Variable flow rate summer air-conditioning systems with low energy consumption for small buildings

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ABSTRACT

The energy performances of a summer air-conditioning system with variable airflow destined for a sole air-conditioned open-space environment which hosts a call-centre are presented. With a simple plant configuration, implementing an opportune control strategy aimed at the simultaneous satisfaction of sensitive and latent loads, the principle energy indexes and thermal comfort with reference to a location with a typical Mediterranean climate (Cosenza, Southern Italy) were identified. The entire building-plant system was studied within a TRNSYS dynamic simulation environment, whose flexibility permitted the use of a special subroutine created by the authors to implement the required control strategy to optimise system performances.

1. INTRODUCTION

Recent awareness campaigns in residential and tertiary sectors on the importance of employing solutions capable of providing energy saving, directly involve air-conditioning systems which use primary variable air flow rate (VAV systems) [Directive 2002/91/CE]. Such systems are largely indicated to condition single areas, such as conference rooms, gyms, shopping centres, open-space etc. The greater initial costs which are caused by the systems, compared to those with a constant airflow, can be recovered in a reasonably short time, because variable airflow conditioning systems permit the attainment of relevant savings. From an energy viewpoint, the simple reduction of the airflow consents a notable saving of electrical energy as an effect of the reduction of the power requested by the ventilators. The air flow rate can, in fact, vary linearly in relation to the load, from a maximum of 100% identified in set point conditions, to a minimum of 50% necessary constraint to guarantee the external fresh air exchange and the Coanda effect in the diffusers [Vio et al., 2005]. Air flow rate variation occurs regulating, by means of the inverter, the number of ventilator rotations. Within the area it is necessary to guarantee a minimum external fresh air exchange in relation to the number of people present, for which the intake air flow rate value cannot go below a determined limit [UNI 10339, 1995]. In summer it is possible to carry out, by means of a heat exchanger, energy recovery on the expulsion airflow of the area, that is much more efficient when its temperature is lower. For this reason, the evaporative cooling of expulsion air is used. The effect of the variable air flow rate allows benefits both the heat recovery and the adiabatic humidifier, in that these systems present efficiency which increases with a decrease of the air flow rate. From a thermal comfort viewpoint, the possibility of modulating the intake air flow rate in relation to the load, permits the lowering of temperature fluctuations and humidity of the internal air of the air-conditioned area. This consents the improvement of the comfort indexes and of guaranteeing better internal air quality properties as an effect of the minor raising of pollutants in the air-conditioned volume. In this work, the performance of an all air variable air flow rate conditioning plant for summer air conditioning is compared critically to a traditional constant air flow rate plant. The merits of the proposed plant configuration with particular regard to energy efficiency and to thermal comfort conditions reached within the simulated environment are highlighted. In Figure 1 the plan of the proposed plant is reported.

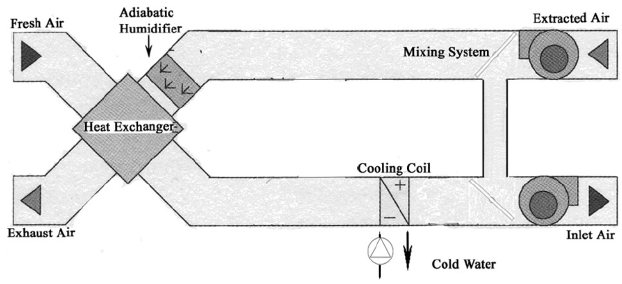

Figure 1: Plan of the considered air conditioning plant

The preparation of the intake airflow occurs by means of thermal recovery of the expelled air, opportunely cooled by means of an adiabatic humidifier, and through a cold battery in which the air flow rate reaches definite temperature and specific humidity conditions. The battery is powered by cold water whose flow rate varies in a linear manner according to the same law of the airflow. An opportune control strategy was created which, based on sensible and latent loads requested by the air-conditioned environment, determines the air flow rate value and the inlet water temperature of the cold battery, necessary to maintain the internal air temperature around the value of 26°C, and relative humidity around 50%. The possibility of varying the battery inlet temperature, consents, at the heat pump employed to produce the fluid, of operating with higher performance coefficients.

