Numerical Analysis of Hybrid Ventilation Performance Depending on Climate Characteristics

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Abstract

This study, which formed part of the Annex 35 “Hybrid Ventilation in New and Retrofitted Office Buildings” project, was completed at LEPTAB and supported by the French Research Ministry and the ADEME (Agence De l’Environnement et de la Maîtrise de l’Energie). It consisted of modelling a typical classroom and comparing different control strategies to estimate the performance of a hybrid ventilation system for different climates.

The intention was that investigated classrooms were assumed to be on the middle level of a three-storey building, oriented South and surrounded by other classrooms subjected to the same conditions. Two mechanical ventilation systems were taken as references. These were: a mechanical exhaust system with a low consumption fan and without heat recovery, and a balanced mechanical ventilation system with two fans and some heat recovery. The hybrid ventilation approach investigated was a fan assisted natural ventilation system incorporating a demand control strategy based on indoor air temperature and CO_2 concentration.

The performance of this hybrid ventilation system was analysed in terms of energy consumption, indoor air and dry resultant temperatures, and CO_2 concentration level. Simulations of specific weeks in the year were performed for ten French cities and gave quite detailed patterns of behaviour. The study was extended to include yearly mean values of energy consumption. The results for both short and long time periods showed the potential of this specific hybrid ventilation system according to climate and control strategy. Hybrid ventilation is shown to provide improved air quality. Also, in relation to delivered energy, energy savings are possible but, except for Mediterranean cities, are not as much as with a mechanical system with heat recovery.

Key words: hybrid ventilation, control strategy, thermal comfort, CO_2 concentration, energy consumption.

1. Introduction

Recently, there has been increased interest in hybrid ventilation for the design of office buildings. The work of the International Energy Agency (IEA) Annex 35 “Hybrid Ventilation in New and Retrofitted Office Buildings” has resulted in the publication of a state of the art report (Annex 35, 2000) and a booklet with a CD-ROM (Heiselberg, 2002). These documents describe hybrid ventilation principles and cover design, control strategies, analysis methods and examples of existing buildings.

It is at the early planning stage that the designer should evaluate which ventilation system is most appropriate in terms of indoor air quality and energy consumption. Several tools are available for such a ventilation performance analysis (Delsante and Vik, 2000, Frascatoro and Perino, 2002). Various other studies on hybrid ventilation have already been undertaken. Comparisons between results obtained with different evaluation tools were presented by Delsante et al (2002a and 2000b). The potential of a hybrid ventilation system compared to more traditional ones was also presented by Cron et al (2002b) and by Elmankibi et al (2002) but was limited to a couple of locations. The influence of climate on hybrid cooling systems was reported by Spindler et al (2002). Barriers to the implementation of hybrid ventilation in Canadian offices and educational buildings due to regulations and climate have been outlined by Bourgeois et al (2002). Simulations of existing buildings have also been undertaken by Jeong and Haghighat (2002). However none of these studies have so far presented an analysis of the potential of a hybrid ventilation system in relation to climatic characteristics.

Thus the purpose of this paper is to broaden numerical studies on hybrid ventilation to include several cities and to draw some conclusions on the potential of this type of ventilation system. We first give a description of the typical classroom that was simulated for ten French cities. Then, after a presentation of the tools and the models used, results are given in terms of indoor air quality and dry resultant temperatures, CO_2 concentration and energy consumption. Simulations of specific weeks show the room air flow and thermal behaviour according to occupancy and outdoor conditions. Results were also extended to a whole year. The results for all the locations allowed us to see the potential of this hybrid ventilation system, given the climate and the control strategy used.

2. Classroom Description

2.1 Description

The simulated room was a single classroom in the middle level of a three-storey building. It was 9 m wide (along the external facade), 6 m deep and 3 m high. The room was oriented south and had a 1 m high double pane window along the whole facade. The external wall was assumed to consist of 80 mm of mineral wool and 150 mm of massive concrete, and the window to have a U-value of 2.7 $W/(m^2.°C)$. All external values were taken from meteorological data, except the CO_2 concentration that was assumed to be constant and equal to 400 ppm.

Normal school hours were set from 8h00 to 15h00 from Monday to Friday. Ventilation was provided during this schedule by the opening of two inlets and one outlet. Heating hours were from 7h00 to 15h00, from Monday to Friday. The set point air temperature was 21 °C during the heating hours and was reduced to 18 °C during the non-heating hours. Preheating brought the ventilation supply air to 18 °C for all the ventilation systems. The heating power depended on the climate and a radiative/convective partitioning of 0.5/0.5 was assumed. For the whole year study, we used the French regulation heating period i.e. from the 1st of October until the 20th of May.

A teacher and 24 pupils were assumed to occupy the room from 8h00 to 12h00 and from 13h00 to 15h00 on Mondays to Fridays. On Tuesdays, from 10h00 to 11h00, only 13 people were in the room, and at 14h00 everybody left the school. There was no school on Wednesday afternoons either. During occupancy, lighting loads were 10 W/m^2 and internal heat loads were 80 W/person and CO_2 production was 18 l/(h.person). These sensible internal gains were assumed to be 50% radiative and 50% convective. A solar shading factor was taken into account, depending on the location and the season, which was between 0.2 and 0.4 of the total incident solar radiation when this was more than 200 W/m^2.

