International Journal of Heat and Mass Transfer 64 (2013) 1–9

Contents lists available at SciVerse ScienceDirect

International Journal of Heat and Mass Transfer

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

Frost growth characteristics of spirally-coiled circular fin-tube heat exchangers under frosting conditions

Sang Hun Lee, Mooyeon Lee, Won Jae Yoon, Yongchan Kim *

Department of Mechanical Engineering, Korea University Anam-Dong, Sungbuk-Gu, Seoul 136-713, Republic of Korea

ARTICLE INFO

Article history:
Received 8 September 2012
Received in revised form 7 April 2013
Accepted 10 April 2013

Keywords:
Correlation
Frost
Growth
Heat transfer
Heat exchanger

ABSTRACT

The objective of this study is to investigate frost formation and growth in a spirally-coiled circular fin-tube heat exchanger. The frost thickness and growth rate were measured in the transverse and circumferential directions by varying the relative humidity (or humidity ratio), air flow rate, inlet air temperature, and fin pitch. The frost thickness was affected primarily by the relative humidity, inlet air temperature, and air flow rate. The relative humidity showed a dominant effect on the frost growth among all parameters tested. An empirical correlation for the frost thickness was developed for the spirally-coiled circular fin-tube heat exchanger as a function of the Reynolds number, Fourier number, humidity ratio, and dimensionless temperature. The mean and average deviations of the predictions using the present correlation from the measured data were 4.32% and 3.96%, respectively.

1. Introduction

Flat plate (circular and rectangular) fin-tube heat exchangers have been used in various HVAC systems due to their simple geometry and easy disposal of water condensate. Especially, they have been used as evaporators in refrigeration systems under frosting conditions for less frosting and relatively easy defrosting. The frost forming in the heat exchanger reduces air flow area by blocking the flow path, which leads to a degradation in heat transfer performance [1]. Therefore, the frost should be periodically removed from the heat exchanger surface to recover the system performance. This normally involves electric heating to melt the frost on the surface, which is a costly operation since energy is required and the heat exchanger is out of service during the process. Therefore, it is very important to understand the frost mechanism when designing a heat exchanger that can be used as an evaporator in household refrigerators. Especially, the variation of the frost thickness according to the operating condition has to be investigated to optimize the interval of the defrosting process, fin pitch arrangement, and specification of the circulation fan.

Various studies have been conducted experimentally and numerically on frost formation and growth on a horizontal plate. Sahin [2] studied initiation and growth of frost formation on a horizontal plate. He clarified the fundamental nature of the early stage of the crystal growth period of frost formation. Cheng and Shiu [3] studied frost formation and frost crystal growth on a cold plate in atmospheric air flow. Their study examined spatial variations of frost thickness as well as patterns of frost crystals formed on a cold plate. Yun et al. [4] modeled frost growth and frost properties with air flow over a flat plate. They developed a physical model of frost layer growth and frost properties with air flow over a flat plate at subfreezing temperatures. Tso et al. [5] developed a general distributed model for predicting the performance of a fin-tube heat exchanger under frosting conditions with a variation of frost thickness along a rectangular fin. Hermes et al. [6] studied frost growth and densification on flat surfaces. They suggested a physical basis for the development of a theoretical model to predict the variation of the frost thickness and mass over time. Kim et al. [7] proposed a numerical model for predicting frost behavior on a heat exchanger fin under frosting conditions, taking into account fin heat conduction and three-dimensional air flow. Kandular [8] developed a numerical model to predict frost density and effective thermal conductivity to be applicable to a wide range of frost circumstances.

Extensive studies have been conducted on flat plate fin-tube heat exchangers under frosting conditions. However, studies focusing on spirally-coiled circular fin-tube heat exchangers used as evaporators in household refrigerators under frosting conditions are very limited in the open literature [9]. The objective of this study is to investigate frost formation and growth characteristics in a spirally-coiled circular fin-tube heat exchanger by comparison with a flat plate fin-tube heat exchanger. The frost thickness and growth rate were measured by varying the relative humidity, air flow rate, inlet air temperature, and fin pitch. In addition, an empirical correlation for predicting the frost thickness in the spirally-coiled circular fin-tube heat exchangers was developed as a

* Corresponding author. Tel.: +82 2 3290 3366; fax: +82 2 921 5439.
E-mail address: yongckim@korea.ac.kr (Y. Kim).

0017-9310/$ - see front matter © 2013 Elsevier Ltd. All rights reserved.
http://dx.doi.org/10.1016/j.ijheatmasstransfer.2013.04.018