2. DESCRIPTION OF THE SIMULATED BUILDING

The building taken into consideration has a parallelepipedon form. It has an air-conditioned volume of 825 m³ with a useful surface of 15 × 20 m and a height of 2.75 m. The greater dimensions are along a North-South axis, the ceiling is flat and the internal partitions have a total surface area of 371.25 m². Each external vertical wall is equipped with a glazed surface which occupies 60% of the whole surface area. The windows are double glazed, with an air-space containing air, presenting a overall loss coefficient of 2.7 W/m²K and a solar gain equal to 78%. In order to limit the entering solar flow during summer, the windows are equipped with external shading, which reduces the solar load by 30%, and which are activated according to orientation: the South facing one from 11:00 a.m. to 15:00 p.m., East facing from 8:00 a.m. to 11:00 a.m., West facing from 15:00 p.m. to 18:00 p.m. The vertical walls which delimit the air-conditioned environment externally are a enbloc wall and have a overall loss coefficient equal to 0.513 W/m²K. They have a thickness of 4 cm of insulating material (λ=0.035 W/mK) placed externally, a solar absorption coefficient of external and internal surfaces equal to 0.3. The covering ceiling, consisting of a cement brick loft is equipped with a 4 cm thickness of insulation, has a overall loss coefficient of 0.528 W/m²K and a solar absorption coefficient equal to 0.3 on the internal side, and 0.35 on the external surface. The insulated floor is placed directly on the round and presents conductivity equal to 0.406 W/m²K also with a solar absorption coefficient equal to 0.3. In the building, different endogenous heat sources are present, such as the artificial lighting system, continually lit from 08.00 a.m. until 18.00 p.m. which employs lamps which supply a thermal flow of 5 W/m². Furthermore, the environment hosts a number of people which varies, according to the time of day, from a minimum of 10 to a maximum of 36. Each person uses a computer which supplies a load of 140 W and by means of metabolic activity, delivers a sensible load of 60 W and a latent load of 40 W. The building is operative every day of the week from 08.00 a.m. until 18.00 p.m. and the period of air-conditioning runs from the 1st of June until the 30th of September. For the evaluation of thermal comfort UNI EN ISO 7730 was referred to, hypothesising a clothing covering factor equal to 0.5 clo and a relative air speed of 0.3 m/s [UNI EN ISO 7730, 1997].

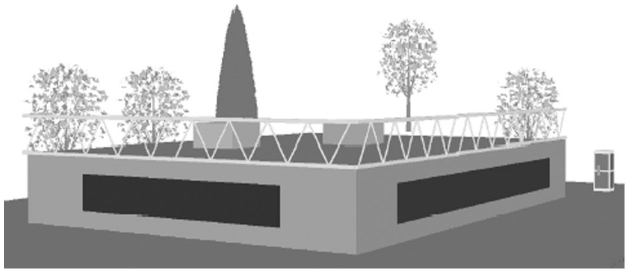

Figure 2: Perspective view of the simulated building

3. CONTROL STRATEGY OF THE AIR-CONDITIONING PLANT

The entire building-plant system was simulated, with a time step of 3 minutes, by means of the TRNSYS dynamic simulation code which, based on the climatic characteristics of the site, on the building and air-conditioning plant characteristics, provides the information necessary to carry out both energy and thermal comfort considerations. Moreover, the transient simulations consents the verification and calibration of the control strategy employed with the aim of optimising plant performance. With reference to the previously described plant configuration, the employed control strategy consents the determination of the intake air flow rate, varying it by the value 1716 kg/h, which guarantees the minimum exchange airflow, to a maximum of 3432 kg/h, identified under set point conditions, and the inlet water temperature of the of the cold battery, varying from 6°C to 21°C, in relation to the sensible and latent loads requested by the air-conditioned environment.