2.2 Ventilation Systems

Infiltration was assumed to be constant and equal to 0.2 ach all the time. Window airing was defined by a constant air flow equal to 4 ach and could occur during normal school hours only. This value was additional to the other air flow rates and independent of the weather and the ventilation system. Window opening was scheduled when both the indoor and outdoor air temperatures were respectively higher than 23 °C and 12 °C, and closed again as soon as the indoor air temperature went below 21 °C. Some night cooling was provided in hot periods of the year, including Saturdays and Sundays. The night cooling was set to operate between 22h00 and 7h00 when the indoor temperature was higher than 24 °C and when the indoor-outdoor temperature difference was higher than 2 °C. It was turned off whenever the indoor temperature was below 18 °C.

Two mechanical ventilation systems were taken as references against which the performance of the hybrid ventilation system was compared. The first mechanical ventilation alternative (MV1) was a mechanical exhaust system in which a constant exhaust air flow of 0.15 m^3/s was assumed and supply air was provided through inlet grilles in the facade. The system was operated during normal school hours (from 8h00 to 15h00) and also, when necessary, to provide night cooling. The low fan power consumption was equal to 1000 $W/(m^3/s)$.

The second mechanical ventilation arrangement (MV2) was a balanced system that incorporated both an exhaust fan and a supply fan for the exhaust and supply of air at 0.15 m^3/s. A heat recovery unit with a temperature efficiency of 0.6 was included before the preheating system. Both fans were also operated for night cooling. The fan combined consumption was 2500 $W/(m^3/s)$.

For both mechanical ventilation systems, the fan air flow rate was set to maintain an indoor CO_2 concentration below 1200 ppm during occupancy.

The hybrid ventilation approach was based on fan assisted natural ventilation. The opening of the inlets and outlets was not only dependent on the school hours, but also on the CO_2 concentration in the classroom. The two inlet grilles were 0.5 m above the classroom floor. The first one opened when the CO_2 concentration was higher than 800 ppm, the second when it was higher than 1000 ppm. Finally the fan was turned on when the CO_2 reached 1200 ppm. The dead band values were 100 ppm. For this hybrid ventilation system, the classroom incorporated a 4 m high exhaust chimney to enhance the natural stack effect. No conductive heat transfer was taken into account through the chimney walls. Wind effects were taken into account: C_p values (wind pressure coefficients) from the AIVC literature were used to make an interpolation (see Table 1). For the chimney, the C_p was equal to -0.6 and was independent of the wind direction. The assisting fan was based on a low energy consumption device equal to 200 W/(m^3/s) and provided 0.15 m^3/s when it was on. Two night cooling modes were considered: the first one with the natural stack effect only (HVa) and the second using the fan (HVb).

Table 1. C_p values depending on the wind direction

Wind angle	0°	45°	90°	135°	180°
C_p value	0.25	0.06	-0.35	-0.60	-0.50

2.3 Meteorological Data and Locations

The ten French cities chosen for the simulation are given in Table 2.

Table 2. Cities chosen for simulations

French Cities	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 has a temperate climate covering an oceanic climate, a Mediterranean climate and a semi-continental climate. Thus government regulation divides France into winter and summer climatic zones. Three zones are considered in winter from the coldest one (H1) to the warmest one (H3) and four zones in summer from the coolest one (Ea) to the warmest one (Ed), see Figure 1 and Figure 2. Degree-days based on 18 °C and also on 21 °C (temperature set point during heating hours) for the heating period are given in Figure 3.

Figure 1. Winter climatic zones in France

Figure 2. Summer climatic zones in France

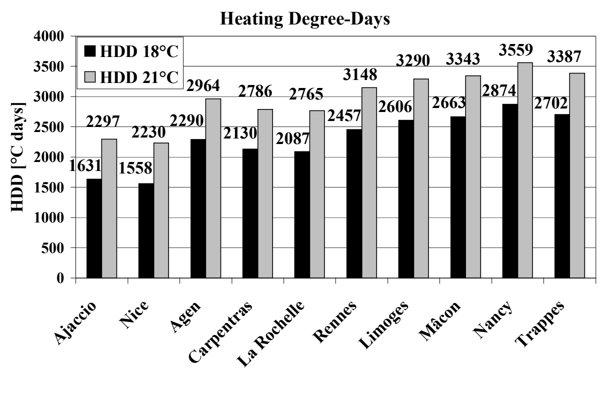

Figure 3. Heating degree-days during the heating period

Cooling degree-days were defined by computing the degree-days of outside temperatures above 26 °C, when air conditioning may be needed. The degree-days between 21 °C and 26 °C were also calculated. They were both taken during the non-heating period, see Figure 4.

