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Ventilation rates and control strategies in VAV systems: effects on energy consumption in office buildings in consideration of the EPBD and EN 16798 standards

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Abstract. The study extends previous work on determining outdoor air flow rates in non-residential buildings, shifting the focus from simply complying with indoor air quality requirements to assessing the energy effects on systems. The design flow rates obtained using different calculation methods, including EN 16798 part 1 and the Italian National Annex, are taken as a basis for analysing the energy performance of a variable flow ventilation system, in line with the objectives of EPBD 2024 on decarbonisation and in-door environmental quality. The assessment applies a dynamic simulation model of a variable flow primary air system, representative of a typical office building. The model is applied to a matrix of scenarios that combines the flow rate calculation method, air quality category, air diffusion strategy in the environment and fan control logic. The results quantify the energy impact associated with different regulations and control methods, highlighting the trade-offs between energy consumption, indoor air quality and thermal comfort in occupied spaces, and providing selection criteria for the design and refurbishment of ventilation systems in office buildings; in particular, it is noted that the two control methods investigated enable annual savings in primary energy of 51% and 53%.

1 Introduction

The present analysis is based on the extension of research [1] previously conducted on the functioning of a primary air system with variable flow rate and, in particular, on the comparison between different types of control strategies.
The system analysed serves a meeting room with occupancy profiles compatible with typical office work situations. The study encompasses different office types with varying workstation counts, meeting rooms used sporadically, and other accessory spaces. The occupancy schedule for different spaces was defined to highlight occupancy variation during normal working hours, with a maximum occupancy peak of 35 people which, considering also the floor area and building type (non-polluting building), results in a nominal flow rate

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under design conditions equal to 1,116 $m^3 h^{-1}$. Figure 1 shows the hourly occupancy profile on working days.

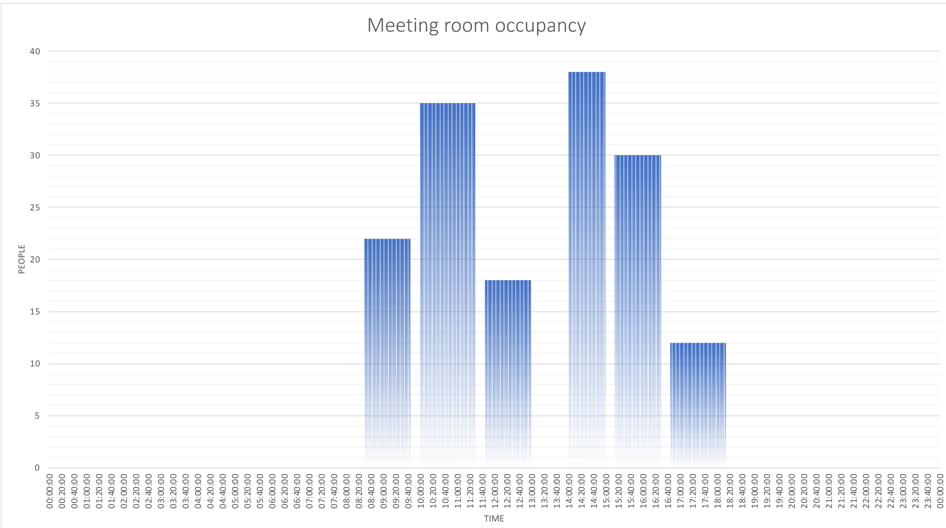

Fig. 1. Hourly occupancy profile.

The study identified two main flow rate control strategies: the first strategy (referred to as STD) is of open-loop type and refers to determining the flow rate to be sup-plied at each instant in the space based on a survey (or an estimate derived from occupancy schedules) of the number of people present and the real-time application of the sizing methods reported in the EN16798-1 Standard [2]. The second strategy is based on feedback regulation through the presence of air quality sensors positioned inside individual spaces and proportional (PI) control logic. The simulations performed report the relationship between CO_2 concentration values in the space and the trends of supplied air flow rates. The time evolution of CO_2 concentration is deter-mined at each step by considering the initial concentration, the supplied air flow rate, the CO_2 concentration in the outside air, and aspects related to the buffer effect due to the volume of the served spaces. The two control logics considered were compared with the reference situation typical for this type of system, which is the constant air volume (CAV) system.

