# AIR QUALITY ANALYSIS

The aim of this chapter is to provide some examples of analysis that can be performed by the whole building model as defined at chapters 2 to 6.

The building model can be used to evaluate the performance of different ventilation systems regarding *indoor air quality* during occupancy. The $CO_2$ concentration can be used as air quality indicator. The outdoor $CO_2$ concentration is supposed to be equal to 300 *ppm*. A maximum indoor-outdoor threshold of 1000 *ppm* can be considered for comfort (chapter 1, table 1.1), corresponding to a maximum absolute concentration of 1300 *ppm*.

Both natural and combined natural/fan powered ventilation systems can be considered. When the absolute concentration threshold of 1300 *ppm* is exceeded, different improvement strategies can be tested to improve the indoor air quality level.

As wind speed and wind orientation influence natural ventilation air flows, Belgian Uccle Meteonorm Year Weather Data can be considered because they include wind speed and wind orientation. Those weather data totalises 2237 *15/15 Degree-Days per year*, while Uccle Reference Mean Year data add up 2149 *15/15 Degree-Days per year*, which is very close.

Indoor quality analysis can be performed for a house (§ 7.1) and for an office room (§ 7.2) equipped with natural only or combined natural/fan powered ventilation devices.

## 7.1. House air quality analysis

The natural ventilation model described on fig 6.3 can be applied to Seneffe house (Annex 2, §2.3) considered with two party walls interconnecting the house with the neighbors (fig. 7.1).

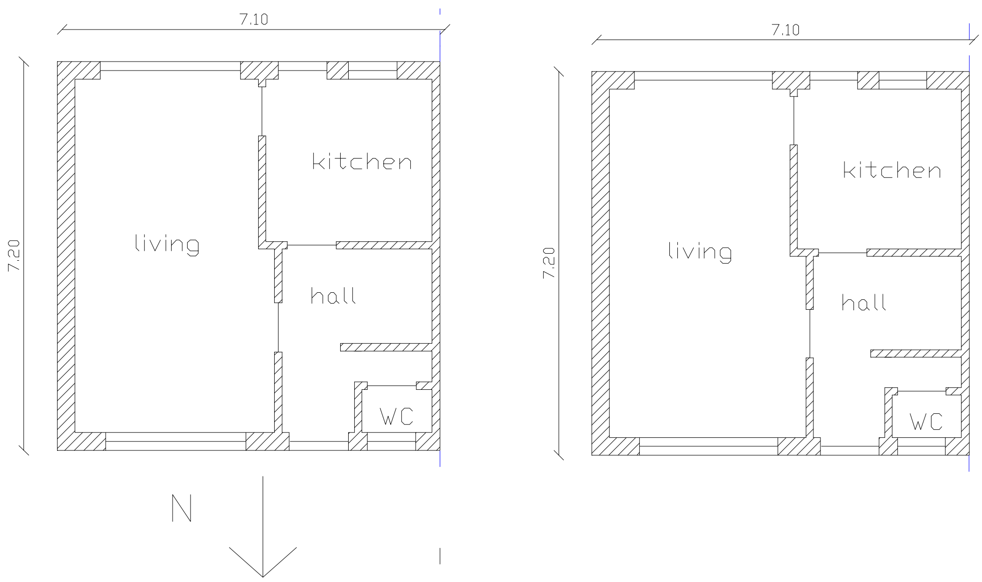



*Fig. 7.1: Seneffe house plans*

Ventilation systems A, B, C and D can be applied successively (chapter 6 § 6.2.3). Ventilation supply and exhaust air flow rates, Controlled Supply Orifices as well as Transfer Orifices can be sized to globally reach *3,6 m³/h* per m² of floor area for a pressure difference of 2 *Pa* (minima air flow rates can be considered for each room, in agreement with Belgian standards) .

The supply air flow rate, related to dry rooms, equals 272 *m³/h* while the exhaust air flow rate, related to humid rooms, is 150 *m³/h* for a 257 *m³* house volume and 91.4 *m²* total heated floor area. Air flow through infiltration cracks can equal 12 *m³/h.m²* of external wall area for a pressure difference of 50 *Pa*

A wind exposed site can be considered and pressure factors can range from -0.2 for North orientation, to 0.7 for South, when wind comes from SW.

The ground floor (zone 1) can be occupied during the day from 7 AM till 11 PM, while the first floor (zone 2) can be occupied during the night from 10 PM till 7 AM. Zone 1 can be heated at 22°C by radiators during the occupancy, while zone 2 can only be maintained at a minimum 10°C temperature (annex 7, fig A 7.1).

Four occupants can be considered, each of them producing $0.016.10^{-3}$ $kgCO_2/s$. Simulations can be performed on a quarter of hour basis.

Fig. 7.2 displays the evolution of indoor $CO_2$ concentration for a natural type A ventilation system: the $CO_2$ concentration in zone 1 exceeds the 1300 *ppm* limit as soon as the wind speed falls under 5 *m/s*. In zone 2, it reaches the limit even when wind speed exceeds 5 *m/s*.

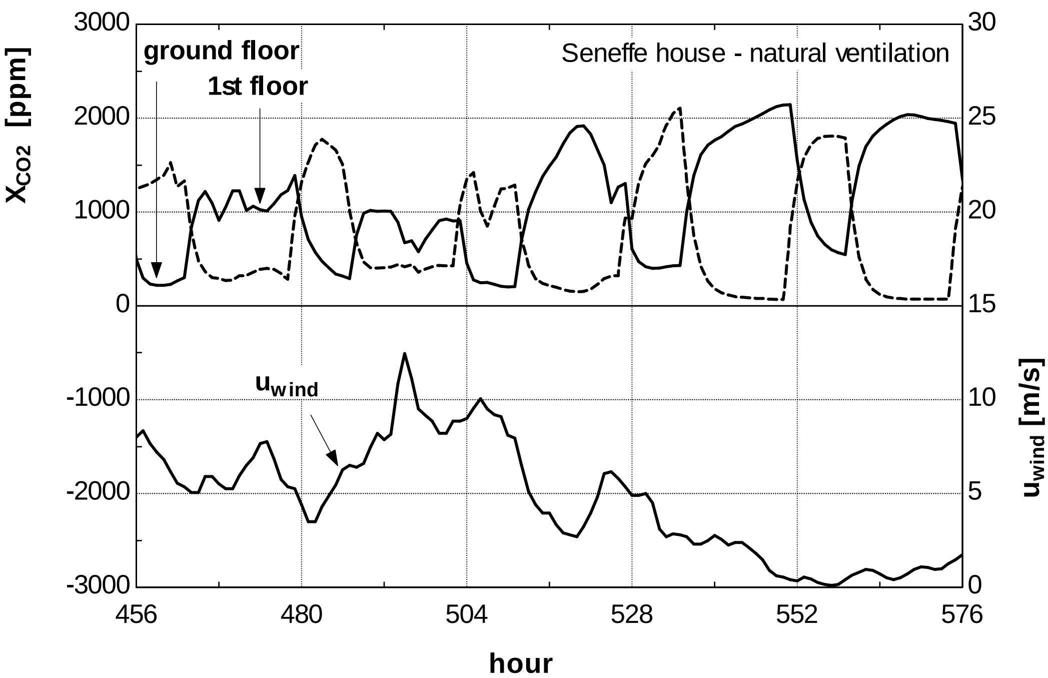



*Fig. 7.2: Evolution of the indoor $CO_2$ concentration in zone 1 (ground floor) and 2 (first floor) in parallel with wind speed, for a house natural type A ventilation system.*

The average excess of $CO_2$ concentration can be computed over the whole year in both zones, during the occupancy hours. It can be compared to a maximum 1300 *ppm* threshold (fig. 7.3).

