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ANALYSIS OF HYBRID VENTILATION PERFORMANCE IN FRANCE

Florence Cron and Christian Inard
LEPTAB, Université de La Rochelle
Pôle Sciences et Technologie, Avenue Michel Crépeau
F-17042 La Rochelle Cedex 01, France
Emails: fcron@univ-lr.fr and cinard@univ-lr.fr

ABSTRACT

This study was part of the International Energy Agency Annex 35 “Hybrid Ventilation in New and Retrofitted Office Buildings”. It consisted in modeling a typical classroom and in predicting performance of a hybrid ventilation system compared to two traditional mechanical systems: a mechanical exhaust ventilation system and a balanced ventilation system. The hybrid system considered here was a fan assisted natural ventilation with a temperature and CO_2 based control strategy. The study of this specific hybrid ventilation system was performed for several cities in France and predicted its potential in terms of thermal comfort, CO_2 concentration and energy consumption given the climate in France.

INTRODUCTION

Hybrid ventilation has lately aroused interests in the design of office buildings. The State of the Art (Annex 35, 2000) and the booklet (Heiselberg, 2002) of the International Energy Agency (IEA) Annex 35 “Hybrid Ventilation in New and Retrofitted Office Buildings” are about hybrid ventilation principles, design, control strategies, analysis methods and existing building examples.

Many tools can be used to do a ventilation performance analysis (Delsante and Vik, 2000 and Fracatoro and Perino, 2002) and their consistency and robustness have been analyzed by Delsante et al. (2002b). Delsante at al. (2002a) and Cron et al. (2000b) have compared a specific hybrid ventilation system to more traditional ones but these studies were limited to a few cities with different climates.

The potential and performance of a hybrid ventilation system depend on the climate. For instance in Canada, climate is more a barrier to the implementation of a hybrid ventilation system (Bourgeois et al., 2002). Thus the objective of this study is to perform simulations for several cities in France and to analyze the potential of a specific hybrid ventilation system given the climate characteristics. The room to be simulated is first presented, followed by a description of the tools and the models used for the numerical study. Then results are given in terms of indoor air and operative temperatures, CO_2 concentration and energy consumption. Finally, the potential of this specific hybrid system is outlined given the climate and the control strategy chosen for these simulations.

CASE STUDY DESCRIPTION

Classroom description

The room to simulate was assumed to be a single classroom at the second level of a three-storey building and surrounded by other classroom at the same conditions. The room was 9 m wide, 6 m deep and 3 m high. The external facade was oriented south and consisted of 150 mm of concrete and 80 mm of mineral wool indoors. It was assumed to have a 1 m double pane window with a U-value of 2.7 $W/(m^2/K)$ along the whole facade. Two ventilation inlet grilles were placed on the facade, 0.5 m above the classroom floor. The outlet grille was at the opposite, on the ceiling. Meteorological data were taken for outdoor boundary conditions except CO_2 that was considered constant and equal to 400 ppm. Solar shading was provided whenever the total incident solar radiation was higher than 200 W/m^2.

A teacher and 24 pupils were assumed to be in the room from 8:00 to 12:00 and from 13:00 to 15:00 on Mondays to Fridays. On Tuesdays, from 10:00 to 11:00, only half of the children were in the classroom and everybody was out at 14:00. There was no school on Wednesday afternoons either. The internal heat gains were 10 W/m^2 for lighting and 80 W per person. CO_2 production was 18 l/(h.person).

Heating hours were from 7:00 to 15:00 from Monday to Friday. During these hours, the set point temperature was 21 °C, whereas it was reduced to 18 °C during the non-heating hours. A preheating system was assumed to preheat the ventilation supply air to 18 °C.

Ventilation systems

Infiltration was assumed to be equal to 0.2 ach all the time and independently of the ventilation system. Window airing occurred when both the indoor and outdoor air temperatures were higher than 23 °C and 12 °C respectively. The window closed when the indoor air temperature went back down to 21 °C. Window airing brought 4 ach independently of the weather conditions. This value was additional to the infiltration and the ventilation airflow rates. During spring and summer, night cooling could occur between 22:00 and 7:00 when the indoor air temperature was higher than 24 °C and when the indoor-outdoor air temperature difference was higher than 2 °C. It turned off when the indoor air temperature went below 18 °C.

