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Some Aspects of Gravity Driven Air Flow through Large Apertures in Buildings

J. van der Maas, Ph.D. **C.A. Roulet, Ph.D.** **J.A. Hertig, Ph.D.**

ABSTRACT

The authors compare the available algorithms for gravity-driven airflows through large openings with the requirements for multizone air-infiltration modeling.

It appears that two extreme situations are addressed in the literature. The first, providing a time-independent airflow, is formulated in terms of the constant temperature difference between a zone and the outside. The other, considering adiabatic walls, provides a transient flow, with the cold outside air filling the zone space completely so that the airflow comes to a halt.

As an intermediate situation we present an algorithm that takes into account the heat transfer with the walls. The coupled equations for airflow and heat transfer are solved for the inside air temperature. The latter being lower than the wall temperature and always higher than the outside temperature, the proposed model predicts an air and heat flow rate, which is intermediate in magnitude with respect to the extreme situations addressed in the literature. For constant wall temperatures, the model predicts that the air flow rate is lower for zones with smaller heat exchanging wall surface area. Coupled to a thermal model for the wall surface temperature, the algorithm predicts the air flow to decrease with time. Experiments were performed on a full-scale test room with a window to the outside. During cold windless nights, velocity profiles were measured in the window plane, and the temperatures of the inside air and the walls were recorded with time.

Good agreement with the model is obtained, considering only the fraction of the wall surface area active in the heat transfer process, and measuring the inside air temperature in the outstreaming air. Effects which have been observed and should be included in the algorithm for detailed multizone air infiltration modeling are wind effects, the time development of temperature stratification, and the initial transient, whose time constant depends on furniture in the zone.

INTRODUCTION

In order to model infiltration and exfiltration of air and the airflows between rooms of a multizone building, a number of computer programs have been developed (Herrlin 1985; Clarke 1985; Liddament 1986) or are under development (International Workshop COMIS 1988, 1989).

All of these models can deal with simple crack or sharp edge opening flow, but there is a need for improved algorithms in particular for the description of bi-directional flow through large openings in the presence of temperature stratification and single-sided ventilation.

Single-sided ventilation was considered by de Gids and Phaff (1982) and by Warren (1986). Their simple models need the knowledge of the inside-outside temperature difference and the wind velocity.

In this paper we address the problem of single-sided ventilation to a colder environment where flow is gravity-driven not perturbed by wind, and in the absence of a heater. The question is "how to determine the inside air temperature once a window has been opened?"

It is well-known from daily experience that the room air cools after a window has been opened, and that the degree of cooling (at a given temperature) depends on the window opening, the size of the room, and time; this is most noticeable in the absence of local heating systems (e.g., electrical heaters).

Kiel and Wilson (1986), Linden and Simpson (1985), and Lane-Serff et al. (1987) considered the airflow during the opening and closing cycle of a door as a transient phenomenon without mixing or heat exchange with the walls; the air temperature in the room drops rapidly to the outside temperature, and the airflow comes to a halt once the space is filled with outside air.

Indeed, the initial large temperature difference is not maintained, but it does not drop to zero either, due to the heat stored in the walls. To predict a time dependent mass and heat flow rate requires thermal modeling to be included in the usual large aperture algorithms (Herrlin 1985; Clarke 1985; Liddament 1986).

How is Single-Sided Ventilation Implemented in the Existing Computer Programs?

The multizone models (Herrlin 1985; Clarke 1985; Liddament 1986; International Workshop COMIS 1988, 1989) are node models, which means that one zone is

J. van der Maas and **C.A. Roulet**, Research Group Leader, are physicists with LESO-PB, Solar Energy and Building Physics Research Laboratory, and **J.A. Hertig** is Research Group Leader, LASEN-GC, Institute for Energy Systems with Applications to Wind Tunnel Research, EPFL, Federal Institute of Technology, Lausanne, Switzerland.

represented by (at least) one node; to every node one initially assigns a temperature and a pressure (unknown). The nodes are interconnected by non-linear flow conductances for cracks or by specific algorithms for special cases like large openings.

Assuming that there must be a mass flow balance in every node, a system of non-linear equations can be set up with the node pressures as variables, which is solvable with an iterative method.

In the case of vertical large openings, the flow is bi-directional—with warm air going out at the top and cold air entering at the bottom. This cannot be represented by a single flow conductance, and not all existing models are able to account for single-sided ventilation (Herrlin 1985).

The building energy simulation program ESP (Clarke 1985) includes an airflow module, ESPAIR, which uses an algorithm for large openings based on the Bernoulli equation, formulated in terms of the zone temperatures. ESPAIR itself is a static program which predicts a constant flow rate as long as the zone temperatures are constant. To obtain a time-dependent behavior, ESPAIR can be coupled to the thermal modeling of ESP. However, this is a very CPU-time-consuming process and therefore simplified thermal models are needed (Pelletret 1988).

In the following paragraphs, we present our thermal model and compare the predictions of the airflow algorithm coupled to the thermal model with the experimental results. We conclude with a discussion and recommendations for future work.

GRAVITY DRIVEN FLOW

With gravity driven air flow we think in the first place of a cold air mass which moves at uniform velocity in a less dense environment. Nevertheless we will use it here also for situations where buoyancy driven flow would be more appropriate. Both terms are used in opposition to pressure driven flow which will not be considered in this paper.

Most authors consider a steady state situation that is not realistic for occupied buildings. One of the reasons is that steady state flow is still not well understood. For example, stratification in buildings is included in the calculations by only a few authors (Hill et al. 1986), and measurements of discharge coefficients, which characterize the flow loss through apertures, vary considerably between authors (Mahajan and Hill 1987).

Few papers describe airflow through apertures as a transient phenomena. When the flow through doors and windows that open and close is considered, the concept of gravity current is very useful (Simpson 1982; Kiel and Wilson 1986; Lane-Serff et al. 1987). Cold air then enters a room as a gravity wave [Figure 1a], which flows at constant velocity until it reflects from the backwall [Figure 1b] filling up the room completely until the top of the door level, leaving trapped warm air at the ceiling [Figure 1c].

Steady State Flow

Steady flow conditions are realized when the temperature distributions on both sides of the door do not change with time. When there is no net flow through the door, the volume flow rate of cold air through the lower part of the door equals the flow rate of warm air at the top of the door. If the temperatures T_{in} and T_{out} are fixed and constant on both sides, the neutral height at the interface separating the two flows is situated in the middle of the door and is defined by a zero pressure difference, $\Delta P = 0$. The heat and mass transfer through the opening is then described by a simple model based on the Bernoulli equation (Brown and Solvason 1962).

In a recent study of interzonal natural convection heat transfer in buildings, Hill et al. (1986) extended the Bernoulli model to include linear temperature stratification and variable neutral heights. The important conclusion was reached that using the experimental temperature profiles in the connecting zones, the model gives a correct prediction of the neutral heights, the velocity profiles, the mass flow rate, and the heat transfer between building zones. We use the Bernoulli model, presented briefly in the following, as the basis to calculate air flow from given zone temperatures. A parameter in the Bernoulli model, the discharge coefficient of the aperture, will also be discussed.

