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Prediction of Two-Phase Heat Transfer in Horizontal Multi-Port Microchannels Using Probabilistic Flow Regime Maps

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ABSTRACT

Flow regime based heat transfer models are developed for refrigerants in horizontal multiport microchannels under evaporation and condensation conditions utilizing probabilistic two-phase flow regime maps developed for multiport microchannels. Probabilistic flow regime map time fraction information is used to provide a physically based weighting of heat transfer models developed for different flow regimes with ease of implementation. The developed models are compared with modified large tube models found in the literature and with condensation and evaporation data for R134a and R410A in 1.58mm hydraulic diameter horizontal four port microchannels.

1. INTRODUCTION

Probabilistic two-phase flow regime map based models have been found in the literature to accurately predict refrigerant pressure drop, void fraction and heat transfer, three important parameters in air conditioning and refrigeration system design, in large tubes by weighting flow regime specific models with consistent quantitative flow regime information and with ease of implementation. Similar probabilistic two-phase flow regime based void fraction and pressure drop models have been developed for refrigerants in multiport microchannels. In the present study probabilistic two-phase flow regime map based heat transfer models are developed for multiport microchannels on a consistent flow regime basis as the probabilistic two-phase flow map based void fraction and pressure drop models found in the literature. The present heat transfer models are evaluated by comparison with modified large tube models found in the literature and with condensation and evaporation data from Newell and Newell (2009) for R134a and R410A in 1.58mm hydraulic diameter horizontal four port microchannels.

2. LITERATURE REVIEW

Flow regime map based heat transfer models are seen by Jassim et al. (2008b) to be most accurate in large tubes with hydraulic diameters larger than approximately 4mm. They evaluated the flow regime map based condensation heat transfer models of Dobson and Chato (1998), Cavallini et al. (2003), and Thome et al. [2003] based on a database of R11, R12, R134a, R22, R410A, and R32/R125 (60/40% by mass) condensation heat transfer for a wide range of mass fluxes and tube diameters (the minimum diameter was 3.9mm). These flow regime map based

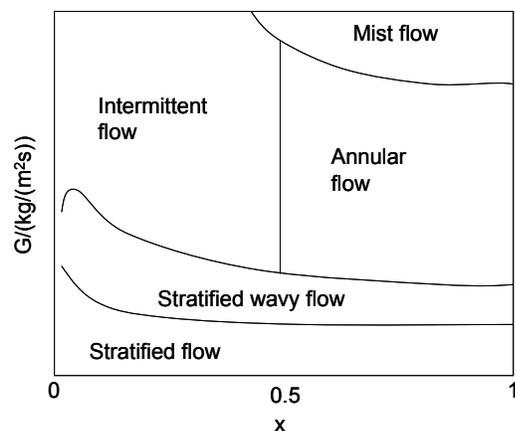
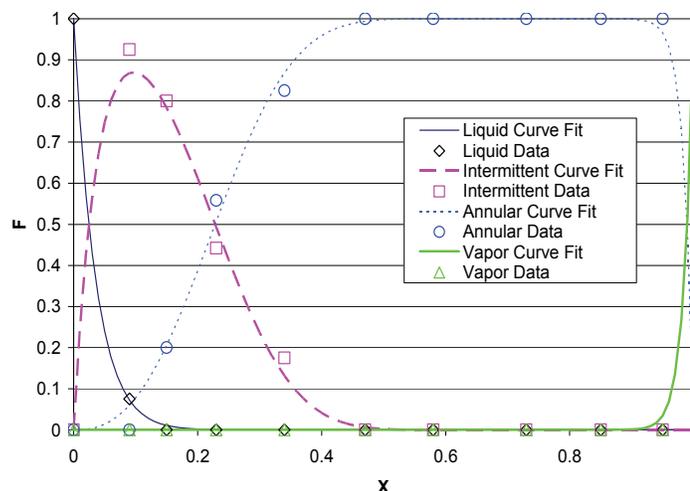


