Primary air treatment vs energy saving: comparison between different design solutions

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Primary air treatment vs energy saving: comparison between different design solutions

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Abstract. In civil application the most diffuse type of HVAC is the mixed one made by fancoils and primary air system. Usually the primary air has not only the aim of replace the exhaust air in the rooms. During the cooling season air plants have the task of dehumidification of the indoor air which is humidified by the indoor humidity sources, like the human skin transpiration and breathing of the people. On the other hand, during the heating period the fresh air must be humidified to compensate the low humidity ratio of the external air. With the aim of energy saving related to the ventilation in buildings, from the beginning 2018 the new European Regulation EU 1253/2014 has been introduced some limits in terms of energy efficiencies. The present work investigates different layouts of plants for the primary air treatment in HVAC systems. Several types of heat recovery units have been considered in the study in order to find which is the best solution for the optimization of the air treatment. Several case studies have been considered in the energy analysis and the results obtained through numerical simulations have highlighted the critical nature of the interventions aimed at limiting energy consumption in primary air treatment systems.

1. Introduction

During the last decades, many works have been carried out for evaluating the thermal and energy behaviour of HVAC (heating, ventilating and air conditioning) systems [1, 2]. In non-residential applications, the most widespread HVAC system is the mixed one, i.e. consisting of fan coils coupled with primary air ventilation system. In this type of system, an Air Handling Unit (AHU) is used to treat the fresh air in a dedicated plant as part of a complete HVAC system. Normally, in addition to the normal task of renewing the indoor ambient air with fresh one, during the summer period, primary air is supplied to the rooms for controlling the latent loads. The increase of the vapour content into the air is due to the infiltration of the outdoor air and to the vapour release by people inside the building. On the other hand, in winter period, the primary air is humidified if needed by the AHU. The humidity produced by a person in the office can be considered of about 100 g_v/h (equal to about 70 W of latent load), it is therefore necessary to evaluate what should be the air flow to balance this latent load. To do this, it has to be considered the conditions that can be obtained downstream of a cooling and dehumidification coil.
For example, it can be considered the air treatment which takes place by means of an 8-row finned coil with 2,5 mm spacing between fins, fed with heat carrier fluid at 7°C/12°C, crossed by air with frontal velocity v = 2,5 m/s. By using the approximate method of the bypass-factor the output conditions can be easily obtain for an incoming air flow under the design conditions. If the external ambient has the following conditions, T_e = 32°C and RH_e = 50%, the output conditions after the finned coil are T_{bo} = 13,1°C and RH_{bo} = 98% corresponding to specific humidity x_{bo} = 9,2 g_v/kg_{da}.

For indoor ambient conditions, T_a = 26°C with RH_a = 50% the specific humidity is x_a = 10.7 g_v/kg_{da}. From this simple consideration it follows that the dehumidification capacity of the renewal air thus treated is 1,5 g_v/kg_{da}. The required air flow for each person is therefore about 67 kg_{da}/h which corresponds to about 56 m^3/h. Similar flow rates generally involve a change of air around 2.5 vol/h of

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the environment in which the air is introduced. The magnitude of the air flow rate has an important impact on energy consumption for air conditioning, hence the need to adopt effective strategies to contain energy consumption deriving from the treatment of fresh air.

With the aim of limiting energy consumption for ventilation in buildings, in November 2014, in the framework of the European regulation ErP 2009/125/CE known as the Ecodesign directive [3], the EU Commission Regulation 1253/2014 entered into force concerning the ventilation units of which it fixes minimum requirements [4]. This regulation provided for the implementation in two phases and from 1 January 2018 the second phase took place which envisages the most restrictive limits.

In the case of air handling units, they must have at least a F7 filtration section on the outside air and an average M5 filtration on the air in expulsion. The presence of pressure probes is used for signalling dirty filter. Moreover, a heat recovery unit must be used with no less than 73% in terms of thermal efficiency in dry conditions. Some application aspects of the regulation appear controversial and do not respond to certain air conditioning needs. This work, regardless of the aforementioned regulation, aims to find the most suitable solutions in order to optimize the working conditions of an AHU.

