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International Journal of Heat and Mass Transfer 45 (2002) 2003–2010

International Journal of HEAT and MASS TRANSFER

www.elsevier.com/locate/ijhmt

A generalized correlation for evaporation heat transfer of refrigerants in micro-fin tubes

Rin Yun, Yongchan Kim *, Kookjeong Seo, Ho Young Kim

Department of Mechanical Engineering, Korea University, Anam-Dong, Sungbuk-Gu, Seoul 136-701, Republic of Korea

Received 23 March 2001; received in revised form 18 October 2001

Abstract

A generalized correlation for flow boiling heat transfer in horizontal micro-fin tubes was developed by implementing non-dimensional parameters accounting for heat transfer enhancement over smooth tubes and physical phenomena into the basic form of a smooth tube correlation. The enhancement factor in nucleate boiling consists of surface tension and turbulence effects generated by the liquid flow. A modified Reynolds number having some similarities with the roughness Reynolds number is introduced into the model to estimate heat transfer enhancement in convective boiling. The ratio of the liquid film thickness to the fin height is also employed in the correlation. The database of the present correlation includes 749 data points for five different refrigerants. The present correlation has a relatively simpler form to employ and yields closer fit to the experimental data with a mean deviation of 20.5% as compared to the existing correlations in the literature.

Keywords: Evaporation; Heat transfer coefficient; Micro-fin tube; Correlation

1. Introduction

During the past few decades, many aspects of boiling heat transfer have been investigated and a large number of correlations have been proposed for flow boiling of refrigerants inside horizontal smooth tubes. However, a generalized correlation for evaporation heat transfer in micro-fin tubes is limited in the open literature due to lack of database and difficulties in a generalization of complicated phenomena. The lack of the correlation for micro-fin tubes with alternative refrigerants of R22 is one of the limitations in the design of efficient heat exchangers. The objective of the present study is to develop a simple form of a generalized correlation for the evaporation heat transfer inside horizontal micro-fin tubes. This paper focuses on the micro-fin tubes typically used in air-conditioners and refrigeration system as shown in Table 1.

Many data for the evaporation heat transfer in micro-fin tubes have been published. Borgart and Thors [1] and Muzzio et al. [2] conducted an experimental work on the evaporation of refrigerants with different micro-fin tubes. Muzzio et al. [2] compared their results with the Kandlikar and Raykoff correlation [3]. Chamra and Webb [4] reported that a micro-fin tended to spread out the liquid around the circumference of a cross-grooved tube. Reid et al. [5] measured boiling heat transfer coefficients of R113 with eight micro-fin tube geometries, and Kaul et al. [6] investigated the effects of heat flux on heat transfer coefficients with seven different refrigerants. Khanpara et al. [7,8] conducted a detailed study on the performance of a micro-fin tube with R22 and R113. They reported that the evaporation heat transfer coefficient was dependent on heat flux and mass flux, but it was relatively independent on vapor quality. Murata and Hashizume [9] investigated evaporation heat transfer coefficients for a refrigerant mixture of R123/R134a in grooved tubes, and Goto et al. [10] reported those for R410A and R407C in micro-fin tubes. Khanpara et al. [8] and Darabi et al. [11] commented that the measured heat transfer coefficient could vary with respect to a heating method to the test section. A

* Corresponding author. Tel.: +82-2-3290-3366; fax: +82-2-921-5439.

E-mail address: yongckim@korea.ac.kr (Y. Kim).

