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Numerical evaluation on performances of AHU equipped with a cross flow heat exchanger in wet and dry operation

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Abstract

In this paper the comparison between the performance of a cross flow heat exchanger in wet and dry operation for air handling process has been investigated. In addition, a case study of application of the component to perform indirect evaporative cooling in a AHU was studied with the software TRNSYS.

Using experimental data and an appropriate analytical method, energy saving performances of the system has been evaluated through the entire cooling season for a typical Mediterranean site. Results show that high energy saving potential can be obtained if the component is operated in wet operation in term of reduction of electricity consumption.

Keywords: Indirect evaporative cooling; precooling of outside air; energy saving in HVAC

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Nomenclature

AHU	Air handling unit	SEER	Seasonal energy efficiency ratio
DEC	Desiccant and evaporative cooling	WB	Wet bulb
ε	Thermal efficiency	WBD	Wet-bulb depression
HX	Heat exchanger		

1. Introduction

Evaporative cooling is a simple and effective technique for cooling an air flow. The sensible heat of the air is transferred to water molecules in the form of latent heat, in order to allow evaporation. The amount of sensible heat absorbed depends on the amount of water that can be evaporated at given conditions. Evaporative cooling works by using water's large enthalpy of vaporization. It is a function of the wet-bulb depression (WBD) of the air flow treated defined as the difference between its dry-bulb and wet-bulb temperature.

Cross flow plate heat exchangers are commonly used in air conditioning systems for heat recovery purposes during winter time. When indirect evaporative cooling is used, the same heat exchangers can be used to efficiently cool the outside air during summer, wetting the channels of the exhaust air with water [1,2,3]. In addition, this component can efficiently be integrated in a desiccant cooling cycle (DEC) downstream of the adsorption process to maximize the indirect evaporative cooling potential associated with the exhaust air stream.

2. Description of the work

The main goal of this work is first to experimentally evaluate the energy performance of a cross flow heat exchanger in wet and dry operation. Afterwards, the influence of using the wet heat exchanger instead of a standard cross flow heat exchanger in a common AHU was numerically investigated by means of the simulation software TRNSYS. In particular results presented show the comparison between seasonal energy performances of the system in both cases for cooling operation.

2.1. Experimental setup

The system consists of a plate heat exchanger designed for dry running, which is here investigated under wet conditions with the aim to use the component as indirect evaporative cooler. In figure 1 a scheme of the experimental setup is reported. Rated process air flow rate of the tested heat exchanger is 1200 m^3/h whereas its rated efficiency for dry operation is 57.4% (T_{1-1} = 30°C, T_{2-1} = 25°C, T_{1-2} = 27.1°C, T_{2-2} = 27.9°C). Cooled air and wet air are in a cross flow arrangement. The surface of secondary flow (return air from the building) air channels is wetted by water sprayed by nozzles, such that a water film evaporates into the cooling air and decreases the temperature of the heat exchange surface. Spray nozzles used have relatively large orifices and operate with low water pressure and thus it has been possible to use an inexpensive pump for water circulation. Process air flowing in the primary airflow channels is cooled down due to the lower temperature surface of the separating wall of the heat exchangers. The temperature of the saturated air film on the wet air side depends on the local wet-bulb temperature of the air stream that gradually increases during the humidifying process. The average temperature of the saturated water film on the wet air side, is approximately equal to the average temperature in the water sump or slightly lower [2].

According to the common definition, the efficiency for a wet heat exchanger can be calculated with the following formula:

$$\varepsilon = \frac{(T_{11} - T_{12})}{(T_{11} - T_{21wb})}; \quad [-] \tag{1}$$

