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MODELLING HEAT EXCHANGES BETWEEN A RADIATOR AND A DWELLING CELL IN STEADY STATE CONDITIONS

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1- INTRODUCTION.

Classical methods for dimensioning heating systems state that the ambient air is at uniform temperature and assume that convective coefficient do not depend on the heating system.

These assumptions do not allow realistic comparisons of thermal comfort given by different heating systems.

The aim of this study has been to undertake these comparisons by developping an anisothermic steady state model of heat transfer inside a heated room.

2- MODELLING PRINCIPLES.

The inside air volume is split into several isothermal zones, coupled with massic conductances. The type of partition must be as close as possible of the physic actuality. Experimental studies [1] to [4] show that, in most cases, the air flows have a main circulation : 1. radiator, 2. trail, 3. ceiling, 4. vertical walls, 5. floor, 1. radiator. So, the inside air volume is cut out into five isothermal zones which each is representative of this circulation as shown in figure 1.

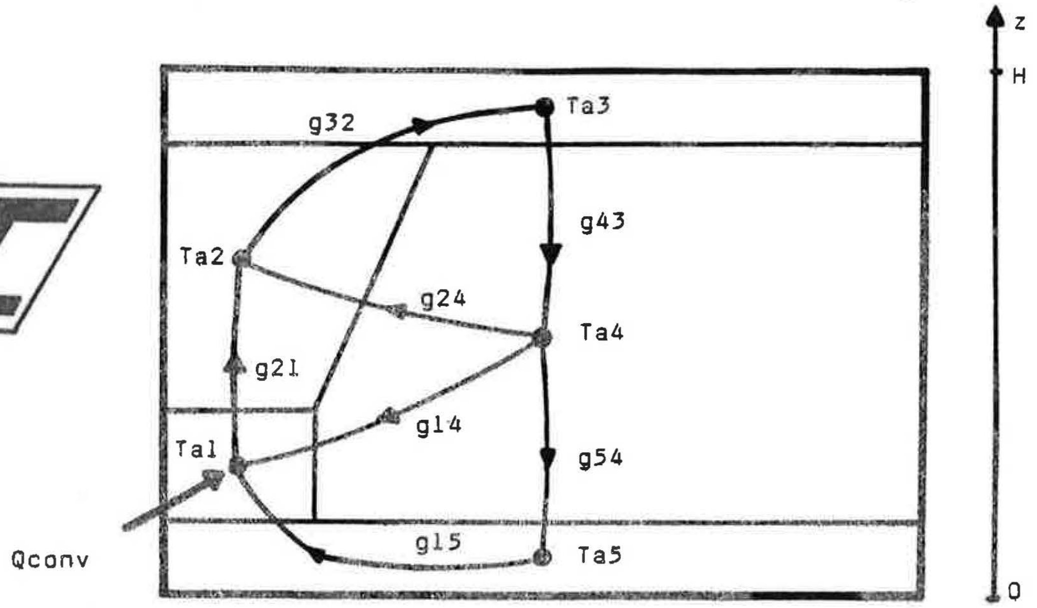


Figure 1 - Partition into five zones of a room heated with a radiator

The variables are the air temperatures of the zones. The boundary conditions are the external temperatures and the mean surface temperature of the emittor.

To complete the description of the model, the emittor convective heating capacity, the different convective heat transfer coefficients, and three massic conductances must be given or identified. Experimentations in climatic test rooms with steady state conditions allow us to get these data.

3- EXPERIMENTAL PROGRAM.

3.1 Methodology.

The tests were performed in climatic test rooms [5] on different radiators and on one electric linear heat source. Two kinds of measurements were done :

- Surface temperature inside and outside the walls, to measure conductives rates, and compute radiative heat exchanges between walls and the radiator.

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

3.2 Emittor convective heating capacity.

From the experimentation, it was found that convective heating capacity can be expressed as :

$$Qconv = K\,(Tsrad - Ta)^{\alpha} \qquad \text{(W)}$$

α being close to 1.25, characteristic of laminar flow, compatible with Rayleigh numbers, based on radiator height, in the range of 2.10^9.

As an example, figure 2 shows the experimental correlations between Qconv and ΔT obtainded from 4 different panel radiators.

