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Annual performance monitoring of a demand controlled ventilation system in a university library

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Abstract

Demand controlled ventilation (DCV) is an important opportunity to reduce energy requirement especially in presence of variable occupancy. An evaluation of the possible amount of the energy savings consequent this more flexible control strategy are here presented in a real application case. This refers to the case of an ancient building in Venice. A part of this building was recently transformed in a modern university library. By recording all the measured data from the supervisory system an analysis of the annual performance of the DCV system was carried on. The investigation has pointed out the possibility of remarkable energy savings without compromising internal comfort conditions.

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Keywords: demand control ventilation; smart building, energy saving, annual monitoring.

1. Introduction

Modern energy regulations have strongly reduced the heat losses of the building envelope by imposing high thermal insulation performances for the opaque surfaces and the windows. To obtain a further significant reduction of the energy requirement of the building now it is fundamental a major attention devoted to minimize the energy consumption due to ventilation also considering the general trend to increase the design air change rate for comfort exigencies. Demand controlled ventilation (DCV) is a control strategy of the ventilation rate which results recently

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very attracting as applicable to traditional plants also in the case of retrofit and therefore studied for many possible application cases [1]. DCV is highly suitable of those places where occupancy rate shows higher fluctuation such as office, meeting room, break room, school, sport facility. However its installation results more simple and cost profitable in presence of great open spaces rather than in buildings subdivided in small rooms because a fine-grained occupancy information is required in this second eventuality [2, 3].

The number of presences can be assessed in real-time indirectly by monitoring indoor CO_2 concentration variations as in buildings the occupants are normally the major source of carbon dioxide. In several guidelines and standard therefore CO_2 concentration is considered as an air quality indicator to control the outdoor ventilation rate in order to maintain reasonable IAQ without resulting an excessive energy consumption. CO_2-based DCV is most likely to be cost-effective when there are unpredictable variations in occupancy and a climate where heating and air conditioning are required for most of the year. CO_2 sensors can provide a correct average measurement if installed in the main return ducts. They can be connected to programmable controllers of dampers and inverter devices which feed the electric motors of the fans in order to modulate the speed and consequently also the air flow rate.

The real amount of the energy savings is however strong dependent from the particular application case. The occupancy profile and the characteristics of building and HVAC system deeply influence the final performance. In addition fundamental are the settings chosen for the modulating devices and their controller. An ideal control approach should keep indoor CO_2 concentration as close as possible to the CO_2 set point during occupied periods.

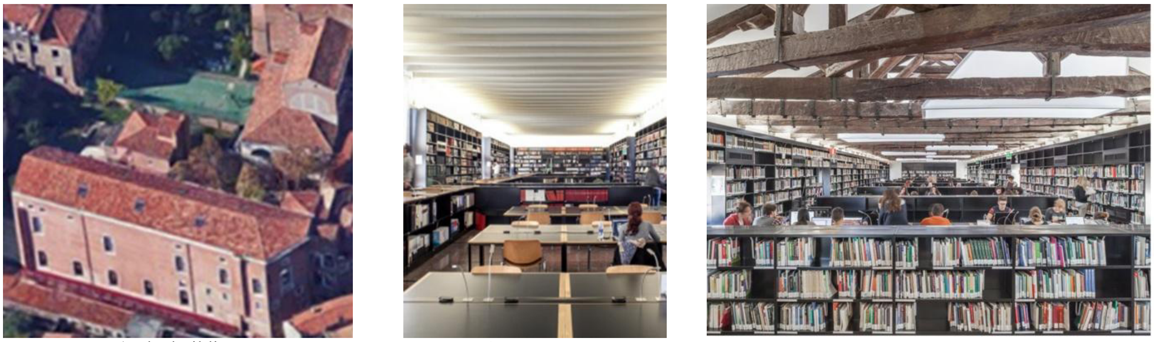
Different control algorithms can be adopted among which proportional and exponential controls are the most discussed and popular ones. Both approaches modulate the ventilation between a lower set point of indoor CO_2 and an upper set point that individuate the equilibrium concentration of CO_2 corresponding to the design ventilation rate. Exponential control is able to adjust ventilation rate more quickly to changes in CO_2 concentration by using a standard proportional-integral (PI) or proportional-integral-derivative (PID) control algorithm. However the potential energy saving is affected by the factor how fast the actual design ventilation rate achieves the ventilation required for the actual occupancy in the space. In practice, the design ventilation rate is calculated based on the assumption of a CO_2 steady state condition. But actual occupant density is frequently more variable than the action of the control speed and therefore it is possible to assist to wide oscillations of the CO_2 concentration in the rooms.

