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Heat transfer correlations for evaporation of refrigerant mixtures flowing inside horizontal microfin tubes

Xiaoyan Zhang [a,b,*], Xiuling Yuan [b]

[a] School of Energy Engineering, Xi'an University of Science and Technology, 58 Yanta Street, Xi'an, Shaanxi 710054, PR China
[b] School of Energy and Power Engineering, Xi'an Jiaotong University, 28 Xianning Road, Xi'an, Shaanxi 710049, PR China

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ABSTRACT

Based on the experimental results of R417A flowing inside horizontal microfin tubes, the present work deals with the development of prediction methods for evaporation heat transfer of refrigerant mixtures in microfin tube. The microfin model by Thome et al. is modified by adjusting the convective heat transfer term, and the other microfin model is developed by introducing the enhancement factor into the modified-Kattan model. The comparison of the calculations by several microfin models and the experimental results reveals that the new microfin models developed at the present study are in much better agreement with the experimental results with the reducing average deviation by 30–50% than the models by Thome et al. and Cavallini et al., and are recommended for the prediction of evaporation heat transfer coefficients for non-azeotropic refrigerant mixtures inside microfin tubes.

1. Introduction

During recent decades, for the need of substituting for HCFCs, there are many binary and ternary non-azeotropic or near-azeotropic refrigerant mixtures to be developed, and many aspects of evaporation heat transfer have been investigated and some correlations have been proposed for evaporation of refrigerant mixtures inside horizontal smooth tubes. Contrast to pure and azeotropic refrigerant mixtures, the heat transfer degradation occurs in non-azeotropic refrigerant mixtures during phase change. The degradation can be mended by heat transfer enhancement inside heat transfer tube. So the technology of heat transfer enhancement becomes the main measure to reform the heat transfer degradation of non-azeotropic refrigerant mixtures and to improve heat transfer effect.

Many kinds of heat transfer tubes with enhanced surface have been developed to improve the performance of heat exchanger. The microfin tubes have outstanding performance from the point of a balance between two-phase heat transfer enhancement and pressure drop enlargement and have been widely used to save energy in refrigeration and air conditioning industries. However, the correlations for evaporation heat transfer of refrigerant mixtures in microfin tubes are very limited in the open literature due to lack of database, the complexity of two-phase heat transfer process, the influence of enhanced surface on heat transfer, etc., and it becomes one of the limitations in the design of efficient heat exchangers.

Based on the experimental results of evaporation heat transfer for R417A, which is a long-term substitute for R22, flowing inside two internally grooved tubes with different microfin geometrical parameters, the objective of this paper is to develop the prediction methods of evaporation heat transfer for non-azeotropic refrigerant mixtures flowing inside microfin tubes. Knowledge of new empirical correlations can provide the reference for design of evaporators, a key component that determines the performance of refrigerating and air conditioning systems.

2. Literature survey

The evaporation of refrigerant mixtures inside enhanced surface tubes is very complex due to the influencing of mixture properties and different enhanced surface on heat transfer, so there is only few models developed for mixtures evaporating in enhanced surface tubes. For pure refrigerants, Yun et al. [1] suggested a model considering the enhancement effect of turbulence generated by microfins, fin height and numbers of fin; Thome et al. [2], Cavallini et al. [3] and Koyama et al. [4] suggested the microfin models by adopting Chen's idea [5] and by introducing different enhancement factors into smooth tube correlations, respectively, these correlations were derived and the influences of microfin were considered into enhancement factors from different point of view. Thome et al. [2] correlation was a generalized model for evaporation inside

* Corresponding author. Address: School of Energy Engineering, Xi'an University of Science and Technology, Xi'an 710054, China. Tel.: +86 013720582706; fax: +86 029 88138913.
E-mail addresses: gqzxy@sohu.com, zhangxy2065@sina.com (X.Y. Zhang).