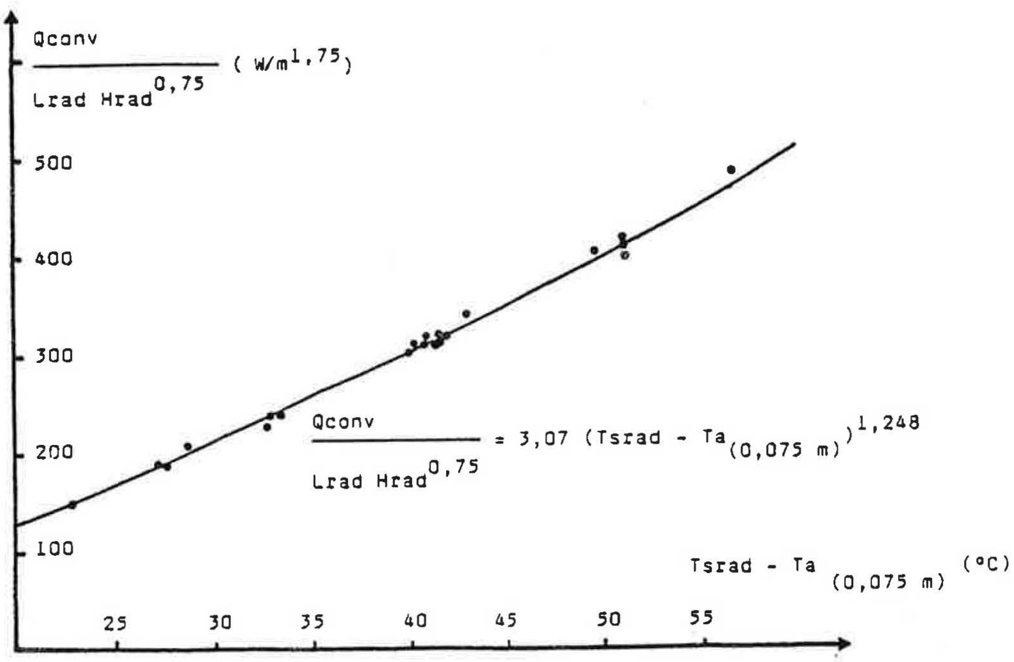


Figure 2 - Relationship between Qconv and ΔT for panel radiators

3.3 Plume characteristics.

The thermal balance inside the plume is written :

$$\frac{dQ(z)}{dz} = -\varphi_{conv2}(z) - C_p V(z) \frac{dT_\infty}{dz} \qquad (1)$$

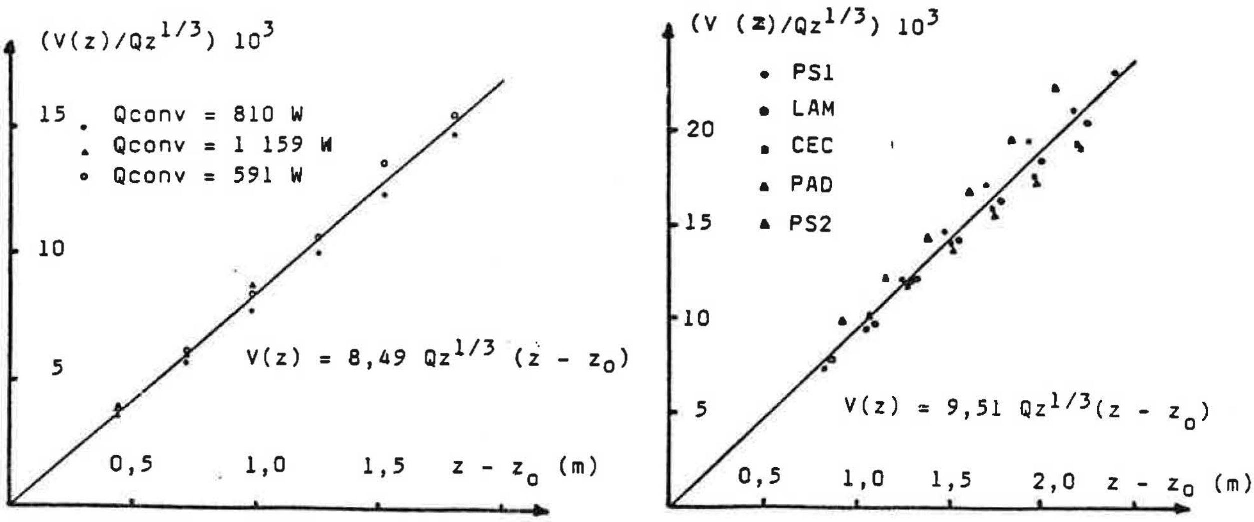
where : Q (z) is the heat flow inside the plume, function of the height z

