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Research Paper

A modified approach for numerical simulation of capillary tube-suction line heat exchangers

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HIGHLIGHTS

- A modified approach for capillary tube-suction line heat exchangers is presented.
- A two-step predictor–corrector model is proposed.
- Numerical results are confronted against experimental data from the literature.
- A parametric analysis is performed for two geometrical parameters.
- Absence of convergence problems in the numerical code is observed.

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ABSTRACT

This paper presents a modified approach for numerical simulations of capillary tube-suction line heat exchangers (CTSLHE). This approach is based on an approximate solution methodology previously presented. This methodology models the expansion process separately from the heat transfer process, giving rise to two sub-models, namely hydrodynamic and thermal sub-models. In the present paper, a two-step predictor–corrector model is proposed, one for the refrigerant mass flux and another for the refrigerant enthalpy path. Simulation results of mass flow rate and suction line outlet temperature were compared with experimental data, empirical correlations and well ranked distributed models. The simulation results showed good agreement with experimental data, with almost all mass flow rate results were within ±20% error bands and most suction line outlet temperature results within ±5 °C. Reasonable agreement was observed when the predicted results of mass flow rate, critical pressure and suction line outlet temperature were compared against the results from well ranked literature numerical models. Both, the influence of the capillary tube inner diameter and the suction line inner diameter over performance parameters were also investigated. No convergence problems were observed during the simulations.

1. Introduction

In small refrigeration and air conditioning systems, the expansion device usually is a capillary tube, which is a long tube (length from 1 to 6 m) with small diameter (0.5–2.0 mm). According to Stoecker and Jones [1], its functions are to reduce the refrigerant pressure before entering the evaporator and regulate the refrigerant mass flow rate. These devices are simple, low cost and allow the equalization of the high and low pressures during the off-cycle, with reduction of the compressor starting torque, as mentioned by Peixoto and Bullard [2]. As simple devices from the structural point of view, the phenomena involved during the refrigerant flow inside capillary tubes are quite complex: two-phase compressible flow associated with heat transfer, non-equilibrium effects, among others.

In household refrigerators, a capillary tube-suction line heat exchanger (CTSLHE) is formed by putting a portion of the capillary tube in thermal contact with the suction line. As mentioned by Dirik et al. [3], this counter flow heat exchanger provides heat exchange between superheated refrigerant vapor passing through suction line to the compressor and relatively warm refrigerant

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Nomenclature

A	heat transfer area (m^2)
c_p	specific heat capacity at constant pressure (J/kg K)
dz	differential element of the tube (m)
dP	differential pressure (Pa)
D	diameter (m)
G	mass velocity (kg/m^2 s)
$\dot{m}$	mass flow rate (kg/s)
v	specific volume (kg/m^3)
P	pressure (Pa)
h	specific enthalpy (J/kg)
f	friction factor
L	length (m)
NTU	number of transfer units
$\dot{Q}$	heat transfer rate (W)
U	overall heat transfer coefficient (W/m^2 K)
x	parameter of two-phase flow region
Greek symbols	
ε	heat exchanger thermal effectiveness
μ	dynamics viscosity (Pa s)
ϕ	parameter of two-phase flow region
φ	under-relaxation factor
Dimensionless groups	
Re	Reynolds number
Pr	Prandtl number
Subscripts	
a	anulli
c	capillary
$calc$	calculated
$crit$	critical
$ctshle$	capillary tube-suction line heat exchanger
f	flash point
i	inlet
int	internal
e	exit
$evap$	evaporation
ext	external
liq	liquid
s	suction line
tp	two-phase
v	vapor

passing through capillary tube. Structurally, these heat exchangers are presented in two possible configurations, namely concentric and lateral, as represented in Fig. 1, Hermes et al. [4].

Due to the increasing number of household refrigerators in operation nowadays, several and extensive studies have been performed to investigate adiabatic and non-adiabatic capillary tubes thermal–hydraulic behavior and performance. In Qiao et al. [5], a systematic and complete review about the modeling approaches for each component of vapor compression systems is presented. In this study, for capillary tubes simulations, empirical/semi-empirical correlations and numerical models (approximate solutions, distributed models) have been constantly applied as alternative or complementary approaches to experimental evaluation. Melo et al. [6] developed an experimental analysis of the performance of concentric capillary tubes operating with HFC-134a as working fluid. The experimental analysis was carried over a wide range of operating boundary conditions and heat exchanger geometries typical of household refrigerators and freezers. The authors reported that the operating condition having the greatest effect on the mass flow rate is the inlet pressure. In addition, the inner diameter of the capillary tube is the geometrical variable with the greatest impact on the mass flow rate.

Correlation-based models calculate the mass flow rate based on inlet and outlet conditions, obtained from experimental data and, generally, present good accuracy only within the range of validity of the regression data [7–9]. Wolf and Pate [7] developed an empirical correlation for mass flow rate predictions for HFC-134a, HCFC-22, R-410A and HC-600a. This empirical correlation was developed only for sub-cooled inlet condition. Melo et al. [8] developed empirical correlations for mass flow rate and the suction line outlet temperature for concentric non-adiabatic capillary tubes operating with HC-600a, under different conditions and geometrical parameters. Park et al. [9] also developed an empirical correlation model for mass flow rate prediction of capillary tubes operating with HCFC-22 and R-410A.

The study carried by Sarker and Jeong [10] proposed the development of empirical correlations for non-adiabatic capillary tubes based on mechanistic numerical model. The aim of the authors was to include a wider range of geometrical parameters and operating conditions into the developed correlations. Results were compared with experimental data and excellent agreement was observed.

Approximate solutions for adiabatic capillary tubes are based on governing equations, where the assumption of isenthalpic flow is applied [11–13]. Yilmaz and Unal [11] developed an algebraic model for mass flow rate predictions for different pure refrigerants flowing inside adiabatic capillary tubes. The refrigerant flow was considered isenthalpic, which yielded to an approximate implicit solution for mass flow rate, due to its dependency on the friction factor. Later, Zhang and Ding [12] modified this approximate solution to include the possibility of simulating refrigerant mixtures and presented a two-step predictor–corrector algorithm to compute the mass flow rate. The semi-empirical model developed by Hermes et al. [13] for simulation of adiabatic capillary tubes revealed a novel approach, since the authors presented a physically consistent formulation which allowed the development of an explicit expression for computation of the mass flow rate and suction line outlet temperature.

Detailed numerical models, also known as distributed models, are derived from governing equations of mass, momentum and energy, where homogeneous model [14–16] or two-fluid model can be applied for two-phase flow modeling, such as [17]. Hermes et al. [14] developed a numerical model based on the mass, momentum and energy conservation equations using finite volume method. This numerical model allowed simulation under steady state and transient regimes. García-Valladares et al. [15] developed a distributed numerical model which took into account the metastable region. Sinpiboon and Wongwises [16] developed a mathematical model for non-adiabatic capillary tubes based on a distributed approach. The model allowed the three cases of simulations, depending on the position of the starting point of the heat exchange process: in the single-phase flow region, in the end of single-phase flow region or in the two-phase flow region. Experimental data were used to validate the numerical model and well agreement was observed. García-Valladares [18,19] carried numerical simulations with experimental validation of non-adiabatic capillary tubes considering a metastable region in the numerical model. The developed model allowed steady and transient