VALIDATION OF DISPLACEMENT VENTILATION SIMPLIFIED MODELS

Dominique Manzoni & Pierre Guitton
Thermal Transfer and Aerodynamics Section
Research and Development Division
Electricité de France
France

ABSTRACT

Optimization and control of displacement ventilation systems in buildings require accurate modeling of aeraulic and thermal phenomena involved in the establishment or the destruction of thermal stratification in the room. We carried out an analysis and developed global models on the basis of data from the Poitiers "Laboratoire d'Etudes Thermiques" experimental facility (see Figure°2). The goal is to provide tools accurate enough to describe the thermal comfort implied by such systems but of easier use than CFD codes [3].

We propose two types of simplified models for evaluation of the temperature in the room. They are called zonal model and one-dimensional model and have been presented in a previous paper [5]. After briefly coming back on their description we compare calculations and experimental data in terms of temperature field. We then focus on the plume representation.

We also integrated the models in the CLIM2000 Building Energy Simulation Tool [12]. This means that the elementary models can be connected to other phenomena (complex wall composition, solar input, climate scenarios, etc.) to allow a realistic representation of a room equipped with a displacement ventilation system.

INTRODUCTION

A displacement ventilation system works as follows : due to natural convection a plume develops over a hot air source ; fresh air is relieved to the room at low velocity and close to the floor level ; the plume entrains part of the fresh air to the upper region, where it pours out (Figure 1). The air in the room is therefore divided into two superimposed regions, with a hot, polluted zone floating above fresh air. Note that several thermal plumes, appearing above heat sources such as people or machines, can cohabit [1, 2].

The two existing displacement ventilation modeling ways are the CFD calculations [3] and the two zones-global models [4]. This paper deals with global models able to quantify thermal gradients in rooms.

We present two types of simplified dynamic model. These models differs by their physical modeling and spatial precision. One is a zonal model based on mass and energy exchanges between the zones, and the other is a continuous one-dimensional model, obtained by integration of the energy equation on the vertical axis. In each case, walls are integrated to the models so that the boundary conditions are the walls external surface temperatures.

Then we compare calculations with both models to experimental data obtained in the Poitiers "Laboratoire d'Etudes Thermiques" test chamber. This chamber represents simplified but typical displacement ventilation configurations.

In a previous publication [5] the models have been compared to experimental data in order to evaluate their ability to reproduce the temperature field pattern in the room and the "stratification-damaging" effect of a cold wall. Here we present a more complete validation study based on recent measurements.

Then we focus on the plume representation by comparing velocity and temperature on the plume axis both measured and calculated. We also present a sensitivity study of an important physical parameter of the plume modeling : the entrainment coefficient.

ZONAL MODEL DESCRIPTION

On the basis of experimental observations, we decided to subdivide the initial space into a lower, cold zone, and a upper, hot zone. Both zones are assumed to be thermally homogeneous. Pressure is taken hydrostatic throughout the room. An intermediate zone, fed by boundary layers, appears too (see below). Mass and energy balances are calculated in each zone. Convective transfers between zones are driven (see figure 3) by the plume region and the wall surface boundary layer regions, which are not included in the zones. Walls are integrated in the model and their internal surface temperature is taken uniform in each zone. The boundary condition is the external temperature.

Note that this model is just sketched here, details being available in [5].

Plume modeling

Since the plume develops over a punctual source, it is supposed to be axisymmetric. Note that the following method is applicable to a linear source. Furthermore, we assume that velocity and temperature profiles in the plume are auto-similar in form, and that they follow a Gaussian profile :

$$\begin{cases} W(r,z) = W_0(z)e^{-\left(\frac{r}{R(z)}\right)^2} \\ \Delta\theta(r,z) = \Delta\theta_0(z)e^{-\left(\frac{r}{\lambda R(z)}\right)^2} \end{cases} \quad (1)$$

Applying these expressions to the mass, momentum and energy balances and integrating them (see [10]) we find the following equations :

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

Note that that system uses Boussinesq's hypothesis, which is valid in our range of temperature variation (about 10%).

