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NATURAL AND MIXED CONVECTION IN ROOMS. PREDICTION OF THERMAL STRATIFICATION AND HEAT TRANSFER BY ZONAL MODELS

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ABSTRACT

The present paper gives a review of zonal models developed in order to predict thermal stratification and heat transfer in heated or ventilated rooms.

After a short description of pioneer works, we present various levels of zonal models used in the prediction of the thermal coupled behavior of a room and its heating system. A reference case solved by IEA annex 20 expert group enables us to compare the solutions given by three different zonal models together with the results of a k-ε low Reynolds number code.

In a second part of the paper we focus on the recent developments of zonal models concerning their extension to unsteady regimes, their application to ventilation studies, and their generalization to pressure zonal models.

KEYWORDS: Zonal models, room thermal behavior, thermal stratification.

INTRODUCTION

In building physics, we can notice during the two last decades new orientations of the objectives of research activity. The whole goal of our activity has been reoriented from the prediction of thermal building behavior and thermal needs to consumption evaluation integrating the system efficiency, thermal comfort and air quality prediction. New problem arise from this evolution: on the one hand the room by room isothermal assumption is no more sufficient to reach these goals, on the other hand even if CFD codes are useful they are not devoted to large scale building simulation.

In order to fill this gap, Lebrun (1970) proposed to split a room into different zones characterising the main driving flows. In such a way it appeared that it was possible to predict more accurately the thermal behavior of rooms and to make easier the coupling between rooms and heating systems.

Since this pioneer work of Lebrun in 1970, various teams have developed this idea and these methods are known today under the generic name of zonal models.

This paper presents at first an overview of the evolution of zonal models in typical configurations, we discuss the physical requirements and present various applications. Then, we consider some existing comparisons carried out within the frame of IEA annex 20 with experiments or CFD codes in the case of couplings with different heating systems.

If zonal models have been previously dedicated to the prediction of steady state behavior of heated rooms, it appears that their use is extended now to more general problems such as unsteady state studies or ventilation. In the second part we present some of these applications and we show the ability of zonal models to give the general information necessary to compare different systems or predict their behavior. With respect to this extension of zonal models, we describe a new generation of zonal models based on the evaluation of the pressure as an additional state variable.

THE PIONEER WORK

During the seventeens, and as a consequence of the first energy crisis, a strong effort has been made in energy management in buildings. Many authors have been working at this time on numerical prediction of thermal behavior of buildings in order to provide better insulation and save energy. Nevertheless few validations on laboratory experiments have been made, and the few results obtained show a systematic gap between the air temperature predicted in a room and the central temperature supposed to be a good approximation of the temperature of convective equilibrium of the air mass contained in a room (Lebrun 1978). At first, Lebrun presented this problem as a bad knowledge of the convective behavior of rooms and a consequence of the heterogeneousness of air temperature in the room. As it was impossible to predict this non isothermal behavior, he proposed to adjust empirically the models by changing the value of the thermal capacity of air.

At the same period, various studies concerning the prediction of comfort conditions (Fanger 1974) pointed out thermal stratification to be a leading parameter for comfort conditions estimates.

We can notice with these two examples how an even simplified model giving a rough estimate of the non isothermal behavior of a room can help. The first contribution to this problem has been given by Lebrun (1970). Considering flow visualisation of a heated room, he suggested to split the room into different zones, and to connect these zones by mass conductances. Figure 1 describes a typical flow pattern in a room heated by a convector, and a view of the model proposed by Lebrun.