From the perspective of implementing the control logics, it is important to emphasize that both approaches suffer from intrinsic limitations due to, for example, the maximum admissible flow rate reduction at the diffusers without incurring a drastic degradation of the ability to distribute air correctly in the space. An additional potential difficulty consists in limitations that many control programmers tend to impose in terms of minimum fan speed. To highlight the potential energy benefits associated with maximum flow reduction, these limitations were not applied in the simulations performed.

The comparison between the investigated control types enables highlighting how a PI-type logic with environmental sensors is capable of adapting system behaviour to actual needs and reveals a conceptual limitation related to the models used to calculate the flow rate reported in Standards. It is useful to consider that when occupancy levels are zero, the minimum flow rate provided to address contaminants derived from the building provides a minimum flow rate value to be supplied. This flow rate level is considered sufficient to address the building's pollutant loads. When occupancy ceases to be null, there exists a minimum number of people whose contamination can be eliminated by this same minimum flow rate, since contaminants from people and those from the environment can be diluted by the same air. However,

Method 1 of EN 16798-1 provides that, from the first person entering the room, the flow rate should increase. Conversely, feedback control based on CO_2 value will tend not to modify the flow rate until increased occupancy impacts the CO_2 level.
Figure 2 shows the flow rate trends in the two cases, applying the two regulation modalities. The third case represented is the one assumed as reference and consists of constant volume control with a value equal to the sizing value.

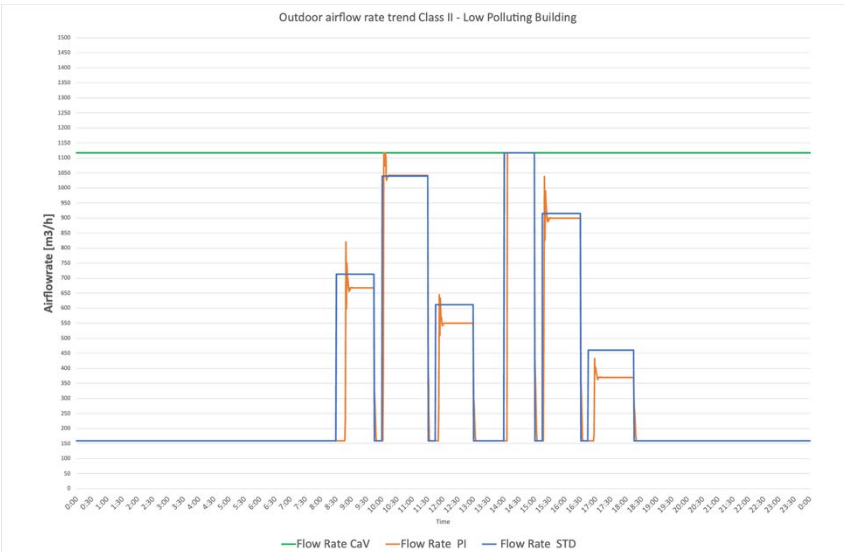

Fig. 2. Airflow profiles for the three control strategies applied to a system sized according to Class II – Low Polluting Building.

The previous analysis stopped at the comparison of flow rates and, beyond the limits of maximum and minimum flow rate, it is verified that feedback control takes into account accumulation and buffer aspects of real spaces, generally delaying the effects of flow rate adjustment compared to the reduction in the number of people. Such evaluations are of limited interest if not accompanied by an energy analysis. The continuation of this analysis therefore focuses on evaluating the energy consequences and will ac-count for electrical consumption of the fans and auxiliary consumption (pumps, valves) as well as thermal consumption related to the variation in the ventilation load. All the measured consumption figures were then converted into electricity consumption in or-der to be able to aggregate the results.

