Manuscript Details

Manuscript number	ATE_2017_3624_R2
Title	The Design of Countercurrent Evaporative Condensers with the Hybrid Method
Article type	Research Paper

Abstract

The heat and mass transfer in evaporative condensers are complex to model analytically and numerical simulations, when applied to multi-phase fluid dynamics in complex paths, often involve too high computational costs. Experimental campaigns at full scale of different heat transfer geometries and tube arrangements involve long lead times and high costs as well. The aim of the present work is to overcome the present limitations and to apply a new method to evaluate the overall performance of the countercurrent evaporative condensers, starting from the experimental, numerical or analytical data with a small scale approach. A test bench has been purposely designed and built up in order to reach and keep constant all the parameters determining the evaporative condenser heat transfer performance. In previous experimental contributions available in the literature, the air conditions were not controlled: here, an air handling unit placed before the evaporative condenser inlet allows to set up temperature and relative humidity of air in large ranges. An extended experimental campaign has been carried out to get affordable data to be used to find a relationship correlating the dry bulb temperature and relative humidity of air after its interaction with water and the condenser tubes surfaces, while all the parameters were set up and controlled. The regression function fits well the experimental data as the predicted values of temperature and relative humidity are characterized by a maximum percent deviation lower than 2.5 % and 4 % respectively. An iterative procedure was then implemented to determine the conditions of air going through the evaporative condenser in order to extend small scale results to full scale performance according to real geometries. The effect of the water flow rate on the cooling capacity was investigated and the results show that an increase of 50 % of the sprayed water leads to an increase of 14 % of the performance.

Keywords	Heat exchanger design; Hybrid method; Evaporative Condensers; Experimentals

Corresponding Author	Giuseppe Starace
Corresponding Author's Institution	University of Salento
Order of Authors	Giuseppe Starace, Maria Fiorentino

Suggested reviewers	Paulo Smith Schneider, Filippo de Rossi, Magesh Thiruvengadam

Submission Files Included in this PDF

File Name [File Type]

RESPONSES TO REVIEWER COMMENTS_R1.docx [Response to Reviewers]

Highlights - Hybrid Method Evap Condensers.docx [Highlights]

Hybrid Method - Performance of Evap Condensers_R1.docx [Manuscript File]

To view all the submission files, including those not included in the PDF, click on the manuscript title on your EVISE Homepage, then click 'Download zip file'.

RESPONSES TO REVIEWER COMMENTS

The Design of Countercurrent Evaporative Condensers with the Hybrid Method

M. Fiorentino[a], G. Starace[b,*]

[a]Decsa S.r.l., Via Cappelletta, Voghera 27058, Italy
[b]Department of Engineering for Innovation, University of Salento, via per Monteroni, Lecce 73100, Italy
*Corresponding author: giuseppe.starace@unisalento.it
tel.: +39 0832 297753 fax: +39 0832 297777

Many thanks to all reviewers and to the editor for their valuable contribution to the article that we authors hope will be in the new revision suitable for publication on Applied Thermal Engineering.

Hereafter you'll find all the reviewers' comments and observations as well as our responses to each of them.

Sincerely,

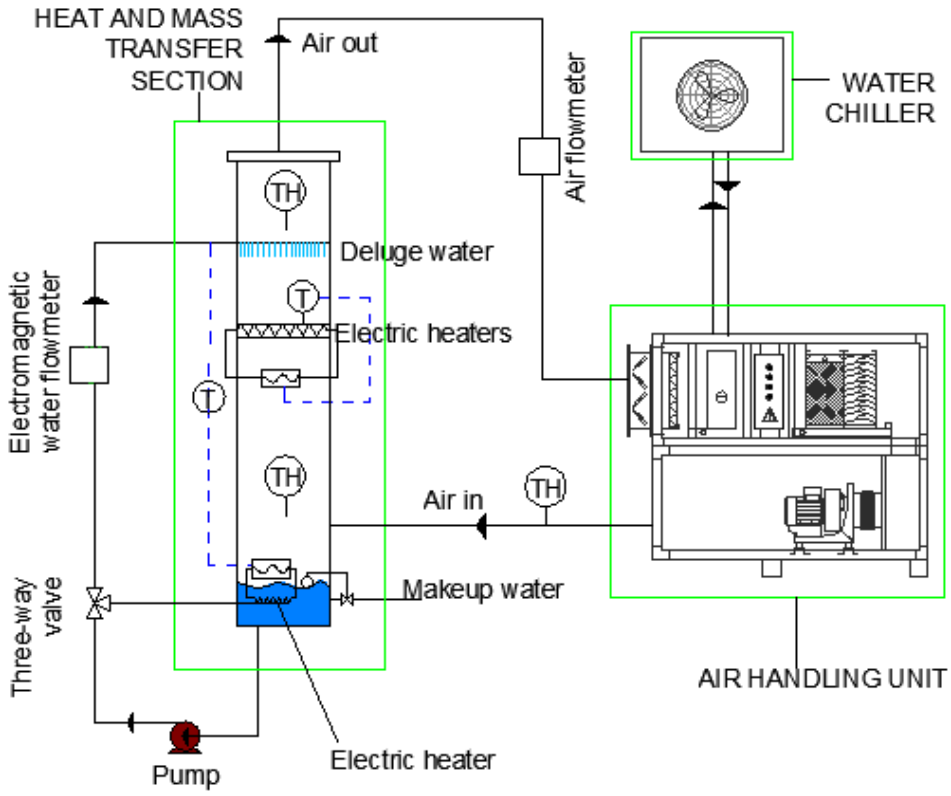
The authors.

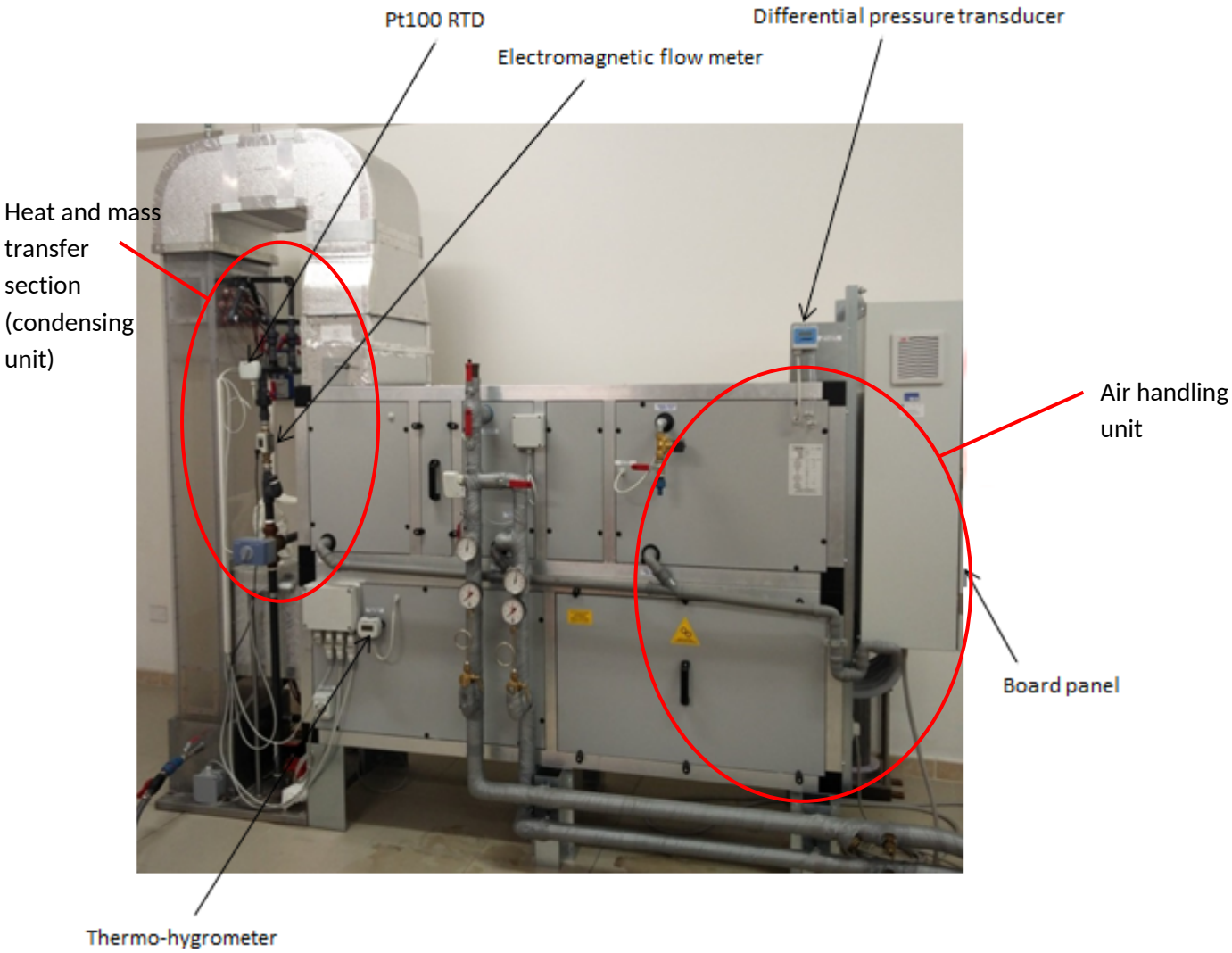
Reviewer #1: Major comments

1. ***I agree with the author that his system is an Evaporative Condenser but I still disagree with him to call it an Air handling unit (AHU). AHU never be called a condenser. AHU is a heat exchanger between chilled water and air or refrigerant and air. Please do search on AHU. My suggestion is to change the word "AHU" by "condensing unit".***