0017-9310/02/$ - see front matter
PII: S0017-9310(01)00321-0

Nomenclature

Bo	Boiling number
C	constant, coefficient
c_f	local friction coefficient
D_i	maximum inside diameter of a micro-fin tube (mm)
D_r	diameter of a micro-fin tube at fin root (mm)
D_{od}	outside diameter (mm)
E	modified Reynolds number
EH	electric-resistance heating method
FH	fluid heating method
E'_{cb}	modified augmentation factor for convective boiling term
E'_{nb}	modified augmentation factor for nucleate boiling term
f	fin height (mm)
F_P	modified reduced pressure $((P/P_C)^{0.23}/[1-0.99(P/P_C)]^{0.9})$
G	mass flux (kg/m^2 s)
g	gravitational acceleration (m^2/s)
h	evaporation heat transfer coefficient (W/m^2 K)
k	thermal conductivity (W/m K)
n	exponent of Reynolds number
M	molecular weight (kg/kmol)
P	pressure (Pa)
Pr_l	liquid Prandtl number
q	heat flux (W/m^2)
Re_l	Reynolds number for liquid phase flowing alone
Re_f	Reynolds number (characteristic length = fin height)
Re_k	roughness Reynolds number
S	non-dimensional variables
T_e	evaporating temperature (K)
u	velocity (m/s)
X_{tt}	Martinelli parameter
x	vapor quality

Greek symbols

α	apex angle (°)
ε	void fraction
δ	liquid film thickness (mm)
ρ	density (kg/m^3)
σ	surface tension (N/m)
β	helix angle (°)
μ	dynamic viscosity (mPa s)
ν	kinematic viscosity (m^2/s)

Subscripts

C	critical value
cv	convective boiling
g	vapor
l	liquid
PB	pool boiling
sat	saturation
tp	two-phase

Table 1
Typical geometries of micro-fin tubes

D_{od} (mm)	No. of micro-fins	β (°)	f (mm)	Fin shape	α (°)
4–15	50–70	6–30	0.10–0.3	Triangular or trapezoidal	25–90

more comprehensive review on the evaporation heat transfer can be found in [12].

Kandlikar and Rayoff [3], and Kandlikar [13] predicted the evaporating heat transfer coefficient in micro-fin tubes by introducing two enhancement factors into a smooth tube correlation, but their enhancement factors were specific empirical constants for each refrigerant and micro-fin tube. The Koyama et al. [14] correlation was derived by the superposition of nucleate boiling and convective boiling including the effects of surface tension and bubble detach size. However, their model did not have a general format because the dimensions of micro-fins could not be input into the correlation. Thome et al. [15] developed a generalized model for evaporation inside micro-fin tubes. The nucleate boiling was expressed as a function of heat flux, while the convective boiling was calculated from a turbulent film flow equation including the liquid film thickness. However, Thome et al. [15] had a limited spectrum of database and did not properly employ vapor quality into their correlation. Cavallini et al. [16] also presented a micro-fin prediction model by employing their correlation form of condensation into convective boiling. They included the fin height and surface tension in their model, but their model was considerably complicate to use.

Even though, lots of work has been performed as to micro-fin tubes, there are some limitations on the application of the correlations due to an improper selection of enhancement factors and a narrow spectrum of database. It would be desirable for a new correlation to contain non-dimensional variables derived with physical meaning and to have a very broad spectrum of database. In addition, the correlation needs to be generated in a simple format for easy applications. The correlation proposed in this paper is intended to satisfy these requirements.

2. Database of the model

The experimental data obtained from the literature were represented as a function of saturation temperature, mass flux, heat flux, quality, fin height, helix angle, tube diameter and number of fins. The database for the present micro-fin tube model is summarized in Table 2. The test ranges and geometries for each data set are also included. The database have a wide range of inner tube diameters from 8.82 to 14.66 mm, fin heights from 0.12 to 0.381 mm, spiral angles from 16° to 30°, mass fluxes from 50 to 637 $\mathrm{kg/(m^2\ s)}$, heat fluxes from 5 to 39.5 $\mathrm{kW/m^2}$, and evaporating temperatures from −15 to 70°C. The range of the database will impose limitations on the application of the correlation.

All data from a given reference have been employed within the limits to avoid any subjectivity in selecting data points. A possible shortcoming of the procedure is that the database becomes unduly weighted towards just one or two sources including a very large number of data [17]. In this study, the major data sources are [18] (419 points) and [19] (593 points). Although these two references covered a reasonable range of test conditions and geometries, only 151 points from [18] and 214 points from [19], which were evenly sampled from the original data sets, were used in the development of the correlation. The database of the present correlation has 749 points for five different refrigerants: R22, R113, R123, R134a, and R410A. However, all data were utilized in the comparison of the correlation with the database. Therefore, the deviations were determined based on the extended database of 1330 points for five refrigerants.