In this paper an analysis of the annual performance of a CO_2 based -DCV system is presented in comparison with a corresponding constant air volume (CAV) system in the case of a university library. The aims of this analysis were to measure the energy saving in a precise context and in addition to verify the ability to achieve the level of comfort required.

The study is based on the experimental data provided by the building management system (BMS) which also records the information necessary for energy and comfort control. DCV system adopts a typical PID algorithm rather than one of the new and more sophisticated controllers recently proposed. But the continuous monitoring of the DCV working have permitted to verify the validity of the values established for the PID parameters and optimized for this particular case study.

Nomenclature

n	actual rotor speed (rpm)
n_0	asynchronous speed (1500 rpm)
f	line frequency (Hz)
p	number of poles (4)
s	slip (-)
$\dot{V}$	air flow rate (m^3/h)
$\dot{V}_{nom}$	nominal air flow rate (m^3/h)

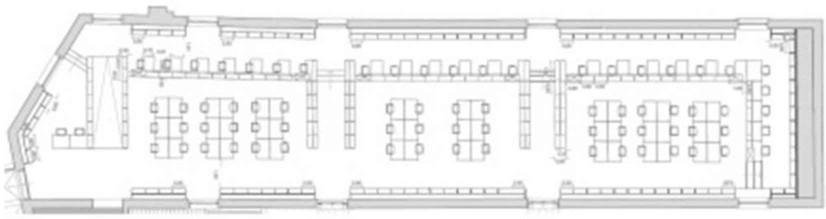


a) The building b) Room C c) Room D

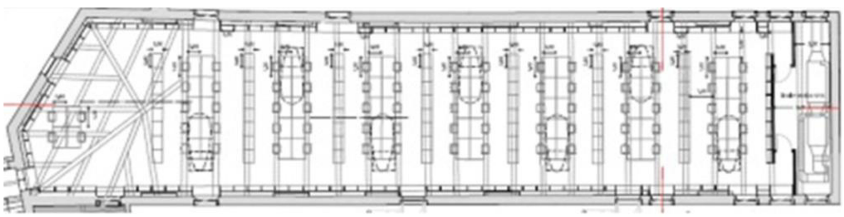
Fig. 1. Views of the building

2. The DCV system

In Fig. 1a an aerial photo of the building is reported. The building is a wing of the ancient complex of the Tolentini that was originally built as a monastery, at the end of the 16th century. Since the 50s of the last century, the property is available to the University IUAV of Venice to become its main building. The relocation of the educational activity into other buildings has allowed a refurbishment to transform the second and third floors, previously used as classrooms, into open spaces used as library reading rooms and named room C and room D. Frontal photos of the room C and D are showed respectively in Fig. 1b and Fig. 1c. Room C is at the second floor of the building and it has a surface area of 430 m^2, a flat ceiling with height of 5.2 m, total volume is 2250 m^3. Room D is at the third floor it has a surface area of 462 m^2, pitched roof with internal height at the top of 5.75 m, total volume of 2070 m^3. In Fig. 2 the plants of the two rooms are reported. A very difficult operation, however, was the adaptation of new HVAC systems inside this historic building subject to heavy monumental protection restrictions. The new plant equipment replaced the previous heating system based on mere radiators. In the reading rooms, the



Room C



Room D

Fig. 2. Plants of the two reading rooms

HVAC system consists of a primary air system with a dedicated air handling unit (AHU) and fan-coils appropriately masked in the furnishing and fed by hot or chilled water provided by a ground source heat pump (GSHP) with borehole heat exchangers installed in the adjacent garden of the company kindergarten of IUAV [4]. The fundamental characteristics of the AHU are reported in table 1.