$\varphi_{conv2}(z)$ is the convective heat transfer to the wall

V (z) is the air mass flow in the plume

dT_∞/dz is the stratification of the ambiant air

Figure 3 gives the correlations obtained for the air mass flow in function of the heigh, for the radiators and for the linear heat source.



Linear heat source Radiators

Figure 3 - air mass flow in the plume of different emittors

It is interesting to notice that air mass flow in the plume of a radiator is similar to that of a linear source. This was already reported by Lebrun and Marret [6]. The general equation is the following :

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

where zo is the fictitious origin of an equivalent linear source, zo is experimentally identified.

Moreover, for sufficiently large Reynolds number values, a local similarity of the velocity and temperature profiles has been observed. From an integral analysis of the plumes, the dimensionless quantities are :

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

$$(Tm - T_\infty)^* = (Tm - T_\infty) / \frac{g\beta}{\rho Cp}^{2/3} \frac{Q(z)}{g\beta(z - zo)}^{2/3} = k\, Fr^{-2/3}$$

where : Um and Tm are the maximum values of the velocity and the temperature in the plume

Fr is the Froude number

k is function of the dimensionless integrals and the entrainment constant of the plume.

At last, the convective heat flux to the wall is expressed by the dimensionless Stanton number :

$$St = \frac{\varphi_{conv2}}{\rho Cp\, Um\, (Tm - Ts2)}$$

The table 1 slows the mean values of the quantities obtained in our experiments.

	Radiators [5]	Linear source [5]	Free linear source [7]	Linear source along an adiabatic wall [8]	Linear source along isothermal wall [9]
Um*	2,86	2,88	2,27	3,09	2,81
$(Tm - T_\infty)^*$	5,40	6,44	4,12	7,91	5,37
Fr	5,13	4,87	3,42	5,42	4,70
St	0,0075	0,0095	0	0	0,0090
k	16,0	18,5	9,36	24,4	15,1

Table 1 - Plumes dimensionless numbers

From all these results, we can solve analytically equation (1) which in turn allows us to get a relationship for the convective heat flux to the wall (eq. 2) :

$$\varphi_{conv2}(z) = \alpha 1\, (T_\infty - Ts2)\, Q^{1/3}(z) + \alpha 2 \frac{Q(z)}{z - zo} \quad (W/m^2) \qquad (2)$$

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

$\alpha 2 = kSt$

Therefore, the plume over a radiator has a complete description.

3.4 Convective heat transfer to the walls.

As examples, the figures 4 and 5 show the convective fluxes obtained at the backwall of the radiators and at the ceiling.

