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Modeling of condensation heat transfer of refrigerant mixture in micro-fin tubes

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Abstract

This paper presents a new refrigerant-mixture model for condensation in micro-fin tubes. The new refrigerant-mixture model is developed based on a theoretical analysis of turbulent film condensation inside smooth tubes. Several modifications have been implemented in the original smooth-tube model to account for mass transfer thermal resistance between the liquid and vapor phases. The new condensation model is compared with a set of around 200 experimental data points for refrigerant mixtures. The comparison shows that the new model is capable of producing consistent prediction results with a mean absolute deviation (MAD) less than 22% for most of the available data sets. The MAD values obtained with the new model are lower compared to the results obtained using another models.

Keywords: Condensation; Micro-fin tubes; Refrigerant mixtures; Heat transfer

1. Introduction

Hydrochlorofluorocarbons refrigerants used in the refrigeration and air conditioning industry have been identified as one of the major contributors to the depletion of the ozone layer of Earth. In response to the phase-out of hydrochlorofluorocarbon refrigerants and changes in efficiency standards for air conditioning equipment, the refrigeration and air conditioning industry is developing more compact equipment with higher system operating efficiency. Much of the effort to replace hydrochlorofluorocarbon refrigerants have been concentrated on development of refrigerant mixtures to be used the refrigeration and air conditioning industry. New systems with enhanced surface need to be developed to be able to accommodate high heat fluxes as well as use new refrigerant mixtures. Micro-fin tubes are commonly used as enhanced surfaces in heat exchangers and represent a technology that has been able to beneficially enhance heat transfer in both single-phase and two-phase applications without causing drastic increases in pressure-drop.

Horizontal micro-fin tubes have been commonly used for heat exchangers of refrigerators and air conditioners due to their better heat transfer performance. Many researches have performed experimental studies on the effects of fin geometry, tube diameter, refrigerant, etc. on the condensation heat transfer and pressure drop of the micro-fin tubes have. Several researches have shown that the used of micro-fin tubes is one of the most efficient heat transfer enhancement techniques to improve the performance of the heat exchangers. Some of these researchers are: Cavallini et al. [1,2], Yu and Koyama [3], and Kedzierski and Goncalves [4], Dongsoo et al. [5], among others.

According to Cavallini et al. [1,2], micro-fin tubes generally exhibit a heat transfer enhancement, with respect to equivalent smooth tubes under the same operating conditions, from 80% to 140%, with a pressure loss increase from 20% to 80%. Similarly, Yu and Koyama [3] showed that due to the enlargement of heat transfer area the local heat transfer coefficient in a horizontal micro-fin tube is two times higher than those of a smooth tube with the same

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Nomenclature

c_p	specific heat (J kg^{-1} K^{-1})
d_o	outer-tube diameter (m)
e	fin height (m)
g	gravitational acceleration (m s^{-2})
G	total mass flux (kg m^{-2} s^{-1})
h	heat transfer coefficient (W m^{-2} K^{-1})
i_{fg_m}	specific enthalpy of isobaric condensation of the mixture (J kg^{-1})
k	thermal conductivity (W m^{-1} K^{-1})
L	heated test section length (m)
MAD	mean absolute deviation
M_1, M_2, M_3	empirical constants defined in Eq. (4)
N	total number of data points
n_g	number of micro-fins
P	pressure (Pa)
Pr	Prandtl number (dimensionless)
q	surface heat flux (W m^{-2})
Re	Reynolds number (dimensionless)
Rx	enhancement factor defined in Eq. (3)
th	tube-wall thickness (without micro-fin) (m)
T	temperature (K)
T^+	dimensionless temperature
ΔTG	temperature glide for refrigerant mixture effect (K), which refers to a refrigerant blend that has a range of boiling points or condensing points
x	vapor quality