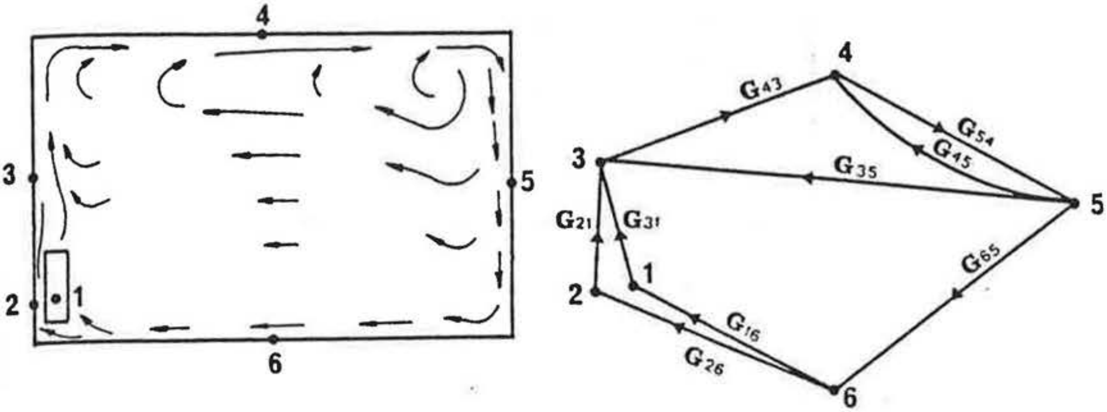


Figure 1: First model developed by Lebrun (1970)

The sketch defined by Lebrun privileges the heat transfer between the surfaces of the room, it supposes a flow circulation from the heating system, to the wall jet, the ceiling, the opposite vertical wall and the floor. Nevertheless, it can also take into account the recirculating flow located at the upper right corner of the cavity. At this very first step, the idea was mainly to take into account the heat transfer by convection from one part of the room to an other part, the idea was more to give an estimate of the convective flows circulating in the room than to predict its real behavior.

This kind of approach has then be followed by various authors dealing with the same kind of preoccupations: the coupled behavior of a room with its heating system. The goal of these studies is to predict the thermal stratification in a room, to increase the quality of the heat transfer description, and as a consequence to evaluate the efficiency of heating systems with respect to comfort criteria. We describe here the most representative steady state models developed in this way.

SHORT REVIEW OF TYPICAL ZONAL MODELS FOR HEATED ROOMS

Analytical Model (Laret 1980)

As a logical following to the pioneer work done by Lebrun, Laret (1980) proposed an analytical model predicting the thermal stratification in a room submitted to a convective heating system. Furthermore, this period corresponds also to a strong demand in analyzing the thermal couplings between heating systems and rooms; various authors contributed to this task using zonal models.

Figure 2 presents a sketch of this model.

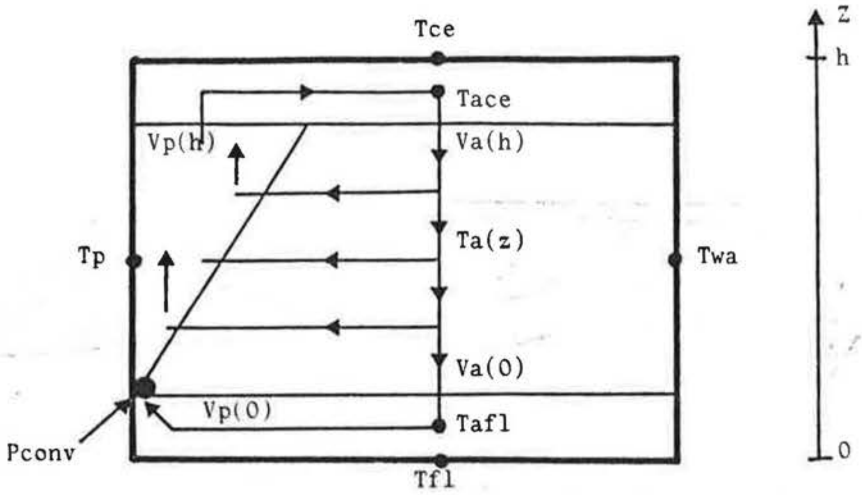


Figure 2 : Scheme of flow pattern by Laret (1980)

In this model, the heat source is described as a linear convective source. Writing the energy balance of each zone leads to energy conservation equations for:

- the ceiling:

$$Cp\ Vp(h)\ (Tace-Tap(h)) + hcec\ Ace\ (Tace-Tce) = 0 \tag{1}$$

- the floor,

$$Cp\ Va(0)\ (Tafl - Ta(0)) + hflc\ Afl\ (Tafl - Tfl) = 0 \tag{2}$$

- the central zone,

$$Cp\ Va\ \frac{dTa}{dz} = \sum_i hwac_i\ Lwa_i\ (Twa_i - Ta) \tag{3}$$

- the plume.