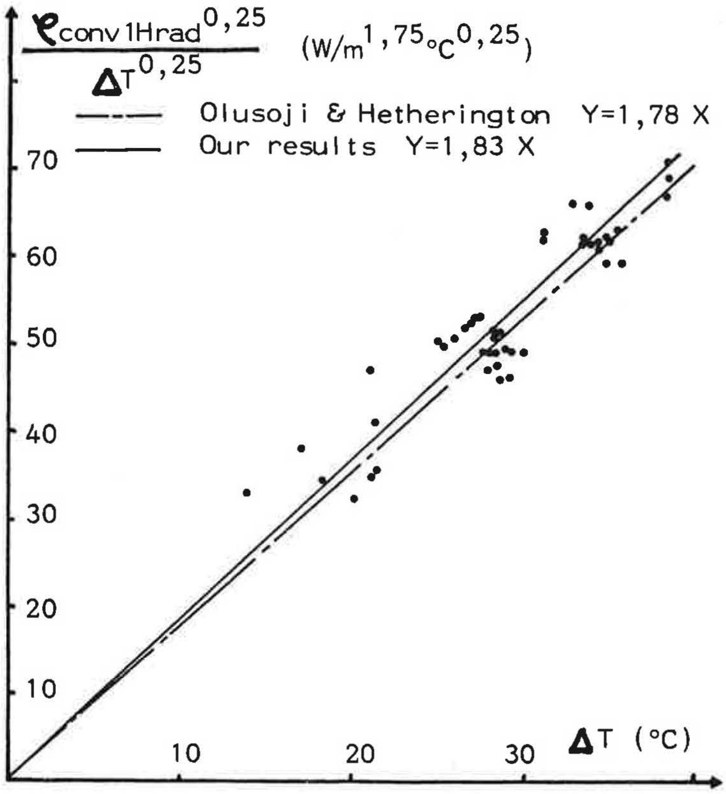


Figure 4 - Convective heat flux at the backwall of the radiators

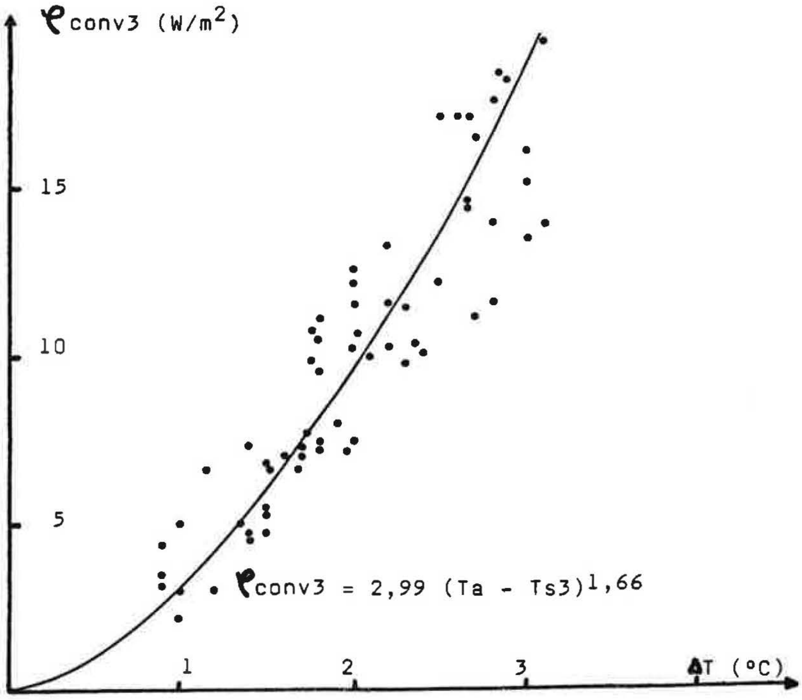


Figure 5 - Convective heat flux at the ceiling for the radiator

For the backwall, the temperature difference (ΔT) is built with the mean surface temperature of the wall and the radiator, and the air temperature measured at 7,5 cm of the floor.

On the figure 4, we have also reported the results obtained by Olusoji and Hetherington [10] for an open vertical chanel heated at different surface temperature.

In the case of the ceiling, ΔT represent the difference between the mean surface temperature of the ceiling and the air temperature measured at 7,5 cm of this surface.

4- VALIDATION OF THE MODEL.

The convective balance inside the room is computed together with a radiative balance based on the method of radiosities, by using an iterative process.

The ratio g15/g14 has been identified with the experimental values of Ta5. We have found a mean value of 0,05. This shows that in our experiments the floor has a very low convective exchange which can be explain by the lack of ventilation and the weak cooling of the vertical walls.

Taking into account conduction inside the walls gives access to internal surface temperatures for given external conditions. By this way, comfort level, assumed to be function of the air temperature and mean radiant temperature, can be computed at any point inside the room. The calculation also gives the emittor total heating capacity for the simulated conditions, which can differ slightly from standard test procedures, and sets apart convective and radiative ratios.