Ventilation hours were scheduled from 8:00 to 15:00 from Monday to Friday. Two mechanical ventilation systems, with a constant airflow rate during ventilation hours, and a hybrid ventilation system were compared.

The first mechanical ventilation system was a mechanical exhaust in which a constant airflow rate of 0.15 m^3/s was assumed through the inlet and outlet grilles (MV1). The low exhaust fan power consumption was 1000 W/(m3/s).

The second one (MV2) was a balanced system with an exhaust and a supply fan to ensure an airflow rate of 0.15 m^3/s. The heat recovery had a temperature efficiency of 0.6 and the fan combined consumption was 2500 $W/(m^3/s)$.

The fan airflow rate for both mechanical systems was set to have a maximum indoor CO_2 concentration of 1200 ppm when the pupils were in the classroom.

The specific hybrid ventilation system considered here (HV) was based on fan assisted natural ventilation with a 4 m high exhaust chimney to improve the natural stack effect. No conductive heat transfer through the chimney walls was considered. The inlet and outlet opening control was not only dependent on the ventilation hours, but also on the indoor CO_2 concentration. The first inlet grille opened when CO_2 reached 800 ppm, the second one when the CO_2 was higher than 1000 ppm and finally the fan turned on when the CO_2 went up to 1200 ppm. The fan stopped and the grilles closed in the opposite order at a 100 ppm lower set point. The assisting fan was assumed to have a consumption of 200 $W/(m^3/s)$ and to provide 0.15 m^3/s. *Cp* values for the wind effect on the facade were taken from the AIVC literature (Liddament, 1996) and are given in Table 1. For the simulations, an interpolation was made from these values. The *Cp* value for the chimney was equal to -0.6 independently of the wind direction. For the three systems, night cooling was ensured by the fans that were turned on.

Table 1
Cp values on the facade given the wind direction

WIND ANGLE	0°	45°	90°	135°	180°
Cp value	0.25	0.06	-0.35	-0.60	0.50

Locations

The simulations were performed for ten French cities listed in the Table 2.

Table 2
City geographical locations

CITIES IN FRANCE	LATITUDE °	LONGITUDE °
Agen	44.2 N	0.6 E
Ajaccio	41.9 N	8.7 E
Carpentras	44.0 N	5.0 E
La Rochelle	46.2 N	1.2 W
Limoges	45.8 N	1.3 E
Mâcon	46.3 N	4.8 E
Nancy	48.7 N	6.2 E
Nice	43.7 N	7.3 E
Rennes	48.1 N	1.7 W
Trappes	48.8 N	2.0 E

France is divided into three zones in winter, from the coldest H1 to the coolest H3, and four zones in summer, from the coolest Ea to the warmest Ed, see Figure 1 and Figure 2.

Figure 1 Winter climatic zones in France

Figure 2 Summer climatic zones in France

In France the regulation heating period is from the 1st of October until the 20th of May. Heating degree-days give some information on the climate and were calculated during this heating period based on the indoor set point temperature of 21 °C (see Figure 3). Cooling degree-day were obtained by computing all the degree-days for an outdoor temperature higher than 26 °C when air conditioning may be required.

Degree-days between 21 °C and 26 °C were also calculated. The last two degree-day values were taken during the non-heating period, see Figure 4.

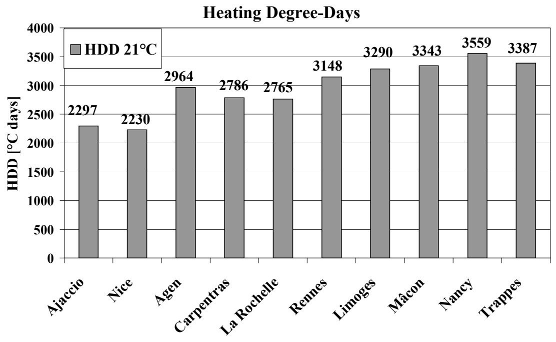

Figure 3 21 °C based heating-degree days during the heating period

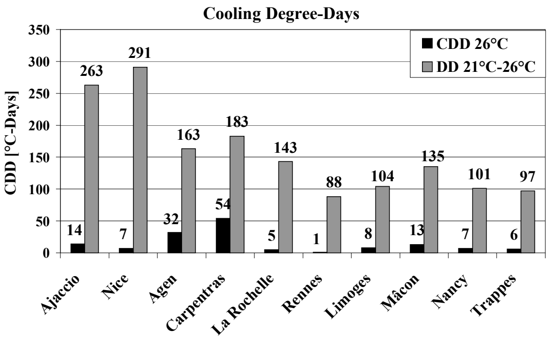

Figure 4 26 °C based cooling degree-days and degree-days between 21 °C and 26 °C during the non-heating period

