International Journal of Heat and Mass Transfer 108 (2017) 1750–1763

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

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

An approach to the analysis of heat and mass transfer characteristics in indirect evaporative cooling with counter flow configurations

Yangda Wan, Chengqin Ren *, Li Xing

State Key Laboratory of Advanced Design and Manufacturing for Vehicle Body, College of Mechanical and Vehicle Engineering, Hunan University, Changsha 410082, China

ARTICLE INFO

Article history:
Received 9 August 2016
Received in revised form 30 November 2016
Accepted 6 January 2017

Keywords:
Indirect evaporative cooling
Counter flow configurations
Coupled heat and mass transfer
Average Nusselt and Sherwood numbers

ABSTRACT

Indirect evaporative cooler (IEC), regarded as a zero pollution, inexpensive and energy efficient cooling device, has been used in many occasions. This paper aims at developing a new approach for analyzing the coupled heat and mass transfer characteristics in indirect evaporative cooling with counter flow configurations. Firstly, a two-dimensional computational fluid dynamics (2-D CFD) model is used to simulate the coupled heat and mass transfer processes. Then, a one-dimensional (1-D) model is used to analyze the fluid stream side average Nusselt number and the cooling air side average Nusselt and Sherwood numbers based on the numerical results of the 2-D model. Thus, the 1-D model can be used to retrieve the 2-D model results using those average Nusselt and Sherwood numbers obtained. This improved accuracy will help to promote the control technologies based on an electronic control unit (ECU) for the processes in similar indirect evaporative cooling devices involved. Further, a similarity analysis is presented to deduce the 2-D model equations into dimensionless forms with the purpose of obtaining the minimum set of grouped dimensionless factors affecting the average Nusselt and Sherwood numbers. Finally, the effects of the ratio of liquid water film to cooling air mass flow rate, cooling air inlet Reynolds number, cooling air inlet dimensionless temperature, fluid stream inlet Reynolds number, liquid water film inlet dimensionless temperature, the ratio of channel length to half width of cooling air channel and the ratio of fluid to cooling air channel width on the average Nusselt and Sherwood numbers are examined in detail.

1. Introduction

Indirect evaporative cooler (IEC) is an environmental friendly and energy efficient cooling device which has been widely used to cool the air or other fluids [1–4]. In the IEC, heat is transferred from the air or other fluids to the cooling air through the separating wall. The surface of the cooling air channel is wetted by liquid water, so that the liquid water evaporates into the cooling air and decreases the temperature of the wetted surface. In this manner, heat is transferred from the fluids to the cooling air without the introduction of moisture into the fluids. The IEC has been studied for improved efficiency [5–8].

Problems related to the heat and mass transfer processes in IEC have received considerable attention in the past few decades, aiming to provide simulation models for parameter analysis. In general, modeling methods can be classified into analytical approach and numerical approach. For analytical approach, Maclaine-cross and Banks [9] proposed an approximate linear model of wet surface heat exchangers by analogizing with dry surface heat exchangers. Their predicted performance of the heat exchanger is substantially higher than that measured by Pescod [10]. Erens and Dreyer [11] made a review among Poppe model, Merkel model and simplified model, and then suggested an optimal cooler shape. Stoitchkov and Dimitrov [12] developed a method for calculating the effectiveness of wet surface crossflow plate heat exchangers based on an improvement to Maclaine-Cross and Banks's model [9]. Alonso et al. [13] developed a more user-friendly simplified model for predicting the effectiveness of an IEC based on the models developed by Maclain-Cross and Banks [9] and Erens and Dreyer [11], and validated the theoretical model using experimental data. Ren and Yang [14] developed an analytical model for the heat and mass transfer processes in indirect evaporative cooling with parallel and counter flow configurations considering variable values of surface wettability and Lewis factor. Hasan [15] developed an analytical model for an IEC. It was shown that the model could be based on the ε-NTU method when proper adjustments were made by redefining the potential gradients, transfer coefficient, heat capacity rate parameters and assuming a linear saturation temperature-enthalpy relation of air. The model results were found to be in good agreement with the results from a numerical model and experimental measurements.

* Corresponding author.
E-mail address: renchengqin@163.com (C. Ren).

