ANALYSIS OF ROOM DISPLACEMENT VENTILATION IN THE PRESENCE OF SEVERAL HEAT SOURCES

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ABSTRACT

An analysis of the flow and temperature field patterns induced by several heat sources of different loads in a room equipped with a displacement ventilation system were performed in order to enhance the abilities of a displacement ventilation simplified model (called VENDA), which is to be used as a dimensioning tool.

This paper presents temperature measurements in a room equipped with a displacement ventilation system and in the presence of two heat sources. This room is an experimental facility where temperature and mass rate of the fresh air diffuser and both heat sources are controlled.

VENDA model is obtained by integrating the energy conservation equation in a horizontal plane. It includes plume and radiation modeling. It is implemented in the building simulation software Clim2000.

Experimental results prove that modeling several heat loads by a single one could give unaccurate results.

Measurement results were compared to the VENDA calculations. It is shown that the model is able to represent the temperature vertical profile induced by several heat sources of different loads.

KEYWORDS

Displacement ventilation
Experiments
Modeling
Plumes
Thermal stratification

INTRODUCTION

A simplified thermal model of a room equipped with a displacement ventilation system was developed in order to get an easy-to-use dimensioning tool for displacement ventilation systems. This model ,called VENDA, is integrated in the building simulation software Clim2000.

VENDA calculations were previously compared with experimental data in the case of a room with a single heat load (Manzoni, 1997). Since, it was improved, and presently, radiative heat transfer between walls as well as the presence of several heat sources in the room are taken into account.

In this paper, we point out the perturbation induced by the presence of a supplementary heat load and we evaluate the model ability to predict the thermal field in the presence of several heat sources. Experimental results are analyzed and compared to VENDA calculations.

METHODS

Test room description

The test facility is set-up in a large air-conditioned room of 400 m^3. It is mainly composed of a test room which is a parallellepipedic enclosure (Figure 1).

Heat sources are obtained using two independent and thermally controlled low velocity air jets located at the bottom of the enclosure. In these conditions the rate of convective heat release of each source can be easily determined. The area of each inlet is 0.0324 m^2. Flowrate and load could vary in the following range :

- 0 kW< load <4.5 kW

- $0\ m^3/s <$ flow rate $< 0.056\ m^3/s$

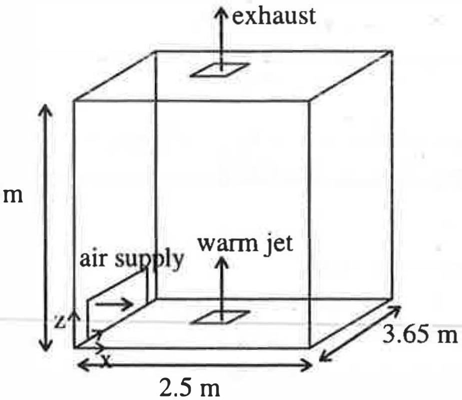

Figure 1 : test room

Fresh air supply is located at the bottom of a vertical wall. The inlet is 0.5 high and 3.4m long. Fresh air temperature is controlled by a water-air cooler in the range of 15-25°C. Air is discharged through grids at a low (about 0.05 m/s) and uniform velocity. Flowrate could reach up to $0.083\ m^3/s$.

The exhaust is a 20cmx20cm square opening located in the center of the ceiling.

Walls are thermally insulated with a 3 cm thick insulating material covered on both sides with an aluminum film in order to minimize radiation. The heat loss coefficient is equal to $0.38 \pm 0.05 W/m^2K$.

Velocity and temperature were simultaneously performed using respectively a two-component Laser Doppler Velocimetry device and a 250 µm diameter K-thermocouple. Measurements. on a 2500 point 3D-grid inside the room. .

Model description

Principles and equations :

VENDA model allows to determine the thermal stratification inside a room equipped with a displacement ventilation system. It takes into account convective, conductive and radiative heat transfer.

