EXPERIMENTAL STUDY OF VENTILATION PERFORMANCE IN DWELLING-CELLS

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ABSTRACT

In order to assess ventilation systems, ventilation and thermal comfort parameters are calculated. Parameters are temperature and ventilation efficiency and PMV / PPD. Two ventilation configurations are set : the supply grille is under the ceiling and tests are performed for 2 exhaust positions. Both are opposite the ceiling : the first one is under the ceiling and the second one is on the floor. In regards with extract position, the ventilation system is better when extract is on the floor. It appears that the air renewal does not influence neither ventilation nor temperature efficiency. Concerning thermal comfort, generally, the thermal feeling remains the same and is good except when the air supply is low and cold. In this case, the jet reaches the occupied zone resulting in a slight feeling of cool.

KEYWORDS

Ventilation efficiency, Temperature efficiency, Comfort, Full-scale experiments, Tracer gas.

INTRODUCTION

Since reduction of ventilation rates in dwellings for economical reasons, it has been necessary to study whether this reduction had not been done to the detriment of indoor air quality. Our work, which is mainly experimental, makes part of a CNRS/ECODEV French Research Group « Ventilation and Air Quality in Dwellings ». It deals with the ability of a ventilation system to settle a suitable indoor air quality for occupants according to different thermal conditions.

METHODS

Ventilation and comfort parameters

To quantify air quality and thermal comfort, 3 parameters were calculated :

Temperature and ventilation efficiency (Sandberg 1981, Qingyan et al. 1988) for air quality : ε_T and ε_C. They express the ability to respectively remove heat and pollutants. They respectively depend on measurements of temperature (T) and concentration (C) taken at supply (s), extract (e) and on the mean value in the occupancy zone (oz). The temperature efficiency is :

$$\varepsilon_T = \frac{T_e - T_s}{T_{oz} - T_s} \qquad (1)$$

Regarding pollutant removal, in this work, the pollutant source is placed at the centre of the room; this thus implies that Cs = 0. Ventilation efficiency is therefore expressed as follows :

$$\varepsilon_c = \frac{C_e}{C_{oz}} \qquad (2)$$

PMV / PPD (Predicted Mean Vote / Predicted Percentage of Dissatisfied) are calculated to assess thermal comfort (ISO 7730, 1984): the PMV is an index which predicts the mean value of votes of a group expressing its thermal feeling according to a 7-point scale from 'cold' to 'hot'. It is determined with the estimations of clothing and subjects activity whereas the following environment factors are measured : mean radiant temperature, air temperature, relative air velocity (V) and vapour pressure. Individual votes are dispersed around mean values and it is particularly interesting to predict the percentage of subjects likely to be disturbed by heat or cold. Because of differences between persons, conditions in ISO 7730 Standard are specified in order to be agreed by 80 % of the occupants at least. All in all, this corresponds to a PPD lesser or equal to 10 % and to a PMV between –0.5 and +0.5. PPD and PMV are linked by the following expression :

$$PPD = 100 - 95 \exp[-03353\ PMV^4 + 0.2179\ PMV^2] \quad (3)$$

Experimental set up

The experimental set up is the full scale test-cell, "Minibat", see Figure 1. It is

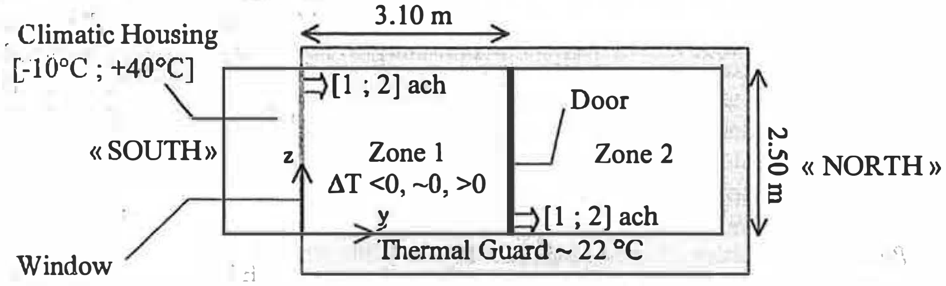

Figure 1 Experimental test-cell "Minibat".

made of 2 identical rooms (Zone 1 and Zone 2) which measure $3.1 \times 3.1 \times 2.5\ m^3$, each.