- *The system is made of the following units:*
    - *a condensing unit where the heat transfer from the electrical heaters (simulating the refrigerant) to air and water vaporization occur;*
    - *an air handling unit (named AHU) able to get the temperature and relative humidity set by the user at the entrance of the former condensing unit.*

They have been clearly indicated in the scheme of the experimental setup (Fig.1) and in the picture representing the test rig (Fig.2). Both illustrations are reported hereafter. No changes were made to the paper as the authors think that the difference between the units is already sufficiently described already in the present version.


HEAT AND MASS TRANSFER SECTION
Air out
Air flowmeter
WATER CHILLER
Deluge water
Electromagnetic water flowmeter
Electric heaters
Air in
Makeup water
Three-way valve
AIR HANDLING UNIT
Pump
Electric heater


Pt100 RTD
Differential pressure transducer
Electromagnetic flow meter
Heat and mass transfer section (condensing unit)
Air handling unit
Board panel
Thermo-hygrometer

The Design of Countercurrent Evaporative Condensers with the Hybrid Method

M. Fiorentino[a], G. Starace[b,*]

[a]*Decsa S.r.l., Via Cappelletta, Voghera 27058, Italy*
[b]*Department of Engineering for Innovation, University of Salento, via per Monteroni, Lecce 73100, Italy*
*Corresponding author: giuseppe.starace@unisalento.it
tel.: +39 0832 297753 fax: +39 0832 297777

HIGHLIGHTS

- Limitations to the of evaporative condensers performance evaluation were overcome
- The hybrid method was applied to the countercurrent evaporative condenser design
- A wide experimental campaign was carried out on a purposely built up test rig
- A 50% increase of water flow rate was found leading to a 14% heat transfer increase

The Design of Countercurrent Evaporative Condensers with the Hybrid Method

M. Fiorentino[a], G. Starace[b,*]

[a]*Decsa S.r.l., Via Cappelletta, Voghera 27058, Italy*
[b]*Department of Engineering for Innovation, University of Salento, via per Monteroni, Lecce 73100, Italy*
*Corresponding author: giuseppe.starace@unisalento.it
tel.: +39 0832 297753 fax: +39 0832 297777

Abstract

The heat and mass transfer in evaporative condensers are complex to model analytically and numerical simulations, when applied to multi-phase fluid dynamics in complex paths, often involve too high computational costs. Experimental campaigns at full scale of different heat transfer geometries and tube arrangements involve long lead times and high costs as well. The aim of the present work is to overcome the present limitations and to apply a new method to evaluate the overall performance of the countercurrent evaporative condensers, starting from the experimental, numerical or analytical data with a small scale approach. A test bench has been purposely designed and built up in order to reach and keep constant all the parameters determining the evaporative condenser heat transfer performance. In previous experimental contributions available in the literature, the air conditions were not controlled: here, an air handling unit placed before the evaporative condenser inlet allows to set up temperature and relative humidity of air in large ranges. An extended experimental campaign has been carried out to get affordable data to be used to find a relationship correlating the dry bulb temperature and relative humidity of air after its interaction with water and the condenser tubes surfaces, while all the parameters were set up and controlled. The regression function fits well the experimental data as the predicted values of temperature and relative humidity are characterized by a maximum percent deviation lower than 2.5 % and 4 % respectively. An iterative procedure was then implemented to determine the conditions of air going through the evaporative condenser in order to extend small scale results to full scale performance according to real geometries. The effect of the water flow rate on the cooling capacity was investigated and the results show that an increase of 50 % of the sprayed water leads to an increase of 14 % of the performance.

Nomenclature

AHU	Air Handling Unit
PID	Proportional Integral Derivative
RTD	Resistance Temperature Detector
$\dot{G}$	Flow rate [$m^3 \cdot h^{-1}$] for air, [$l \cdot min^{-1}$] for water
$\dot{m}$	Mass flow rate [$kg \cdot s^{-1}$]
n	Rows number
Q	Heat transfer rate [W]
R_{adj}^2	Adjusted coefficient of determination
RH	Relative humidity [%]
T_{db}	Dry bulb temperature [°C]
T_{wall}	Outer surface temperature of electric heaters[°C]
T_{water}	Water temperature [°C]

x	Specific humidity [kg·kg$_{dry\ air}^{-1}$]
Greek symbols	
$(\beta_0\ \beta_1\ \beta_2\ \beta_3\ \beta_4\ \beta_5\ \beta_6\ \beta_7\ \beta_8\ \beta_9\ \beta_{10}\ \beta_{11}\ \beta_{12}\ \beta_{13}\ \beta_{14}$	Regression coefficients
ε	Convergence tolerance
Subscripts	
$evap$	Evaporated
in	Inlet section
out	Outlet section
$setpoint$	Set conditions at the outlet of the air handling unit

1. Introduction

The evaporative cooling is a well-established technology used to improve the heat rejection, even if the related heat and mass transfer phenomena are not perfectly predictable.

This is why researchers are still making efforts to model the heat transfer in cooling towers, evaporative coolers and condensers.

The contributions in the literature can be classified as approaching the problem with:

- mathematical and numerical models [1-9];
- theoretical and experimental investigations [10-14];
- experimental studies [15-20].

Bykov et al. [1] proposed a method able to compute the water and air temperature variations and air enthalpy change in three different zones: a. the area above the tube bundle b. the tube bundle c. the area between the water basin and the tube bundle.

Webb [2] developed a unified theoretical treatment based on the Merkel hypothesis for cooling towers, evaporative coolers and condensers, since the air side heat and mass transfer phenomena are governed by the same process.

Qureshi and Zubair [3, 4] evaluated the effect of fouling on the cooling capacity of evaporative coolers and condensers with a mathematical model. Then Qureshi and Zubair [5] carried out a sensitivity analysis and detected the condensing temperature and inlet relative humidity as the main parameters affecting the evaporative condenser performance.

Acunha Jr and Schneider [6] simulated the air and water flows in evaporative condenser by commercial available CFD codes (Ansys Fluent), but without modeling the heat and mass transfer.

Jahangeer et al. [7] presented a numerical model for a single straight tube of an evaporative condenser, with water and air in crossflow configuration. They assumed a constant wall temperature, solved the equations using the finite difference technique and computed the heat transfer coefficients as well.

Fiorentino and Starace [8] carried out two-dimensional numerical analyses on the heat transfer core of an evaporative condenser, consisting in the portion of fluid between two staggered tubes.

They simulated the condensing refrigerant by assuming a constant wall temperature hypotesis. They studied the influence of air conditions and wall temperature on the heat transfer coefficient. Then Fiorentino and Starace [9] continued their analysis on the established water flow modes, by varying the water-to-air ratio and the tubes arrangement.

Leidenfrost and Korenic [10] implemented a mathematical model that simulated the evaporative condenser during steady state operation. Then Leidenfrost and Korenic [11] analyzed the system behavior at different air, water and refrigerant flow rates and under different ambient conditions. They carried out an experimental campaign in order to validate the results with experimental data.

Nasr and Hassan [12] designed an innovative evaporative condenser for small size refrigeration system application and studied its performance through experimental tests. They implemented a simplified theoretical model, validated with experimental results, for the heat and mass transfer phenomena.