Assuming that the isothermal lines are nearly horizontal inside the room, air temperature $T(x,y,z,t)$ (in the cartesian coordinate system represented in Figure 1) varies little along x and y axis (but the plume zone), the following method was implemented :

- the air temperature $T(x,y,z,t)$ is integrated on horizontal planes $\Sigma(z)$ (apart from the plume zone) to obtain the average temperature $\theta(z,t)$;
- integrating the energy conservation equation (1) on $\Sigma(z)$ we use equation (2) for the determination of $\theta(z,t)$.

$$\rho_0 C_p \partial_t T(x,y,z,t) + \rho_0 C_p V \cdot \nabla T = \lambda \Delta T \qquad (1)$$

where T = local air temperature
V = local air velocity
ρ_0 = air density
Cp = air heat capacity
λ = air thermal conductivity

$$\begin{aligned} &\rho_0 C_p |\Sigma| \partial_t \theta(z,t) - \lambda |\Sigma| \Delta\theta \\ &+ \rho_0 C_p \partial_z (A\theta(z,t)) + \rho_0 C_p B \\ &= h.Per.(\theta_w - \theta) \end{aligned} \qquad (2)$$

where θ = average air temperature in the horizontal plane $\Sigma(z)$
θ_w = average wall temperature at height z
$|\Sigma|$ = $\Sigma(z)$ area
h = wall convective heat transfer coefficient

Terms A and B depend on boundary conditions (including entrainment by the plume) and are obtained by integrating the mass conservation equation in the plane $\Sigma(z)$:

$$\partial_z A = -\int_{\partial\Sigma} V.nds \qquad (3)$$

$$B = \int_{\partial\Sigma} TV.nds \qquad (4)$$

In the same way, we obtain the averaged wall surface temperature $\theta_w(z,t)$ from the following equation†:

$$\rho_w C_{pw} \cdot e_w \cdot \partial_t \theta_w(z,t) - \lambda_w e_w \Delta\theta_w = h(\theta - \theta_w) + U(T_{out} - \theta_w) \qquad (5)$$

where e_w = wall thickness

U = heat loss coefficient

T_{out} = air temperature outside the room

We obtain a partial differential equation system with eight variables : the air temperature, four wall temperatures and three quantities describing the plume (see below). This system is solved by applying a finite difference method using an implicit scheme.

Plume modeling :

The plume is supposed to develop over a point source and is then considered as axisymmetric. Furthermore, we assume that the temperature and velocity profiles are auto-similar (gaussian profile) :

$$\begin{cases} W(r,z) = W_0(z) e^{-\left(\frac{r}{R(z)}\right)^2} \\ \Delta\theta(r,z) = \Delta\theta_0(z) e^{-\left(\frac{r}{\tilde{\lambda} R(z)}\right)^2} \end{cases} \qquad (5)$$

where W = vertical velocity

$\Delta\theta$= temperature difference between plume and surroundings.

R = plume radius

Applying equations (5) to the mass, momentum (with Boussinesq hypothesis) and energy conservation equations and integrating them (Morton 1956), we obtain the following equations :

$$\begin{cases} \dfrac{d(R^2 W_0)}{dz} = 2\alpha R W_0 \\ \dfrac{d(R^2 W_0^2)}{dz} = 2g\beta\tilde{\lambda}^2 R^2 \delta\theta_0 \\ \dfrac{d(R^2 W_0 \delta\theta_0)}{dz} = -\dfrac{1+\tilde{\lambda}^2}{\tilde{\lambda}^2} \dfrac{dT_\infty}{dz} R^2 W_0 \end{cases} \qquad (6)$$

where α = plume entrainment factor

β = thermal expansion coefficient

T_∞ = surroundingtemperature

g = gravitational acceleration

Radiation modeling :

The radiosity method is used to calculate radiative heat transfer between walls

Radiosities are solution of the linear system (7) :

$$\left(I - \left[\rho_i F_{ij}\right]\right)\left[J_j\right] = \sigma\left[\varepsilon_i T_i^4\right] \qquad (7)$$

where J_i = radiosity of area n°i

F_{ij} = shape factor from wall element n°i to wall element n°j

ε_i = emissivity of wall element i

ρ_i = reflectivity of wall elementi

σ = Stefan-Boltzmann constant

Shape factors F_{ij} are calculated analytically (Siegel,1985).