Greek symbols

β	micro-fin apex angle (°)
γ	micro-fin helix angle (°)
μ	dynamic viscosity (kg m^{-1} s^{-1})
τ	shear stress (Pa)
δ	liquid film thickness (m)
δ^+	dimensionless liquid film thickness
ρ	density (kg m^{-3})

Subscripts

exp	experimental results
l	liquid-phase only
m	two-phase mixture
pre	prediction results
sat	saturation
v	vapor-phase only
w	wall

inner diameter. Similarly, Kedzierski and Goncalves [4] reported an enhancement of the heat transfer but in their case it was due to the fins behaving as a surface roughness. Dongsoo et al. [5] reported that the condensation heat transfer coefficient of a micro-fin tube were 2–3 times higher than those of a plain tube and the heat transfer enhancement factor decreased as the mass flux increased for all refrigerants tested (R22, R134a, R407C, and R410A).

Many researchers have participated actively in determining the performance of the micro-fins tubes by using pure refrigerants as working fluids. There are a lot of techniques for predicting the heat transfer coefficients during condensation insides both smooth tubes and micro-fin tubes over the past decades. Generally, empirical methods are most often used to compute the heat transfer coefficient during condensation inside the horizontal tubes. Akers et al. [6], Cavallini and Zecchin [7], and Shah [8] are some of the empirical models that have been developed for condensation inside smooth tubes. However, most of these models are developed based on their own data and valid for certain refrigerants only. After the invention of the micro-fin tubes, researchers have tried to account for the heat transfer enhancement due to the presence of the micro-fins by introducing new parameters and generating new empirical constants to their original smooth tube correlations. Cavallini et al. [1], Yu and Koyama [3], Kedziersky and Goncalves [4], and Chamra et al. [9] among others have developed models to predict the condensation heat transfer of refrigerants in micro-fin tubes.

The implementation of Montreal Protocol in 1987 mandated the replacement of many pure refrigerants by the new alternative refrigerants, which have zero ozone depletion potential and reduced global-warming potential. These alternative refrigerants include pure HFC (Hydrofluorocarbon) refrigerants and mixture of HFC refrigerants. The efficiencies of these alternative refrigerants are not clearly understood and only limited research has been conducted on them. According to Cavallini et al. [2], these new refrigerant mixtures generally exhibit heat transfer degradation during the condensation process because of mass transfer diffusion, sensible heating effects, and non-equilibrium effects in two-phase flow (liquid and vapor phases). Thus, several modifications or corrections had been made to the pure-refrigerant condensation heat transfer models in order to account for the mass transfer thermal resistance during the condensation process.

The main objective of the current research is to develop a semi-empirical model for predicting heat transfer coefficients during condensation of refrigerant mixtures flowing inside different geometries of micro-fin tubes. The validity of the new semi-empirical model is to be evaluated with available experimental data.

2. Experimental database

A database was created from the available experimental data of refrigerant mixtures flowing inside micro-fin tubes. These data were collected from published work to help to develop the new refrigerant-mixture model and test the validity of the refrigerant-mixture model. These data were also used to validate some of the existing condensation heat transfer models. Tables 1 and 2 present the refrigerant mixtures experimental data for flow inside micro-fin tubes. Table 1 lists the flow conditions (saturation pressure, saturation temperature, heat flux, mass flux, and mean vapor quality) and Table 2 delineates the tube geometries (outer tube diameter, minimum wall thickness, fin height, number of fins, helix angle, apex angle, and the heated test section length). Most of the experimental data were presented at constant vapor quality with varying mass flux. The experimental data reported by Jeong et al. [15] were presented at constant mass flux with varying vapor quality.

The mean absolute deviation (MAD) is set as the condition to determine the effectiveness of a heat transfer model. MAD is defined as the average of the normalized difference between the predicted heat transfer coefficient and the experimental heat transfer coefficient.

$$\mathrm{MAD} = \frac{1}{N} \cdot \sum \frac{|h_{\mathrm{exp}} - h_{\mathrm{pre}}|}{h_{\mathrm{exp}}} \tag{1}$$

The heat transfer model is considered acceptable if the achieved MAD value is less than 30%.

