Development of a design numerical model of a hybrid cooler

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Abstract

This paper presents the development of the numerical model of a hybrid cooler. It is based on a modular and generic coil geometry that deals with any staggered coil and pipe arrangement. Particular attention is given to model the spray water system and to the calculation of water evaporation on the coil surface. It is validated by monitored data from a pilot system installation in which a commercial hybrid cooler is operated under typical summer south European working conditions. The numerical model has an average error of 7 % for the rejected heat and 0.1 % for the fan power consumption. The model is compatible with TRNSYS simulation software. It can be used for design and product development purposes by HVAC manufactures and thermal engineers.

1. Introduction

An efficient and cost-effective heat rejection system is a key requisite of any cooling system. Two categories of air-based condenser units are typically distinguished: dry coolers (DCs) and wet cooling towers (WCTs).

Thanks to the evaporation of water deposited on the coil surface, WCTs can reach an outlet water temperature lower than dry-bulb ambient temperature. Their average specific nominal consumption is therefore lower (0.017 kW_{el}/kW_t) when compared to DCs (0.033 kW_{el}/kW_t) (D'Antoni et al., 2014). Conversely, they are characterized by higher operation (e.g. water consumption) and maintenance (e.g. legionella growth risk) costs (D'Antoni et al., 2014). On the other hand, DCs are relatively cheaper (22-27 €/kW_t against 49-107€/kW_t) and lighter (97-185 kg/m^2 against 208-376 kg/m^2) than WCTs (D'Antoni et al., 2014).

Both WCTs and DCs have technological limitations that can turn into important economic issues up to the extent to prevent their installation in certain climatic conditions. These problems can be partly overcome by hybrid coolers (HCs). A hybrid cooler is a dry cooler that wets the coil surface by spraying water upwards in a co-current direction with the airflow. Thanks to this solution, the use of water can be largely limited with respect to WCTs by spraying water only when necessary, thus achieving at the same time electricity savings compared to DCs due to fan operation.

HCs are not a new concept and several studies have been produced in the past 30 years (Sen, 1973; Yang and Clark, 1975; Nakayama et al., 1988; Dreyer et al., 1992). An extended literature review (Romeli, 2014) has revealed that numerical models of hybrid cooler still show room for further improvements.

- The calculation of the surface wettability is in some cases estimated through empirical correlations valid only for some specific geometries and in other fixed as a constant.
- None of the reviewed models has a modular definition of coil geometry.
- The level of detail in the control volume definition is quite diversified, ranging from the control volume identified as the whole coil, a single row, a single pass in a row or a fraction of a single tube.
- Effectiveness-NTU methods are more suitable for heat and mass transfer problems when inlet fluid conditions are imposed and for a given coil geometry, whereas mean-log enthalpy difference (LMED) can be successfully used for design purposes.

The ambition of the present paper is to present the development of a hybrid cooler numerical model characterized by:

- a modular and generic coil geometry that deals with any staggered coil and pipe arrangement;
- a model for the spray water system specifically conceived for hybrid coolers;
- a detailed calculation of wettability factors for fins and tubes at each row in which specificities of coil geometry, nozzles characteristics and position are taken into account;
- dedicated control strategies to regulate the fan's speed and the amount of sprayed water.

The numerical code is compatible with TRNSYS (Klein et al., 2010), which allows for the assessment the performance of a hybrid cooler in a whole system simulation. By reviewing already existing TRNSYS models ("Types"), we can observe that they lack:

- the possibility of specifying a generic coil geometry (e.g. Type 32);
- fins surfaces are assumed fully wet (e.g. Type 51, Type 510) and the use of the wettability factor concept;
- a clear focus on the control strategy for fans and spray water system operations.

2. Heat and mass transfer balances

2.1 Governing equations and infinitesimal control volume

The heat and mass transfer problem in a hybrid cooler can be studied on an infinitesimal control volume dA=b×dl (Fig. 1). It is characterized by the presence of three fluids: (1) process fluid flowing in coil pipes (water or a water-glycol mixture), (2) water sprayed by nozzles onto the coil surface and (3) moist air passing across both fins and tubes. For each of these a subsystem is identified on which heat balance and mass conservation is written. In order to do this, a list of assumptions and simplifications are necessary and in particular:

- The system is in a steady state and it is well insulated from the surrounding environment.
- Radiative heat transfer between the coil's external surface and the environment is negligible.
- Heat and mass transfer coefficients are assumed as a constant within the infinitesimal volume.
- The model is one-dimensional.
- The spray water temperature is assumed as a constant.
- The temperature of the interface T_{int} between the water film on the external surface (fins and tubes) and the air is equal to the average bulk temperature of the film.
- The Lewis factor Lef = hc / (hm×cp,a) ≈ 1.