A thermal guard maintains an uniform temperature all around the cell thus replacing the presence of neighbouring dwellings. One side of the cell, called 'South' side, is made of glass and is in contact with a climatic housing whose temperature is set between –5 °C and +35 °C. The ventilation system has a fixed air supply (either mixing or displacement supply) and a mobile extract : air supply temperature is controlled between +5 °C and +35 °C whereas supply and extract air flows are set between 1 and 5 ach (air changes per hour).

2 × 9 K-type thermocouples distributed on the inner and on the outer surfaces of each of the 11 walls provide the thermal state of the envelope. At the centre of each zone, relative air hygrometry and operative temperature are measured. Besides, air flow and air temperature are taken both at supply and extract. A source pollutant continuously injects tracer gas (SF_6) at the centre of Zone 1.

A programmed system scans the vertical mid plane of each zone so that temperature and velocity of the air (mean value over 3 minutes) and SF_6 concentration are automatically measured. Temperature is measured with a K-type thermocouple, velocity is measured with a TSI hot film omnidirectional anemometer and concentration is measured through photoacoustic effect (Brüel & Kjaer).

The regular grid has an elementary mesh of $0.1 \times 0.1\ (m^2)$.

RESULTS

Experiments consist in setting a thermal configuration. When temperature and concentration steady states are obtained, 600 points of T, V and C are measured in the vertical mid-plane of the cell. Each experiment is validated through both SF_6 concentration and thermal balances.

2 positions of the air exhaust were set: position A and B (see Figure 2)

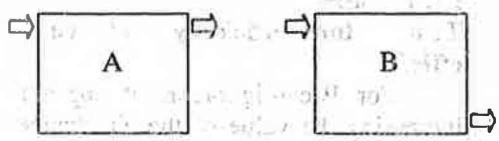

Figure 2 Tested positions of supply.

The different tested configurations are committed to Table 1.

Table 1 Tested configurations.

Extract	Test N°	ΔT (T_s-T_{oz}) (°C)	a.r. (ach)	Mean air flow velocity (m/s)	Ar	Re
A	1	13.5	1.0	1.4	0.0151	6983
	2	-10.6	1.0	1.3	0.0144	6919
B	1	10.6	2.0	2.7	0.0032	13566
	2	-10.7	1.0	1.4	0.0134	7205
	3	-12.4	2.0	2.7	0.0042	13999
	4	0.4	1.0	1.4	0.0005	7106
	5	-0.5	2.0	2.7	0.0002	13628

where a.r. = air renewal

Ar = Archimed's Number

$$Ar = \frac{g\beta|T_s - T_{oz}|d}{V^2_{supply}} \quad (4)$$

Re = Reynolds' Number

$$Re = \frac{\rho V_{supply}\, d}{\mu} \quad (5)$$

with β = thermal expansion coefficient (K^{-1})

g = gravitational acceleration (m/s^2)

S_{supply} = Supply area (= 0.005 m^2)

$d = \sqrt{S_{supply}}$ (m)

ρ = air density (kg/m^3)

μ = dynamic viscosity of air (Pa.s)

For each test, temperature and velocity of the air and SF_6 concentration were processed in order to display isovalue lines.

DISCUSSION

Temperature efficiency and ventilation efficiency

For B-configuration, it appears that increasing the value of the air change rate does not affect efficiencies. Whatever the case, these values are close to 1 (see Table 2). It means that the air movements are close to perfect mixing. Isotherms (cases 4 and 5) clearly display the homogeneity of temperatures.

Table 2 Experimental results.