3. Development of the micro-fin tube flow boiling model

Generally, the correlations for saturated flow boiling in smooth tubes were expressed by the combination of nucleate boiling and forced convection [17]. The nucleate boiling is characterized by the presence of active nucleation sites, and it is expressed as a function of Boiling number, Bo (Eq. (1)). The two-phase forced convection is characterized not only by the suppression of nucleate boiling but also by evaporation heat transfer between a liquid film and a vapor core. The forced convection heat transfer is governed by mass flow rate and vapor quality, which is expressed as a function of Martinelli parameter, X_{tt} (Eq. (2)).

$$Bo = \frac{q}{\Delta h G}, \tag{1}$$

Table 2
Database for the present correlation

Data source	Refrigerant	D_i (mm)	f (mm)	β (°)	G ($\mathrm{kg/m^2\ s}$)	q ($\mathrm{kW/m^2\ s}$)	T_e (°C)	Heating method	No. of data point
Chamra and Webb [4]	R22	14.66	0.35/0.17	15/−15	151/255/327	17.3/26.5	24.0	FH	34
Kaul et al. [6]	R22	8.525			314	20/30	4.4	FH	9
Khanpara et al. [7,8]	R22	8.825	0.22	16–17.5	270–540	5–13	2.5	EH/FH	36
Seo and Kim [18]	R22	8.82	0.12	25	70–211	5–15	−15 to 5	EH	151 (456[a])
Reid et al. [5]	R113	8.509/8.712	0.211/0.381	17.5/30.0	234–637	15.1–39.5	70	EH	124
Khanpara et al. [7,8]	R113	8.825	0.22	16–17.5	248–600	17–40	70	EH/FH	63
Murata et al. [9]	R123	10.7	0.3	30.0	93–278	10/30	47	EH	22
Kattan et al. [24]	R123	11.90	0.25	18	101.1–300.5	5–25	29.9	FH	27
Singh et al. [23]	R134a	11.78	0.3	18	50	5–30	20.15	EH	24
Kattan et al. [24]	R134a	11.90	0.25	18	102.3–301.6	5–25	10.1	FH	45
Kim et al. [19]	R410A	8.82	0.12	25	70–211	5–15	−15 to 5	EH	214 (493[a])

[a] It indicates number of data points used in the calculation of deviations shown in Table 5.

$$X_{\mathrm{tt}} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_{\mathrm{v}}}{\rho_{\mathrm{l}}}\right)^{0.5}\left(\frac{\mu_{\mathrm{l}}}{\mu_{\mathrm{v}}}\right)^{0.1}. \tag{2}$$

The present correlation for micro-fin tubes basically has the form of a smooth tube correlation. Generally, the ratio of flow boiling heat transfer coefficient, h_{tp}, in smooth tubes to single-phase convection heat transfer coefficient, h_{l}, can be represented as a function of Bo and X_{tt}.

$$\frac{h_{\mathrm{tp}}}{h_{\mathrm{l}}} = f(Bo, X_{\mathrm{tt}}), \tag{3}$$

where

$$h_{\mathrm{l}} = 0.023 Re_{\mathrm{l}}^{0.8} Pr_{\mathrm{l}}^{0.4}\left(\frac{k_l}{D_{\mathrm{i}}}\right). \tag{4}$$