Tissot et al. [13] studied the improvement effect of evaporative cooling on the COP (about 22 %) by spraying water on the heat transfer surface of a dry condensing unit. Then they proposed a numerical model whose results were in good agreement with experimental data.

Islam et al. [14] carried out an experimental campaign on an air conditioning system operating with an evaporative condenser. A theoretical model for a small segment of the condenser tube was described in detail and a good agreement with experimental data was observed.

Ettouney et al. [15] experimentally investigated the evaporative condensers performance: they noticed that the efficiency increased with the water to air flow ratio and obtained empirical relationships for the heat transfer coefficients.

Hajidavaloo and Eghtedari [16] compared the performance of an air conditioning system using dry and evaporative condensing units. The experimental tests showed that the power consumption can be decreased down to 20 % and the Coefficient of Performance can be improved around 50 %.

Liu et al. [17] studied the cooling performance, the heat transfer coefficient and the COP of an air conditioning system using an evaporative condenser at different compressor frequencies, dry bulb temperatures, air velocities and water flow rates.

Junior and Smith-Schneider [18] performed experimental analyses on a small scale evaporative condenser. They collected 40 samples and obtained an empirical relationship to predict the heat rejection as a function of the refrigerant and water temperatures, the dry and wet bulb temperatures and the water to air flow ratio. This analysis was carried out without any control on air conditions.

Fiorentino and Starace [19-22] designed and built up a test rig capable to measure and control all the parameters affecting the cooling capacity of evaporative condensers. An AHU placed before the heat and mass transfer test section allows setting up a wide range of air conditions. Experimental

tests at small scale were performed that allowed to make a sensitivity analysis depending on the inlet dry bulb temperature and relative humidity, the sprayed water flow rate and the temperature.

In the present work, a hybrid method adapted to the evaporative condensers design and optimization [23] used to extend the experimental tests (at small scale) to the overall system is described. The whole heat transfer domain is divided in micro-volumes and regression function is determined to describe their performance depending on working parameters.

The regression technique is based on data (either experimental, or numerical, or theoretical), referred to micro-volumes to be used to obtain predictor functions for each previously identified response variable depending on all the explanatory variables. An iterative procedure, taking into account the real geometry, allows then to evaluate the overall heat exchanger performance as an integration of different local contributions.

2. The experimental campaign

2.1 The setup

A scheme of the experimental setup with all components and measuring devices is illustrated in Fig.1.

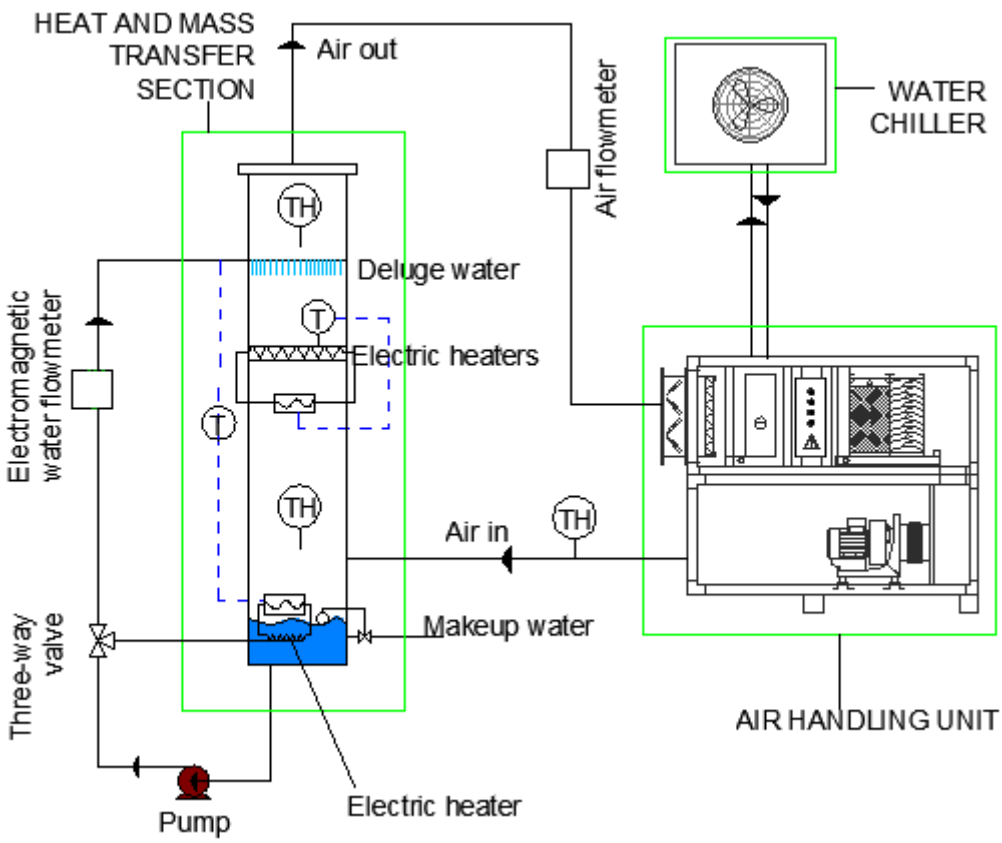


Figure 1. Scheme of the experimental setup and measurement equipment (TH: thermo-hygrometer, T: thermo-regulator)

The main components are the AHU, the water chiller and the heat and mass transfer section.

Electrical heaters and chilled water are used for heating and cooling, while the humidifying system consists of a high efficiency evaporative package.

The fan is driven by a Variable Frequency Drive to control the air flow rate.

The cooling water is provided by a chiller operating with R410A and condensed by air.

The specifications of the air handling unit and chiller are summarized in Table 1.

Table 1. Specifications of air handling unit and chiller

	Unit	Nominal Capacity
AHU heating section	kW	8
AHU cooling section	kW	3
Water chiller	kW	4

The heat and mass transfer section is the core of the test bench. Four transparent panels are assembled to form a rectangular channel of overall dimensions 450x200x1800 mm^3. The heat transfer geometry is made up of staggered tubes mounted on plates so that different arrangements can be tested by replacing only them.

The condensing refrigerant is simulated by three electrical heaters equipped with RTD-Pt100 sensors connected with a PID for the outer surface temperature control. The remaining tubes establish the water and air countercurrent flows typical of an evaporative condenser.

Water is sent from the basin to three pipes distributing it on the electrical heaters. The water flow rate is regulated by selecting the velocity level of the circulating pump and through a three-way valve, while its temperature is controlled by an electrical heater in the basin.

The air is totally recirculated from the handling unit to the mass transfer section, where its enthalpy increases due to the interaction with the electrical heaters and water vaporization.

The measuring equipment with the maximum relative uncertainty is listed in Table 2.

The test rig is shown in Fig.2.

Table 2. Measuring equipment used during the experimental accuracy with the maximum uncertainty

Measurement	Measuring device	Uncertainty
Air flow rate	Differential pressure transducer	±5%
Air dry bulb temperature	Thermo-hygrometer	±1%
Air relative humidity	Thermo-hygrometer	±4%
Water flow rate	Electromagnetic flow meter	±5%
Water temperature	RTD-Pt100	±1%
Surface temperature of electrical heaters	RTD-Pt100	±1%