Figure 1. Steiner (1993) type flow map depiction

Figure 2. Probabilistic flow map for R410A, 10°C, 300kg/(m²s) in a 6-port 1.54 mm hydraulic dia. microchannel taken from Jassim and Newell (2006)

models, with average absolute errors between 11 and 13%, were found to be more accurate than two-phase flow models that did not utilize flow regime information from two-phase flow maps. All of the flow regime map based heat transfer models mentioned above utilize traditional Steiner (1993) or Taitel and Dukler (1976) type flow regime maps which indicate a particular flow regime at a given quality and mass flux. The distinct lines seem to lack a physical basis as Coleman and Garimella (2003), El Hajal et al. (2003), and Niño (2002) indicate that more than one flow regime can exist near the boundaries or within a given flow regime on a Steiner (1993) type flow map. Traditional flow maps such as Mandhane (1974), Steiner (1993), and Taitel and Dukler (1976) type flow maps can not be represented as continuous functions for the entire quality range as can be seen from a Steiner (1993) type flow map depicted in Figure 1. As a result, interpolation schemes are used to eliminate discontinuities as the flow regime boundaries, but these models remain to be difficult to implement as a result of the nature of the flow regime maps that they utilize.

Jassim et al. (2008b) developed a flow regime map based heat transfer model utilizing probabilistic flow regime maps for large tubes developed by Jassim et al. (2007) and Jassim (2006). Probabilistic flow regime maps, depicted in Figure 2 for multiport microchannels, indicate the fraction of time that a particular flow regime is observed under specific flow conditions. The flow regime time fractions are represented as continuous functions for the entire quality range, which eliminates discontinuities associated with traditional flow regime maps at the flow regime boundaries. Furthermore, the transitions are based on observations instead of interpolations. The probabilistic two-phase flow regime maps are generalized as explicit functions of physical parameters by Jassim (2006) for smooth tubes. The heat transfer modeling method used by Jassim et al. (2008b) weighted heat transfer models developed for particular flow regimes by the respective time fractions as shown in equation 1 below.

$$h_{total} = F_{int+liq} h_{int+liq} + F_{strat} h_{strat} + F_{ann} h_{ann} \quad (1)$$

In this way the model applies the appropriate assumptions (flow regime information) in a quantitative and explicit manner for a full range of flow conditions. This probabilistic flow map model is flexible in that it readily accepts different heat transfer relations as new relations are developed for particular flow regimes. To illustrate the versatility of the method the authors used the annular flow heat transfer models of both Dobson and Chato (1998) and Thome et al. (2003). This method, utilizing either of the annular flow models, was found to have comparable predictive capabilities to the traditional flow regime map based models of Dobson and Chato (1998), Cavallini et al. (2003), and Thome et al. (2003), but is easier to implement due to the nature of the probabilistic two-phase flow regime maps. Furthermore, this modeling technique allows for pressure drop, void fraction, and heat transfer to be modeled on a consistent flow regime time fraction weighted basis, as Jassim (2006a) and Jassim (2006) developed similar void fraction and pressure drop models, respectively.

Single tubes of approximately 3mm in diameter and larger are found to contain a stratified flow regime which is absent in the 1.54mm hydraulic diameter microchannels of Nino (2002). Damianides and Westwater (1988) support this observation because they indicate that the transition from “microchannel” behavior to “large tube”

behavior occurs in the 3mm tube diameter range. Furthermore, vapor only flow is not present below a quality of 100% in single tubes. Consequently, the probabilistic two-phase flow regime map based heat transfer models developed by Jassim et al. (2008) may not be directly applied to multiport microchannels with hydraulic diameters less than 3 mm.

Jassim and Newell (2006) developed a probabilistic two-phase flow map void fraction and pressure models for R410A, R134a, and air water in 1.54mm 6-port microchannels and is given in equations 2 and 3, respectively

$$\alpha_{total} = F_{liq} \alpha_{liq} + F_{int} \alpha_{int} + F_{vap} \alpha_{vap} + F_{ann} \alpha_{ann} \quad (2)$$

$$\left(\frac{dP}{dz}\right)_{total} = F_{liq} \left(\frac{dP}{dz}\right)_{liq} + F_{int} \left(\frac{dP}{dz}\right)_{int} + F_{vap} \left(\frac{dP}{dz}\right)_{vap} + F_{ann} \left(\frac{dP}{dz}\right)_{ann} \quad (3)$$