Zukosky correlation ([6], equations (3)), which is the system (2) solution when the ambient temperature is constant, can then be used to quantify mass and energy rates in the plume :

$$\begin{cases} R(z) = \frac{6\alpha}{5} z \\ W_0(z) = \left(\frac{3F_0\left(1+\lambda^2\right)}{2\pi}\right)\left(\frac{5}{6\alpha}\right) z^{-1/3} \\ \Delta\theta_0(z) = \left(\frac{2}{3\pi^2}\right)^{1/3} \frac{\left(F_0\left(1+\lambda^2\right)\right)^{2/3}}{g\beta\lambda^2} z^{-5/3} \end{cases} \quad (3)$$

These equations are valid in the case of a uniform medium only. On transition between cold zone and hot zone, we compute the virtual origin of the plume in a hot medium, so as to conserve the mass, applying Morton's method [7].

Boundary layers modeling

Ascending or descending currents can appear along the vertical wall surfaces, depending on the difference between wall surface temperature and ambient temperature. Mass and enthalpy transfers along the vertical wall surfaces are modeled by Jaluria correlations [8] in a turbulent medium. Boundary layer thickness d and velocity U are computed by :

$$\begin{cases} U = 1.185\frac{\nu}{z}(Gr)^{1/2}\left[1+0.494(Pr)^{2/3}\right]^{-1/2} \\ \delta = 0.565z(Gr)^{-1/10}(Pr)^{-8/15}\left[1+0.494(Pr)^{2/3}\right]^{1/10} \end{cases} \quad (4)$$

Mass and energy quantities advected in the boundary layer, per unit of perimeter, are given by :

$$\begin{cases} \dot{m} = 0.1463\rho\delta U \\ q = 0.0366\rho C_p \delta U\left(T_p - T_\infty\right) \end{cases} \quad (5)$$

These correlations are valid only in case of homogeneous medium. A particular treatment is applied to transition between two zones, and each possible scenario of temperature hierarchy between wall and zones is took into account (see [5] for details).

Principal zones equations

In order to determine each zone temperature, we have to establish the mass balance and energy balance in each zone. For example we treat the following case : $T_C<T_W<T_H$, meaning that heat transfers between walls and air tend to cool the hot zone, warm up the cold zone and thicken the intermediate region.

The mass balance and energy balance are computed in each zone :

$$\begin{cases} \frac{dM_C}{dt} = \dot{m}_{diff} - \dot{m}_{drag} - \dot{m}_{blC} \\ \frac{dM_I}{dt} = \dot{m}_{blC} + \dot{m}_{blH} \\ \frac{dM_H}{dt} = \dot{m}_{plume} - \dot{m}_{extract} - \dot{m}_{blH} \end{cases} \quad (6)$$

$$\begin{cases} \frac{dH_C}{dt} = Q_{diff} - Q_{source} - Q_{blC} + hS_C\left(T_{WC} - T_C\right) \\ \frac{\partial H_I}{\partial t} = \lambda\left(\frac{M_C}{\rho_C}\right)\Delta T_I + Q_{blC} + Q_{blH} + hS_I\left(T_{WI} - T_I\right) \\ \frac{dH_H}{dt} = Q_{plume} - Q_{extract} - Q_{blH} + hS_H\left(T_{WH} - T_H\right) \end{cases} \quad (7)$$

where the C, H, I and W indices refer respectively to cold, hot, intermediate regions and wall.