$$Cp\ \frac{dVp}{dz}\ (Tap - Ta) + Cp\ Vp\ \frac{dTap}{dz} = hpc\ Lp\ (Tp - Tap) \tag{4}$$

The mass balance for each zone leads to :

- Ceiling : $Vp(h) = - Va(h)$ (5)
- Floor : $Vp(o) = - Va(o)$ (6)

- Central zone:

$$\frac{dVa}{dz} = -\frac{dVp}{dz} \tag{7}$$

Assuming the mass flow variation independent of the altitude within the plume, Laret proceeds to the integration of this set of equations which delivers the mean temperature of each zone, and by consequence the thermal stratification in the room.

Two Zone Model (Howarth 1980)

The objective of this model, described by A.T. Howarth (1980) in his Ph. D. thesis, is to evaluate the temperature stratification in a room heated by a radiator with a simple model. This technique requires information and assumptions on the convective flows and heat transfers which occur in the room. Figure 3 gives a typical scheme of flow pattern described by Howarth.

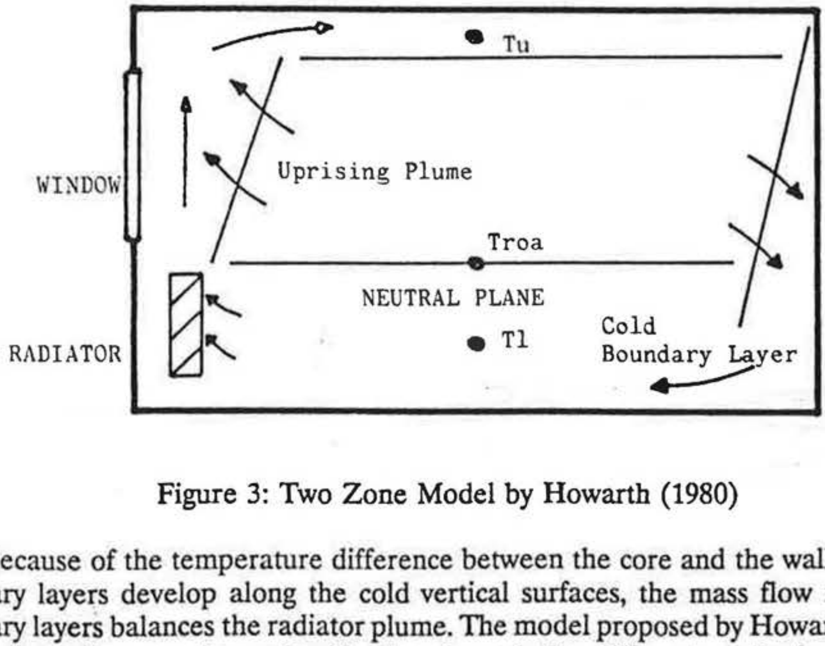


Figure 3: Two Zone Model by Howarth (1980)

Because of the temperature difference between the core and the walls, downward boundary layers develop along the cold vertical surfaces, the mass flow rate in these boundary layers balances the radiator plume. The model proposed by Howarth calculates all the heat flows exchanged with the air and the different walls from empirical equations. The characteristics of the heating system and its plume are defined by experimental measurements and expressed by Howarth with experimental correlations for each parameter.

* Mass flow rate at the top of the radiator:

$$Vp(Hrad) = K1\ (Tra - Troa)^n \tag{8}$$

- Mean air temperature leaving the radiator:

$$\mathrm{Tap(Hrad)} = \mathrm{Troa} + \mathrm{K2}\ (\mathrm{Tra} - \mathrm{Troa}) \qquad (9)$$

- Convective heat transfer between the radiator and the room:

$$\mathrm{Pconv} = \mathrm{Cp}\ \mathrm{K1}\ \mathrm{K2}\ (\mathrm{Tra} - \mathrm{Troa})^{n+1} \qquad (10)$$

- Mass flow rate in the plume:

$$\mathrm{Vp}(z) = \mathrm{Vp(Hrad)}\ [\ (Z-Z_0)\ /\ (\mathrm{Hrad}-Z_0)]^{m} \qquad (11)$$

For each kind of heating system, the parameters K1,K2,n,m, and Z_0 are defined by empirical equations resulting from experimental studies.