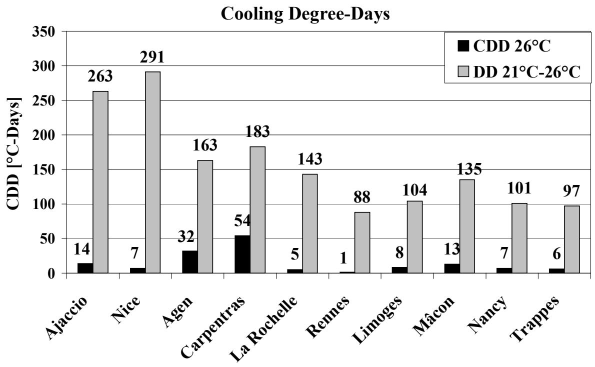

Figure4. Cooling degree-days and degree-days between 21 °C and 26 °C during the non-heating period

The simulation goals were to determine indoor temperatures, indoor air quality and performance results for specific weeks and for a whole year

3. Models used for the Simulations

The object oriented solver SPARK was used to perform the whole problem with differential and non linear equations. They were all implemented in SPARK for a previous study undertaken within tIEA Annex 35 (Cron and Inard, 2002a) with a time step equal to 60 s. Comparisons between simulation results for another case study and experimental data were also reported earlier (Cron et al, 2002b).

3.1 The Room Air Flow and Thermal Models

A single well-mixed zone with pressure varying hydrostatically was assumed for the models.

For a zone i with n openings, the “pure” air mass balance and the pollutant mass balance equations were:

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

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

As described previously, the infiltration and the window airing air flows were assumed to be constant. For both mechanical ventilation systems, the air flow rate was equal to 0.15 m^3/s.

For the hybrid ventilation system, when the fan was turned off, a simple power law relation was used for the inlet grilles and the chimney:

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

where
$C = 0.053$ $m^3/(Pa^{0.5}.s)$ for each grille
$C = 0.088$ $m^3/(Pa^{0.5}.s)$ for the chimney.

From Equation (3), the thermal buoyancy difference was incorporated to obtain the mass air flow rate, using a correction factor K_Q (Feustel et al, 1990):

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

with $\rho_{ij} = \dfrac{\rho_i + \rho_j}{2}$ the mean air density of the zone i and the zone j.

The thermal balance equation was:

$$\sum_{j=1}^{n}\left(m_{as\,ji}cp_{as} + m_{es\,ji}cp_{es}\right)T_j - \sum_{j=1}^{n}\left(m_{as\,ij}cp_{as} + m_{es\,ij}cp_{es}\right)T$$
$$+ S_{es}cp_{es}\left(T_{es} - T\right) + P_{heat\,conv} + P_{load\,conv} + \Phi_{conv}$$
$$= \left(\rho_{as}cp_{as} + \rho_{es}cp_{es}\right)V\frac{dT}{dt} + cp_{es}VT\frac{d\rho_{es}}{dt} \qquad (5)$$

3.2 The Room Envelope Models

Conduction heat transfer was described by an electrical 2R-3C model (Rumianowsky et al, 1989) that gives a good response to a high frequency excitation indoors, see Figure 5.

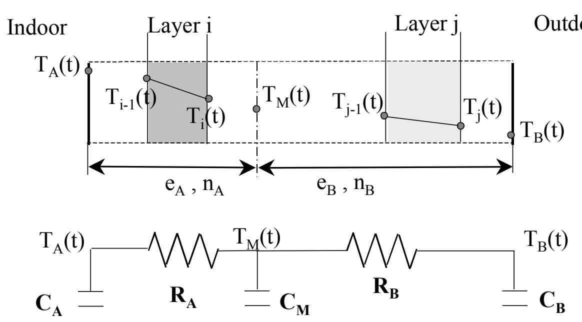

Figure 5. Description of the 2R-3C model

Considering the steady state, we have:

$$R_A = \sum_{k=1}^{n_A} \frac{e_k}{\lambda_k S} \quad (6)$$

$$R_B = \sum_{k=nA+1}^{n_B} \frac{e_k}{\lambda_k S} \quad (7)$$

with $n_A + n_B = n$, the number of layers.

Accumulation of heat in the wall was expressed by:

$$dQ_{accum} = C_A dT_A + C_M dT_M + C_B dT_B \quad (8)$$

where:

$$C_A = \sum_{j=1}^{n_A} \rho_j cp_j e_j S(1-\beta_j) \quad (9)$$

$$C_B = \sum_{i=n_A+1}^{n_B} \rho_i cp_i e_i S \delta_i \quad (10)$$

$$C_M = \sum_{j=1}^{n_A} \rho_j cp_j e_j S \beta_j + \sum_{i=n_A+1}^{n_B} \rho_i cp_i e_i S(1-\delta_i) \quad (11)$$

$$\beta_j = \frac{\sum_{k=1}^{j-1} \frac{e_k}{\lambda_k S} + \frac{e_j}{2\lambda_j S}}{R_A} \quad (12)$$

$$\delta_i = \frac{\sum_{k=n_A+1}^{i-1} \frac{e_k}{\lambda_k S} + \frac{e_i}{2\lambda_i S}}{R_B} \quad (13)$$

A comparison between this model response and a finite difference model response was undertaken. The position of the intermediate node was set to the optimum value of e_A that provided the minimum error E_{eA} where E_{eA} was defined by:

$$E_{eA} = \int_0^\infty \left| T_{A\,finite\,difference} - T_A \right| dt \quad (14)$$

The simulated room was assumed to be surrounded by rooms at the same conditions.