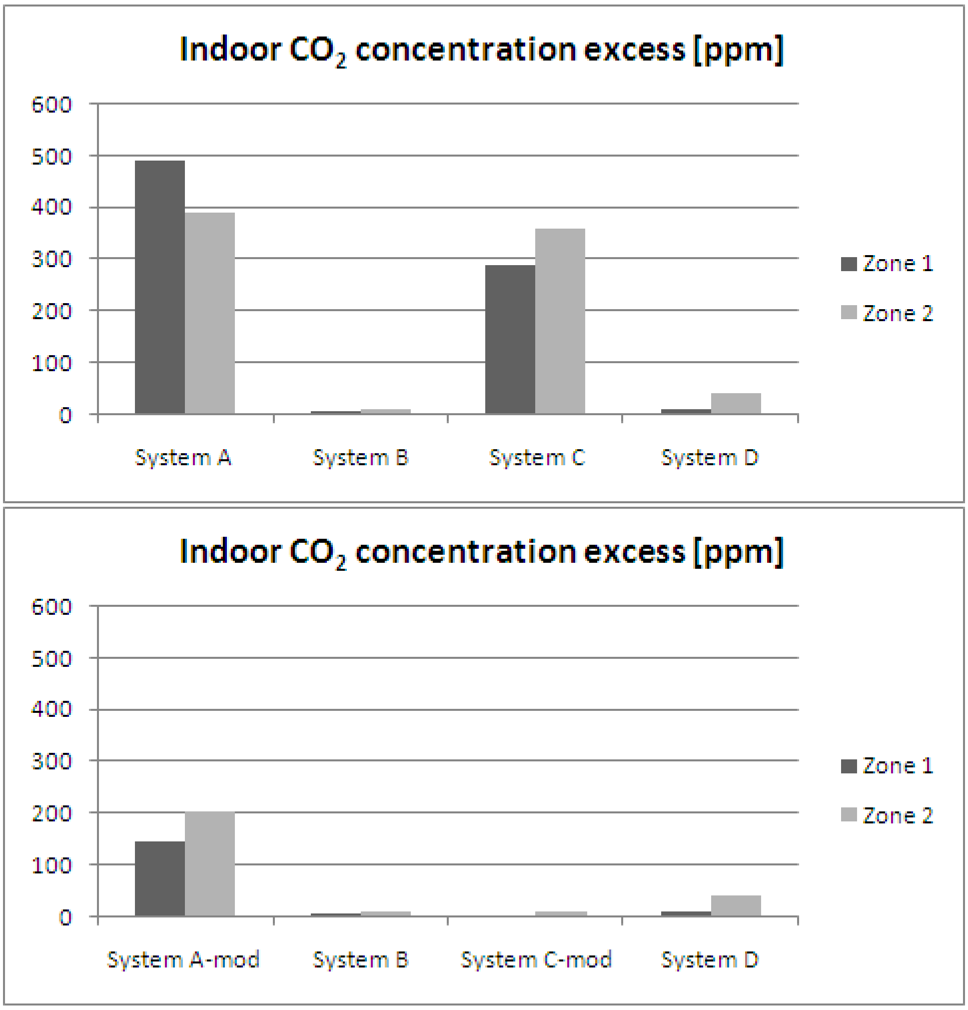



*Fig. 7.3: Yearly mean indoor $CO_2$ excess of concentration in zone 1 (ground floor) and 2 (first floor) related to types A, B, C and D ventilation systems, before (up) and after (down) improvement of types A and C sizing criteria.*

The maximum $CO_2$ comfort threshold is respected with systems B and D, while it isn't with systems A and C (fig. 7.3, up). The reason is that the Belgium standards imposed 150 *m³/h* exhaust air flow rate is too low, because it is computed on the basis of the whole humid rooms floor area (kitchen, bathroom, lavatories), while the 272 *m³/h* supply air flow rate depends on the dry rooms total floor area (living room, office, bedrooms), which is higher than the humid rooms floor area.

The ventilation Controlled Exhaust Orifices of system A as well as the exhaust fan of system C are sized on the basis of that low 150 *m³/h* exhaust air flow rate, so both systems are undersized and they can only provide an yearly average fresh air flow rate equal to 170 *m³/h* which is lower than the recommended 272 *m³/h* supply air flow.

When the natural and mechanical ventilation devices of system A and C are sized on the basis of the recommended 272 *m³/h* supply air flow rate, the results are better (fig. 7.3, down).

System C fulfills the $CO_2$ concentration comfort requirements while system A improves its performances.

The yearly average fresh air flow rates provided by the four systems during occupancy are, after improvement of types A and C sizing criteria:

A: 245 *m³/h* B: 288 *m³/h* C: 280 *m³/h* D: 280 *m³/h*

This means a minimum air renewal rate of $1.1\ h^{-1}$ during occupancy for fan powered ventilation systems.

## 7.2. Office air quality analysis

Indoor air quality can be analysed in an office room provided either with natural or with combined natural/fan powered ventilation system. Different ventilation improvement strategies can be tested if the $CO_2$ concentration exceeds a maximum of 1300 *ppm* during occupancy hours.