SIMULATION

The simulations were undertaken to compute the indoor air and operative temperatures, the indoor air quality in terms of CO_2 concentration and the energy performance. SPARK is the object-oriented solver we used to solve the entire problem with differential and non-linear equations. The model equations we used here were implemented in SPARK for a previous IEA Annex 35 project (Cron and Inard, 2002a). The time step was 60 s. An experimental cell presented by El Mankibi et al. (2001) allowed to do comparisons between simulation results and experimental data and to calibrate the model.

Air flow and thermal models

A single zone model with a hydrostatic pressure variation was assumed. "Pure" air and pollutant mass balance equations for a zone *i* with *n* openings were:

$$\sum_{j=1}^{n} m_{as\,ji} - \sum_{j=1}^{n} m_{as\,ij} = 0 \tag{1}$$

$$\sum_{j=1}^{n} m_{es\,ji} - \sum_{j=1}^{n} m_{es\,ij} + S_{es} = V\frac{d\rho_{es}}{dt} \tag{2}$$

Airflow rates derived from infiltration, window airings, and both mechanical systems were constant. Concerning the specific hybrid ventilation system, a simple power law relation was used for the inlet grilles and the chimney when the fan was turned off:

$$Q_{ji} = C(\Delta P)^{0.5} = C\left(P_j - P_i\right)^{0.5} \tag{3}$$

C is equal to 0.053 $m^3/(Pa^{0.5}.s)$ for each grille and to 0.088 $m^3/(Pa^{0.5}.s)$ for the chimney.

Thermal buoyancy and a correction factor K_Q (Feustel et al, 1990) were used to obtain the mass air flow rate:

$$m_{as\,ji} = \rho_{ij} K_Q Q_{ji} = \rho_{ij} K_Q C\left(P_j - P_i\right)^{0.5} \tag{4}$$

where $\rho_{ij} = \dfrac{\rho_i + \rho_j}{2}$.

The thermal balance equation was:

$$\begin{aligned}&\sum_{j=1}^{n}\left(m_{as\,ji}cp_{as} + m_{es\,ji}cp_{es}\right)T_j - \sum_{j=1}^{n}\left(m_{asij}cp_{as} + m_{esij}cp_{es}\right)T \\ &+ S_{es}cp_{es}\left(T_{es} - T\right) + P_{heatconv} + P_{loadconv} + \Phi_{conv} \\ &= \left(\rho_{as}cp_{as} + \rho_{es}cp_{es}\right)V\frac{dT}{dt} + cp_{es}VT\frac{d\rho_{es}}{dt}\end{aligned} \tag{5}$$

Room envelope Models

Conduction heat transfer was described by an electrical 2R-3C model (Rumianowsky et al., 1989), see Figure 5.

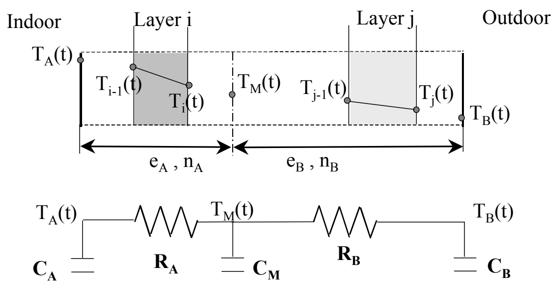

Figure 5 Electrical 2R - 3C model

The intermediate node M allows us to take into account differences in material thermal capacities. A comparison between this model response and a finite difference model response was undertaken to define the position of the node M. The latter was set to the optimum value of e_A that gave the minimum error E_{eA}. E_{eA} was obtained by:

$$E_{eA} = \int_0^{\infty}\left|T_{A\,finite\,difference} - T_A\right|dt \tag{6}$$

The outdoor long-wave radiation was obtained by:

$$\Phi_{netGLO\,o} = \varepsilon_o \sigma S_o \left(\frac{(1-\cos\beta)}{2} \left(T_o^4 - T_{so}^4\right)\right) + \varepsilon_o \sigma S_o \left(\frac{(1+\cos\beta)}{2} \left(T_v^4 - T_{so}^4\right)\right) \quad (7)$$

The short-wave radiation absorbed by the external wall surface was calculated from the incident solar radiation and the surface solar properties.