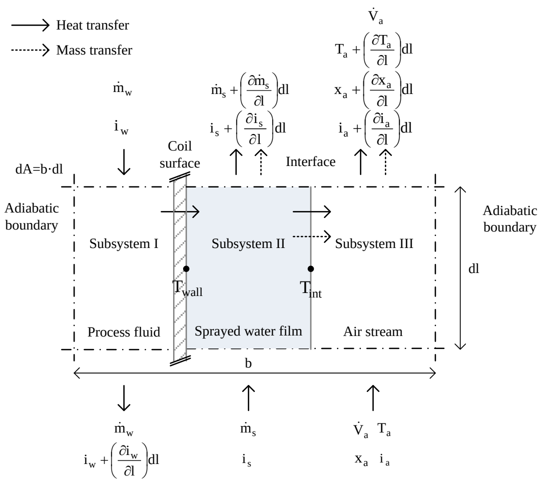

Fig. 1 – Infinitesimal control volume dA of the hybrid cooler

Looking at the overall infinitesimal control volume dA, a mass balance is written as:

$$dx_a = -\frac{d\dot{m}_s}{\dot{m}_a} dl \tag{1}$$

The mass balance on the subsystem II permits to calculate the evaporated rate of sprayed water on coil surface.

$$d\dot{m}_s = -h_m \left(x_{a,sat}(T_{int}) - x_a \right) dl \tag{2}$$

Here $x_{a,sat}(T_{int})$ is the humidity ratio of the air stream at saturation conditions and at temperature T_{int} and h_m is the mass transfer coefficient.

The air stream (subsystem III) varies its thermo-hygrometric conditions as a consequence of water evaporation from coil surface and heat release from process fluid. The consequent variation in terms of enthalpy can be expressed as follows:

$$di_a = \frac{h_m}{\dot{m}_a} \left(i_{a,sat}(T_{int}) - i_a \right) dl \tag{3}$$

As done for subsystems II and III, the variation of process fluid temperature results from an energy balance imposed on subsystem I.

$$dT_w = -\frac{U_i\left(T_w - T_{wall}\right)}{\dot{m}_w c_{p,w}} dl \tag{4}$$

In Eq. 4 U_i is the overall internal heat transfer coefficient, which is a function of the average internal convective heat transfer coefficient and of the tube thermal conductivity as calculated in (Shah and London, 1978; Gnielinski, 1976).
Therefore, the heat and mass transfer problem reduces to solving the system of differential equation of Eq. 5.

$$\begin{cases} dx_a = -\dfrac{d\dot{m}_s}{\dot{m}_a} dl \\ d\dot{m}_s = -h_m\left(x_{a,sat}\left(T_{int}\right) - x_a\right) dl \\ di_a = \dfrac{h_m}{\dot{m}_a}\left(i_{a,sat}\left(T_{int}\right) - i_a\right) dl \\ dT_w = -\dfrac{U_i\left(T_w - T_{wall}\right)}{\dot{m}_w c_{p,w}} dl \end{cases} \tag{5}$$

2.2 Mass balance and wettability factors

The calculation of the wet surface area in a hybrid cooler is fundamental in order to predict the performance of the unit under different working conditions. Most of the numerical models of the sprayed water coils assume the existence of fully dry or wet conditions on tubes and fins. This assumption cannot be made because it is hard to wet uniformly the surface coil with a water jet sprayed upwards (Fig. 2 and Fig. 3).
When water sprinklers are activated, only a fraction of the spray water rate $\dot{m}_s$ deposits on the coil surface since the remaining part passes through. The deposited spray water rate ($\dot{m}_{dep}$) determines the so-called wettability factor R_{wet} of fins and tubes. When the spray water forms a film deposit on the coil surface, this might result in evaporation. The evaporation rate depends on the humidity ration difference at the water-air interface (considered at saturation conditions) and the airflow. The evaporated mass flow rate can be calculated for each j row as:

$$\dot{m}_{evap} = \eta_{o,wet} h_m \left(x_{a,sat}\left(T_{int}\right) - x_a\right) A_{wet} \tag{6}$$