Five Zone Model (Inard 1988)

Inard dealt also with the thermal couplings between a radiator and a room. In his five zone model, the inside air volume is split into five isothermal zones, coupled with mass flow conductances.

In the case of a radiator or a convector heating a room, experimental studies (Inard 87, Howarth 80) showed that, in most cases, the air flows have a main circulation along the way radiator, trail ceiling, vertical walls, floor and radiator. Thus, the inside air volume is split into five zones representative of this flow pattern as shown in Figure 4.

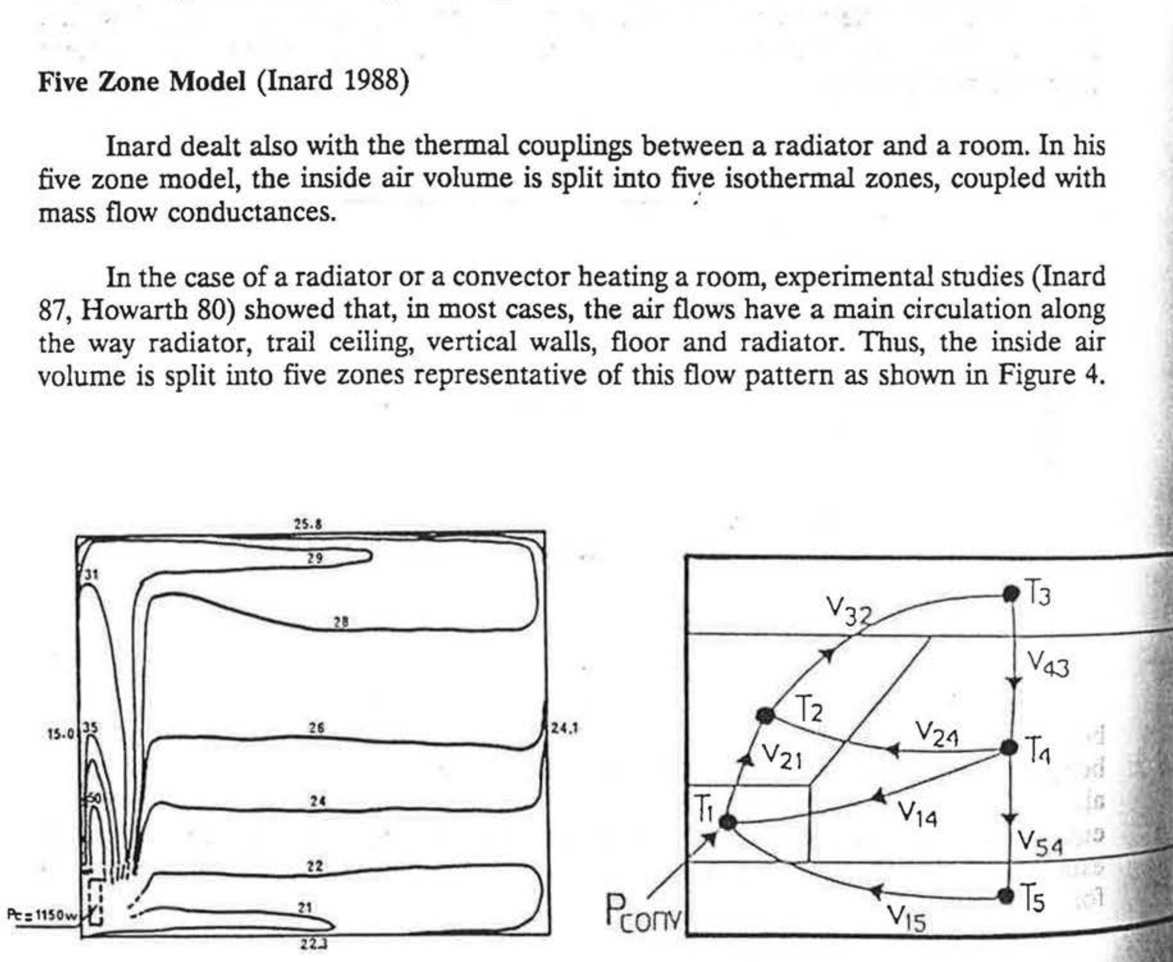


Figure 4 : Typical thermal field in the vertical mid plane of a heated room and five zone model proposed by Inard (1988).