The outdoor long-wave radiation was given by:

$$\Phi_{netGLO\,o} = \varepsilon_o \sigma S_o \left(\frac{(1-\cos\beta)}{2} \left(T_o^4 - T_{so}^4 \right) \right) + \varepsilon_o \sigma S_o \left(\frac{(1+\cos\beta)}{2} \left(T_v^4 - T_{so}^4 \right) \right) \quad (15)$$

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

The convective heat transfer at the outdoor surface was calculated by:

$$\Phi_{conv\,o} = h_{conv\,o} S_o (T_o - T_{so}) \quad (16)$$

$$h_{conv\,o} = c + dU_{met}^n \quad (17)$$

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

A mean radiative temperature model was chosen to describe the long-wave heat transfer indoors. Part of the internal loads and the heating system was taken into account in the model. Given a surface i we modelled this exchange as:

$$\Phi_{netGLO\,i} = h_{rm\,i} S_i (T_{rm} - T_{si}) \quad (18)$$

$$h_{rm\,i} = 4\sigma\varepsilon_i \left(\frac{T_{si} + T_{rm}}{2} \right)^3 \quad (19)$$

$$T_{rm} = \frac{\sum_{i=1}^{n} (h_{rm\,i} S_i T_{si}) + P_{heat\,rad} + P_{load\,rad}}{\sum_{i=1}^{n} h_{rm\,i} S_i} \quad (20)$$

Indoors, solar radiation transmitted through the window was assumed to be entirely incident on the floor. One part was absorbed, depending on the solar properties of the floor, the other part was assumed to be reflected in a diffuse way. Both the transmitted and the reflected solar radiation were then absorbed over all the surfaces, depending on the surface ratio.

Convection indoors was obtained by:

$$\Phi_{conv\,i} = h_{conv\,i} S_i (T - T_{si}) \quad (21)$$

where $h_{conv\,i} = a \left| T - T_{si} \right|^b$ (22)

a and b were taken from Allard (1987):

a = 3 and b = 0 for a horizontal surface
a = 1.5 and b = 0.33 for a vertical surface.

The dry resultant temperature is a simplified indication of thermal comfort and was obtained by:

$$T_{rs} = \frac{h_{conv}\,T + h_{rm}\,T_{rm}}{h_{conv} + h_{rm}} \tag{23}$$

$$h_{rm} = 4\sigma\varepsilon\left(\frac{T_{rs} + T_{rm}}{2}\right)^3 \tag{24}$$

$$h_{conv} = a\left|\frac{T - T_{rs}}{D}\right|^b \tag{25}$$

h_{conv} is here the globe convection heat transfer, a and b are coefficients equal to 1.4 and 0.25 respectively and D is the globe diameter equal to 0.15 m.

4. Simulation Results and Discussion

Simulations of specific weeks for different seasons were performed for each city and results are given in terms of indoor and dry resultant temperatures, CO_2 concentration and energy consumption. Simulations were extended over a whole year, to better estimate the potential of this hybrid ventilation system according to the climate specifications. For every simulation, the two mechanical ventilation systems give the same temperatures and CO_2 concentrations since they differ only by the heat recovery and the fan consumption. Hybrid ventilation HVa and HVb are identical except when there is some night cooling, but in that case, results remain quite close. So we consider, except for energy consumption, only the results for mechanical ventilation (MV) and hybrid ventilation with the mechanical night cooling (HV).

4.1 Short Period Analysis

Results for specific days for both Trappes and Nice are given here: the first day is a Tuesday in winter (the 16th of January) and the second is a Thursday in summer (the 28th of June).

In winter, Figure 6 and Figure 7 show the set point temperature change in the morning (at 7h00) and in the afternoon (at 15h00). For both Trappes and Nice, the maximum temperature occurs with hybrid ventilation since, when the pupils enter the room at 8h00 or 13h00, there is at first no ventilation due to the low CO_2 level. The first grille opens when the CO_2 concentration reaches 800 ppm. Half of the pupils are in the classroom from 10h to 11h and we see a little temperature decrease due to lower internal loads.

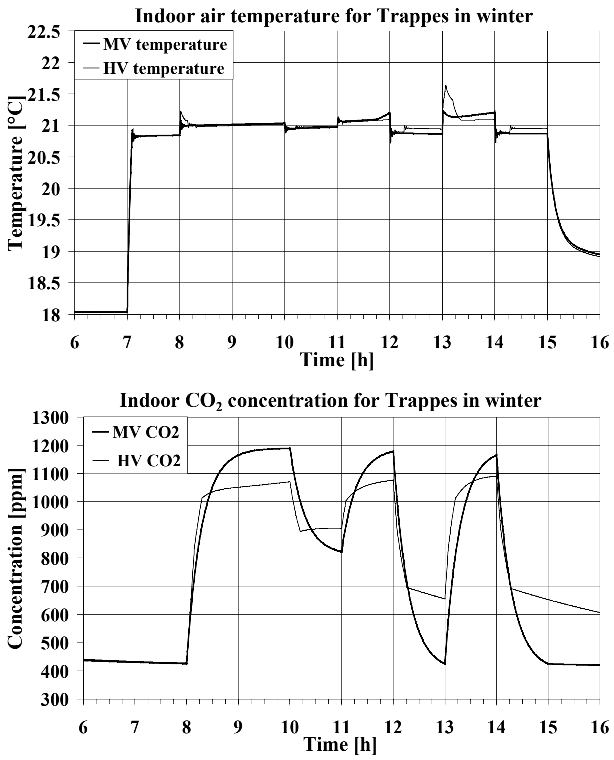

Figure 6 . Temperatures and CO_2 concentration in winter for Trappes (MV and HV systems)