3. New refrigerant mixtures model

The new refrigerant-mixture model presented in this paper is based on the model for pure-refrigerant developed by Chamra et al. [9]. The pure-refrigerant model developed by Chamra et al. [9] was based on a theoretical analysis of turbulent film condensation inside smooth tubes. The flow was assumed to be in the annular regime over the length of the tube. The expression to compute the heat transfer coefficient for pure-refrigerant during condensation inside micro-fin tubes reported by Chamra et al. [9] is given by

$$h = \frac{0.208 \cdot \rho_{\mathrm{l}} \cdot c_{\mathrm{pl}} \cdot \left(\frac{\tau_{\mathrm{w}}}{\rho_{\mathrm{l}}}\right)^{0.224}}{T^{+}} \cdot Rx^{1.321} \tag{2}$$

where h is the heat transfer coefficient in W m^{-2} K^{-1}, T^{+} is the dimensionless temperature, τ_{w} is the wall shear stress, ρ_{l} is the density of the liquid phase, c_{pl} is the specific heat of the liquid phase, and Rx is the enhancement factor, which can be determine as

$$Rx = \left[\frac{2 \cdot e \cdot n_{\mathrm{g}} \cdot \left(1 - \sin\left(\frac{\beta}{2}\right)\right)}{\pi \cdot d_i \cdot \cos\left(\frac{\beta}{2}\right)} + 1\right] \cdot \frac{1}{\cos(\gamma)} \tag{3}$$

A detailed description of the procedure used to develop Eq. (2) can be found in Chamra et al. [9]. Eq. (2) was modified using the Silver [17] and Bell and Ghaly [18] correction factors to account for mass transfer thermal resistance between the liquid and vapor phases. The use of these two correction factors were suggested by Cavallini et al.

Table 1
Flow conditions for refrigerant mixtures flowing inside micro-fin tubes

Reference	Runs	Fluid	P_{sat} (kPa)	T_{sat} (°C)	q (kW m^{-2})	G (kg m^{-2} s^{-1})	x (mean)
Bogart and Thors [10]	11	R410A		40.6		200–850	0.80–0.10
Ebisu et al. [11]	7	R32 + R134a		50		85–530	0.80–0.10
Ebisu et al. [12]	4	R407C	1974.4	47	7.5	140–400	0.10–0.70
Eckels et al. [13]	13	R410A		40		100–630	0.95–0.05
	9	R407C		40			
Goto et al. [14]	28	R407C		40		70–600	0.50
Jeong et al. [15]	21	R410A		31		90–210	0.10–0.90
Tang et al. [16]	37	R410A		40		250–850	0.50

Table 2
Tube geometries for refrigerant mixtures flowing inside micro-fin tubes

Reference	Tube material	d_{o} (mm)	th (mm)	e (mm)	n_{g}	γ (°)	β (°)	L (m)
Bogart and Thors [10]	Copper	9.53	0.33	0.203	60	18	50	3.66
Ebisu et al. [11]	Copper	7.0	0.30	0.18	50	18	40	3.0
Ebisu et al. [12]	Copper	7.0	0.25	0.18	50	18	40	0.54
Eckels et al. [13]	Copper	9.53	0.305	0.203	60	18	51	3.78
		15.88	0.635	0.305	60	27	45	3.81
		7.94	0.3	0.203	50	18	57	3.78
Goto et al. [14]	Copper	9.52	0.36	0.17	60	25	50	1.0
		6.35	0.34	0.14	55	16	50	1.0
Jeong et al. [15]	Copper	9.52	0.30	0.20	60	18	53	0.63
Tang et al. [16]	Copper	9.52	0.36	0.2	60	0	15	2.83
					72	18	40	