2 Methods

2.1 Fan power consumption

The estimation of fan consumption was performed by assuming variable flow rate control set on constant static pressure control, completed with VAV terminal cassettes controlled according to the considered logic. The sizing of the Air Handling Unit (AHU) was performed using the certified Eurovent sizing software made available by a manufacturer. The unit was selected based on the maximum flow rate value of 1116 $m^3 h^{-1}$. The air treatment section is characterized functionally based on the capabilities of sensible heat recovery, heating, cooling with dehumidification, adiabatic humidification, post-heating, and filtration. Internal losses from the air handling unit were calculated in accordance with European Regulation EU 1253/2014 [3], which defines the maximum specific power consumption (Specific Fan Power) of the supply and return fans (expressed in $Wm^{-3}s^{-1}$) permitted for an air handling unit on the European Union market. The size and configuration of the AHU thus obtained were then maintained by modifying the treated flow rate and accounting for the variation of the corresponding operating point of the supply and return fans. The software recalculates the internal losses and fan efficiency, providing for each partially loaded condition the effective estimated consumption.
The pressure values for the distribution system (supply and return) were assumed constant and equal to 300 Pa. The variation of cumulative electrical power consumption for both the supply and return fans was analysed for a series of reference conditions starting from the nominal flow rate value down to the value corresponding to minimum occupancy. The calculation was then performed through least-squares regression after applying a log-log transformation.

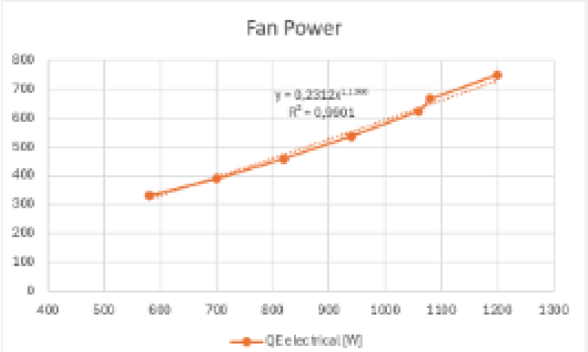

Fig. 3. Power consumption curve for fans.

2.2 Thermal Energy Consumption

The quantification of thermal requirements was based essentially on the consideration of the ventilation load, that is, the energy effort required to bring the conditions of ex-ternal air downstream of the heat exchanger to the environmental conditions.
The situation referred to the cooling regime involves the activation of components necessary for dehumidification and cooling. The power considered is the refrigeration pow-er for the months of June, July, and August, while the power for the heating period was considered from October 15 to April 15. The periods between April 15 and May 31 and between

September 1 and October 15 are considered periods in which treatment of out-side air is not necessary and, for the sake of simplicity, it has been decided to refer to these periods as '*free cooling periods*'.
For each month or part of month considered, an analysis was performed based on the trend of hourly climatic condition records from the last six years for the city of Milano (2020-2025, and a typical day was defined that summarizes the average trend of each considered month. For the months of April and October, the average was calculated considering only the half of the month in which thermal and hygrometric treatment is considered to be operational.
In the summer case, account was taken of the dehumidification process which requires reducing the temperature well beyond the set point temperature to dehumidify the air. The treatment performed by a cooling coil was calculated to bring the humidity ratio of the inlet air to a value equal to that desired (environmental set point) with outlet conditions reasonably close to saturation (90%). The treatment performed generally requires post-heating action; however, for the simulation, the consumption in the summer regime of post-heating is not considered in light of the fact that, in the presence of water/water chillers or more generally polivalent units, the possibility of recovering this power through condensation heat is applied. The calculation also considered the presence of a sensible heat exchanger with efficiency equal to 73%.
The estimate of thermal power required in the winter regime is based on the assumption that the primary air supplied is supplied at the same temperature and humidity required in the space. Consequently, the power necessary to heat and humidity the air is evaluated through an adiabatic humidification system net of the contribution from the heat ex-changer (sensible only).

2.3 Free cooling.

In the periods mentioned above in which air treatment is not necessary and the combi-nation between external climatic conditions and indoor temperature setpoints leads to the shutdown of hot and cold generators, the fan power consumption was nevertheless accounted for.

2.4 Primary Energy Assessment

The conversion of electrical consumption into primary energy was performed by applying the procedure prescribed by UNI/TS 11300 Parts 2, 3, 4 and 5 [4,5,6,7], adopting the conversion factors from Ministerial Decree (DM) of 26 June 2015 [8] and subsequent amendments for the energy carrier «grid electricity»: $f_{p,nren}$ = 1.95 and $f_{p,ren}$ = 0.47 ($f_{p,tot}$ = 2.42). The calculation accounts for the specific features related to the use of a reversible air-to-water heat pump as the generator for both heating and cooling, with a hydronic circuit connecting to the AHU.