Water droplets that do not evaporate are removed either by gravitational forces or by the air stream. In both cases, these forces have to overcome the water surface tension driving the water retention. The retained water on the coil surface increases the pressure drop across the heat exchanger by restricting the flow of air (McQuiston, 1978; Wang et al. 1997; Korte & Jacobi, 1997). The dominance of airflow forces on gravity (causing a trailed spray mass flow rate) or the opposite situation (water dripping) is determined through the comparison between the critical film velocity v_{crit} and the maximum air velocity $v_{a,max}$ evaluated on the minimum free-flow area (Dreyer et al., 1992; Kriel, 1991; Wallis, 1969).

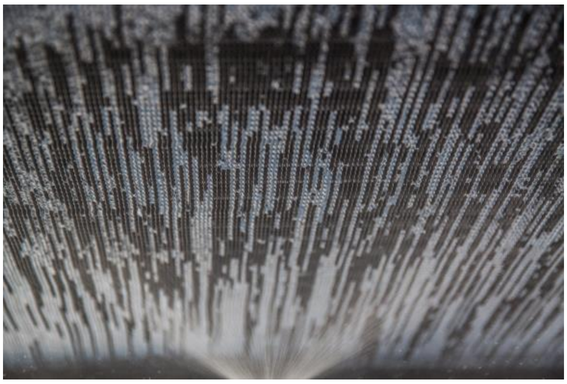

Fig. 2 – Picture of the 1st row (bottom part) of the coil while sprinkles are active

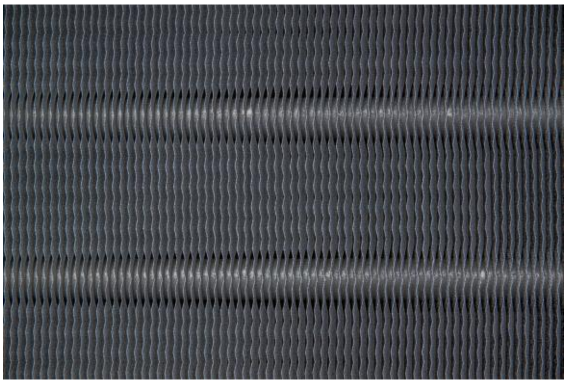

Fig. 3 – Picture of the 6th row (upper part) of the coil while sprinkles are active

2.2.1 Wettability factor

The wettability factor is defined as the ratio of wet finned-tube surface A_{wet} over the total finned-tube area. It is a fraction that depends upon several factors such as the type of nozzles, the size of droplets, the coil-geometry or the spray water pressure, just to name a few.

The research on the droplet impact is in continuous progress (Cossali et al., 2005). A physical model of the spray water system is elaborated here. Five main assumptions are required in order to formulate the solution of the problem.

- The trajectory of water droplets is linear.
- The water is sprayed in a cone of a variable height.
- The spray water devices are designed in a way that nozzles wet potentially the entire coil frontal area A_{fr}.
- The spray evaporation before the deposit on coil surface has been neglected.
- The sprayed water that impacts the coil surface is accounted as completely deposited (specific studies on water deposition on heat exchanger are not in complete agreement).

The spray water rate deposited on the coil's surface is calculated using the notion of collection efficiency for fins E_{fin} and tubes E_{tube}. These parameters can be calculated as a function of Stoke number and Langmuir's parameter as proposed in Stuemple (1973) and Finstad et al. (1987).

$$\dot{m}_{dep,fin} = E_{fin} \cdot \dot{m}_s \tag{7}$$

$$\dot{m}_{dep,tube} = E_{tube} \cdot (1 - E_{fin}) \cdot \dot{m}_s \tag{8}$$

After some simplifications (Romeli, 2014), the wet fin $A_{wet,fin}$ and tube $A_{wet,tube}$ surfaces are calculated as follows:

$$A_{wet} = \frac{3 \cdot d^2_{drop,max} \cdot \dot{m}_{dep} \cdot SFT \cdot \Delta\tau}{2 \cdot \rho_s \cdot d^3_{drop}} \tag{9}$$

Where:

- $d_{drop,max}$ is the maximum diameter of deposited droplets (Cossali et al. 2005; Scheller and Bousfield, 1995);
- d_{drop} is the volume median diameter maximum diameter of deposited droplets (J.E. Braun., 1988);
- SFT is the average spray cycle;
- $\Delta\tau$ is the simulation time step.