Bernoulli algorithm. Assuming one neutral level in the opening at the height, $z = H_n$, the pressure difference at z equals

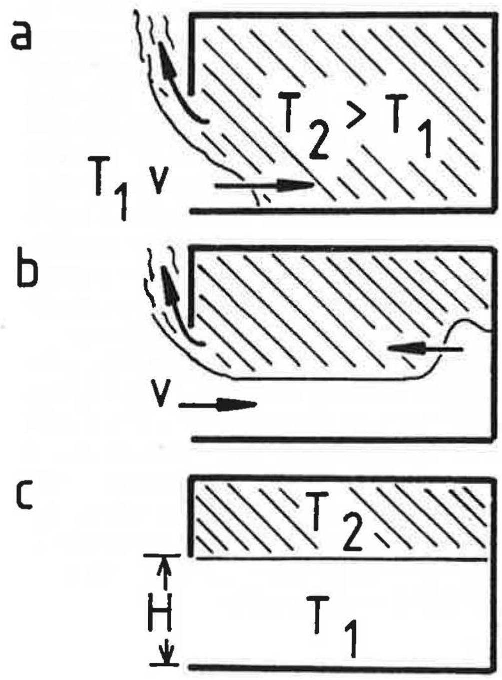

Figure 1 *Transient flow model. a) At the opening of the door a gravity wave enters the room with constant velocity v, driven by the temperature difference $T_2 - T_1$. b) The wave reflects at the wall and returns. c) The room is filled with dense fluid up to the door height level. Low density fluid is trapped at the ceiling (T_2). Ventilation has stopped.*

$$\Delta P(z\quad) = g\,(\rho_{out} - \rho_{in})\,(z - H_n)$$

Assuming inviscid and incompressible flow, application of the Bernoulli equation yields the velocity

$$v(z) = C_1 \sqrt{\frac{g}{T}}\sqrt{2\,|z - H_n|\,\Delta T} \qquad 0 \leq z \leq H \tag{1}$$

where we have used the ideal gas law to convert relative density differences in relative temperature differences, and

$$\Delta T = T_{in} - T_{out} \tag{2}$$

C_1 is the discharge coefficient accounting for flow losses in the aperture. In the general case, T_{in} depends on z, and assuming a linear temperature profile we have

$$\Delta T(z) = \Delta T(0) + a\,z \tag{3}$$

Temperature stratification is insignificant as long as

$$\Delta T(0) >> a\,H \tag{4}$$

this is the condition for isothermal zones; the velocity profile (Equation 1) is parabolic. Inside buildings the inverse is often true, $\Delta T(0) << a\,H$. From Equations 1 and 3 it follows that the velocity profile is then linear above the neutral height, a feature of many data presented by Hill et al. (1986) and Mahajan (1986, 1987).

The neutral height is obtained by integrating Equation 1. For one-sided ventilation the condition is that the volume flow above the neutral level equals the volume flow below. This means solving the following expression for H_n:

$$\int_0^{H_n} v\,dz - \int_{H_n}^{H} v\,dz = 0 \tag{5}$$

The ventilation rate through the aperture equals

$$V = W \int_0^{H_n} v\,dz \tag{6}$$

For isothermal zones this reduces to

$$V = \tfrac{1}{3}\,W\,H\,v_{max} \tag{7}$$

where the maximum velocity, $v_{max} = v(0) = v(H)$, and the heat flow becomes

$$Q = \rho\,C_p\,V\,\Delta T \tag{8}$$

The Discharge Coefficient. The importance of flow losses in the opening is characterized by the discharge coefficient C_1 (Equation 1). Its value is expected to depend on the degree of turbulence and therefore on the local Reynolds number. and C_1 may vary with the distance from the neutral level. It is customary, to define a discharge coefficient for the flow through the aperture due to the interzone temperature difference. However, it is important to distinguish isothermal-zones (IZ, Equation 4), from non-isothermal-zones (NIZ), and IZ-models (Equations 7 and 8) from NIZ-modeling. Indeed, in a NIZ- experiment, C_1 depends on the location of the temperature measuring points, and when, moreover, data are interpreted with the IZ-model, odd values for C_1 may result.

While the theoretical value for a sharp-edged orifice, C_1 = 0.61 (IZ model), is widely used, values of up to 0.75 are being reported (Hill et al. 1986, NIZ model). Mahajan (1986, 1987) studying various aperture configurations (NIZ experiment) found values as low as 0.33 for the mass transfer through a center door (IZ model). Moreover, the coefficients for mass and heat transfer are quite different.

Both full-scale and model experiments were performed by Kiel and Wilson (1986). While the (water-) model data (IZ) are in very good agreement with C_1 = 0.6, the full-scale results (NIZ) are consistently lower, showing a correlation with interzone temperature difference, $C_1 = (0.40 + 0.0045\,\Delta T)$. The authors show that this is because of the interfacial mixing across the counterflow interface, and at smaller temperature differences a stronger temperature gradient is observed. Nevertheless, the IZ model was used to obtain C_1 values.

Finally, Riffat (1988) correlated the mass and heat transfer between the lower and upper level of a dwelling (NIZ experimental situation), to the temperature difference between the zone centers (IZ model). The discharge coefficient C_1, characterizing the transfer through both door and staircase, is now 0.6 for interzone temperature differences of ≈ 1 K, decreasing to 0.25 for ΔT = 10 K, in complete contrast with the results of Kiel and Wilson (1986). It is clear that situations where Equation 1 does not strictly apply, yield C_1 values which have not the same meaning and are not comparable.

To see whether large deviations from the usual value C_1 = 0.6 are readily found, we have performed two tests in our laboratory.

(1) *Reduced Scale Model.* We have repeated some of the experiments of Lane-Serff et al. (1987), who studied transient flow through doorways produced by temperature differences.

The two-dimensional flow model we used had three compartments separated by vertically sliding doors. The model of the room or house was placed in the middle part in water. The other parts were filled with water in which salt was dissolved, and to which a coloring agent was added. The densities were measured with a calibrated density meter.The flow of the denser liquid through doorways of various height was videotaped, allowing us to measure the velocities with a time resolution of 1/25 s (one frame).

For a model door 1:10 in scale, the condition of dynamic similarity (same Reynolds and Froude numbers) was satisfied. The maximum Reynolds number we considered was 28,000 and the maximum density variation was 5% (ΔT = 5 K).

We noted an influence of the ceiling-to-door height ratio, Y, on the frontal gravity wave velocity (Simpson 1982). We reproduced the value C_1 = 0.5 for Y = 1 (the case where the ceiling and door height are the same, which was studied in Lane- serff et al. (1987), see Figure 2, dashed line). All experiments where Y > 1.5,

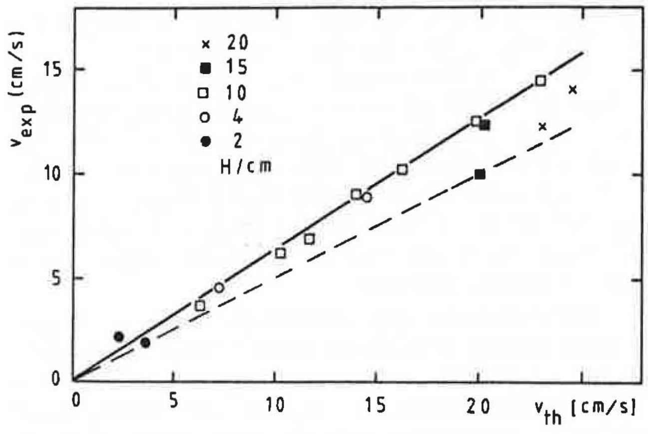

Figure 2 *The frontal velocity of gravity waves measured in a scale model,* v_{exp}*, as a function of the maximum theoretical velocity* $v_{th} = (\Delta\rho/\rho\ g\ H)^{1/2}$*. The discharge coefficient,* $C_1 = v_{exp}/v_{th}$*. The dashed line:* $C_1 = 0.5$*, for* $Y = 1$*. The full curve:* $C_1 = (0.63 \pm 0.02)$ *for* $Y > 1.5$*.*

yielded the same value $C_1 = (0.63 \pm 0.02)$ (see Figure 2, solid line).