Table 1 Characteristics of the AHU

Supply fan		
Fan	Motor	
Type: Backward curved double inlet	Type: asynchronous - number of poles: 4	
Nominal flow rate: 5000 m^3/h	Nominal power 3 kW	
Total pressure: 1199 Pa	Supply: three phase AC - 400V/50 Hz	
Nominal speed: 3528 rpm	Nominal speed: 1420 rpm	
Return fan		
Fan	Motor	
Type: Forward curved double inlet	Type: asynchronous - number of poles: 4	
Nominal flow rate: 4500 m^3/h	Nominal power 1.5 kW	
Total pressure: 509 Pa	Supply: three phase AC - 400V/50 Hz	
Nominal speed: 1337 rpm	Nominal speed: 1410 rpm	
Heat recovery system: cross flow plate heat exchanger nominal efficiency 66%		
Humidification section: with spray humidifier – nominal efficiency 85%		
	Capacity	Inlet/outlet water temperature
Heating/cooling coil	45 kW(H) - 53 kW(C)	45/40°C (H) – 7/12°C (C)
Post-heating coil	20.2 kW(H)	45/40°C (H)

In Fig. 3 performance curves of the two fan of the AHU are showed. They were used to test the corresponding values measured during the monitoring. A further air source heat pump is present to serve the rest of the Tolentini complex and the peak load of the library. In summer, partial recovery from the condensers of the machines takes

a b

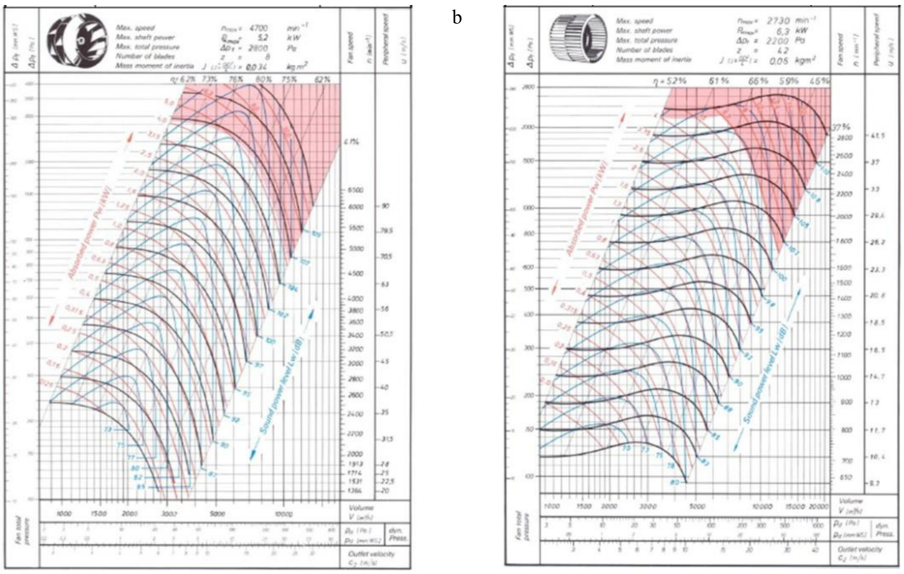


Fig. 3. Fan performance curves (a) supply fan, (b) return fan.

place by refrigerant gas desuperheating at the compressor outlet to feed the post-heating of the air handling unit. The HVAC system is completely supervised by a building management system (BMS) which allows an easy interface with the user for controls and adjustments. In addition, the BMS measures and records all the data required to assess plant performance. This registration has allowed the authors to monitor the behavior of the DCV system and to elaborate the analysis reported in this paper. Fig. 4 shows a screenshot of BMS page related to the AHU control.

3. The DCV system

As recommended by national standard UNI 10339 [5] a design ventilation rate of 7 l/s (25 m^3/h) per person has been considered. The total air flow rate of 5000 m^3/h is equally divided between the rooms considering a design occupancy of 100 persons for each one. As frequently happens in university buildings, the real crowding has high peaks, but also an appreciable variability during the activity period. It is therefore foreseen the presence of a variable ventilation rate on the basis of the real exigency.

