundary layers and we use experimental results obtained by Allard (18). In the case of one vertical cold surface and five hot surfaces, the author recommend the following relationship:

$$hc_v = 1.45 \ (T_3 - Tw_v)^{1/3} \qquad (W/m^2K)$$

4.2.2.3. THE CEILING

From the experimental and the numerical isothermal lines, we can see that the convective heat transfer with the ceiling is certainly weak. Furthermore, very few studies are dealing with this subject. So, we adopt 2 W/m^2K for the ceiling convective heat transfer coefficient.

4.2.3. AIR MASS FLOW BETWEEN ZONES

We make the choice of identifying the wall mass flow rate along vertical wall, then mass balance equations can be used to evaluate other mass flows. According to vertical walls convective heat transfer, we use Allard experimental results (18) obtained from cold boundary layer velocity and temperature measurements. The proposed relationship is:

$$Vbl(z) = 0.0083.10^{-3} Ra^{1/3}(z) \qquad (Kg/ms)$$

which can be expressed as:

$$Vbl(z) = 0.0039z(T_3 - Tw_v)^{1/3} \qquad (Kg/ms)$$

5. VENTILATION HEAT LOSSES

We take into account ventilation with a simplified method. We assume that air change does not affect directly the air flow patterns showed in Figures 9 and 12. From supply and exhaust opening locations, we arbitrarily define air flow scenario. Figure 13 gives an example for a cell heated with a radiator. The air supply is in the plume and the air exhaust near the floor.

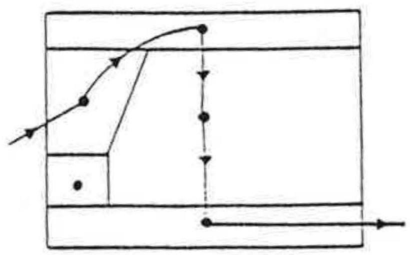


Fig. 13: Example of ventilation scenario.

We take into account ventilation effect in mass balance and energy equations.

6. RADIANT HEAT FLUXES CALCULATION

The radiant heat fluxes are computed with the radiosity method. For each surface, we write:

$$\phi rad = \frac{\epsilon}{1-\epsilon}(\sigma_o\, Tw^4 - J) \qquad (W/m^2)$$

The radiosities are calculated by solving the system:

$$\sum_j (\delta_{ij} - (1-\epsilon_i) F_{ij})\, J_j = \epsilon_i\, \sigma_o\, Tw_i^4$$

In a cell heated with a radiator, the view factors (F_{ij}) are computed with two assumptions:
- the radiator is split into two surfaces: the front and the rear surface
- the view factor between the radiator rear surface and the backwall is equal to unity.

So, the backwall radiant heat flux is given by:

$$\phi rad_b = \frac{\epsilon ra\ \epsilon_b\ \sigma_o\ (Tw_b^4 - Tra^4)}{\epsilon ra + \epsilon_b + \epsilon ra \epsilon_b} \qquad (W/m^2)$$

The knowledge of the radiosity of each surface enable us to calculate a radiant temperature at any point inside the room:

$$Tror = \left[\frac{\sum_i Fc_i\ J_i}{\sigma_o}\right]^{1/4} \qquad (K)$$

where Fc_i is the view factor between a spherical black body and surface i.

7. COMPARISON BETWEEN EXPERIMENTAL AND COMPUTED RESULTS

An iterative process combining convective, radiative balance and conduction inside the walls delivers internal surface temperatures and air temperatures under given external conditions. The calculation gives the total heating power (total, convective and radiative rates).

The validation of the models has been done with regard to experiments carried out by Marret at University of Liège (19). The experimental set up is a real scale room (4.74x3.45x2.70 m) with a facade in contact with ambient air (Tout = -3°C). Two facade insulation levels have been tested:
- Single Glazing (SG)
- Double Glazing (DG)

The ventilation air supplies were located around the glazing, simulating cracks, and the exhaust was done near the floor in the wall opposing facade. Several heating systems were tested, we are only dealing with single panel hot water radiator and heated floor. At last, Marret (19) has normalized all the experimental results with respect to an indoor-outdoor temperature difference of 25°C, the reference temperature were dry bulb temperatures at the center of the cell, and at 1.50m high.

7.1. RESULTS FOR SINGLE PANEL RADIATOR

The radiator is located beneath the window and the ventilation heat losses are taken into account according to Figure 13. An iterative process is used in order to valuate the radiator surface temperature in agreement with the dry bulb temperature value of 22°C.
Table 1 shows the results obtained concerning the total radiator heat Power (P) and the total convective power injected in the cell (Pconv + $\Phi conv_b$).

