INTERNATIONAL JOURNAL OF REFRIGERATION 33 (2010) 334–346

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Effect of some geometric parameters on performance of PF^2 heat exchangers in periodic frosting

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ARTICLE INFO

Article history:
Received 14 October 2008
Received in revised form
30 October 2009
Accepted 2 November 2009
Available online 10 November 2009

Keywords:
Heat exchanger
Finned tube
Experiment
Performance
Parameter
Geometry
Frosting

ABSTRACT

The thermal performance under conditions of initial frost growth and subsequent refrosting after a defrost is experimentally studied for the parallel flow parallel fin (PF^2) heat exchanger, a new style of heat exchanger that uses louvered bent fins on flat tubes to enhance water drainage when the flat tubes are horizontal (typically outdoor heat exchanger in heat pump operation). This paper focuses on quantification of the effects of geometry (i.e. fin pitch 12–22 fpi and louver pitch 1.4–2.8 mm) on defrost and refrost times. Eight heat exchangers differing in louver pitch and fin spacing are studied. A series of tests are conducted in search for the best geometry. The effects of geometry on heat transfer (thermal performance) and pressure drop for air face velocities of 0.9, 2, and 3 m/s are determined and used for comparison. Characteristics of initial heat transfer coefficient and pressure drop during the first frosting cycle are reported in terms of Colburn j_0 factor and Fanning friction f_0 factor, as a function of Re_{L_p} and geometry. The newly developed air-side correlations for PF^2 heat exchangers (Colburn j_0 factor and Fanning friction f_0 factor) predict the test data within rms error of ±10%, and with mean deviation of 2.95% and 4.98%, respectively. The correlations are base on a very low Reynolds numbers in the range of 100–620, and 8 PF^2 heat exchangers using 48 experimental data.

Impact de certains paramètres sur la performance d'échangeurs à écoulements et aux ailettes parallèles lors d'épisodes périodiques de givrage

Mots clés : Échangeur de chaleur ; Tube ailеté ; Expérimentation ; Performance ; Paramètre ; Géométrie ; Givrage

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0140-7007/$ – see front matter
doi:10.1016/j.ijrefrig.2009.11.004

Nomenclature

A	area (m^2)
C_p	specific heat ($J\,kg^{-1}\,K^{-1}$)
D	hydraulic diameter (m)
D_{avg}	average deviation (%)
D_{mean}	mean deviation (%)
dP	pressure drop (kPa)
F	cross–flow correction factor (–)
F_d	fin depth (mm)
F_h	fin height (mm)
F_p	fin pitch (fpi)
F_s	fin spacing (on centers) (mm)
f	friction factor (–)
G	air-flow mass velocity ($kg\,m^{-2}\,s^{-1}$)
h	convective heat transfer coefficient ($W\,m^{-2}\,°C^{-1}$)
h_{sg}	latent heat of ablimation (for water vapor) ($J\,kg^{-1}$)
j	Colburn factor (–)
k	thermal conductivity ($W\,m^{-1}\,°C^{-1}$)
L_p	louver pitch (mm)
$\dot{m}$	mass flow rate ($kg\,s^{-1}$)
Nu	Nusselt number (–)
Pr	Prandtl number (–)
q	heat transfer rate (W)
Re	Reynolds number based on hydraulic diameter (–)
Re_{L_p}	air side Reynolds number based on louver pitch (–)
T	temperature (°C or K)
T_L	Tube length (m)
T_{major}	tube major (mm)
T_{minor}	tube minor (mm)
T_s	tube spacing (on centers) (mm)
t	time (min)
U	overall heat transfer coefficient ($W\,K^{-1}\,m^{-2}$)
UA	overall conductance ($W\,K^{-1}$)
x	the data element (–)
ΔT_{lm}	log-mean temperature difference (°C)
δ	thickness (m)
ϕ	relative humidity (%)
n_a	overall surfaceefficiency (–)
n_f	frosted-fin efficiency (–)
μ	viscosity of fluid (Pa-s)
θ	louver angle
ρ	density ($kg\,m^{-3}$)
υ	air face velocity ($m\,s^{-1}$)

Subscripts

0	initial value
cor	by correlation
d	defrost
dp	dew-point
exp	by experiment
f	fin
ff	free-flow
fr	frost
in	inlet
l	latent heat
n	nozzle
out	outlet
r	refrigerant (or refrigerant side)
S	sensible heat
tot	total friction
w	wall
a	moist air (or air side)

1. Introduction

When a heat exchanger's surface temperature is below the freezing point of water and dew-point of the inlet moist air, the heat exchanger's surface will frost. The accumulation of frost blocks the airflow passages and increases heat transfer resistance, leading to a degradation of system capacity and efficiency. A defrost cycle melts the accumulated frost and brings performance close to the original value. However, only part of the heat brought to the evaporator is used for defrost. That energy leaves the evaporator along with water, while the rest of the heat becomes an additional load for the next refrigeration cycle.

The louvered-fin, flat-tube heat exchangers are finding wider applications as performance, compactness and cost concerns continue to drive heat exchanger design. Louver directed flow generates a higher heat transfer coefficient, which improves performance; Achaichia and Cowell (1988). Frost buildup on the louvers strongly affects air-side performance by augmenting the flow from louver directed flow to duct directed flow. Thus, to explore geometry that will be more frost tolerant, it is important to understand the effects of the geometrical parameters on the airflow during frosting.

Several studies of frost properties and frost growth mechanism on the round-tube-plate-fin heat exchanger have been reported (Kondepudi and O'Neal, 1987, 1989; Machielsen and Kerschbaumer, 1989; Yan et al., 2005). Kondepudi and O'Neal (1987) reviewed the literature on the effect of frost formation on finned-tube heat exchanger performance, and Kondepudi and O'Neal (1989) conducted frost growth research on louver-fin-round-tube heat exchangers. Machielsen and Kerschbaumer (1989) conducted research on the effects of frosting and defrosting on heat exchanger performance. Yan et al. (2005) investigated the performance of frosted finned-tube heat exchangers with plain fins, single-bank louvered-fins, and multi-louvered fins, and found that the heat transfer rate, the overall heat transfer coefficient, and the pressure drop for multi-louvered fins were higher than for the others. Only a few studies dealing with the effects of frost on the flat tube louvered compact heat exchanger have been published in the open literature to date (xia et al. 2006; Xia and Jacobe, 2004; Itoh et al., 1996; Kim and Groll, 2002). Xia et al. (2006) studied the effect of frost, defrost and refrost on the thermal-hydraulic performance of louvered-fin flat-tube heat exchangers. They developed a numerical model, which was experimentally validated, to predict the frost thickness and blockage ratio. Xia and Jacobi (2004) developed an exact solution to two-dimensional, steady heat conduction on a one-dimensional fin for frosted heat exchanger. The heat conduction inside the fin was modeled as one-dimensional