Available online at www.sciencedirect.com
ScienceDirect
Energy Conversion and Management 48 (2007) 1338–1348
ENERGY CONVERSION & MANAGEMENT
www.elsevier.com/locate/enconman

Theoretical investigation of adiabatic capillary tubes working with propane/*n*-butane/iso-butane blends

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Received 1 January 2006; accepted 15 July 2006
Available online 5 February 2007

Abstract

In this paper, a theoretical model is developed to predict the refrigerant flow characteristics in adiabatic capillary tubes using propane/*n*-butane/iso-butane mixtures as working fluids in a domestic refrigerator. This model is based on the mass, energy and momentum conservation equations for a homogeneous refrigerant flow under different inlet conditions, such as subcooled, saturated and two phase flow. The effects of the inlet pressure (8–16 bar), inlet vapor quality (0.001–15%), inlet subcooling degree (1–15 °C), mass flow rate (1–5 kg/h), propane mass fraction (0.5–0.7), capillary tube inner diameter (0.6–1.0 mm) and the tube surface roughness on the capillary tube length are predicted.

The results showed that the present model predicts data that are very close to the available experimental data in the literature with an average error of 2.65%. The pressure of the hydrocarbon mixture (HCM) decreases, while its vapor quality, specific volume and Mach number increase along the capillary tube. Also, the results indicated that the capillary tube length is largely dependent on the capillary tube diameter. Other parameters such as mass flow rate, inlet pressure, subcooling degree (or quality) and relative roughness influence the capillary tube length in that order. The capillary tube length as a function of the significant parameters is presented in equation form. Also, capillary tube selection charts either to predict the mass flow rates of propane/*n*-butane/iso-butane mixtures through adiabatic capillary tubes or to select the capillary tube size according to the required applications are developed. The comparison between R12, R134a and the hydrocarbon mixture (HCM) of propane/*n*-butane/iso-butane indicated that for a given mass flow rate, the pressure drop per unit length is about 4.13, 5.0 and 12.0 bar/m for R12, R134a and HCM, respectively. The ratios of the average mass flow rate of the HCM with a propane mass fraction of 0.6 to those of R12 and R134a are about 0.62 and 0.67, respectively. The average capillary tube length for the HCM with a propane mass fraction of 0.6 is longer than those of R134a and R12 by about 30% and 48%, respectively.

Keywords: Capillary tube sizing; Flow characteristics; Alternative refrigerants; Hydrocarbon mixtures

1. Introduction

A capillary tube is made of a small internal diameter copper tube of a varying length depending upon the application. It is installed in the liquid line between the condnser and evaporator of a vapor compression system to reduce the condenser pressure to the evaporator pressure. It has several advantages, such as simple construction, no moving parts (which wear or stick, i.e. no maintenance is required), no receiver is necessary, low starting torque motor (low cost motor) and is less expensive. When the refrigerant expands from the condenser pressure to the evaporator pressure adiabatically, i.e. the tube is fully insulated, the capillary tube is called an adiabatic tube. In some refrigeration systems, the capillary tube is soldered to the suction line, and the combination is called a capillary tube-suction line heat exchanger. This type of capillary tube is known as a non-adiabatic capillary tube.

The sizing of capillary tubes commonly used as expansion devices in household refrigerators and freezers

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0196-8904/$ - see front matter © 2006 Published by Elsevier Ltd.
doi:10.1016/j.enconman.2006.07.003

Nomenclature

A	cross section area (m^2)
c	fluid velocity (m/s)
C	contraction coefficient (–)
d	capillary tube diameter (m)
f	friction factor (–)
G	refrigerant mass flux (kg/m^2 s)
h	specific enthalpy (kJ/kg)
HCM	hydrocarbon mixture
L	capillary tube length (m)
m	mass flow rate (kg/s)
M	Mach number (–)
p	pressure (MPa)
PMF	propane mass fraction (–)
Re	Reynolds number
s	specific entropy (kJ/kg K)
T	temperature (K)
v	specific volume (m^3/kg)
x	dryness fraction (–)

Greek letters

ε	roughness (mm)
φ	flow factor (–)
μ	dynamic viscosity (Pa s)
ρ	density (kg/m^3)
Δ	difference (–)

Subscripts

$1-n$	refers to Fig. 1
c	contraction
con	condensation
ch	choke
ds	down stream
in	inlet
m	mean
r	refrigerant
sat	saturated
sl	saturated liquid
sp	single phase
std	standard
sv	saturated vapor
sub	subcooling
tot	total
tp	two phase
us	up stream

working with R12 alternatives has been investigated by many researchers using either theoretical, empirical [1,2] or experimental approaches [3]. An experimental approach is very expensive, and the empirical approach depends upon the available experimental data over a wide range of operating conditions. In the present work, a theoretical approach is adopted to investigate the flow characteristics through adiabatic capillary tubes and to determine their sizing.

Bittle and Pate [4] presented a theoretical model to predict the flow rate of R134a through an adiabatic capillary tube. The model was validated with data of the previous studies over a range of operating conditions. It was found that the model agrees reasonably well with the experimental data for R134a. A numerical model for predicting capillary tube performance using pure refrigerants or binary mixtures was proposed by Sami and Tribes [5]. The model was established for a homogeneous refrigerant flow under saturated, subcooled and two phase conditions. Numerical results of the proposed model and experimental data showed fair agreement. Numerical models have been developed to size adiabatic and non-adiabatic capillary tubes by Sami and Maltais [6] for R22 substitutes (R407C, R410A and R410B) and Sami et al. [7] for R502 alternatives, such as azeotropic mixtures (R507, R404A) and a quaternary mixture (R32/R125/R134a/R143a). Their results indicated that the system using either R407C or R507C would experience a smaller pressure drop across the capillary tube when compared to the other alternatives.

An adiabatic capillary tube model was developed by Wongwises et al. [8] and Wongwises and Pirompak [9] to study the flow characteristics in capillary tubes. Numerical results showed that the traditional refrigerants gave lower pressure drops (for both single phase and two phase flows) than the environmentally acceptable alternative refrigerants, which resulted in longer tube lengths for them. The model was validated via comparing its simulation results with published experimental data for R12 and R134a. Fatouh and El-Kafafy [10] have reported that a hydrocarbon mixture (HCM) of propane/iso-butane/*n*-butane with the propane mass fraction of about 0.6 is a promising drop in refrigerant for R134a in domestic refrigerators. This mixture offers many advantages, such as its local availability and low price [11], compared with other refrigerants.

Despite the many investigations reported to size capillary tubes working with either ozone depleting substances, such as R12, R22 and R502, or ozone safe refrigerants, such as R134a, the sizing of capillary tubes working with HCMs needs to be investigated. Thus, the main objectives of the present work concern sizing the adiabatic capillary tubes that use the propane/iso-butane/*n*-butane mixture and predicting the flow characteristics through them while evaluating the effects of operating parameters on their sizes. In order to achieve these objectives, a computer program based on the mass, energy and momentum conservation equations has been developed. The input data are thermophysical properties, inlet conditions (subcooled, saturated or two phase), refrigerant mass flow rate and capillary tube