(2) *Full Scale Test.* The aim of the second test was to measure the velocity profile in an open door between two zones at different temperature, and to compare the NIZ- and the IZ-interpretation. We heated one office room (2.8 x 3.4 x 4.2 m^3) of our LESO laboratory with an electrical fan heater (2.4 kW).

For practical reasons, the upper part of the door was covered with a board, reducing its height to 1.4 m. A velocity and temperature probe traveled up and down in the opening at a speed of 1 cm/s and the signals were recorded (probe time constant <1 s; low frequency turbulence (<1 Hz) of amplitude <5 cm/s. A typical (smoothed) result is shown in Figure 3. Using a fit of the experimental curve to the NIZ Bernoulli model with linear stratification (Equations 1 and 3, dashed curve) yielded $C_1 = 0.75$, consistent with a temperature difference of $\Delta T(z) = 1.5 + 1.5\ z$[°C], and a neutral level at 0.55% of the opening height. This result is in full agreement with Hill et al. (1986). However, using the IZ-model, the 20cm/s velocity measured at floor level, combined with the temperature difference between out and inflowing air of 3.6°C, yields a value of $C_1 = 0.5$. With decreasing interzone temperature difference, we expect the relative importance of stratification to increase (Equation 3), and the C_1 determined from a fit to the IZ-model to decrease, just as was found by Kiel and Wilson (1986).

We conclude that the discharge coefficients used in the Bernoulli model can confidently be chosen in the range 0.6 to 0.75, provided temperature stratification is taken into account. Presently, its precise numerical value can not be predicted with confidence, and for precision it is best to determine C_1 for any given experimental situation.

Transient Flow, A Filling Box

If multizone air infiltration modeling is to incorporate user behavior, allowing for the opening of doors and windows, then it is necessary to have a clear picture of the transient flow during the opening-closing cycle (Kiel and Wilson 1986).

As a door is opened, the transient response takes the form of a gravity current of cold air traveling along the floor and spreading out into the warmer room. In the case of a corridor, the advancing cold air in the gravity current front reflects from the end walls and returns to the door. The flow decreases until the space is filled with incoming air (Figure 1). Frontal velocities of gravity waves have been measured on scale models and in a full-scale experiment (Simpson 1982; Linden and Simpson 1985; Kiel and Wilson 1986; Lane-Serff et al. 1987). The constant velocity, u, of the gravity wave derives from an equilibrium between the inertial and buoyancy forces

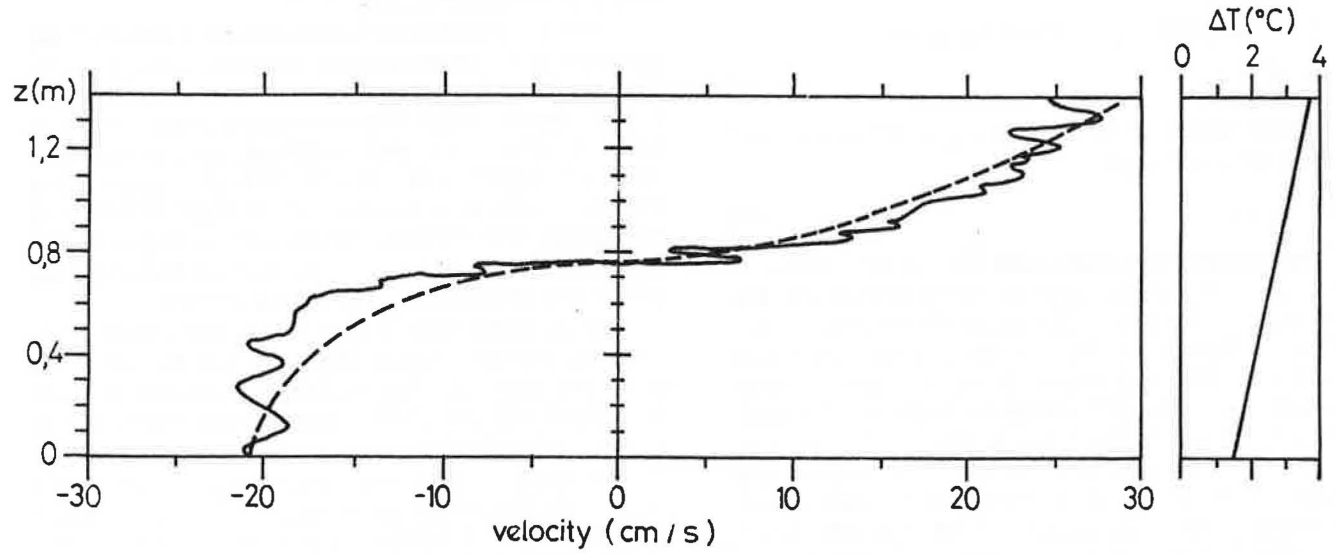

Figure 3 *Air velocity as measured in the vertical midplane of a door (H = 1.4 m, W = 0.85 cm). Dashed curve is calculated from the Bernoulli model with the parameters* $C_1 = 0.75$*,* $\Delta T(Z) = 1.5 + 1.5Z$ *°C.*

$$\frac{1}{2} \rho u^2 \approx (\rho_{in} - \rho_{out}) g z$$

where z is the height of the wave, which equals about half the aperture height, $z = H/2$. The Froude number $u/(g'H)^{1/2}$ now replaces the orifice coefficient, C_1, in Equation 1 (Simpson 1982).

When the open time interval is much longer than the opening and closing times, the latter can be neglected. The gravity-driven flow model then predicts a transient flow, steady flow persisting only until the finite interior volume causes a flow reduction. This reduction starts when the gravity wave is reflected, and continues to decrease exponentially with time.

The final situation is one of a very neat stratification at the top of the door level (Figure 1c). We have compared the predictions of this model, with the observed ventilation rate and resulting stratification for long opening times.

In full-scale measurements, it was immediately apparent that the heat stored in the walls plays an important role. While the inside air temperature decreases and becomes stratified, the process is slower and less pronounced than the gravity wave model predicts, because the entering cold air is heated by the floor and the walls. In fact, the latter model describes the dynamics of the process, i.e. where the exchange of heat with the walls is negligibly small (Kiel and Wilson 1986). The filling box model (Figure 1) then predicts that at most one volume air change takes place. The smallest amount of heat capacity of the walls, however, will make the flow to last and this requires thermal features to be included in the modeling.

THERMAL MODELS

The basic assumption of the following approach is that before opening a window to the outside, the zone is in thermal equilibrium with the building. Therefore, we consider only perturbations of an equilibrium situation, without having to know the steady state energy balance of the zone. One of the consequences of initial thermal equilibrium is the neglect of radiation. Radiation heat transfer couples the walls to each other, but is assumed to have negligible influence on the heat loss through the window. The heat loss after the opening of a window is treated here, as if a step in heating-power was applied to the room. The transient response of the average zone temperature is observed and modeled.