The following wall surface equations are added to this system :

$$\begin{cases} Per.h_1.\rho_{wC}.C_w \dfrac{\partial T_{wC}}{\partial t} \\ = \lambda_w Per.e_w \Delta T_{wC} + h.Per.h_1\left(T_C - T_{wC}\right) \\ + k.Per.h_1.\left(T_{out} - T_{wC}\right) \\ \\ Per.\left(h_2 - h_1\right).\rho_{wI}.C_w \dfrac{\partial T_{wI}}{\partial t} \\ = \lambda_w Per.e_w \Delta T_{wI} + h.Per\left(h_2 - h_1\right)\left(T_I - T_{wI}\right) \\ + k.Per.\left(h_2 - h_1\right).\left(T_{out} - T_{wI}\right) \\ \\ Per.\left(H - h_2\right).\rho_{wH}.C_w \dfrac{\partial T_{wH}}{\partial t} \\ = \lambda_w Per.e_w \Delta T_{wH} + h.Per\left(H - h_2\right)\left(T_H - T_{wH}\right) \\ + k.Per.\left(H - h_2\right).\left(T_{out} - T_{wH}\right) \end{cases} \quad (8)$$

We then have to solve a 9 equations system for 9 unknowns : T_C, T_I, T_H, T_{WC}, T_{WI}, T_{WH}, h_1, h_2, $m_{extract}$.

ONE-DIMENSIONAL MODEL DESCRIPTION

The one-dimensional model aims to provide a more accurate description of the temperature gradient inside the room than the zonal model. At the moment it does not include boundary layer nor radiant transfers modeling.

Principles and equations

As the isotherms are nearly horizontal inside the room, air temperature T(x,y,z,t) (in the Cartesian coordinates system represented on Figure 2) varies little along x and y axes outside the plume. It steered us to the following method :

- temperature T(x,y,z,t) is integrated on horizontal planes $\Sigma(z)$ (excluding the plume zone) to obtain the average temperature $\theta(z,t)$;
- integrating the energy balance equation (9) on $\Sigma(z)$, we obtain the equation (10) for the unknown $\theta(z,t)$.

$$\rho_0 C_p \partial_t T\left(x,y,z,t\right) + \rho_0 C_p V \cdot \nabla T = \lambda \Delta T \quad (9)$$

$$\begin{aligned} &\rho_0 C_p |\Sigma| \partial_t \theta(z,t) - \lambda |\Sigma| \Delta\theta \\ &+ \rho_0 C_p \partial_z \left(A\theta(z,t)\right). + \rho_0 C_p B \\ &= h.Per.\left(\theta_w - \theta\right) \end{aligned} \quad (10)$$

Terms A and B depend on boundary conditions (which include exchanges with the plume), and are obtained by integrating the mass balance equation on $\Sigma(z)$:

$$\partial_z A = -\int_{\partial\Sigma} V.nds \quad (11)$$

$$B = \int_{\partial\Sigma} TV.nds \quad (12)$$

The needed information about the plume are given by the equations system (2) given previously. Hence we take into account the non-isotherm medium effects.

We apply the same integration method for each wall surface temperature $T_w(x,y,z,t)$. We thus have averaged wall surface temperatures $\theta_w(z,t)$, and associated averaged heat equation (13) :

$$\begin{aligned} &\rho_w C_{pw}.e_w.\partial_t \theta_w(z,t) - \lambda_w e_w \Delta\theta_w \\ &= h\left(\theta - \theta_w\right) + U\left(T_{out} - \theta_w\right) \end{aligned} \quad (13)$$

We obtain a PDE system of eight equations, the unknowns being the air temperature, four wall temperatures and three quantities describing the plume. It is solved by applying a finite difference method using an implicit scheme.

TEST CHAMBER 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 square enclosure (Figure 2) equipped with two independent air loops and with measurement equipment.

The heat source is obtained using a thermally controlled low velocity air jet located at the bottom of the enclosure. In these conditions the rate of convective heat release can be easily determined. Its area is 0.0324 m^2. Flow rate 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

The fresh air supply is located at the bottom of a vertical wall. Its area is 1.7 m^2. Fresh air temperature is controlled by a water-air cooler in the 15-25°C range. Air is discharged through grids at a low (about 0.05 m/s) and uniform velocity. Flow rate could reach up to 0.083 m^3/s.