	Experimentation		Simulation	
	P (W)	Pconv + $\Phi conv_b$ (W)	P (W)	Pconv + $\Phi conv_b$ (W)
SG 0.ach	945	566	934	544
SG 1.2ach	1264	839	1298	775
DG 0.ach	565	348	558	336
DG 1.2ach	852	540	892	551

Table 1: Comparison between experimental and computed radiator heat powers.

Table 1 shows experimental and computed results agreement. Concerning the total heat power, the maximum gap is -4.7% (DG, 1.2 ach). The results are not so satisfactory for the convective power with a maximum gap of 7.6% (SG, 1.2 ach). This may be due to the uncertainties about convective correlations. On Figure 14, we show the experimental and the calculated air temperature profiles. The air temperatures T_3 and T_5 computed by the model have been arbitrarily located at the highest and lowest measured temperatures. The calculated air temperature T_4 represents an energy balance within the core, so we placed it at mid height of the cell. A linear air temperature profile is then assumed between (T_3,T_4) and (T_4,T_5).

The agreement between experimental and calulated temperatures is quite good except for floor surface temperature with ventilation.

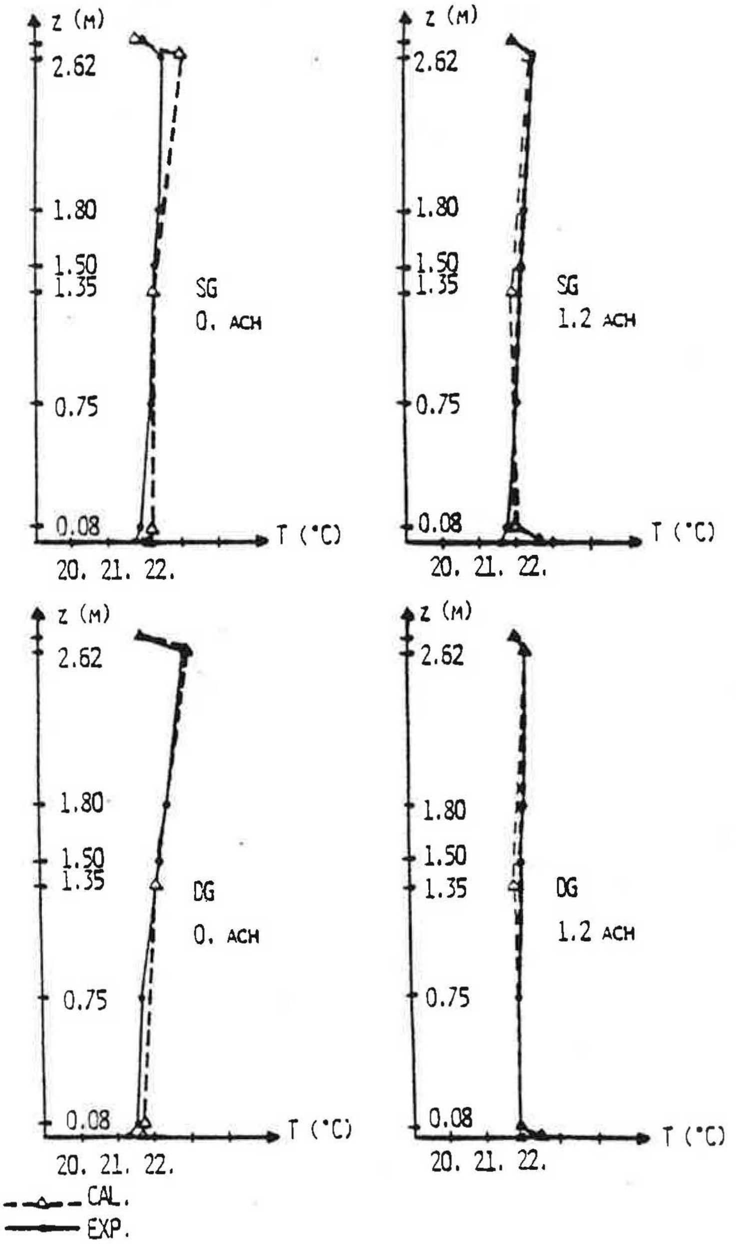


Fig. 14: Comparison between experimental and computed air temperature profiles (Single Panel Radiator).

