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Transient single sided ventilation through large openings in buildings

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Summary

To model transient single sided ventilation through a large opening of a room we take heat transfer between the air and walls into account, and a varying wall temperature. The ventilation rate expressed as a function of inside-outside temperature difference varies with time due to the cooling of the walls. Parameters used in the model are the room wall surface area and the thermal properties of the walls.

The velocities and temperatures in the window opening determine the ventilation heat loss, which is measured independently through the analysis of the time dependence of the wall surface temperatures. Tests are performed on full scale rooms (heavy and light structures). with the heating system turned off. It is found that for wind velocities near the window opening less that 1m/s and temperature differences inside-outside larger than 5 °C, the model predicts correctly the magnitude of the observed temperature drop after opening a window. The indoor climat, the transient ventilation and heat loss rate predicted by the model can be used in building simulations to evaluate user behaviour.

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Abstract

To model transient single sided ventilation of a room we take heat transfer between the air and walls, and a varying wall temperature into account. The ventilation rate expressed as a function of inside-outside temperature difference varies with time due to the cooling of the walls. Parameters used in the model are the room wall surface area and the thermal properties of the walls.

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Introduction

The modeling of single sided ventilation is an essential part of single and multizone air infiltration and ventilation models [1,2]. In order to be realistic, user behaviour in regard to the opening of doors and windows must be implemented [3,4]. Because the problem of ventilation and energy loss by single sided ventilation is of transient character, it is necessary to give a description of the flow as a function of time [5].

Single sided ventilation has been studied by various groups notably in Holland and the UK [6, 7, 8 and 9]. Ten years ago De Gids and Phaff [6,7] investigated the consequences of opening one window on the internal climate of a room on homes in Holland, deriving an empirical expression for the ventilation rate due to the combined effect of temperature difference and wind. The influence of wind turbulence in the window opening was the subject of mainly windtunnel studies [8,9].

Transient flow has been studied by various authors [10,11,12], by studying the way a gravity wave of cold air enters through a door and how cold air fills up the space [10]. However these models assume no heat exchange with the walls, i.e. adiabatic walls, the air temperature in the room drops rapidly to the outside temperature and the airflow comes to a halt.

However our day to day experience tells us that when we open a window in the winter the temperature is reduced but the room temperature is maintained well above the outside temperature (even when the heating equipment is turned off) by the heat stored in the walls. Ventilation is coupled to thermal modeling.

In a previous paper [5] we have presented a model which describes what happens after the first moments when the gravity current of cold air has filled the space once with fresh air.

The model was first tested on a (large thermal mass) room of our experimental LESO building [5], and found to predict correctly the initial drop in inside air temperature and the subsequent decrease with time. The thermal modeling is very simple and requires to know (1) the inside outside temperature difference before opening (2) the room wall surface area and, necessary for the time dependence over a few hours, (3) the thermal properties of the room walls. Future developments of the model will include wind effects, the time development of temperature stratification and the transient of the first minutes which depends on furniture in the zone.

In the following we present new measurements on a light-weight structure (where the time variation of temperature is very significant), and compare our model predictions with the temperature drops observed by De Gids and Phaff [6].

Coupled ventilation and thermal model

Ventilation modeling

Isothermal zone approximation. We have found previously [5] that the simple Bernoulli model is satisfactorily when the zones connected by the large opening can be considered as isothermal.

Because temperature stratification can hardly be avoided we call a situation isothermal, when the room temperature variation between the top and bottom of the opening is small compared to the interzone temperature difference.

This means for example that a gradient of 2 [°C/m] in a room with an open window can be called isothermal when the window height is one meter and the inside-outside temperature difference ΔT is larger than about 10 °C. The velocity profile in the opening will be close to symmetric.

On the other hand a gradient of 1 [°C/m] in a room with an open door to a zone which differs just 2 °C, is a strongly non-isothermal situation with an asymmetric velocity profile in the door.

Discharge Coefficient. Provided temperature stratification is taken into account, we concluded that the discharge coefficient is in the range C_1=0.6-0.7 [5]. For isothermal situations no values differing by more than 10% from 0.6 have been reported. Therefore as long as we do not know the origin of the spread in values we adopt the universal value of 0.6.