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

Walls were thermally insulated with a 3 cm thick insulating material covered on both sides with an aluminum film in order to minimize radiation. The global heat exchange coefficient is around 0.4±0.1W/m^2K.

Though walls are integrated in the models (hence inside surface temperatures are not taken to be boundary conditions) radiant exchanges are not calculated. The introduced error on calculated heat transfers between hot and cold regions is estimated at about 10% referred to the heat source load. Integration of radiant heat transfer in the models will be performed in order to correct that error.

This experimental set-up is completed by air temperature and velocity measuring equipments. Velocity and temperature are obtained using respectively a two-components Laser Doppler Velocimetry device and a 250 mm diameter K-thermocouple. Their measures are performed on a 1500 points 3D-grid in the room and are simultaneous at a given point.

TEMPERATURE FIELD STUDY

Both models were used simulating the four following experimental cases (the source load is referred to the diffuser temperature).

Case	1	2	3	4
diffuser flow rate (m^3/s)	0.051	0.051	0.051	0.051
T_{diff} (°C)	21	21	21,6	20,9
source flow rate(m^3/s)	0.028	0.028	0.052	0.028
source load (W)	807	807	1464	1652
T_{out} (°C)	17	23,6	24	25

Figures 4 to 7 present comparisons of zonal and one-dimensional models with experiments in the vertical temperature field (regarding the measurements we took the average value outside the plume at a given height). Concerning the zonal model, results are presented so that in the intermediate zone, the mean temperature value is replaced by a sloping segment joining boundaries points. Hence we have a better visualization of the stratification calculated by the model.

Calculations are in good agreement with measurements, although we can detect weaknesses for each model.

Globally the zonal model proves to be more accurate on average temperature in cold and hot zones than the one-dimensional model. The later tends to overestimate the hot zone temperature, to underestimate the cold zone temperature and to overestimate the temperature gradient in the transition zone. We suggest that this is due to the influence of mass and energy transfers by boundary layers, phenomenon which is not taken into account in the one-dimensional model.

In case 3 (high mass rate of the heat source) the zonal model does not predict the thermocline height accurately, because in that case the plume entrainment is strongly influenced by the non-uniform ambient temperature field (which is overestimated by the zonal model). In case 4, boundary layers are important so the one-dimensional model is not accurate on thermocline height.

PLUME STUDY

In order to discuss our plume model we compare air temperature (figures 8 and 9) and velocity (figures 10 and 11) calculations and measurements along the plume axis.

Calculated temperatures agree well with measurements in the upper region. The larger difference in the lower zone is due to the plume modeling, since the plume has not a gaussian profile just above the heat source.

Velocities comparisons are more problematic, especially in the upper zone. Nevertheless we notice that in the zonal model case the plume velocity is not as important in the hot zone as in the cold zone, since the plume entrainment in the lower part only acts on the thermocline height. Second, given the plume fluctuations, its velocity is difficult to measure experimentally.

We also performed a sensitivity study of the one-dimensional model temperature field on the plume entrainment coefficient α. On the case 1 basis we tested the four following values : 0.085 [9], 0.093 [10], 0.116 [7], 0.16 [11]. The figure 12 logically indicates that a higher entrainment coefficient implies a lower thermocline and a colder hot zone (since more fresh air melts into the hot air). One can see that α influence is important : between 0.085 and 0.16, the thermocline height increases by a factor 2 and the hot zone temperature increases by more than 1°C. However, the value α=0.16 [11], which is dedicated to fire plumes, is not adequate for displacement ventilation calculations.

