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A model for the performance assessment of hybrid coolers by means of transient numerical simulation

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HIGHLIGHTS

- Numerical modelling of heat and mass transfer phenomena in an air-based heat rejection component.
- Definition of a modular coil geometry for staggered finned-tube heat exchangers.
- Calculation of wettability factors for tube and fins.
- Definition of control strategies for fan speed and spray water cycles.

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ABSTRACT

This paper presents the development of a numerical model of a hybrid cooler for transient simulation purposes as a component of a thermal energy system. The model uses a modular definition of the control volume, and is suitable for modelling any staggered coil geometry. The set of parameters required for modelling the hybrid cooler is typical of the so-called design models. A rigorous analysis of sensible and latent heat fluxes due to spray water evaporation is included. Further, the model can be exploited for the development of user-defined control strategies of fans and water spray systems. The model is validated using monitored data from a pilot system installation in which a commercial hybrid cooler is operated under typical summer south European working conditions.

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1. Introduction

A hybrid cooler is a dry cooler equipped with a spray water system. The water system wets the coil surface by spraying from the bottom of the heat exchanger to the top, in a co-current direction with the air flow. Hybrid coolers are not a new concept and several studies, particularly experimental [1–4], have been performed in the last 30 years. Successful design of a dry/wet heat exchanger depends on having a basic understanding of the performance of spray cooled heat exchangers, combined with efforts to save water resources and reduce electricity consumption.

A hybrid cooler can be used in the heat rejection processes of cooling systems. Efficient heat rejection is vital for achieving a cost-effective primary energy balance, which is especially important in the case of thermally activated cooling systems using renewable energy sources (IEA Task 38 [5] and 48 [6]). Dry coolers (DCs) and wet cooling towers (WCTs) are the most common heat rejection devices available on the market [7]. As shown in Fig. 1, the fan electricity consumption is highly sensitive to the desired outlet process fluid temperature. When the temperature difference of the process fluid ΔT_w (=$T_{w,in} - T_{w,out}$) is relatively small, the advantages in terms of fan electrical savings of WCTs (dashed line) over DCs (continuous line) are almost negligible. Conversely, when the outlet process fluid temperature approaches the dry-bulb temperature, the electrical power of fans for DCs is very large compared to WCTs. The main advantage of hybrid coolers with respect to DCs and WCTs is the fact that the water is periodically sprayed only when necessary. Therefore, hybrid coolers can simultaneously compensate for the limitations of both dry coolers (i.e. high electrical consumption) and wet cooling towers (i.e. high water consumption).

In order to promote the development of new and more efficient concepts of hybrid coolers and to demonstrate the benefit in terms of primary energy savings at system level, it is necessary to have a numerical model of a hybrid cooler. Such a model should be

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Nomenclature

A	area [m^2]
A^*	equivalent area [m^2]
c_1, c_2, c_3, c_4	experimentally derived coefficients
c_{pa}	specific heat of dry air [kJ/(kg K)]
c_{pm}	specific heat of moist air [kJ/(kg K)]
c_{pv}	specific heat of water vapor [kJ/(kg K)]
c_{pw}	specific heat of water [kJ/(kg K)]
C	thermal capacity [W/K]
C_r	thermal capacity ratio [–]
d_i, d_o	internal/external tube diameter [m]
D_h	hydraulic diameter [m]
E	collection efficiency [m]
f	friction factor [–]
Gz	Graetz number [–]
h_c	heat transfer coefficient [W/(m^2 K)]
h_m	mass transfer coefficient [kg/(m^2 s)]
i	enthalpy [kJ/kg]
i_{fg}	latent heat of vaporization [J/kg]
j_h	Colburn j-factor [–]
K	heat and mass transfer coefficient [kg/(m^2 s)]
l	length [m]
l^*	equivalent length [m]
Le	Lewis number [–]
$\dot{m}$	mass flow rate [kg/s]
M	mass resistance [(m^2 s)/kg]
Nu	Nusselt number [–]
K	heat and mass transfer coefficient [kg/(m^2 s)]
p_l	longitudinal pipe spacing [m]
p_t	transversal pipe spacing [m]
P	pressure [Pa]
P_{el}	electrical energy [kW h]
$\dot{P}_{el}$	electrical power [W]
Pr	Prandtl number [–]
Q	thermal energy [kW h]
$\dot{Q}$	thermal power [W]
N	number
R	thermal resistance [(m^2 K)/W]
Re	Reynolds number [–]
R_{wet}	wettability factor [%]
s	spacing [m]
S	Sommerfeld number [–]
t	thickness [m]
T	temperature [K]
U	heat transfer coefficient [W/(m^2 K)]
v	velocity [m/s]
$\dot{V}$	volumetric flow [m^3/h]
w	width [m]
w^*	equivalent width [m]
We	Weber number [–]
x	humidity ratio [kg_{vapour}/kg_{air}]
x'', y'', z''	equivalent length [m]

Abbreviations

COP	coefficient of performance [–]
EFC	evaporative fluid cooler
NTU	number of transfer units [–]
SFT	average spray water cycle [–]
SPF	seasonal performance factor [–]
WM	working mode

Greek letters

Δ	variation/difference
ε	effectiveness [–]
η	fin efficiency [–]
λ	thermal conductivity [W/(m K)]
μ	dynamic viscosity [kg/(ms)]
ϕ	relative humidity [%]
ω	rotational speed [rpm]
ρ	density [kg/m^3]
σ	tolerance [–]
Σ	finite control volume
τ	time [s]

Subscripts

a	air
dep	deposited
ff	free-floating
fr	frontal
H	referred to heat transfer
H + M	referred to heat and mass transfer
i	internal
in	inlet
int	interface
j	row number
k	iteration number
n	row number
o	external
out	outlet
s	spray water
sat	saturation
series	referred to a series configuration
paral	referred to a parallel configuration
tot	total
w	referred to the process fluid
wall	coil surface

reliable and flexible enough to easily adapt to a change of coil geometry, working conditions or system layout and control strategy. However, as shown in following sections, such a model in literature is still lacking. In order to overcome this problem, a novel numerical model is developed. Conversely to existing air-liquid heat exchanger models, it utilizes a modular definition of the control volume and it can be adapted to any staggered coil arrangement. Additionally, the hybrid cooler integrates a model of the spraying system which can wet coil surfaces with upwards water jets when requested. The set of parameters required for modelling the hybrid cooler is typical of so-called design models [8]. Finally, the model is validated using monitored data from a pilot system installation in which a commercial hybrid cooler is operated under typical summer working conditions. The model can be successfully exploited as heat rejection component as a part of a cooling system or process for evaluating annual performances or for testing user-defined control strategies of fans and water spray system.

In section two, a literature review is performed in order to understand to which extend the available numerical models can be used for present modelling scopes. The review focuses on the most popular calculation methods of heat and mass transfer problems in cooling coils, highlighting single limitations connected to the control volume definition. In the third section, the governing equations of sensible and latent heat transfer are rigorously formalized on an infinitesimal portion of the coil. The definition of heat and mass transfer coefficients in the case of water evaporation is also reported. In the fourth section, the governing heat transfer equations are integrated over a finite control volume (Σ) and used for developing a numerical model. The identification of the control volume is motivated by the aim of deriving modular coil geometry

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