3. Development of a numerical model

3.1 Geometrical model

Typically, the coil used in a hybrid cooler is a multirow-multipass overall counter flow configuration with unmixed cross flow at each row. Tubes are arranged in a staggered configuration with their axes perpendicular to the air stream.

In a real HC configuration and with reference to the pipe coil length, it is possible to identify fans in series $N_{fan,series}$ or in parallel $N_{fan,paral}$. Then each single fan is defined by a length l_{fan} and width w_{fan}.

The geometry of commercial coils can be very different and in order to adapt the geometry model for a generic coil, it is of practical use to define a modular equivalent coil arrangement. The following features characterise it:

- real and equivalent coils have the same internal and external pipe diameters, a staggered arrangement of tubes, the same longitudinal and transversal tube pitch;
- the equivalent coil geometry has one single pass for each row by defining an equivalent fan length l^*_{fan} and width w^*_{fan}

$$l^*_{fan} = l_{fan} \cdot \frac{N_{tubes}}{N_{rows} \cdot N_{circuits}} \tag{10}$$

$$w^*_{fan} = w_{fan} \cdot \frac{l_{fan}}{l^*_{fan}} \tag{11}$$

where l_{fan} is the length of the fan, N_{tubes} is the sum of all the tubes that can be counted in a vertical cross section of the coil, N_{rows} is the number of finned tube banks crossed by the airflow and $N_{circuits}$ identified by the number of parallel fluid loops in which the process fluid is divided when entering the coil.

In the present work, a single row-finned tube is selected as the finite control volume Σ on which the system of differential equations is solved (Eq. 5).

This approach is compatible for the application of the ε-NTU method under incomplete wet conditions (Braun, 1988; Braun et al., 1989), where the heat transfer is the sum of dry and wet contributions as follows:

$$\dot{Q} = \varepsilon \cdot \dot{Q}_{max} = \varepsilon_{dry} \cdot \dot{Q}_{max,dry} + \varepsilon_{wet} \cdot \dot{Q}_{max,wet} \qquad (12)$$

where $\dot{Q}_{max,dry}$, $\dot{Q}_{max,wet}$ and ε_{dry}, ε_{wet} are the maximum heat transfer rate and the efficiency (as calculated in ESDU, 1991) under dry and wet conditions, respectively.

Kriel studied the impact of this decision (Kriel, 1991). They demonstrate that despite 1 % loss in the calculation of the rejected heat, a simplified control volume (identified with a single row) in favour of a more detailed approach (a fraction of a tube) takes 1/8 of the simulation time of the latter.

3.2 Convergence method

For a counter-flow arrangement of cooling coil with n-rows (Fig. 4), the following continuity conditions hold:

$$T_{w,1}^{(n)} = T_{w,i} \text{ and } T_{w,1}^{(j-1)} = T_{w,2}^{(j)} \qquad (13)$$

$$T_{a,1}^{(1)} = T_{a,i} \text{ and } T_{a,2}^{(j-1)} = T_{a,1}^{(j)} \qquad (14)$$

$$x_{a,1}^{(1)} = a_{a,i} \text{ and } x_{a,2}^{(j-1)} = x_{a,1}^{(j)} \qquad (15)$$

$$\dot{m}_a = \dot{m}_a^{(j-1)} = \dot{m}_a^{(j)} \qquad (16)$$

$$\dot{m}_w = \dot{m}_w^{(j-1)} = \dot{m}_w^{(j)} \qquad (17)$$

In Eq. 13-17 subscripts 1 and 2 denote inlet and outlet conditions for the j row. Inlet temperature $T_{w,in}$ and mass flow rate $\dot{m}_w$ of the process fluid as well as the thermo-hygrometric conditions ($T_{a,in}$, $x_{a,in}$) and mass flow rate $\dot{m}_a$ of the air stream are typically given as inputs. In order to find the zeros of the system of equations (Eq. 5), a root-finding method is necessary. After having reviewed several alternatives, Brent's method (Brent, 1973) is chosen. Brent's method is a hybrid root-finding method and it combines secant method, bisection method and inverse quadratic