The control system acts on air dampers installed in the return and supply ductworks and on the central AHU which has fans equipped with variable speed devices. In detail the air distribution is divided on leaving between the ducts which serve the two rooms. The same for the return ducts. In this way it is possible an independent DCV based on the measure of the CO_2 by the infrared sensors installed in the last part of the return air duct of each room. Air dumpers on supply and return are synchronized and they modulate the air flow rate acting between a minimum aperture section up to the total aperture. A PID system controls the movement of the air dampers of each room basing on a set point equal to 850 ppm with a switching differential of 100 ppm. In other words the modulating dampers completely open at 900 ppm and close at 800 ppm. However a minimum aperture share equal to 20% of the total one is always foreseen in order to ensure a base ventilation during the working hours.

A good air quality is generally associated to a limit value of 1000 ppm. In particular ASHRAE [6] indicates under 1000 ppm the concentration which ensures a percentage superior than 80% of the people which do not express dissatisfaction. However recently ASHRAE has clearly clarifies that the CO_2 level of 1000 ppm is a guideline for comfort acceptability rather than a ceiling value for IAQ. In many standards, for example DIN [7] and SIA [8], the maximum acceptable threshold in offices and schools is fixed at 1500 ppm. British authority prescribes to limit the mean level over a school day to 1500 ppm [9]. In this case study the design setpoint has been decided in order to maintain CO_2 concentration under 1500 ppm and preferably near 1000 ppm.

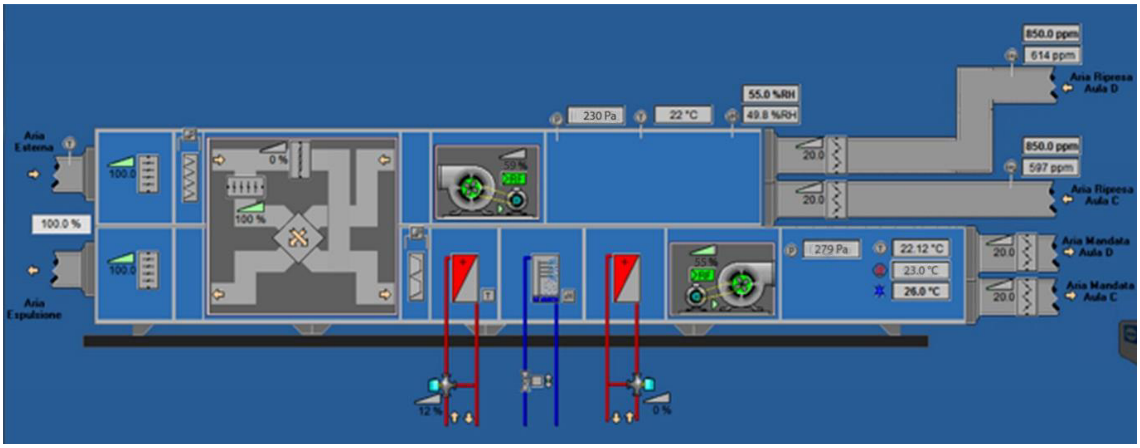


Fig. 4. A screenshot of the AHU in the BMS

The variation of the flow area in the motorised dampers involves a modification of the air rate consequent the variation of the flow resistance and then of the pressure drop. But in order to avoid unnecessary fan energy use and unbalancing between the ductworks of the two rooms, the static pressures downstream the supply fan and upstream the return fan are maintained constant. In fact in both these positions a static pressure sensor is installed and connected to the speed controller of the fan. This variable speed drive (VSD) is integrated in the BMS and acts on the inverter devices which feed the fan electric motors. The setpoints of the static pressure are equals to the values measured by the sensors when the air dampers are fully open: 300 Pa downstream the supply fan and -250 Pa upstream the return fan. For each fan the PID algorithm operates in a range of 50 Pa around the setpoint and gives back a control signal between 0% and 100% (DC 0÷10 V) which is submitted to the inverter controller and it corresponds to a ramp of the electric supply frequency between 20 Hz (0%) and 50 Hz (100%).

In fact the best mode to vary the speed of an induction motor is by varying the supply frequency. The frequency inverter transforms the line voltage, with constant amplitude and frequency, into a voltage with variable amplitude and frequency in order to ensure a constant electromagnetic torque. In this way the speed of the rotating field and consequently the mechanical speed of the motor is changed by varying the frequency of the supply voltage.