2. The possible plant solutions

Several layouts of the AHU can be used in HVAC plants for the same aims. In this section, four plant solutions for the AHUs are described and analysed in the following section of the paper. Each case is characterised by peculiar properties and components that distinguish it.

2.1. The common solution with sensible heat recovery (Case A)

This solution is the most common layout that can be found in the HVAC plants. A scheme of an AHU with the sensible heat recovery unit is shown in Figure 1. The use of heat recovery unit with efficiency above the minimum required by regulations or laws would always be advantageous. On the other hand, one of the main issue are the pressure drops that have to be limited, consequently there will be larger exchange surfaces and lower front speeds; the limit is represented by the overall dimensions which, increasing, make it difficult to install the AHU in small spaces. The energy efficiency regulation refers exclusively to dry efficiency of the system, completely ignoring the problems deriving from humidity control which give rise to significant energy consumption for humidification and dehumidification in heating and cooling operation periods respectively.

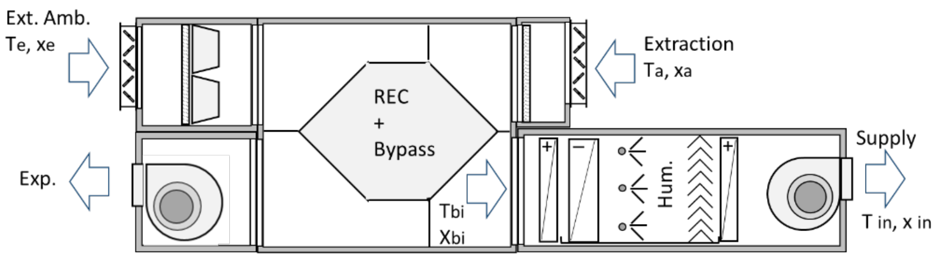

Figure 1. Layout of an Air Handling Unit equipped with a sensible heat recovery.

2.2. The solution with heat recovery wheel (Case B)

Based on the considerations summarized in the previous subsection it is therefore important to evaluate the possibility of installing a heat recovery wheel for the combined recovery of sensible and latent heat [5]. Considering only the cooling process with dehumidification, the installation of a heat recovery wheel could be advantageous compared to the use of a simple sensible heat recovery unit, but it is also important evaluating its behaviour during the heating period. On the other hand, a sensible heat recovery would be effective in winter, the period of the year in which the difference between interior and exterior temperatures is higher than in summer.

The choice is therefore strongly influenced by the climate of the location (favouring hot-humid and cold-dry climates), therefore an overall assessment is required over the whole year. In a climate

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where seasonal values are subject to wide thermo-hygrometric variations, it is essential to have a recovery system that is as flexible as possible and able to guarantee good efficiency in all conditions.

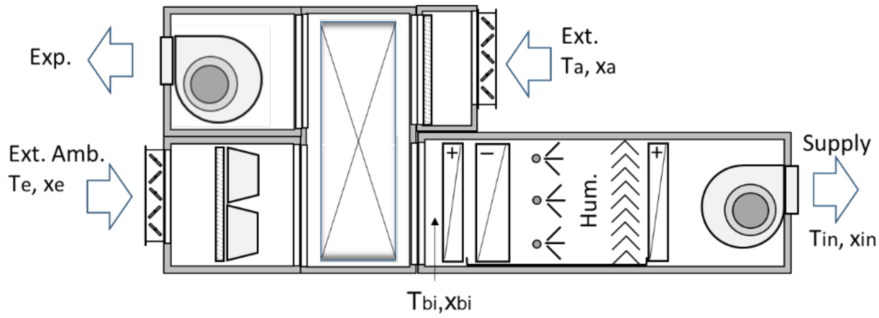

Figure 2. Layout of an Air Handling Unit equipped with a rotary enthalpy heat recovery.