http://dx.doi.org/10.1016/j.ijheatmasstransfer.2017.01.019
0017-9310/

Nomenclature

a	thermal diffusivity [$m^2\ s^{-1}$]
A	total interface area for heat transfer in the heat exchanger unit [m^2]
c	mass fraction of water vapor
$c_{sf,i}$, $W_{sf,i}$	mass fraction of water vapor and humidity ratio for saturated air at inlet fluid temperature respectively
c_p	specific heat capacity [$kJ\ kg^{-1}\ °C^{-1}$]
c_{pg}	specific heat capacity of cooling air [$kJ\ kg^{-1}\ °C^{-1}$] ($c_{pg} = c_{p,dg} + W_g c_{pv}$)
C_w^*, C_f^*	liquid water and fluid to dry air heat capacity rate ratios respectively
D, L	half width and length of the cooling air channel respectively [m]
D_s	diffusion coefficient of water vapor in air [$m^2\ s^{-1}$]
g	gravitational acceleration [$m\ s^{-2}$]
Gr	Grashof number
h	heat transfer coefficient [$kW\ m^{-2}\ °C^{-1}$]
h_D	cooling air side mass transfer coefficient [$kg\ m^{-2}\ s^{-1}$]
h_{fg}	evaporation heat of water [$kJ\ kg^{-1}$]
h_{fg}^0	evaporation heat of water at reference temperature condition (0 °C) [$kJ\ kg^{-1}$]
$\overline{h_{fg}}$	a normalized heat of evaporation at reference temperature condition (0 °C)
H	half width of fluid channel [m]
Le_f	Lewis factor defined as $h_g/h_D c_{p,dg}$
m	mass flow rate [$kg\ m^{-1}\ s^{-1}$]
$\dot{m}_I$	mass flux of water evaporation [$kg\ m^{-2}\ s^{-1}$]
NTU	number of heat transfer units
Nu_M, Nu_Y	average and local Nusselt numbers respectively
p	mixture pressure [Pa]
P	dimensionless mixture pressure defined as $p/(\rho_g v_{g,i}^2)$
Pr	Prandtl number
r	fluid to cooling air heat transfer coefficient ratio
Re	Reynolds number
R_{cv}, R_{cw}	water vapor and liquid water to dry air specific heat capacity ratios respectively
R_{wg}	ratio of liquid water film to cooling air mass flow rate
Sc	Schmidt number
Sh_M, Sh_Y	average and local Sherwood numbers respectively
T	temperature [°C]
u, v	velocity components in x, y coordinate directions, respectively [$m\ s^{-1}$]
U, V	dimensionless velocity components in X, Y coordinate directions, respectively
W	humidity ratio of air [$kg\ kg^{-1}$]
x, y	space coordinate as shown in Fig. 1 [m]
X, Y	dimensionless space coordinate as defined in Eq.(42)

Greek letters

α_H, α_L	half width of fluid channel and channel length to half width of cooling air channel ratios respectively
α_δ	liquid water film thickness to half width of cooling air channel ratio
α_t	dimensionless liquid water film thickness
δ	liquid water film thickness [m]
θ	dimensionless temperature defined as $(T - T_{wg,i})/(T_{f,i} - T_{wg,i})$
λ	thermal conductivity [$W\ m^{-1}\ °C^{-1}$]
μ	dynamic viscosity [Pa s]
υ	kinematic viscosity [$m^2\ s^{-1}$]
ρ	density [$kg\ m^{-3}$]

Subscripts

b	bulk quantity
f	fluid stream
g, dg	cooling air, dry air respectively
i, o	inlet, outlet respectively
I	condition at the gas-liquid interface
PW	condition at the plate wall
qb	saturation vapor pressure
w, v	liquid water film and water vapor respectively
wg	cooling air wet bulb temperature

For numerical approach, Kettleborough and Hsieh [16] used a wettability factor to describe the effects of incomplete wetting for a counter flow IEC. The changes of the spray water temperature along the heat exchanger surface were also taken into consideration. Numerical analysis was utilized to study the thermal performances. Results showed that the agreement between measured and calculated performance results was improved. Tsay [17] also carried out a numerical analysis to study the heat and mass transfer processes in a counter flow IEC. Results showed that the energy transported across the liquid film is primarily absorbed by the film vaporization process. Guo and Zhao [18] investigated the effects of a wide variety of parameters such as the channel width, the air inlet velocity and relative humidity and the wettability of the plate on the thermal performance of an IEC. The results are of significance for the design of the IEC. Hettiarachchi et al. [19] investigated the effects of the longitudinal heat conduction in the exchanger wall of an IEC. Results showed that thermal performance of the IEC was significantly deteriorated due to longitudinal wall conduction at some operational conditions such as larger NTU or conduction factor values or lower air inlet wet bulb temperatures. Cui et al. [20,21] studied the performance of a novel counter-flow closed-loop dew-point IEC. The model was validated using experiment data acquired from literature and demonstrated good agreement with experimental results. Their results showed that the IEC can achieve a higher dew-point effectiveness and wet-bulb effectiveness with smaller channel height, lower air velocity or larger length to height ratio. Heidarinejad and Moshari [22] proposed a model of an IEC with consideration of wall longitudinal heat conduction and effect of spray water temperature variation along the exchanger surface in a cross-flow configuration. The resultant coupled equations of heat and mass transfer were discretized using finite difference method (FDM) and solved by an iterative method. The numerical results of this study showed applicability of the presented model for both sub- wet bulb and above-wet bulb cooling applications.

It can be observed from the previous works that the diverse factors such as surface wettability, spray water temperature variation, wall longitudinal heat conduction and variable Lewis factor have received great interest. However, due to the complexity of the coupled heat and mass transfer in film evaporation, the analysis of average Nusselt and Sherwood numbers for the IEC has not been well addressed yet. For a fully developed laminar flow in a channel without film evaporation, the Nusselt number is constant either at uniform wall temperature or uniform wall heat flux [23]. Both types of the Nusselt number were employed to approximate the heat transfer coefficient in some papers [24–27]. But in the actual situation, the wall surface is neither of the two conditions. Both heat and mass transfer mechanisms are involved in the processes in the wet channel with film evaporation unlike that in the dry channel without film evaporation. However, some literatures