7.2. RESULTS FOR HEATING FLOOR

An iterative process is used to evaluate the total heat power in agreement with the dry bulb temperature of 22°C. In this case, taking into account ventilation heat losses is not so easy. After several attempts, we imposed the following air infiltration pattern:

- 70% of air ventilation supplied in zone 1 (cold boundary layer)
- 30% of air ventilation supplied in zone 3 (core)
- 100% of air exhausted in zone 2 (floor)

Table 2 shows the results obtained for the floor total heat power (P) and the floor convective power (Pconv).

	Experimentation		Simulation	
	P (W)	Pconv (W)	P (W)	Pconv (W)
SG 0.ach	745	221	753	228
SG 1.2ach	1111	367	1167	454
DG 0.ach	351	35	345	87
DG 1.2ach	825	335	819	323

Table 2: Comparison between experimental and computed floor heat powers.

The maximum gap between experimental and calculated total heat power is 5% (SG, 1.2 ach). Ventilation increase convective exchange along the floor and leads to a greater convective power. Except for the weaker total heat power, the ratio between radiative power and total power is between 60% and 70% as shown by Kast and Klan (15).

Figure 15 shows experimental and calculated air temperature profiles. As for the radiator, the computed T_2 temperature has been located at the lowest measured temperature and T_3 temperature at mid height of the cell.

We can notice that the calculated air temperature T_2 which represents an energy balance of zone 2 is in agreement with air temperature measured at the center of the cell and at 0.08 m from the floor.

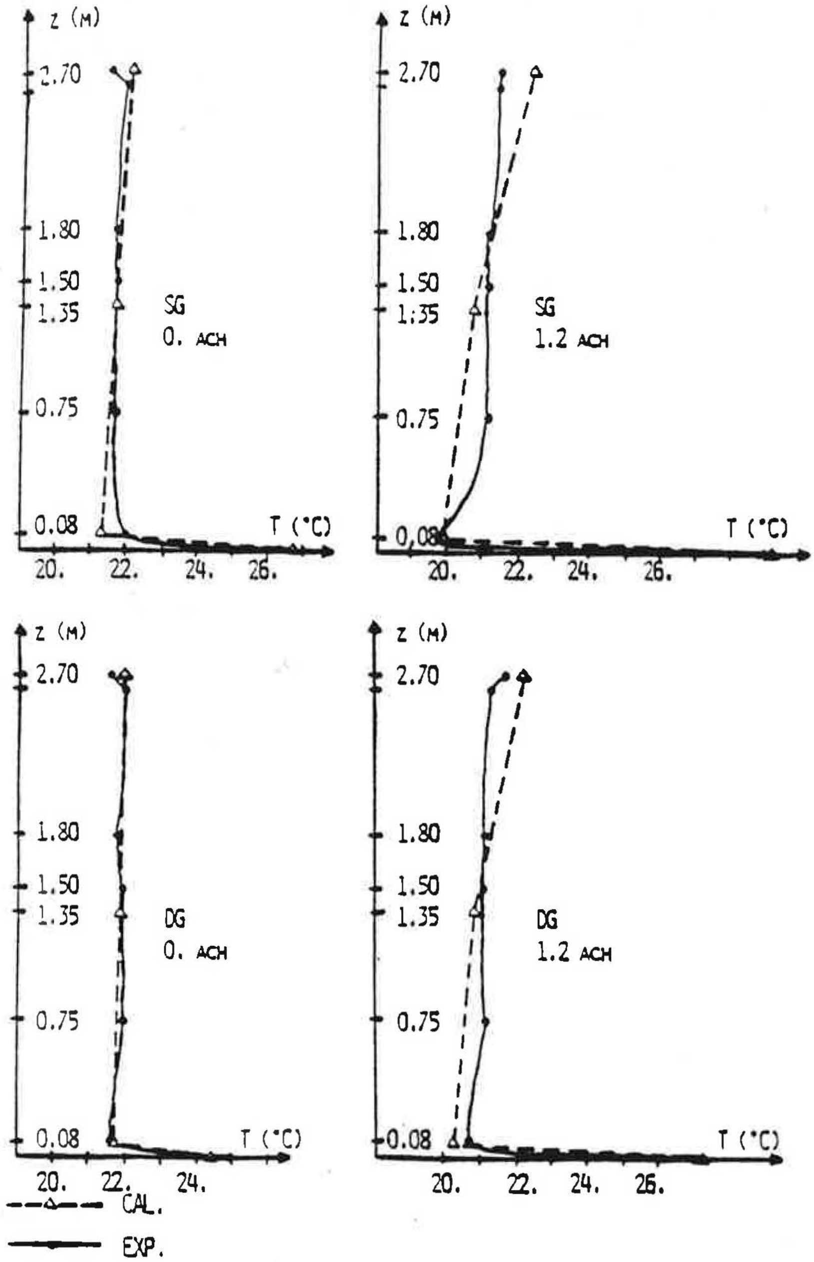


Fig. 15: Comparison between experimental and computed air temperature profiles (Heated Floor).