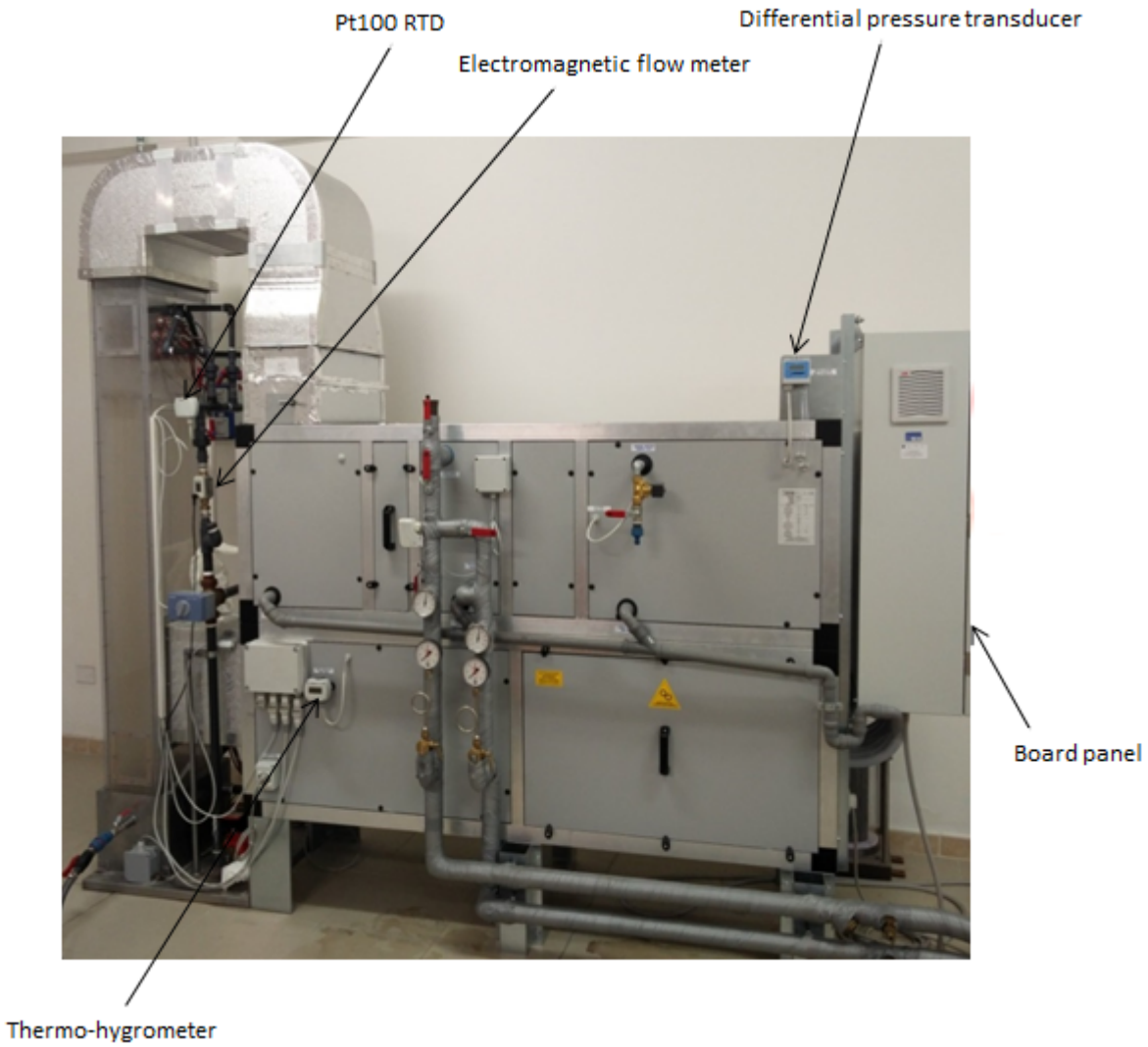


Figure 2. Test rig

2.2 The procedure

The controlled parameters, set by user on the board panel are:

- air flow rate $\dot{G}_{air}$;
- air dry bulb temperature $T_{db,setpoint}$ and relative humidity $RH_{setpoint}$ at the outlet of the handling unit;
- water flow rate $\dot{G}_{water}$ and temperature T_{water};
- outer surface temperature of the electrical heaters T_{wall}.

When steady operating conditions are reached, the thermo-hygrometers TH illustrated in Fig.1 in the heat and mass transfer section read the following parameters:

- air dry bulb temperature $T_{db,in}$ and relative humidity RH_{in} before the interaction with the electrical heaters (different from those set by the user, because of the air cooling occurring in the channel portion between the basin and the electrical heaters);

- air dry bulb temperature $T_{db,out}$ and relative humidity RH_{out} after the interaction with the electrical heaters.

Each test at the same operating conditions is repeated to collect a number of samples to be treated statistically in order to ensure the affordability of experimental data.

3. The method

The experimental results are used to obtain for each response variable a predictive relationship depending on all the explanatory variables [23]. Tests are carried out keeping constant the air flow rate and the wall temperature of tubes equipped with the electrical heaters, while the other parameters are varied in the ranges indicated in Table 3.

The water temperature ranges from the minimum value depending on the physical phenomena occurring in the heat and mass transfer section and the wall temperature.

Table 3. Test operating conditions

	Unit	
$\dot{G}_{air}$	m^3h^{-1}	800
T_{wall}	°C	28
$\dot{G}_{water}$	$l \cdot min^{-1}$	[1.2÷4.2]
T_{water}	°C	[25÷27]
$T_{db,setpoint}$	°C	[19÷30]
$RH_{setpoint}$	%	[50÷90]

The explanatory variables are:

- air dry bulb temperature before the interaction with the electrical heaters;
- relative humidity before the interaction with the electrical heaters;
- water flow rate;
- water temperature.

The response variables are:

- air dry bulb temperature after the interaction with the electrical heaters;
- relative humidity after the interaction with the electrical heaters.

Each response variable is determined as the regression coefficients:

$$\left(\beta_0\, \beta_1\ \beta_2\ \beta_3\ \beta_4\ \beta_5\ \beta_6\ \beta_7\, \beta_8\, \beta_9\ \beta_{10}\ \beta_{11}\ \beta_{12}\ \beta_{13}\, \beta_{14}\right)$$

multiplied by constant, linear, interaction and squared terms as follows:

$$\left(1\ T_{db,in}\ RH_{in}\ \dot{G}_{water}\ T_{water}\ \ T_{db,in}RH_{in}\ T_{db,in}\dot{G}_{water}\ T_{db,in}T_{water}\ RH_{in}\dot{G}_{water}\ RH_{in}T_{water}\ \dot{G}_{water}T_{water}\ T_{db,in}^{2}\ RH_{in}^{2}\ \dot{G}_{water}^{2}\ T_{water}^{2}\right)^{T}$$

The samples generated by the experimental campaign are summarized in Table 4 and are used as control points for the regression.

1

Table 4. Experimental samples

N. test	Explanatory variables				Response variables	
	$\dot{G}_{water}$ [l·min^{-1}]	T_{water} [°C]	$T_{db,in}$[°C]	RH_{in} [%]	$T_{db,out}$ [°C]	RH_{out} [%]
1	1.20	25.8	21.60	57.00	22.20	62.60
2	1.20	26	21.80	67.20	22.40	71.30
3	1.20	25.9	21.85	76.25	22.40	79.50
4	1.20	26.2	22.30	84.15	22.75	86.75
5	1.20	26.1	22.60	91.75	23.15	92.55
6	1.20	25.9	18.40	75.20	19.10	78.80
7	1.20	26	18.90	83.20	19.70	85.90
8	2.00	26.2	18.30	86.40	19.30	88.70
9	2.00	25.8	19.00	93.60	19.90	94.60
10	2.00	26.2	17.85	78.40	19.05	82.35
11	2.00	25.4	27.90	66.10	27.90	72.20
12	2.00	25.1	26.95	66.65	27.05	72.20
13	2.00	25.8	27.80	74.40	27.90	77.80
14	2.00	24.7	25.90	54.80	26.10	61.60
15	2.00	24.8	26.30	66.40	26.40	71.60
16	2.00	25.2	26.80	76.80	26.80	79.70
17	2.00	26	26.60	83.70	26.80	85.80
18	2.00	24.8	22.30	98.40	22.80	96.50
19	3.20	25.15	21.50	72.30	22.40	75.85
20	3.20	25.3	21.60	92.10	23.00	89.20
21	3.20	24.9	22.15	96.70	22.90	95.50
22	3.20	25	20.30	70.75	22.20	69.50
23	3.20	25.05	22.10	65.05	23.65	66.45
24	3.20	25	22.30	75.10	23.70	75.00
25	3.20	24.8	23.20	90.90	24.30	88.40
26	3.20	25.4	23.60	95.20	24.50	92.70
27	3.20	25.1	17.30	78.20	19.10	77.80
28	3.20	25	17.55	82.20	19.15	83.00
29	3.20	25.05	17.85	89.95	19.40	88.55
30	3.20	25	18.40	96.45	19.50	96.00
31	3.2	26.3	18.75	97.35	19.85	97.05
32	3.2	25.8	18.40	97.45	19.50	96.95
33	3.2	26.1	17.70	94.80	19.30	92.20
34	3.2	26	17.70	83.20	19.20	84.70
35	3.2	26	17.60	77.15	19.20	78.70
36	3	25.8	21.20	63.90	22.00	68.30
37	3	26	21.20	73.70	22.40	75.55
38	3	25.9	21.30	82.40	22.40	82.40
39	3	25.9	21.70	90.50	22.60	89.00
40	3	26.1	22.30	96.60	23.00	95.50
41	3	26.2	22.10	64.10	23.60	65.80
42	3	26	22.75	73.55	23.95	73.70
43	3	25.95	23.20	81.45	24.15	81.40
44	3	26.1	23.70	90.50	24.50	88.45
45	3	26	24.00	96.60	24.60	94.20
46	4	25.1	21.90	67.25	23.70	67.75
47	4	25	22.50	74.30	24.00	74.90
48	4	25.1	22.80	84.40	24.10	82.10
49	4	25.4	23.70	91.10	24.70	89.10
50	4	24.7	23.50	91.80	24.40	89.10
51	4	25.5	24.00	97.15	24.70	95.40
52	4	25	20.60	67.70	22.30	70.90
53	4	25	21.10	72.90	22.50	75.00
54	4	24.9	21.00	83.45	22.50	82.15
55	4	24.9	21.50	91.90	22.80	89.30
56	4	24.6	21.80	96.80	22.60	95.20
57	4	25.2	22.00	97.45	22.85	95.70
58	4	25	17.60	85.50	19.50	84.50
59	4	25	18.10	91.00	19.80	89.95
60	4	24.95	18.55	97.10	19.90	96.40
61	4	26.1	18.00	85.20	19.70	86.10
62	4	26	18.30	89.90	19.90	90.50

