Experimental Thermal and Fluid Science 35 (2011) 706–716

Contents lists available at ScienceDirect

Experimental Thermal and Fluid Science

journal homepage: www.elsevier.com/locate/etfs

Condensation heat transfer characteristics of R-22, R-134a and R-410A in a single circular microtube

Hoo-Kyu Oh, Chang-Hyo Son *

Department of Refrigeration and Air-Conditioning Engineering, College of Engineering, Pukyong National University, San 100, Yongdang-dong, Nam-gu, Pusan 608-739, South Korea

A R T I C L E I N F O

Article history:
Received 20 May 2010
Received in revised form 4 September 2010
Accepted 12 January 2011
Available online 18 January 2011

Keywords:
Condensation heat transfer
Mixture refrigerant (R-410A)
Single circular microtube
Single-phase heat transfer

A B S T R A C T

The condensation heat transfer coefficients of R-22, R-134a and R-410A in a single circular microtube were investigated experimentally. The experiments are conducted without oil in the refrigerant loop. The test section is a smooth, horizontal copper tube of 1.77 mm inner diameter. The experiments were conducted at mass flux of 450–1050 kg/m^2 s, saturation temperature of 40 °C. The test results showed that in case of single-phase flow, the single-phase Nusselt Number measured by experimental data was higher than that calculated by Gnielinski correlation. In case of two-phase flow, the condensation heat transfer coefficient of R-410A was higher than that of R-22 and R-134a at the given mass flux. The condensation heat transfer coefficient of R-22 showed almost a similar value to that of R-134a. Most of the existing correlations which were proposed in the large diameter tube failed to predict condensing heat transfer. And also, recently proposed correlation in the single circular microtube is considered not adequate for small diameter tube. Therefore, it is necessary to develop accurate and reliable correlation to predict heat transfer characteristics in the single circular microtube.

1. Introduction

Small diameter tubes are widely being used to achieve high heat transfer rates with compact heat exchangers. Condensation inside small diameter tubes finds applications in heat pipes and compact heat exchangers for electronic equipment in residential air conditioning and refrigeration applications. Small diameter tubes are viewed as appropriate options for reducing inventories of hazardous fluids and also reducing greenhouse gas emissions by improving component and system energy efficiency. The compactness of small diameter tube elements in air-cooled condensers improves the system efficiency through out the reduction of air-side pressure drops and the increase of the heat transfer coefficients. The adoption of small diameter tubes also promotes the reduction of the refrigerant charge, which is favourable to the use of toxic or flammable refrigerants. Furthermore, small diameter tubes can be well used with high pressure fluids, for instance with carbon dioxide in transcritical cycle equipment, since these elements are able to withstand high system pressures [1].

No specific criteria have been developed for the definition of small diameter tubes during forced convective condensation. In the present paper, the classification taken from Kandlikar & Grande [2] is adopted and therefore small diameter tubes can be single circular microtubes or multiport extruded aluminum microchannels having inner hydraulic diameters in the range 0.2–3 mm. (3 mm is used as the threshold diameter to distinct small and large diameter tubes here [3].)

Previous reviews on condensation heat transfer in small diameter tubes have been conducted by many researchers [4–18]. Of these studies, Yang and Webb [4], Webb and Ermis [5], Vardhan and Dunn [6] measured the heat transfer coefficient during condensation of refrigerants inside microchannels, both plain and microfinned. Also recently Wei-Wen et al. [7], Garimella [8], Koyama et al. [9], Baird et al. [10], Cavallini et al. [11], Bandhauer et al. [12] and Baummer et al. [13] collected experimental heat transfer coefficient data for condensation in microchannels. Yan & Lin [14], Zhang & Webb [15], Kim et al. [16,17] carried out experimental condensation heat transfer coefficient data in microtubes.

As shown literature reviews mentioned earlier, there are many studies related to the heat transfer in microchannels, but only few studies related to the heat transfer in microtubes. More experimental studies are necessary to develop heat transfer database and correlations in microtubes because some experimental data are much different from others.

Accordingly, the purpose of this study is to present new experimental data and search the suitable existing predictions which describe the present data and also analyze the experimental data to find condensation characteristics in a single circular microtube. In this study, the condensation heat transfer characteristics of R-22, R-134a and R-410A flowing in a single circular microtube of 1.77 mm were investigated experimentally. The present experimental results are compared with previous correlations proposed for circular microtubes.

* Corresponding author.
E-mail address: sonch@pknu.ac.kr (C.-H. Son).