2.4.1 Plant Subsystem Efficiencies

The thermal power at the generator Q_{gn} was determined from the thermal power re-quired by the AHU Q_{UTA} through the efficiencies of the emission η_e, control η_{rg} and water distribution η_{dw}, according to formulas (3)–(5) of UNI/TS 11300-3:

$$Q_{gn}(h) = Q_{UTA}(h) / (\eta_{rg} \times \eta_{dw}) \tag{1}$$

The adopted values are: η_{rg} = 0.97 (control, modulating zone ±1°C, Table 7); $\eta_{dw,C}$ = 0.98 (chilled water distribution, Table A.16); $\eta_{dw,H}$ = 0.95 (hot water distribution, 11300-2). The overall losses amount to 11.8% in heating and 8.4% in cooling.

2.4.2 Heat Pump COP in Heating Mode

The full-load COP was determined by interpolation of the exergetic efficiency η_{II} according to the method prescribed by UNI/TS 11300-4, clause 9.4.3.2 (formulas 43–48). For each manufacturer-declared operating point, the following is computed:

$$\eta_{II} = COP \times (Tc - Tf) / (Tc + 273{,}15) \tag{2}$$

where T_c = 45 °C (hot sink temperature) and $T_f = T_{ext}$ (cold source temperature). η_{II} is linearly interpolated in T_f and the full-load COP is reconstructed:

$$COP_{full}(h) = \eta_{II,interp} \times (Tc + 273{,}15) / (Tc - T_{ext}(h)) \tag{3}$$

For inverter-driven units, UNI/TS 11300-4 (§9.4.4.2, clause 3) prescribes a correction factor f_{CR} = 1 down to the minimum modulation ratio CR_{min} (= 0.30). Below this threshold the unit cycles on/off and formula (57) applies:

$$COP_{eff} = COP_{full} \times CR / ((1 - Cc) + CR \times Cc) \tag{4}$$

with C_c = 0.90 (default value for air-to-water units). The constraint $Q_{aero} \geq 0$ was imposed to prevent negative aerothermal energy values during hours with very low CR.

2.4.3 Heat Pump EER in Cooling Mode

The mean generator efficiency in cooling mode η_{mm} was determined according to formula (15) of UNI/TS 11300-3, by interpolating the EER from the manufacturer-declared 4-point curve (EN 14825) [9] and applying the correction factor κ1 from Tables C.1–C.4 of Appendix C for the actual temperature conditions:

$$\eta_{mm}(h) = EER(F) \times \kappa_1(F, T_{ext}) \times \kappa_2 \ldots \kappa_7 \tag{5}$$

where $F = Q_{gn,c}/\Phi_{n,C}$ is the load factor and the factors $\kappa_2 \ldots \kappa_7$ = 1.00 for standard conditions (monobloc unit, without glycol).

2.4.4 Condensation Heat Recovery

In summer operation, the heat rejected at the condenser $Q_{cond} = Q_{gn,c} + W_{HP,c}$ was recovered to supply the air reheat coil, thereby eliminating the additional electrical consumption of the heat pump in heating mode. This system design solution, consistent with the use of polyvalent units or dedicated heat exchangers on the condensing circuit, enables $W_{HP,post}$ to be zeroed during the summer months.

2.5 Fan Power

The electrical power of the supply and return fans was determined on an hourly basis as a function of the actual airflow rate for each control scenario, starting from the fan operating point calculated by the certified sizing software. The values range from a minimum of 0.07 kW (minimum airflow, PI and Std scenarios during night-time hours) to 0.67 kW (maximum airflow, CAV scenario), with a ratio of approximately 1:10 reflecting the fan affinity laws.