Table 2 gives experimental and calculated data for emittor heating capacity (total and convective), and air and dry resultant temperatures at 1,50 m in eight different cases. Differences appear to be very small (1 % for powers and 0,2 °C for temperatures)

	CALCULATION				EXPERIMENTATION			
	Q (W)	Qconv (W)	Ta (°C)	TRS (°C)	Q (W)	Qconv (W)	Ta (°C)	TRS (°C)
Lamella type radiator (LAM)	1020 790	692 536	24,0 22,9	23,7 22,7	1018 790	689 535	24,0 23,0	23,5 22,6
Panel radiator (PS1)	900 696	380 293	23,0 22,1	23,0 22,1	896 696	380 295	23,2 22,3	23,0 22,1
Panel radiator (PS2)	1136 883	493 387	24,5 23,4	24,3 23,3	1138 884	499 387	24,3 23,3	23,9 23,0
Double panel radiator (PAD)	1087 862	716 570	24,3 23,1	24,2 23,0	1092 865	720 572	24,5 23,4	23,9 22,9

Table 2 - Comparison between experimental and calculated data

The experimental and calculated air gradients inside the room are reported on figures 6 and 7. The mean temperature difference is about 0,3 °C.

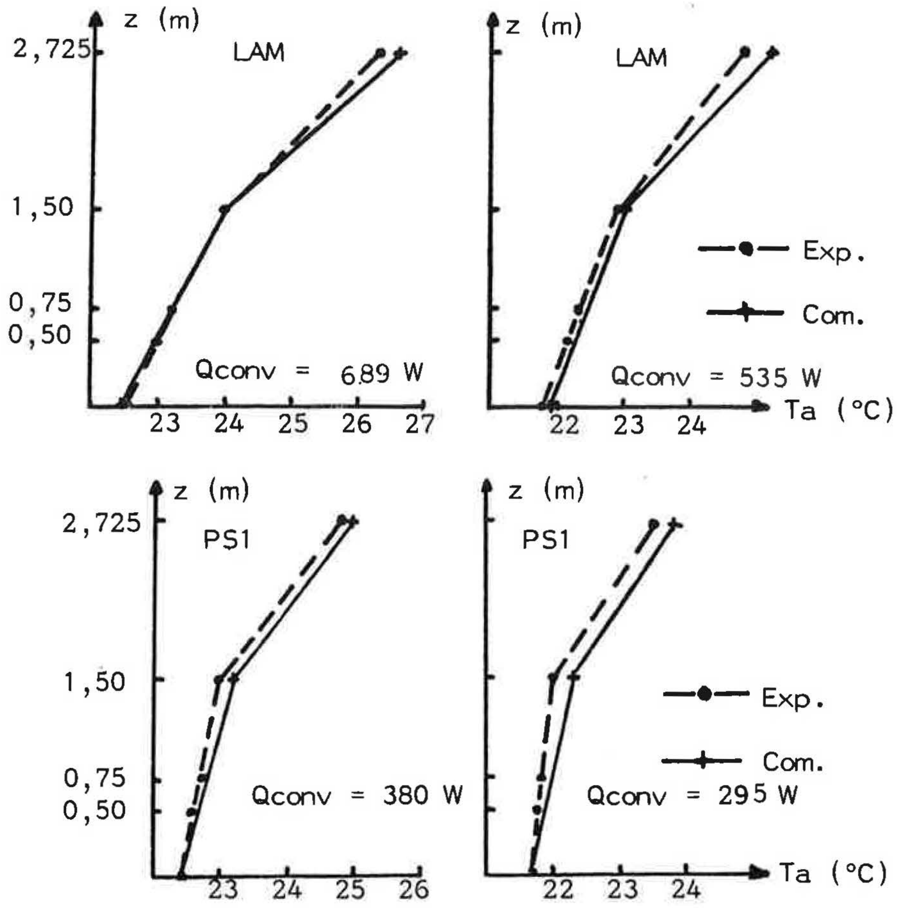


Figure 6 - Comparison of air temperature profiles (LAM, PS1)