The probabilistic flow regime maps used were developed specifically for multiport microchannels with liquid, intermittent, vapor, and annular flow regimes present. The void fraction and pressure drop values are simply predicted as the sum of the products of the time fractions of each flow regime and the models representative of the respective flow regimes. A sample probabilistic flow map developed by Jassim and Newell (2006) is depicted in Figure 1b for R410A, 10°C, 300 kg/(m²s) in a 6-port 1.54 mm hydraulic diameter microchannel. Curve fits were made for the time fractions of each flow regime so that they can be explicitly determined in a model. However, the time fraction functions developed for multiport microchannels were not linked to physical parameters. Like the large tube models of Jassim et al. (2008a), and Jassim (2006) these models do not contain discontinuities because of the nature of the probabilistic flow regime maps used, and it allows for easy replacement of the flow regime models as more accurate models are identified for each flow regime.

3. PROBABILISTIC TWO-PHASE HEAT TRANSFER MODEL FOR MULTI-PORT MICROCHANNELS

The probabilistic two-phase heat transfer model for multiport microchannels proposed in the present study is given as

$$h_{total} = F_{liq} h_{liq} + F_{int} h_{int} + F_{vap} h_{vap} + F_{ann} h_{ann}, \quad (4)$$

where the heat transfer coefficient is predicted as the sum of the time fractions of each flow regime present multiplied by a heat transfer model that are representative of the respective flow regimes. The present heat transfer model is similar to that of Jassim et al. (2008b) but it utilizes flow regime maps, flow regimes, and heat transfer models that are specific to multiport microchannels. The time fractions utilized in this heat transfer model are obtained from the probabilistic two-phase flow regime maps developed by Jassim and Newell (2006) for multiport microchannels. In this way heat transfer can be predicted for multiport microchannels on the same flow regime basis as void fraction and pressure drop are predicted by Jassim and Newell (2006).

3.1 Liquid flow heat transfer model

The heat transfer model developed by Jassim et al. (2008) is utilized for the liquid flow regime. It is based on a modified Dittus-Boelter relation, given in equation 5, is developed for the intermittent/liquid flow regime.

$$h_{liq} = 0.023 \left(\frac{k_l}{D} \right) \left(\frac{GD}{\mu_l} \right)^{0.8} \text{Pr}_l^{0.3}, \text{ where} \quad (5)$$

$$\text{Pr}_l = \frac{\mu_l C_{p_l}}{k_l} \quad (6)$$

The Dittus-Boelter equation, which was developed for turbulent single-phase flow is modified by using the two-phase mass flux while using the liquid refrigerant properties. This model assumes that the vapor phase is not contributing to the heat transfer other than the fact that it is a contributor to the total mass flux. As the quality approaches zero, equation 5 approaches the liquid heat transfer equation, which is the correct physical limit. It should be noted that the same equation was used for both condensation and evaporation for the purpose of simplicity.

3.2 Intermittent flow heat transfer model

The stratified flow regime microchannel condensation model presented by Chen et al. (2008) is utilized for the intermittent flow regime given in equation 7.

$$h_{\text{int}} = 0.555 \left(\frac{k_l}{D} \right) \frac{\left(\frac{\rho_l (\rho_l - \rho_v) g h_{fg} D_h^3}{k_l \mu_l (T_{\text{sat}} - T_w)} \right)^{1/4}}{\left(1 + \frac{1-x}{x} \left(\frac{\rho_v}{\rho_l} \right)^{2/3} \right)^{-1}} + 0.023 \left(1 - \left(1 + \frac{1-x}{x} \left(\frac{\rho_v}{\rho_l} \right)^{2/3} \right)^{-1} \right) \text{Re}_l^{0.8} \text{Pr}_l^{0.4} \quad (7)$$

It should be noted that the stratified flow regime defined in Chen et al. (2008) is similar to the intermittent flow regime defined in the present paper.