The outdoor convective heat transfer was calculated by:

$$\Phi_{conv\,o} = h_{conv\,o} S_o \left(T_o - T_{so}\right) \quad (8)$$

$$h_{conv\,o} = c + d U_{met}^n \quad (9)$$

where $c = 2.5$, $d = 3.5$ and $n = 1$ (Ferries, 1980).

A mean radiative temperature model described the long-wave heat transfer indoors. Part of the internal loads and the heating system were taken into account in the model. Given a surface i, we used:

$$\Phi_{netGLO\,i} = h_{rm\,i} S_i \left(T_{rm} - T_{si}\right) \quad (10)$$

$$h_{rm\,i} = 4\sigma\varepsilon_i \left(\frac{T_{si} + T_{rm}}{2}\right)^3 \quad (11)$$

$$T_{rm} = \frac{\sum_{i=1}^{n}\left(h_{rm\,i} S_i T_{si}\right) + P_{heat\,rad} + P_{load\,rad}}{\sum_{i=1}^{n} h_{rm\,i} S_i} \quad (12)$$

Indoors, solar radiation transmitted through the window was assumed to be entirely incident on the floor. One part was absorbed, the other part reflected in a diffuse way. The global diffuse solar radiation was distributed over all the surfaces depending on their area ratio and then absorbed.

Convection indoors was obtained by:

$$\Phi_{conv\,i} = h_{conv\,i} S_i \left(T - T_{si}\right) \quad (13)$$

where $h_{conv\,i} = a\left|T - T_{si}\right|^b$ (14)

a and b were taken from Allard (1987):

- $a = 3$ and $b = 0$ for an horizontal surface
- $a = 1.5$ and $b = 0.33$ for a vertical surface

The operative temperature is a simplified indication of thermal comfort and was computed by:

$$T_{op} = \frac{h_{conv}\,T + h_{rm}\,T_{rm}}{h_{conv} + h_{rm}} \quad (15)$$

$$h_{rm} = 4\sigma\varepsilon \left(\frac{T_{op} + T_{rm}}{2}\right)^3 \quad (16)$$

$$h_{conv} = a\left|\frac{T - T_{op}}{D}\right|^b \quad (17)$$

h_{conv} is the globe convection heat transfer, $a = 1.4$ and $b = 0.25$. D is the globe diameter equal to 0.15 m.

DISCUSSION AND RESULT ANALYSIS

Simulations were performed for each of the ten cities over a whole year and results were given in terms of indoor air and operative temperatures, CO_2 concentration and energy consumption. For every simulation, the two mechanical system results were identical except for the preheating and the ventilation consumption.

Climatic differences

As presented in Ghiaus and Allard (2002), two indoor air temperatures for a free-running building (balance temperatures) were calculated for the mechanical systems to outline the room heating, cooling and air-conditioning needs, see Figure 6. Both balance temperatures with and without window airing were obtained with the steady state assumption.

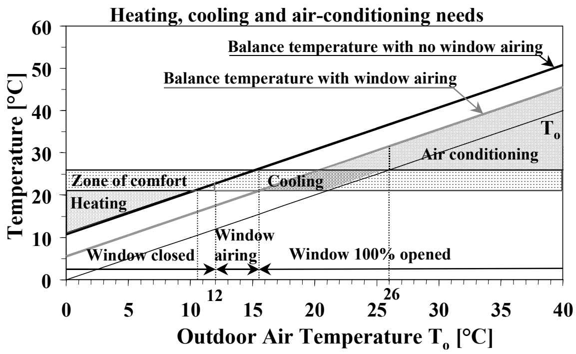

Figure 6 Heating, cooling and air-conditioning needs during occupancy

Figure 6 shows heating is required for an outdoor air temperature lower than 10.5°C. When the outdoor air temperature is between 12 °C and 16 °C, the window can be either opened or closed, it depends on the internal and solar gains. For an outdoor air temperature higher than 16 °C, the window is opened constantly during occupancy. This window airing can provide some free cooling, but thermal comfort may be a concern since solar radiation hasn't been taken into account to define the balance temperature. For hybrid ventilation, it is also critical since, within this range of temperature, natural stack effect is relatively low. For an outdoor air temperature higher than 26°C, air-conditioning is required to ensure a satisfying thermal comfort. The annual outdoor air temperature distribution during occupancy seemed to be another important information to analyze the needs depending on the outdoor air temperature. This annual distribution was calculated from the weather data files, see Table 3.