Extract	Test N°	T_{oz} (°C)	σ_T^*	V_{oz} (m/s)	ε_T	ε_c	PPD_{oz}^{**}
A	1	20.0	0.2	0.01	0.8	0.6	-10.2
	2	21.7	0.2	0.08	1.1	0.9	-12.4
B	1	20.8	0.6	0.02	1.1	0.9	-8.7
	2	21.9	0.2	0.07	1.0	1.0	-11.5
	3	22.1	0.1	0.06	1.0	1.0	-8.3
	4	21.9	0.1	0.01	1.1	1.2	-6.2
	5	21.8	0.0	0.04	0.9	1.0	-6.8

* Standard deviation to mean value

** Minus in front of PPD values is used here to indicate that the predicted thermal sensation is below the neutral point (corresponding to negative PMV : Gan 1995).

In the other cases, one can see a thermal buoyancy effect. When supply is warm (test B1), the highest temperatures remain in the upper part of the cell (above 1.60 m) and thus do not affect the occupied zone. The strongest thermal buoyancy effect is obtained when air change rate is 1 ach and supply temperature is 11.2 °C (test B3). In this case, one may see (Figure 3) the jet

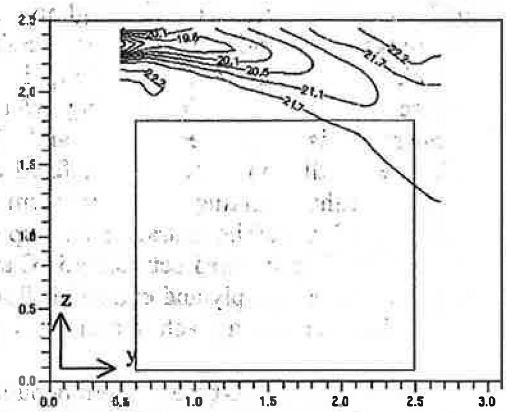

Figure 3 Isotherms (°C) for Test B2. a.r. = 1 ach, Ts = 11.2 °C (for every graphs, occupied zone is in dotted lines and axes unity is meter).

falling towards the occupied zone but actually it is quite quickly reheated by ambient air induction.

When extract is under the ceiling, and for the same supply conditions, isotherms have the same appearance. See Figure 4.

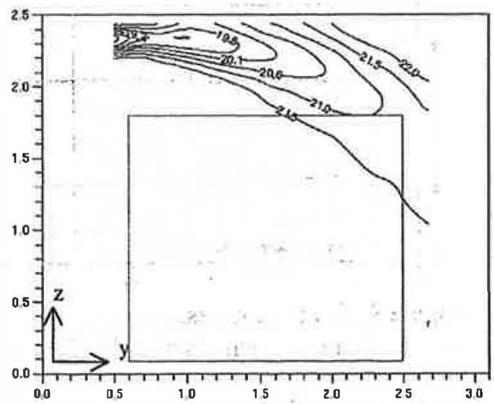

Figure 4 Isotherms for Test A2.
a.r. = 1 ach, Ts = 11.1 °C.

In this case, temperature efficiency is a little better than for configuration B2 : as the jet falls towards the occupied zone, the exhaust does not retrieve cold air much. Supply velocity (V_{supply} = 1.4 m/s) carries the jet over the occupied zone. See Figure 5.

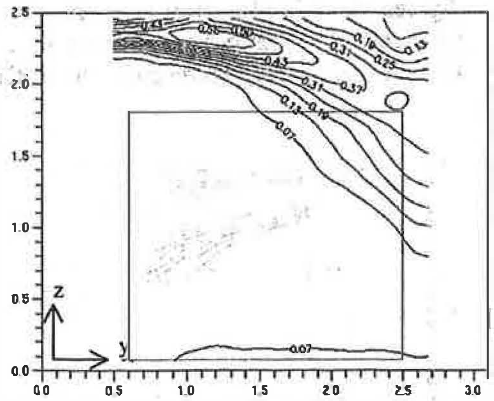

Figure 5 Iso-velocities (m/s) for Test B2.
a.r. = 1 ach, Ts = 11.2 °C.

When air supply velocity increases (test B3) the cold jet goes even further : it reaches the opposite wall and does not cool the occupied zone (see Figure 6).