0894-1777/$ - see front matter Crown Copyright © 2011 Published by Elsevier Inc. All rights reserved.
doi:10.1016/j.expthermflusci.2011.01.005

Nomenclature

A	area (m^2)
C	frictional flow constant
c_p	specific heat at constant pressure (kJ/(kg K))
d	diameter (m)
D_h	hydraulic diameter (m)
F	Adams et al. correlation
G	mass velocity (kg/(m^2 s))
g	standard gravitational acceleration (m/s^2)
h	heat transfer coefficient (kW/(m^2 K))
i	enthalpy (kJ/kg)
i_{fg}	latent heat (kJ/kg)
J_g	$xG/[gd_i(\rho_l - \rho_g)]^{0.5}$
J_g^T	transition dimensionless gas velocity (Eq. (19))
L	total condensing length (m)
m	mass flowrate (kg/s)
N	number of data
P	pressure (kPa)
q	heat flux (kW/m^2)
Q	heat capacity (kW)
T	temperature (K)
U_{SG}	gas superficial velocity (m/s)
U_{SL}	liquid superficial velocity (m/s)
x	vapor quality
z	length of test section (m)

Dimensionless groups

f	friction factor
Nu	Nusselt number
Pr	Prandtl number
Re	Reynold number
We	Weber number
X_{tt}	Martinelli parameter
Φ	two-phase friction multiplier

Greek symbols

δ	film thickness (m)
ρ	density (kg/m^3)
μ	dynamic viscosity (Pas)
κ	thermal conductivity (W/mK)
σ	deviation, surface tension (N/m)

Subscripts

A	ΔT-independent flow regime
abs	mean
avg	average
c	condensation
cal	calculated
cs	source water
D	ΔT-dependent flow regime
exp	experimental
g	gas phase
Gn	Gnielinski
i	inner, inside
l	liquid phase
lo	liquid phase with total flow
L	local
m	average
in	inlet
o	outer, outside
out	outlet
pre	pre-heater
r	refrigerant
s	saturation
sub	subsection
w	tube wall
wi	inside tube wall

2. Experimental apparatus and procedures

2.1. Test facility

The experimental apparatus as schematically shown in Fig. 1 is designed to investigate the heat transfer coefficient and pressure drop of R-22, R-134a and R-410A in a single microtube. The main loops of the system are a refrigerant loop and a water loop. Detailed descriptions of the two loops of the test facility are provided below. A schematic diagram of the test rig and Pyrex sight glass tube is shown in Fig. 2.

As seen in Fig. 1, the refrigerant loop contains a magnetic gear pump, a mass flow meter, a pre-heater, a test section, a subcooler and a receiver, etc. The refrigerant then passes in series through the magnetic gear pump, the refrigerant flow meter, the pre-heater, and enters the test section. The subcooled refrigerant is charged in the receiver where it is further cooled to increases its density. Liquid refrigerant is pumped by a magnetic gear pump which can be regulated by means of an inverter, and then flows through a flow meter. The refrigerant flow rate is measured by the mass flow meter of Oval corporation.

The inlet quality and temperature before entering the test section are controlled by the pre-heater. The DC power supply is used to apply the imposed heat flux to the pre-heater, which can be controlled by adjusting the supply voltage and current. Electrically insulated heating wires are wrapped around the surface of copper tubes in the pre-heater. The pre-heater is insulated with glass fibers and rubber. The amount of heat loss from the pre-heater is calibrated through pretests with water; this is the correlated to the voltage input. In passing through the test section, the two-phase refrigerant is completely condensed and subcooled by cooling water. Leaving the test section, the refrigerant vapor then condenses in a subcooler and is later collected in a receiver; it eventually returns to the refrigerant pump to complete the cycle. The subcooler is a counterflow heat exchanger with refrigerant flowing in the inner tube and the water flowing in the annulus. The pressure of the cycle is controlled by the charged amount of refrigerant.

The water loop is composed of a centrifugal pump, an in-line electric heater and a heat exchanger. The cooling water is pumped to the circular-tube annulus, where it absorbs the heat of the condensing refrigerant. The water flow rate was set by the pump and the by-pass. The mass flow rate of cooling water is also measured by a turbine type flow meter of Schlumberger corporation. The inlet temperature of the cooling water at the test section is controlled by both the electric heater and the refrigeration unit.

2.2. Test section

The complete schematic diagram of the test section is shown in Fig. 2. The dimensions of the test section are also listed in Table 1. The test section is a horizontal and smooth tube-in-tube type heat exchanger, in which each subsection has an internal diameter of 1.77 mm, 160 mm long. The inner tube of the test section is made of smooth copper tube, The outer tube is made of a clear polyvinyl chloride (PVC) tube. Sight glass is mounted at the middle and