3.4 Vapor flow heat transfer model

The heat transfer model for the vapor flow regime, like the liquid flow regime, is based on a modified Dittus-Boelter relation, given in equation 8.

$$h_{\text{vap}} = 0.023 \left(\frac{k_v}{D} \right) \left(\frac{GD}{\mu_v} \right)^{0.8} \text{Pr}_v^{0.3}, \text{ where} \quad (8)$$

$$\text{Pr}_v = \frac{\mu_v C_{p_v}}{k_v} \quad (9)$$

This model assumes that the liquid phase is not contributing to the heat transfer other than the fact that it is a contributor to the total mass flux. As the quality approaches one, equation 8 approaches the vapor heat transfer equation, which is the correct physical limit. It should be noted that, like the liquid model, the same equation was used for both condensation and evaporation for the purpose of simplicity.

3.5 Annular flow heat transfer model

Two different simple annular flow models are used in the present model for illustration purposes. A modified Shah (1979) model for annular flow, given in equation 10, is used to represent h_{ann} . The Shah (1979) model is a two-phase multiplier model for annular flow in large tubes. This model is found to collapse the heat transfer data for microchannels in annular flow, however it seems to be off by a factor of two. Therefore, the present annular flow model is the Shah model divided by two.

$$h_{\text{ann}} = \frac{0.023}{2} \left(\frac{k_l}{D} \right) \text{Re}_l^{0.8} \text{Pr}_l^{0.4} \left(1 + \frac{3.8 P_{\text{crit}}}{P} \right)^{0.38} \left(\frac{x}{1-x} \right)^{0.76}, \text{ where} \quad (10)$$

$$\text{Re}_l = \frac{GD(1-x)}{\mu_l} \quad (11)$$

The modified annular flow portion of the Dobson and Chato (1998) model, a two-phase multiplier/ Lockhart-Martinelli based model, was also used for h_{ann} in the present model. Like the modified Shah (1979) model, the annular flow portion of the Dobson and Chato (1998) model was divided by two as seen in equation 12 below.

$$h_{\text{ann}} = \frac{0.023}{2} \left(\frac{k_l}{D} \right) \text{Re}_l^{0.8} \text{Pr}_l^{0.4} \left(1 + \frac{2.22}{X_u^{0.889}} \right), \text{ where} \quad (12)$$

$$X_u = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_v}{\rho_l} \right)^{0.5} \left(\frac{\mu_l}{\mu_v} \right)^{0.1}, \text{ and} \quad (13)$$

4. PRESENT MODEL EVALUATION

The present heat transfer models are compared with evaporation and condensation average heat transfer data for R410A and R134a in a 1.58 mm hydraulic diameter four port microchannel with mass fluxes ranging from 100 to 300 kg/m²s, and a full quality range from Newell and Newell (2009). Heat transfer models from the literature are also compared against the present models. The two present models utilize the same vapor, liquid, and intermittent heat transfer relations above, but use different annular flow models, namely, the modified Dobson and Chato (1998) model and the modified Shah(1979) model. However, since the heat transfer data have large quality changes between the inlet and outlet of the test section the heat transfer models must be integrated to properly predict average heat transfer coefficients. The local heat transfer coefficients were computed in increments of change in quality, Δx , of 1%. The heat transfer associated with a 1% change in quality at n locations along the length of the microchannel is given as

$$\dot{m} h_{fg} \Delta x = h_1 A_1 \Delta T, \quad \dot{m} h_{fg} \Delta x = h_2 A_2 \Delta T, \quad \dots \quad \dot{m} h_{fg} \Delta x = h_n A_n \Delta T \quad (14)$$