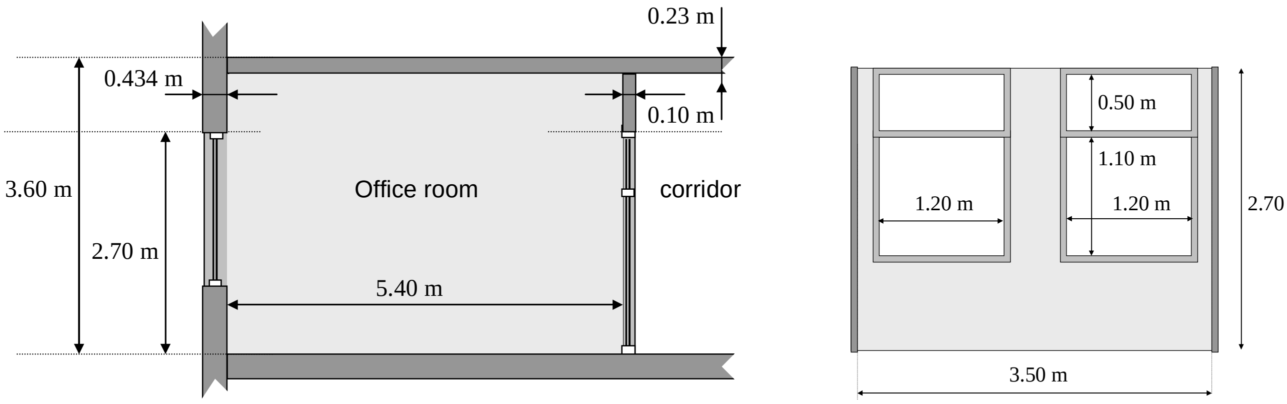



*Fig. 7.4. Office room (IEA 27): section (left) and facade (right).*

### 7.2.1. Natural ventilation systems

The office room described in chapter 4 §4.1.3 can be repeated four times with different orientations (South, West, North, East) in order to generate a simplified dynamic five zones model (chapter 5, fig. 5.34) that can be combined to the natural ventilation model defined in (chapter 6, fig. 6.5). Indoor air quality analysis can be performed with that model.

Offices can be occupied from 7 AM till 6 PM every day except on the week-end. Controlled Supply Orifices as well as Transfer Orifices can be sized to reach *2,9 m³/h* per m² of floor area for a pressure difference of 2 *Pa*. The corridor can be connected to a ventilation shaft through a Controlled Exhaust Orifice, sized on the basis of the total four offices air flow.

Offices can be located at ground floor. Two types of site can be considered: a *wind half-exposed site* for which pressure factors can range from -0.3 for West and East orientations, to 0.4 for South, and a *wind exposed site* for which pressure factors can range from -0.5 for West and East orientations, to 0.7 for South, when wind comes from SW.

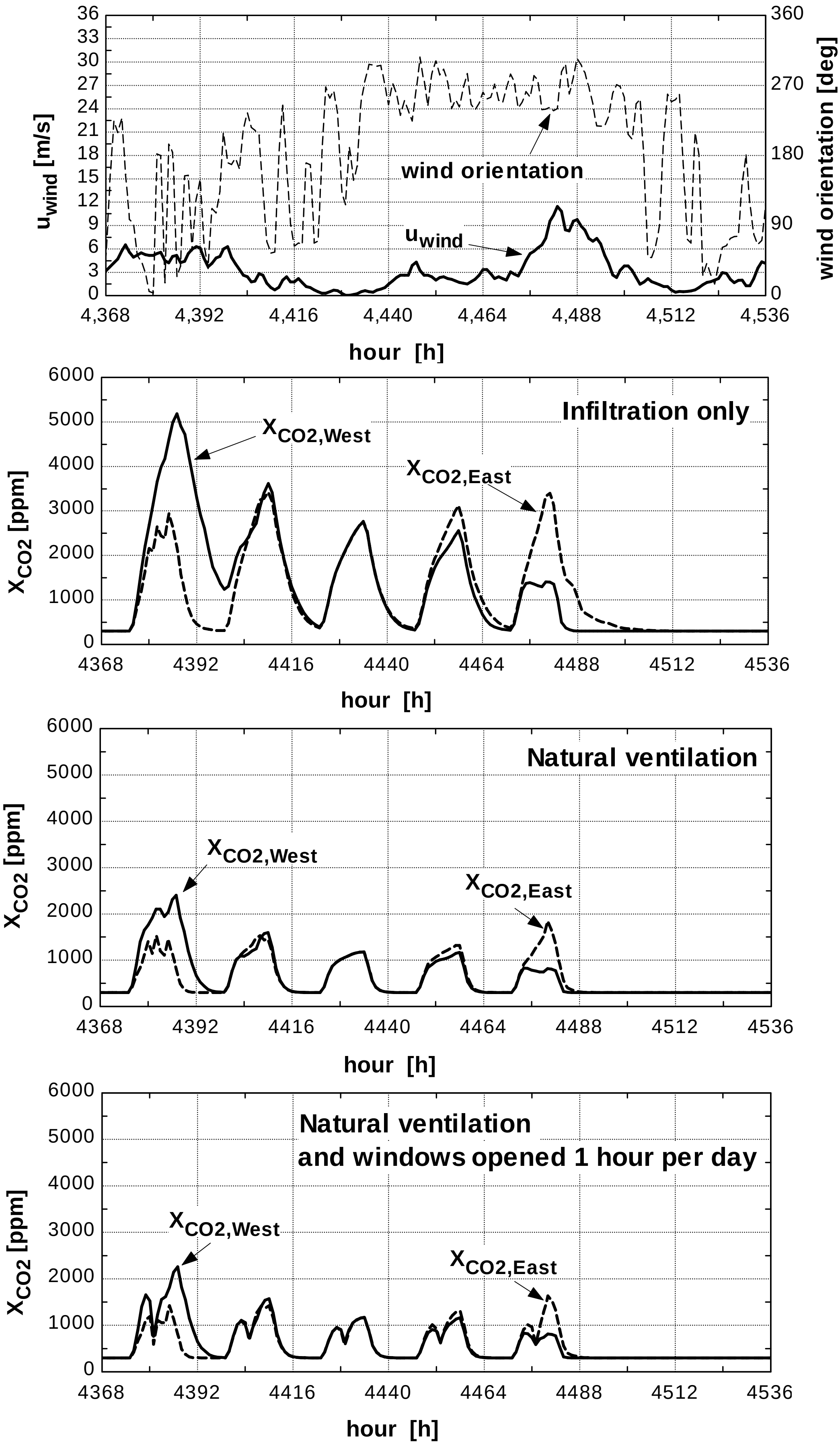
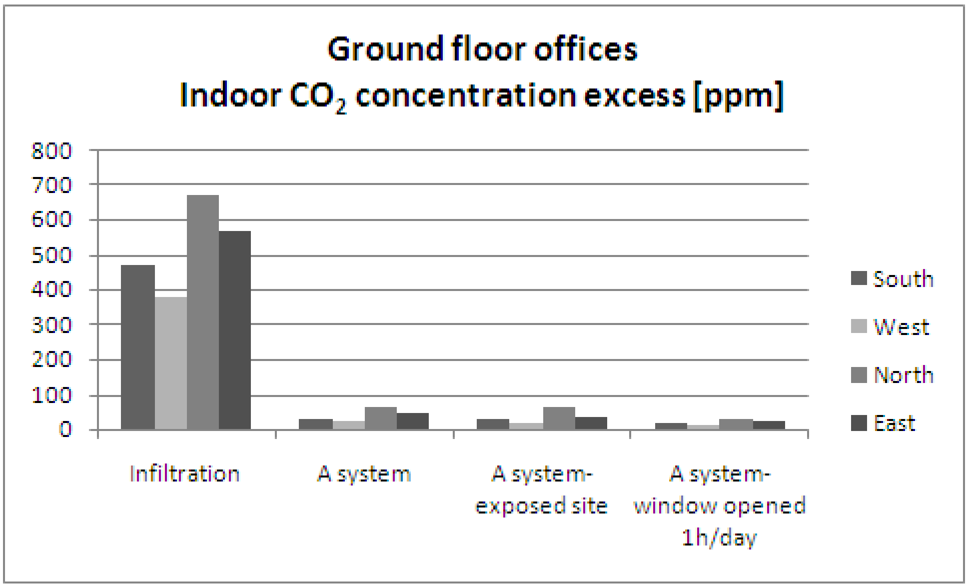



*Fig 7.5 Indoor $CO_2$ concentration for west an east oriented offices in a half exposed site.*

Fig 7.5 displays the evolution of indoor $CO_2$ concentration over a week for west and east oriented ground floor offices, in a half exposed site.

