International Journal of Heat and Mass Transfer 81 (2015) 207–221

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Performance study of the Maisotsenko Cycle heat exchangers in different air-conditioning applications

Demis Pandelidis [a,*], Sergey Anisimov [a], William M. Worek [b]

[a] Department of Environmental Engineering, Wroclaw University of Technology, 27 Wyspiański st., 50-370 Wroclaw, Poland
[b] Stony Brook University, College of Engineering and Applied Sciences, Department of Mechanical Engineering, 127 Engineering Bldg., Stony Brook, NY 11794, USA

ARTICLE INFO

Article history:
Received 12 August 2014
Received in revised form 13 October 2014
Accepted 13 October 2014
Available online 1 November 2014

Keywords:
Maisotsenko Cycle
M-Cycle
Indirect evaporative cooling
Mathematical model
Cross-flow recuperator

ABSTRACT

This paper investigates a mathematical simulation of the heat and mass transfer in the two different Maisotsenko Cycle (M-Cycle) heat and mass exchangers used for the indirect evaporative cooling in different air-conditioning systems. A two-dimensional heat and mass transfer model is developed to perform the thermal calculations of the indirect evaporative cooling process, thus quantifying the overall heat exchangers' performance. The mathematical model was validated against the experimental data. Numerical simulations reveal many unique features of the considered units, enabling an accurate prediction of their performance. Results of the model allow for comparison of the two types of heat exchangers in different applications for air conditioning systems in order to obtain optimal efficiency.

1. Introduction

In recent years, the increase in summer temperatures, improved insulation of the buildings, and a growth of indoor facilities have led to an increased requirement for air conditioning in buildings. Conventional mechanical vapor–compression air-conditioning systems consume a large amount of the electrical energy that is largely dependent upon a fossil fuel. This mode of air conditioning is, therefore, neither sustainable nor environmentally-friendly. Due to the increasing need for air conditioning and the growing interest in energy savings, seeking ways to reduce fossil fuel consumption and to increase usage of the renewable energy during air-conditioning process in building sector is a matter of great importance.

Evaporative air cooling is an alternative to the conventional vapor–compression systems to meet above mentioned economic, environmental, and regulatory challenges. Direct evaporative cooling is the process of evaporating liquid water into the surrounding air and causing its temperature to decease. A typical direct evaporative cooler uses a fan to draw in outside air through a pad-wetting media and circulates the cool air through the building. Theoretically, the ultimate temperature for the direct evaporative cooling process is the ambient air wet-bulb temperature, however this temperature is not easy reached and the resulting air stream is humid. Therefore, new methods and technologies are needed for cooling of buildings.

One of the best solutions to this limitation is the sub-wet bulb temperature evaporative cooling. There are several studies on achieving sub-wet bulb temperatures by the evaporative cooling and some innovative ideas do exist. Stoichkov [1] presented a mathematical model describing a cross-flow heat exchanger with a flowing down water film. In this study the wet-bulb temperature of an ambient air was not reached at the exit. Ren and Yang [2] developed an analytical model for the coupled heat and mass transfer processes in an indirect evaporative cooling with parallel/counter-flow configurations. Maclaine-Cross and Banks [3] referred to that for regenerative evaporative cooling the process can approach the dew point temperature of the ambient air if appropriate mass flows and cooler geometry are chosen. Hasan [4,5] numerically simulated various configurations of indirect evaporative coolers. The results showed that the performance of the system could be improved by manipulating the air flow by branching the working air from the product air, which is indirectly pre-cooled. Zhao et al. [6] presented a numerical study of the counter-flow heat and mass exchanger for the dew point evaporative cooling purpose. They proposed a range of design conditions and flow rates to improve the cooler performance. Wang [7] studied the effect of the wettability (the surface wettability factor is the parameter used to estimate the effect of an incomplete wetting) of aluminum plates to the cooling performance of the indirect evaporative systems. A dynamic contact analyzer was applied to

* Corresponding author.
E-mail address: demis.pandelidis@pwr.wroc.pl (D. Pandelidis).

http://dx.doi.org/10.1016/j.ijheatmasstransfer.2014.10.033
0017-9310/

Nomenclature

c_p	specific heat capacity of moist air [J/(kg K)]
d	hydraulic diameter [m]
F	surface area [m^2]
h	specific enthalpy of the moist air [kJ/kg]
H	height [m]
G	moist air mass flow rate [kg/s]
L, l	streamwise length of cooler [m]
M	water vapor mass transfer rate [kg/s]
NTU	number of transfer units, $\mathrm{NTU} = \alpha F/(Gc_p)$ [–]
q	heat flux [W/m^2]
q^o	latent heat water [kJ/kg]
Q	rate of heat transfer [W]
$\widehat{Q}$	specific cooling capacity per cubic meter of the heat exchanger's structure [kW/m^3]
RH	relative humidity [%]
t	temperature [°C]
$\bar{t}$	average temperature [°C]
v	air stream velocity [m/s]
W	heat capacity rate of the fluid [W/K]
x	humidity ratio [kg/kg]
X	coordinate along primary air flow direction [m]
Y	coordinate along working air flow direction in the wet channels [m]