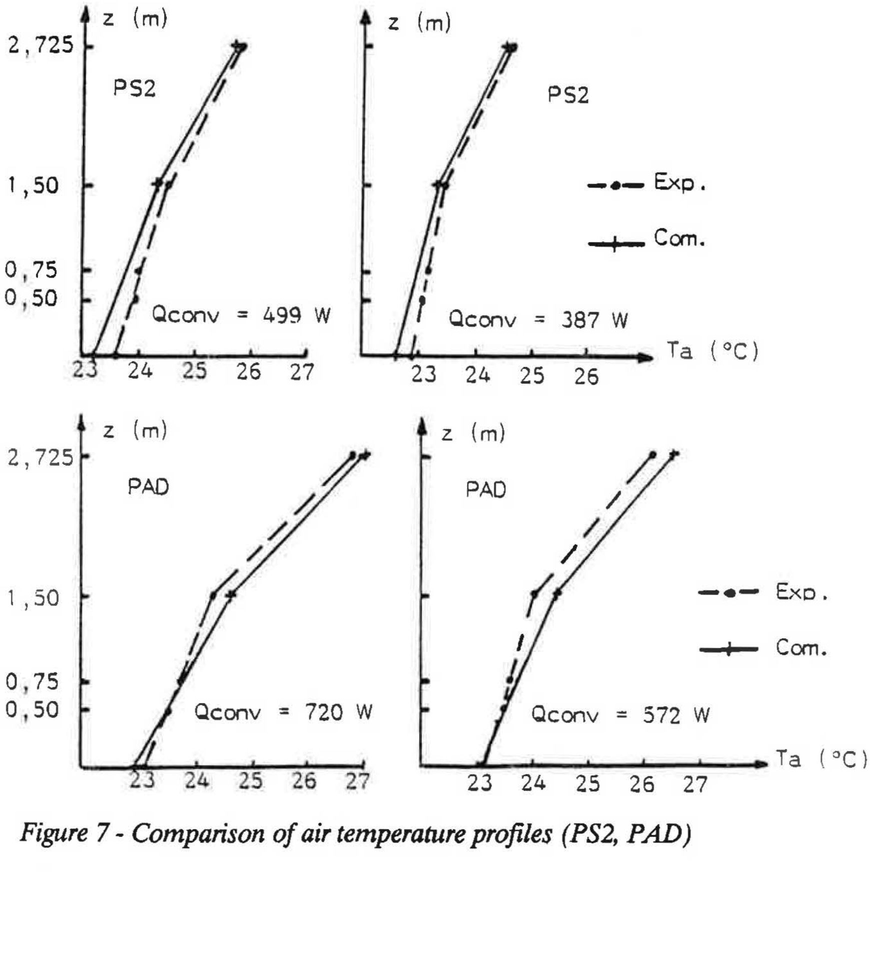
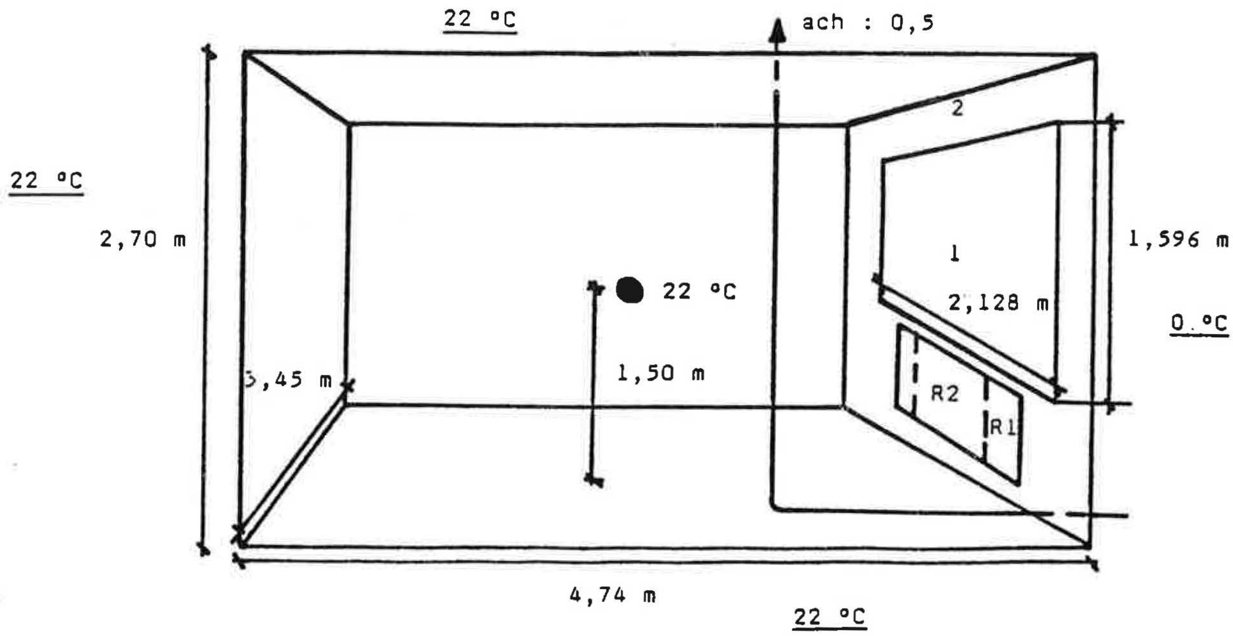