8. ARE ZONAL MODELS USEFUL?

In order to compare zonal models with more simplified models. We calculate the total power of the heating systems considering:

- a reference temperature
- global and constant values of heat transfer coefficients at the walls.

The reference temperature is 22°C and the heat transfer coefficients are those used by the french building regulation. Taking experimental results as reference, Tables 3 and 4 show the values of the ratio between zonal models results and experimental results (Pzonal/Pexp.), and the ratio between isothermal model results and experimental results (Pisot./Pexp.).

Single Panel Radiator	Pzonal/Pexp.	Pisot./Pexp.
SG 0.ach	0.99	0.86
SG 1.2ach	1.03	0.96
DG 0.ach	0.99	0.92
DG 1.2ach	1.05	1.01

Table 3: Comparison between zonal models and isothermal model.

Heating Floor	Pzonal/Pexp.	Pisot./Pexp.
SG 0.ach	1.01	1.01
SG 1.2ach	1.05	1.06
DG 0.ach	0.98	0.99
DG 1.2ach	0.99	1.01

Table 4: Comparison between zonal models and isothermal model.

Concerning single panel radiators without ventilation, zonal model gives better results than isothermal model. Furthermore, simplified model underpredicts total heat power. In this case, zonal model allows to take into account direct thermal coupling between heating systems and the wall (trail, backwall...). The gap between the two models decreases when insulation level

increases. With ventilation, the results are not so clear. We think that zonal models are always suitable but the way to take into account ventilation is perhaps too simplified.
With heating floor, the gap between zonal and simplified models is very low in all cases. Furthermore, except for one test (DG, 1.2 ach), the total heat power computed are very close to experimental ones. As a result, it seems that sufficient accuracy in predicting total heating power under steady state conditions with an isothermal model is obtained, concerning this kind of heating system.

9. CONCLUSION

From experimental and numerical results, we built zonal models dealing with dwelling-cells heated with radiator and heating floor system.
Comparison between experimental and computed heat powers and air temperature profiles show good agreement.
The results have also been compared with those issued from an isothermal simplified model. We have shown the ability of zonal models to predict heat losses due to direct thermal coupling between the heating system and the walls, and the influence of the ventilation.
Until now, this study was limited to steady state conditions. Laret (2) showed that zonal models give better results than isothermal models in unsteady conditions. That is the next step of our study.

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DISCUSSION

HUTTER E. (France)

Comparisons between zonal models and isothermal models show that the first are better than the second when there is no ventilation but the prediction of the total power is quite the same with ventilation. What are, from your point of view, the possible ways of improvement of ventilation "scenario" or parameters ?

ANSWER :

A better way to take into account ventilation is to introduce ventilation momentum in the model. Presently, in the models we have developed, it is not possible. One way of investigation could be to mix zonal models as we have presented and zonal models which compute air mass flow from the relation :