CONCLUSIONS

Since displacement ventilation systems dimensioning requires simplified tools, we propose two models with differing degrees of accuracy according to specific needs. These are a zonal model and a one-dimensional model. Both models are able to represent a displacement ventilation temperature field pattern and its sensitivity to characteristics of the heat sources. At the moment the zonal model is more accurate in some cases because of the wall boundary limit modeling but the one-dimensional model proved to gather more information about thermal comfort in the lower zone in stratification damaging cases.

The plume modeling by gaussian profiles seems to be satisfactory since temperatures are accurately calculated out of the source neighborhood. The velocity profile calculations being not so good, a sensitivity study will be performed on the plume boundary conditions.

The models described in this paper have been developed as elementary models (or Formal Types) integrated in the CLIM2000 Building Energy Simulation Tool [12]. In this environment they can be connected to other components or phenomena (complex walls, solar input, air conditioning scenarios, etc.) in order to simulate complete systems (air and building in time dependent thermal solicitations). The study of such systems on industrial cases is opened.

Thereafter the study is directed towards the analysis of multiple source configurations. The objective is to determine whether or not the presence of heat sources of different values is liable to degrade thermal stratification, and thus operation of the displacement ventilation system.

Following the results presented in this paper, we also start integration of radiative heat transfer between walls and natural convection currents along walls phenomena in the one-dimensional model.

REFERENCES

1. A.G. Svensson, "Nordic Experiences of Displacement Ventilation Systems", ASHRAE, Vol.95, Part 2.
2. H. Skaret, "Displacement Ventilation", Research Studies Press Ltd
3. D. Manzoni, P. Vialle, D. Blay, "Convective Phenomena involved in a Displacement Ventilation System", INDOOR'AIR 96 Proceedings.
4. H.M. Mathissen, "Case Studies of Displacement Ventilation in Public Halls", ASHRAE, Vol.95, Part 2.
5. D. Manzoni, "Simplified models of displacement ventilation systems", Proc. RoomVent, Yokohama, 1996
6. E.E. Zukosky, T. Kubota, B. Cetegen, "Entrainment in Fire Plumes", Fire Safety Journal, Vol.3, pp. 107-121, 1981.
7. B.P. Morton, "Forced Plume, Journal of Fluid Mechanics", 5.1, 151-163, 1959
8. J. Jaluria and K. Kapoor, "Importance of Wall Flows at the Early Stages of Fire Growth in the Mathematical Modeling of Enclosure Fires", Combust. Sci. and Tech., 59, 355-369, 1988
9. G. Abraham, "Entrainment principle and its restriction to solve problems of jets", J. Hydraulic Res., Vol.31, pp 1-23, 1965.
10. B.R. Morton, G. Taylor, J.S. Turner, "Turbulent gravitational convection from maintened and instantaneous sources", Proc. Royal Soc., Vol.234A, pp 1-23, 1956.
11. B.J. Mac Caffrey, "Purely buoyant diffusion flames : some experimental results", NBSIR 79, 1910 Nat. Bureau of Stand.
12. B Gautier, F-X Rongère, D Bonneau, "Clim2000 : The Building Energy Simulation Tool and The Modeling Method", Building Simulation, Nice, 1991

NOMENCLATURE

C_P	mass heat (J/kg.K)
e_W	wall thickness
F_0	source buoyancy
g	gravitational acceleration (m/s²)
G_r	Grashof number
	$\left(Gr = g\beta\,(T_p - T_\infty)z^3 / \nu^2\right)$
h	coefficient of convective transfer on the wall surface (W.m².K^{-3})
h_1, h_2	interface height (m)
H	enthalpy (W), room height (m)
k	conducto-convective resistance between wall and outside air (W.m².K^{-1})
K	global coefficient of heat loss on (W.m².K^{-1})
M	weight (kg)
$\dot{m}$	mass rate (kg/s)
Per	perimeter (m)
Pr	Prandtl number
Q	heat rate (W)
S	surface (m²)
T	temperature (°C)
u	mean velocity of boundary layer (m/s)
U	wall surface conductance
W	vertical velocity in plume (m/s)