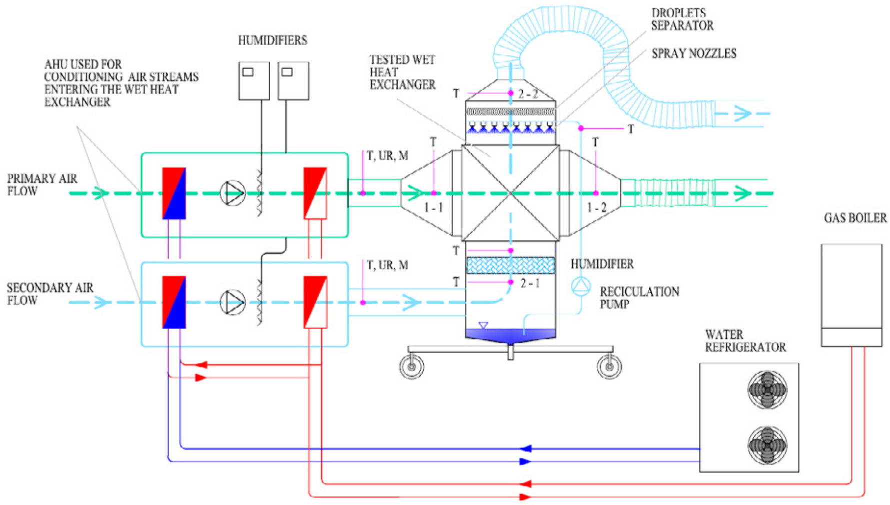


Fig. 1. Scheme of the experimental setup

2.2. Results on heat exchanger performances

Tests were conducted by two main investigations: first, the energy efficiency of the component was evaluated with the same volumetric flow rates. Then, the assessment was aimed to study the decrease of performances with decreasing volume flows on the secondary side. In both cases, the system was studied in dry and wet operation.

In order to compare the behaviour of the heat exchanger in dry and wet modes a wide monitoring campaign was conducted in order to obtain experimental data to be used later for numerical simulation. Figure 2 shows the cooling power ratio of the heat exchanger defined as the ratio between the cooling power measured in wet operation to the cooling power measured in dry operation.

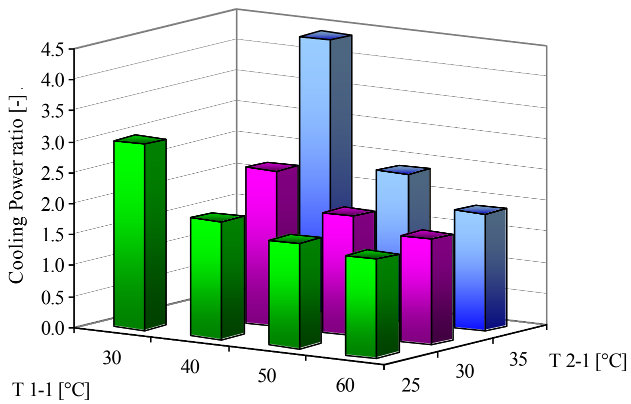


Fig. 2. Cooling power ratio between wet and dry operation of the heat exchanger

For the considered inlet air conditions it ranges from a minimum value of 1.6 to a maximum value of 4.4. Wet bulb temperatures of secondary air flow ($T_{2\text{-}1w\,b}$) are 17.4°C, 22.7°C, 27.1°C respectively for $T_{2\text{-}1}$ = 25°C, 30°C, 35°C. For $T_{1\text{-}1}$=30°C and $T_{2\text{-}1}$=25°C (typical summer case) cooling power in wet operation is three times the one measured in dry operation. Therefore, thanks to the mentioned effect, the cooling power exchanged in dry running could also be obtained with a smaller heat exchanger reducing purchase cost.

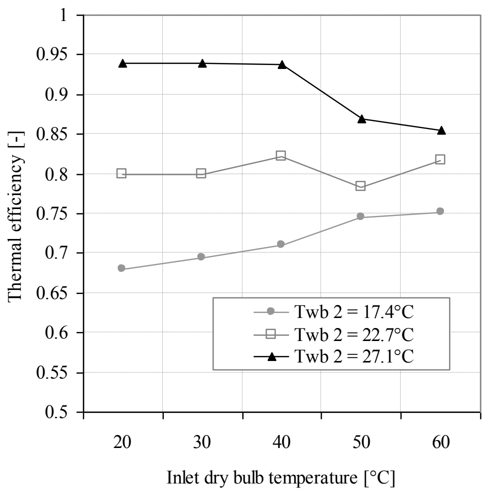


Fig. 3. Experimental data on the heat exchanger thermal efficiency in wet operation

The influence of reducing the flow rate of secondary air stream was also studied. This is an important issue since in most system configurations return air extracted from the building is used as secondary air flow in the heat exchanger. Return air flow rate is often smaller than the one on the supply side.