Bernoulli Model. The expressions for the maximum velocity in the opening, the inside outside temperature difference, ΔT, the volume flow, $\dot{V}$, and the ventilation heat loss rate, Q_v, are :

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

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

(3) $$\dot{V}=1/3\ W\ H\ v_{max}$$

(4) $$Q_v=C_p\, \rho\, \dot{V}\ \Delta T$$

The ventilation heat loss rate, Q_v, is a non-linear function of the inside-outside temperature difference through (1).

Thermal modeling

Basic Assumption. The basic assumption of the following approach is that before opening a window to the outside, the zone is in quasi 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 opening. The heat loss after the opening of a window or door 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 compared with the model prediction.

Heat transfer with the walls. It is assumed that the internal walls of the zone are homogeneous or that average wall properties can be defined. The heat load to the room is transmitted to the walls by convection through a thermal boundary layer resistance. Assuming that an average heat transfer coefficient h_c can be defined, the thermal resistance equals $1/S_i h_c$ [W/K], where S_i is the total wall surface area, including ceiling and floor.

For a heater power Q_H applied to a room, the average heat flow density is $q = Q_H/S_i$ and the difference between the air temperature, T_{in}, and the wall surface temperature, T_{wall}, is

(5) $$T_{in} - T_{wall} = q / h_c = Q_H/S_i h_c$$

Before the heater is switched on, q=0 and $T_{in} = T_{wall}$. As was shortly discussed in [5], and in more detail in [15], after putting the heater on at t=0, the air temperature rises exponentially to the limiting value given in expression (5). The time constant depends on the heat capacity of the room air and the furnishing of the room. In most practical cases the time constant is of the order of minutes. In this paper we have neglected this second order effect, but it is object of further study.

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 wall, T_{wall} , submitted to a boundary condition of heat flow density q. The solution of the heat equation for this situation can be found in Carslaw and Jaeger (1959) and is for a semi-infinite solid :

(6) $$\Delta T_{wall}\ (t) = T_{wall}\ (t) - T_{wall}\ (0) = \frac{1}{\lambda}\sqrt{\frac{a}{\pi}} \int_0^t q\,(t-\tau)\,\tau^{-1/2}\, d\tau$$

where the wall material constants, λ and a, are the thermal conductivity and diffusivity respectively. For a constant value of q, Equation 7 simplifies to

(7) $$\Delta\, T_{wall}\ (t) = \frac{2\,q}{\lambda}\sqrt{\frac{a\ t}{\pi}} = R_{dyn}\, q$$

where R_{dyn} is the dynamic thermal resistance of the wall, which is zero at time t=0, and increases with the square root of time. Even when q does vary with time, the simpler expression (8) can sometimes replace (6) as a first approximation. This means that the real time history of q is replaced by a history of constant q. Whether this is justified should be discussed for every particular situation.

Thin walls. For long time periods and thin walls, the wall cannot be considered semi infinite (as required by Eq. 6 and 7) and it will be necessary to formulate the problem differently. But when should the wall be considered thin? The thermal penetration depth is $\sqrt{a\ t}$ and a long time period means that the wall thickness, e, becomes smaller than this value. Equivalently from Eq. 7, we require that the dynamic resistance, R_{dyn} , is smaller than the static thermal resistance (e/λ).

In the LESO test room we have e=10 cm concrete walls ($a = 5 \times 10^{-7}$ [m^2/s], λ=0.76 [W/m^2K]) backed with 10 cm glass wool insulation on most of the surface area. From the thermal penetration depth, we expect to reach the limit of the model after three hours.. However, experimentally we have observed no significant departures from the $\sqrt{t}$ time dependence after a 10 hour period.

We have also tested a lightweight structure (Portakabin) with e=5cm thermal insulation (polystyren, for which nominally $a=15 \times 10^{-7}$ [m^2/s] and λ=0.04 [W/m-K]) and a square-root time dependence was observed over 4 hours, while the limit of validity is expected to be 30 minutes

From these tests we expect that in most practical cases the model is valid for at least one hour.

