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Modeling of non-adiabatic capillary tube flows: A simplified approach and comprehensive experimental validation

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ABSTRACT

A simplified computational model for simulating refrigerant flow through capillary tubes is proposed and validated using a dataset composed of more than 1400 experimental data points, including adiabatic flows of refrigerants CFC-12, HCFC-22, HFC-134a, HC-600a, R-404A, R-407C and R-507A, and non-adiabatic flows of refrigerants HFC-134a and HC-600a, in both concentric and lateral capillary tube–suction line heat exchanger configurations. The model is based on the mass, energy and momentum conservation equations written according to their one-dimensional differential formulation. Some simplifications were added to the model in order to improve both numerical stability and computational performance. It was found that the model predicts 91.5% of the measured refrigerant mass flow rate for adiabatic and 79.3% for non-adiabatic flows within an error band of ±10%. Also, the model solves non-adiabatic flows as fast as adiabatic ones.

Modélisation de l'écoulement non adiabatique à l'intérieur des capillaires : approche simplifiée et validation expérimentale complète

Mots clés : Système frigorifique ; Capillaire ; Modelisation mathematique ; Non-adiabatique écoulement ; Valider expérimental

1. Introduction

In refrigeration systems, a capillary tube is simply a small bore tube connecting the condenser to the evaporator. Liquid refrigerant flows into one end and expands until reaching the evaporating pressure. In doing so it maintains the refrigerant at the desired mass flow rate. A capillary tube appears to be quite simple, but the refrigerant flow inside this

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Nomenclature

Roman

A	cross-sectional area [m^2]
c_p	specific heat at constant pressure [J kg^{-1} K^{-1}]
d	capillary tube inner diameter [m]
d_s	suction line inner diameter [m]
D	capillary tube outer diameter [m]
e	rugosity [m]
G	mass flux [kg s^{-1} m^{-2}]
h	specific enthalpy [J kg^{-1}]
$I(p,h)$	integrand of Eq. (9)
k	thermal conductivity [W m^{-1} K^{-1}]
L	length [m]
p	pressure [Pa]
P	tube inner perimeter [m]
q	heat flux [W m^{-2}]
t_c	capillary tube refrigerant temperature [K]
t_s	suction line refrigerant temperature [K]
U	overall heat transfer coefficient [W m^{-2} K^{-1}]
v	specific volume [m^3 kg^{-1}]
w	mass flow rate [kg h^{-1}]
z	axial coordinate [m]

Greek

α	heat transfer coefficient on tube walls [W m^{-2} K^{-1}]
η	viscosity [Pa s]
κ	coefficient of Eq. (13) [$=(1-v/v_f)/(1-p_f/p)$]
τ	shear stress on tube walls [Pa]

Dimensionless parameters

ε	temperature effectiveness
f	friction factor
NTU	number of transfer units
x	vapor quality

Subscripts

c	capillary tube
en	entrance region
ex	exit region
hx	heat exchanger region
i	inlet
l	liquid
o	outlet
s	suction line
sat	saturation
tp	two-phase
v	vapor

component is rather complex. The flow offers several challenges for a phenomenological description: turbulence, heat transfer, phase-change, compressibility and non-equilibrium effects all occur in the flow through capillary tubes. The expansion process is driven by two major effects: shear stress between the fluid flow and the tube walls, and flow acceleration when the liquid turns into vapor. The refrigerant pressure drop, as it passes through the capillary tube, is accompanied by a reduction in temperature brought about by the transfer of enthalpy from the remaining liquid to provide the enthalpy of evaporation of the flash vapor. At any stage during the expansion process the vapor formed at that point has performed its function and has no other function until it is recompressed by the compressor.

The application of capillary tubes in vapor-compression refrigeration systems started in the 1920s, when the first hermetic domestic refrigerators were developed (Nagengast, 1996). Most manufacturers, however, initially resisted the use of capillary tubes as expansion devices. The main reason for this was that SO_2, a popular working fluid at the time, requires very small capillary diameters and thus the tubes may eventually become clogged (Swart, 1946). Other issues included the low cooling capacity (the capillary tube-suction line heat exchanger had not yet been invented), and the restrictions imposed by a patent, which postponed the dissemination of capillary tubes until the early 1930s, when the Freon working fluids were introduced (Swart, 1946). Since then, the application of capillary tubes has increased steadily, being nowadays the expansion devices most widely used in small capacity vapor-compression refrigeration systems, such as household refrigerators and freezers, dehumidifiers, and room air conditioners.

In many applications the capillary tube forms a counterflow heat exchanger with the suction line in order to increase evaporator capacity and to prevent slugging of the compressor and sweating of the suction line. Two types of the so-called capillary tube-suction line heat exchanger (CT-SL HX) are usually found: lateral and concentric. In the lateral configuration the capillary tube is brazed to the suction line, whereas it passes inside the suction line in the concentric arrangement. Fig. 1 shows a schematic diagram of both a concentric (Fig. 1a) and a lateral (Fig. 1b) CT-SL HX, where three distinct flow regions can be observed: entrance region (L_{en}), heat exchanger region (L_{hx}), and exit region (L_{ex}). Both L_{en} and L_{ex} are usually assumed to be adiabatic.

As a matter of fact, the capillary tube flow is considerably affected by the heat transfer to the suction line. Fig. 2 compares the flow patterns described by adiabatic and non-adiabatic flows of the refrigerant HFC-134a in the same capillary tube. The adiabatic flow follows a path that is close to an isenthalpic line, whilst the non-adiabatic flow is projected toward the line of saturated liquid, increasing the amount of liquid in the two-phase mixture and decreasing the vapor quality at the evaporator inlet. As a consequence, the refrigerating effect in the evaporator is increased.

Due to the importance of capillary tubes to the refrigeration industry, their thermo-hydrodynamic behavior has been extensively investigated at the POLO Research Laboratories for Emerging Technologies for Cooling and Thermophysics of the Federal University of Santa Catarina for almost two decades. Here, studies have been carried out both theoretically (Melo et al., 1992; Boabaid Neto, 1994; Seixlack et al., 1996; Mezavila and Melo, 1996; Hermes et al., 2000; Negrão