The non-dimensional terms of Bo and X_{tt} in Eq. (3) can be applied to the micro-fin tube model with a modification using correction factors. Generally, the effects of heat flux on the heat transfer coefficient are more pronounced in a smooth tube than in a micro-fin tube [9]. For micro-fin tubes, the turbulence effects generated by the liquid film are more dominant factor for heat transfer coefficients at low qualities as compared to an enhancement of nucleating boiling resulted from heat flux. However, the enhancement of heat transfer in micro-fin tubes with an increase of heat flux tends to have the similar pattern as that in smooth tubes. Murata and Hashizume [9] reported that the correlations of nucleate boiling heat transfer for micro-fin and smooth tubes, which are given in Eqs. (5) and (6), respectively, had the same exponent of heat flux. Therefore, the Bo can be employed in the correlation for micro-fin tubes, but it has to be modified by including the effects of turbulence and surface tension.

$$h_{\mathrm{PB,plain}} = 31.4\left(\frac{P_{\mathrm{C}}^{0.2}F_P}{M^{0.1}T_{\mathrm{C}}^{0.9}}\right)q_{\mathrm{PB}}^{0.8}, \tag{5}$$

$$h_{\mathrm{PB,micro\text{-}fin}} = 48\left(\frac{P_{\mathrm{C}}^{0.2}F_P}{M^{0.1}T_{\mathrm{C}}^{0.9}}\right)q_{\mathrm{PB}}^{0.8}. \tag{6}$$

For two-phase flow, the changes of liquid film thickness, velocity and void fraction are obviously dependent on quality, x. Although the trends of those change are different between a smooth and micro-fin tube, these phenomena can be well implemented in terms of X_{tt} by introducing additional correction factors for micro-fin tubes. In addition to Bo and X_{tt}, the following operating parameters are considered in the present correlation: surface tension, turbulence effects, fin height, liquid film thickness, evaporating temperature, and fluid properties. Table 3 summarizes the non-dimensional parameters employed in the present correlation for micro-fin tubes.

The variation of surface tension of the thin liquid film can alter wetting characteristics of the surface inside circumferential of a micro-fin tube. The turbulence effects generated by micro-fins also produce considerable influence on nucleate boiling heat transfer. These two factors happen to simultaneously improve the evaporation heat transfer at low vapor quality, which are combined into a non-dimensional factor as given in Eq. (7) to correct the effects of Bo on nucleate boiling. Evaporating pressure is also included to form the non-dimensional parameter.

Table 3
Variables employed in the present correlation

Effect factors	Non-dimensional parameters
Heat flux	Bo
Fluid properties	Pr
Evaporating temperature and surface tension	$(P_{\mathrm{sat}}D_{\mathrm{i}}/\sigma)$
Turbulence effect	$(Gf/\mu_{\mathrm{l}})Re_{\mathrm{l}}$
Fin height and liquid film thickness	δ/f
Vapor quality	X_{tt}

$$S = \left(\frac{P_{\mathrm{sat}}D_{\mathrm{i}}}{\sigma}\right)Re_{\mathrm{l}}. \tag{7}$$

To consider the turbulence effects produced by micro-fins in convective boiling, a non-dimensional parameter of E, which is a modified Reynolds number as given in Eq. (8), is included in the correlation. There are some similarities between E and the roughness Reynolds number, Re_k (Eqs. (9) and (10)), which depicts the effects of turbulence induced by surface roughness on single-phase convection heat transfer. The E is employed as an enhancement factor for the turbulence effects generated by micro-fins in forced convective boiling.

$$E = \left(\frac{Gf}{\mu_{\mathrm{l}}}\right)Re_{\mathrm{l}}, \tag{8}$$

$$Re_k = \frac{u_\tau f}{\nu_{\mathrm{l}}}, \tag{9}$$

where

$$u_\tau = uf\left(\frac{c_{\mathrm{f}}}{2}\right) = uf\left(\frac{1}{Re_{\mathrm{l}}}\right). \tag{10}$$

The effects of fin height and number of fins can be expressed by a relation between liquid film thickness, δ, and fin height, f. When the fin height is by far greater than the liquid film thickness, the increases of heat transfer area and turbulence effects, which are resulted from the increments of the fin height and number of fins, are relatively inefficient in an improvement of evaporation heat transfer. When the liquid film thickness is by far greater than the fin height, not only the liquid film becomes a thermal-resistance, but also the