[1]. The expression to compute the heat transfer coefficient for refrigerant mixtures during condensation inside micro-fin tubes can be expressed as

$$h_{\mathrm{m}} = \left[\frac{1}{\frac{M_1 \cdot \rho_1 \cdot c_{\mathrm{pl}} \cdot \left(\frac{\tau_{\mathrm{w}}}{\rho_1}\right)^{M_2}}{T^+} \cdot Rx^{M_3}} + \left(\frac{\delta Q_{\mathrm{SV}}}{\delta Q_{\mathrm{T}}}\right) h_{\mathrm{v}}^{-1} \right]^{-1} \tag{4}$$

where h_{v} is the heat transfer coefficient of the vapor phase flowing alone in the duct, $\delta Q_{\mathrm{SV}}/\delta Q_{\mathrm{T}}$ is the ratio between the sensible heat duty removed by cooling the vapor and the total heat flow rate exchanged, and M_1, M_2, and M_3 are empirical constants.

The heat transfer coefficient of the vapor phase flowing alone in the duct is given by

$$h_{\mathrm{v}} = 0.023 \cdot \frac{k_{\mathrm{v}}}{d_i} \cdot \left(\frac{G_{\mathrm{v}} \cdot d_i}{\mu_{\mathrm{v}}}\right)^{0.8} \cdot \left(\frac{c_{\mathrm{pv}} \cdot \mu_{\mathrm{v}}}{k_{\mathrm{v}}}\right)^{0.3} \tag{5}$$

and the term $\delta Q_{\mathrm{SV}}/\delta Q_{\mathrm{T}}$ can be determined from:

$$\frac{\delta Q_{\mathrm{SV}}}{\delta Q_{\mathrm{T}}} \approx x \cdot c_{\mathrm{pv}} \cdot \left(\frac{\Delta TG}{i_{\mathrm{fg\_m}}}\right) \tag{6}$$

where $i_{\mathrm{fg\_m}}$ is the enthalpy of isobaric condensation of the mixture, and ΔTG is the temperature glide.

The dimensionless temperature, T^+, is defined as

$$T^+ = \delta^+ \cdot Pr_1 \quad \delta^+ \leqslant 5 \tag{7}$$

$$T^+ = 5 \cdot \left[Pr_1 + \ln\left[1 + Pr_1\left(\frac{\delta^+}{5} - 1\right)\right]\right] \quad 5 < \delta^+ \leqslant 30 \tag{8}$$

$$T^+ = 5 \cdot \left[Pr_1 + \ln(1 + 5 \cdot Pr_1) + 0.5 \cdot \ln\left(\frac{\delta^+ - 2.5}{27.5}\right)\right] \quad \delta^+ > 30 \tag{9}$$

The dimensionless condensate film thickness, δ^+, for laminar flow can be found by using the Nusselt [19] correlation:

$$ta^+ = 0.866 Re_1^{0.5} \quad \text{for } Re_1 \leqslant 1600 \tag{10}$$

For turbulent flow, the dimensionless film thickness can be found by using the Ganchev et al. [20] empirical correlation,

$$\delta^+ = 0.051 Re_1^{0.87} \quad \text{for } Re_1 > 1600 \tag{11}$$

The MathCad minimize function [21] is used to evaluate the three empirical constants presented in Eq. (3). The new empirical constants for the new refrigerant-mixture model were determined using 110 data points collected from six different sources listed in Table 3. This yields values of $M_1 = 0.31$, $M_2 = 0.314$, and $M_3 = 0.993$.

Table 3
Refrigerant-mixture data sets used for generating the new empirical constants

Reference	Refrigerant	Number of data points
Bogart and Thors [10]	R410A	11
Jeong et al. [15]	R410A	21
Tang et al. [16]	R410A	37
Ebisu et al. [12]	R407C	4
Eckels et al. [13]	R407C	9
Goto et al. [14]	R407C	28

Introducing the values of the empirical constants into Eq. (3) the final expression to compute the heat transfer coefficient for refrigerant mixtures during condensation inside micro-fin tubes can be expressed as

$$h_{\mathrm{m}} = \left[\frac{1}{\frac{0.31 \rho_1 \cdot c_{\mathrm{pl}} \cdot \left(\frac{\tau_{\mathrm{w}}}{\rho_1}\right)^{0.314}}{T^+} \cdot Rx^{0.993}} + \frac{x \cdot c_{\mathrm{pv}} \cdot \left(\frac{\Delta TG}{i_{\mathrm{fg\_m}}}\right)}{h_{\mathrm{v}}} \right]^{-1} \tag{12}$$

where the heat transfer coefficient is in W m^{-2} K^{-1}.