In order to identify a more suitable control strategy, the sensible and latent loads which the considered plant subtracts from the environment were identified, altering the air flow rate and the inlet water temperature of the cold battery in the ranges previously identified, having hypothesised a temperature of 26°C with relative humidity of 50% in the air-conditioned environment. In figures 3 and 4 the identified sensible and latent load trends are illustrated, related to the intake air flow rate and the inlet water temperature of the cold battery, having hypothesised an external air temperature of 34°C equal to the set point value. It has been possible to notice that the influence of the external air temperature is negligible, therefore both the sensible and latent loads removed from the plant, with good approximation, can be made dependant only on the m_a air flow rate and on the inlet water temperature T_{in} in the cold battery.

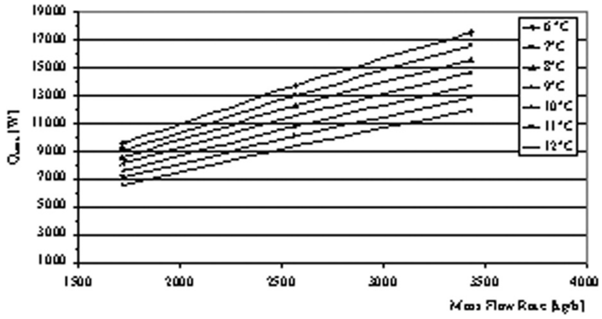

Figure 3: Sensible load trend in relation to the fresh air flow rate and the inlet water temperature

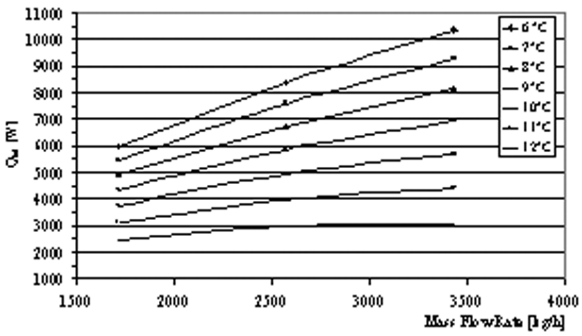

Figure 4 – Latent load trend in relation to the fresh air flow rate and the inlet water temperature

The curves reported in figures 3 and 4 are described by the equations:

$$Q_{sens}[W] = (-5.1141 \cdot T_{in}^2 + 48.643 \cdot T_{in} + 1285.1) + \\ + (-0.2296 \cdot T_{in} + 5.9816) \cdot m_a \qquad (1)$$

$$Q_{lat}[W] = (-12.244 \cdot T_{in}^2 + 265.99 \cdot T_{in} + 264.46) + \\ (-0.3898 \cdot T_{in} + 4.2922) \cdot m_a \qquad (2)$$

Equations. (1) and (2) are diagrammed in figure 5. It is possible to observe that the same sensible or latent power is obtainable increasing the air flow rate and the inlet water temperature, but that notes the sensible and latent power to be extracted from the environment only one pair of values exists m_a and T_{in} that are capable of contemporaneously satisfying the desired loads. With reference to the environment to be air-conditioned, the thermal loads to be subtracted were estimated supposing to make the sensible power linearly dependant on the difference in temperature between the internal air and the set point one of 26°C, and the latent power dependant on the difference in specific humidity between the internal air and that of the set point condition of 0.0105 kg_V/kg_a. The equations used to identify the load to be extracted are therefore the following:

$$Q_{sens} = k_1 \cdot \left(T_{\dot{a}} - 26\right) \qquad (3)$$

$$Q_{lat} = k_2 \cdot \left(x_{\dot{a}} - 0.0105\right) \qquad (4)$$

TRNSYS code flexibility has permitted the creation of a specific automatic procedure which, from temperature and specific humidity reading of the internal air, by means of (3) and (4) evaluates the sensitive and latent loads to be extracted, once these factors are known, it calculates the maximum fresh air flow rate and the inlet temperature of the cold battery which satisfy (1) and (2). The implemented procedure uses an iterative process to determine a pair of airflow and inlet water temperature in the cold battery values which satisfy equations (3) and (4). Some times the pair of airflow and temperature values fall out with the range of variability set. In such a case, the found value is approximated with the closest value receding in the work domain.