4. The monitoring

The recording of the measured data takes place every 10 minutes. As regards measurement accuracy, NTC temperature sensors are normally used, with ±0.8 K accuracy, capacity humidity sensors with ±3% accuracy and electric power meters with 1% of full scale accuracy. Among the monitoring data, for this analysis relative humidity and air temperature upstream and downstream each coil, recovery unit and outdoor have been used to calculate the enthalpy variation of the handled air in the various components of the AHU. Carbon dioxide concentrations are measured by non dispersive infrared (NDIR) sensors (accuracy ±50 ppm+2%). In addition the monitoring provides the value of all the parameters which describe VSD operation mode: the opening share of each air damper, the static pressure values and the percentage signals transmitted to the inverters of the two fans. The values of these signals permit to know the supply frequencies and consequently the effective speeds of the two fan motors. In fact the relationship between the motor speed, the supply frequency, the number of poles and the slip of an induction motor is given by the following equations:

$$n = \frac{120 \cdot f \cdot (1-s)}{P} \tag{1}$$

$$s = \frac{n_0 - n_{nom}}{n_{nom}} \tag{2}$$

Owing to the costant transmission ratio based on a belt and pulley system, the fan speed varies with the same ratio of the electric motor. Applying the affinity laws of fans, with the impeller diameter held constant, the new operating point is characterized by a flow rate $\dot{V}$ of the fan proportional to the shaft speed as indicated in Eq. 3:

$$\dot{V} = V_{nom} \cdot \left(\frac{n}{n_{nom}} \right) \tag{3}$$

The heating/cooling powers supplied to the AHU can be calculated at every step of the monitoring by multiplying the air flow rate for the enthalpy gap in each coil. The electric absorptions of the fans are directly measured.

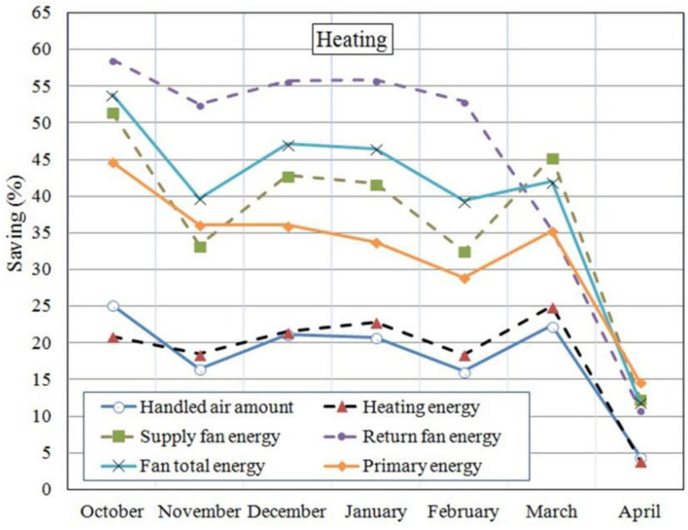


Fig.5. Monthly percentage savings by DCV in the heating season.

5. The results

For this analysis the monitoring period from October 2014 to September 2015 has been considered. Air conditioning was working from the beginning of June to the end of September, heating season is from the 15th of October to the 15th of April as fixed by national regulation for this climatic area. The library was closed two weeks for Christmas holidays and the same in August. Fig. 5 shows the percentage savings of handled air amount and energy consumptions obtained with DCV for each month of the heating season. These savings are referred to the corresponding values in the case of CAV during the opening hours. In addition to the electric needs of the fans, primary energy consumption of the AHU considers the quota of electric energy absorbed by the heat pumps to feed the heating/cooling coils calculated by the measured COP/EER [4].