In terms of ventilation efficiency, results for configuration B are of the same type as for temperature efficiency. As noticed before, the air movements resulting of the different thermal conditions are close to perfect mixing.

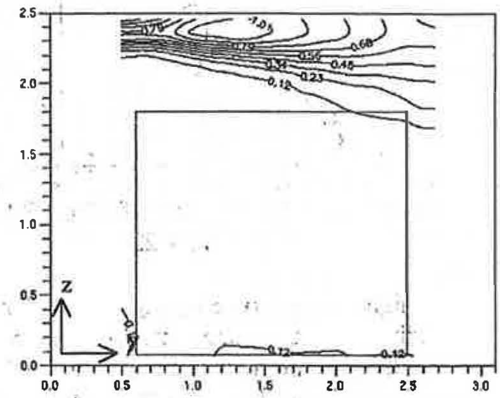

Figure 6 Iso-velocities (m/s) for Test B3.
a.r. = 2 ach, Ts = 9.7 °C.

When exhaust is under the ceiling, (Test A2) ventilation efficiency is a little worse than when it is on the floor. As the jet falls in the occupied zone, the pollutant is dragged towards occupied zone hence a ventilation efficiency of 0.9. This value is not that bad probably because the cold jet enhances a perfect mixing effect. As Nielsen (1993) mentioned, the return opening has only a very small influence on the velocity distribution in a room but may have a large influence on the ventilation efficiency.

The best result is obtained for Test B4. All in all, in this case, measurements show that concentrations in the lower part of the cell are relatively low, compared to exhaust concentration. In Test B4, values are even lower above 1.40 m. Iso-velocities show that the jet remains stuck to the ceiling; the shape of the isotherms confirms it. Because of the absence of air movements in the room (V_{oz} ~0 m/s for Test B4, see Figure 7), the pollutant spreads and naturally remains in the lower part of the room.

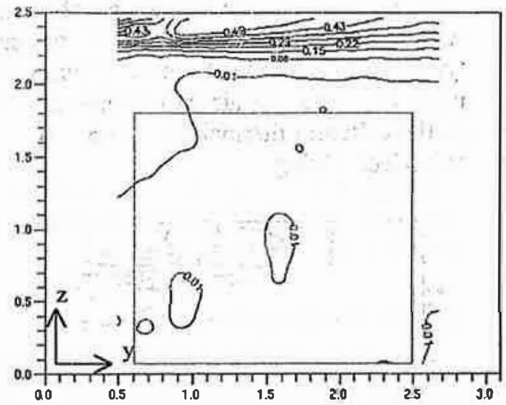

Figure 7 Iso-velocities (m/s) for Test B4.
a.r. = 1 ach, Ts = 22.3 °C.

The high density of SF_6 (d_{SF_6} = 6.3) may contribute to this phenomenon. Figure 8 displays concentration measurements taken during Test B4. In order to take exhaust concentration into account, it displays local ventilation efficiency iso-lines, iso-εc_p (Equation 6) :

$$\varepsilon_{c_p} = \frac{C_E}{C_p} \qquad (6)$$

where p is any point of the vertical mid plane of the cell.

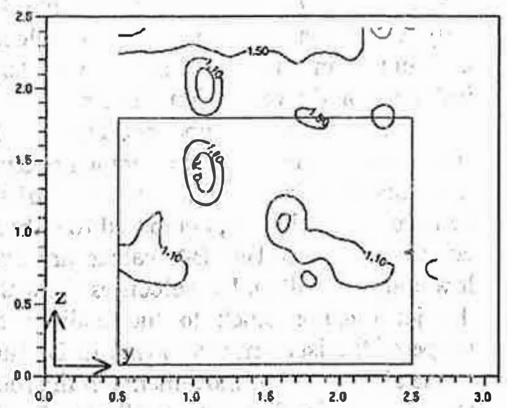

Figure 8 Iso-εc_p for Test B4.
a.r. = 1 ach, Ts = 22.3 °C.

The worst ventilation efficiency is obtained for Test A1. The combination of both thermal buoyancy and exhaust position is such that pollutants are enclosed in the lower part of the room. See Figure 9.