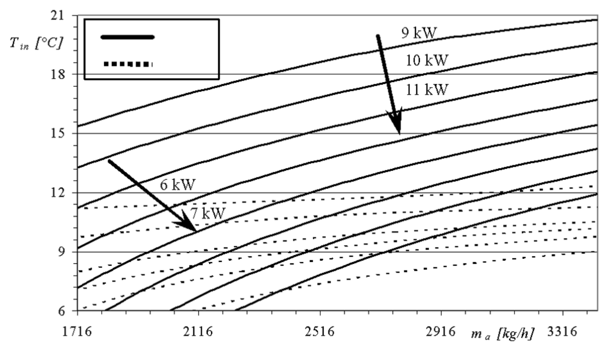

Figure 5: Sensible and latent load trends in relation to the air flow rate and the inlet temperature of the cold battery.

The simulation procedure in the TRNSYS environment carries out a series of controls within the air-conditioning plant: if the exit temperature from the adiabatic humidifier is greater than the external air temperature, a situation which is easily encountered in the early hours of the morning, or during cooler months, the heat exchanger is by-passed since otherwise the renewal air would be pre-heated instead of being pre-cooled. The cold battery is also by-passed in the case in which the outlet temperature from the heat exchanger is inferior to the temperature of the inlet water of the cold battery determined by the preciously described procedure. In the latter case, one would arrive at the paradox of heating the air flow rate instead of cooling it further. In this case, not being able to carry out a fine regulation on the thermo-igrometric conditions of the air exiting the battery, the air-conditioning would directly use the exiting airflow from the heat exchanger mixes with a fraction of internal air. In this situation, it is not possible to intervene on the regulation of the latent loads, and the control system intervenes to determine the conditions of the intake airflow to satisfy the sensible loads.

A simulation campaign spreading to the period considered consented the determination of values of the constant in equations (3) and (4) which better optimise system performance. The values obtained are: k_1= 11750 W/°C; k_2= 1704167 W /(kg_v/kg_a)

4. SIMULATION RESULTS

In figures 6 and 7 relative to the city of Cosenza (Italy, 39.3° N) trends for internal air temperature, relative humidity, entering water temperature and air flow rate in the cold battery are reported for a day in the month of June and for the hottest day in July in a period comprising between 8:00 a.m. and 18:00 p.m.. In the month of June, the sensible and latent loads requested by the building are not very high, therefore the system modulates the air flow rate and regulates the inlet temperature of the battery on relatively high values, slightly lower than 15°C and such as to guarantee the eventual dehumidification of the air within the battery. Both the temperature and relative humidity of the air in the environment assume values extremely closet o the set point ones (T_{ia}=26°C e φ_{ia}=50%). The airflow assumes lower values in the first hours of the day, during which loads are reduced. The inlet temperature assumes higher values as the specific humidity within the air-conditioned environment is low. In the month of July, given that in this period the loads are higher, the control system identifies high external fresh air loads and low feed temperatures for the battery. In the case in which it is not possible to simultaneously satisfy the two loads, preference is given to satisfying the sensible loads, therefore in some hours the system uses the maximum airflow and the minimum feed temperature. In such a case a better dehumidification is obtained compared to that requested, with acceptable relative humidity values. With regards to the internal air temperature, this undergoes greater fluctuations compared to the month of June due to sudden changes in the intake air conditions, remaining however more than satisfactory values.

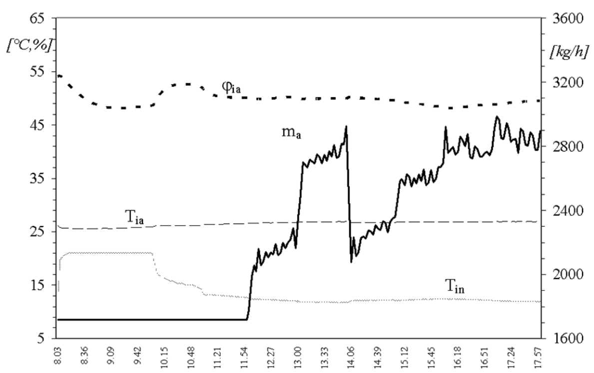

Figure 6: Temperature trends of the internal air T_{ia}, relative humidity φ_{ia}, airflow m_a and inlet temperature of the cold battery T_{in} (Cosenza - Italy, 3rd of June).