Table 3
Annual distribution of outdoor air temperature during occupancy (percentage)

CITIES	12°C> T_O	12°C≤T_O <16°C	16°C≤T_O <26°C	26°C≤ T_O
Ajaccio	17	34	46	3
Nice	25	30	45	0
Agen	50	19	27	4
Carpentras	44	17	33	6
La Rochelle	51	23	26	0
Rennes	59	22	19	0
Limoges	60	16	24	0
Mâcon	59	14	26	1
Nancy	62	14	23	1
Trappes	65	14	21	0

This table shows different categories of cities.

Ajaccio and Nice are the warmest ones: they have an annual outdoor air temperature higher than 16 °C during more than 45 % of the occupancy time and the outdoor air temperature is lower than 12 °C during less than 25 % of the occupancy. These two cities have also the lowest heating degree-days and the highest degree-days between 21 °C and 26 °C.

Agen and Carpentras have outdoor air temperatures higher than 16°C less frequently than Ajaccio and Nice but the distribution is in fact different. Indeed they have more 26 °C based cooling degree-days and fewer degree-days between 21 °C and 26 °C.

The definition of climatic zones and degree-days aren't enough to describe the climate of an area. For instance, Agen and La Rochelle are in the same zones (H2 and Ec), but La Rochelle is cooler since the outside air temperature is never above 26 °C. Although La Rochelle and Carpentras have about the same heating degree-days, the cooling degree-days and the annual distribution are different. In fact, Rennes and La Rochelle are located on the Western coast of France and both have a mild climate because of the oceanic influence. Table 3 shows the frequency of outdoor air temperature between 12 °C and 16 °C and the one of temperature between 16 °C and 26 °C are about the same for both cities. Rennes has somehow a cooler summer since the cooling degree-days and those between 21 °C and 26°C are lower than La Rochelle ones.

Finally Limoges, Macôn, Nancy and Trappes have an outdoor air temperature lower than 12 °C during around 60 % of the occupancy time, and also the highest 21 °C based heating degree-days.

Thermal comfort

As mentioned before, the two mechanical systems differ only in consumption. Figure 7 shows the annual operative temperatures during occupancy.

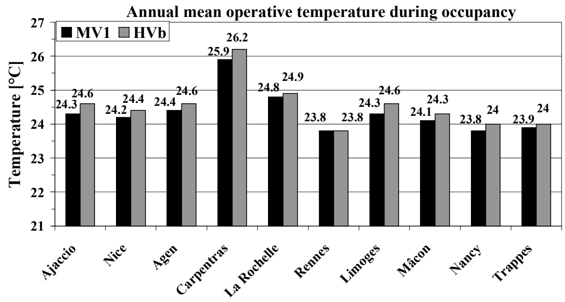

Figure 7 Annual mean operative resultant temperature during occupancy

For each city, the operative temperature is a little higher with the hybrid ventilation system than with the mechanical ones. In fact, during winter time the natural driving forces induce a higher air flow rate than the one provided by the fan, but during the intermediate seasons and the warmer months with occupancy, air flow rates for the hybrid ventilation system are lower than for the mechanical one. This is first due to a lower stack effect when the outdoor temperature is cool or warm. Then, when there is some window airing, the fan is still on for the mechanical modes, whereas for the hybrid ventilation, since the CO_2 level is reasonable, only one inlet grille is opened. So these combined factors explain the lower airflow rate and thus the higher operative temperature for the hybrid ventilation. For these reasons, temperature differences are a little higher for cities with a warm climate. For instance Carpentras in the zone Ed has the highest operative temperature and the highest temperature difference between the two modes, that is in correlation with the highest cooling degree-day value. On the other hand, Rennes, that is in the coolest zone Ea, has the lowest mean operative temperature and almost no difference between the mechanical and the hybrid ventilation temperature mean values. Rennes has in fact the lowest cooling degree-days and degree-days between 21 °C and 26 °C.