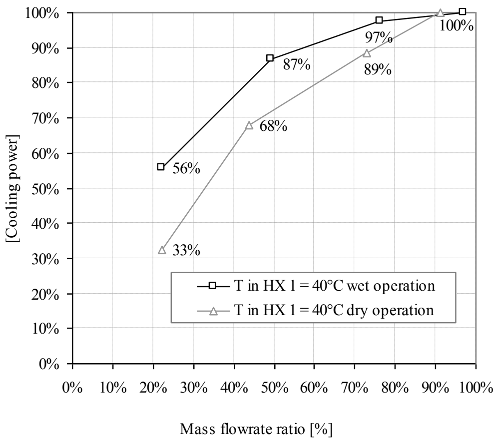


Fig. 4. Cooling power decrease versus air flow rate ratio

Figure 3 describes the decrease of cooling power versus the mass flow rate ratio for the cases of dry and wet operation and defined as following:

$$Mass\ flow\ rate\ ratio = \frac{\dot{m}_2}{\dot{m}_2};\quad [-] \tag{2}$$

It can be observed that, up to 50% of mass flow rate ratio, the reduction of cooling power turns out to be sufficiently small (10-12%) whereas a decrease of only 2-3% occurs for a mass flow rate ratio of 80%.

2.3. System simulation

Second step of the analysis was to investigate the influence of using a wet heat exchanger instead of a standard cross flow heat exchanger in a common AHU. Analyses were carried out using the TRNSYS platform and refer to a building equipped with an full-air AHU able to control temperature and humidity. The reference building is located in a typical Mediterranean site (Palermo - Italy) with a maximum and minimum seasonal outside temperatures of 35.7 °C in summer and 5 °C in winter. Maximum humidity ratio in summer reaches 24 g/kg, while solar radiation on the horizontal plane is 1664 KWh/m^2 year. Wet bulb temperature of outside air is normally lower than 25°C.

Conditioned room has a total area of about 100 m^2 and an internal air volume of about 450 m^3. External surfaces are the roof (U = 1.39 W/m^2K) and two walls (U = 1.42 W/m^2K) facing North-West and South-East. The glazed surfaces are made of clear double-paned glass mounted on aluminium frames (overall fenestration U = 3.2 W/m^2K). They face South-East and account for 10% of the total external area of the building. No shading device is present on the glazed surfaces. Adjacent spaces are conditioned only during the winter season.

Two different AHU were considered equipped mainly with standard components (cooling coil, heating coil, fans) and including the mentioned heat exchanger in dry (Fig. 5 left) and wet (Fig. 5 right) operation. Cooling coil and heating coil are fed respectively by an electric refrigerator and a gas boiler. Thermal energy needed for the reheating process is also considered. The type of heat exchanger is the same for both cases whereas the thermal efficiency of heat exchanger in dry operation was assumed constant and equal to 0.7. This value corresponds to an optimized and high quality cross flow heat exchanger, in order to consider the best possible case for dry operation.

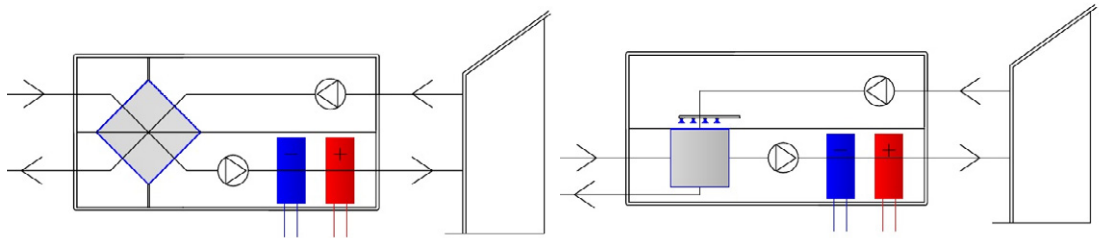


Fig. 5. AHU equipped with sensible heat exchanger (left) and wet heat exchanger (right)

Simulated systems are intended to be provided with variable speed fans, having a minimum flow rate of process air related to required building air change (see Table 1).