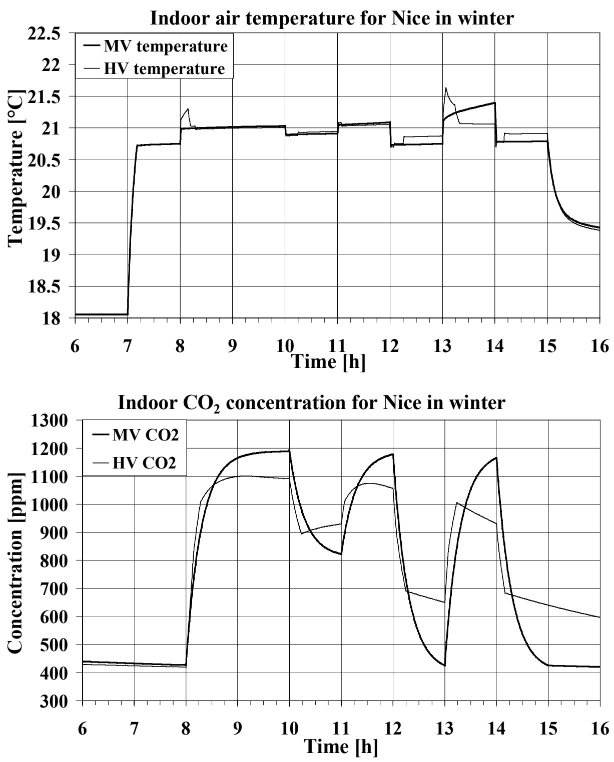

Figure 7. Temperatures and CO_2 concentration in winter for Nice (MV and HV systems)

During lunch time and between 14h and 15h, there is no occupancy and, whereas the fans are still working for the mechanical mode, the inlet grilles close for the hybrid system because of the low CO_2 concentration in the room. This gives, of course, some energy savings.

CO_2 concentration evolution is quite different between mechanical and hybrid ventilation modes. With all the pupils, the CO_2 tends to the constant value of 1200 ppm with mechanical ventilation, whereas with hybrid ventilation, the ventilation rate varies both according to CO_2 control and the stack effect. The latter is sufficient to ensure good indoor air quality and the fan doesn't need to be turned on. As a consequence CO_2 concentration is higher with mechanical than with hybrid ventilation. Also, in the morning, a lower outside temperature in Trappes than in Nice induces a more important stack effect and thus a lower CO_2 level for Trappes in the morning. When half of the pupils are in the class, the CO_2 level decreases and goes to around 800 ppm with mechanical ventilation. The hybrid ventilation control closes one inlet grille at 900 ppm, so the air flow rate is reduced, whereas mechanical ventilation continues to provide the constant 0.15 m^3/s. Overall, the hybrid system leads to a concentration of between 900 and 1000 ppm which is, on average, higher than with the mechanical mode. However, we can say that for this specific winter day, the maximum value of CO_2 concentration is lower with hybrid ventilation than with mechanical ventilation for both cities. There is also less exposure to a concentration higher than 1000 ppm for Nice (where 1000 ppm is the value given in the French regulation).

In summer, Figure 8 and Figure 9 show the results of simulations. For Trappes, as there is at first no ventilation for hybrid ventilation, the temperature is seen to rise quite quickly and window airing is provided earlier for the hybrid mode than for the mechanical one. Nice has higher temperatures, so window airing is provided as soon as the pupils are in the room. When the window is opened for both systems, the hybrid ventilation temperature is higher than the mechanical ventilation one. This is initially due to a lower stack effect in summer compared to winter. Then, with the window airing, only one inlet grille is opened for hybrid ventilation. Hence there is a lower air flow rate with the hybrid system than with the mechanical one. For these reasons, the maximum value of temperature is obtained with hybrid ventilation, but doesn't exceed 26 °C in both

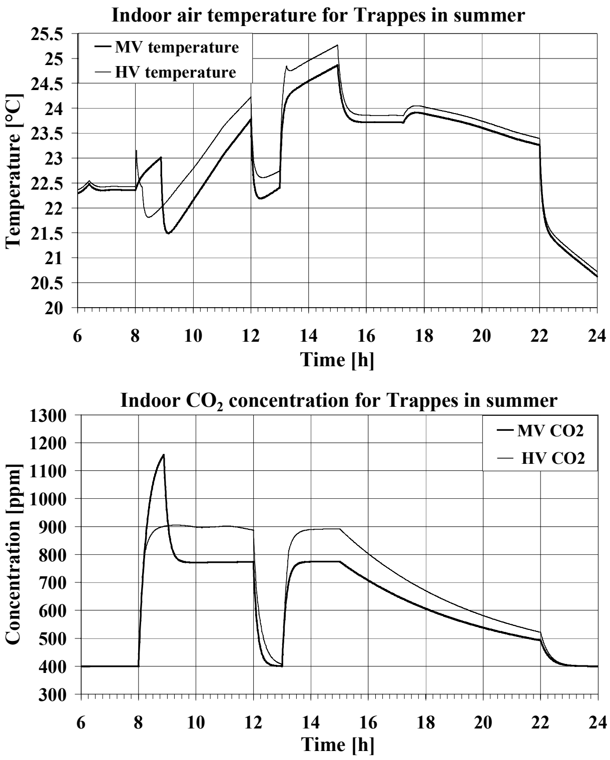

Figure 8. Temperatures and CO_2 concentration in summer for Trappes (MV and HV systems)