2.3. The solution with adiabatic cooling section (Case C)

The simple use of a heat recovery unit, even if at high efficiency, as foreseen by the legislation, is not enough to optimize the energy consumption connected to the renewal of the air. The low contribution that a sensible heat recovery can offer during the summer period can be significantly increased by the use of indirect adiabatic cooling (IAC). In this case, an adiabatic saturator is installed on the extracted air flow from the air-conditioned rooms and upstream of the heat recovery unit, which evaporates the water atomised at the expense of the sensitive heat of the air in transit, reducing its temperature. In this way the heat recovery will find itself operating with a higher temperature difference and a consequent greater cooling of the incoming external air. The scheme of this solution is shown in Figure 3.

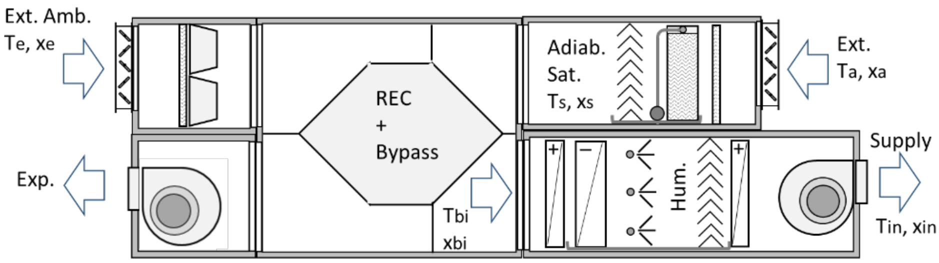

Figure 3. Layout of an Air Handling Unit equipped with adiabatic cooling section.

2.4. The solution with two heat recovery units (Case D)

In this work, a novel layout of the AHU with a second heat recovery unit has been investigated. In the cooling and dehumidification phase, the second heat recovery exchanges heat between the cold air exiting the coil and the hot incoming air to simultaneously realize a pre-cooling phase with dehumidification and post-heating. The complexity of the system does not allow to evaluate in a simple way the result obtainable in the design conditions as it was for the previous cases, it therefore refers to the analysis of the results obtained through the numerical simulation of the various cases. The scheme of this solution is shown in Figure 4.

3. The description of the systems investigated

In the first step of the work, the boundary conditions used in the design process have been considered for the calculation of the maximum flow rate of the fans. In particular, a dedicated control strategy has been considered for the evaluation of how the AHU works. The description of three different conditions of working are summarized in Table 1.

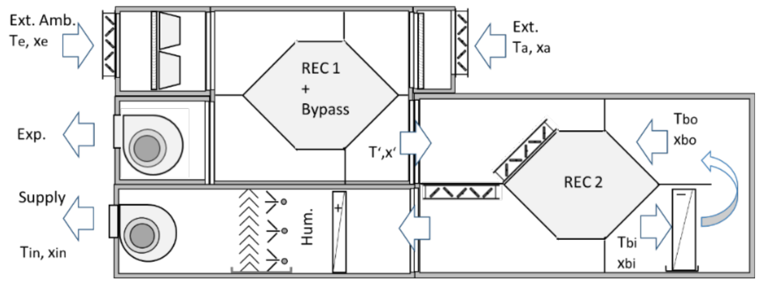

Figure 4. Layout of an Air Handling Unit equipped with two heat recovery units.

Table 1. Boundary conditions used in the energy analysis.

	Condition 1 (heating)	Condition 2 (middle season)	Condition 3 (cooling)
External Air Temperature	$T_e < 20°C$	$20\ °C \leq T_e \leq 26°C$	$T_e > 26°C$
Indoor Air Temperature	$T_a = 20°C$	$T_a = T_e$	$T_a = 26°C$
Relative Humidity of the Indoor Air Temperature	$RH_a = 50\%$	$RH_a = 50\%$	$RH_a = 50\%$

In the case study with the sensible heat recovery unit (Case A), the amount of heat exchanged between the incoming external air and the exhaust air depends on the thermal efficiency of the recovery heat exchanger, defined according to EU Regulation 1253/2014 as:

$$\eta = (T_{e'}-T_e) / (T_a-T_e) \quad (1)$$

where $T_{e'}$ is the outlet air temperature of the outside air (fresh air) after having crossed the heat exchanger.