CO_2 concentration

The annual mean CO_2 concentration results show a difference between the warm climates and the mild or the cold ones. The value of the mean CO_2 concentration is higher with the hybrid system for Mediterranean cities, but remains below 1000 ppm. For all the other cities, CO_2 is in fact higher with the mechanical ventilation systems (the difference for Mâcon is too little to give a conclusion). As shown previously, the hybrid ventilation system provides a lower air flow rate than the mechanical systems when the window is opened, so there is a higher mean CO_2

concentration in that case than for the mechanical ventilation. In fact, two different ventilation phenomena occur. For Mediterranean cities, stack effect is not very important, thus ventilation over the year is mostly due to window airing, whereas for colder cities, stack effect prevails over window airing. For Mâcon, these effects are almost equal.

All climatic parameters are important and have to be taken into account to estimate hybrid ventilation system performance in terms of CO_2 concentration. Mâcon and Rennes both have an outdoor temperature lower than 12 °C during 59 % of the occupancy time, but they are respectively in the summer zones Ec and Ea which explains the differences in Figure 8. Likewise La Rochelle and Agen have a similar frequency of outdoor temperature lower than 12 °C and are both in the climatic zones H2 and Ec, but the mean CO_2 concentration is different. In fact, according to the degree-days, La Rochelle is cooler in summer and warmer in winter than Agen.

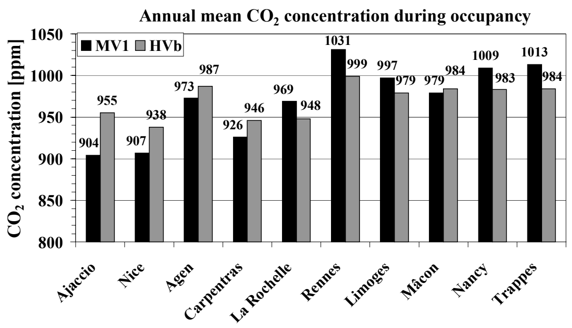

Figure 8 Annual mean CO_2 concentration during occupancy

Finally there is, for all cities, less exposure to a high CO_2 concentration with the hybrid system (see Figure 9), so this hybrid ventilation system provides a better indoor air quality than the mechanical ones. Indeed there can be, with the mechanical ventilation, an exposure to a concentration higher than 1000 ppm when the window is still closed (in the early morning for instance) and a lower mean concentration value as long as the window is opened.

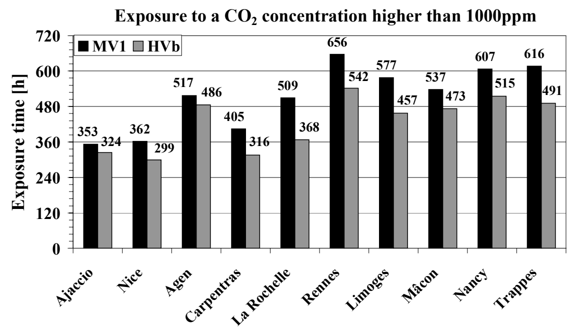

Figure 9 Exposure to a CO_2 level higher than 1000 ppm

The major conclusion is that, when applying this specific control strategy, indoor air quality in terms of CO_2 concentration is, independently of the climate, better with hybrid ventilation than with mechanical ventilation.

Energy consumption

Hybrid ventilation provides some energy savings on heating and preheating for all cities (see Table 4), mostly because the inlets are closed during non-occupancy. There are fewer energy savings for colder cities than for warmer ones, since a higher stack effect leads to a higher air flow rate and thus to more preheating and heating consumption. For Ajaccio and Nice, hybrid ventilation induces fewer energy savings for the fan consumption than for the other cities. This is due to more night cooling requirements.

Table 4
Annual energy savings between HV compared to MV1 (percentage)

ENERGY SAVINGS	HEATING HV/MV1	PREHEAT. HV/MV1	FAN HV/MV1
Ajaccio	27	27	92
Nice	21	16	92
Agen	11	14	94
Carpentras	24	12	93
La Rochelle	16	7	96
Rennes	11	7	98
Limoges	7	7	97
Mâcon	6	11	95
Nancy	5	8	97
Trappes	6	6	98

Table 5 corroborates these conclusions. Ajaccio and Nice have actually less preheating and more fan energy consumption than the other cities, thus high energy savings are done for Ajaccio and Nice. This is due to the low fan power consumption in the hybrid mode and fewer heating and preheating needs. For the other cities, hybrid ventilation provides energy savings for heating, preheating and the fan consumption compared to the exhaust mechanical ventilation mode (MV1) but energy consumption is higher with hybrid ventilation than with the balance mechanical system (MV2). This is due a high preheating need, especially during the cold days, so preheating consumption prevails over the fan consumption for these cities.