Then the net radiative flux is given by :

$$\Phi_i^{net} = \frac{\varepsilon_i}{1-\varepsilon_i} S_i\left(\sigma T_i^4 - J_i\right) \qquad (8)$$

where S_i = surface of wall element i.

The influence of radiative heat transfer on the calculation results is as shown in Figure 2. We can observe a better agreement with the experimental results specially in the lower part of the room when radiation is considered.

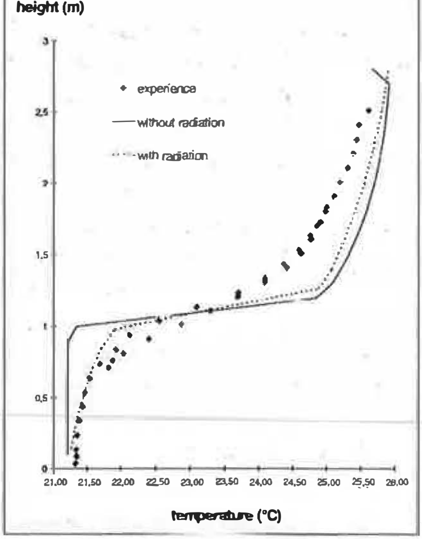

Figure 2 Case 3 -Influence of radiative heat transfer.

RESULTS

Experimental study

Four experimental were studied and are described in table 1. Case 1 has only one heat source, the other cases have two heat sources. Characteristics of the first heat source for cases 2 to 4 are constant.

Table 1 Experimental cases

case	1	2	3	4
fresh air flow rate (m^3/h)	486	486	486	486
T_{diff} (°C)	21.1	21.2	21.2	21.2
Flowrate (m^3/h)				
source 1	140	95	95	95
source 2		33	59	33
Total	140	136	149	128
heat release (W)				
source1	1200	870	770	800
source2		270	310	170
Total	1200	1140	1080	970

are averaged vertical temperature profiles.

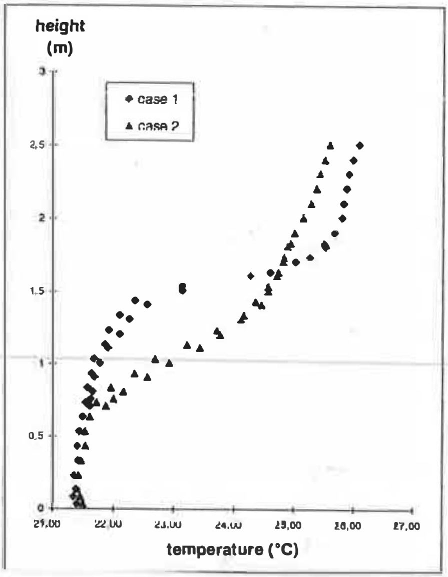

Figure 3 Vertical temperature profile for case 1 (single source) and case 2 (2 sources)

Comparison of temperature profiles between case1 and case 2 (Figure 3) shows that the plumes induced by two heat sources have not the same behaviour than the plume induced by a single source with the same global heat release. This is quite normal since plume equations are non-linear.

Applied to displacement ventilation, in configurations with several plumes, a "single plume" modeling could induce important errors.