63	4	25.9	18.90	95.20	20.10	95.90
64	4.2	25.9	20.65	68.55	22.30	71.50
65	4.2	26.1	21.10	74.50	22.60	77.30
66	4.2	26.1	21.50	82.75	23.00	82.65
67	4.2	25.8	21.70	91.00	22.90	89.90
68	4.2	26	22.80	96.50	23.40	96.30
69	4	24.9	25.30	63.40	26.10	66.40
70	4	24.9	25.60	70.50	26.10	73.40
71	4	25.9	26.30	81.20	26.80	81.60
72	4	26.6	26.80	90.30	27.20	88.50

Table 5. Regression coefficients for the response variables

	$T_{db,out}$	UR_{out}
β_0	$2.7487 \cdot 10^{1}$	$-3.4397 \cdot 10^{2}$
β_1	$6.1476 \cdot 10^{-1}$	4.0274
β_2	$-2.2048 \cdot 10^{-2}$	1.0428
β_3	2.1846	-4.7227
β_4	-1.7599	$2.3766 \cdot 10^{1}$
β_5	$1.0287 \cdot 10^{-3}$	$-7.5346 \cdot 10^{-3}$
β_6	$-2.5763 \cdot 10^{-2}$	$-6.2088 \cdot 10^{-2}$
β_7	$2.3708 \cdot 10^{-2}$	$-2.1677 \cdot 10^{-1}$
β_8	$-3.2956 \cdot 10^{-3}$	$-1.2428 \cdot 10^{-2}$
β_9	$2.8714 \cdot 10^{-3}$	$-1.1723 \cdot 10^{-2}$
β_{10}	$-3.0326 \cdot 10^{-2}$	$1.1442 \cdot 10^{-1}$
β_{11}	$-8.1315 \cdot 10^{-3}$	$5.2008 \cdot 10^{-2}$
β_{12}	$-4.7430 \cdot 10^{-4}$	$1.9950 \cdot 10^{-3}$
β_{13}	$-5.8362 \cdot 10^{-2}$	$6.3344 \cdot 10^{-1}$
β_{14}	$2.0904 \cdot 10^{-2}$	$-3.4848 \cdot 10^{-1}$

Table 6. Adjusted coefficients of determination

	$T_{db,out}$	RH_{out}
R^2_{adj}	0.9948	0.9891

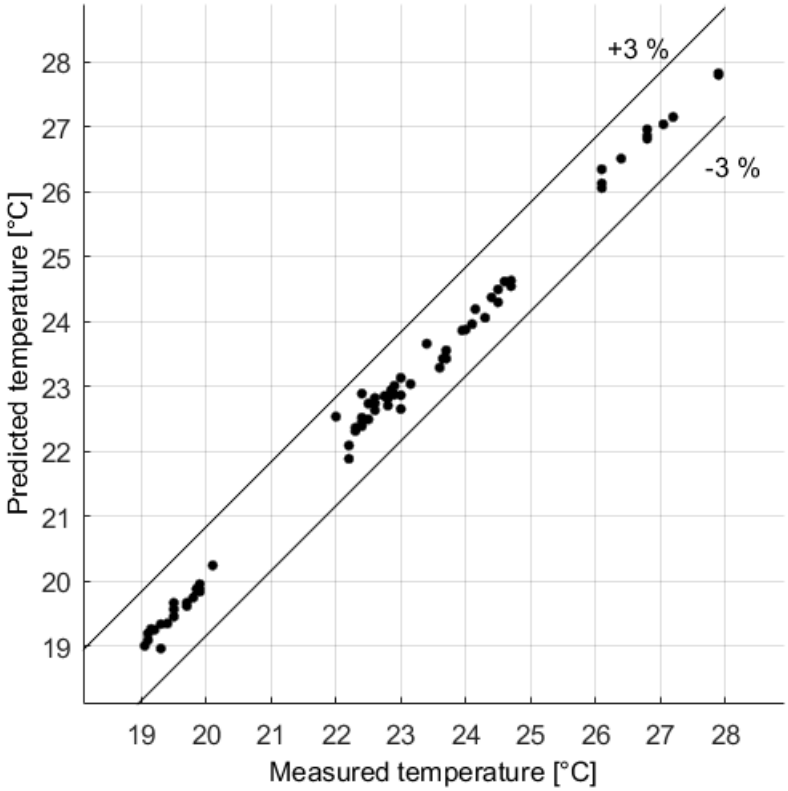
The calculated regression coefficients for each response variable and the adjusted coefficient of determination can be read in Table 5 and 6 respectively. The relationships obtained with the regression fit data within ±3 % for temperature and ±4 % for relative humidity, as shown in Fig. 3(a) and 3(b). The adjusted coefficients of determination are close to 1, and this guarantees that the regression is characterized by a good level of accuracy for both the response variables. The regression coefficients allow to determine the conditions reached by the air after crossing a tubes row. The evaporative condenser is schematized as a sequence of rows, as illustrated in Fig.4.

The rows are numbered from the bottom upwards, along the airflow direction.

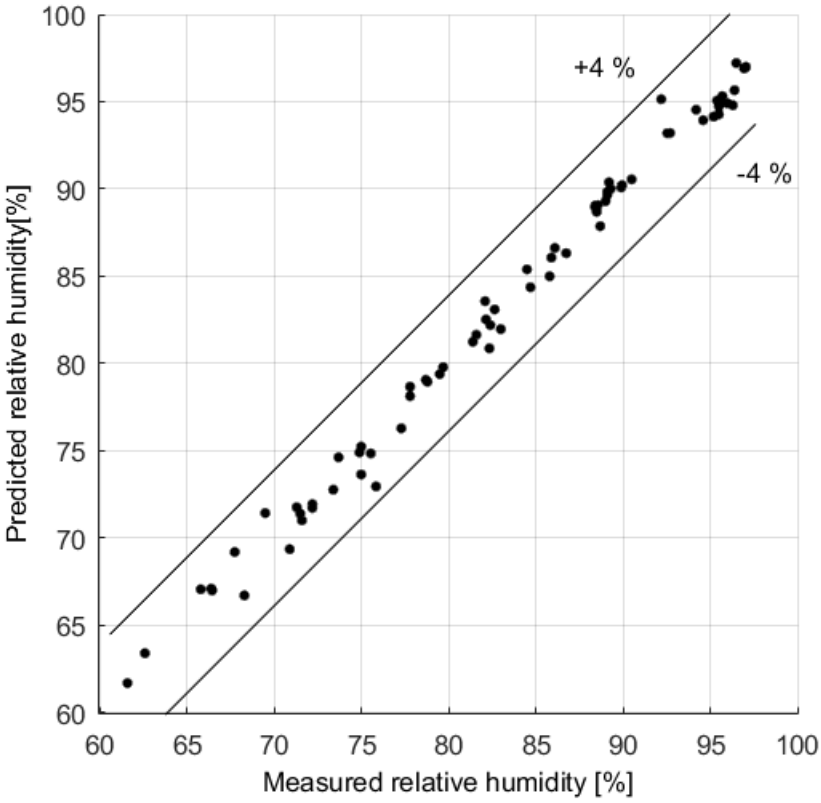
The air operating conditions at the inlet of a micro-volume correspond to those at the outlet of the previous one. The water temperature is assumed constant.