Table 4
Coefficients of the present correlation (Eq. (13))

Coefficient	C_1	C_2	C_3	C_4	C_5	C_6	C_7	C_8	C_9
Value	0.009622	0.1106	0.3814	7.6850	0.5100	−0.7360	0.2045	0.7452	−0.1302

turbulence effects can be reduced. When the ratio of the liquid film thickness to the fin height is close to unity, the thin liquid film forms around the fin tips, and periodic liquid wave caused by the fins also makes the liquid flow unstable, which maximizes thermal efficiency and turbulence effects. Therefore, the non-dimensional parameter of δ/f is included in the correlation. The liquid film thickness, δ, is determined using the void fraction correlation for plain tubes suggested by Rouhani and Axelsson [20] as given in Eq. (12).

$$\delta = \frac{D_r(1-\varepsilon)}{4}, \tag{11}$$

where

$$\varepsilon = \left(\frac{x}{\rho_g}\right)\left[(1+0.12(1-x))\left(\frac{x}{\rho_g}+\frac{(1-x)}{\rho_g}\right) + \left(\frac{1.18(1-x)\left(g\sigma(\rho_l-\rho_g)\right)^{0.25}}{G\rho_l^{0.5}}\right)\right]^{-1}. \tag{12}$$

Several researches on the enhancement of the evaporation heat transfer by designing δ/f close to unity were conducted. Muzzio et al. [2] reported that higher and lower fin tubes had an advantage in establishing δ/f near unity. Miyara et al. [21] also mentioned that a herringbone tube more easily made δ/f close by unity. Itoh et al. [22] developed a more complex fin geometry, which increased the thin liquid film thickness over the fin tips.

The non-dimensional factors for micro-fin tubes derived in this study are added into Eq. (3) to account for heat transfer enhancements and physical phenomena. The enhancement factor for nucleate boiling in micro-fin tubes is $(P_{sat}D_i)/\sigma$, while that for convective boiling is $(Gf)/\mu_l$. Since Re_l, Pr_l, and δ/f represent significant influences on nucleate boiling as well as forced convection, they are implemented as the correction factors for both heat transfer mechanisms. Combining and rearranging the non-dimensional parameters, the present correlation for the evaporation heat transfer in micro-fin tubes is given by

$$\frac{h_{tp}}{h_l} = \left[C_1 \times Bo^{C_2}\left(\frac{P_{sat}\times D_i}{\sigma}\right)^{C_3} + C_4 \times \left(\frac{1}{X_{tt}}\right)^{C_5}\left(\frac{Gf}{\mu_l}\right)^{C_6}\right] Re_l^{C_7} Pr_l^{C_8}\left(\frac{\delta}{f}\right)^{C_9}. \tag{13}$$

The coefficients in Eq. (13), which are given in Table 4, were determined by a multiple regression analysis with the database listed in Table 2. It should be noted that as shown in Eq. (13), the influence of δ/f on the heat transfer coefficients is modified by the fin height included in the term of $(Gf)/\mu_l$.

4. Comparison with experimental data

To validate the new correlation, the predicted heat transfer coefficients were compared with the database of 1333 points for five refrigerants. In addition, the Thome et al. [15] and Cavallini et al. [16] correlations were also included in the comparison. Table 5 shows the comparison of the correlations with each data set in the database. A mean and an average deviation of the present correlation are 20.5% and −11.7%, respectively. The Thome et al. [15] correlation yields a mean deviation of 64.4% and an average deviation of 51.5%, while the Cavallini et al. [16] correlation provides a mean deviation of 36.4% and an average deviation of −16.1%. Fig. 1 shows the predicted heat transfer ratio of flow boiling to single-phase heat transfer coefficient vs. the measured value. Approximately 90% of the experimental data are correlated within a deviation of ±30%.