For each zone, the mass and energy balances are written :

$$\sum_{j=1}^{M} G_{ij} + Gs_i = \sum_{i=1}^{M} G_{ji} + Gr_i \tag{12}$$

$$\sum_{j}^{M} Cp\ G_{ij}(T_i - T_j) + Cp\ Gs_i\ (T_i - Ts_i) + \Phi conv_i = Pconv_i \tag{13}$$

In these equations, the variables are the air temperature of each zone. To complete the description of the model, the radiator convective heating power, the different convective heat transfer coefficients and the mass flow conductances must be given or identified. For that reason, Inard (1988) provides experimental results. The tests were performed in a climatic test room on different radiators and on one electric linear heat source and two kinds of measurements were carried out:

- surface temperature inside and outside the walls, to measure 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 rate and the heat flow inside the plume.

This model has been extended later on to various heating systems such as radiators, convectors, heating ceiling and heating floor (Inard 1991 a)

Comparative Evaluation of the Models

Within the frame of IEA annex 20, a comparative evaluation of these models has been realised by Inard and Buty (1990). The case characteristics are given in Figure 5.

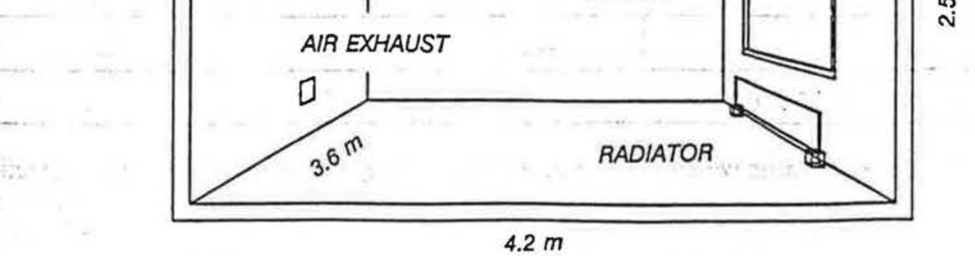


Figure 5: Description of test case d of IEA annex 20.

Different cases were tested corresponding to different radiator and window surface temperatures. Figure 6 presents the air temperature vertical profile obtained with the three models compared with a low Reynolds number k-ε model developed by Chen (1990).

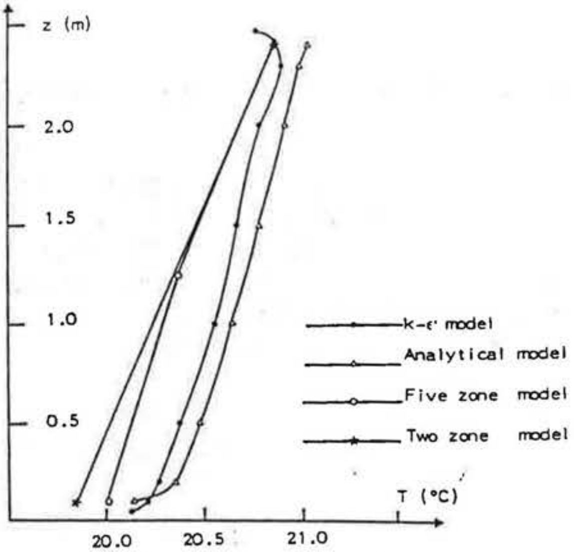


Figure 6: Air temperature vertical profiles.

Table 1 presents the results obtained in convective heat fluxes exchanged at the heat source and along the glazing. The underlined numbers correspond to necessary data for the models.

Table 1: Comparison between three zonal models.

	GLAZING		HEAT SOURCE	
MODELS	Temp °C	Convec. Heat Flux (W)	Temp °C	Convec. Heat Flux (W)
K-ε	0.0	-217	65.0	288
Analytical	0.0	-271	---	415
Two Zones	0.0	-429	72.4	586
Five Zones	0.5	-275	65.0	392

From these results we can see that, except for the two zone model, the convective heat fluxes computed are of the same order of magnitude. The values calculated by the two zone model are roughly 100% higher. As this has been pointed out by Inard before, these values strongly depend on the convective heat exchange coefficient between the radiator plume and the glazing.