Figure 7 - Comparison of air temperature profiles (PS2, PAD)

5- APPLICATION.

A case study is described on figure 8. Simulations were performed, aiming at comparing a panel radiator (R1) and a lamella type radiator (R2).



R1 : panel radiator (0,75 x 1,55 m)

R2 : lamella type radiator (0,75x0,62 m)

1 : U_1 = 5,79 W/m^2 °C (single glazing)

2 : U_2 = 1,39 W/m^2 °C

Figure 8 - Simulated room and boundary conditions

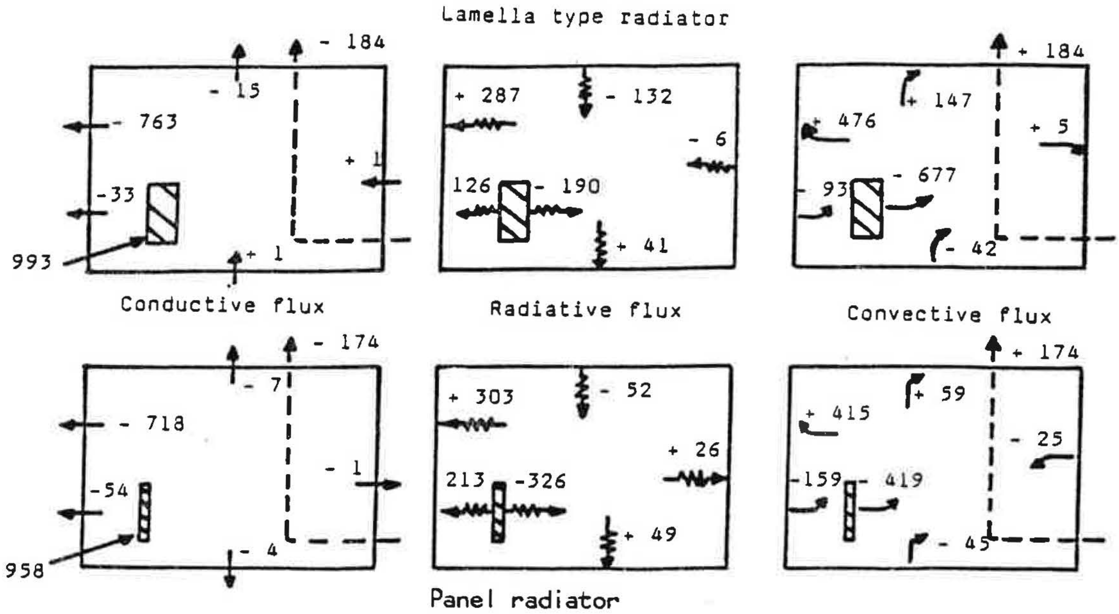


Figure 9 - Heat rates on wall surfaces and out of the emittor

Figure 9 shows the heat rates on the wall surfaces, and out of the emittors. It can be seen that for given external conditions, a resultant temperature of 22 °C at 1,50 meter in the center of the room is obtained with different heat rates 993 W with the lamella type radiator versus 958 W with the panel radiator.

6- CONCLUSION.

The model which has been developped gives satisfactory results, compared to experimentation. Such realistic description of convective flows inside a heated room allows the model to be used for estimating comfort level and heating efficiency of radiators for given climatic conditions, building enveloppe insulation level, ventilation rates, emittor dimensioning and position inside the room, and so on ...

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