<u>Greek letter symbols</u>

α	plume entrainment factor
β	coefficient of thermal expansion
δ	plume thickness (m)
l	thermal radius/dynamic radius, conductivity (W.m^{-1}.K^{-1})
ν	diffusivity (m²/s)
$\Delta\theta$	difference between plume temperature and ambient temperature (°C)
$\theta(z)$	averaged temperature at the height z (°C).
ρ	density (kg/m^3)

<u>Indices</u>

0	plume axis
bl	boundary layer
out	external condition
C	cold zone
H	hot zone
I	intermediate zone
W	wall condition
∞	ambient medium

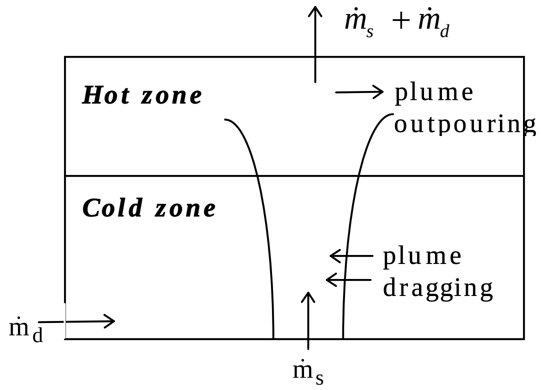