Thermal Losses of the Room

The heat loss of a room due to airflow by an opening to the environment can be calculated in terms of the heat exchanged with the air, flowing in through the lower half of the aperture and flowing out through the upper half. Assuming the condition for isothermal zones to apply Equation 4, which simplifies considerably the mathematics, the neutral level is simply at mid-height. For isothermal zones, the inflowing air temperature equals the outdoor air temperature, T_{out}, the outflowing air temperature equals the inside air temperature, T_{in}, and the heat transfer through the aperture is given by Equation 8.

Heat Transfer with the Walls

It is assumed that the walls of the zone are homogeneous, initially in thermal equilibrium with the building and at the same temperature.

The heat load to the room is transmitted to the walls by convection through a thermal boundary layer resistance. Assuming an average heat transfer coefficient hc, can be defined, the thermal resistance is $1/S_i\, h_c$. The difference between the wall temperature, T_{wall}, and the air temperature, T_{in}, is

$$T_{wall} - T_{in} = Q/S_i\, h_c$$

As we will see, the experimental results can only be understood when part of the total surface area, S_i, is effective in the heat transfer; this is the consequence of temperature stratification. In order to be able to quantify this effect, we substitute S_i by $C_2 S_i$, where the fraction C_2 is the only free parameter in the model, and we get

$$T_{wall} - T_{in} = Q/C_2\, S_i\, h_c \tag{9}$$

The average heat load per unit surface area, $Q/C_2\, S_i$, is the parameter that determines the heating or cooling of the walls and the inside air temperature.

In the following we will consider first the simple situation where Q is a constant source of heat (electrical fan heater). Later we will take for Q the heat flow through an aperture (Equation 8), so that Q depends on $T_{out} - T_{in}$. Equations 8 and 9 must then, be solved self-consistently in order to obtain a value for T_{in}.

The Wall Thermal Model

In studying the transient thermal behavior of the room we are interested in the time dependence of the surface temperature of the walls, T_{wall}, submitted to a boundary condition of heat-flux-density $Q/C_2\, S_i$. The solution of the heat equation for this situation can be found in Carslaw and Jaeger (1959) and is for a semi-infinite solid:

$$\Delta T_{wall}(t) = T_{wall}(t) - T_{wall}(0)$$

$$= \frac{1}{\lambda\, C_2\, S_i} \sqrt{\frac{a}{\pi}} \int_0^t Q\,(t-\tau)\, \tau^{-1/2}\, d\tau \tag{10}$$

where λ and a are the thermal conductivity and diffusivity, respectively. For a constant heat flux Q this simplifies into

$$\Delta T_{wall}(t) = \frac{2Q}{\lambda\, C_2\, S_i} \sqrt{\frac{a\, t}{\pi}} \tag{11}$$

It is seen that the model predicts a square root time dependence of T_{wall}.

For long time periods and thin walls, the wall cannot be considered semi infinite and it will be necessary to formulate the problem differently. A long time period here means that the wall thickness becomes smaller or equal to $\sqrt{a\, t}$. In our test room we have 10 cm concrete walls backed with 10 cm glass wool insulation on most of the surface area (a = 5 x 10^{-7} m^2/s). We expect to reach the limit of the validity of this model after five hours or less. However, experimentally we have observed no significant departures from the $\sqrt{t}$ time dependence after a 5 to 10 hour period.

The Heat Transfer Coefficient

It is assumed that an average heat transfer coefficient, h_c, can be defined. From the literature this idea receives little support at first sight. Free natural convection heat transfer in a cavity is invariably formulated in terms of correlations between Nusselt and Grashof numbers (ASHRAE 1985; Bauman et al. 1983). However convective heat transfer coefficients calculated from these correlations are small, typically ≤ 3 [W/m² - K] in buildings. Other sources learn that higher transfer rates are frequently encountered. Brau et al. (1983) use as overall transfer coefficient 6 [W/m² - K]. Lebrun and Ngendakumana (1987) study h_c as a function of heating power and measure (at 2 kW heating power) values on the floor, vertical surfaces and ceiling of 4, 5 and 7 [W/m² - K] respectively. Qingyan et al. (1989) measure convective heat transfer in rooms with mixed convection and find values for h_c as large as (5 ± 2) [W/m² - K] for air velocities below 1 m/s.

The heat transfer between a gravity current of air and the floor is certainly not a situation of "free convection," and we expect that values of h_c typical for mixed convection are more realistic. In view of these uncertainties we assume in the following a value

$$h_c = 6\ W/m^2 - K \tag{12}$$

This constant value for h_c, is certainly an approximation which needs further justification in the future. For the present simplified ventilation model, however, it turns out to be satisfactorily. If needed, replacement by a better and more detailed model for h_c, is possible.

Experimental Test of the Thermal Model

The thermal model has been tested for the simple situation of a closed room and a constant heat source Q. To avoid temperature stratification we used an electrical fan heater on the floor, which mixes the air during heating. Indeed stratification was not observed in this case. However, using a convective heat source alone, air temperature stratification is a natural consequence (Inard 1988).

We used heating powers between 0.6 and 3.6 kW. The air temperature was measured with a DISA flow analyzer system, the wall temperatures were scanned with a radiation infrared thermometer (see next section for experimental details). The readings of the latter device were checked against thermocouple thermometer readings in fixed locations. The radiation thermometer allowed us to detect very rapidly local changes in surface temperatures.

We expect to observe an initial, transient time-dependence of the air temperature, whose time constant is typical, for the heating of the air and the establishment of a steady convective regime. From Equation 9, measuring simultaneously the average temperature of the air and of the wall, we expect T_w - T_{air} to be proportional to Q, and to yield a value for h_c.

A typical test result is shown in Figure 4, where the measured air temperature is given as a function of time. The solid line is for the room with cupboard and desk, the dashed line is for an empty room. Applying Equation 11 (C_2 = 1), ΔT_w is calculated as a function of time and subtracted from the measured air temperature, yielding the dot-dashed curve. The wall-air temperature difference reaches a constant value after 10 to 20 minutes for the empty room, and almost double this time for the office room with furniture. Using Equation 9, we deduce for the heat transfer coefficient a value h_c = 6.6 ± 0.5 W/m²K, (Q = 0.6 to 3.6 kW). There is no systematic dependence on the temperature difference, T_{air} - T_{wall}, as expected for free natural convection (Bauman et al. 1983), and this supports our choice of a fixed value for h_c (Equation 12). This experiment also shows that the thermal model can be improved in the future, by including a description of the initial transient period, whose time constant depends on the presence of furniture.