Infiltration only (fig 7.5 up) means that there are neither Controlled Supply Orifices (only facade infiltration) nor Transfer Orifices (only 40 *cm²* aperture under the doors). The Controlled Exhaust Orifice and the exhaust ventilation shaft are those located in lavatories, thus sized for a rather small air flow. The internal $CO_2$ concentration reaches more than 2000 *ppm*. It is smaller in east offices at the beginning of the week, because wind is east oriented, and it is smaller in west offices at the end of the week, the wind being west oriented (wind orientation is measured clockwise from the north).

A natural ventilation system (fig 7.5 middle and down) is sufficient to maintain a moderate indoor air quality i.e. a maximum concentration of 1300 *ppm* most of the week.



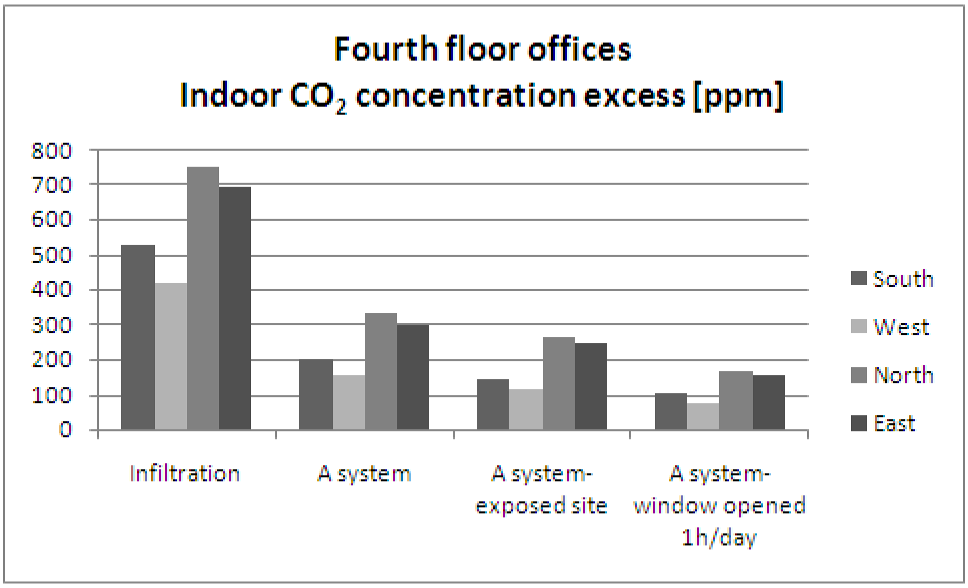




*Fig 7.6 Indoor $CO_2$ excess of concentrations yield by natural ventilation for different office orientations at ground floor (up) and at fourth floor (down) for a four levels office building.*

The average excess of $CO_2$ concentration, compared to a 1300 *ppm* threshold, can be computed on a quarter of hour basis over the whole year during occupancy (fig 7.6). Occupancy gains due to lighting and appliances can be equal to *10 W/m²* of floor area.

The model shows that a natural type A ventilation system can be sufficient to maintain a moderate indoor air quality level over a whole year in ground floor offices, while it can't in fourth floor offices, for a four levels building. The reason is that stack effect is not strong enough for the fourth building level.

The model can also evaluate the effect of ventilation improvement strategies in fourth floor offices. For example, opening the windows one hour per day until 10 % of their area, when outdoor temperature is higher than 5°C, can improve air quality but it can't fulfill air quality requirements.

Other strategies could be simulated by the model: ventilation air flow could be increased by introducing a stronger stack effect i.e. by increasing the height of ventilation shaft (it is here 1 *m* higher than the roof) or by introducing fan powered ventilation.

The model shows higher excesses of $CO_2$ concentration for North and East oriented offices (fig 7.6). Those offices are mainly wind depressurized over the whole year, while South and West offices are wind pressurized. So East and North offices mainly receive polluted air coming from South and West offices and their indoor air quality level decreases.

### 7.2.2. Combined natural/fan powered ventilation systems

The office room described in chapter 4 §4.1.3 can be repeated 12 times with different orientations (South, West, North, East) in order to compose a wing area. The wing can be identically repeated once, in order to create two wings per floor (fig. 7.7). The whole floor can be repeated again to generate four identical levels. So the whole building can add up 8 identical offices wings, i.e. 96 office rooms, each of 19 *m²* floor area.

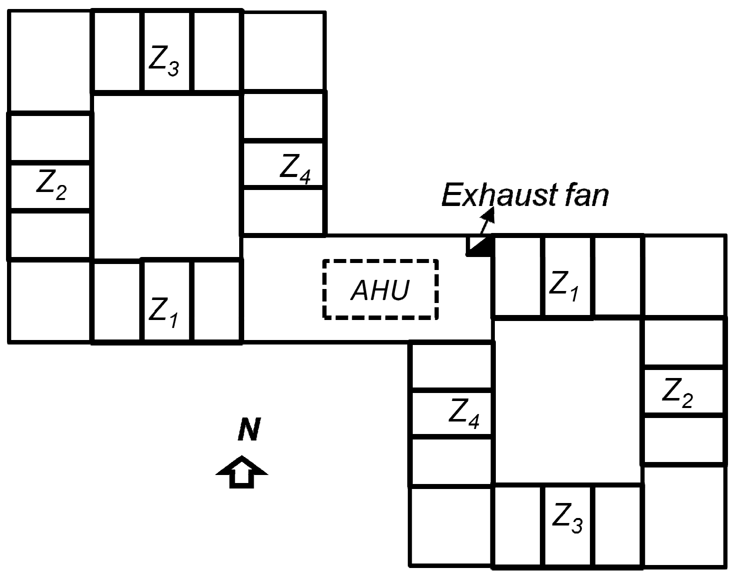



*Fig 7.7 Building floor layout: two wings per floor and 4 offices per wing, with different offices orientations.*

Computation can be performed for the offices of one wing located on ground floor. The results can be first multiplied by the number of wings in order to estimate fan air flow rates related to the whole building. Fans electricity consumptions can be divided by the number of wings in order to get the consumption related to one offices wing.