The air and water conditions at the bottom and top, respectively, are known, i.e.:

- dry air mass flow rate $\dot{m}_{dryair}$;
- dry bulb temperature and relative humidity at the inlet of the first row $T_{in}(1)$,$RH_{in}(1)$;
- water mass flow rate at the inlet of the nth row $\dot{G}_{water,in}(n)$.



(a)



(b)

Figure 3. Comparison between predicted and experimental values (a) Temperature (b) Relative humidity.

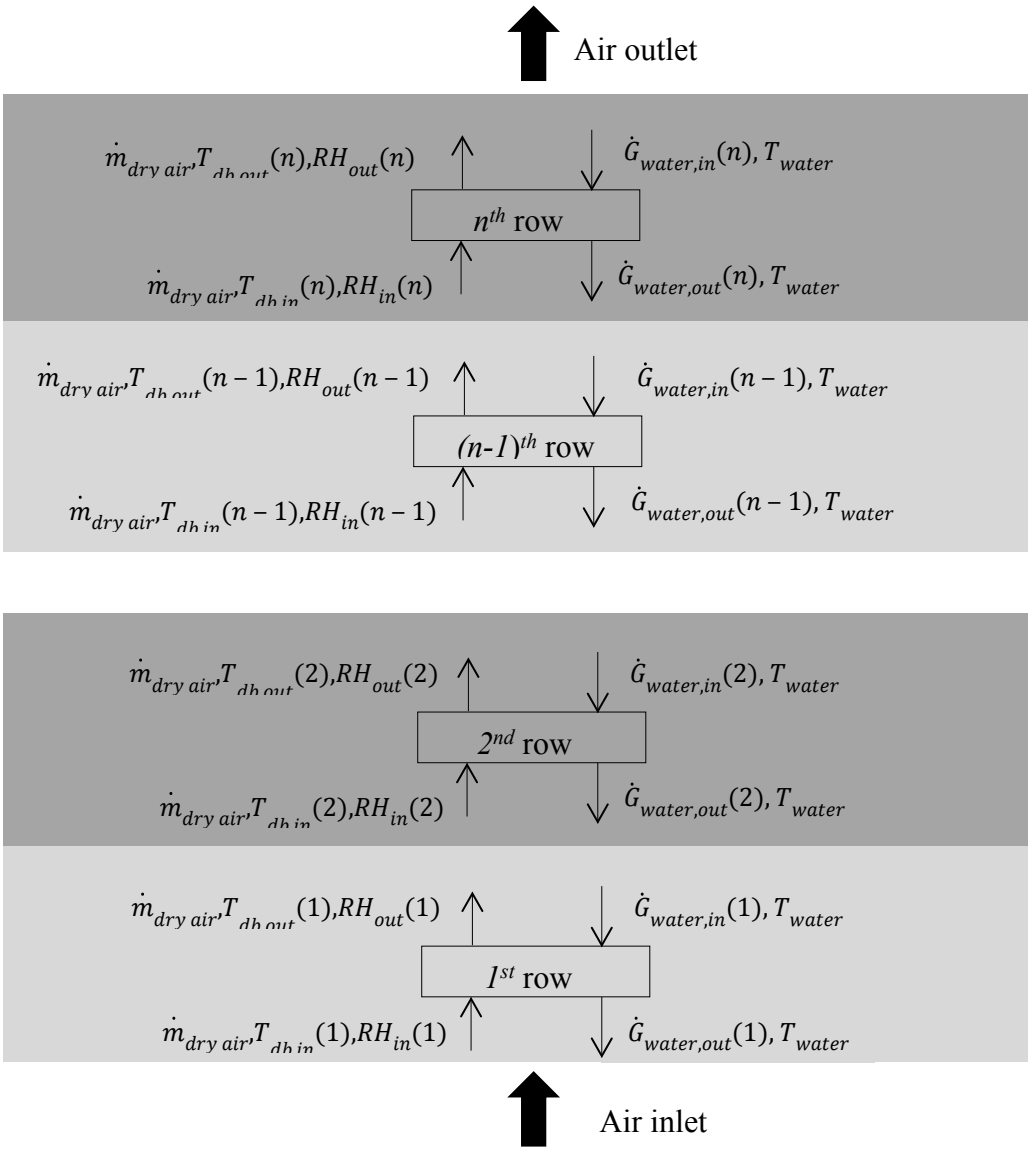


Figure 4. Schematization of the evaporative condenser in the hybrid method approach

The algorithm, implemented in Matlab, works as follows:

1st row

Considering the first row, only some of the explanatory variables are known; actually, the water flow rate at the inlet $\dot{G}_{water,in}(n)$ depends on the amount of vaporized water and therefore it has to be assumed.

$$\dot{G}_{water,in}(1) = \dot{G}_{water,in}(n) \cdot C_1 \tag{1}$$

Then the response variables $T_{out}(1), RH_{out}(1)$, $\dot{G}_{water,out}(1)$ can be calculated with the regression coefficients.

2nd row

The known parameters for the second row are the air dry bulb temperature and relative humidity at the inlet section and the water flow rate at the outlet.

$$T_{in}(2) = T_{out}(1) \tag{2}$$

$$RH_{in}(2) = RH_{out}(1) \tag{3}$$

$$\dot{G}_{water,out}(2) = \dot{G}_{water,in}(1) \tag{4}$$

The water flow rate at the inlet has been assumed:

$$\dot{G}_{water,in}(2) = \dot{G}_{water,out}(2) \cdot C_2 \tag{5}$$

and the response variables $T_{out}(2)$,$RH_{out}(2)$ are calculated. The specific humidity variation allows to compute the water flow rate at the outlet section:

$$\dot{G}_{water,out,\,computed}(2) = \dot{G}_{water,in}(2) - m_{dry,air}(x_{out}(2) - x_{in}(2)) \tag{6}$$

The difference $(\dot{G}_{water,out,\,computed}(2) - \dot{G}_{water,out}(2))$ is determined and the coefficient C_2 is updated until the convergence is not reached.

n^{th} row

The calculation procedure explained for the second row is repeated for all the consecutive rows.

The calculated water flow rate at the inlet of the last row:

$$\dot{G}_{water,in,computed}(n) = \dot{G}_{water,out}(n) \cdot C_2 \tag{7}$$

is compared to the design value and until the convergence condition (Eq.8) is not reached a new value of water flow rate at the inlet of the first row has to be assumed, by varying the coefficient C_1.

$$\dot{G}_{water,in,\,computed}(n) - \dot{G}_{water,in}(n) < \varepsilon \tag{8}$$

4. Results

The number of rows, the flow rate and temperature of water distributed over the electric heaters, the air temperature and relative humidity at the inlet section for the discussed case study are summarized in Table 7.

Table 7. Operating conditions for the case study

	Unit			
n	-		12	
T_{wall}	°C		28	
		Case A	Case B	Case C
$\dot{G}_{water}$	l·min^{-1}	1.5	3	4
T_{water}	°C		25	
$\dot{G}_{air}$	m^3·h^{-1}		800	
$T_{db,in}$	°C		18	
RH_{in}	%		60	

The conditions of air, the heat transfer rate and the amount of vaporized water for each row are shown in Fig.5. The dry bulb temperature and absolute humidity increase when air crosses the rows

and as expected, the heat transfer rejected from the electrical heaters decreases from 1 kW to 0.5 kW in the outlet section.

The relative humidity in the last row is 90%, and, as consequence, other rows would be needed to reach saturation.

The overall heat transfer rate and the evaporated water flow rate are summarized in Table 8: the amount of evaporated water represents about 11% of the sprayed one; it cannot be therefore considered negligible and accurate calculations have to take it into account.

Then a comparison of the evaporative condenser performance for different sprayed water flow rates has been made, by keeping constant the other operating conditions. As in Fig. 6, the heat transfer improves with the water flow rate for the first seven rows, while in the following rows a trend reversal occurs, due to the higher relative humidity of the incoming air.

The overall heat transfer rate increases with the water flow rate, as summarized in Table 9.