In a second test a 1 kW electrical heater (placed under the window) was used. In Figure 5, the rise in air temperature is given as a function of time, for different heights at the center of the room. The upper curve (squares) was measured near the ceiling, the lowest curve near the floor. The solid curve from Figure 4, given for comparison (stars), follows closely the heating curve measured at 1.8 m height. It is worth noting, that the three upper curves have the same form, suggesting that they were all built up under the same conditions of constant heat flux. If this were true, the relative magnitude of the heat flux could be estimated from their

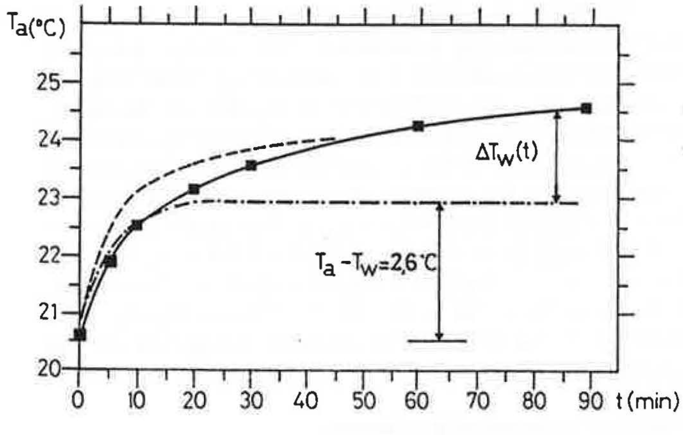

Figure 4 *The air temperature, T_a, measured in the middle of the room, as a function of time with (solid curve) and without furniture (dashed curve). A power step (1.2kW) is applied to the room at t=0. An electrical fan-heater was used to avoid temperature stratification. $\Delta T_w(t)$ was calculated and subtracted from T_a to yield the function, T_a - T_w (dot-dashed). The constant value for T_a - T_w (after an initial transient period of about 10 min without and 20 min with the furniture in the room) defines an average heat transfer coefficient.*

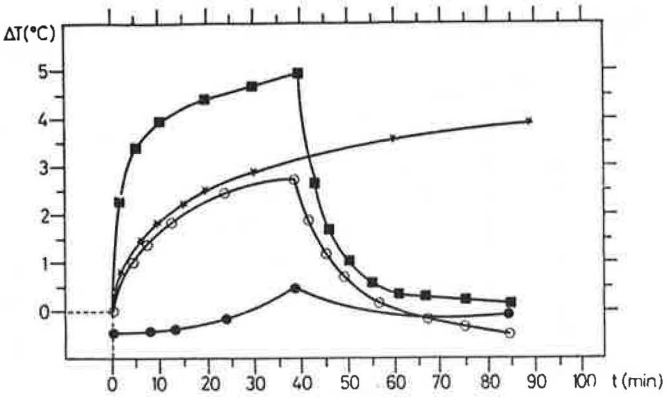

Figure 5 *At t = 0, an electrical heater-convector (1kW), placed below the window was put on for 40 minutes. Initially the room containing a desk and cupboard was, in thermal equilibrium with the building, at 20°C. Air temperatures were measured at the ceiling (squares, z = 2.8 m), at z = 1.8 m (open circles), and near the floor (filled circles, z = 0.1 m). For comparison with the non-stratified case is given the solid curve from figure 4 (stars).*

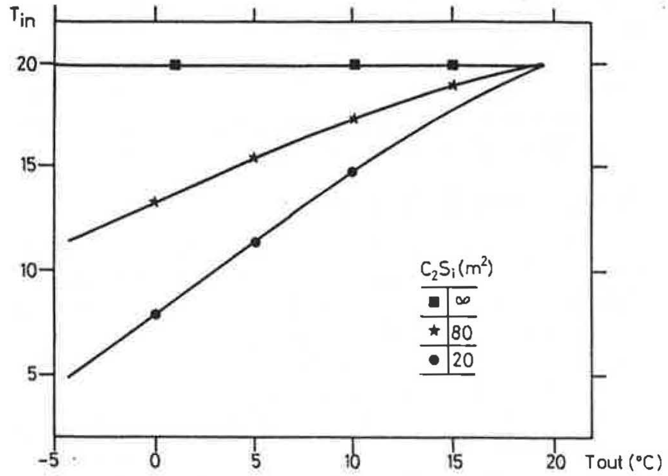

Figure 6 *The inside air temperature,* T_{in}, *as a function of* T_{out}, *calculated as the self-consistent solution of the airflow and the heat transfer equations of the present model. The curves correspond to difference values of the heat exchanging surface area* $C_2 S_i$ *(*C_1 *= 0.6,* T_w *is 20°C).*

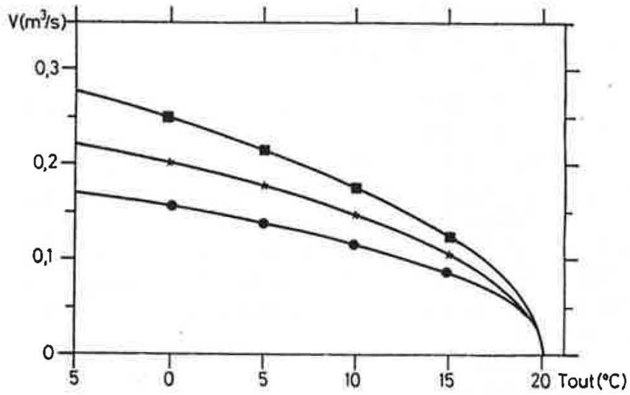

Figure 7 *Predicted ventilation rates corresponding to Figure 6.*

amplitude. The lowest curve is different in form, but in view of the large degree of temperature stratification, radiation coupling between ceiling and floor can be an important mechanism for the rise in floor temperature. Interestingly, we noted that the temperature stratification of the air and the vertical temperature profile on the wall are closely correlated.

COMBINED MODEL

Algorithms

By combining the following expressions

$$v_{max} = C_1 \sqrt{g/T}\sqrt{H \Delta T} \quad (1)$$

$$\Delta T = T_{in} - T_{out} \quad (2)$$

$$V = 1/3\, W H\, v_{max} \quad (6)$$

$$Q = C_p \rho\, V \Delta T \quad (8)$$

$$T_{in} = T_{wall} - Q/(C_2 S_i h_c) \quad (9)$$

$$T_{wall} = T_w(0) - \frac{2Q}{C_2 S_i \lambda}\sqrt{\frac{a\,t}{\pi}} \quad (11)$$

$$h_c = 6\ W/m^2 - K \quad (12)$$

and solving self-consistently for T_{in}, the inside air temperature is obtained in the approximation of isothermal zones (Equation 4).

Predictions of the Combined Model

In Figures 6 and 7 some typical predictions of the model for the inside air temperature and the ventilation rate are given, as a function of outside air temperature. The parameters chosen for this simulation are T_{wall} = 20°C, W = 0.8 m, H = 1.5 m, C_1 = 0.6. The parameter is the heat exchanging surface area $C_2 S_i$. For large values of this surface area, the inside air temperature is independent from T_{out}. Taking 0°C outside temperature as an example, the temperature in the room is lowered from 20° to 8°C upon opening the window in a small room, and to 13°C in a larger room (Figure 6). The air change rate is at the same time reduced by 10%, and 20% respectively (Figure 7). However, in reality where the air is stratified, temperature changes are more important, because C_2 is found to be smaller than 1.

EXPERIMENTS

Room Description

The room we used for the experiments is one of the offices of the experimental LESO building; its volume is 2.8 x 3.4 x 4.2 = 40 m³ and the total wall surface area is 70 m². The room has 10 cm thick concrete walls except for one side wall, which is 1 cm plaster (Figure

9, IV). All the walls, floor, and ceiling have 10 cm glass wool insulation as a second layer. The windows (total area = 5 m^2) are double glazed with a selective coating. Their U-value is 1.5 $W/m^2 \cdot K$. The openable window has dimensions of 1.1 x 0.85 m^2 and provisions were made to be able to partly cover the aperture with a board.