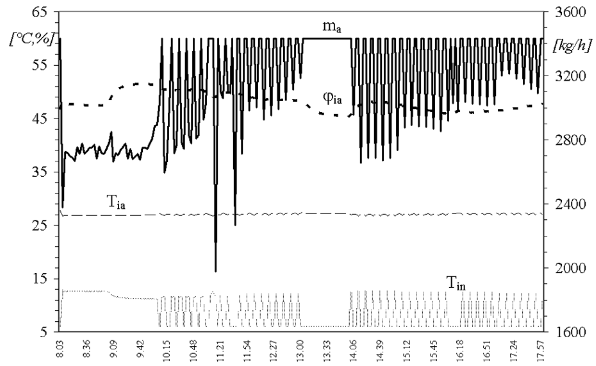

Figure 7: Temperature trends of the internal air T_{ia}, relative humidity φ_{ia}, airflow m_a and inlet temperature of the cold battery T_{in} (Cosenza - Italy, 23rd of July).

In Table 1 the number of percentage hours in which the temperature and relative humidity of the internal air fall with the three identified intervals are reported, for the same location and for different months included in the air-conditioned period. It is possible to observe that the plant consents the control in a quite sufficient way both the temperature and the relative humidity of the internal air. The internal air temperature never rises above the value of 27.5°C and the relative humidity always remains below 60%, almost 78% of the operative hours of the air-conditioned environment present internal air temperatures between 25°C and 27°C and the relative humidity is even better, almost always between 40% and 60%.

Table 1: Percentage of the hours in which the temperature and relative humidity of the air are included in different intervals of values (Cosenza – Italy, Lat. 39.3° N)

Month	T_{ia}<25	25<T_{ia}<27	27<T_{ia}<27.5	φ_{ia}<40	40<φ_{ia}<60	φ_{ia}>60
[-]	[°C]	[°C]	[°C]	[%]	[%]	[%]
June	4.5%	86.6%	8.9%	2.5%	97.5%	0.0%
July	0.2%	64.7%	35.1%	0.2%	99.8%	0.0%
August	0.0%	73.3%	26.7%	0.0%	100.0%	0.0%
September	10.5%	87.2%	2.3%	1.6%	98.4%	0.1%
Seasonal	3.7%	77.8%	18.5%	1.1%	98.9%	0.0%

In figure 8 the percentages of hours during which the PMV comfort index is included in the intervals (-1; 0) and (0;+1) are reported. The graphic shows that the control satisfies the comfort conditions, the PMV index always being between -1 e +1. In Figure 9 the number of percentage hours in which the plant works with different air flow rate values is reported. The control favours a functioning with minimum airflow above all during cooler periods (June and September), while it uses higher airflows during hotter months. In 82.6% of the functioning hours the plant works with lower airflow than the maximum, with an evident energetic advantage linked to the reduction of the energy requested by the ventilators. Figure 10 reports the percentage of the hours in which the inlet temperature of the cold battery is included in different intervals. It is interesting to observe that for 80% of the functioning hours, a water temperature greater than 10°C is requested with a consequent increase in the heat pump performance coefficient, which permits a further saving of electrical energy. From a plant viewpoint, hypothesising that the airflow and the water in the cold battery request power for the ventilators and for the pump of 700W and of 200W, the energy consumption following the variation of the airflow is reported in Table 2. In the same table, the energy required in order to operate the ventilators and the pump, in the case that the plant operates with a constant airflow, is reported. The saving of electrical energy is 27.2% for the ventilators and 32.7% for the pump.