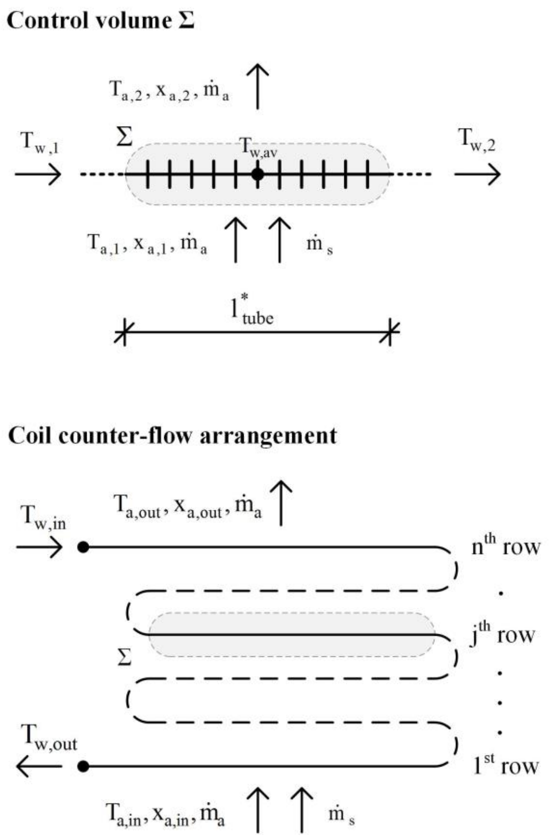

Fig. 4 – Identification of the finite control volume Σ as a coil row and its arrangement in a counter-flow configuration

interpolation based on Dekker's method (Dekker, 1969). It incorporates the guarantee of convergence as in the bisection method, but also takes advantage of the rapid rate of convergence of the less reliable methods (secant method or inverse quadratic interpolation).

This method can be used to determine a root α in the interval [a,b] as long as f(a) and f(b) have different signs. In this problem the function f is the difference $\Delta\dot{Q}$ between the heat transfer rate on the air and process fluid. This function is calculated at any time step, and convergence is reached when the absolute value of the difference between the actual (k) and previous (k-1) iteration is lower than a user-defined tolerance σ (in Watt units).

$$\left\| \Delta\dot{Q}^{(k)} \right| - \left| \Delta\dot{Q}^{(k-1)} \right\| \le \sigma \qquad (18)$$

These convergence properties can be applied for a different set of operational conditions. In real applications outlet water temperature $T_{w,out}$ is controlled by varying the fan speed. In alternative

when the fan speed is set to the maximum value, the outlet temperature is sought.
The performance of Brent's convergence method is compared to the bisection method for the cases listed in Table 1.

Table 1 – Simulation boundary conditions

Inputs		Cases 01	02	03	04	05
Relative fan speed	ω/ω_{max} [-]	1	0.5	-	-	-
Process fluid inlet temp.	$T_{w,in}$ [°C]	40				
Process fluid mass flow rate	$\dot{m}_w$ [kg/h]	3000				
Air inlet temp.	$T_{a,in}$ [°C]	25				
Air relative humidity	$\phi_{a,in}$ [%]	50				
Process fluid	-	Water				
Setpoint outlet process fluid temp.	$T_{w,out,set}$ [°C]	-	-	35	30	25

For each of these, dry and wet conditions (denoted with "D" and "W", respectively) are considered. Wet conditions are determined by spraying water at a rate of 100 kg/h. When convergence is reached, equal values of rejected heat and fan electrical consumption is achieved. A relative measure of the convergence efficiency is proportional to the number of iterations. It can be appreciated as Brent method reaches.
Comparing the number of iterations, we can notice that the computational efforts determined by Brent's method are less than bisection method in the range of 10-69 % (Fig. 5).
Furthermore, the influence of the convergence tolerance σ for cases "D03", "D04" and "D05" of Table 1 is further investigated. The overall number of iterations and the relative error in the dissipated heat is calculated. Assuming that a low tolerance value ($\sigma=1\text{x}10^{-4}$ W) will lead to an exact solution, we can notice that for the considered cases a good