The constant M_1 must have the dimension m$^{0.338}$/s$^{0.338}$ in order for the equation to yield the correct dimensions, W m^{-2} K, for the heat transfer coefficient. The new refrigerant-mixture model is used to predict existing experimental data and the mean absolute deviation (MAD) value is calculated. The prediction results using the new refrigerant-mixture model for the available refrigerant-mixture data sets are presented in Table 4.

Table 4
Mean absolute deviation (MAD) achieved by the new refrigerant-mixture model for the data sets used in generating the new empirical constants

Reference	Refrigerant	MAD value (%)
Bogart and Thors [10]	R410A	3.7
Jeong et al. [15]	R410A	21.4
Tang et al. [16]	R410A	5.8
Ebisu et al. [12]	R407C	7.9
Eckels et al. [13]	R407C	4.8
Goto et al. [14]	R407C	10.9

Table 4 shows that the prediction results using the new refrigerant-mixture model are excellent since the mean absolute deviations for all these refrigerant mixtures data sets are less than 22%. Most of the mixture-refrigerant data sets are predicted within 10%. The prediction results for the R410A and R407C data sets using the new refrigerant-mixture model are illustrated in Figs. 1 and 2, respectively.

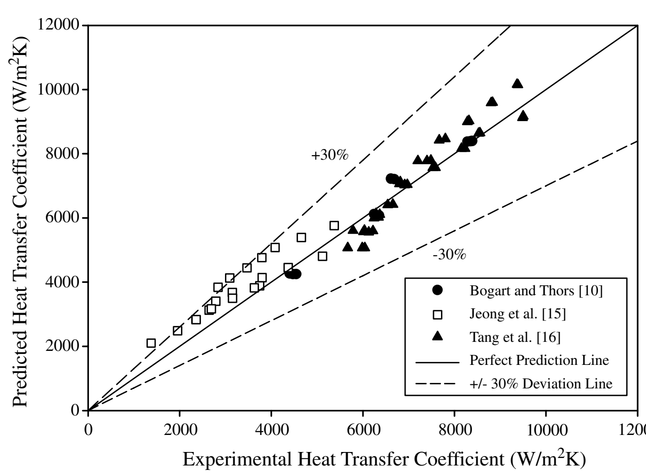

Fig. 1. New refrigerant-mixture model for the R410A data sets used in generating the new empirical constants.

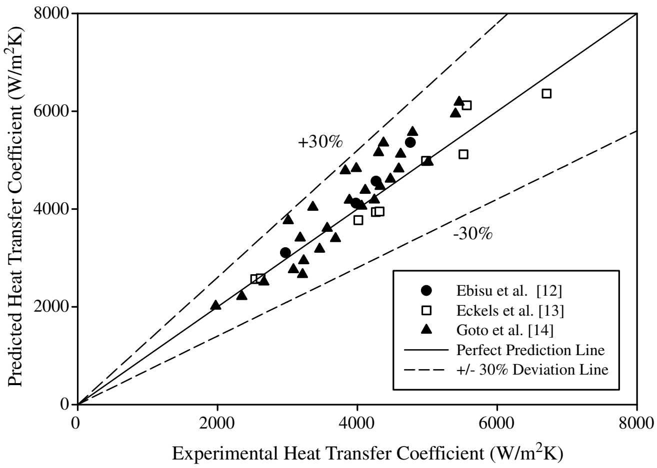

Fig. 2. New refrigerant-mixture model for the R407C data sets used in generating the new empirical constants.