Measurement Methods Used

The air temperature and velocity ware measured with a DISA flow analyzer system. The temperature transducer is a 50 micron diameter thermistor. The absolute accuracy of the temperature measurements is ± 0.5°C (at air velocities > 5 cm/s), with a time constant < 1 s. The temperature differences in our experiment never exceeded 20°C, and Inard (1988) has shown that for small dimensions of the sensor, the influence of radiation on the temperature measurement is negligible (< 0.1°C). The velocity sensor is a calibrated, omni-directional, constant temperature anemometer, with a time constant < 0.1 s. Its measuring accuracy is 2.5% of reading and the influence of the air temperature is < 0.2% / °C.

Wall temperatures were scanned with a radiation infrared thermometer (Thermophil-Infra T-203). Using the built-in black cell for calibration, measurements were reproducible within 0.5 °C. The readings were checked against the temperatures measured with thermocouples in fixed locations on the wall. The radiation thermometer allowed us to detect rapidly local changes in surface temperature.

We used heating powers in steps of 0.6 kW. The air velocity in front of each fan heater (max 1.2kW) decreased from 2m/s, to <0.2m/s at a distance of 2m. At the level of the DISA thermometer the velocities varied between 5 to 10 cm/s. Velocity and temperature profiles were recorded by mounting probes on a motor-driven trolley.

Experimental Results

In order to test our ventilation model, which takes the cooling of the room air into account, we measured the air and wall temperatures as a function of time after the opening of a window (Figure 8). The measurement was performed during a windless cold night. The outside temperature, initially at 5.3°C, drifted slowly to 6°C the following morning, for which correction was made.

During the experiment, the temperature stratification of the air at the window level was about 1°C/m, and the inside outside temperature difference was sufficiently high for the isothermal zone approximation to be valid (Equation 4). Measurements of velocity profiles in the window showed consistency with C_1 = 0.6, a value which we will use in the following. The room air temperature, T_{in}, which was considered relevant for the ventilation process, was measured in the outflowing air.

In Figure 8, the time dependence of T_{in} is given for a ten hour period. The initial homogeneous room temperature was 21.5°C. The outflowing air temperature dropped rapidly to 15°C, decreasing then slowly to 12°C in the following 10 hours. The model can explain the initial drop to 15°C provided the effective heat

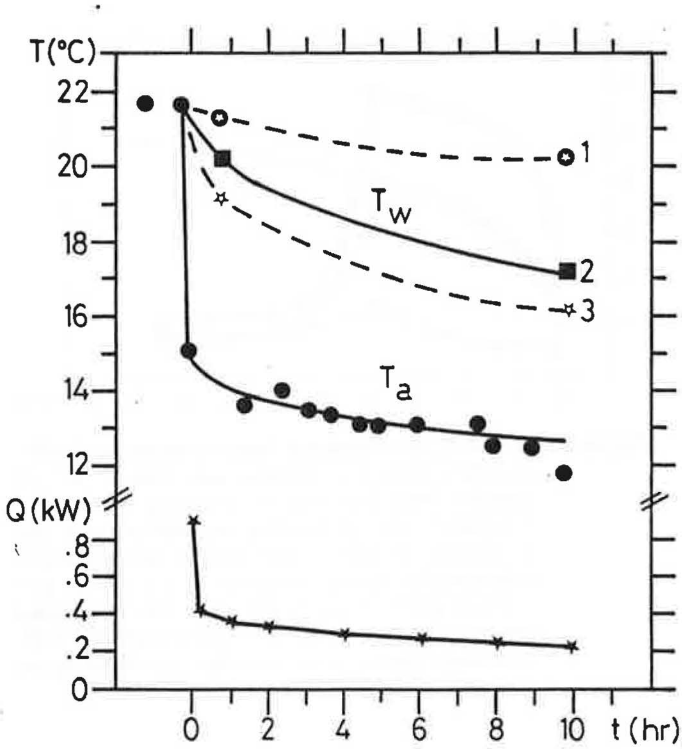

Figure 8 *Inside air temperature, T_a, and wall temperature, T_w, after opening the window at t = 0 to the environment (5.5°C). The curves for T_w, T_a and the heatflow Q, are self-consistent solutions of the airflow and the heat transfer equations of the present model (C_1 = 0.6 and C_2 = 0.5). The inside air temperature, T_a, was measured in the out-streaming air near the top of the window (filled circles). Average values for the wall temperature, T_w (see also Figure 9), at the ceiling (1.: z = 2.8m), at the floor (3.: z = 0.1 m), and at the level of the outstreaming air (2, z = 1.8 m). Wall surface area: 70 m^2. Window: H x W = 1.1 x 0.85 m^2.*

exchanging surface area is 50% of the total area (S_i = 70 m^2, C_2 = 0.5). The surface area coefficient C_2 is the only free parameter in the model, and its value of 0.5 corresponds roughly to the wall surface below the upper level of the window. This would be the surface predicted by the filling box model (Figure 1).

Wall temperatures, initially at 21.5°C, have been measured after one hour, and at the end of the experiment. Some average values are given in Figure 8, a more detailed picture at the end of the experiment is given in Figure 9. The ceiling temperature (Figure 8) changed by 0.5°C in the first hour and was still at 20°C at the end of the run. The temperature at a height of 1.5 to 1.8 m, changed 1.5° in the first hour and dropped further to 17°C at the rear wall. The floor temperature dropped from 21.5° to 19°C in the first hour, with a final average temperature of 16°C.

In the lower part of Figure 8, the heat transfer through the window as used in the self consistent calculation is given. The heat transfer rate drops to about 25% of the prediction of the steady flow model. The map of the wall temperatures at the end of the run

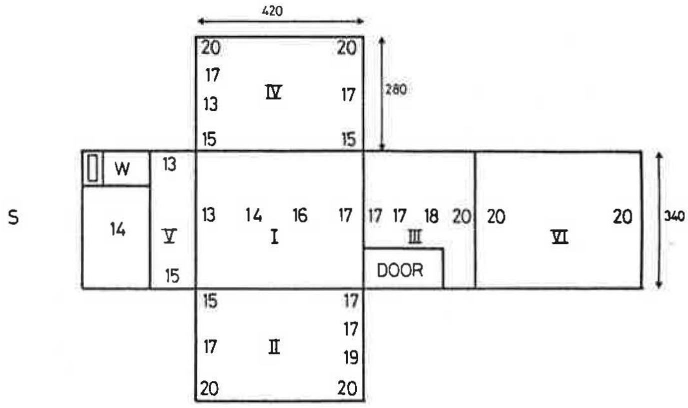

Figure 9 *The wall temperatures in the test room after opening the window for 10 hours (compare Figure 8). T_{out} = 6.6°C. Initial room temperature 21.5°C.*

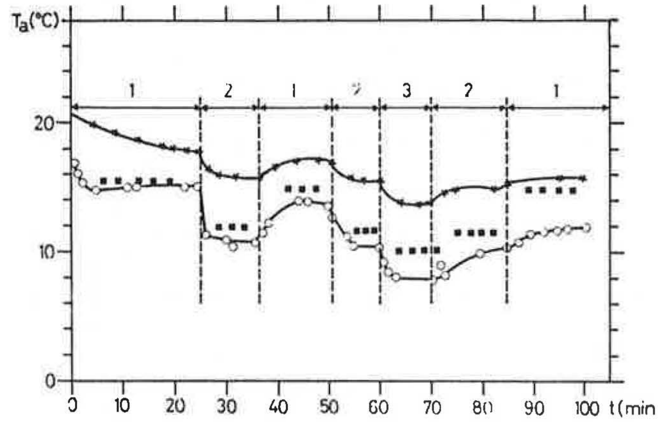

Figure 10 *The air temperature T_a after opening the window at t = 0; outside temperature T_{out} = 0°C. The width of the window is varied about every 10 min.: 1) W = 27 cm; 2) W = 63 cm; 3) W = 85 cm. T_a is measured in the middle of the room, near the floor (circles, z = 0.1 m) and at a level corresponding to the outflowing air stream (stars, z = 1.8 m). Squares: the inside air temperature calculated as the self consistent solution of the airflow and the heat transfer equations of the present model (C_1 = 0.6 and C_2 = 0.5).*

(Figure 9) shows that the temperature distribution is rather homogeneous below the upper level of the window frame, with a vertical gradient of about 2°C/m above it. Also, we note in Figure 9 a horizontal temperature gradient of about 1°C/m on the floor.