Table 1. Technical data of AHU

		AHU equipped with sensible heat exchanger	AHU equipped with wet heat exchanger
Max flow rate of process air	[kg/h]	5000	5000
Min flow rate of process air	[kg/h]	2000	2000
Mass flow rate ratio	[-]	85%	85%
Thermal efficiency of heat exchanger	[-]	0.7	See Fig.1
Installed fan power	[kW]	1.1	1.2
Rated circulation pump power	[kW]	0	0.15

All simulations were performed during the cooling season over a time interval of six months. After a preliminary phase of sizing and optimisation, a complete mathematical TRNSYS model, including weather data processing, reference building, AHU components, auxiliaries and control strategy, was created. Main data introduced in the model are reported in Table 2.

Table 2. Simulation data introduced in the TRNSYS platform

Technical data of the main components		
Efficiency of gas boiler	[-]	93
Rated EER of compression chiller	[-]	2.5
Rated fan power of the heat rejection unit	[kW]	0.35
Temperature set point of building	[°C]	26
Absolute humidity set point of building	[g/kg]	10.5
Simulation time step	[h]	0.25
Simulation time interval		1th May - 31th Oct
Total operation hours	[h]	1577

The heat exchanger in wet operation was simulated using the Type 757 of the TRNSYS TESS library. The external file containing the performance data was modified according to the measurements done on the component. In addition, the Type used was updated by means of experimental data in order to take in consideration also unbalanced flow rate ratio between primary and secondary air stream. Figure 6 shows a simplified scheme of the TRNSYS project used for the simulations.

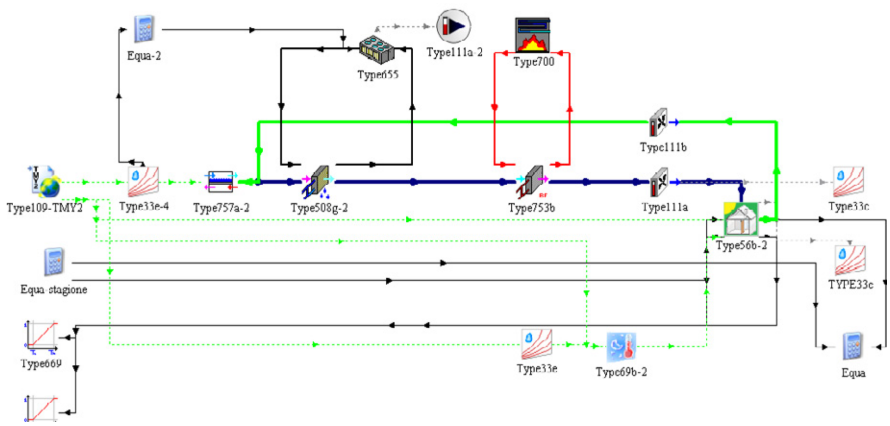
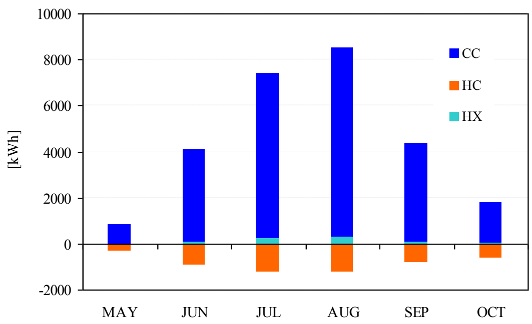


Fig. 6. Simplified scheme of the TRNSYS project

2.4. Simulation results

For the considered site, simulation results for the whole summer season have shown that the contribution of the heat exchanger in dry operation is very low (≈ 3%) whereas in wet operation about 30% of cooling energy can be delivered by the wet heat exchanger. In other terms, cooling energy saved by means of the heat exchanger in wet operation is ten times the one exchanged in case of dry operation for the whole time period considered.