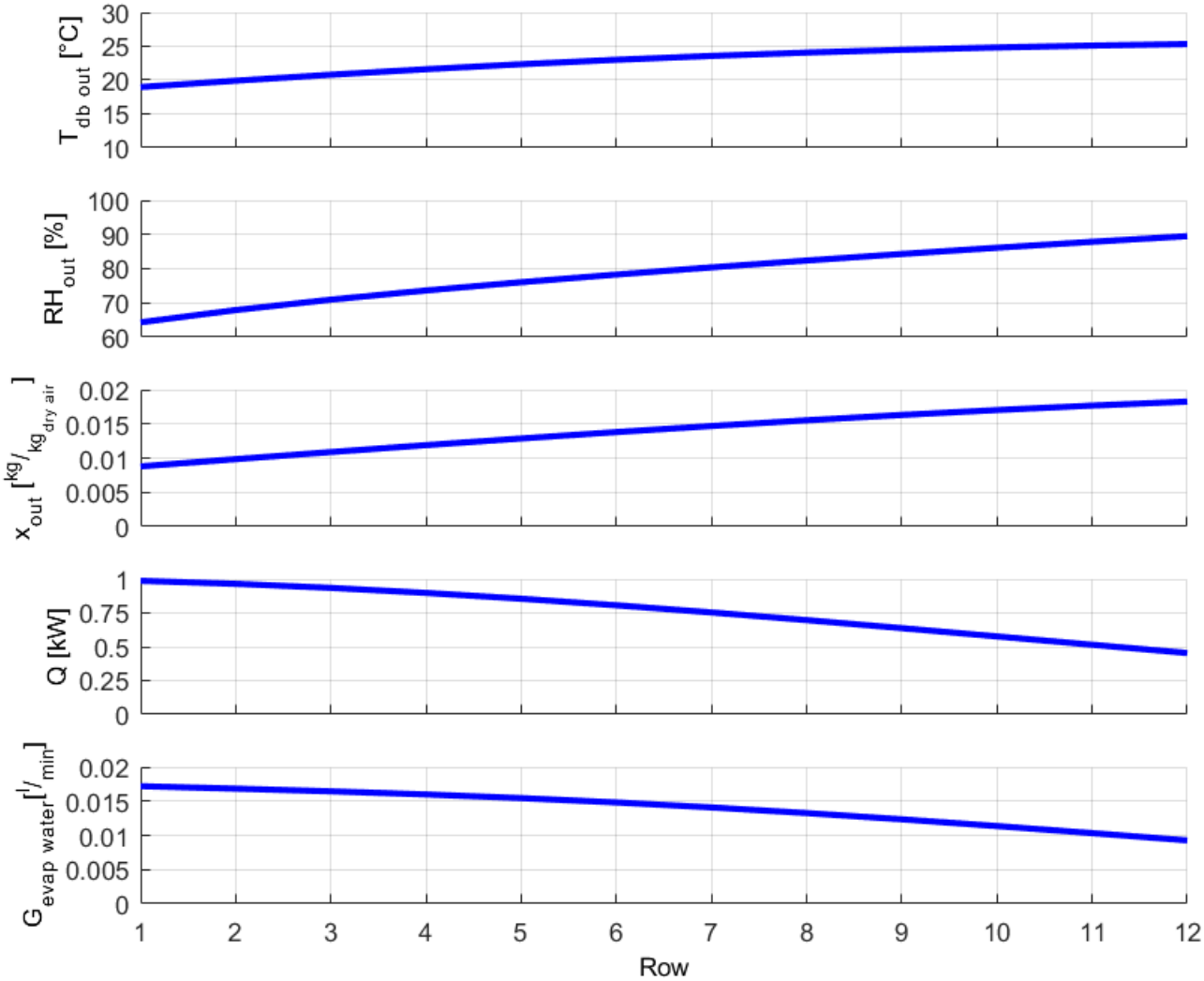


Figure 5. Evolution of physical quantities through the rows (Case A)

Table 8. Cooling capacity and evaporated water (Case A)

Parameter	Unit	Value
Evaporated water flow rate	$l \cdot min^{-1}$	0.167
Heat transfer rate	kW	9.079

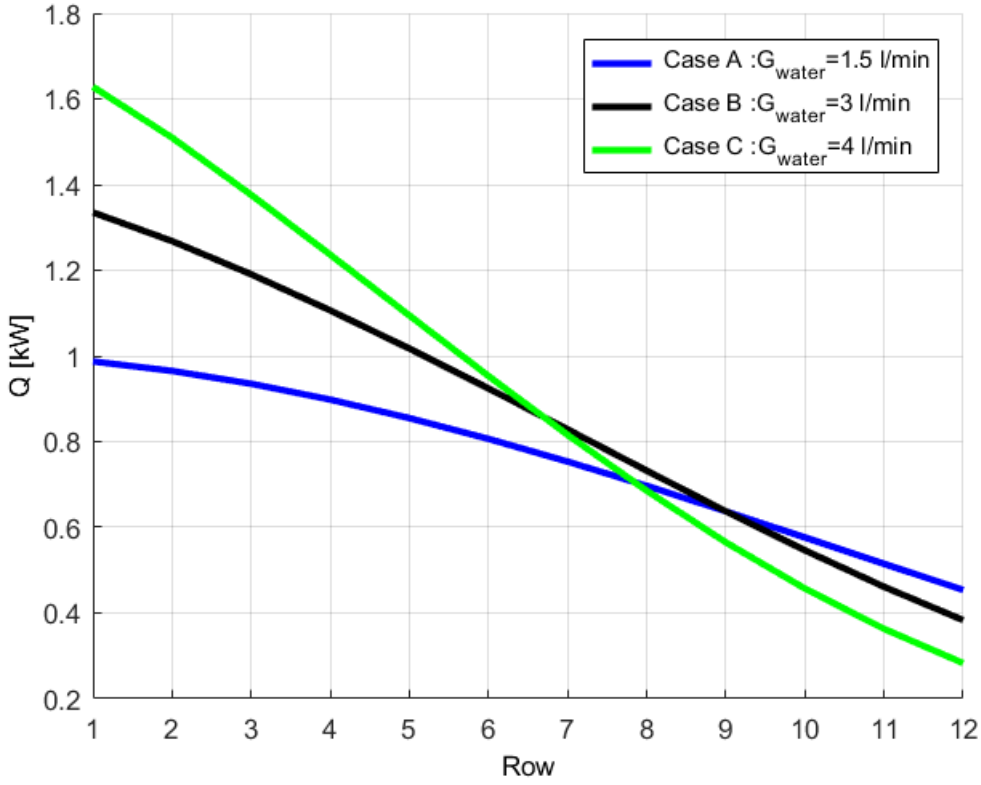


Figure 6. Evolution of the heat transfer rate through the rows at different water flow rates.

Table 9. Cooling capacity at different water flow rates.

	Water flow rate [$l \cdot min^{-1}$]	Cooling capacity [kW]
Case A	1.5	9.1
Case B	3	10.4
Case C	4	11.0

Fig.7 shows for the three cases of Tables 7 and 9 how the enthalpy variations decrease, in terms of both sensible and latent heat when air approaches to the exit.

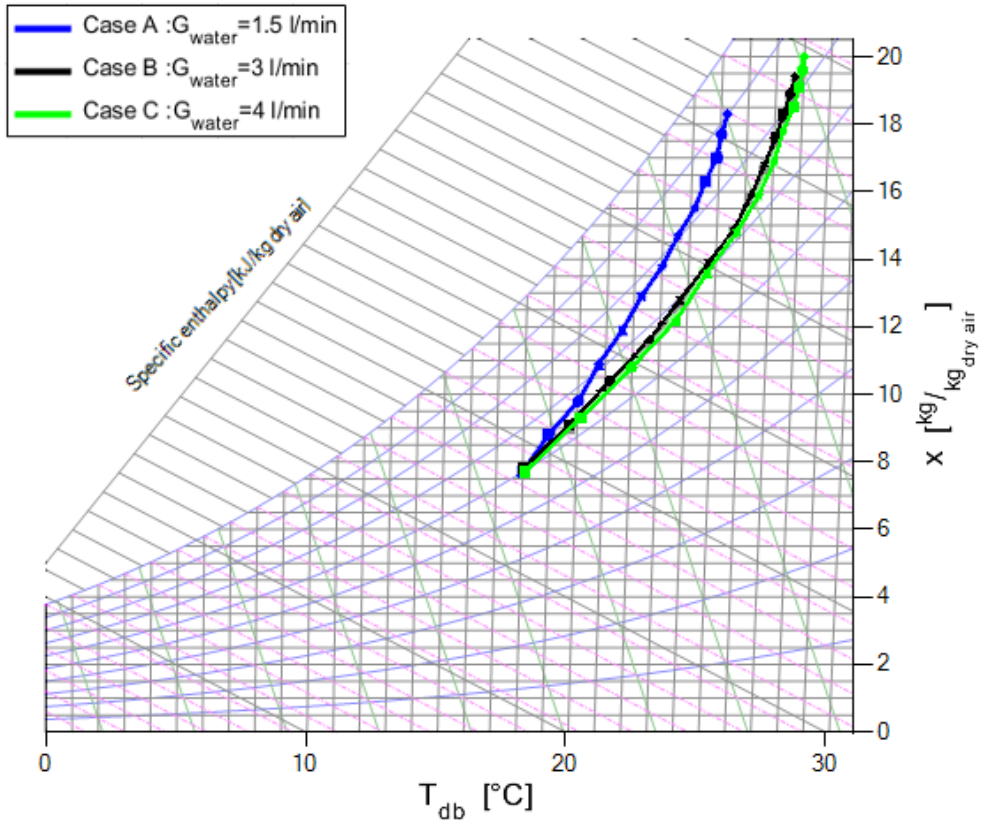


Figure 7. Air processes plotted on the psychrometric ASHRAE chart (sea level)

5. Conclusions

A new method to evaluate the overall performance of an evaporative condenser starting from experimental results at small scale was presented.