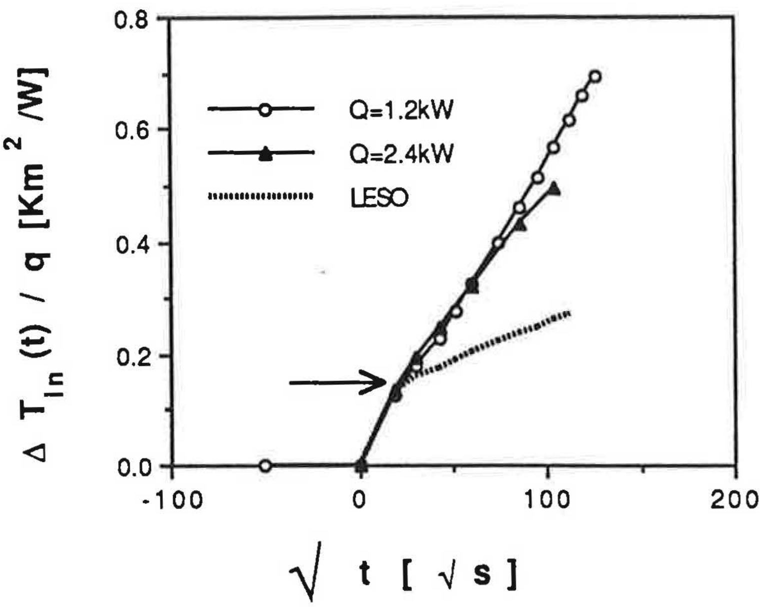

Fig.1 The measured change in room air temperature, normalized to unit heat flow density plotted as a function of the square root of time. From Eq.8, $\Delta T_{in}/q$, corresponds to $R_{dyn}(t)+1/h_c$. , with neglect of the heat capacity of the air and of the furnishing. The data for the light weight structured Portakabin are given by symbols and the more heavy structure of the LESO (Ref. [5]) by a dashed line. The furnishings of both rooms are comparable. The arrow indicates the experimental value of the boundary layer resistance $1/h_c$=(0.15±0.02) [Km^2/W] typical for the fan heater.

Experimental Test of the Thermal Model. The thermal model has been tested for the situation of a furnished closed room and a constant heat source Q_H. 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.

The average room air temperature T_{in} is measured, and combining Eq. (5) and (7), the change in room air temperature $\Delta T_{in}(t) = T_{in}(t) - T_{in}(0)$ equals

$$(8) \qquad \Delta T_{in}(t) = \Delta T_{wall}(t) + Q_H / S_i h_c = (R_{dyn}(t) + 1/h_c)\, q$$

Therefore $\Delta T_{in}/q$ plotted as a function of $\sqrt{t}$ is expected to yield the constant value $1/h_c$, and R_{dyn} from the slope of the following linear variation.

Results are given in Figure 1 (symbols, Portakabin; dashed line, LESO data [5]). Instead of jumping to the value of $1/h_c$ at t=0, the rise takes some time (arrow) because of the heat capacity of the air and the furnishing, which are neglected in the model. The slopes beyond the arrow yield experimental values for $\sqrt{a}/\lambda$ of $(1.0\pm0.3)\times10^{-3}$ and $(4\pm1)\times10^{-3}$ $[Km^2/(W\sqrt{s}]$ respectively for the LESO and the Portakabine.

The arrow indicates the value of the heat transfer coefficient (6.6 ± 0.5) $[W/Km^{2}]$, which is consistently the same in all our experiments with the fan heater. This seems to be quite a characteristic value for a fan heater system, because a similar value of 6.31 $[W/Km^{2}]$ was reported in a field study of over thirty rooms of various shapes and sizes, furnished and unfurnished, and heated by an electrical fan convector heater [15].

In this way it is possible to measure experimentally the $\sqrt{a}/\lambda$ value for the test room and to use this value for the test of the combined ventilation and thermal model. While the value for the LESO corresponds to the characteristics of the homogeneous building material, it was a surprise that an experimentally well defined value was obtained for the Portakabin which has not a homogeneous wall structure due to various thermal bridges.

Coupled Ventilation and Thermal modeling

It is not obvious how ventilation heat loss can be modeled with the thermal model from the previous section. While the heat source Q_H can be replaced by the ventilation heat loss rate $-Q_v$ (Eq. 4), it is not clear how to determine the heat flow density q, or the heat transfer coefficient h_c.

Temperature stratification coefficient C_2. In many cases the opening does not reach up to the ceiling, and only the wall surface below the upper level of the opening will receive the full cooling load. Temperature stratification will result, the walls above the opening will stay warmer. (The ceiling will cool however by radiation from the cold floor). In order to be able to quantify this effect, we substitute S_i by $C_2 S_i$, where the stratification coefficient C_2 becomes the only free parameter in the model which in most cases is between 0.5 and 1. From (5) the inside air temperature below the upper level of the large opening becomes

(9) $$T_{in}(t) = T_{wall}(t) - Q_V / C_2 S_i h_c$$

The stratification as caracterized by C_2, is not at the level of the opening and therefore it should have no incidence on the validity of the isothermal zone approximation for the Bernoulli flow model.

