Applied Energy 87 (2010) 1187–1197

Contents lists available at ScienceDirect

Applied Energy

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

Comparison of heat pump performance using fin-and-tube and microchannel heat exchangers under frost conditions

Liang-Liang Shao [a], Liang Yang [a,b], Chun-Lu Zhang [b,*]

[a] Institute of Refrigeration and Cryogenics, Shanghai Jiaotong University, Shanghai 200240, China
[b] China R&D Center, Carrier Corporation, 3239 Shen Jiang Road, Pudong, Shanghai 201206, China

ARTICLE INFO

Article history:
Received 31 March 2009
Received in revised form 12 July 2009
Accepted 14 August 2009
Available online 11 September 2009

Keywords:
Heat pump
Heat exchanger
Microchannel
Frost
Model
Experiment

ABSTRACT

Vapor compression heat pumps are drawing more attention in energy saving applications. Microchannel heat exchangers can provide higher performance via less core volume and reduce system refrigerant charge, but little is known about their performance in heat pump systems under frosting conditions. In this study, the system performance of a commercial heat pump using microchannel heat exchangers as evaporator is compared with that using conventional finned-tube heat exchangers numerically and experimentally. The microchannel and finned-tube heat pump system models used for comparison of the microchannel and finned-tube evaporator performance under frosting conditions were developed, considering the effect of maldistribution on both refrigerant and air sides. The quasi-steady-state modeling results are in reasonable agreement with the test data under frost conditions. The refrigerant-side maldistribution is found remarkable impact on the microchannel heat pump system performance under the frost conditions. Parametric study on the fan speed and the fin density under frost conditions are conducted as well to figure out the best trade-off in the design of frost tolerant evaporators.

1. Introduction

Air-source heat pumps are becoming popular for energy saving. Frost is one of the challenges for air-source heat pumps. When evaporator surface temperature is below the freezing point and the ambient air humidity is higher than the saturation humidity at the evaporator surface temperature, frost forms on the coil surfaces. Frost increases the heat resistance of the evaporators and the air-side pressure drop through the coil, which decreases heat pump system performance.

Microchannel heat exchangers, which are commonly used in automotive air-conditioning systems, are compact and offer higher performance per unit weight than conventional finned-tube heat exchangers. Microchannel heat exchangers consist of aluminum flat tubes and louvered fins, which makes less material cost than finned-tube heat exchangers. Refrigerant inventory of microchannel heat exchanger systems is also less than that of finned-tube heat exchangers. Therefore, the replacement of conventional finned-tube heat exchangers with microchannel heat exchangers is drawing more attention.

The characteristics of conventional finned-tube heat exchangers in heat pump and refrigerator systems have been investigated by many researchers [1–5]. Some researchers have experimentally and numerically investigated the frost formation and the performance of finned-tube heat exchangers under frosting conditions. Hayashi et al. [6] reported there were three stages of the frost formation: crystal growth period, frost layer growth period, and frost layer full growth period. Yonko and Sepey [7] found the thermal conductivity of frost layer varies with the frost density. Yang and Lee [8,9] investigated frost properties on a cold plate experimentally, and frost density and thermal conductivity correlations were given. Xia and Jacobi [10] gave an exact solution of the frosted fin efficiency with the one-dimensional fin and two-dimensional frost assumption. Seker et al. [11] developed a quasi-steady state finned-tube heat exchanger model to investigate the frost formation on a finned-tube heat exchanger. The frost growth on the finned-tube heat exchangers in Seker's model was divided into two parts: the frost thickness growth and the frost density growth. The model results were also compared with experimental data [12]. Chen et al. [13] developed a finned-tube heat exchanger frost model, and investigated the frosted heat exchanger performance combined with a fan. Yao et al. [14] developed a frosted finned-tube heat exchanger model to investigate the frost formation and its impact on performance in air-source heat pump water heater/chiller units.

Some researchers have developed various models of the microchannel heat exchangers. Yin et al. [15] developed a CO_2 microchannel gas cooler model. In their model, each pass was separated

* Corresponding author. Tel.: +86 21 3860 3010; fax: +86 21 3860 3156.
E-mail address: chunlu.zhang@gmail.com (C.-L. Zhang).