From Figs. 1 and 2 it can be seen that the new refrigerant-mixture model has the capability to produce excellent prediction results for the available data sets. The new refrigerant-mixture model is further tested with additional data sets not used in developing the present model.

4. Comparison between the prediction results of the new refrigerant-mixture model and experimental data from a tube manufacturing

A manufacturing tube company provided two experimental condensation data sets for the Turbo-A and the Turbo-A crosscut geometries using R410A as the working fluid. The flow conditions of the R410A experimental data are listed in Table 5, and the geometries of the micro-fin tubes used are presented in Table 6.

The experimental data provided by the manufacture are plotted and compared with the predicted results of the new refrigerant-mixture model. Figs. 3 and 4 present the comparison for the R410A data sets for the Turbo-A and Turbo-A crosscut, respectively.

From Figs. 3a and 4a can be observed that the new model results agree well with the experimental results provided by the manufacturing company. Also, Figs. 3b and 4b show that the MAD values for both cases are within ±30%. The mean absolute deviation for R410A Turbo-A data set is 10.51%, while for R410A Turbo-A with crosscut data set is 7.24%. The new refrigerant-mixture model accurately predicts both the R410A data sets from the manufacture. Table 7 presents the summary of the MAD achieved by all the R410A data sets.

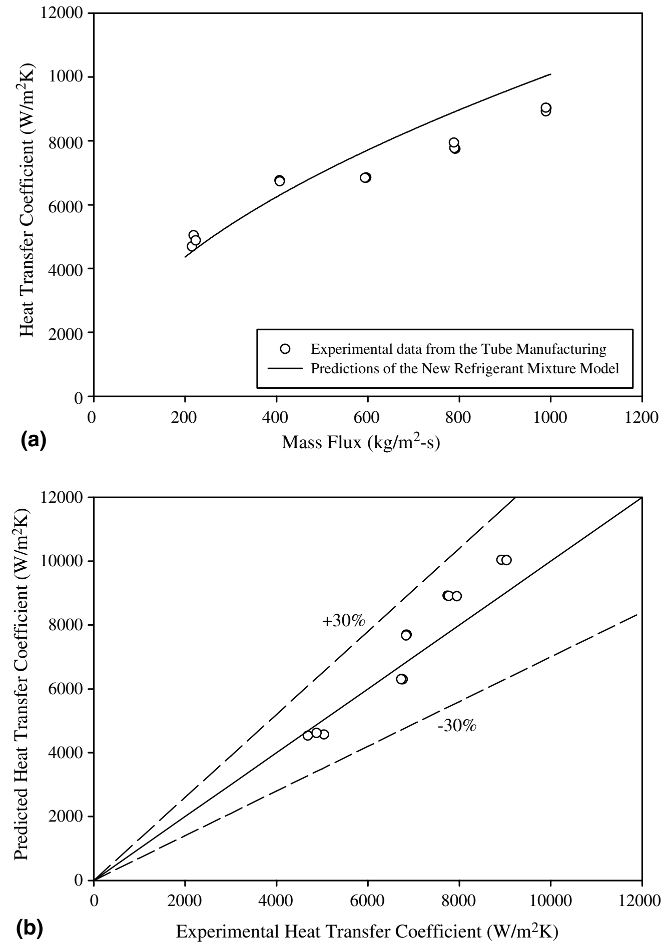

Fig. 3. New refrigerant-mixture model for the R410A Turbo-A data set from the tube manufacturing.