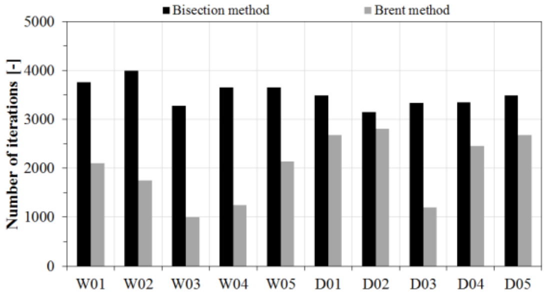

Fig. 5 – Influence of the convergence tolerance on the accuracy of the model and the related computational time for cases "D03", "D04", and "D05": comparison of bisection and Brent methods

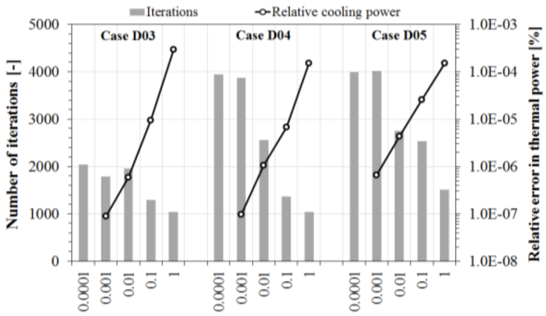

Fig. 6 – Influence of the convergence tolerance σ on the accuracy of the model and the related computational time for cases "D03", "D04", and "D05"

compromise between accuracy and computational time can be found (Fig. 6). This trade-off might change from case to case, according to the coil configuration and application.

3.3 Model validation

Heat and mass transfer equations are written in FORTRAN with the aim of deriving a numerical model for TRNSYS simulation environment.
Geometry and technical characteristics of a commercial air-to-water hybrid cooler (model "RCS-08" provided by SorTech AG, Table 2) are used to setup the numerical model. Then it is validated using experimental data from a real installation where the hybrid cooler is operated under typical summer south European working conditions.
The process fluid is a mixture of 60 % water and 40 % glycol. This component was monitored continuously between May and September 2013 with a measurement interval of 1 minute. During this period, the following measurements were taken: dry-bulb temperature and relative humidity of ambient air, inlet/outlet process fluid

temperature, mass flow rate and fan electrical consumption. The spray water system was active for 10 seconds in a minute with an average mass flow rate of 200 kg/h. Tap water is used for the spray water system with an average temperature of about 10°C.

The uncertainty in water mass flow rate and temperature measurements is about 2 % and ±0.3 K, respectively. The experimental error of the electrical sensor is identified at 0.25 % of measured values.

Table 2 – Nominal performance characteristics of the hybrid cooler "RCS-08" SorTech AG

Description		Value
Process fluid inlet temp.	$T_{w,in}$ [°C]	31.8
Process fluid mass flow rate	$\dot{m}_w$ [kg/h]	3684
Air inlet temp.	$T_{a,in}$ [°C]	24.5
Air relative humidity	$\phi_{a,in}$ [%]	50
Process fluid	-	Water
Spray water mass flow rate	$\dot{m}_{spray}$ [kg/h]	0
Process fluid outlet temp.	$T_{w,out}$ [°C]	27
Heat transfer rate	$\dot{Q}$ [W]	21000
Air volumetric flow rate	$\dot{V}_a$ [m³/h]	13000
Max fan speed	ω_{max} [rpm]	890
Max fan electrical power	$\dot{P}_{el}$ [W]	1320

3.3.1 Wettability factor

The validation of the wettability factor is not an easy task to perform, in particular on a real installation. The compactness of a coil assembly prevents the observation or the measurement of the water deposited on tubes and fins for each row. Although a detailed validation of the wettability factor cannot be performed, the sensitivity analysis to a possible error in the calculation of the wet area could already provide a good insight.