0306-2619/$ - see front matter © 2009 Elsevier Ltd. All rights reserved.
doi:10.1016/j.apenergy.2009.08.021

Nomenclature

A	area (m^2)
Bo	boiling number ($qG^{-1}\ h_{fg}^{-1}$)
C	constant
COP	coefficient of performance = capacity/power consumption
c_p	specific heat ($kJ\ kg^{-1}\ K^{-1}$)
D	diameter (m)
D_S	diffusivity of water vapor in air ($m^2\ s^{-1}$)
F_h	fin height (m)
F_p	fin pitch (m)
f	friction factor
G	mass flux ($kg\ s^{-1}\ m^{-2}$)
H	tube height (m)
h	enthalpy ($kJ\ kg^{-1}$)
h	heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
hd	mass transfer coefficient ($kg\ s^{-1}\ m^{-2}$)
k	thermal conductivity ($W\ m^{-1}\ K^{-1}$)
Le	Lewis number
M	refrigerant charge (kg)
m	frosting rate ($kg\ s^{-1}$), mass flow rate ($kg\ s^{-1}$)
P	power (W)
p	pressure (Pa)
Q	heat transfer (W)
R	universal gas constant ($kJ\ kmol^{-1}\ K^{-1}$), heat resistance ($m^2\ K\ W^{-1}$)
Re	Reynolds number
T	temperature (°C)
Ts	saturation temperature (°C)
t	time (s)
v	specific volume ($m^3\ kg^{-1}$)
v_m	average specific volume ($m^3\ kg^{-1}$)
V	volumetric flow rate ($m^3\ s^{-1}$)
V_c	air velocity through minimum free-flow area ($m\ s^{-1}$)
W	humidity ratio ($kg\ kga^{-1}$)
We_l	Weber number based on liquid, ($G^2 D/\rho_l \sigma$)
x	coordinate

Greek symbols

δ	thickness (m)
η	efficiency
θ	louver angle (°)
λ	eigenvalue
μ	viscosity (Pa s)
ρ	density ($kg\ m^{-3}$)
σ	contraction ratio of the fin array surface tension ($N\ m^{-1}$)

Subscripts

a	air
cal	calculation
d	dry air
dis	discharge
f	fluid
fin	fin
fr	frost
g	gas
i	ice
in	inlet
lat	latent heat
m	metal
out	outlet
r	refrigerant
s	surface
sb	sublimation
sc	subcooling
sens	sensible heat
sh	superheat
suc	suction
v	vapor
wat	water
δ	thickness
ρ	density

into 10 equal-length element. Asinari et al. [16] developed a microchannel gas cooler model considering the thermal conduction in heat exchanger. They found the accuracy of capacity prediction could be improved if thermal conduction was included in the gas cooler model. Shao et al. [17] developed a port-by-port serpentine microchannel condenser model. In their model, each port was divided into several control volumes and a set of conservation equations were solved to calculate the air and refrigerant states. The thermal conductivity between ports was also considered in the model. Kim and Bullard [18] developed a microchannel evaporator model for CO_2 systems. Each slab of the microchannel evaporator was divided into several segments. Yun et al. [19] developed a microchannel evaporator model with finite volume method. The air and refrigerant leaving properties of each element were calculated with the energy and mass conservation equations. So far, the studies on the microchannel evaporator modeling under frosting conditions haven't yet been found in open literature.

The investigation on the comparison of microchannel and finned-tube heat exchangers is rare, too. Kim and Groll [20] experimentally compared four different microchannel evaporators with a baseline finned-tube evaporator in a unitary split heat pump system. The defrost frequency of the microchannel heat exchangers systems were about three times of that of the baseline finned-tube heat exchanger system. The microchannel heat exchangers system showed low average heating capacity and system performance lower than those of the finned-tube heat exchanger system. Modeling study of microchannel evaporators is not sufficient compared to the investigation on modeling of finned-tube evaporators.

In this study, the comparison of heat pump performance using microchannel and finned-tube evaporators under frost conditions are conducted numerically and experimentally. First, tube-by-tube models for microchannel and finned-tube evaporators under dry, wet, and frost conditions and a quasi-steady-state heat pump system model are developed for the numerical investigation. Then, the microchannel heat pump and the finned-tube heat pump are tested under frosting conditions. The test results are compared with the simulation results using the numerical models developed in this study. The effects of the refrigerant-side maldistribution, the fan speed, and the fin density on the system performance under frost conditions are numerically investigated as well.

2. Heat pump system model

The air-source heat pump system model is a quasi-steady state model. In each time interval, the system model is solved with the steady state assumption. The growth of the frost layers is calculated at the end of the time interval. The calculated frost layer growth is used in the next time interval. Including the microchannel evaporator model and the finned-tube evaporator model in the previous sections, the heat pump system model also contains a compressor model, a parallel-flow shell-and-tube condenser mod-