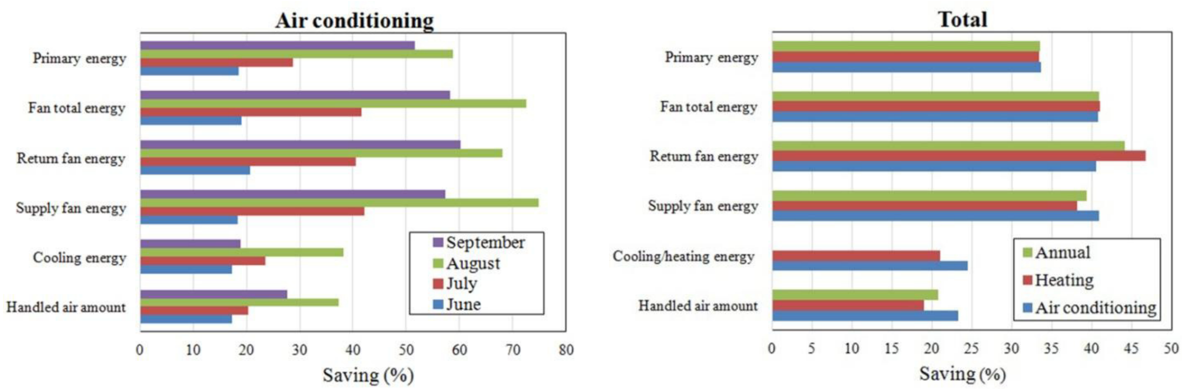


Fig. 6. Monthly percentage savings by DCV in the air conditioning period and the total ones

Fig. 6 shows the analogous monthly savings for the air conditioning period and the total ones. The savings result very significant, but sometimes strong different in the various month owing to the extreme variability of the occupancy. In the summer period primary energy saving is over 50% in August and September as consequence of scarce presences. As example in Fig. 7 the trends of measured CO_2 concentrations in the two rooms are reported for the second half of January. In the week-end the library is closed. It is evident the higher occupancy in room C.

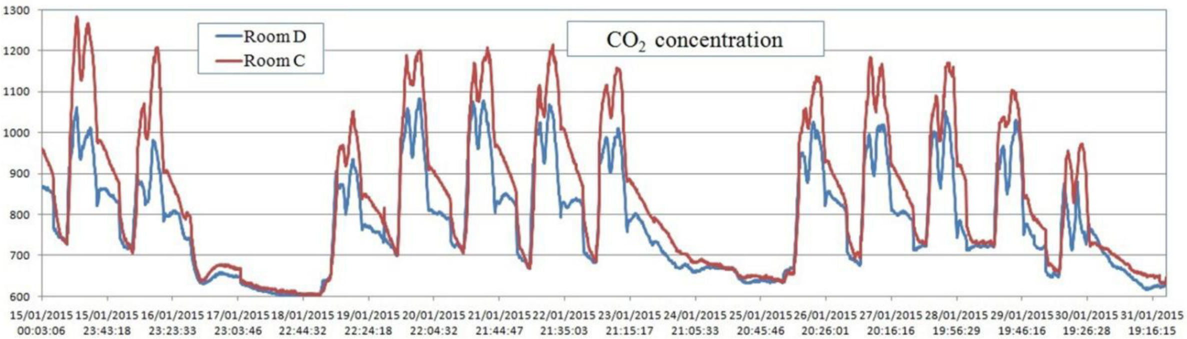


Fig. 7. Trends of the measured CO_2 concentration in rooms from 15th to 31th of January

However its variability during the opening days makes it difficult to predict. The CO_2 concentration always remains under 1500 ppm, but some remarkable peaks justify that the full ventilation setpoint is at 850 ppm to anticipate the demand requirement. The values during the nights and weekends are not realistic because the ventilation system is stopped when the library is closed and the CO_2 sensors are installed inside the return ducts. However the high values observed in the morning starting have suggested to begin the base ventilation two hours before the opening of the library.

6. Conclusions

The DCV system based on the use of NDIR sensors to drive the fan speed has allowed a reduction of the air handled amount equal to 21% in the monitored year. Consequently the total primary energy saving was 33%. The most significant saving quota refers to the heat pump consumption to supply the heat and cold necessary for the air treatment in the coils rather than the electric absorption of the fans. DCV technology confirms to be a valid tool to reach the goal of nZEB in building characterized by variable occupancy. The presence of BMS with user friendly interface permits the optimization of the control settings which must be adapted to the characteristic of each building.

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