Three ventilation strategies can be treated by the model (annex 6, fig A 6.5):

- A type C ventilation system (annex 6, fig A 6.5 up left): offices can be equipped with Controlled Supply Orifices, and with air Transfer Orifices connected to corridors. Corridors can be equipped with Controlled Exhaust Orifices (one exhaust grid per floor) connected to a ventilation shaft surmounted by an exhaust fan. The same fan can exhaust air for all the building wings and can work only during building occupancy hours (fig. 7.8).

- A first type D ventilation system (annex 6, fig A 6.5 up right): fresh air can be supplied to the different offices by a fan through a duct network. Offices can be connected to corridors through Transfer Orifices. Corridors can be equipped with Controlled Exhaust Orifices (one exhaust grid per floor) connected to a ventilation shaft surmounted by an exhaust fan. Both fans can supply/exhaust air for all the building wings and can work only during building occupancy hours. The scheme would be a combination of fig 7.9 supply network with fig 7.8 exhaust network.

- A second type D ventilation system (annex 6, fig A 6.5 down): fresh air can be supplied to the different offices by a fan through a duct network and offices polluted air can be also returned by a fan through a duct network. The system can be provided with a plate heat recovery exchanger or not. The supply and return fans can be involved in an Air Handling Unit that deliveries the whole building during occupancy hours. The air duct system model can be combined with natural ventilation entities (infiltration, apertures under doors and lavatories ventilation shaft) to account for a natural cross ventilation occurring through the corridor during no occupancy hours (fig. 7.9).

Each of those ventilation models can be combined to the simplified dynamic five zones model (chapter 5, fig. 5.34) in order to generate a whole building model able to provide resulting air flows, pressure drops, indoor $CO_2$ concentrations, heating and cooling building demands and energy consumptions.

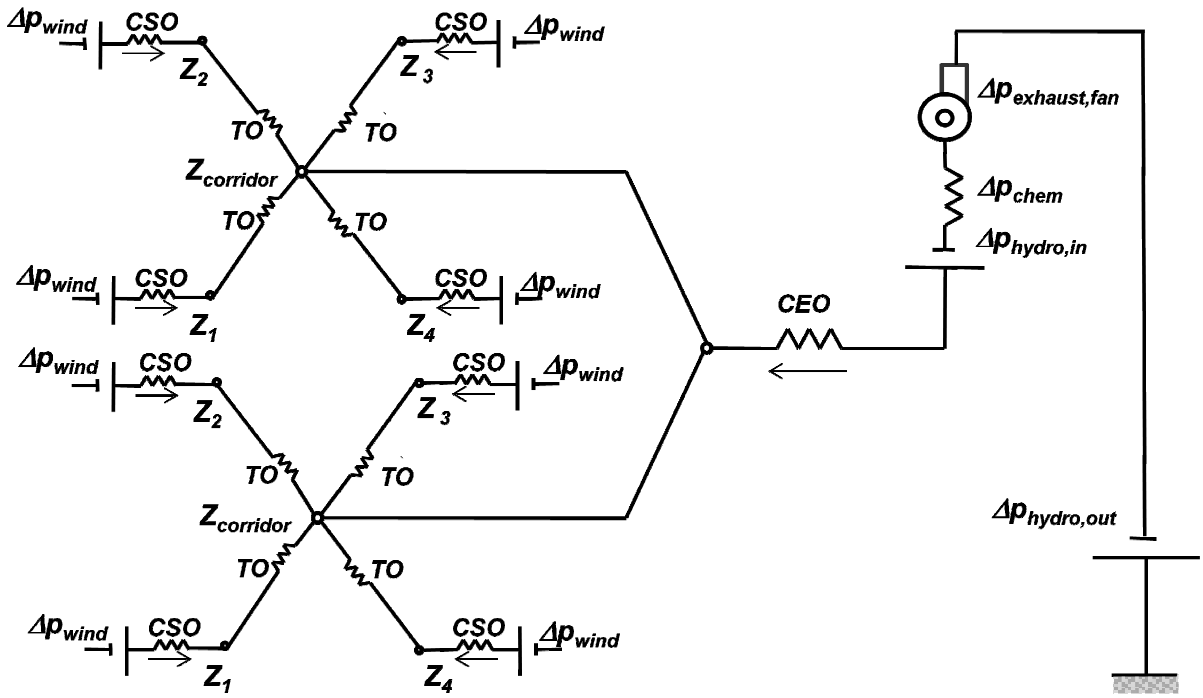



*Fig 7.8 Type C ventilation system for a 4 offices/3 wings building*

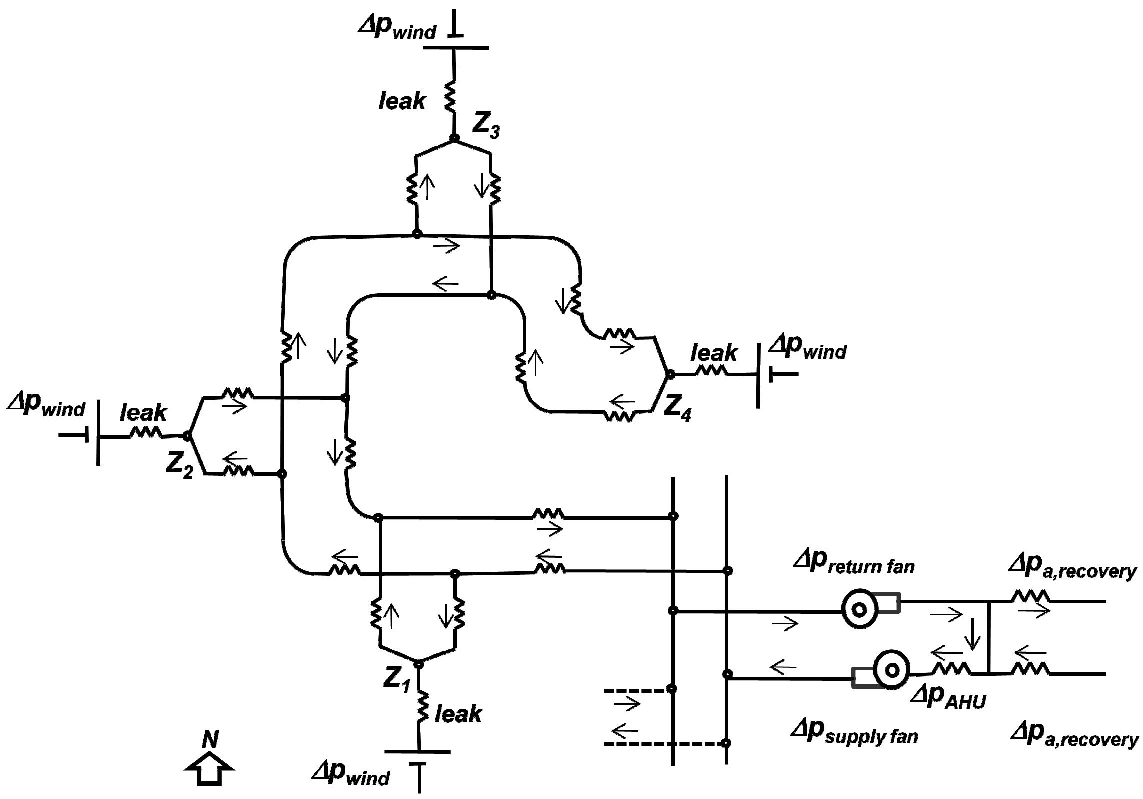



*Fig 7.9 Second type D ventilation system: layout of one building wing made of 4 offices with different orientations.*