Nomenclature

A	surface area, m^2
A_c	minimum free flow area, m^2
c_p	specific heat at constant pressure, J kg^{-1} K^{-1}
D	mass diffusivity, m^2 s^{-1}
D_h	hydraulic diameter, $4A_cL_s/A$
d_f	fin diameter, mm
d_t	tube diameter, mm
Fo	Fourier number, $\alpha_a t/L^2$
f_p	fin pitch, mm
f_t	fin thickness, mm
G	mass flux, kg m^{-2} s^{-1}
h_h	heat transfer coefficient, W m^{-2} K^{-1}
h_m	mass transfer coefficient, kg $m^{-2}K^{-1}$
k	thermal conductivity, W m^{-1} K^{-1}
i_{sv}	latent heat of sublimation, J kg^{-1}
L	tube length, mm
L_s	streamwise length, mm
Le	Lewis number, α/D
$\dot{m}$	mass flow rate, kg s^{-1}
$\dot{m}_{fr}$	frost deposition rate, kg s^{-1}
n	number of tubes
Nu	Nusselt number, $\mathrm{Nu} = hD_h/k$
Q	heat transfer rate, W
Re_{Dh}	Reynolds number based on the hydraulic diameter, GD_h/μ
T	temperature, °C
t	time, s
T^*	dimensionless temperature,$(T_{tp} - T_{fs})/(T_{fs} - T_a)$
W	humidity ratio, $kg_v \cdot kg_a^{-1}$

Greek symbols

α	thermal diffusivity, m^2 s^{-1}
δ	frost thickness, mm
η	fin efficiency
μ	dynamic viscosity, Ns m^{-2}
ρ	density, kg m^{-3}

Subscripts

a	air
$cond$	conduction
$conv$	convection
eff	effective
f	fin
fr	frost
fs	frost surface
in	inlet
out	outlet
tp	tube plate
v	water vapor

function of the Reynolds number based on hydraulic diameter, Fourier number, dimensionless temperature, and humidity ratio.

2. Experimental setup and test procedure

Fig. 1 shows a schematic diagram of the experimental apparatus. The test setup consisted of a test section and a chiller unit for controlling the refrigerant temperature. The test section had a removable front door made of transparent acrylic resin for observation of the frost thickness. The outer size of the test section was 115 mm in depth, 320 mm in width, and 950 mm in height. The test setup was installed in an environmental chamber to maintain air at a constant temperature and humidity. An ethylene glycol–water mixture (50:50% by weight) was used as a working fluid. The refrigerant was circulated by a magnetic gear pump and its temperature was precisely controlled by the chiller unit.

The refrigerant temperature was measured by using thermocouples located at the inlet and outlet of the heat exchanger. The refrigerant flow rate was measured by a Coriolis-type mass flow meter with an accuracy of ±0.2% of reading. The temperature of the air before and after the test section was measured using three thermocouples with an accuracy of ±0.1 °C; these were located in the inlet and outlet ducts of the test section. The absolute humidity of the air was measured using a chilled mirror dew point sensor with an accuracy of ±0.2 °C. The air flow rate across the heat exchanger was measured using a differential pressure transducer across a nozzle with an estimated accuracy of ±3.0%.

The frost thickness was measured using a cathetometer and a micrometer with an accuracy of ±3.3% of reading [10]. The frost thickness was measured as the direct distance between the fin surface and the end of the frost. The frost thickness in the spirally-coiled circular fin-tube heat exchanger was calculated by averaging the values for windward and leeward to account for the asymmetric frost growth. The frost thickness in the tested heat exchanger was measured at the center of the middle fin in the first tube of the spirally-coiled circular fin-tube heat exchanger with 2 columns and 2 rows, because the deviation of the frost thickness at all fins in the first tube was negligible. The frost thickness at each fin in the first tube grew nearly uniform because of the uniform air flow rate past the fins. For all fin pitches, the frost thickness was measured for 120 min with an interval of 10 or 20 min. The variation in the surface roughness of the frost with time was not considered in this study because of the relatively large interval time. Frost growth patterns were observed using a digital camera at the middle fin in the tested heat exchanger.

Fig. 2 shows a schematic diagram of a spirally-coiled circular fin-tube heat exchanger with the inline tube alignment. The tested heat exchangers used individual spirally-coiled circular fins. The spirally-coiled circular fin was tilted at an angle as shown in Fig. 2(a). When air flows from the bottom, the left side of the fin is windward and the right side is leeward. The longitudinal and transverse tube pitches of the heat exchangers were 30.0 mm. The diameters of the spirally-coiled circular fin and the tube were 24.5 and 8.0 mm, respectively. Table 1 shows the specifications of the tested spirally-coiled circular fin-tube heat exchangers. The tested heat exchangers had the inline fin and tube alignments.

Table 2 shows the test conditions used in the present experiments. The refrigerant temperature was varied from −30 to −20 °C. The refrigerant flow rate was maintained at 100 kg h^{-1}. The inlet air temperature and inlet air relative humidity were varied from 3 to 15 °C and from 70% to 90%, respectively. The initial air flow rate was varied from 0.55 to 1.5 m^3 min^{-1}, which corresponded to the face velocity from 0.76 to 2.08 m s^{-1}.

The air side heat transfer rate under frosting conditions was calculated by Eq. (1), which is sum of sensible and latent heat transfer rates resulted from the air temperature change and the mass transfer of water vapor, respectively.

$$Q_a = \dot{m}_a c_{p,a}(T_{a,in} - T_{a,out}) + \dot{m}_a(W_{a,in} - W_{a,out})i_{sv} \tag{1}$$

The frost layer surface temperature was estimated by the energy balance between the conduction heat transfer rate in the frost layer and the air side heat transfer rate. It was assumed that fins and tubes of the spirally-coiled circular fin-tube heat exchanger were