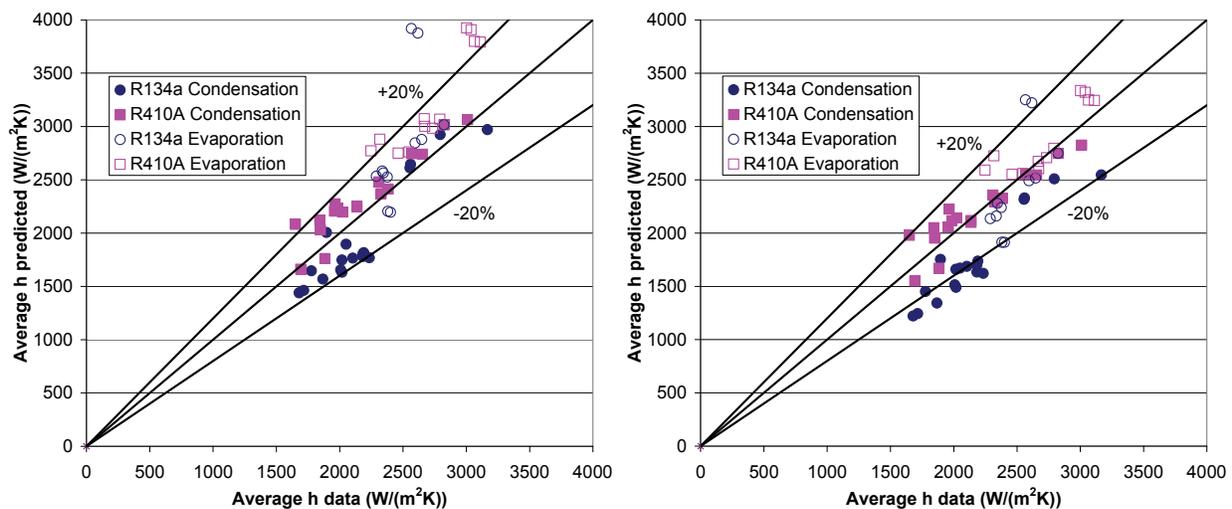
The heat transfer associated with an average heat transfer coefficient along the entire length of the microchannel with a quality change of $n \Delta x$ is given as:

$$\dot{m} h_{fg} n \Delta x = \bar{h} A_{total} \Delta T, \quad \text{where } A_{total} = A_1 + A_2 + \dots A_n \quad (15)$$

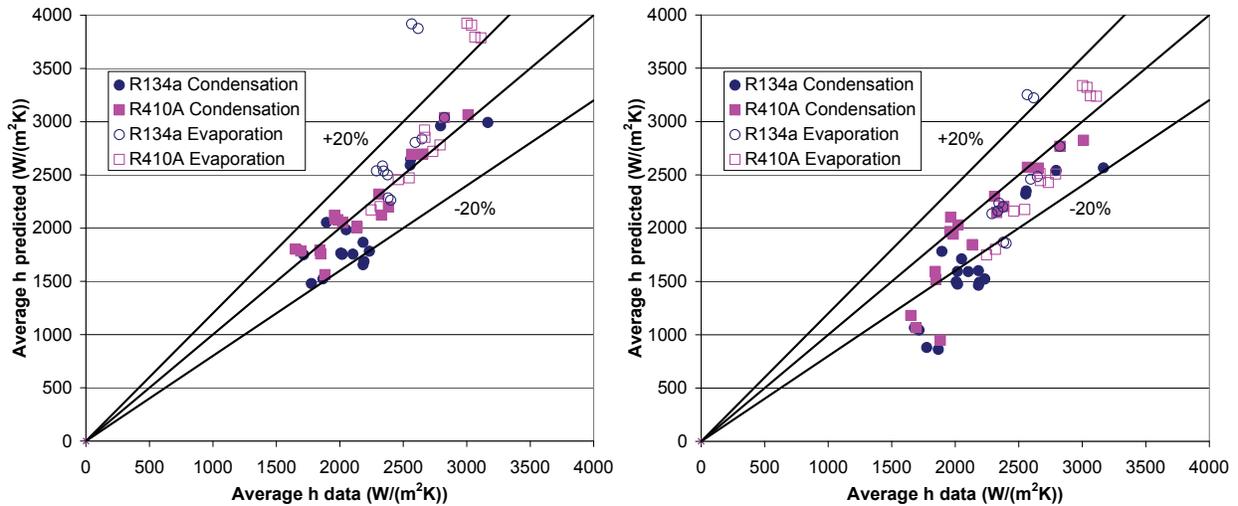
The difference in temperature between the microchannel wall and the refrigerant, ΔT , is relatively constant over the length of the microchannel for the Newell and Newell (2009) data. Consequently combining equations 14 and 15 above yields the following expression for the average heat transfer coefficient