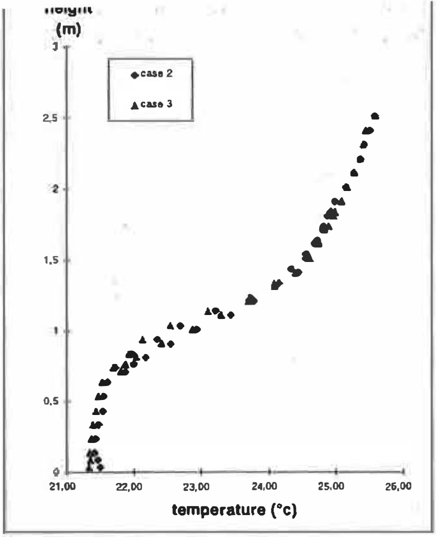

Figure 4 Influence of the second source flowrate. Case 2 : 33 m^3/h
Case 3 : 59 m^3/h

Comparison of temperature profiles of cases 2 and 3 (Figure 4) points out the influence of the second source flowrate for a given heat release. Results show that a variation of the second source flowrate from 33 to 59 m3/h does not affect the thermal stratification.

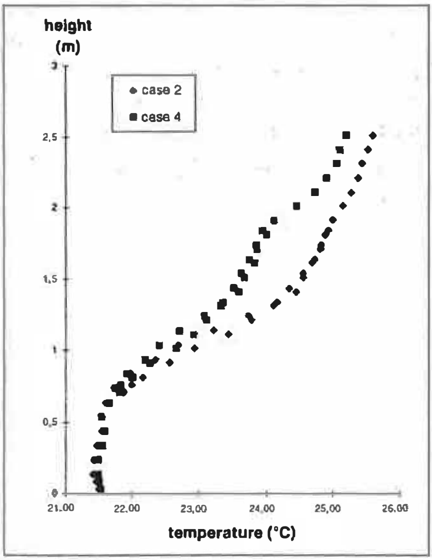

Figure 5 Influence of the second source heat release. Case 2 : 1140W
Case 4 : 970W

Comparison of temperature profiles of cases 2 and 4 (Figure 5) points out the influence of the second source heat release for a given flowrate. Results show that a variation of the heat release of the second source (from 1140W to 970 W) affects the thermal stratification. This is due to the fact that in case 4 the temperature of the plume is lower than in case 2 and consequently the plume does not rise as high as in case 2 and spreads into the thermocline zone which reduces the thermal gradient.

Model validation

Considering previous conclusions, it was decided to take into account several heat sources in the Venda model.

The following diagrams show comparisons of experimental temperature profiles and numerical results.

Figure 6 confirms (Manzoni, 1997) the good agreement between Venda calculations and measurements in the case of a single heat source.

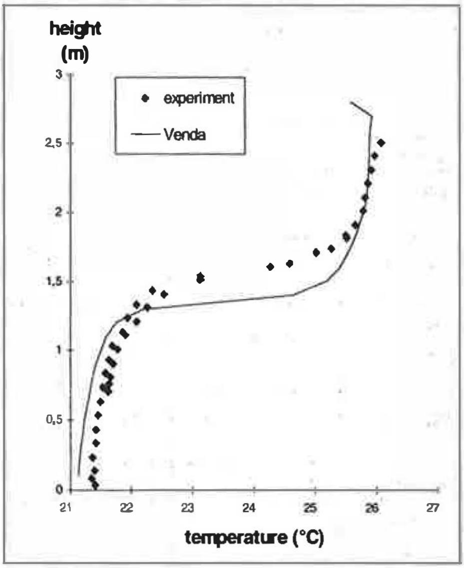

Figure 6 Case 1 : comparison between measurements and calculations

Figure 7, Figure 8 and Figure 9 show the comparison between numerical and sources.