2.6 Primary Energy Conversion

The total hourly electrical energy includes the heat pump, the hydronic circulation pump and the AHU fans:

$$W_{el}(h) = W_{HP}(h) + P_{pump} + P_{fan}(h) \quad (6)$$

The conversion into primary energy according to UNI/TS 11300-5 (formulas 12–14) is performed by applying the factors from DM 26/06/2015 and subsequent amendments:

$$EP_{,nren} = \Sigma\, W_{el}(h) \times 1.95 \qquad (7)$$

For heating, the on-site renewable aerothermal energy is additionally accounted for (UNI/TS 11300-4, §9.3):

$$Q_{aero}(h) = max(0,\, Q_{gn}(h) - W_{HP}(h)) \qquad (8)$$

$$EP_{,ren} = \Sigma\, W_{el}(h) \times 0.47 + \Sigma\, Q_{aero}(h) \qquad (9)$$

For cooling, the aerothermal component is not accounted for (the condensation heat is rejected to the outdoor environment) and the renewable EP is given by the electrical component only. During the intermediate months (April, May, September, October) in which the thermal load is governed by outdoor conditions and the heat pump is not active, the primary energy consists solely of the fan contribution.
The total monthly primary energy is obtained by multiplying the daily values by the number of days in the month, and the annual value from the sum of the 12 months:

$$E_{P,tot} = \Sigma\ m{=}1..12\ E_{P,tot,m} = \Sigma\ m\ (E_{P,nren,m} + E_{P,ren,m}) \quad (10)$$

3 Results and conclusions

The results of the analysis are presented in the form of bar charts. Contributions in terms of estimated primary energy are shown in different colours, referring to the genera-tor (HP), to fan consumption, and to the system of auxiliaries and pumping of the water systems serving the air treatment section. Figure 4 shows the reference values, month by month, referring to constant volume (CAV) control. Figure 5 shows the values referring to control based on the Standard model, and finally Figure 6 shows the values referring to feedback control (PI).

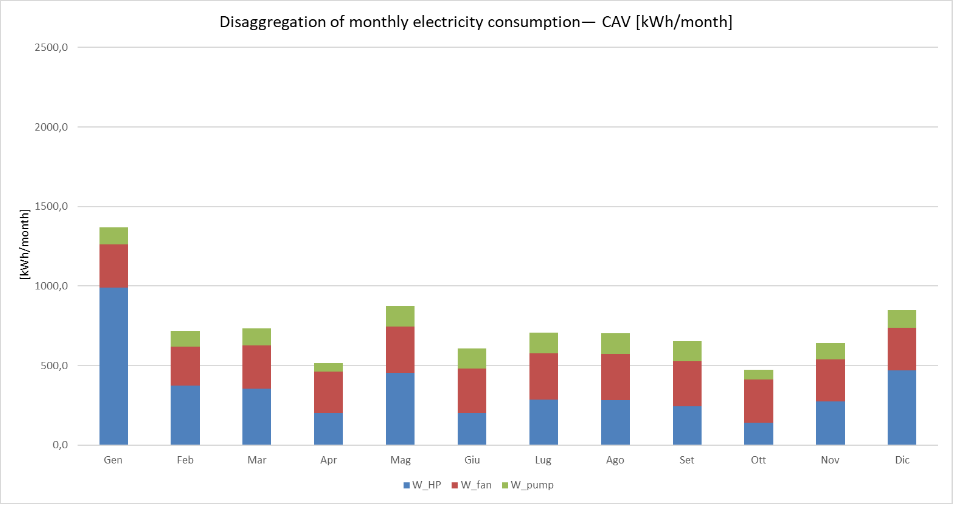

Fig. 4. Primary energy distribution - Constant Air Volume (CAV) system.