This approach consists in assuming a given error in the calculation of the fins and tubes wet surface ΔA_{wet}. The wettability factors R_{wet} for each row are then calculated together with the total transfer rate $\Delta\dot{Q}_{tot}$. For practical reasons, the variation of the wet area and the total heat transfer rate variations are provided in relative terms.

$$\delta A_{wet} = \Delta A_{wet} / A_{wet} \tag{19}$$

$$\delta\dot{Q}_{tot} = \Delta\dot{Q}_{tot} / \dot{Q}_{tot} \tag{20}$$

This test is carried out by operating the hybrid cooler with an inlet water temperature of 40 °C, an inlet air dry-bulb temperature of 25 °C, an air relative humidity of 50 %, and water mass flow rate of 3000 kg/h. The fan speed is operated at its maximum speed rate (890 rpm). Sprayed water rate amounts to 100 kg/h with a spray water cycle SFT of 0.1667.

The deviation in relative and absolute terms is shown in Fig. 7 and Fig. 8. It can be seen that despite a large deviation of wet surface area (±60 %), the total heat transfer varies less than ±5 %. On the contrary, a much larger variability is shown latent heat transfer (+30/-36 %). In absolute terms the deviation for the total heat transfer is between 42.9 kW and 46.4 kW.

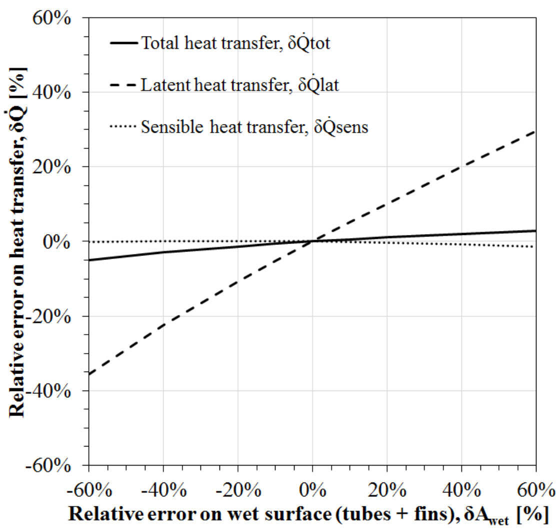

Fig. 7 – Relative error on heat transfer as a function of the relative error on wet surface calculation

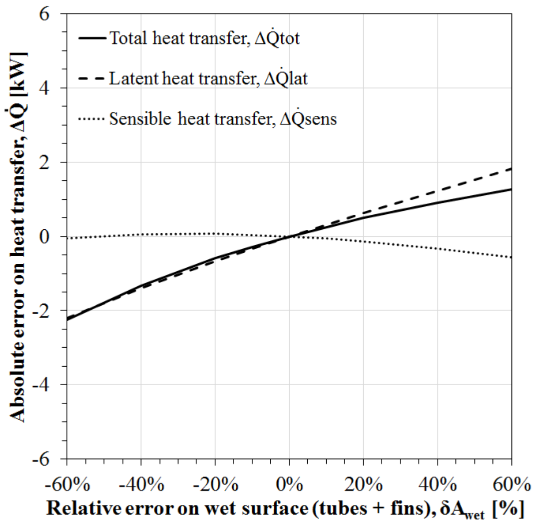

Fig. 8 – Absolute error on heat transfer as a function of the relative error on wet surface calculation

3.3.2 Heat transfer and electrical power

Fig. 9 and Fig. 10 show the comparison between the predicted and monitored instantaneous performances of the hybrid cooler with respect to the rejected heat $\dot{Q}_{ch}$ and the fans electrical power $\dot{P}_{el}$, respectively.

The numerical model has an average error of 7 % for the rejected heat and 0.1 % for the fan power consumption. The major discrepancies are observed at low values of relative fan speed. This is due mainly to the fact that the pressure loss curve of the fan provided by the manufacturers does not contain specifications for an installed component.