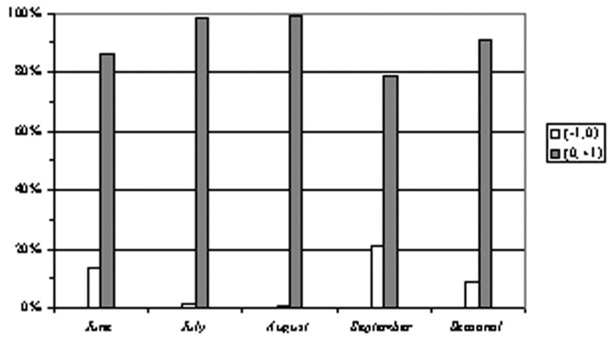

Figure 8 – Percentage of the hours in which the PMV index falls within the intervals (-1;0) (0;+1)

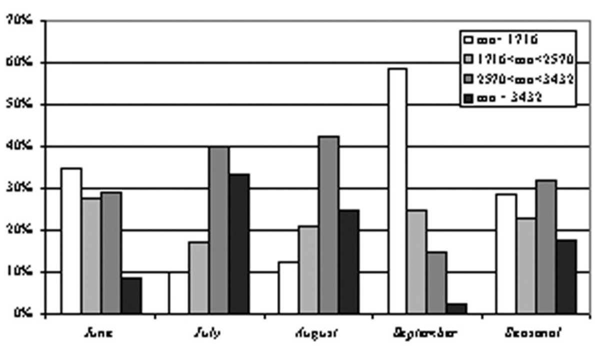

Figure 9 – Percentage of the hours in which the plant works with a minimum air flow rate, maximum air flow rate, and other intervals

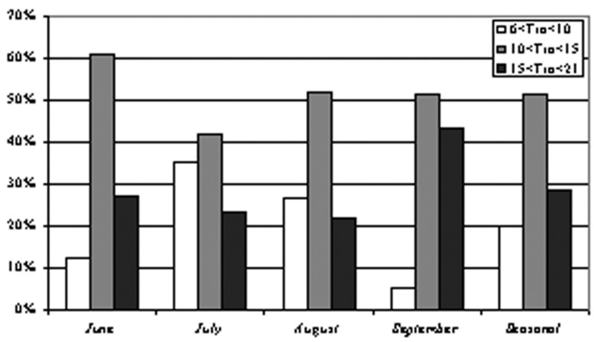

Figure 10 – Percentage of the hours in which the plant works with a inlet temperature of the battery falling in different intervals

Table 2 – Electrical energy consumption of the fans and the pump in the case of variable and constant airflow, and relative percentage of energy saving.

Month	FAN VAV	PUMP VAV	FAN COST	PUMP COST	Saving FAN	Saving PUMP
[-]	[kWh]	[kWh]	[kWh]	[kWh]	[%]	[%]
June	113.51	29.28	168.00	48.00	32.4%	39.0%
July	144.16	40.69	173.60	49.60	17.0%	18.0%
August	139.86	39.12	173.60	49.60	19.4%	21.1%
September	99.82	22.32	168.00	48.00	40.6%	53.5%
Seasonal	497.36	131.41	683.20	195.20	27.2%	32.7%

5. CONCLUSIONS

The energy performance of a summer air-conditioning system with a variable airflow destined for an open-space environment situated in Southern Italy were evaluated. This required the creation of a control strategy aimed at optimising plant performance which consents the regulation of the airflow and of the cold battery water temperature in order to simultaneously satisfy the sensible and latent loads requested by the air-conditioned environment. The comparison with a constant airflow plant highlighted seasonal electrical energy saving of 27,2% for the ventilators and of 32,7% for the circulation pump. From a viewpoint of physiological comfort of the occupants, the control consents the obtainment of an air temperature in the range of 25°C÷27°C in 78% of the hours and a relative humidity between 40% and 60% in 99% of the hours. The PMV comfort index is always maintained within the optimal range (-1;+1). Since the actual European normative framework in terms of energy efficiency is addressed to the adoption of always more stringent measures, great importance must be given to variable airflow systems due to obtainable energy saving.

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