Following diagrams show the energy distribution between the two cases considered. Heating energy due to reheating process is shown as a negative quantity. System seasonal mean value of EER (SEER) obtained for the considered cases are 1.72 and 2.07 respectively for the case of dry and wet operation of the heat exchanger. This value accounts for all electric components presented in the system (chiller, fans, pumps, etc..). Cooling energy provided by the wet heat exchanger during summer is 6930 kWh with an average cooling power of 4.4 kW over the entire time period. The total cooling energy delivered by the AHU is about 22000 kWh.



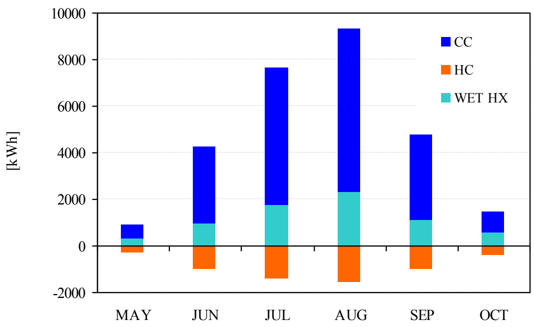


Fig. 7. Energy distribution in the AHU equipped with sensible heat exchanger (left) and wet heat exchanger (right)

Total electricity consumption in case of dry operation is 12570 kWh. Electricity saving for the case of wet operation amounts to 14% (1710 kWh) included electricity consumed by the circulation pump and assuming the same pressure drops in both cases. Considering a tariff of 0.18 €/kWh for electricity, it turns out that a seasonal energy cost saving of about 300 € can be obtained using the wet heat exchanger instead of a standard one. Therefore, extra costs due to components necessary to make the heat exchanger able to operate in wet mode as recirculation pumps, spray nozzles and basin could be rapidly covered by increased electricity savings.

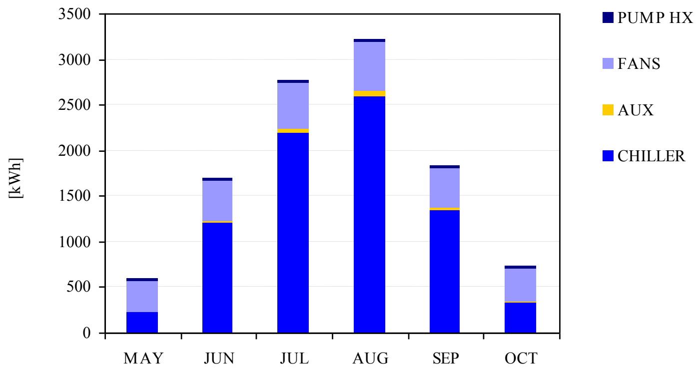


Fig. 8. Electricity consumption distribution in the AHU

3. Conclusions

In this work the influence of using a wet heat exchanger instead of a standard cross flow heat exchanger in common AHUs was numerically investigated by means of the simulation software TRNSYS. Data concerning the heat exchanger in wet and dry operation were obtained experimentally.

The study demonstrates the opportunity of using a wet heat exchanger in combination with standard AHUs in climates with relevant cooling needs as indirect evaporative cooler. For the considered case study, cooling energy saved by means of the wet operation of the heat exchanger is about ten times the one exchanged in case of dry operation accounting for about 30% of the total cooling energy delivered by the AHU. The use of return air as secondary air flow in the heat exchanger can guarantee quite low and constant wet bulb temperature permitting high and stable performance of the component.

The proposed solution requires some changes on the component. In order to permit the wet operation, the heat exchanger has to be equipped with spray nozzles, circulation pump and basin. A detailed cost benefit analysis should be carried out in order to quantify production cost of the component at industrial level.

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