The present correlation and Cavallini et al. [16] correlation yield a reasonable agreement with the experimental data. A maximum mean deviation of the present correlation is 32.7% for Murata et al. [9] data. However, the Cavallini et al. [16] correlation has relatively high mean deviations for Singh et al. [23] and Katten et al. [24], which are 65.8% and 53.2%, respectively. The Thome et al. [15] correlation generally overpredicts heat transfer coefficients as compared with the experimental data, which provides a maximum mean deviation of 204.3% for Kim et al. [19] data. Large deviations are observed when the correlations are extended beyond their test range of the database such as test conditions, tube geometries, and micro-fin shape.

Average deviations with vapor quality are shown in Fig. 2. Based on the test conditions of Khanpara et al. [7,8] data, the deviations of the correlations were estimated. As the quality increases, the Thome et al. [15] correlation begins to overpredict heat transfer coefficients, while the Cavallini et al. [16] correlation starts to underpredict heat transfer coefficient as compared to

Table 5
Percentage deviations between the correlations and the experimental data

Data source	Chamra and Webb [4]	Kaul et al. [6]	Khanpara et al. [7,8]	Seo and Kim [18]	Reid et al. [5]	Khanpara et al. [7,8]	Singh et al. [23]	Kattan et al. [24]	Murata et al. [9]	Kattan et al. [24]	Kim et al. [19]	Total
Refrigerant	R22	R22	R22	R22	R113	R113	R134a	R134a	R123	R123	R410A	
No. of data	34	9	36	456	124	63	24	45	22	27	493	1333
Present correlation												
Mean dev. [a]	32.3	11.2	13.0	13.1	13.3	16.3	19.8	31.0	32.7	27.6	15.4	20.5
Average dev. [b]	−32.3	−7.4	−10.2	4.6	0.6	−9.2	−5.4	−17.9	−32.7	−27.6	8.6	−11.7
Thome et al. [15]												
Mean dev.	21.6	23.3	62.5	138.6	39.4	21.7	97.6	26.5	47.5	25.1	204.3	64.4
Average dev.	−3.4	23.3	52.8	138.6	17.6	2.2	95.6	4.4	47.5	−16.0	204.3	51.5
Cavallini et al. [16]												
Mean dev.	37.0	10.4	25.0	20.8	39.3	48.8	65.8	34.5	36.6	53.2	29.4	36.4
Average dev.	−37.0	7.9	−25.0	−11.7	−38.9	−48.8	65.8	−27.7	−36.6	−53.2	28.2	−16.1

[a] Mean dev. $= \frac{1}{n}\sum_1^n \mathrm{ABS}[(h_{\mathrm{pred}} - h_{\mathrm{exp}}) \times 100)/h_{\mathrm{exp}}]$.

[b] Ave dev. $= \frac{1}{n}\sum_1^n [(h_{\mathrm{pred}} - h_{\mathrm{exp}}) \times 100)/h_{\mathrm{exp}}]$.

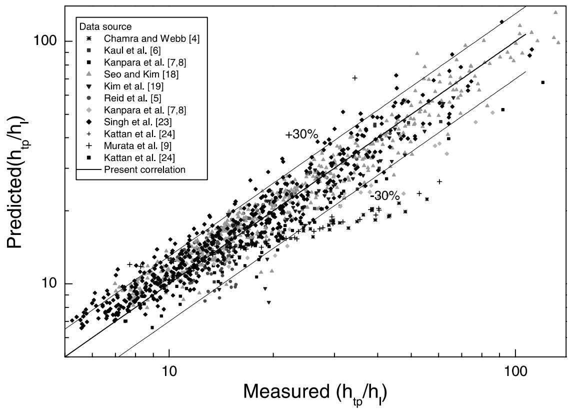

Fig. 1. Comparison of the correlations with the experimental data.

the data. The present correlation underpredicts the heat transfer coefficient at qualities near zero, but the predictions yields relatively good agreement with the data as the quality increases over 0.04. Fig. 3 shows a typical trend of the predicted heat transfer coefficients as a function of mass flux as well as a comparison with Schlager et al. [25] data. Both the present correlation and Cavallini et al. [16] correlation yield relatively good agreement with the data. Besides, the Thome et al. [15] correlation represents relatively little dependency of heat transfer coefficients on mass flux as compared to the other correlations.