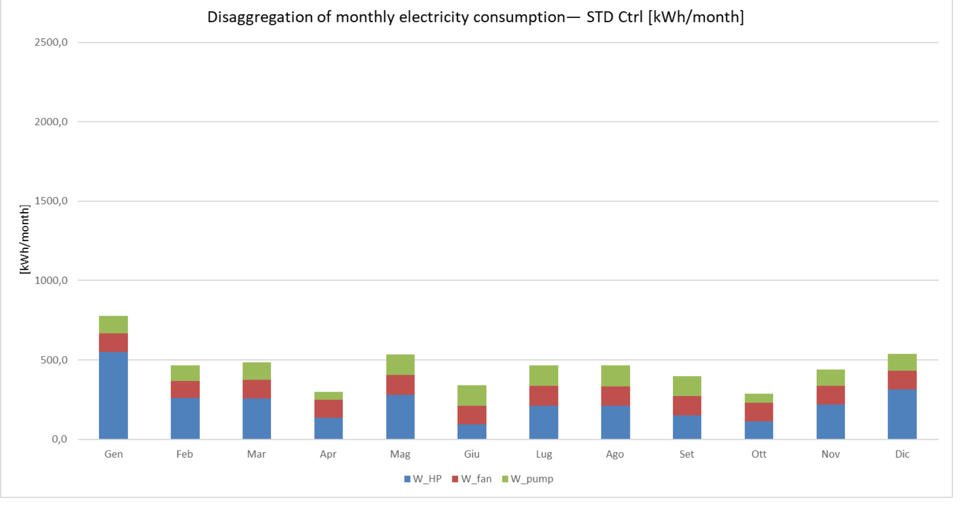

Fig. 5. Primary energy distribution - Standard-based control (STD) system.

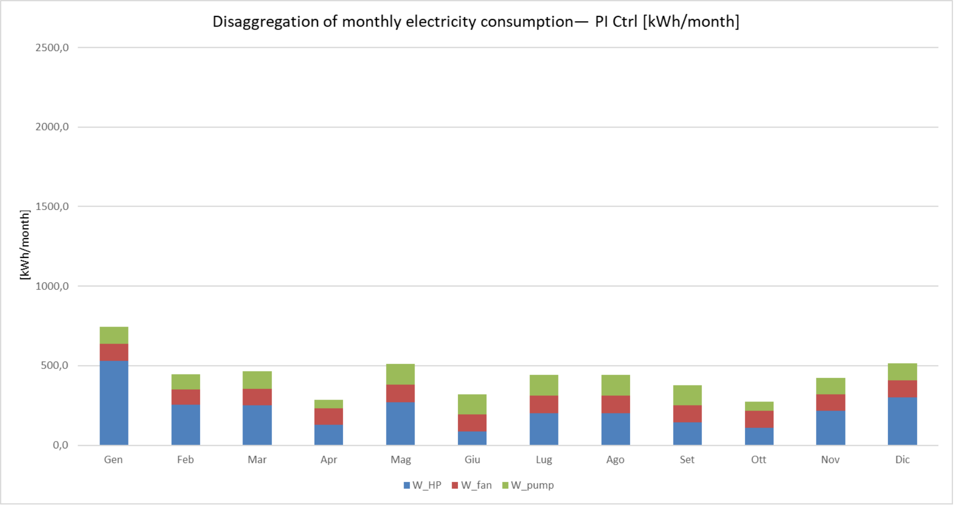

Fig. 6. Primary energy distribution - PI closed loop control system.

The direct comparison between primary energy consumption values for the three cases is illustrated in Figure 7:

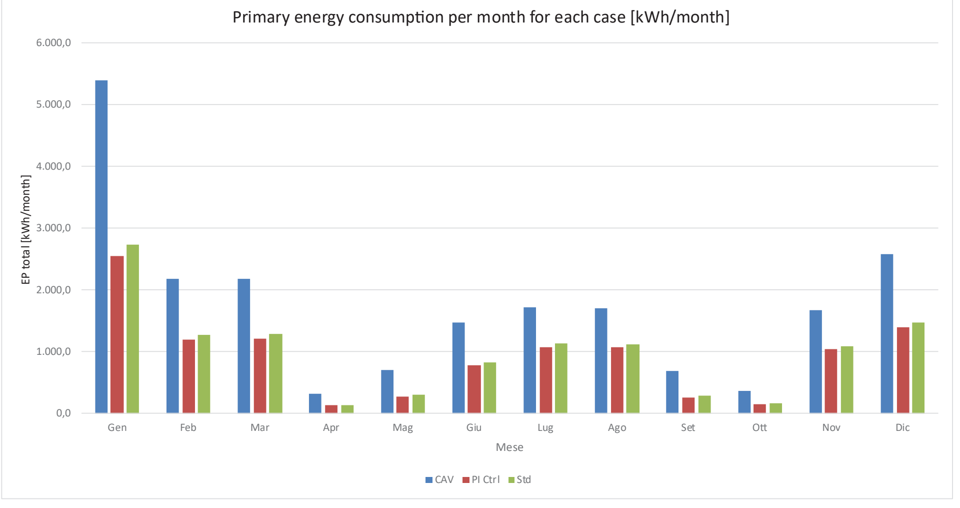

Fig. 7. Annual primary energy comparison between control strategies.