Heat Transfer Coefficient. The assumption of an average heat transfer coefficient, h_c, of fixed value $h_c = 6\ [W/m^2 K]$ was discussed in [5].

A main argument is that the heat transfer between a gravity current of cold air and the floor is certainly not a situation of "free convection", and we expect that rather high values for h_c typical for mixed or forced convection are more realistic.The heat transfer situation of cold air flowing over the floor, is similar to where a heater plume flows over the ceiling a case for which Lebrun and Ngendakumana [14] found 7 $[W/m^2 K]$. This constant value for h_c, is certainly an approximation which needs further justification in the future. One can imagine for example that for situations with a strong turbulent wind, h_c increases even more. For the present simplified ventilation model, however, it turns out to be satisfactorily, giving a consistent picture without contradictions. (Taking h_c lower in value for example, leads to C_2 larger than one in certain cases).

Solving for T_{in} Solving equations (4), (7), (8) and (9) self-consistently for T_{in}, the inside air temperature is obtained in a few iterations. The model is illustrated in Figure 2. Although it does not make a difference for the present case, the Bernoulli model is represented by a heat source (1) rather than by a non-linear conductance [16].

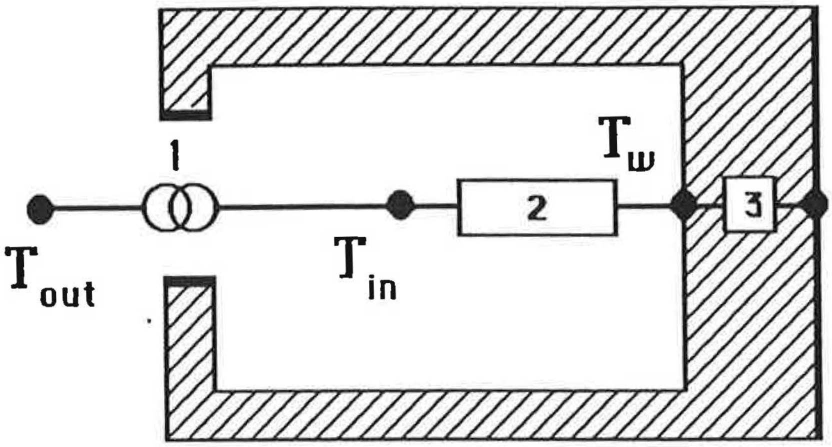

Fig. 2. The ventilation and thermal models are coupled in a three node network. The ventilation heat loss (Eq. 4) is represented as a heat current source Q_v: (1); the term $1/C_2\ Si\ h_c$ from Eq. 9 corresponds to the boundary layer resistance: (2); the dynamic thermal resistance of the wall, (3), follows from Eq. 7 and equals $R_{dyn}/C_2\ S_i$; T_b is the initial wall temperature T_{wall} (0).

Experiments

Experimental Details

Room Description The room of the LESO building we used for the first experiments (2.8 x 3.4 x 4.2m^3) is described in [5]. The room is characterized by concrete walls with an experimental value of $\sqrt{a}/\lambda \approx 1\mathrm{x}10^{-3}[\mathrm{Km}^2/(\mathrm{W}\sqrt{\mathrm{s}})]$, a wall surface area of 70m^2, an openable window, WxH=0.8x1.1 m^2 in size, and a wall surface area above the upper window level of 25m^2 (the theoretical value for the stratification coefficient is then C_2=0.65).

For the second experiment we choose a light-weight structure, the room in the Portakabin (2.1 x 2.6 x 6.5m^3) with 5 cm of polystyren insulation mounted on the inside. The room is characterized by walls with an experimental value of $\sqrt{a}/\lambda \approx 4\mathrm{x}10^{-3}[\mathrm{Km}^2/(\mathrm{W}\sqrt{\mathrm{s}})]$ (Figure 1), a wall surface area of 72 m^2, and a door opening WxH=0.7x2.0 m^2. Because the dooropening reaches close to the ceiling, the stratification coefficient C_2 is expected to be near unity.

Measurement Methods Used The air temperature and velocity were 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 [17] 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 the reading, 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. Velocity and temperature profiles were recorded by mounting probes on a motor-driven trolley.