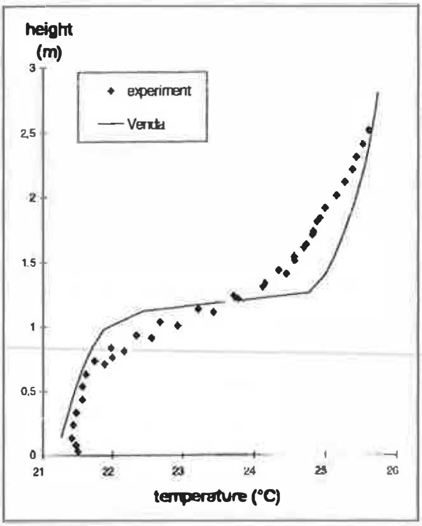

Figure 7 Case 2 : comparison between measurements and calculations

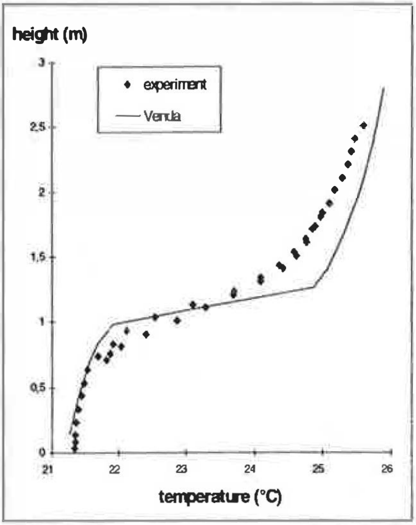

Figure 8 Case 3 : comparison between measurements and calculations

Greater discrepancies appear in Figure 9 between calculations and measurements. This reveals the limitations of the model with regard to the spreading effect of the plume.

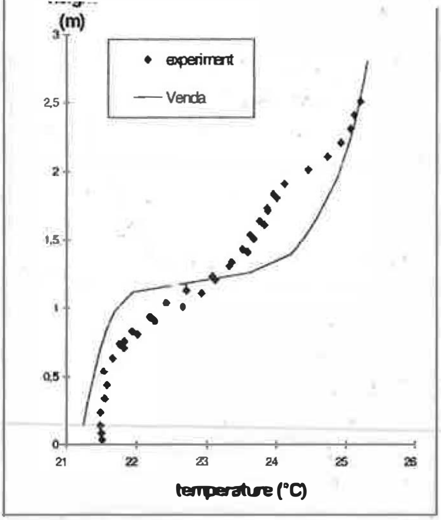

Figure 9 Case 4 : comparison between measurements and calculations

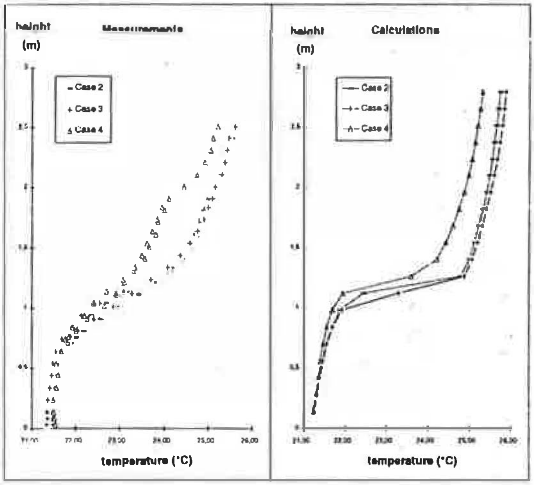

Figure 10 Comparison between cases 2, 3 & 4 - Measurements and calculations.

Figure 10 shows the good agreement between Venda and experiment, concerning qualitative aspects, when changing the plume characteristics. Except for case 4, due to the spreading effect of the plume, we can observe a similar behaviour between Venda and experiment for different heat source conditions

CONCLUSIONS

The present work pointed out the influence of the addition of a second source in a room ventilated with a displacement system. Results show that :

- the thermal stratification obtained in the case with two heat sources and in the case with an equivalent unique heat source are different ;
- the resulting thermal stratification depends mainly on the heat release of the secondary source but is not sensitive to the source flowrate (Froude number).

Comparisons between 1D numerical results and experimental results are satisfactory, but they point out the difficulty for the model to take into account the behaviour of the secondary plume for low heat release rates.

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