This example show the ability of zonal models to predict thermal stratification. However, the results presented show how the convective heat fluxes are sensitive to the choice of the convective heat transfer coefficient especially to the glazing. A very good knowledge of convective heat transfer between the air and all walls in the different flows considered is necessary, and a strong effort in experimental studies is needed in this field.

NEW DEVELOPMENTS IN ZONAL MODELS

Extension to Unsteady Regime

In order to improve the use of zonal model, various authors extended the previous models to unsteady state. Overby and Steen-Thode (1990) used a two zone model similar to Howarth's model and the comparison of their numerical predictions with experiments in a laboratory test room agrees reasonably well.

Inard (1991) followed the same way in introducing thermal capacities at each temperature node of his five zone model.

For each zone, the energy balance equation becomes :

$$\rho CpV_i \frac{dTa_i}{dt} + \sum_{j}^{M} Cp\ G_{ij}(T_i - T_j) + Cp\ Gs_i\ (T_i - Ts_i) + \Phi conv_i = Pconv_i \tag{14}$$

Using this model Inard shows that in order to predict the dynamical stratification, five zones are not sufficient and he proposes a twelve zone dynamical model. Figure 7 presents this model.

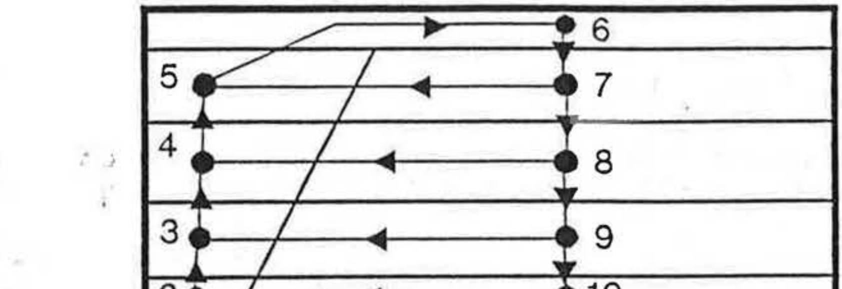
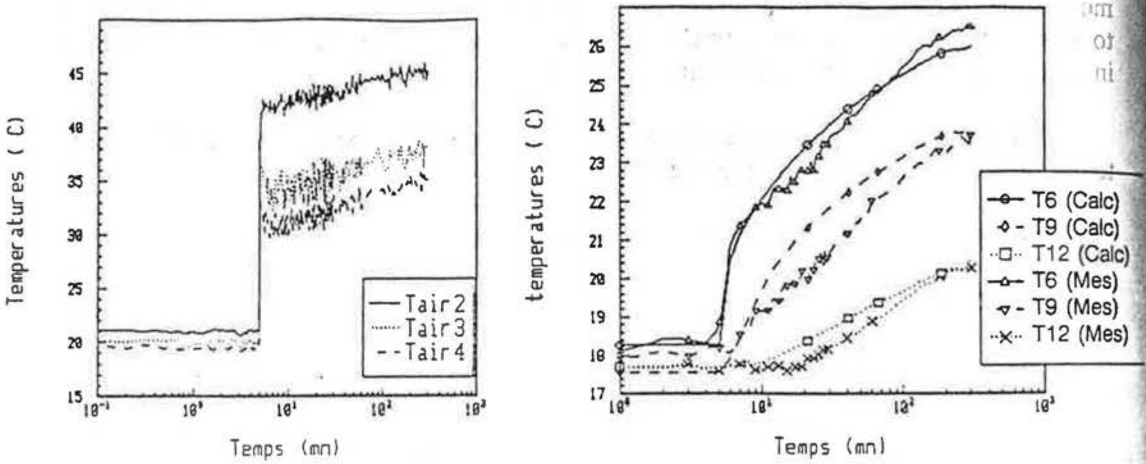


Figure 7: Twelve zone dynamical model proposed by Inard (1991)

Inard realised an experiment using a purely convective source and generating a temperature step. The comparison between experimental data and the predicted values shows a good prediction of the temperature in the upper zones of the room. In the lower part the differences we can notice can be due to the fact that the correlations used for the evaluation of convective heat transfer were established in steady state conditions. Furthermore as the model has no diffusion term, in the central part with low velocities, this can slightly modify the results. Figure 8 presents the temperature measured in the plume of the source and the comparison between measured and predicted temperatures in the mid plane of the cavity.