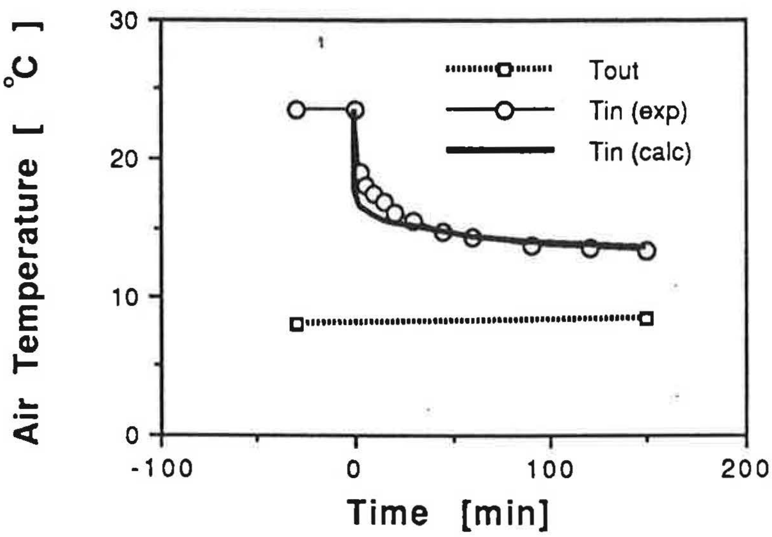

Fig. 3 . The air temperature measured in the middle of the room (Portakabin) as a function of time. At time t=0, the heating equipment is switched off and the door is opened. The symbols correspond to the air temperature measured in the middle of the room at a height of 1.2 m. The calculated curve corresponds to C_2=1, and the experimental value for the wall parameter (Fig.1) was used.

Experimental Test of the Ventilation Model

In order to test our model, we measured and calculated the air- (and wall-) temperatures of a room in the LESO building and in the Portakabin, as a function of time after turning off the heating system and opening the window and door respectively. The former experiment is described in Ref. [5], and the results on the latter are given in Figures 3-5.

In Figure 3, the time dependence of the observed and calculated drop in air-temperature are given, while the corresponding transient ventilation rate and heat loss are shown in Figures 4 and 5. During the experiments, the wind velocity was less than 0.5m/s and there was no sunshine.

The first half hour the measured air temperature in Figure 3 is systematically higher than the temperature predicted by the model. This is expected to be related in large part to the presence of office furniture whose influence is not taken into account by the model. Indeed its heat capacity will slow down any temperature change as noted before in Figure 1. After opening the door a vertical temperature gradient of about 2 °C/m established in the first minutes decreasing to less than 1°C/m at the end of the experiment. T_{in}-T_{out} was sufficiently large however for the isothermal zone approximation to be valid.

No correction was made for the drift in temperature which was observed when the heating system was turned off, but the door kept close. In both cases the corrections were estimated to be not larger than the influence of the furniture and not relevant for the present discussion.

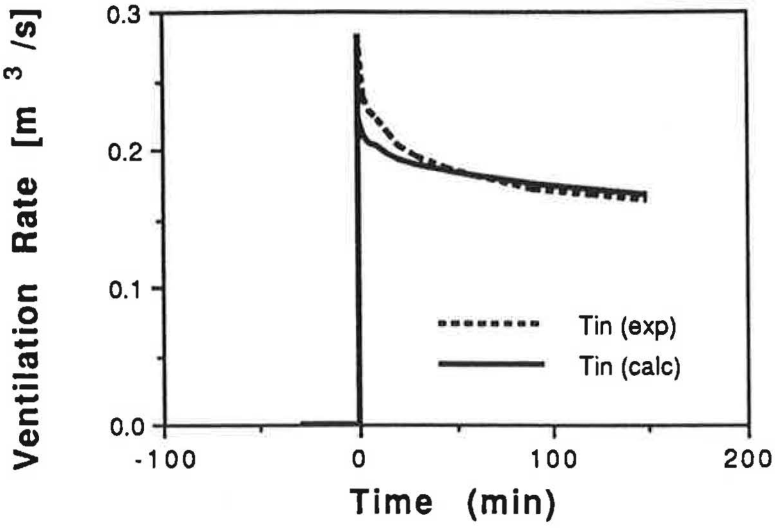

Fig. 4 The ventilation rate as a function of time after opening the door of the test room (Portakabin). The curves are calculated from the measured and calculated room air temperatures T_{in} as in Figure 3. The room volume is 36m^3 ·. Measurements of velocity profiles showed consistency with C_1 = 0.6.