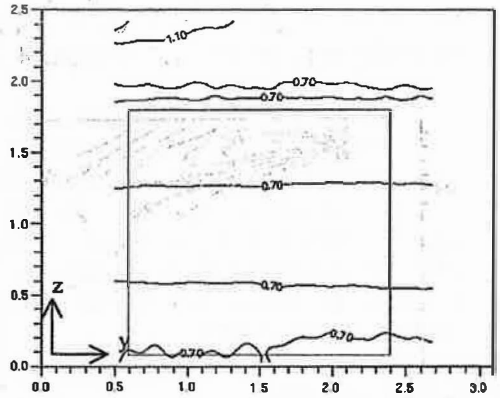

Figure 9 Iso εc_p for Test A1.
a.r. = 1 ach, Ts = 33.5 °C.

Thermal comfort

The global sensation is a slight feeling of cool whatever the supply condition for configuration B. Except Tests A2 and B2, the results satisfy ISO 7730's recommendations Actually, Tests A2 and B2 were the only ones which presented a supply jet drop. PPD isolines (Figure 10) display that discomfort follows both lines of velocities and temperatures (respectively Figures 3 and 5) : the occupied zone is little affected by low temperatures thus a PPD of 12.4 %, which only corresponds to a PMV of –0.6.

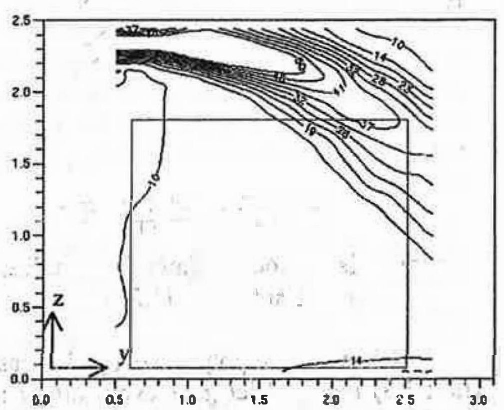

Figure 10 Iso-PPD for Test A2.
a.r. = 1 ach, Ts = 11.1 °C.

In the whole, results are quite close to each other, whatever the configuration. Except for Test A1, measurements seem to prove that air movements are close to perfect mixing (Laret et al. 1976). This is especially true as far as efficiencies are concerned.

As perspectives, this work will be pushed on with further experiments both on mixing and displacement ventilation. Then 2-Zone configurations will be tested according to the same protocol as for single zone. In the mean time, results will be compared to values issued from a zonal model (Inard et al. 1996).

REFERENCES

Gan, G. (1995) Evaluation of room air distribution systems using computational fluid dynamics. *Energy and Buildings*, 23, 83-93.

Inard, C., Bouia, H., Dalicieux, P. (1996) Prediction of air temperature distribution in buildings with a zonal model. *Energy and Buildings*, 24, 125-132.

ISO (1984) Moderate thermal environments – Determination of the PMV and PPD indices and specification of the conditions for thermal comfort. ISO 7730, International Standards Organisation, Geneva, Switzerland.

Laret, L., Lebrun, J., Marret, D., Nusgens, P. (1976) Etude expérimentale sur l'efficacité de la ventilation mécanique contrôlée dans un local d'habitation. *Extrait de la collection des publications de la faculté des sciences appliquées de l'université de Liège*, 62, 68-119.

Nielsen, P. V. (1993) Air Distribution systems – Room air movement and ventilation effectiveness. *Proceedings of 1992 International Symposium on Room Air Convection and Ventilation Effectiveness*, 22-24 July, 1992, Tokyo, Japan, 43-52. Edited by Murakami, S., Kaizuka, M., Yoshino, H. and Kato, S. ASHRAE.

Qingyan, C., Van der Kooi, J., Meyers, A. (1988) Measurements and Computations of Ventilation Efficiency and Temperature Efficiency in a Ventilated Room. *Energy and Buildings*, 12, 85-89.

Sandberg, M. (1981) What is Ventilation Efficiency? *Building and Environment*, 16 (2), 123-135.

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