Special characters

α	convective heat transfer coefficient [W/(m^2 K)]
β	mass transfer coefficient [kg/(m^2 s)]
δ	thickness [m]
λ	thermal conductivity [W/(m K)]
ε	thermal effectiveness [%]
σ	surface wettability factor, $\sigma \in (0.0–1.0)$ [–]

Non dimensional coordinates

Le	Lewis factor $Le = \alpha/(\beta c_p)$ [–]
NTU	number of transfer units $\mathrm{NTU} = \alpha F/(Gc_p)$ [–]
Nu	Nusselt number [–]
Pr	Prandtl number [–]
Re	Reynolds number [–]
$\overline{X}$	$\overline{X} = X/l$ – relative X coordinate [–]
$\overline{Y}$	$\overline{Y} = Y/l$ – relative Y coordinate [–]

Subscripts

1	main (primary) air flow
2	working (working) air flow in the wet channels (product part of exchanger)
3	working (working) air flow in the dry channels (pre-cooling part of exchanger)
4	working (working) air flow in the wet channels (pre-cooling part of exchanger)
cond	heat transfer by thermal conduction
I*cond*	referenced to the first-order boundary conditions
II*cond*	referenced to the second-order boundary conditions
g	water vapor
h	referenced to the height of the channel
heat	heat transfer
i	inlet
l	latent heat flow
mass	mass transfer
o	output
p	plate surface
plt	channel plate
product	referenced to the product section of heat exchanger
s	sensible heat flow
w	water film
work	referenced to the working section of heat exchanger
WB	wet-bulb temperature
X	air streamwise in the dry channel
Y	air streamwise in the wet channel
•	referenced to the elementary plate surface
′	condition at the air/plate interface temperature

quantitatively measure the advancing and receding contact angles and the water-retention capacity of different aluminum surfaces. Riangvilaikul and Kumar [8] carried out experimental studies on a dew point evaporative cooling exchanger. Their results indicated that the wet-bulb effectiveness achieved by the exchanger was 92–114%. Zhou [9] carried out a study to optimize the design of the water distributor to improve the water distribution uniformity in the indirect evaporative air coolers. This research outlined several available water distribution modes applicable to these types of devices. To enhance the cooling performance of the typical evaporative exchangers, a novel thermodynamic cycle, known as the M-Cycle [10–18], was proposed by Professor Valeriy Maisotsenko as the new approach of making and operating the heat and mass exchanger (HMX). This was claimed to enable harnessing extra amount of energy from the ambient using a dedicated flat plate, cross-flow and perforated heat exchanger.

According to the producer, the maximal water consumption for the unit containing six HMXes is 14 kg/h [13,17]. The water needed to provide 1 kW of cooling is only 0.001 kg (for one HMX, at inlet conditions $t_i = 30$ °C and $\mathrm{RH}_i = 40\%$ and primary and working air flow rate equal 330 m^3/h [17]).

In the typical indirect evaporative air coolers (Fig. 1(a)) the working air flow is delivered directly to the wet channel, while the primary air flow is delivered to the dry channels. The principal idea of the Maisotsenko Cycle is to indirectly pre-cool the working airflow before it is delivered to the wet channel (Fig. 1(b)). Passing through the dry channel the primary air flow is cooled without increasing the moisture content (process 1*i*–1*o*). At the outlet of the dry channel a part of the primary flow (working air flow) is delivered to the wet channel, where it realizes the evaporative cooling process on the base of direct air–water contact (process 2*i*–2*o* in Fig. 1(b)). The remaining portion of the primary flow (product air flow) is delivered to the conditioned space.

The currently produced HMX has a unique design to maximize the efficiency of the direct and indirect stages of cooling process. Fig. 1(c) illustrates air-flow arrangement in the HMX produced by the Coolerado Corporation [10].

The working mechanism of the HMX under the M-Cycle is described as follows. Part of the surface on the dry side is used for the primary air flow (Stream 1 in Fig. 1), while the rest is used for the working air flow (Stream 2 in Fig. 1). The primary and the working air streams are guided to flow over the dry side along parallel flow channels. The working air stream is delivered to the dry channels first, in order to be sensibly precooled before it is split into multiple streams that are directed into the wetted section. There are regularly distributed holes in the channel where the working air is retained and each of these allows a certain percentage of the air stream to pass through the wet channels. The working air stream is then gradually delivered to the wet channel (Stream 3 in Fig. 1) as it flows along the dry side, forming an even distribution of the air flow over the wetted surface. The pre-cooled working air, delivered to the wet channel, flows over the wet