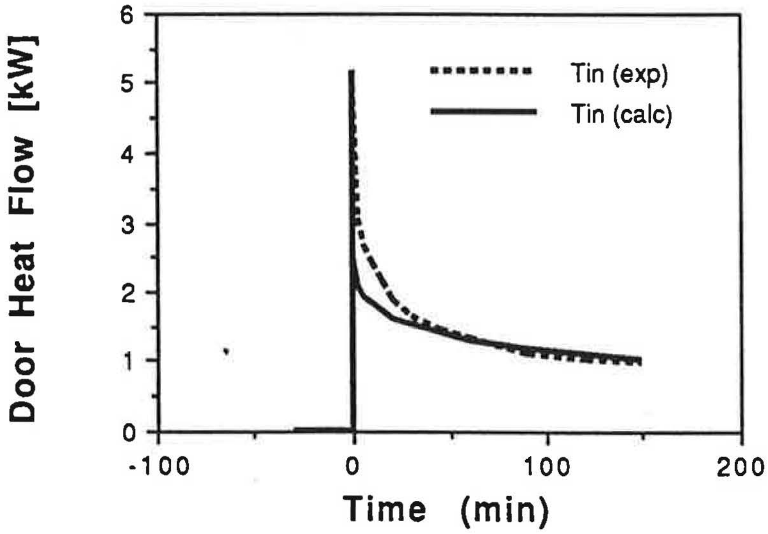

Fig. 5. The transient heat flow through the door as a function of time after opening the door of the test room (Portakabine) The curves are calculated from the measured and calculated room air temperatures T_{in} as in Figure 3.

Discussion

In Figures 3 4 and 5, we have given the room temperature drop, the ventilation rate and heat loss after opening a door for a particular testcase. It is interesting for example that this model allows an estimate to be made of the heat loss through a door which is left open (from Fig.5 the heat loss is dropping from 5kW to 50% of this value after 10 minutes, and decreases to 30% after 30 minutes). The agreement between the measured and calculated temperature drop shows that it is worthwhile to develop the model further. In particular it would be of great value if we could predict values of C_2 , or that we would know the influence of the wind. For the few cases we have tested the model seems surprisingly accurate, but further investigations are needed in order to understand the limits of the model.

In Figure 6 we compare our model with results from the Dutch 'investigation of the consequences of opening one window on the internal climate of a room' [6].

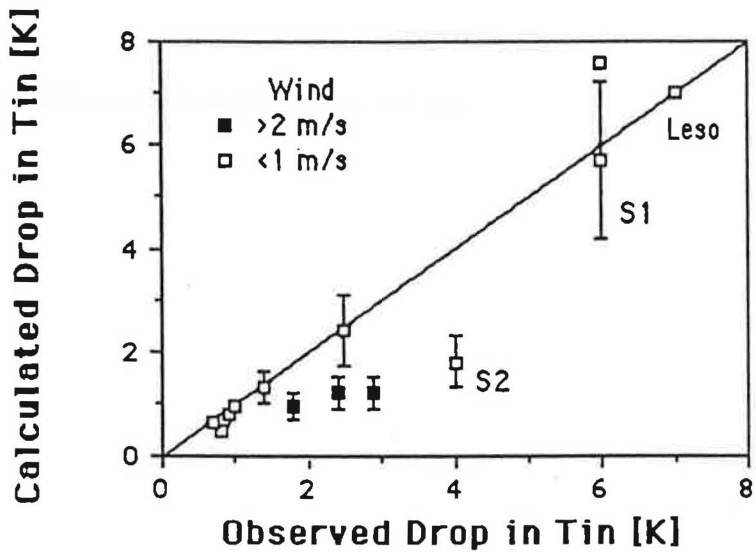

Fig. 6 . The temperature drop after the opening of a window reported by De Gids and Phaff [6] for various rooms and varying wind conditions, compared with the model prediction at t=10min. The intervals of uncertainty in the calculated results correspond to a variation of C_2 from 0.5 (strongest change in T_{in}) to 1. The data points labeled, S, refer to the pivoting window of the room in Schipluiden [6]. When completely open (S1), and when the opening is only 7cm wide (S2),.A few data are available with strong winds about 45° with the facade (flat in Delft), showing a larger decrease in T_{in} (filled symbols). For comparison we give the data point 'LESO' (C_2 =0.5 [5]), and the result of Figure 3 at t=10min (the uppermost data point in the plot).