This expression is valid for balanced mass flow rates and must be evaluated for $T_a = 25°C$ and $T_e = 5°C$, without condensation. As it can be seen, the Regulation refers exclusively to the condition in which $T_e < T_a$ (heating operating condition) Therefore, due to the different density of the incoming and expelling air, different volume flow rates will be obtained and the flow rate ratio is inverse to that of the densities. The variations of temperature and humidity, as well as air density, influence the heat exchange coefficients, making the analytical analysis complex for the calculation of the effective efficiency of the heat recovery unit modifying the operating conditions. In this work we have chosen, in accordance with the Regulations, to operate with balanced mass flow rates or, equivalently, with nominal volumetric flow rate with density equal to $\rho = 1.2\ kg_{da}/m^3$. As reference, a heat recovery unit present on the market, whose performances are known, was used. Based on the reference data provided by the manufacturer of the heat recovery unit, the Eq. 2 has been defined for the evaluation of the thermal efficiency. A similar approach was used to consider the effect of low flow rate at the heat recovery unit. These equations were used in the simulations in order to consider the behaviour of the unit in function of the working conditions.

$$\eta = \eta_{ref} + 2*10^{-4}(T_{fi} + T_{ci} - 15) \text{ where } [\eta_{ref} = 73.4\% \ @\ T_e = -5°C,\ T_a = 20°C] \quad (2)$$

In the design conditions indicated above, for an internal vapour production of 100 g_v/h and an air flow rate of 67 kg_{da}/h, the specific humidity entered must be equal to: $x_{in} = x_{set} - 100/67\ g_v/kg_{da}$.

If $x_{bi} < x_{in}$ it will be necessary to humidify by entering a quantity of steam equal to $x_{in} - x_{bi}$. If the AHU is equipped with variable-flow fans, controlled according to humidity, it will be possible to reduce the amount of outdoor air to a minimum of 36 kg_{da}/h.

On the other hand, if $x_{bi} > x_{in}$ it will instead be necessary to dehumidify; it is therefore necessary to cool the air below its dew point T_{dp} (T_{bi}, x_{bi}). The simulation requires the operating conditions of the cooling coil so that the outlet air has specific humidity x_{in}. By always using the approximate method of the bypass factor, for the coil describe above in the text, with numerical method it is possible to obtain the average temperature T_m of the battery and the enthalpy of the outgoing air; it is therefore possible to calculate the energy required for the air treatment.

The hygrometric control is not carried out even if a cooling with humidification is required. In any case, the dehumidification process means that the temperature of the air coming out of the cooling coil is too cold to be supplied directly into the indoor environment, so the post-heating coil is required. In this work the energy required for post-heating is always recovered from the condensation heat of the chiller and it is not considered for the purposes of comparing the layout solutions investigated.
In the case B, the solution with the heat recovery wheel, it was assumed that a heat recovery unit could be available with the same thermal efficiency as the sensible heat recovery, equal to 73.4%, and a hygroscopic efficiency of 70.3%. As far as the thermal control is concerned, there are no particular problems, differently as regards the hygrometric control.

In the Case C, which consists of indirect adiabatic cooling solution. The IAC system will be active only for $T_e > 26$°C. In this case, the ambient air at conditions T_a and x_a, before reaching the HRU and then being expelled, passes through the adiabatic saturator with a 85% saturation efficiency; this treatment of the air stream satisfies the relation $h(T_a, x_a) = \text{constant}$. In this case, the additional presence of the adiabatic saturator leads to an increase in load losses and a consequent increase in energy consumption for ventilation. In the results obtained, it is therefore necessary to assess the net balance between energy saved for refrigeration and for ventilation.