The comparison between values was also performed numerically using simulation results. With respect to constant volume control, savings were evaluated in percentage terms and refer to the overall estimated electrical consumption value. The adoption of a control logic based on the STD model results in an annual saving equal to 2,986 kWh (40%). The adoption of a control logic with feedback IAQ sensors results in an annual saving equal to 3,210 kWh (43 %).
The detail in terms of electricity consumption and savings on a month-by-month basis is set out in Table 1.

Table 1. Electricity consumption and monthly savings.

Total electricity consumption by scenario [kWh]				**% savings compared to CAV**	
Month	**CAV**	**PI Ctrl**	**Std**	**PI Ctrl**	**Std**
Jan	1,369	745	777	46%	43%
Feb	718	447	466	38%	35%
Mar	734	464	483	37%	34%
Apr	262	103	114	61%	56%
May	292	111	124	62%	58%
Jun	609	319	339	48%	44%
Jul	707	444	467	37%	34%
Aug	704	441	464	37%	34%
Sep	283	108	120	62%	58%
Oct	271	106	118	61%	56%
Nov	642	424	440	34%	31%
Dec	847	516	540	39%	36%
TOTAL	**7,438**	**4,228**	**4,452**	**43%**	**40%**

A closer inspection of the monthly breakdown reveals a marked seasonal dependence of the achievable savings. The highest percentage reductions occur during the intermediate months (April, May, September and October), reaching 61–62% with PI control and 56–58% with the STD strategy. In these periods the thermal load is modest and the primary energy consumption is largely attributable to the fans; consequently, the cubic dependence of fan power on airflow rate, governed by the fan affinity laws, amplifies the benefit of any flow rate reduction. The 1:10 ratio between minimum and maxi-mum absorbed power (0.07 kW vs 0.67 kW) demonstrates that even modest reductions in ventilation rate translate into disproportionately large savings in electrical consumption. During the heating season (November–March, plus the partial months of October and April) savings range from 34% to 46% for PI control and from 31% to 43% for STD, with the highest values observed in January, when both the ventilation thermal load and the fan consumption are at their peak. In the cooling season (June–August) savings range from 37% to 48% (PI) and from 34% to 44% (STD), with the additional complexity of latent load treatment and post-heating partially offset by the condensation heat recovery strategy adopted. It is worth noting that the simulation assumes a daily operating schedule of 14 hours (07:00–20:00), consistent with typical office occupancy; ex-tending the operating period would increase the absolute consumption values while potentially raising the relative savings, since the additional hours would fall during periods of low or zero occupancy where the variable-flow strategies yield the greatest bene-fit. These results must be interpreted taking into account that some of the assumptions made are not currently easy to verify. The first assumption consists in the fact that at partial flow rate, the diffusion system is capable of maintaining performance in terms of perfect mixing of the air supplied to the space. Currently this assumption is fairly easily verifiable for primary air systems (with neutral supply conditions) like the one simulated; however, in the case of all-air systems with hot or cold supply, this assumption would not be easily verifiable unless the implementation of a control system capable of associating the flow rate variation with a variation of the supply temperature.

4 Conclusion

The energy analysis carried out on a VAV primary air system serving a typical office building demonstrates that both variable-flow control strategies investigated yield substantial reductions in annual primary energy consumption compared with the constant air volume

(CAV) reference scenario. In quantitative terms, open-loop control based on the EN 16798-1 flow rate model (STD) achieves an annual saving of 2,986 kWh (40%), while closed-loop PI control with indoor CO_2 sensors achieves 3,210 kWh (43%), out of a CAV baseline of 7,438 kWh.