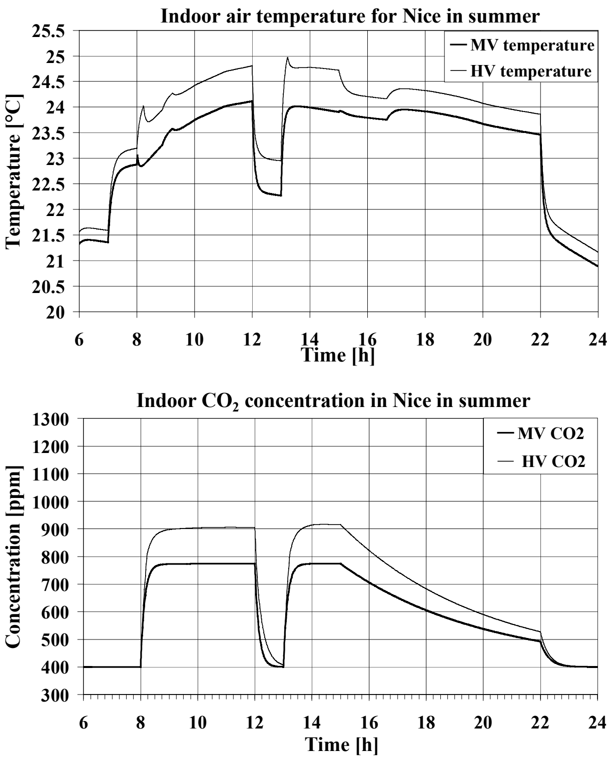

Figure 9. Temperatures and CO_2 concentration in summer for Nice (MV and HV systems)

cities. At lunch time, window airing provides some free cooling. After 17h, solar shading is removed for both cities which explains the slight temperature rise at that moment. Nice is warm enough to have night cooling on the night before and the night after this day, whereas Trappes has some night cooling only during the following night.

Since the window is opened almost immediately with hybrid ventilation, CO_2 concentration stays below 1000 ppm, and is lower than that given by mechanical ventilation where there is no window airing. But, when the mechanical mode has some window airing, the CO_2 concentration is lower than with the hybrid mode. The example of Trappes shows that, even if the mean CO_2 concentration is higher with the hybrid ventilation, the time exposure to a concentration higher than 1000 ppm remains lower with hybrid ventilation.

4.2 Yearly Simulations

As presented by Ghiaus and Allard (2002), two balance temperatures for the room equipped with mechanical ventilation have been calculated. The first is with the window closed and the second is with the window open. Given the requirements of indoor comfort, Figure 10 presents the results subject to the required heating, cooling and air–conditioning needs.

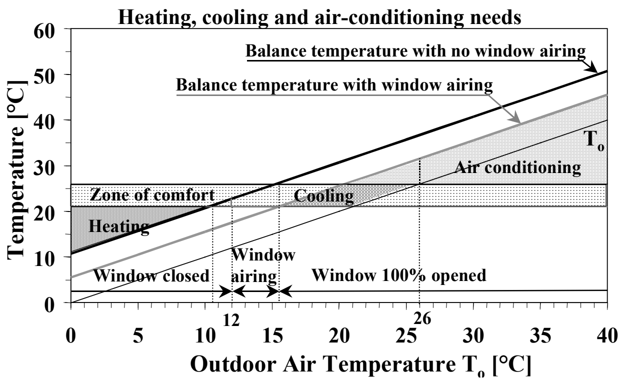

Figure 10. Heating, cooling and air-conditioning needs during occupancy

For mechanical ventilation, Figure 10 shows some heating is needed whenever the outdoor air temperature is below 10.5 °C. When the outdoor temperature is equal to 12 °C, the window opens and can be successively opened and closed between 12 °C and 16 °C, depending on the solar gains that weren't taken into account for this diagram. When the outside temperature is higher than 16 °C, the window is opened all the time during school hours. Even if window airing can provide some free cooling, this part of the figure begins to be critical in terms of thermal comfort when considering solar radiation gains and the lower stack effect for hybrid ventilation in this range of temperatures. For an outdoor air temperature higher than 26 °C, the room should be air-conditioned to provide satisfactory thermal comfort. Figure 10 shows the importance of the needs of thermal comfort conditions in relation to the outdoor air temperature, thus the annual outdoor air temperature distribution during occupancy was calculated from the weather data files, see Table 3.

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

Cities	T_o<12°C	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

Several categories of cities can be distinguished here. First of all, Ajaccio and Nice have an annual outdoor air temperature higher than 16 °C during more than 45 % of the occupancy time and have an outside temperature lower than 12 °C for less than 25% of the occupancy. These two cities are also the ones that have the lowest heating degree-days and the highest degree-days between 21 °C and 26 °C.

