INTERNATIONAL JOURNAL OF REFRIGERATION 67 (2016) 90–101

Available online at www.sciencedirect.com

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Two-dimensional numerical modeling for the air-side of minichannel evaporators accounting for partial dehumidification scenarios and tube-to-tube heat conduction

*Abdelrahman Hussein Hassan, Santiago Martínez-Ballester *, José Gonzálvez-Maciá*

Institute for Energy Engineering, Universitat Politècnica de València, Camino de Vera s/n, Valencia 46022, Spain

ARTICLE INFO

Article history:
Received 8 October 2015
Received in revised form 30 March 2016
Accepted 3 April 2016
Available online 8 April 2016

Keywords:
Minichannel evaporator
Modeling
Air-side analysis
Heat transfer
Mass transfer

ABSTRACT

This paper presents a comparative study between the air-side heat transfer results of a two-dimensional numerical model for minichannel evaporators (Fin2D-W), and the results of the classical ε–NTU approach. This study is carried out for different dehumidifying conditions, and for scenarios with different degree of temperature difference between tubes due to the refrigerant superheat. The Fin2D-W model solves the two-dimensional heat conduction in the fin wall considering also a 2D discretization for the air flow in contact with it. Thus, the presented model is able to capture the partial dehumidification scenarios and the effect of the heat conduction between tubes. The results analyze the differences, due to these phenomena, between the proposed model and the classical approach. Significant deviations between the two models are reported, especially in the cases of partially wet fin and high values of superheat, resulting in being up to ≈ 52% in total heat transfer.

Modélisation numérique bi-dimensionnelle du côté air d'évaporateurs à minicanaux représentant les scénarios de déshumidification partielle et de conduction de chaleur de tube en tube

Mots clés : Évaporateur à minicanaux ; Modélisation ; Analyse du côté air ; Transfert de chaleur ; Transfert de masse

* *Corresponding author.* Institute for Energy Engineering, Universitat Politècnica de València, Camino de Vera s/n, Valencia 46022, Spain. Tel.: +34 963879120; Fax: +34 96389126.

E-mail address: sanmarba@gmail.com (S. Martínez-Ballester).

http://dx.doi.org/10.1016/j.ijrefrig.2016.04.003

Nomenclature

A	surface area [m^2]
a	parameter defined in Eq. (5) [kg$_w$ kg$_{da}$$^{-1}$]
b	slope of saturated humidity ratio line [K^{-1}]
C_p	specific heat [J kg^{-1} K^{-1}]
G	mass flux [kg m^{-2} s^{-1}]
H_f	fin height [m]
h_{fg}	latent heat of water condensation [J kg^{-1}]
k	conductivity [W m^{-1} K^{-1}]
l	distance between two wall cells [m]
Le	Lewis number [–]
$\dot{m}$	mass flow rate [kg s^{-1}]
NTU	number of transfer units [–]
P	perimeter [m]
$\dot{Q}$	heat transfer [W]
$\dot{q}$	heat flux [W m^{-2}]
RH	relative humidity [%]
s	length in the forward direction of a fluid [m]
SH	superheat = T_{ut}–T_{lt} [K]
T	temperature [°C]
T^*	modified moist air temperature [°C]
t	thickness [m]
U_{wet}	overall heat transfer coefficient for wet case [W m^{-2} K^{-1}]
W	humidity ratio [kg$_w$ kg$_{da}$$^{-1}$]
x, y, z	spatial coordinates [m]

Greek symbols

α	sensible heat transfer coefficient [W m^{-2} K^{-1}]
α_D	mass transfer coefficient [kg m^{-2} s^{-1}]
α_{wet}	total heat transfer coefficient for wet case [W m^{-2} K^{-1}]
β	parameter defined in Eq. (6) [K]
ε	thermal effectiveness [–]
η	thermal efficiency [–]
λ	thermal conductance [W K^{-1}]

Subscripts

a	air
c	centroid of wall cell
dp	dew point
f	fin
fB	fin base
i	fluid cell index
j	wall cell index
k	direction index
lt	lower tube
ma	moist air
N, S, W, E	directions of neighbor wall cell
s	surface of wall cell
sat	saturated
tot	total
ut	upper tube

Superscripts

in	inlet
out	outlet

1. Introduction

Currently, several models or simulation tools for heat exchangers are available in the literature: for conventional-channel (Liang et al., 2001; Corberán et al., 2002; Domanski, 2003; Jiang et al., 2006; Singh et al., 2008; IMST-ART, 2010); and for minichannel (Asinari et al., 2004; Gossard et al., 2013; Huang et al., 2015; Martínez-Ballester et al., 2011; Ren et al., 2013; Wu and Webb, 2002; Yin et al., 2015; Zhao et al., 2012).

To allow for good and accurate modeling of the heat exchangers, a discretization process is required. The tubes of the heat exchanger are divided into a number of segments, usually along the refrigerant flow direction, with its corresponding fins. Each segment represents an individual heat exchanger that could be evaluated by classical global methods, i.e., the logarithmic mean temperature difference (LMTD), logarithmic mean enthalpy difference (LMHD), ε–NTU, and so on. However, on the other hand, those segments could be additionally discretized into a number of cells using finite volume method (FVM) or finite element method (FEM). For each, cell heat and mass balances are implemented. The main difference between the two methodologies is that the classical global methods have several implicit assumptions resulting in less freedom to describe the actual processes. These include, for example, no heat conduction between tubes along the fin (adiabatic-fin-tip assumption), negligible effect of longitudinal heat conduction (LHC), and uniform air temperature along the fin height.

The impact of these assumptions on the air-side performance of minichannel gas coolers and condensers were extensively discussed by Asinari et al. (2004), Martínez-Ballester et al. (2011), and Yin et al. (2015). However, more assumptions are employed when modeling minichannel evaporators under dehumidification, such as:

- **Uniform humidity ratio along the fin height.** This assumption is a consequence of using the fin theory, which also assumes a uniform air temperature profile within the same direction. Martínez-Ballester et al. (2011) concluded in their study that a notable air temperature gradient exists near tubes, which also extends far from the boundary layer region. This fact could have an important impact on local effects controlling the heat and mass transfer, e.g. dehumidification.
- **No consideration of partial fin dehumidification.** When the heat exchanger is used as an evaporator, its wall surface temperature is usually below the average dew point of the surrounding moist air. This results in simultaneous heat and mass transfer. However, some locations on the fin surface could be wet, and some others could be dry, due to the fin temperature gradient. The fin temperature profile and, therefore, the local dehumidifying condition depend on the local dew point of surrounding air, fin roots temperatures, and fin efficiency. To identify the surface area below or above the dew point along both the tube and the associated fin is considered to be a challenge in modeling evaporators for simulation tools. Thus, most of evaporator models in the