The small difference between the two control strategies (approximately 3 percentage points on an annual basis) deserves careful interpretation. From a purely energy stand-point, the STD and PI approaches perform almost equivalently; however, they differ conceptually in a significant way. The STD strategy relies on an a priori calculation model that increases the airflow rate from the first occupant entering the space, in strict compliance with Method 1 of EN 16798-1, where the person-related and building-related flow rate components are treated additively. The PI strategy, by contrast, responds to the actual measured CO_2 concentration, inherently accounting for the buffer effect of room volume and the fact that, at low occupancy levels, the building-related minimum flow rate is already sufficient to dilute the contaminant load from a limited number of occupants. This physical effect, whereby the same air volume simultaneously addresses both building and occupant pollutant sources, is not captured by the additive model of EN 16798-1, and represents a conceptual limitation of the standard-based approach.

From the perspective of primary energy accounting, the analysis confirms the significant role of the fan subsystem in the overall energy balance, particularly under variable-flow operation. When the heat pump operates at reduced capacity ratios, the generator contribution to the total primary energy decreases more rapidly than the fan contribution, because fan power follows a cubic law while the heat pump electrical input scales approximately linearly with the thermal load. However, the relative weight of each sub-system is strongly dependent on the assumed operating schedule. The present simulation adopts a 14-hour daily operating period (07:00–20:00), consistent with typical office occupancy; under this assumption, the CAV baseline electricity consumption is ap-proximately half of that which would result from continuous 24-hour operation, and consequently the heat pump accounts for a larger share of the total electrical input relative to the fans. In configurations with extended operating hours, such as continuously occupied buildings, hospitals or data centers, the fan contribution would become even more dominant, and the percentage savings achievable through variable-flow control would be correspondingly higher. This observation underscores the importance of selecting high-efficiency fans and of minimizing the internal pressure losses of the AHU, in line with the provisions of EU Regulation 1253/2014 on Ecodesign requirements for ventilation units.

The results must be interpreted in light of several assumptions that bound the validity of the analysis. First, the simulation assumes that the air diffusion system maintains adequate mixing performance across the entire flow rate range, down to approximately 10% of the nominal value. While this assumption is reasonable for primary air systems operating with near-neutral supply temperatures, it would require experimental verification for all-air systems delivering heated or cooled air, where reduced throw distances at low flow rates may compromise the uniformity of the indoor environment. Second, no minimum fan speed limitation was imposed in the simulations, thereby representing an up-per bound of the achievable savings; in real installations, practical constraints on mini-mum fan turndown ratio, VAV box minimum positions and actuator dead bands would reduce the effective savings. Third, the climatic data used, based on typical days con-structed from six-year hourly records for Milan, represent average conditions and do not capture extreme weather events that could temporarily increase the thermal load and re-duce the part-load savings. Fourth, the plant operating schedule of 14 hours per day on working days represents a significant modelling parameter: extending the operation to 24 hours, as would be the case for continuously occupied buildings, would increase the absolute energy consumption but also raise the relative savings, since the additional operating hours would fall during periods of zero or very low occupancy where the variable-flow strategies deliver the greatest benefit. In the context of the recast Energy Performance of Buildings Directive (EPBD 2024), which establishes ambitious decarbonization targets and places renewed emphasis on indoor environmental quality, the findings of this study carry direct practical implications. The demonstrated savings of over 40% in primary energy consumption confirm that demand-controlled ventilation represents one of the most effective strategies for reducing the energy footprint of office buildings without compromising indoor air quality. The near-equivalence of the two control approaches suggests that, where the installation of indoor air quality sensors is not economically or technically feasible, an open-loop strategy based on occupancy estimation can deliver comparable energy performance, provided that reliable occupancy data are available.

Future developments of this work should extend the analysis to different climatic zones and building typologies, incorporate dynamic building energy simulation to account for the interaction between the ventilation load and the building envelope, and validate the results against monitored data from real installations. Additionally, the extension of the flow rate calculation to the methods described in the Italian National Annex to EN 16798-1, and the comparison with the base standard provisions, would provide a more complete regulatory picture. A parametric study on the influence of the daily operating schedule on the achievable savings would further clarify the sensitivity of the results to this modelling assumption. Finally, the integration of occupancy detection technologies (e.g. presence sensors, access control systems, Wi-Fi device counting) with variable-flow control logic represents a promising research direction that could bridge the gap between the theoretical savings quantified in this study and their practical realization in operational buildings.

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