Figure 1 : principle of a displacement ventilation system

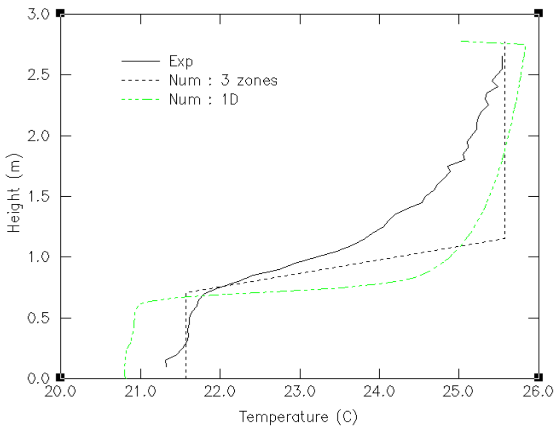


Figure 4 : case 1

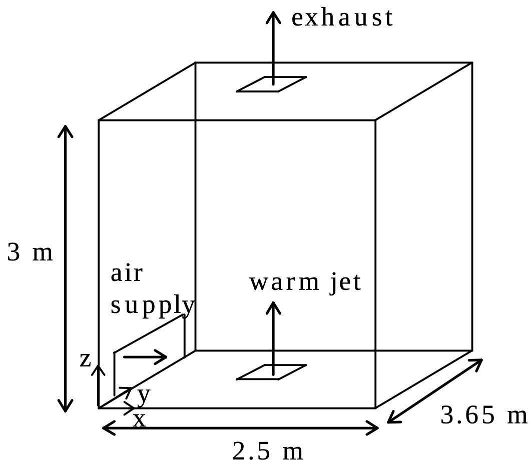


Figure 2 : test room

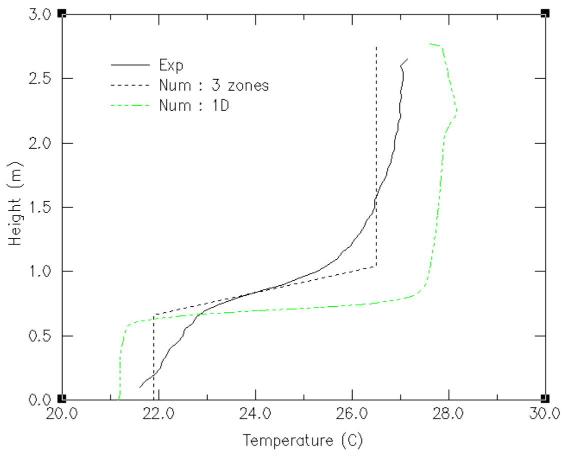
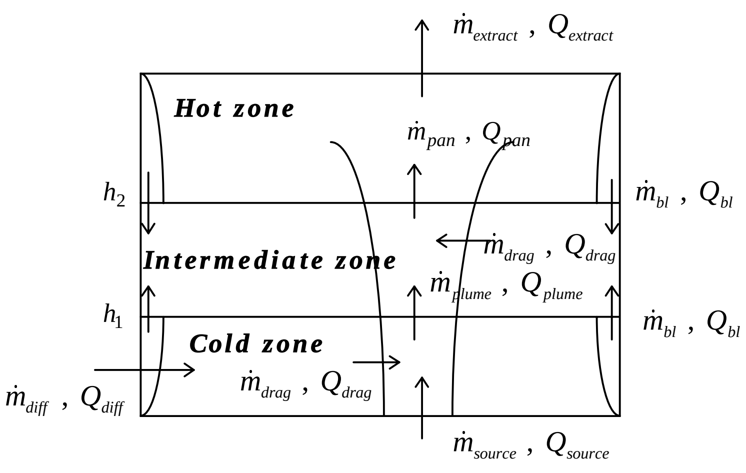


Figure 5 : case 2



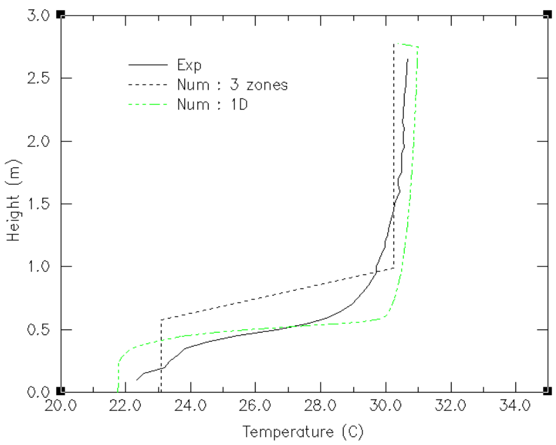


Figure 6 : case 3

Figure 3 : zonal model principle

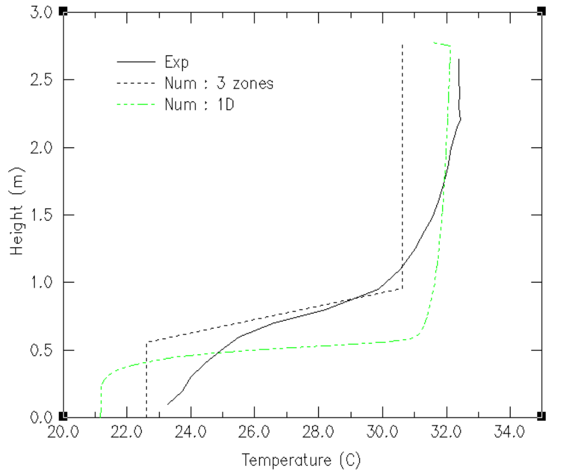


Figure 7 : case 4

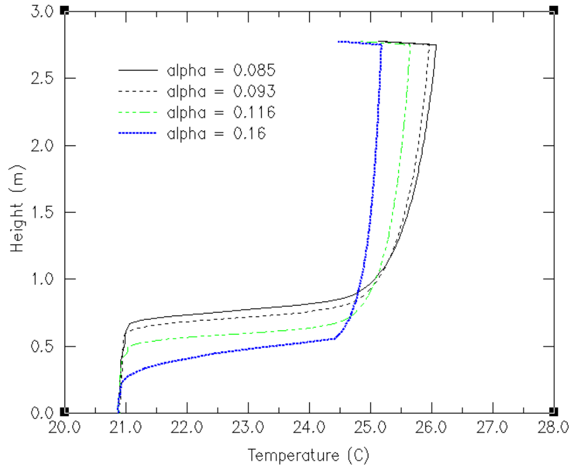


Figure 12 :dragging coefficient sensibility.