Although many research contributions to the evaporative condenser modeling and comprehension have been found in the literature, some limitations to their wide application were found as:

- analytical and numerical studies involve high computational costs;
- experimental campaigns at full scale lead long times and high costs.

The aim of the application of the hybrid method was to evaluate the performance of an evaporative condenser with reduced costs. Its key points were:

I. <u>performing experimental tests at small scale</u>

A purposely designed test rig capable of controlling all the parameters affecting the heat transfer rate was used to carry out the necessary tests, by varying all the parameters except the air flow rate and wall temperature of the condenser heat transfer surfaces.

The air was cooled in the vertical portion that precedes the tube bundle and thus the air conditions before and after the interaction with the heat transfer section were measured in order to estimate the heat released to air.

II. <u>obtaining correlations for the response variables from the collected data</u>

The experimental data represent the control points necessary to build up for the regression relation describing the response variables (outlet relative humidity and temperature) as a function of the explanatory variables (relative humidity and temperature at the inlet, water flow rate and temperature).

III. <u>scale up results to evaluate the overall performance</u>

The evaporative condenser was divided in sub-domains, each of them representing a row: the regression coefficients were used to determine the air conditions after the heat transfer and an iterative procedure was implemented to calculate the amount of vaporized water.

The relations obtained from the data set (72 samples) allowed to predict the temperature and relative humidity at the outlet section with a maximum percent deviation from the experimental data lower than 2.5 % and 4% respectively.

The evolution of physical quantities through the rows has been obtained and the air processes have been plotted on the psychometric chart. The influence of the sprayed water flow rate on the evaporative condenser performance has been investigated as well: an increase of 50 % of the water flow rate causes an increase of 14 % of the cooling capacity.

The method represents a tool to support evaporative condenser accurate design, as it can suggest the geometrical characteristics (i.e. the number of rows) that ensure the air saturation and can be used starting from data sets originated either by analytical, or numerical or experimental investigations.

In the future, further data will be collected by varying the air flow rate and outer surface temperature of the condenser tubes, in order to reach a full description of its behavior. Different heat transfer geometries and tubes arrangements [22] will be characterized by different regression coefficients and the method implemented for each of them will allow comparing the corresponding performance.

References

[1] A. V. Bykov, V. A. Gogolin, N. V. Tovaras, Investigation of heat, mass transfer and fluid flow characteristics in evaporative condensers, Int. J. Refrig. 7 (6) (1984) 342-347.

[2] R. L. Webb, A unified theoretical treatment for thermal analysis of cooling towers, evaporative condensers and fluid coolers. Presented at Conference *ASHRAE transactions* (vol. 90(2) (1984) 398-415) http://www.techstreet.com/ashrae/standards/kc-84-07-3-rp-322-a-unified-theoretical-treatment-for-thermal-analysis-of-cooling-towers-evaporative-condensers-and-fluid-coolers?product_id=1828942

[3] B. A. Qureshi, and S. M. Zubair, The impact of fouling on performance evaluation of evaporative coolers and condensers, International Journal of Energy Research 29 (14) (2005) 1313-1330.

[4] B. A. Qureshi, and S. M. Zubair, A comprehensive design and rating study of evaporative coolers and condensers. Part I. Performance evaluation, International Journal of Refrigeration 29 (4) (2006) 645-658.

[5] B. A. Qureshi, and S. M. Zubair, A comprehensive design and rating study of evaporative coolers and condensers. Part II. Sensitivity analysis, International Journal of Refrigeration 29 (4) (2006) 659-668.

[6] I. C Acunha Jr, and P. S. Schneider, Numerical Simulation of Air-Water Flows in an Evaporative Condenser, Thermal Engineering 8 (1) (2009) 24-30.

[7] K.A. Jahangeer, A.A.O Tay, M.R. Islam, Numerical investigation of transfer coefficients of an evaporatively-cooled condenser, Applied Thermal Engineering 31 (10) (2011) 1655-1663.

[8] M. Fiorentino, and G. Starace, A Numerical Model to Investigate Evaporative Condensers Behaviour at Tube Scale. In: ASME 2014 12th Biennial Conference on Engineering Systems Design and Analysis. American Society of Mechanical Engineers (2014) V003T12A015-V003T12A015.

[9] M. Fiorentino, and G. Starace, Numerical investigations on two-phase flow modes in evaporative condensers, Applied Thermal Engineering 94 (2016) 777-785.

[10] W. Leidenfrost, and B. Korenic, Analysis of evaporative cooling and enhancement of condenser efficiency and of coefficient of performance, Heat and Mass Transfer 12 (1) (1979) 5-23.

[11] W. Leidenfrost, and B. Korenic, Evaporative cooling and heat transfer augmentation related to reduced condenser temperatures, Heat Transfer Engineering 3 (3-4) (1982) 38-59.

[12] M. M. Nasr, and M. S. Hassan, Experimental and theoretical investigation of an innovative evaporative condenser for residential refrigerator, Renewable energy 34 (11) (2009) 2447-2454.

[13] J. Tissot, P. Boulet, F. Trinquet, L. Fournaison, M. Lejeune, F. Liaudet, Improved energy performance of a refrigerating machine using water spray upstream of the condenser, International Journal of Refrigeration, 38 (2014) 93-105.

[14] M. R. Islam, K. A. Jahangeer, K. J. Chua, Experimental and numerical study of an evaporatively-cooled condenser of air-conditioning systems, Energy 87 (2015) 390-399.

[15] H. M. Ettouney, H.T. El-Dessouky, W. Bouhamra, B. Al-Azmi, Performance of evaporative condensers, Heat Transfer Engineering, 22 (4) (2001) 41-55.

[16] E. Hajidavalloo, and H. Eghtedari, Performance improvement of air-cooled refrigeration system by using evaporatively cooled air condenser, International Journal of Refrigeration, 33 (5) (2010) 982-988.

[17] H. Liu, Q. Zhou, Y. Liu, P. Wang, D. Wang, Experimental study on cooling performance of air conditioning system with dual independent evaporative condenser, International Journal of Refrigeration, 55 (2015) 85-92.

[18] I. C. A. Junior, P. Smith-Schneider, Consolidated experimental heat and mass transfer database for a reduced scale evaporative condenser, International Journal of Refrigeration, 66 (2016) 21-31.

[19] M. Fiorentino, and G. Starace, Numerical and Experimental Performance Analysis of Evaporative Condensers, Energy Procedia, 101 (2016) 26-33.

[20] M. Fiorentino, and G. Starace, Experimental Investigations on Evaporative Condensers Performance. In: 50th AICARR International Conference, Matera 10-11 May 2017, to be published on Energy Procedia.

[21] M. Fiorentino, and G. Starace, Experimental Investigations on Air Side Heat and Mass Transfer Phenomena in Evaporative Condensers, International Journal of Heat and Technology 35 (1) (2017) S399-S404.

[22] M. Fiorentino, and G. Starace, Sensitivity Analysis of Evaporative Condensers Performance Using an Experimental Approach, Energy Procedia, 126 (2017) 345-352.

[23] G. Starace, M. Fiorentino, M. P. Longo, E. Carluccio, A hybrid method for the cross flow compact heat exchangers design, Applied Thermal Engineering, 111 (2017) 1129-1142.