The model can compute evolutions of indoor $CO_2$ concentration over a year, providing yearly average values for different orientations and for the different fan powered ventilation systems described here above (fig. 7.11). The model shows that all systems fulfill air quality requirements limiting the indoor $CO_2$ concentration to 1300 *ppm*.

For second type D ventilation system, the model can evaluate the effect of different strategies meant to reduce energy consumptions, while maintaining indoor air quality requirements:

- A plate heat recovery exchanger included in the Air Handling Unit, with a by-pass used either when $t_{out} > 10°C$ or when $t_{in} > 24°C$ in order to avoid overheating risks.

- $CO_2$ probes in offices, measuring the maximum indoor $CO_2$ and controlling the supply and return fans rotation speeds through the following proportional control law, with $X_{CO2,set} = 800\ ppm$ and $1/C = 500\ ppm$:

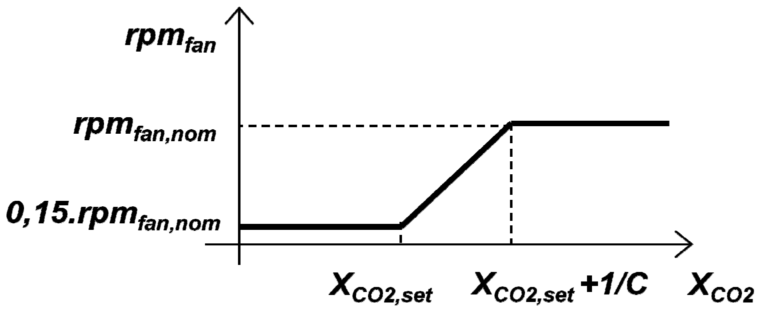



*Fig 7.9: Fan rotation speed controlled by indoor $CO_2$ probe.*

The model can verify that air quality requirements are respected when indoor $CO_2$ probes are used to control fans: yearly average indoor $CO_2$ concentrations are lower than the 1300 *ppm* threshold (fig. 7.11).

An air handling unit can complete the model. It can include a plate heat recovery exchanger and a heating coil. In a first approach, efficiencies can be considered as constant: 0.6 for heat recovery efficiency and 0.4 for heating coil efficiency (the model can also consider a variable efficiency as function of the air flow rate: see chapter 9, §9.2). Fresh air can be provided to the offices at 18°C temperature set point. Supply and return fans can be sized and modelled (annex 5, §5.3).

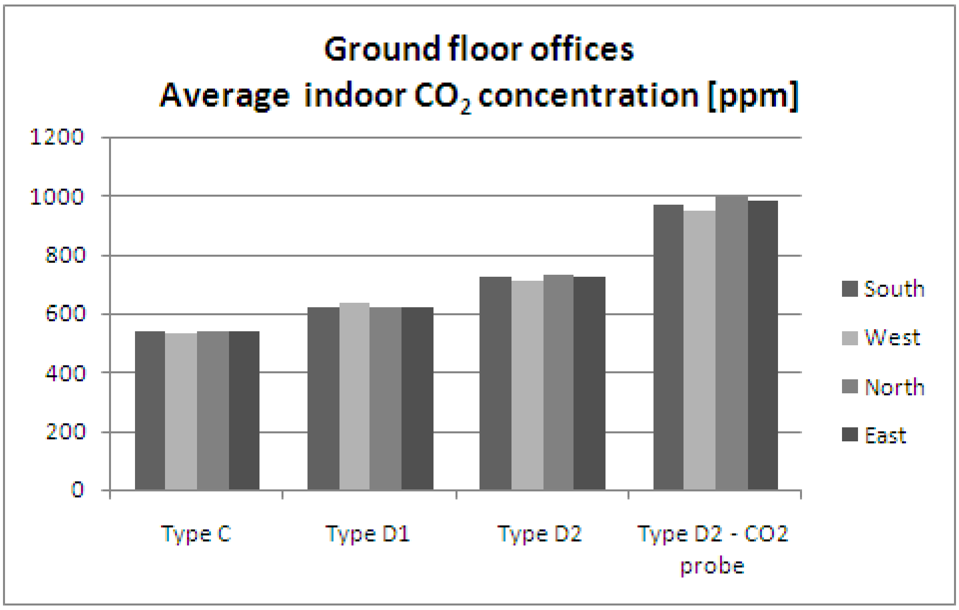
Offices can be equipped with fan coils able to provide heating and sensible cooling. Coils can be modelled as semi isothermal exchangers with constant efficiency and constant air flow rate. Fans power can be estimated at 1.5 % of the coil heating or cooling delivered power. Water supply temperature can be 70°C for heating coils and 10°C for cooling coils. Indoor temperature set points can be 22°C for heating and 25°C for cooling.

Hot water and cold water distribution pumps can be added to the model. Their manometric delivery head can be 70 *kPa*, their constant efficiency 0.6. Water flow rates can be estimated from a 10°C distribution temperature difference.

Finally, the model can handle plants: a boiler with a 0.9 constant efficiency, and an air-cooled condenser chiller for which the COP can be estimated as function of the condenser air supply temperature (supposed to be equal to the outdoor air temperature):

$$COP_{chiller} = 4.97 - 0.06063 \cdot t_{a,su,cd} - 0.0001757 \cdot t_{a,su,cd}^{2}$$

The correlation was established on a typical chiller with air-cooled condenser.