$$\bar{h} = \frac{n}{1/h_1 + 1/h_2 + \dots 1/h_n} \quad (16)$$

Figures 3 through 7 depict the average heat transfer coefficient data found by Newell and Newell (2009) versus the predicted average heat transfer coefficient for the present probabilistic model using the modified Dobson and Chato (1998) annular flow model, the present probabilistic model using the modified Shah (1978) annular flow model, the modified Dobson and Chato (1998) model, the modified Shah (1978) model, and the modified Cavallini et al. (2003) model (Cavallini et al. (2003) model divided by 2, respectively.



Figures 3 (left). and 4 (right). Experimental vs. predicted average heat transfer coefficient for 4 port 1.58mm hydraulic diameter microchannels using the present probabilistic model utilizing the modified Dobson and Chato (1998) annular model and modified Shah (1978) annular model, respectively



Figures 5(left) and 6 (right). Experimental vs. predicted average heat transfer coefficient for 4 port 1.58mm hydraulic diameter microchannels utilizing the modified Dobson and Chato (1998) model and the modified Shah (1978) model, respectively

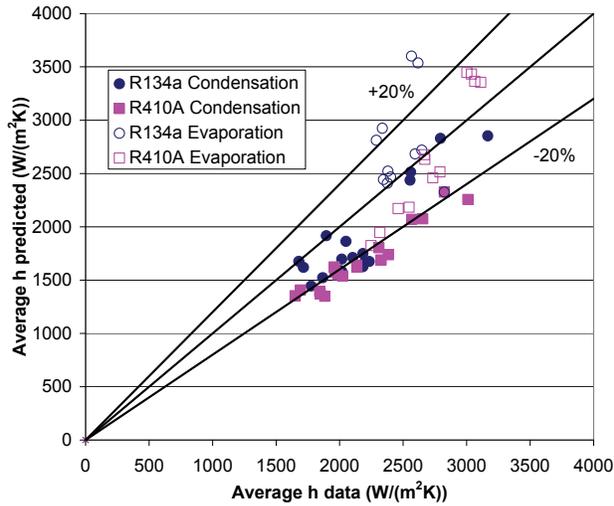


Figure 7. Experimental vs. predicted average heat transfer coefficient for 4 port 1.58mm hydraulic diameter microchannels utilizing the modified Cavallini et al. (2003) model

From figures 3 through 7 it can be seen that all five of the models perform well with the majority of the data points lying within 20% error bars and little variation between condensation and evaporation data. Table 1 contains the mean absolute deviation along with the percentage of predicted points lying with in $\pm 20\%$ error bars for all five models compared to Newell and Newell's (2009) condensation and evaporation data of R134a and R410A in 1.58mm four port microchannels.

Table 1. Statistical comparison of heat transfer models with Newell and Newell (2009) experimental data

Statistical comparison	Present model mod. Shah (1979) annular	Present model mod. Dobson & Chato (1998) annular	Mod. Dobson & Chato (1998)	Mod. Shah (1979)	Mod. Cavallini et al. (2003)
Mean absolute error (%)	11.3	13.0	10.4	16.1	15.9
% of data within 20% error bars	79.7	83.1	84.7	64.4	64.4

It is noteworthy that the large tube models of Dobson and Chato (1998), Cavallini et al. (2003), and Shah (1979) all predict heat transfer well in multiport microchannels when divided by 2. It is not clear why this is the case, but it could be postulated that it is related to the definition of the hydraulic diameter. Further investigation with a larger database consisting of a wide range of hydraulic diameters, geometries, and refrigerants, and flow conditions would be required to validate this postulate. Although the modified Dobson and Chato (1998) model performs very well, it is seen to significantly over predict the average heat transfer coefficient for high mass flux points where mainly annular flow exists. The modified Shah (1978) model more accurately predicts these high mass flux annular points, however it significantly under predicts low mass flux heat transfer coefficients with flows in mixed flow regimes. The present probabilistic model that utilizes both the modified Shah (1978) model along with flow regime appropriate models for the other flow regimes is seen to perform well for all the conditions investigated with all of the data points within or near the 20% error bars. The present model that utilizes the modified Dobson and Chato (1998) model also performs well, with the majority of the data points within 20% error bars, however it contains the same deviation as the Dobson and Chato (1998) for the high mass flux annular flow points where the models are nearly identical. The present probabilistic flow regime map modeling technique readily allows for replacement of heat transfer models for each flow regime as more accurate models are developed or identified.