All these characteristics are also represented in the Figure 10 where we have the annual total energy consumption for all the ventilation modes.

Table 5
Fan consumption over preheating consumption for MV1. Comparison of consumption between hybrid and mechanical ventilation modes (percentage)

CITIES	FAN / PREHEAT. (MV1)	HV/MV1	HV/MV2
Ajaccio	41	-42	-28
Nice	36	-33	-14
Agen	17	-23	+31
Carpentras	24	-27	+27
La Rochelle	16	-19	+65
Rennes	12	-16	+58
Limoges	12	-15	+61
Mâcon	13	-16	+33
Nancy	11	-13	+44
Trappes	11	-13	+48

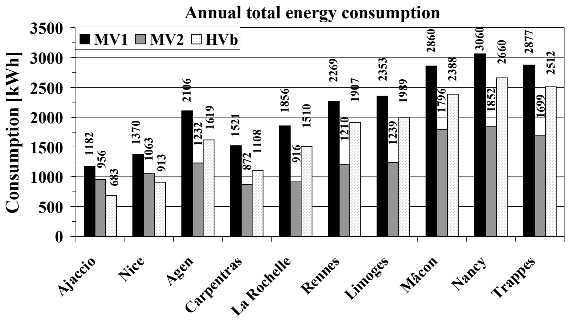

Figure 10 Annual total energy consumption for the different ventilation modes

Finally cities that require less heating and preheating have a better performance with the hybrid system, whereas the ones that have high heating degree-day values have the highest energy savings with the heat recovery system (see Figure 10). These results may be generalised to other buildings, if the yearly temperature range distribution is known. The relation between climate specifications and energy savings over a year can be finally summarise in Table 6

Table 6
Hybrid ventilation energy savings estimated from the climatic conditions

HEATING NEEDS	VENTILA. NEEDS	WINDOW AIRING NEEDS	ENERGY SAVINGS
★★	★★★	★★★★★	☺☺☺☺
★★★★★	★★	★★★	☺☺☺
★★★★★★	★★	★★	☺☺
★★★★★★★	★	★★	☺
★ : ca. 10 % of the occupancy time ($T_o < 12$ °C) ☺ : ca. 10 % of energy savings compared to a mechanical ventilation mode like MV1			

CONCLUSIONS

This study has shown the general behaviour of a fan assisted natural ventilation with control based on temperature and CO_2 concentration. This control provides air flow when needed for indoor air quality considerations and, although the mean CO_2 concentration is a little higher with the hybrid system for warn cities, it remains reasonable and the overall indoor air quality is improved with less exposure to a high CO_2 concentration. A drawback may be that the operative temperature is a little higher and may reach the thermal comfort upper limits, especially in warm cities. The exception to this is Mediterranean cities where the hybrid ventilation system outperforms both mechanical systems.

Heating and cooling degree-days give an initial idea about climate characteristics and the impact on indoor conditions and consumption. But degree-day data alone are not sufficient to estimate hybrid system performance. Hence more information such as the annual outdoor temperature distribution is needed to complete the analysis since it gives better indications on overall room needs. This kind of study is interesting and can be useful for designers. Designers have to define their priorities that are either to improve the indoor air quality, or to reduce energy consumption to as low as possible.

This study was limited to one specific hybrid ventilation system. For future work, it would be very interesting to perform simulations for other control strategies or other hybrid ventilation systems. Another interesting future work would be to extend these simulations to other specific climates such as humid, very cold or very warm climates. Then an atlas of potential and performance of several hybrid ventilation systems could be developed.