Although Agen and Carpentras experience outside temperatures higher than 16°C during occupancy less frequently than Ajaccio and Nice (30% and 40%), they have more cooling degree-days and fewer degree-days between 21 °C and 26 °C than Nice and Ajaccio. So, Agen and Carpentras belong to a second category of cities.

Although Agen and La Rochelle are in the same climatic zones (H2 and Ec) they show climatic differences, La Rochelle is cooler since the outside temperature is never above 26 °C during the occupancy time. Also, although La Rochelle and Carpentras have about the same heating degree-days, the cooling degree-days and the annual temperature distribution are different. In fact both La Rochelle and Rennes have a quite mild climate, with the Atlantic Ocean influence. They have about the same frequency of outside temperature between 12 °C and 16 °C as between 16 °C and 26 °C. But Rennes has a cooler summer since the cooling degree-days and the degree-days between 21 °C and 26 °C are both lower.

The last category includes Limoges, Mâcon, Nancy and Trappes that have an outside temperature lower than 12 °C for about 60% of the occupancy time and have the highest heating degree-day values.

The global performances of hybrid ventilation on indoor thermal comfort and air quality were analysed over the whole year. For all the cases, the indoor air and the mean dry resultant temperatures with the two hybrid systems are both less than 0.4 °C higher than with the mechanical mode (see Figure 11). Air flow rates are in fact lower with hybrid ventilation for reasons explained previously when we analysed specific days. These differences appear to be a little more important for cities with a warmer climate in the summer zone Ed. Figure 11 shows that Carpentras (zone Ed) has the highest mean dry resultant temperature. The reasons are that Carpentras is the city with the highest cooling degree-days and it has an outdoor temperature higher than 26 °C during 6 % of the occupancy time. On the other hand Rennes (zone Ea – the coolest zone) has the lowest mean dry resultant temperature value and has in fact, as shown previously, the lowest number of cooling degree-days and degree-days between 21 °C and 26 °C.

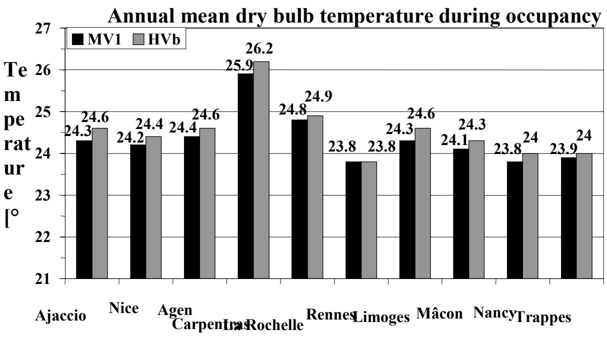

Figure 11. Annual mean dry resultant temperature during occupancy

The annual mean CO_2 concentration results show a real difference between the warm climates and the mild or colder ones. For Ajaccio, Nice, Agen and Carpentras, the value of the mean CO_2 concentration is clearly higher with the hybrid system and it is the opposite for all the other cities (the difference for Mâcon is too small to give a conclusion). Mean CO_2 concentration is, in fact, strongly associated with window airing (i.e. periods with an outdoor temperature higher than 12 °C). There are also some contrary effects between window airing and the stack effect. As shown previously, there is a lower air flow rate when the window is opened for hybrid ventilation since one inlet grille is closed. Hence for Ajaccio, Nice, Agen and Carpentras, the window airing effect prevails over the stack effect. For Mâcon, these effects are approximately equal. For all the other cities, the stack effect prevails over window airing.

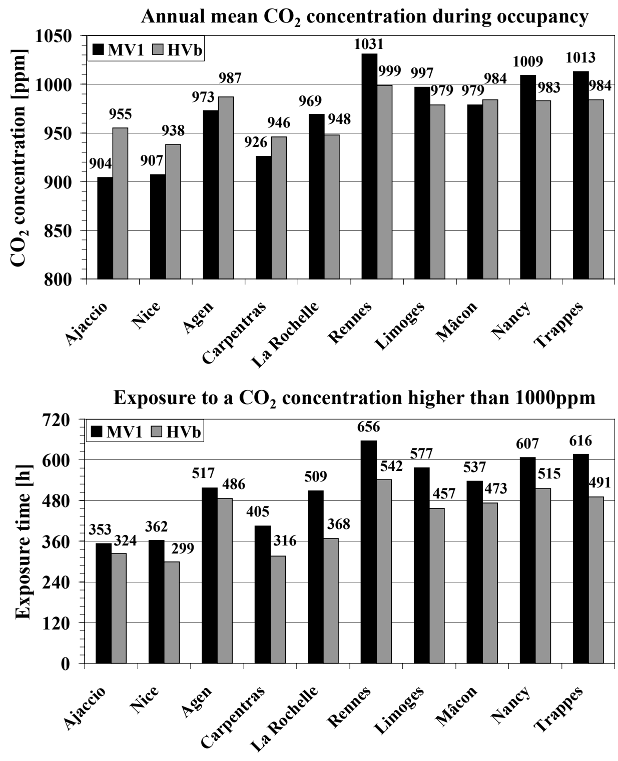

Figure 12. Annual mean CO_2 concentration and exposure to a CO_2 level higher than 1000 ppm

Even if Mâcon and Rennes both have an outdoor temperature lower than 12 °C during 59 % of the occupancy time, they are respectively in the summer zones Ec and Ea which explains the differences in Figure 12. Likewise La Rochelle and Agen have a similar frequency of outdoor temperature lower than 12 °C and are both in the zones H2 and Ec. However they have a difference in terms of mean CO_2 concentration. In fact, according to the degree-days, La Rochelle is less warm in summer and less cold in winter than Agen.