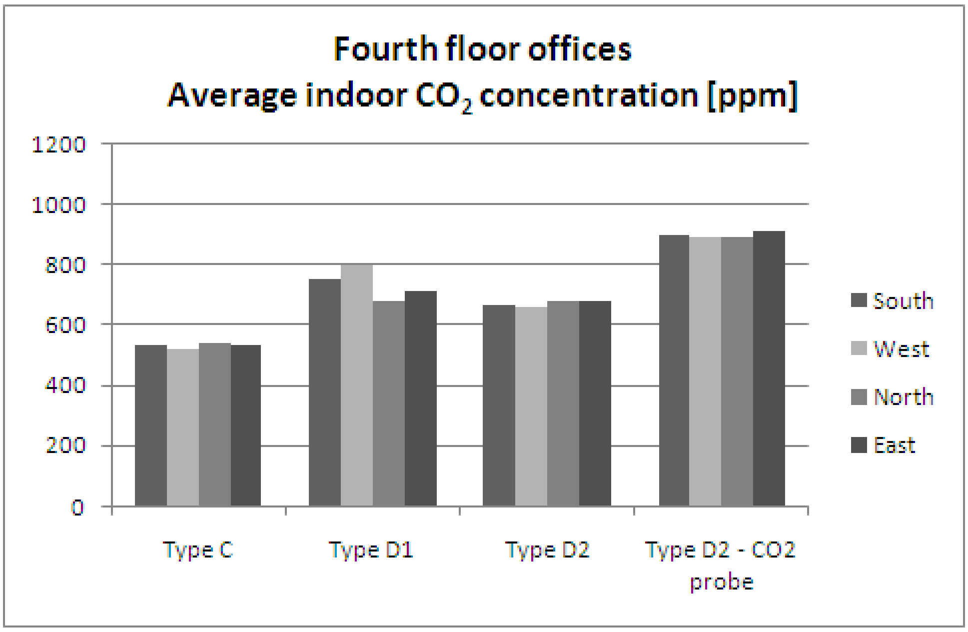



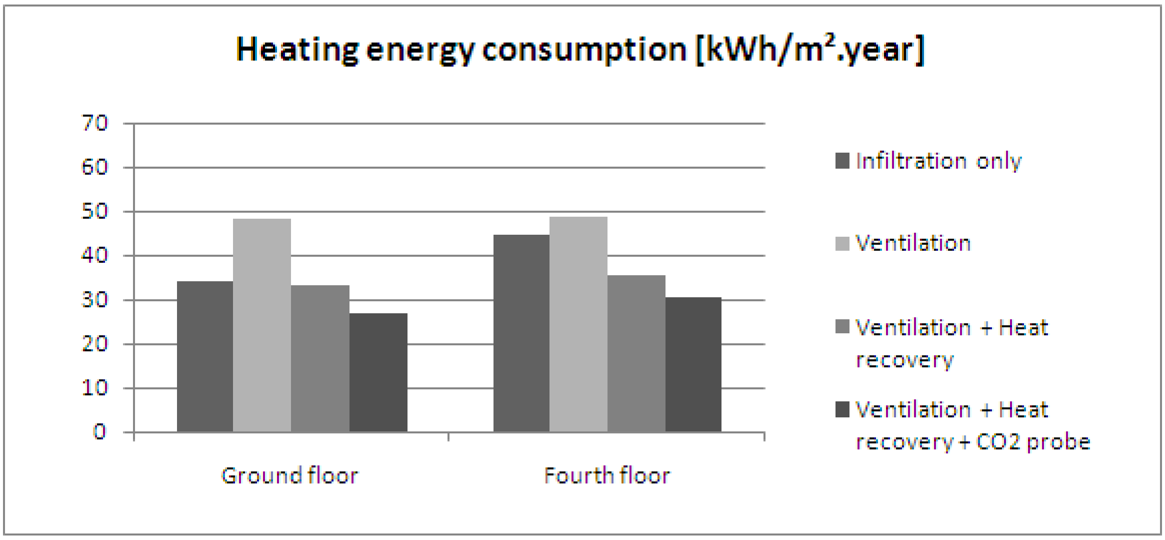

*Fig 7.11 Yearly average indoor $CO_2$ concentrations yield by fan powered ventilation for different office orientations at ground floor (up) and at fourth floor (down) for a four levels office building.*

The model can evaluate the yearly heating energy consumptions and the electricity consumptions including chiller, fans, pumps, lighting and appliances. Those consumptions can be expressed by square meter of office floor area (fig. 7.12).

The *heating energy consumption* can reach up to 50 *kWh/year.m²*. The model yields the performances associated to energy saving strategies (fig. 7.12):

- Heat recovery can save up to 30 % heating energy consumption.
- Fan control through indoor $CO_2$ probes can add up to 12 % energy saving.

The electricity consumption can average 33 *kWh/ year.m²*. Light and appliances can represent 66 % of that consumption, while chiller can account for 11 % and fans for 15 % (fig. 7.13). Chiller energy consumption may be higher, as occupancy gains due to lighting and appliances are equal to *10 W/m²*, which is a rather low estimation.



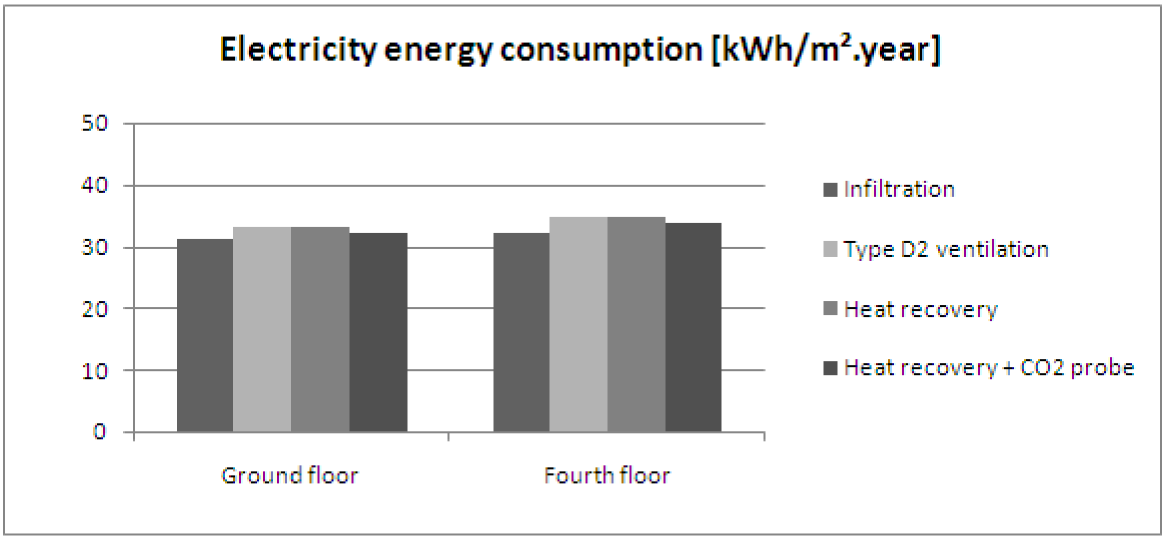




*Fig 7.12: Yearly heating energy and electricity consumptions associated to type D2 fan powered ventilation system, with different energy saving strategies.*