$$Q = K \, (\Delta P)^n$$

with Q : volumetric air flow
K : permeability
ΔP : pressure difference

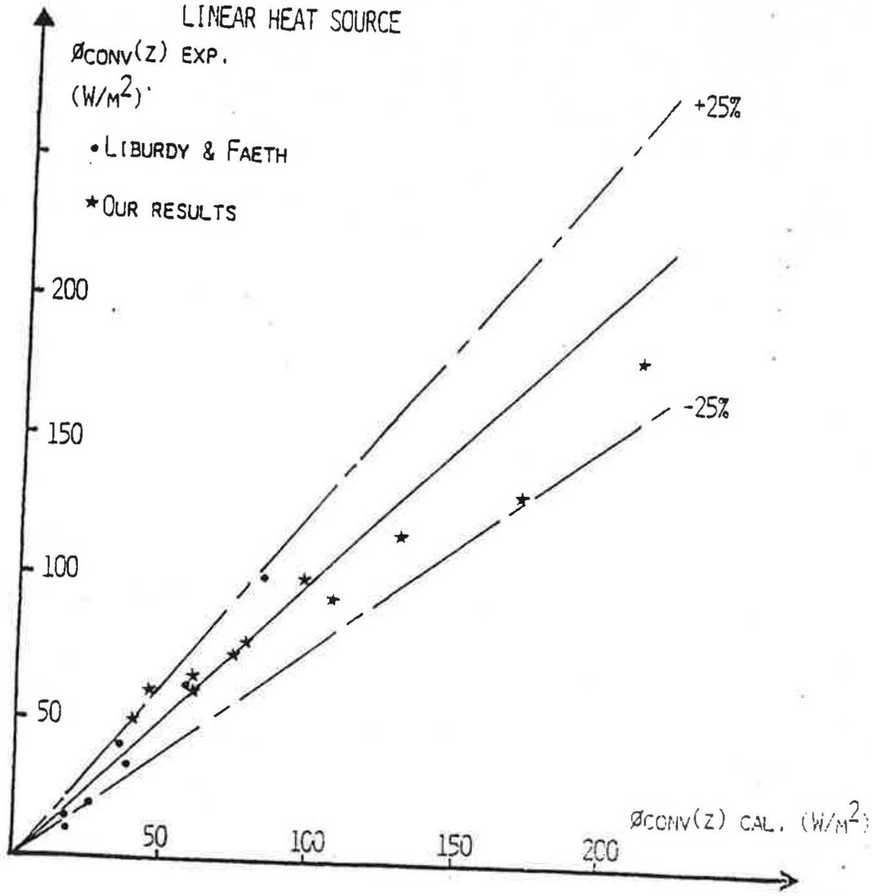
One problem is to determine the permeability K between two zones. Furthermore, the idea is, on the one hand, to use the knowledge about specific air flows like thermal plumes or boundary layers and, on the other hand, to compute air mass flows in the central zone from pressure differences. Then, ventilation can be introduced by either a volumetric air flow either a pressure difference, for example with a fan.

SUTER P. (Switzerland)

It seems that the extremely good comparison between best and zonal models may be due to the fact that here the transfer coefficients used in the model come from direct measurements concerning the same situation, so it is not astonishing. Tests for different cases must be included, e.g. in the frame of Annex 20.

ANSWER :

Concerning the thermal plume of the radiator, integral analysis which allows us to get convective heat transfer along the trail, we think that the results are independent of the situation as shown on figure 1. This figure shows local convective heat fluxes that we measured (see ref. 5) and computed.



<u>Figure 1</u> : Local convective heat fluxes along the trail of a linear heat source.

The experimental local convective values were obtained from the slope of the air temperature profiles very close to the wall, i.e. in the viscous sublayer. Furthermore, on this figure we superimposed values measured by Liburdy and Faeth (1) with heat flux gases.

For vertical walls, we used experimental correlations from Lebrun (see ref.1). In the frame of Annex 20, we tested several zonal models developed by Howarth (see ref. 4), Laret (see ref. 2) and Inard (2).

The results obtained are in agreement with k-ε low Reynolds number code developed by Chen (3). The comparison was made on air temperature profiles and convective heat fluxes. In all cases, the thermal boundary conditions are very close : one façade in contact with ambiant temperature and other walls very insulated. So, we think that zonal models must be validated with regard to other thermal boundary conditions.

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MADJIDI M. (Germany)

If you introduce some furniture pieces, do you think that the calculation with zonal models would still be less complicated than F.E.-method calculations ?

ANSWER :

For the two heating systems we studied and in steady-state, we think that if the furniture pieces do not affect directly convective heat emission or convective heat exchanges, they could be quite easily taken into account, because they don't disturb the air flow pattern we adopted. Otherwise, specific studies about thermoconvective behaviour must be carried out and calculation can be more complicated.

KOHONEN R. (Finland)

Until now, you have studied steady-state problems. Do you see that it could be possible to apply your approach in energy calculations using one hour time steps ?

ANSWER :

The dynamic behaviour of the indoor air volume can briefly be split into three parts :

- time constants of flows near the walls (thermal plume, boundary layers, ...) that are of the order of a few seconds;

- gradient establishment in the cell, around half an hour, depending on air indoor volume;
- variation of air temperature due to convective coupling between air and walls. In this case, the phenomena can be very long (several hours) and depend on the walls inertia.

So, the time step chosen in the simulation must take into account the phenomena observed considering the Shannon theorem. At last, we must consider numerical stability problems due to higher time steps which are important when simulating with zonal models.