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Two-phase flow model of refrigerants flowing through helically coiled capillary tubes

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ABSTRACT

This paper presents a numerical study of the flow characteristics of refrigerants flowing through adiabatic helically coiled capillary tubes. The theoretical model is based on conservation of mass, energy and momentum of the fluids in the capillary tube. The two-phase flow model developed was based on the homogeneous flow assumption. The viscosity model was also based on recommendations from the literature. The developed model can be considered as an effective tool for designing and optimizing capillary tubes working with newer alternative refrigerants. The model is validated by comparison with the experimental data of Kim et al. (2002) for R-22, R-407C and R-410A, and Zhou and Zhang (2006) for R-22. The results obtained from the present model show reasonable agreement with the experimental data. The proposed model can be used to design helical capillary tubes working with various refrigerants.

1. Introduction

In many small refrigeration systems, a capillary tube is used as the expansion device due to its low cost, low starting torque and low maintenance. Normally, it is used in refrigeration systems with cooling capacities less than 10 kW such as household refrigerators and air conditioners. The nominal size of the capillary tube typically ranges between 0.5 and 2.0 mm in diameter and 2–5 m in length.

Several decades ago, the flow characteristics of various refrigerants flowing through capillary tubes were studied both experimentally and theoretically. The most recent articles describing investigations of capillary tubes are summarized as follows:

Wong and Ooi [1] presented comparisons between the predicted results from homogeneous and separated flow models and experimental results from a number of researchers. The separated flow model which uses Miropolskiy's slip ratio combined with Lin's equation for frictional pressure gradient gave better predictions compared to the homogeneous flow model.

Bansal and Rupasinghe [2] investigated a homogeneous two-phase flow model which is called the "CAPIL model" to study the performance of adiabatic capillary tubes using R-134a as working fluid. The REFPROP program was used to calculate the refrigerant properties.

Sami and Tribes [3] reported a numerical model for predicting capillary tube performance for some azeotropic and zeotropic binary mixtures as well as pure HFC refrigerants.

In 1999, Melo et al. [4] studied experimentally the effect of capillary length, diameter, refrigerant subcooling, condensing pressure and type of refrigerant (CFC-12, HFC-134a and HC-600a) on the mass flow rates through capillary tubes. In addition, they proposed correlations to predict the mass flow rate of various refrigerants. The proposed correlations were found to be in good agreement with those obtained from the measured data and other studies in the open literature.

Wongwises et al. [5] developed a two-phase flow model based on homogeneous flow to study the flow characteristics of many pairs of refrigerants flowing through adiabatic capillary tubes. The Colebrook equation was used to determine the two-phase friction factor. It was found that the traditional refrigerants consistently gave lower pressure drops for both single-phase and two-phase regions which resulted from longer capillary tube lengths.

Wongwises et al. [6] presented a two-phase separated flow model to describe the refrigerant flow characteristics in the capillary tubes of refrigeration systems. The agreement between experimental data and simulation results obtained for flows of R-12, R-22 and R-134a through capillary tubes indicated that the separated flow model with appropriate correlations of the frictional pressure gradient and slip ratio can be used to predict the two-phase flow behaviour of refrigerant flowing through capillary tubes.

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Nomenclature

A	cross sectional area of capillary tube (m^2)
d_i	capillary tube internal diameter (m)
D_C	coil diameter (m)
De	Dean Number, $De = \mathrm{Re}\sqrt{(d_i/D_C)}$
e/d_i	relative roughness
$\triangle T_{sub}$	degree of subcooling (°C)
f	friction factor
g	gravitational acceleration (m/s^2)
G	mass flow rate per unit area ($kg/s\text{-}m^2$)
h	specific enthalpy (J/kg)
H_{loss}	head loss (m)
He	Helical number, $He = \mathrm{Re}[(d_i/D_C)/\{1+(p/\pi D_C)^2\}]^{1/2}$
k	entrance loss coefficient
L	length (m)
m	mass flow rate (kg/s)
P	pressure (Pa)
Re	Reynolds number
s	specific entropy (J/kg-K)
T	temperature (°C)
V	velocity (m/s)
x	quality

Greek letters

τ_w	shear stress at wall (N/m^2)
v	specific volume (m^3/kg)
μ	dynamic viscosity (kg/m-s)
ρ	density (kg/m^3)

Subscripts

cond, evap	condenser and evaporator, respectively
f, g	liquid phase and gas phase, respectively
h	homogeneous flow
i	capillary inlet condition
sp, tp	single-phase and two-phase, respective

Wongwises and Pirompak [7] studied the flow characteristics of alternative refrigerants in adiabatic capillary tubes by using an adiabatic capillary tube model. Moreover, they also proposed selection charts for selecting the size of capillary tube based on the flow rate and flow condition. In addition to tube size, the charts are very useful for determining the mass flow rate directly.