(a) Temperatures in the plume

(b)Comparison with experiments

Figure 8: Dynamical 12 Zone Model (Inard, 1991)

Application to Ventilated Rooms

In 1987, Sandberg proposed a two zone unsteady box model to predict the behavior of a room ventilated by displacement. Figure 9 shows the problem treated by Sandberg.

As previously defined by Baines and Turner (1969), Sandberg assumes an axisymetric turbulent buoyant plume starting from a point source. The main characteristics of the plume are given by integral analysis, and this leads to write conservation equations of mass and energy at each altitude.

This model comparable to Howarth's model is very robust and enables Sandberg to predict the evolution of the contaminant concentration in a room ventilated by displacement. The numerical prediction appears to be in good agreement with experimental results obtained in a scale model with water.

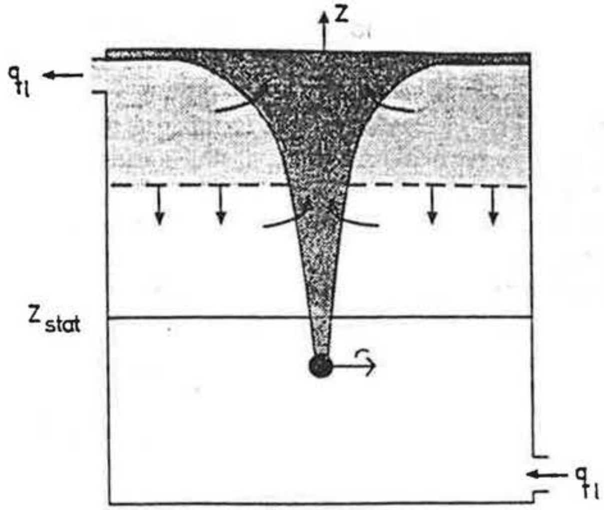


Figure 9: Sketch of development of stratified environment in a ventilated room due to a heat or contaminant source.

Finally, Sandberg points out that the quality of the results is directly related to the quality of the description of the leading flows, mainly the buoyant plume, but he suggests to improve this model by adding boundary layer flows along the walls.

Pressure Zonal Models

As a generalization of the concept of zonal model, various investigators (Grelat 1987, Dalicieux and Bouia 1991) suggested recently to add the pressure in the state variables defining the behavior of a room. The idea is that the main problem of usual zonal model is related to the prediction of the transport terms between zones which are not directly described by the main flow equations. Then in the case of a heated room, we can split the room into two different kind of zones, the zones included in the plume, and the "current" zones. Figure 10 presents this principle of discretization.

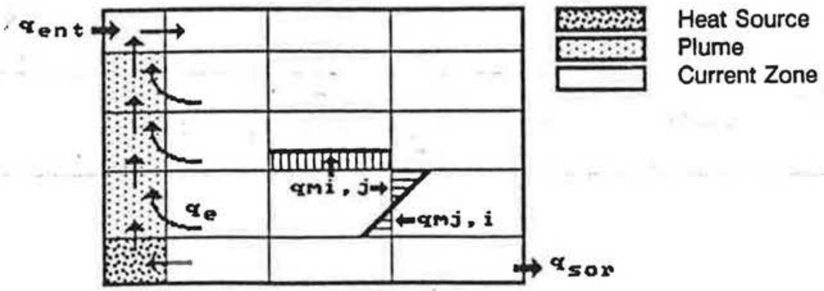
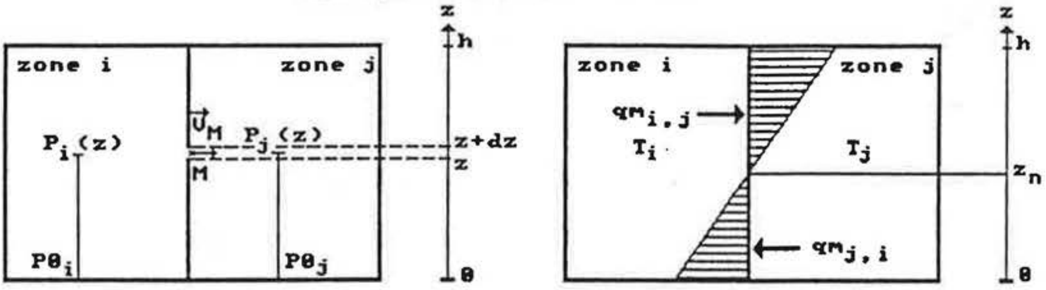


Figure 10: The different zones and associated flows.