Several researchers have investigated the effects of helix angle on the evaporation heat transfer. Recently, Chamra and Webb [26] explored the evaporation heat transfer as a function of helix angle with a typical micro-fin and cross-groove micro-fin. Although an optimum helix angle for the tubes existed, the effects of helix angle were not clear in boiling heat transfer. Since the data as

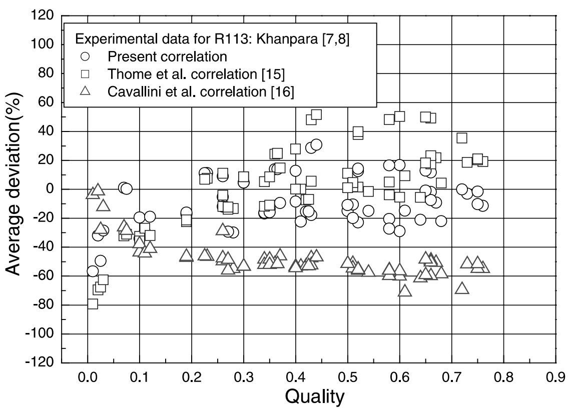

Fig. 2. Average deviations of the correlations as a function of quality.

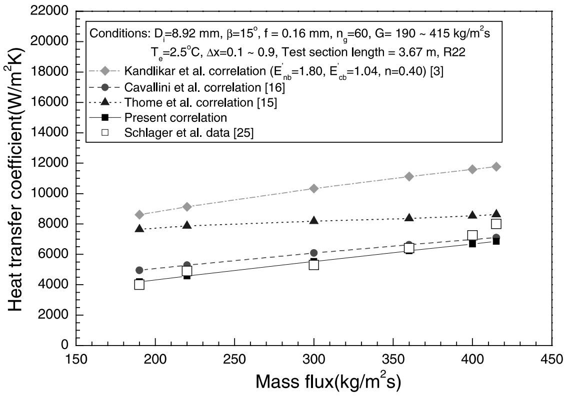

Fig. 3. Comparison of the correlations with Schlager et al. [25] data as a function of mass flux.

a function of helix angle were limited and the observed trends were not clear, the helix angle was not implemented into the present correlation. Further study on the influence of helix angle needs to be performed to improve the present correlation. The generalized form of the present correlation can be applicable to any micro-fin geometry and refrigerants. However, the model needs to be validated against more micro-fin geometry and more fluids to cover more wide range and new micro-fin shape.

5. Conclusions

In the present study, the factors affecting on the evaporating heat transfer in micro-fin tubes were explored, and the non-dimensional parameters were introduced to account for heat transfer enhancement and physical phenomena during the evaporation process. Not only the surface tension acting on the thin liquid film but also the turbulence effects generated by the liquid flow produce considerable influence on nucleate boiling heat transfer. The modified Reynolds number, which has some similarities with the roughness Reynolds number for single-phase convection heat transfer, is implemented as an enhancement factor due to the turbulence effects produced by micro-fins in convective boiling. Besides, the ratio of the liquid film thickness to the fin height plays a very important role on both the nucleate and convective boiling mechanisms in micro-fin tubes.

The generalized correlation for predicting evaporation heat transfer coefficients in micro-fin tubes was developed by including the non-dimensional parameters of $(P_{\mathrm{sat}}D_{\mathrm{i}}/\sigma)$, $(Gf/\mu_{\mathrm{l}})Re_{\mathrm{l}}$, Pr_{l} and δ/f into the basic form of the smooth tube correlation. The present correlation, which is relatively simple and generalized form, provides reasonable agreement with the experimental data. Compared to the extended database of 1333 points, the present correlation yields a mean deviation of 20.5% and an average deviation of −11.7%.

Acknowledgements

This work was supported by a grant No. KOSEF 1999-1-304-006-3 from the Korea Science and Engineering Foundation.

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