This longer-term experiment was complemented by a second experiment (empty room, T_{out} = 0°C), where the variable was the width of the window (Figure 10). The width was (Figure 10, numbers 1 to 3), one-third, two-third, or 100% by placing a board in the opening. Air temperatures have been measured in the middle of the room, the outside temperature is 0°C. About every 10 minutes a change is made, and (Figure 10, upper curve) the air temperature at a height of 1.8 m, is found to follow less rapidly than the air temperature at 0.1 m from the floor (lower curve). This effect is strongest during the first period of 10 minutes. On changing the window width the temperature stabilizes in typically 5 minutes or less. There is an overall cooling of the room, which can be noted while looking at the temperatures for window width 2 at 30, 60, and 90 minutes from the start of the experiment. The calculated inside temperatures with C_2 = 0.5, are indicated by square symbols and fall in between the two curves. Previously we have compared the outstreaming air temperature with the calculated T_{in}, and we expect the latter to be representative for the upper curve.

DISCUSSION

In a multizone air infiltration model, the temperatures of the zones are given for the steady state. In order to account for the drop in air temperature after the opening of a window, we have proposed a thermal model for the heat transfer from the air to the walls. This model seems only to be interesting in cases where wind velocities are very low. This is only partly true, because in order to calculate the heat-losses due to wind induced ventilation the inside temperature should be known, and this can only be achieved by using a heat transfer model. As a first approximation to the ventilation due to the combined effects of wind and temperature difference, the model of De Gids and Phaff (1982) could be used. But new measurements are needed in order to validate a model including wind.

The experimental data to which this model was compared (Figure 8 and 10), satisfied the isothermal zone approximation (Equation 4). When data come available with small inter-zone temperature differences and/or apertures of considerable height (like hangars), this approximation is not longer valid. At best the temperature profile is approximated to be linear, and the neutral height is calculated in every iteration step. Without an additional model for the stratification, coupling the gradient a to $\Delta T(0)$ (Equation 3), the set of equations can not be solved. An arbitrary temperature profile, described by more than two parameters (more than one neutral height is possible, see Hill et al. 1986), seems too complicated for use in an iteration scheme. Also for very small temperature differences the situation complicates, because there is a transition between bulk density difference flow and boundary layer flow regimes (Scott et al. 1988).

The various ingredients of the combined airflow and heat transfer model as exposed in this paper, can be modified and improved. But the essence of what we propose is that the equations are to be solved self-consistently.

We have assumed that no convective heat source is present in the room, which is not realistic. A better modeling of the mixed convection heat transfer is certainly desirable. Indeed, we would expect from the numerous papers on convection heat transfer coefficients, different values for h_c on the floor, ceiling and vertical walls (ASHRAE 1985; Allen et al. 1986; Lebrun and Ngendakumana 1987; Qingyan et al. 1989), and it would be interesting to extend numerical calculations on the mixed convection problem (Simoneau et al. 1988) to single sided ventilation.

A more detailed model must be developed for the case where the walls are not semi-infinite and/or made of very different materials. Also, the use of a constant heat flux for T_{wall} (Equation 11), for the time dependent flux through a window (Figure 8 and 10) needs a justification. Of course, for the general case where the heat flux is a function of time, Equation 10 should be used and the wall temperature depends on its detailed time history. In using Equation 11, we replace the real time history by a history of constant Q. This is acceptable in the present case because our room has a large thermal mass and the wall temperature varies slowly. For rapid variations of T_{wall} and large with respect to T_{air} - T_{wall}, this approximation will not be correct.

Stratification is an important effect when the heat source is a convector. Models have been developed to describe the development of air temperature stratification (Lebrun and Ngendakumana 1987; Inard 1988). The experimental results in Figure 9, show also the development of a strong temperature stratification after opening the window. This stratification resulting from the cold air entering the room and flowing over the floor is similar to the stratification resulting from a local heat source with hot air flowing along the ceiling. The models of Lebrun and Ngendakumana (1987) and Inard (1988), could probably be adapted to the ventilation problem, improving the present situation where various degrees of stratification are only taken into account through the parameter C_2. While we do not consider wind effects in this paper, it is worth mentioning that we have observed that a turbulent wind tends to increase the mixing and to break up stratification. So the effects of wind and temperature are profoundly connected.

Experiments have only been done on one type of test room. While the agreement between the model and the observations is good for the present case, we expect the stratification and therefore the effective heat transfer surface area (C_2S_i), to depend on the vertical location of the window. This effect has not been investigated yet.

Finally, in Figure 11, we show the ventilation rate as used in the calculation for Figure 8, compared to the steady flow model, and the gravity wave / filling box model. The latter two models cannot explain that the ventilation continues at a reduced rate, and we feel that the present model gives a more realistic description of ventilation.

CONCLUSION

We have compared the available algorithms for gravity-driven airflows through large openings with the requirements for multizone air-infiltration modeling.

It appears that two extreme situations are addressed in the literature. The first, providing a time-independent airflow, is formulated in terms of the constant temperature difference between a zone and the outside. The other, considering adiabatic walls, provides a transient flow, with the cold outside air filling

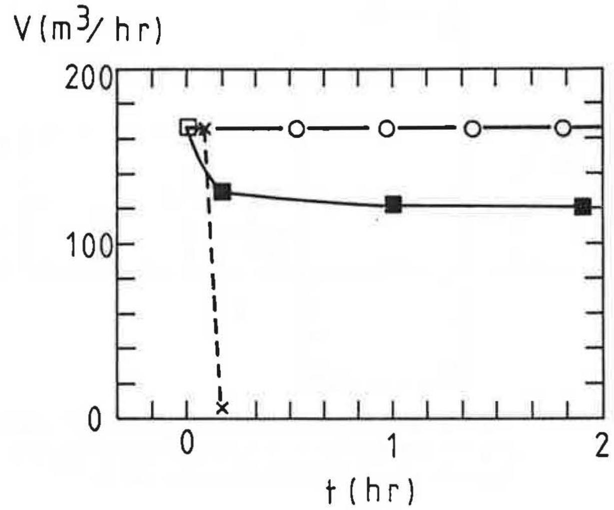

Figure 11 *The ventilation rate due to inter-zone temperature difference after opening a window at t = 0. Circles: steady state model. Dashed: transient flow, a filling box, see Figure 1. Squares: the self-consistent solution of the airflow and the heat transfer equations of the present model (C_1 = 0.6 and C_2 = 0.5).*

the zone space completely so that the airflow comes to a halt.