ACKNOWLEDGEMENTS

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NOMENCLATURE

C	thermal capacity	[J/°C]
cp	specific heat	[J/(kg°C)]
Cp	pressure coefficient	[-]
e	thickness of a layer	[m]
h	heat transfer coefficient	[W/(m^2°C)]
m	mass flow rate	[kg/s]
n	number of layers	[-]
P	internal gains (heating system or internal loads)	[W]
Q	air flow rate	[m^3/s]

R	thermal resistance	[°C/W]
S	surface	[m^2]
S_{es}	internal pollutant mass production	[kg/s]
t	time	[s]
T	temperature	[°C] or [K]
U	wind speed	[m/s]
V	volume of the zone	[m^3]

Greek Symbols

β	angle between the wall and the horizontal plane	[°]
ΔP	pressure difference	[Pa]
ε	radiative emissivity	[-]
λ	material conductivity	[W/(m.°C)]
Φ	heat transfer at the surface of a wall	[W]
ρ	specie density	[kg/m^3]
σ	Stephan-Boltzmann constant	[$W/(m^2.K^4)$]

Subscripts

A	node A
as	pure air
B	node B
conv	convective
es	pollutant
GLO	long-wave
heat	heating system
i	indoors or zone *i*
ij	from the zone *i* to the zone *j*
j	zone *j*
load	internal loads
M	node M
met	meteorological
net	net radiative heat transfer
o	outdoors
op	operative
rad	radiative
rm	mean radiative
s	surface
v	sky vault

REFERENCES

Allard, F. 1987. Contribution à l'étude des transferts de chaleur dans les cavités thermiquement entraînées à grand nombre de Rayleigh, Thèse d'Etat, INSA de Lyon.

Annex 35 2000. CD with the State of the art of Hybrid Ventilation, IEA Annex 35.

Bourgeois, D., Haghighat, F. and Potvin A. 2002. On the applicability of hybrid ventilation in Canadian office and educational buildings: Part 2 – Implementing Annex 35 pilots study projects in Canada, Proceedings of the 4th International Forum on Hybrid Ventilation, Montreal, Canada, 14-15 May 2002, pp24-29.

Cron, F. and Inard, C. 2002a. Hybrid Ventilation Simulation of Control Strategies – WG A1 input to first parameter study simulation in WG B7, Technical report F7-081, IEA Annex 35.

Cron F., El Mankibi, M., Inard, C. and Michel, P. 2000b. Experimental and numerical analysis of a hybrid ventilated room, Proceedings of the 8th International Conference on Air Distribution in Rooms, Copenhagen, Denmark, 8-11 September 2002, pp493-496.

Delsante, A.E. and Vik, T.A. 2000. State-of-the-Art Report on Hybrid Ventilation, IEA Annex 35 report.

Delsante, A.E., Aggerholm, S., Citterio, M., Cron, F., El Mankibi, M. 2002a. The Use of simulation tools to evaluate ventilation systems and control strategies, Proceedings of the 4th International Forum on Hybrid Ventilation, Montreal, Canada, 14-15 May 2002, pp103-110.

Delsante, A.E., Aggerholm, S. 2002b. The use of simulation tools to evaluate hybrid ventilation control strategies, IEA Annex 35 Technical Report.

El Mankibi, M., Michel, P. and Guarracino, G. 2001. Control Strategies for hybrid ventilation development of an experimental device, Proceedings of the 22nd Annual AIVC Conference, Bath, UK, 11-14 September 2001.

Ferries, B. 1980. Contribution à l'étude des enveloppes climatiques et aide à leur conception par micro-ordinateur, Thèse Doct.-Ing., INSA de Lyon.

Feustel, H.E. and Rayner-Hooson, A., Ed 1990. COMIS fundamentals, Technical report LBNL-28560, Lawrence Berkeley National Laboratory.

Frascatoro, G.V. and Perino, M. 2002. Natural vs. Mechanical Ventilation – A Tool to Help Making a Choice, International Journal of Ventilation, 1, No.2, pp101-108.

Ghiaus, C. and Allard, F. 2002. Assessing climatic suitability to natural ventilation by using global and satellite climatic data, Proceedings of the 8th International Conference on Air Distribution in Rooms, Copenhagen, Denmark, 8-11 September 2002, pp625-628.

Heiselberg, P., Ed. 2002. Principles of Hybrid Ventilation, IEA Energy Conservation in Buildings and Community Systems Program Annex 35: Hybrid Ventilation in New and Retrofitted Office Buildings.

Liddament, M.W. 1996. A guide to energy efficient ventilation AIVC, Coventry, UK.

Rumianowski, P., Brau, J. and Roux, J.J. 1989. An adapted model for simulation of the interaction between a wall and the building heating system, Thermal performance of the exterior envelopes of buildings IV Conference, Orlando, USA.