Although in some cities, hybrid ventilation seems to be a priori less efficient with regard to indoor air quality, Figure 12 shows that there is less exposure to high CO_2 concentrations in all cases. As shown in the specific day analysis (Figures 6, 7, 8 and 9), there can be an exposure to a concentration higher than 1000 ppm for mechanical ventilation (when the window is still closed), but the mean concentration value is lower for mechanical ventilation than for hybrid ventilation (when the window is opened). Here again this is due to the antagonist effects of window airing and stack effect. The important conclusion is that, irrespective of climate, when applying this specific control strategy, indoor air quality, in terms of CO_2 concentration, is better with hybrid ventilation than with mechanical ventilation.

Table 4 illustrates that hybrid ventilation provides some energy savings on heating and preheating for all cases, mostly because the inlets are closed during non-occupancy. The energy saving difference is due to a higher stack effect for colder cities since they induce a higher air flow rate and thus more heating and preheating consumption. In relation to fan energy savings, warm cities and especially Ajaccio and Nice, have more fan consumption than cold cities, due to more night cooling requirements. Differences between the two hybrid modes are not significant, so results are only presented for HVb.

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

Energy savings	Heating HVb/MV1	Preheating HVb/MV1	Fan HVb/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 correlates the previous conclusions: Ajaccio and Nice have less preheating and more energy consumption for the fan than the other cities, thus hybrid ventilation with its low fan power leads to high energy savings compared to both mechanical modes. For the other cities, hybrid ventilation provides energy savings for heating, preheating and the fan compared to the first mechanical mode (i.e. extract only) but energy consumption is higher than for the mechanical system with heat recovery. The reason is that preheating prevails over the fan in terms of consumption for these cities. Finally all these characteristics appear also in the annual total energy consumption for all the ventilation modes (Figure 13).

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

Cities	Fan / Preheating (MV1) x 100	HVb compared to MV1	HVb compared to 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

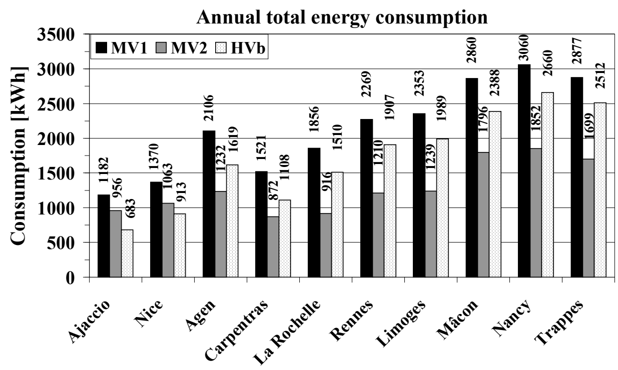

Figure 13. Annual total energy consumption for the different ventilation modes

Finally Figure 13 shows that the performance of fan assisted natural ventilation is dependent on the heating needs. The cities that require more night cooling 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. These results may be generalised to other buildings, as soon as the temperature range distribution over the year is known. With these results, the relation between climate specifications and energy savings over a whole year can be finally summarise, see Table 6.

Table 6. Hybrid ventilation energy savings estimated from the climatic conditions

Heating needs	Ventilation 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			

5. Conclusions

This study has shown the general behaviour of fan assisted natural ventilation with control based on temperature and CO_2 concentration. This approach provides air flow on demand and, even if the mean CO_2 concentration is a little higher with the hybrid system than with the mechanical system, overall indoor air quality is improved since periods of exposure to CO_2 above 1000 ppm are reduced. A drawback may be that the room air temperature is a little higher and may be uncomfortable, especially in really warm cities. This specific hybrid ventilation provides some energy savings, but not as much as a mechanical system with heat recovery, except for Mediterranean cities. Heating and cooling degree-days give an initial idea about climate characteristics and the resultant impact on indoor conditions. However 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 this gives better indications on overall room needs.

This type of analysis may be useful for designers at the beginning of a project since it provides knowledge about the technical solution that may be appropriate for a specific building. Designers also have to decide between improving indoor air quality and reducing energy consumption to as low as possible. An interesting future work would be to extend these simulations to other specific climates such as humid, very cold or very warm climates. This would enable an atlas of suitability of hybrid ventilation performance to be developed.

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Nomenclature

cp	specific heat	[J/(kg°C)]
e	thickness of a layer	[m]
h	heat transfer coefficient	[$W/(m^2 °C)$]
m	mass flow rate	[kg/s]
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
ji	from the zone j to the zone i

load	internal loads
M	node M
met	meteorological
net	net radiative heat transfer
o	outdoors
rad	radiative
rm	mean radiative
rs	dry resultant
s	surface
v	sky vault

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