In 2001, Liang and Wong [8] introduced the homogeneous flow model based on a drift flux model to predict the flow characteristics of R-134a flowing through an adiabatic capillary tube. This model was validated by comparisons between numerical and experimental results by Li et al. [9] and Mikol et al. [10] using R-12 as working fluid.

Jung et al. [11] proposed a model to calculate the size of capillary tubes for various refrigerants of R-22, R-134a, R-407C and R-410A. In their model, the Stocker model was modified and took several factors into account such as area contraction, different equations for viscosity and friction factors, and mixing effects. Numerical results indicated that using the McAdam model for the viscosity equation gave a better calculation than the Dukler model. Finally, semi-empirical correlations for predicting mass flow rate, condensing temperature and subcooling of refrigerant were proposed.

Sinpiboon and Wongwises [12] presented a model to study the flow characteristics in non-adiabatic capillary tubes. In this study, the mathematical model was categorized into three different cases, depending on the position of the heat exchange process. The first case is used when the heat exchange process starts in the single-phase flow region. The second case is determined when the heat exchange process starts at the end of the single-phase flow region. Finally, the last case is considered when the heat exchange process takes place in the two-phase flow region. A set of differential equations were solved by an explicit method in a finite-difference scheme.

Fiorelli et al. [13] performed an experimental study to evaluate the performance of R-22 and its alternatives, such as R-407C and R-410A flowing through adiabatic capillary tubes. The results showed that the performance of capillary tubes for R-410A and R-407C were similar under given conditions. Moreover, they also determined the effect of geometry on the behaviour of capillary tubes. Finally, differences in the flow behaviour of R-410A and R-407C were evaluated.

Bansal and Wang [14] presented a homogeneous and metastable simulation model. The first law of thermodynamics, some fluid mechanics and empirical relations were incorporated into this model. Compared with published experimental data, this model shows good agreement within ±7% for R-22, R-134a and R-600. Moreover, a new numerical analysis for simulating the choked flow of refrigerant under adiabatic conditions was proposed.

Choi et al. [15] presented a generalized mass flow rate correlation based on their experimental data for R-22, R-290 and R-407C. Dimensionless parameters were derived from the Buckingham π theorem by considering the effect of refrigerant properties, capillary tube geometry and inlet conditions.

Similarly, Yang and Wang [16] proposed a generalized mass flow rate correlation based on extensive data for R-12, R-22, R-134a, R-290, R-600a, R-410A, R-407C, and R-404A. The simulation conditions used in their study are as follows: 1) the inner diameter ranges between 0.5 and 2 mm, 2) the tube length ranges between 0.5 and 5 m, 3) the condensing temperature ranges between 20 °C and 60 °C, 4) the subcooling temperature is between 0 °C and 20 °C and 5) Inlet quality varies approximately in the range from 0 to 0.3. Their results showed that the predicted values agree well with the experimental data in the open literature for R-12, R-22, R-134a, R-290, R-407C, R-410A and R-404A refrigerants. Moreover, the model presented gave average and standard deviations of 0.83% and 9.02% compared with the ASHRAE [17] and Choi et al. [15] equations, respectively.

Seixlack and Barbazelli [18] presented a numerical model to predict refrigerant flow along non-adiabatic capillary tubes using a two-fluid model. The flow along the straight and horizontal capillary tube is divided into two regions: single-phase and two-phase flow regions. In comparisons between numerical results and experimental data, the results gave good predictions of the refrigerant mass flow rate. In addition, comparisons with a homogeneous model were also made.

It can be noted that the theoretical and experimental investigations found in the literature and described above have focused on the study of the flow characteristics in horizontal straight capillary tubes; the flow characteristics in coiled capillary tubes has received comparatively little attention in the literature. The most productive studies of coiled capillary tubes have been continuously carried out by the following researchers.

Ali [19] proposed pressure drop correlations which were developed in terms of fluid properties (ρ and μ), flow rate (V) and tube geometry (d_i, D_C, p and L). It should be noted that in most of the