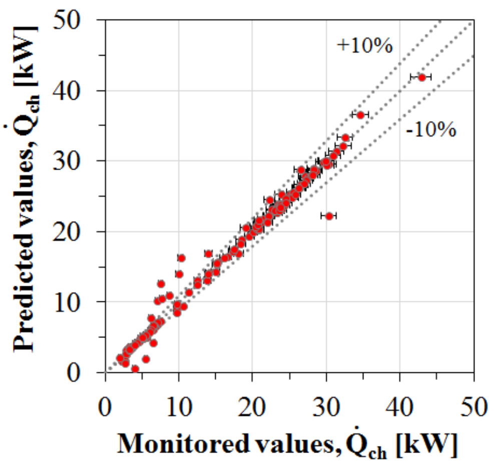

Fig. 9 – Comparison of rejected heat power between monitoring (x-axis) and experimental (y-axis) data

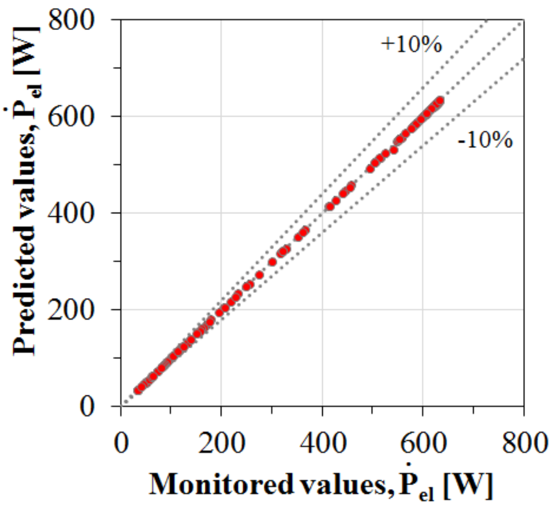

Fig. 10 – Comparison of fan electric power between monitoring (x-axis) and experimental (y-axis) data

4. Conclusion

The aim of the present work is the development of the numerical model of a hybrid cooler to be used for transient simulations purposes. The work is motivated by a raising interest in HVAC manufacturers due to its adaptability to a wide range of applications and the limitation of operation and maintenance costs with respect to traditional air-based condensing units.

The main features of the code elaborated are: (1) the definition of a modular geometry of the cooling coil, (2) the definition of water collection efficiency on coil surface due to a water jet sprayed upwards, (3) the possibility to develop tailored control strategies for water spray cycles, and (4) the implementation of dedicated control strategies to modulate the fan speed.

The model can be exploited for different purposes by HVAC industries and system designers. The influence of a hybrid cooler as an alternative heat rejection component for a whole system layout can be quantified. Dedicated control strategies for fan and spray water systems can be developed and easily implemented by the users in the simulation. Coil geometry and fans arrangements can be optimized for designing a custom-made water coil.

The numerical model was validated by using the monitored data of a pilot plant installation from May throughout September 2013. The model demonstrated to be in good agreement with the monitored performance, with errors of 7 % of the rejected heat and 0.1 % regarding the fan power.

5. Nomenclature

Symbols

A	area (m^2)
C	thermal capacity (W/K)
$c_{p,a}$	specific heat of dry air (kJ/(kg K))
d	diameter (m)
E	collection efficiency (-)
h_c	heat transfer coefficient (W/(m^2 K))
h_m	mass transfer coefficient (kg/(m^2 s))
i	enthalpy (kJ/kg)
l	length (m)
l^*	equivalent length (m)
Le	Lewis number (-)
$\dot{m}$	mass flow rate (kg/s)
NTU	number of transfer units (-)
K	heat and mass transfer coefficient (kg/(m^2s))
$\dot{P}_{el}$	electrical power (W)
$\dot{Q}$	thermal power (W)
R_{wet}	wettability factor (%)
SFT	average spray cycle (-)
T	temperature (K)
t	thickness (m)
U	heat transfer coefficient (W/(m^2 K))
$\dot{V}$	volumetric flow rate (m^3/s)
v	velocity (m/s)
w	width (m)
x	humidity ratio (kg_v/kg_a)
Δ	variation / difference
ε	effectiveness (-)
η	fin efficiency (-)
λ	thermal conductivity (W/(m K))
ϕ	relative humidity (%)
ω	rotational speed (rpm)
σ	tolerance
Σ	finite control volume
τ	time (s)

Subscripts/Superscripts

a	air
dep	deposited
H	referred to heat transfer
H+M	referred to heat and mass transfer
i	internal
in	inlet
int	interface
j	generic row number
k	iteration number
n	row number
o	external
out	outlet
s	spray water
Sat	saturation
W	referred to process fluid
Wall	coil surface

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