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Nomenclature

A	cross-section area (m^2)
B	scaling factor
Bo	Bond number
d	diameter (m)
E_{mf}	microfin enhancement factor
E_{RB}	convection enhancement factor
F_c	mixture correction factor
Fr	Froude number
f	fin height (m)
G	mass flow rate (kg m^{-2} s^{-1})
G_{high}	mass flow rate at the transition annular flow curve (kg m^{-2} s^{-1})
G_{low}	mass flow rate at the transition stratified-wave flow curve (kg m^{-2} s^{-1})
h	heat transfer coefficient (W m^{-2} K^{-1})
Δh_{lv}	latent heat of vaporization (J kg^{-1})
M	molecular weight (kg $kmol^{-1}$)
n	number of grooves
p	fin pitch (m)
P_r	reduced pressure
Pr	Prandtl number ($\mu Cp/\lambda$)
Q	heat rate (W)
q	heat flux (W m^{-2})
Re	Reynolds number (Gd_i/μ)
Rx	geometry enhancement factor
T	temperature (K or °C)
u	velocity (m s^{-1})
X_{tt}	Martinelli parameter
x	vapor quality
x_{max}	vapor quality at the intersection of annular flow and mist flow transition curves

Greek symbols

α	apex angle of microfin (°)
β	helix angle of microfin (°)
β_l	liquid mass transfer coefficient
δ	liquid film thickness (m)
ε	void fraction
η_A	geometry enhancement factor
λ	thermal conductivity (W m^{-1} K^{-1})
μ	dynamic viscosity (kg m^{-1} s^{-1})
θ_{dry}	dry angle (radians)
θ_{max}	dry angle for x_{max} (radians)
θ_{strat}	stratified angle (radians)
ρ	density (kg m^{-3})
σ	surface tension (N m^{-1})

Subscripts

bub	bubble point
cv	convection
dew	dew point
go	gas only
I	ideal
i	inside
l	liquid
nb	nucleate boiling
o	outside
r	root
ref	refrigerant
t	top
tp	two phase
v	vapor
wi	inner wall

microfin tubes, the nucleate boiling heat transfer coefficient defined by the Cooper correlation [6], while the convective boiling heat transfer coefficient was calculated from a turbulent film flow equation. Cavallini et al. [3] correlation was derived by introducing their correlation form of condensation into convective evaporation. The dimension of microfins including fin height, number of grooves and helix angle of microfin, and the mass flow rate and properties of refrigerant were considered into enhancement factors in Thome et al. [2] and Cavallini et al. [3] correlations. The Koyama et al. [4] correlation was developed considering the enhancement effect of microfin on both the nucleate boiling and the convective heat transfer components, however, this correlation did not have a general format because the dimension of microfins could not be input into the correlation. For refrigerant mixtures, above microfin models overpredict heat transfer coefficients rather strongly, and cannot express the local variations in heat transfer coefficients during the whole evaporation process.

3. Experimental work

Fig. 1 shows a schematic diagram of the test facility, a vapor compression system was chosen, which is consisted of three separate circuits, namely the refrigerant circuit, the heating circuit and the cooling circuit. The evaporating test section consists of three double-tube type heat exchangers, the inner tubes adopt a 3.0 m long smooth (after referred to as Tube I) and two 2.4 m long internally grooved tubes (after referred to as Tube II and Tube III) having different geometrical parameters with the outer diameters of 9.52 mm, respectively, the details of these tubes are shown in Table 1 and Fig. 2.

Fig. 3 shows the test section measurement layout diagram for water flowing in the annulus and refrigerant flowing inside inner tubes with counter flow arrangement. The wall temperatures are measured along the tube with T type thermocouples soldered on the inner tube at axial intervals of 300 mm, for each location, the thermocouples are located at the top and bottom. The water temperatures are measured directly with thermocouples in the water flow at the same locations. The refrigerant temperatures are measured at the inlet and outlet of the test tube. The refrigerant pressure is measured at the inlet of the test tube by absolute pressure transducers; and the refrigerant pressure drop through each test tube is measured by differential pressure transducers. The mass flow rate of refrigerant and water are measured by differential pressure flow meter and float meter, respectively. The accuracies of the measurements are predicted to be within ±0.4% over the range of −40 to 350 °C for thermocouples, within ±1.0% for the differential pressure flow meter, within ±1.5% for float meter, within ±0.5% for the absolute pressure transducers and the differential pressure transducers.

In the experiment, the sub-cooled refrigerant liquid from liquid receiver is introduced into the test tube through an expansion valve by assuming isenthalpic expansion, and the refrigerant liquid is instantaneously evaporated to a certain vapor quality. The inlet quality is determined by measuring pressure and temperature. Heat input during evaporation is provided by water flowing in the annulus. The experiment has been carried out over a refrigerant mass flow rate range of 176–344 kg m^{-2} s^{-1}, a heat flux range of 11–32 kW m^{-2}, a vapor quality range of 0.2–1 and an evaporation temperature range of 0–5.5 °C. For every working state, when steady state is reached, the experimental data are recorded and