The uncertainty intervals in the calculated values in Figure 6, are the uncertainties in the choice of the stratification coefficient; T_{in} has been calculated for C_2 =0.5 and 1 (smallest temperature drop) and the midpoint has been assigned the calculated value (Y-axis). The fact that the average value is close to the observed drop in T_{in} suggests that $0.5<C_2<1$ for a large number of data. Indeed stratification is observed in almost all cases reported in [6].

The large temperature drop in T_{in} for the filled data points correlates with high windvelocities (45° incidence with the facade) and relatively high air velocities measured in the window opening [6]. However many data are reported [6] which do not show this correlation and the situation with respect to the influence of wind velocity and turbulence on the ventilation rate is far from clear at present.

Surprisingly for a large fraction of the data, the drop in T_{in} can be explained exclusively by the inside-outside temperature difference, in spite of the fact that significant turbulence and wind velocities at the facade of up to 1m/s are reported [6]. The probable explanation is that by measuring the heat loss of the walls one effectively performs an integration of the heatflow over time, smoothing out all rapid variations.

In Figure 6 we have also given the data point 'LESO' from [5], which is on the straight line of perfect correlation because we had fitted C_2=0.5; a larger value of C_2 would bring the point up. It was also interesting to put the result of Figure 3 on the plot. We have done so for the arbitrary chosen time t=10min (the uppermost data point in the plot).

To conclude, we have seen that for rooms with very different thermal wall, properties, the model correctly predicts the magnitude of the to be expected drop in temperature, the transient ventilation and the heat loss rate after opening a door or window in a building.

Taking heat transfer with the walls into account, the predicted temperature drop gives a more realistic estimate of T_{in}, V and Q_v than existing models. For example De Gids [7] presented 'Basic material for the instruction of occupants of homes. How, when and where to use your windows', but based his estimates on the situation where the inside temperature stays indefinitely at 20°C, largely overestimating the energy loss when windows stay open for a long time with the heater turned off..

Further, the thermal part of the model allows an estimate of the indoor climat as a function of time. When a room is filled with people, and there is no air-conditioning for example, one can make a quick estimate whether the wall will keep the temperature comfortably low after a one hour meeting or not. This was also one of the objectives of ref [15].

CONCLUSION

It was shown that this new model describes quite well the transient air- and energy flows going through large vertical openings. It is simple and does not need a complete coupling between the air flow model and a thermal model of the building. It can replace the preceding large opening models, which were the gravity wave model with adiabatic walls and the isothermal, constant, air temperature model. In fact, the proposed model is more realistic and gives results which are inbetween the results of the preceding models.

As an example, the energy lost by a window opening varies strongly with the model used. For the gravity wave model, it becomes zero after one air change. The model assuming the room temperature to remain constant, the energy losses are constant and greatly over-estimated for small rooms for example. It was shown here, by comparison with several experiments, that the proposed model in spite of the simplifications made, predicts rather well the transient energy loss after opening a window or door, and the use of this model should significantly improve the realism of air infiltration and ventilation programs.

Acknowledgements

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NOMENCLATURE

a	= thermal diffusivity [m^2/s]
g	= gravitational acceleration [m/s^2]
h_c	= convective heat transfer coefficient [W/m^2K]
q	= heat flow density [W/m^2]
t	= time [s]
v	= velocity [m/s]
C_p	= specific heat [J/kg - K]
C_1	= discharge coefficient [-]
C_2	= coefficient of temperature stratification[-]
H	= height of aperture [m]
Q_H	= heat flow rate [W]
Q_v	= ventilation heat loss rate [W]
R_{dyn}	= dynamic thermal wall resistance (Eq.8) [m^2-K/W]
S_i	= total wall surface area [m^2]
T	= mean thermodynamic air temperature[K]
T_{in}	= inside air temperature [°C]
T_w	= wall surface temperature [°C]
T_{out}	= outside air temperature [°C]
V	= volume flow rate [m^3/s]
W	= door width [m]
λ	= thermal conductivity [W/m-K]
ρ	= mean air density [kg/m^3]
ΔT	= $T_{in} - T_{out}$ [°C]
ΔT_w	= $T_{w(t)} - T_{w(0)}$ [°C]

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