For the zones which are not located within a driving flow, using Bernoulli's equation, it is possible to estimate the reference pressure at any location of the room. The pressure difference calculated between two points located at the same altitude enables us to obtain the horizontal velocity due to this pressure drop. If we now consider two adjacent zones with a vertical interface, if the zones are isothermal and if the pressure distribution within a zone is only due to the hydrostatic pressure, then between the two zones we get a linear profile of pressure difference and we can calculate from one hand the neutral level and on the other hand the two way mass flow existing between these zones. Figure 11 presents these principles.



Orifice Flow

Interface between 2 Zones

Figure 11: Two way flow through a vertical interface between two zones.

Defining the same scheme for horizontal interfaces, a pressure difference will lead to a one way flow. Then, writing the mass balance and energy equations of each zone, one converges iteratively to a solution giving the temperature and reference pressure in each zone. Figure 12 presents the results obtained by Dalicieux when modelling a heated and ventilated room.

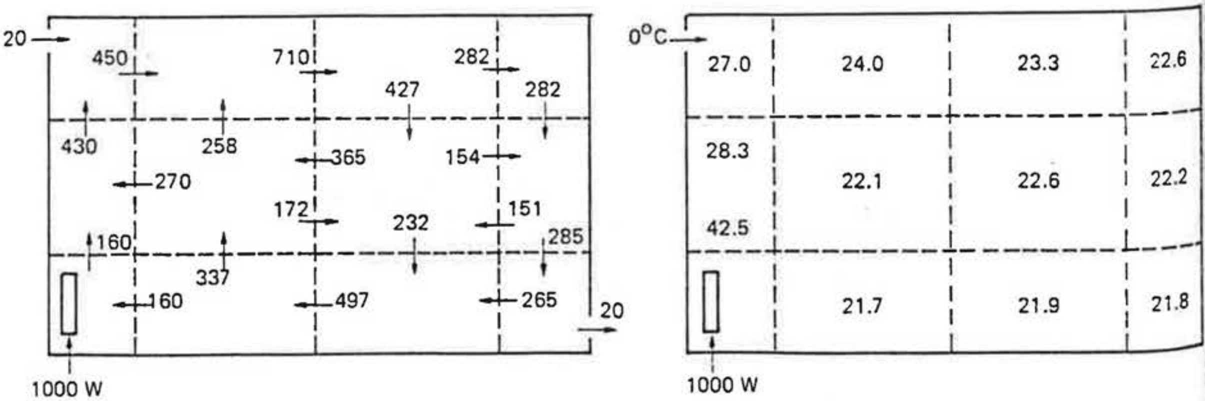


Figure 12: Predicted mass flow rates and temperature in a ventilated and heated room (Dalicieux 1991).

CONCLUSION

This short review of zonal models points out the usefulness of such an approach in order to get a quick estimate of the non isothermal behavior of a room with heating or ventilating systems. Furthermore the existing comparisons with CFD codes or real scale experiments show that most of the time the zonal models are able to predict with a reasonable accuracy air flow rates and heat transfers within a room.

Nevertheless, this kind of models are always based on two main assumptions. On the one hand they suppose that we are able to predict the main driving flow (boundary layer, jet or thermal plume), on the other hand they assume that we have a sufficiently good empirical knowledge of these structures to calculate their main characteristics.

These two assumptions are really limiting the use of these models in a prediction process. In a general case, it is not always possible to take a clear decision on the main flow pattern and much more work is still needed to get a better knowledge of heat and mass transfer in buoyancy driven flows developing in a non isothermal and non adiabatic environment. Real scale experiments, or in singular cases CFD codes, can bring us this necessary information.

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