In the Case D, which has the installation of a second heat recovery unit, as far as the first recovery unit is concerned, all the already treated operating conditions are repeated, while for the second unit new considerations have to be done. The second heat recovery unit was assumed the same as the first and will be operational only when the cooling coil is active, otherwise the recovery unit and the cooling coil will be bypassed. In this case the bypass will be total, i.e. it will divert the air immediately downstream of the first heat recovery unit, thus avoiding double passage and transit through the cooling coil. The proposed scheme also allows eliminating the preheating coil. In order to obtain the value of x_{in} that satisfies the indoor set-point condition it is necessary to operate in an iterative way to reach the equilibrium conditions. The temperature range in which the second recovery unit operates is different from the one considered for the first, moreover condensation conditions almost always occur and, in this case, we are interested in knowing specific temperature and humidity of the outgoing hot flow. Using the data provided by the manufacturer, the correlation functions have been obtained that allow the estimation of the increase in the outlet temperature of both the hot and the cold flow in the field of recovery temperatures. As in the previous case, the presence of the second heat recovery unit determines the increase in pressure losses inside the AHU.

4. Results of the simulations

The simulations have been carried out on an hourly basis with reference to the TRY (Test Reference Year) of Milan, Italy. The primary air system has been planned to work 10 hours a day (from 7:00 a.m. to 5:00 p.m.) for an office building and a volume flow rate of 9000 m^3/h. In Table 2 the annual energy demand for the different layouts and the total amount in terms of kg of vapour needed by the AHU are summarized. For a homogeneous comparison of the different values, the energy demand is also expressed in terms of TOE (Tonnes Oil Equivalent) in Table 3.

Table 2. Summary of the results of the simulations.

Energy and Vapour		Case A		Case B	Case C	Case D
		Constant Flow Rate	Variable Flow Rate			
Heating	kWh_t	19707	13898	41127	13898	13898
Cooling	kWh_t	49061	49061	43780	44742	38045
Humidification	kg_v	25276	5637	299	5637	5637
Ventilation	kWh_e	17311	12669	17015	13455	14090

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Table 3. Results of the simulations in terms of TOE.

Primary Energy			Case A: Constant Flow Rate	Case A: Variable Flow Rate	Case B	Case C	Case D
Heating	Thermal Efficiency	0.9	1.88	1.33	3.93	1.33	1.33
Cooling	SEER	3.5	2.62	2.62	2.34	2.39	2.03
Humidification	Efficiency	0.5	3.01	0.67	0.04	0.67	0.67
Ventilation	Efficiency	1	3.24	2.37	3.18	2.52	2.63
	Total		10.75	6.99	9.49	6.91	6.67
Var. %	Reference Case A Const.		-	-35%	-12%	-36%	-38%
	Reference Case A Var.		54%	-	36%	-1%	-5%

5. Conclusions

The results obtained through numerical simulation have highlighted the criticality of the interventions aimed at containing energy consumption in primary air treatment systems. The increase in the efficiency of the heat recovery unit has certainly brought significant energy savings during the heating period, but its effect is of little relevance during the summer period. The use of a heat recovery wheel, although it presents a better result in the summer period. Against a limited summer improvement, it was found a considerable increase in consumption for heating and ventilation and even if the need for humidification is almost eliminated, the global energy balance remains clearly negative. On the other hand, the use of fans that are not simply multi-speed or with continuous variation of the flow rate, as required by the Regulation, is fundamental, but the flow rate must be regulated automatically according to the internal hygrometric conditions or air quality. This solution presented the maximum energy savings. It has to be noted that the Regulation is aimed exclusively at containing energy consumption in the winter period, a condition that is suitable for cold climates but not for a Mediterranean climate in which consumption for cooling is clearly higher than that for heating. The cases examined (indirect adiabatic cooling and double heat recovery units) are aimed at containing consumption for refrigeration and both lead to a positive result even if of different entity. Furthermore the best solution (Case D) in terms of energy demand has only another component (i.e. the second heat recovery unit) if compared to the reference layout system (Case A). On the other hand, it requires more space for the installation but this can be justifiable by the higher energy efficiency of the system.

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