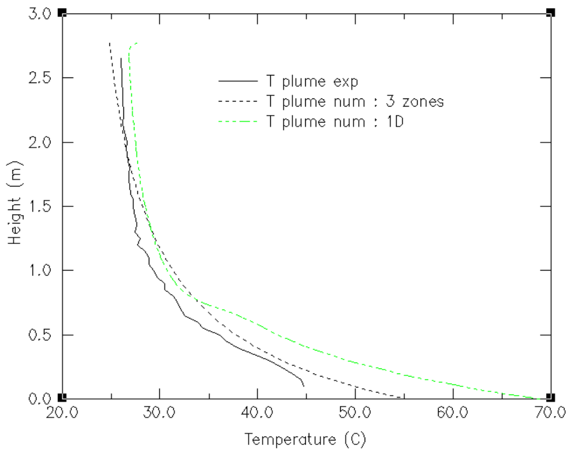


Figure 8 : plume temperature in case 1

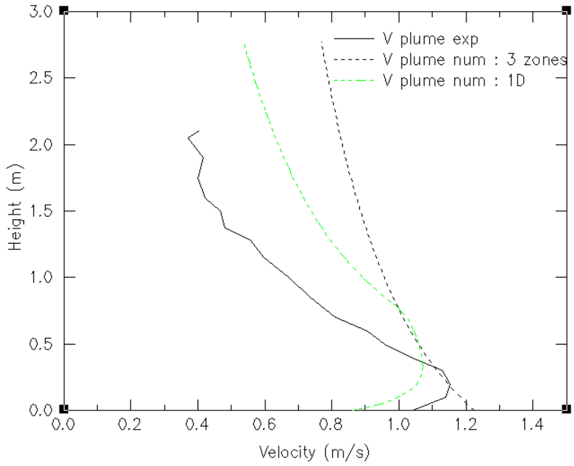


Figure 10 : plume velocity in case 1

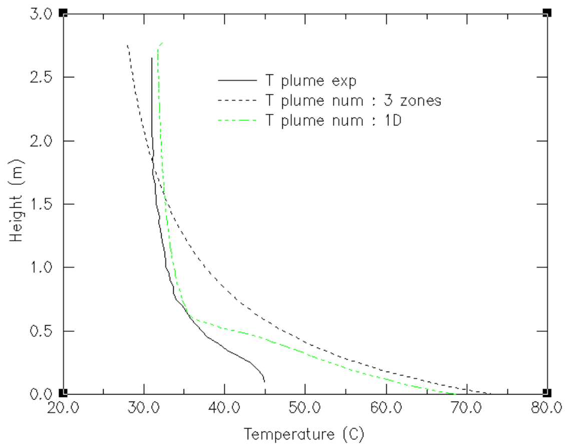


Figure 9 : plume temperature in case 3

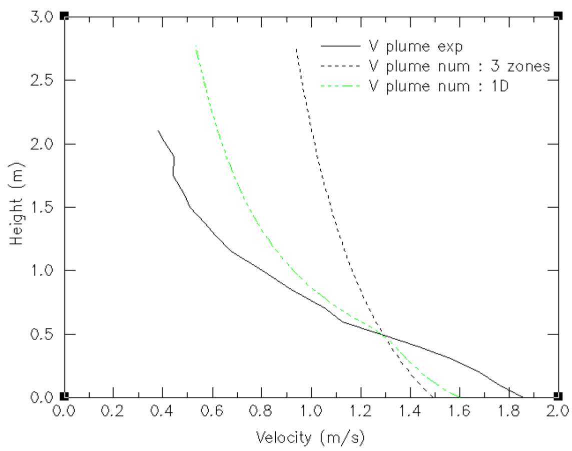


Figure 11 : plume velocity in case 3