Table 5
Flow conditions for R410A flowing inside micro-fin tubes

Experimental condition	Range
Saturation temperature (°C)	37.8
Saturation pressure (kPa)	2290
Mass flux (kg m^{-2} s)	180–1100
Mean vapor quality	0.5

5. Comparison with other experimental data sets

The new refrigerant-mixture model was also tested with additional experimental data sets from other sources. These data sets were not included in developing the new refrigerant-mixture model. The flow conditions of the new experimental data points and the tube geometries used for the

Table 6
Tube geometries of the micro-fin tubes

Tube type	Tube material	d_o (mm)	th (mm)	e (mm)	n_g	γ (°)	β (°)	L (m)
Turbo-A	Copper	9.53	0.33	0.203	60	18	50	3.66
Turbo-A crosscut	Copper	9.53	0.33	0.203	60	18	50	3.66

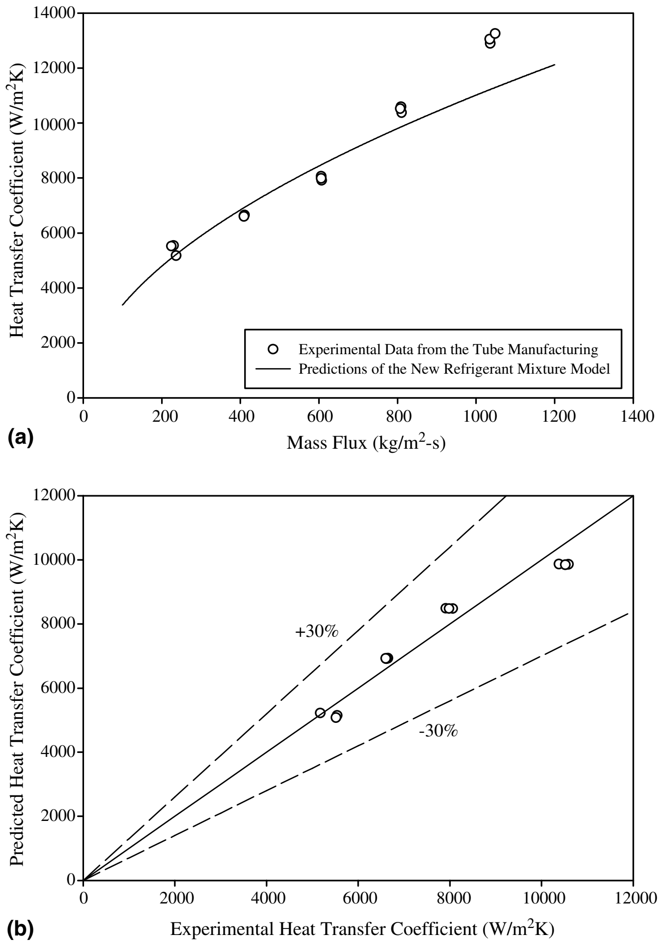

Fig. 4. New refrigerant-mixture model for the R410A Turbo-A crosscut data set from the tube manufacturing.

Table 7
Mean absolute deviation (MAD) achieved by the new refrigerant-mixture model for the R410A data sets from tube manufacturing

Refrigerant	Tube type	MAD (%)
R410A	Turbo-A	10.5
R410A	Turbo-A with crosscut	7.2

Table 8
Mean absolute deviation (MAD) achieved by the new refrigerant-mixture model on the refrigerant-mixture data sets

No.	Reference	Refrigerant	MAD value (%)
1	Ebisu et al. [11]	R32/R134a (30%/70%)	9.2
2	Eckels et al. [13]	R410A	16.2

new data sets are listed in Tables 1 and 2, respectively. The predictions of the new refrigerant-mixture model for the new experimental data points are presented in Table 8.