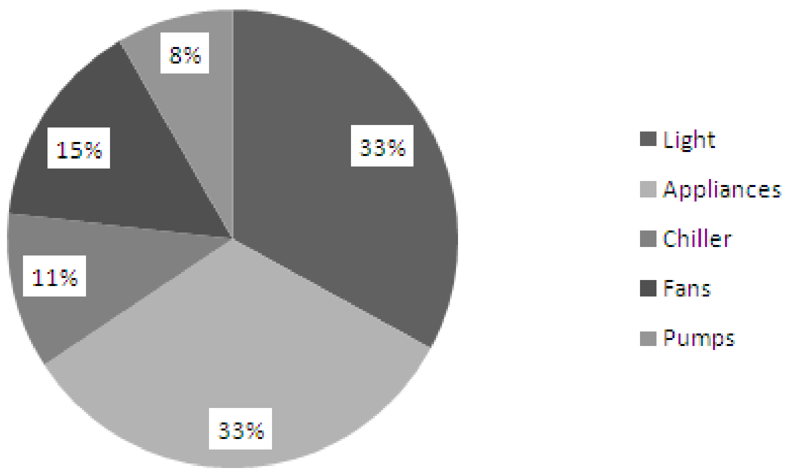



*Fig 7.13: Repartition of yearly electricity consumptions associated to type D2 fan powered ventilation system, without recovery and without $CO_2$ probes, in ground floor office.*

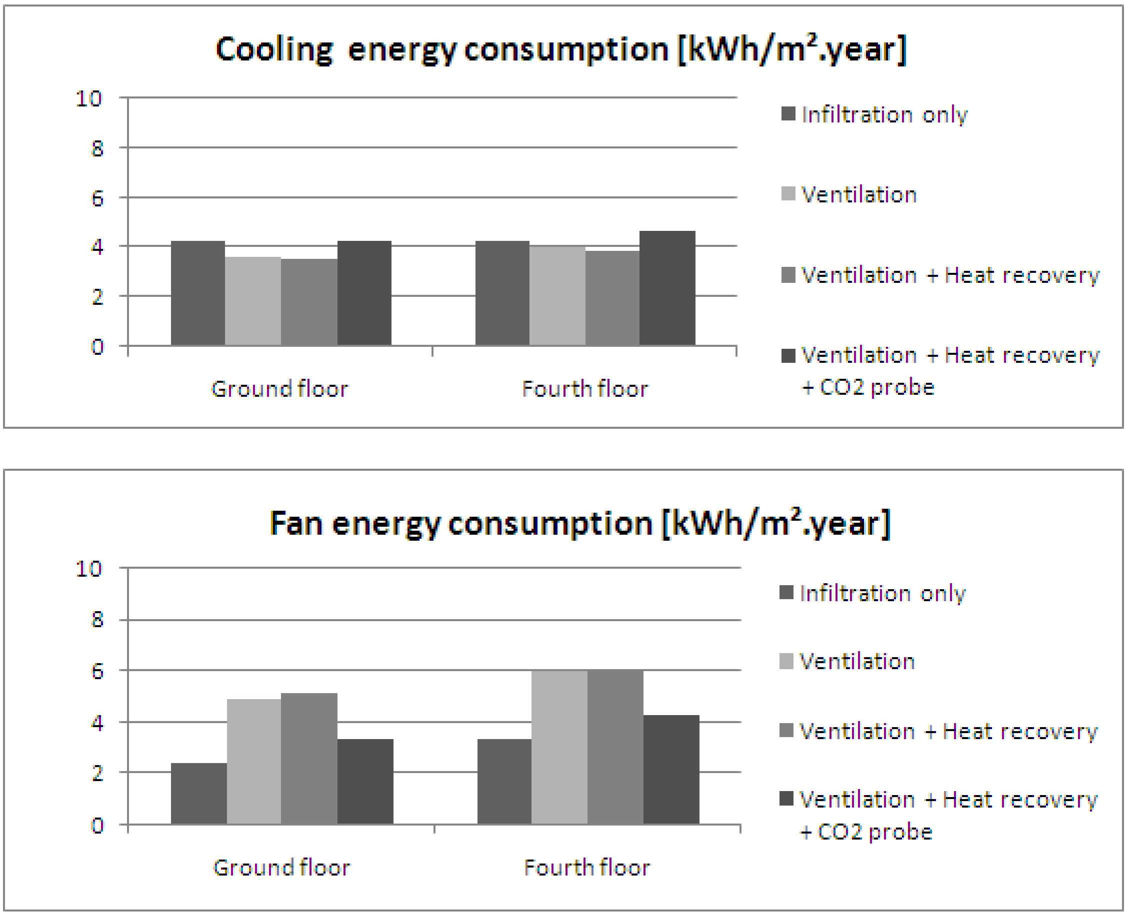



*Fig 7.14: Yearly cooling and fan energy consumption associated to second type D mechanical ventilation system, with different energy saving strategies.*

The model can provide chiller electricity consumptions as well as cooling energy demands, so that an average chiller COP, equal to 4.36, can be deduced.

The fan model can yield fan air flow rates and air delivered pressures, so that average fan efficiencies can be estimated. For ventilation without recovery and without $CO_2$ probe, the average supply fan efficiency equals 0.44 for an average 320 *Pa* total air delivered pressure. The corresponding return fan values are: 0.36 average efficiency and 176 Pa average total air delivered pressure. Higher efficiencies could be obtained by an optimised fan sizing.

The model can estimate the effect of energy saving strategies on chiller and fan electricity consumptions (fig. 7.14):

- Heat recovery doesn't affect cooling energy consumption as it isn't used as soon as $t_{out} > 10°C$, while it can increase fans energy consumption.

- Fan control through $CO_2$ probes decreases fresh air flow rates so that cooling demand increases and chiller consumption too. Anyway, fans consumption significantly decreases. Heat recovery and fan control can decrease electricity consumption by 1*kWh/ year.m²*.

## 7.3. Conclusion

The building model, as defined in chapters 2 to 6, can be used to evaluate the performance of different ventilation systems regarding *indoor air quality* during occupancy. The $CO_2$ concentration can be used as air quality indicator.

Both natural and combined natural/fan powered ventilation systems can be considered by the model and different improvement strategies can be tested to improve the indoor air quality level.

For example, in houses an offices, the model can verify the ability of natural ventilation devices to maintain a given indoor air quality level (Controlled Supply Orifices, Transfer Orifices and Exhaust ventilation shaft). Combined natural/ fan powered ventilation system can also be evaluated as well.

The impact of improvements regarding ventilation can be estimated by the model: resizing fans, opening windows, increasing the ventilation shaft height.

The model allows testing different strategies in order to reduce energy consumptions while maintaining indoor air quality requirements: heat recovery exchanger, fan control through $CO_2$ probes. The effect of those strategies on boiler, chiller, fans and pumps energy consumptions can be evaluated.

So ventilation models can be useful tools to control the air quality level in rooms. They can also provide interesting results related to summer comfort when free cooling strategies are implemented, as can be seen in the following chapter.