As an intermediate situation we have presented an algorithm that takes into account the heat transfer with the walls. The coupled equations for airflow and heat transfer are solved for the inside air temperature. The latter being lower than the wall temperature and always higher than the outside temperature, the proposed model predicts an air and heat flow rate which is intermediate in magnitude with respect to the extreme situations addressed in the literature. For constant wall temperatures, the model predicts that the air flow rate is lower for zones with smaller heat exchanging wall surface area. Coupled to a thermal model for the wall surface temperature, the algorithm predicts the air flow to decrease with time.

Experiments were performed on a full-scale test room with a window to the outside. During cold windless nights, velocity profiles were measured in the window plane, and the temperatures of the inside air and the walls were recorded with time. Good agreement with the model is obtained, the only free parameter being the fraction of the wall surface area active in the heat transfer process, and measuring the inside air temperature in the outstreaming air.

We have presented an improved algorithm for the airflow through large openings, taking heat transfer with the walls into account. This algorithm is better suited for the use in detailed multizone air infiltration models, where user behavior is taken into account.

FUTURE RESEARCH

Heat and mass transfer through large apertures in buildings cannot be disconnected from thermal modeling. In order to complete the model as presented in this paper, the following areas will be part of our future research:

a) Wind effects; the presence of wind increases ventilation, and modifies the heat transfer problem (mixed convection.

b) Non-isothermal zones; the aperture flow is modified by the stratification, and the stratification develops with time.

c) The influence of furniture on the initial transient.

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NOMENCLATURE

a	=	thermal diffusivity [m^2/s]
g	=	gravitational acceleration [m/s^2]
g'	=	g ((insert)out - (insert)in) / (insert) [m/s^2]
h_c	=	heat transfer coefficient [W/m^2K]
t	=	time [s]
v	=	velocity [m/s]
z	=	height in aperture [m]
C_p	=	specific heat [J/kg - K]
C_1	=	discharge coefficient [-]
C_2	=	surface area coefficient [-]
H	=	height of aperture [m]
H_n	=	height of neutral level [m]
Q	=	heat flow [W]
S_i	=	wall surface area [m^2]
T	=	mean absolute air temperature [K]
T_{in}, T_{air}	=	inside air temperature [°C]
T_{wall}	=	wall surface temperature [°C]
T_{out}	=	outside air temperature [°C]
V	=	volume flow rate [m^3/s]
W	=	door width [m]
Y	=	ceiling-to-door height ratio
λ	=	thermal conductivity [W/m-K]
ρ	=	mean air density [kg/m^3]
ρ_{in}	=	density of indoor air [kg/m^3]
ρ_{out}	=	density of outdoor air [kg/m^3]
ΔP	=	pressure difference [Pa]
ΔT	=	$T_{in} - T_{out}$ [°C]

REFERENCES

Allen, J.P., and Whittle, J.K. 1986. "An appraisal of convection coefficient algorithms in dynamic thermal models." *Proc. 10th Congr. Int. Council Bldg. Res, CIB 86*, Vol. 3, pp. 799-806.

ASHRAE. 1985. *ASHRAE handbook—1985 fundamentals*, chapter 3. Atlanta: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.

Bauman, F.; Gadgil, A.; Kammerud, R.; Altmeyer, E.; and Nansteel, M. 1983. "Convective heat transfer in buildings: recent Research Results." *ASHRAE Transactions*, Vol. 89, Part 1A, pp. 215-230.

Brau, J.; Achard, G.; Allard F.; and Pallier, M.M. 1983. "Comparison of different thermal models for buildings with experiments in artificial climatic conditions." *ASHRAE Transactions*, Vol. 89, Part 1A, pp. 23-24.

Brown, W.G., and Solvason, K.R. 1962. "Natural convection through rectangular openings in partitions—1." *Int. J. Heat Mass Transfer*, Vol. 5, pp. 859-868.

Carslaw, H.S., and Jaeger, J.C. 1959. *Conduction of heat in solids*, 2nd edition. London: Oxford University Press.

Clarke, J.A. 1985. *Energy simulation in building design*. Bristol, UK: Adam Hilger Ltd.

De Gids, W., and Phaff, H., 1982. "Ventilation rates and energy consumption due to open windows." *Air Infiltration Review*, Vol. 4, No. 1, pp. 4-5.

Herrlin, M.K. 1985. "MOVECOMP: A static multicell airflow model." *ASHRAE Technical Data Bulletin, Infiltration and Air Leakage*, pp. 128-135.

Hill, D.; Kirkpatrick, A.; and Burns, P. 1986. "Analysis and measurements of interzonal natural convection heat transfer in buildings." *Transactions of the ASME Journal of Solar Energy Engineering*, Vol. 108, pp. 178-184.

Inard, C. 1988. "Contribution a l'Etude du Couplage Thermique entre un Emetteur de Chauffage et un Local." Ph. D. Thesis, INSA, Lyon, France.

International Workshop COMIS. 1988, 1989. Conjunction of Multizone Infiltration Specialists at Lawrence Berkeley Laboratory, Berkeley, CA.

Kiel, D.E., and Wilson, D.J. 1986. "Gravity driven flows through open doors." 7th AIVC Conference, Stratford on Avon, UK, September 29.

Lane-Serff, G.F.; Linden, P.F.; and Simpson, J.E., 1987. "Transient flow through doorways produced by temperature differences." ROOMVENT 87, Session 22.

Lebrun, J., and Ngendakumana, P. 1987. "Air circulation induced by heating emitters and corresponding heat exchanges along the walls: test-room results and mdeling." ROOMVENT 87, 2a, Stockholm.

Liddament, M. 1986. *Air infiltration calculation techniques AIVC*. Warwick, UK: Air Infiltration and Ventilation Center.

Linden, P.F., and Simpson, J.E. 1985. "Buoyancy driven flow through an open door." *Air Infiltration Review*, Vol. 6, pp. 4-5.

Mahajan, B.M., and Hill, D.D. 1986. "Interzonal natural convection for various aperture configurations." *ASME Winter Annual Meeting*, Anaheim, CA.

Mahajan, B.M. 1987. "Measurement of interzonal heat and mass transfer by natural convection." *Solar Energy*, Vol. 38, pp. 437-446.

Pelletret, R. 1988. "Inter-zone air movement and heat transfer Modeling." *Proc. User 1 Work. Conf. SCS Int. KIH*, Ostend, Belgium, September.

Qingyan, C.; Meyers, C.A.; and van der Kooi, J. 1989. "Convective heat transfer in rooms with mixed convection." *Proc. Int. Seminar on Air Flow Patterns in Ventilated Spaces*. Liege, Belgium.

Riffat, S.B. 1988. "A study of heat and mass transfer through a doorway in a conventional house." Research in Building Group, Polytechnic Central London, January, # AIVC 2454.

Scott, D.; Anderson, R.; and Figliola, R. 1988. "Blockage of natural convection boundary layer flow in a multizone enclosure." *Int. J. Heat and Fluid Flow*, Vol. 9, pp. 208-214.

Simoneau, J.P.; Inard, C.; and Allard, F. 1988. "Numerical approach of interaction between an injection and laminar natural convection in a thermally driven cavity." ASME Winter Annual Meeting, Chicago, Vol. 99, pp. 45-51.

Simpson, J.E. 1982. "Gravity currents in the laboratory, atmosphere and ocean." *Annual Review of Fluid Mechanics*, Vol. 14, pp. 219 - 234.

Warren, P.R. 1986. "The analysis of single-sided ventilation measurements." *Air Infiltration Review*, Vol. 7, No. 2, pp. 3-5.