The two new experimental data sets were compared with the predicted results of the new refrigerant-mixture model. The achieved MAD value is within 17%. This shows that the new refrigerant-mixture model is capable of predicting

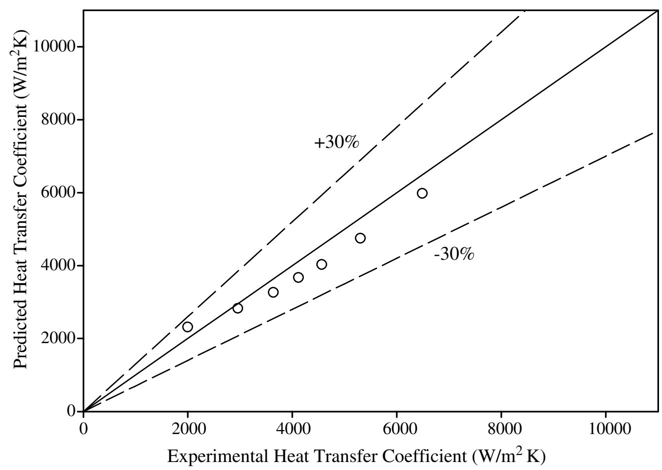

Fig. 5. New refrigerant-mixture model for the Ebisu et al. [11] R32/R134a (30%/70%) data set.

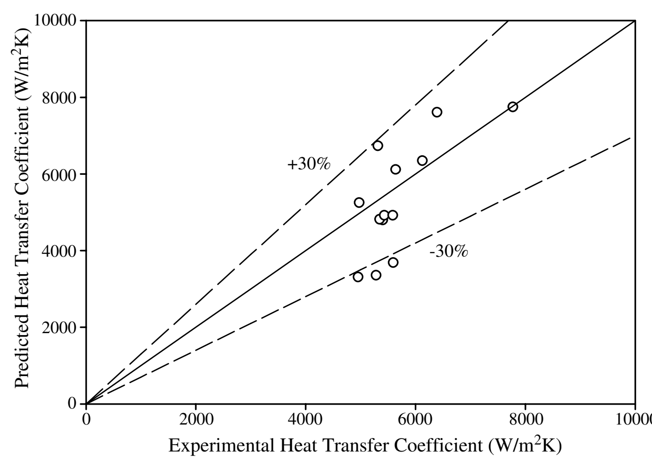

Fig. 6. New refrigerant-mixture model for the Eckels et al. [13] R410A data set.

the experimental data sets accurately. The prediction results of the new refrigerant-mixture model for these two new data sets are illustrated in Figs. 5 and 6.

Table 9 presents a comparison of the mean absolute deviations of the new refrigerant-mixture model, the Cavallini et al. [1] model, and the Kedzierski and Goncalves [4] model with the refrigerant mixtures data sets. As can be seen, the new pure-refrigerant model MAD values are within 22% for most of the experimental data points. It can also be observed that in most cases the MAD values for the new model are lower than those obtained using the Cavallini et al. [1] model and the Kedzierski and Goncalves [4] model.

6. Conclusions

A semi-empirical model for heat transfer coefficient calculation during condensation of refrigerants mixtures inside micro-fin tubes is presented. The new refrigerant-mixture model is developed based on a theoretical analysis of turbulent film condensation inside smooth tubes with

Table 9
Comparison of the mean absolute deviation (MAD) of different models for the refrigerant-mixture data sets

	Reference	Refrigerant	MAD value (%)		
			New mixture model	Cavallini et al. (1999) model	Kedzierski and Goncalves (1999) model
1	Bogart and Thors [10]	R410A	3.7	28.5	32.3
2	Ebisu et al. [11]	R32/R134a (30%/70%)	9.2	41.9	17.0
3	Ebisu et al. [12]	R407C	7.9	52.5	8.5
4	Eckels et al. [13]	R410A	16.2	23.5	14.3
		R407C	4.8	22.6	14.3
5	Goto et al. [14]	R407C	10.9	50.8	49.2
6	Jeong et al. [15]	R410A	21.4	81.2	25.2
7	Tang et al. [16]	R410A	5.8	47.1	11.2

several modifications to account for mass transfer thermal resistance between the liquid and vapor phases. The new refrigerant-mixture model is validated with the other experimental data that are not used in developing the model. Around 200 experimental data points are used to evaluate the validity of the new refrigerant-mixture model. The new refrigerant-mixture model is able of producing consistent prediction results with lower MAD values for most of the available data sets as compares to the other models.

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