6. CONCLUSIONS

In conclusion, probabilistic two-phase flow map modeling is found in the literature to accurately predict refrigerant heat transfer, pressure drop, and void fraction in large tubes on a consistent flow regime basis with ease of implementation. Furthermore, two-phase flow map modeling is found in the literature to accurately predict refrigerant pressure drop, and void fraction in multiport microchannels. In the present work a probabilistic two-phase flow map based heat transfer model is developed for multiport microchannels on the same flow regime basis as the probabilistic pressure drop and void fraction models found in the literature for multiport microchannels. The present probabilistic models developed utilizing the modified Dobson and Chato (1998) annular flow model and the Shah (1979) annular flow model are found to predict condensation and evaporation heat transfer for R134a and R410A in 1.58mm hydraulic diameter multiport microchannels well with average absolute errors of 11.3% and 13.0%, respectively. This probabilistic to phase flow regime map modeling method readily allows for the replacement of the models for each flow regime as more accurate models are developed or identified in the literature. The utility of the present model could be improved if the probabilistic two-phase flow regime maps for multiport microchannels are generalized based on physical parameters as Jassim (2006) has done for large tubes. It is noteworthy that large tube models such as the Dobson and Chato (1998), Shah (1979), and Cavallini et al (2003) are found to predict heat transfer in 1.58mm hydraulic diameter multiport microchannels well if they are divided by two, as they are found to have average absolute errors of 10.4, 16.1, and 15.9%, respectively, and is postulated to be related to the hydraulic diameter definition. Furthermore, there did not seem to be a significant difference in the predictive capabilities of the models between evaporation heat transfer and condensation heat transfer. A larger database with a wide range of refrigerants, hydraulic diameters, geometries, and flow conditions would be required to determine the link between large tube heat transfer and small tube heat transfer and to aid in the development of more accurate and universally applicable probabilistic two-phase flow regime map based heat transfer models.

NOMENCLATURE

A	area (m)	h_{fg}	latent heat of vaporization (kJ/kg)
C_p	specific heat (kJ/(kgK))	k	thermal conductivity (W/(mK))
D	hydraulic diameter (m)	\dot{m}	refrigerant mass flow rate (kg/s)
dP	pressure drop (kPa)	P	pressure (Pa)
dz	length (m)	P_{crit}	critical pressure (Pa)
F	time fraction (-)	Pr	Prandtl number (-)
G	mass flux (kg/(m ² s))	Re_l	superficial liquid Reynolds number (-)
g	gravitational acceleration (9.81 m/s ²)	T	temperature
h	heat transfer coefficient (W/(m ² K))	x	flow quality (-)
\bar{h}	average heat transfer coefficient (W/(m ² K))	X_{tt}	turbulent-turbulent Lockhart-Martinelli parameter (-)

Subscripts

<i>ann</i>	pertaining to the annular flow regime	<i>vap</i>	pertaining to the vapor flow regime
<i>h</i>	homogeneous	<i>w</i>	at the wall
<i>int</i>	pertaining to the intermittent flow regime	<i>l</i>	liquid
<i>int+liq</i>	pertaining to the intermittent and liquid flow regime	1	at location 1 along the microchannel
<i>liq</i>	pertaining to the liquid flow regime	2	at location 2 along the microchannel
<i>n</i>	at location n along the microchannel	Greek symbols	
<i>sat</i>	corresponding to saturation conditions	α	void fraction (-)
<i>strat</i>	pertaining to the stratified flow regime	μ	dynamic viscosity (kg/(ms))
<i>v</i>	vapor	ρ	density (kg/m ³)

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