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Experimental investigations on adiabatic frictional pressure drops of R134a during flow in 5 mm diameter channel

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ABSTRACT

The article presents detailed two-phase adiabatic pressure drops data for refrigerant R134a at a saturation pressure of 5.5 bar corresponding to the saturation temperature of 19.4 °C. Study cases have been set for a mass flux varying from 100 to 500 kg/m^2 s. The frictional pressure drop was characterized for the refrigerant R134a, for vapor qualities ranging from 0 to 1. Long-time thermal stability of test facility allowed to gather a comprehensive experimental database for two-phase frictional pressure drop including multiple data points in transition and dryout flow regions. The effect of transition region on the peak value of two-phase frictional pressure drop, for literature models and experiment, is recognized.

A systematic assessment of predictive techniques for two-phase frictional pressure drop in adiabatic flows for varying vapor quality was conducted. Both qualitative and quantitative analysis of gathered data vs. literature models was presented. Verification of the pressure drop for two-phase adiabatic flow showed that for Zhang and Webb correlation 93% of experimental data fits in the range of ±30%. The model proposed by Thome et al. in other hand predicts almost 33% of data within 10% error, but only 80% of the data is predicted within 30% error. Additional prediction of the peak value of two-phase frictional pressure drop with literature models and the experiment was made.

1. Introduction

Precise control of cooling parameters is required in an ever wider range of technical applications. The boiling phenomena are amongst the most complex transport processes encountered in engineering applications. Phase-change processes include all the complexity of single-phase convective transport and additional problems resulting from the motion and deformation of the vapor–liquid interface. The flow of vapors and liquids in pipes, channels, equipment, etc. is frequently encountered in industry and has been studied intensively for many years. The reliable prediction of pressure drop in two-phase flows is thereby an important aim; yet, pressure gradients predicted using leading methods often differ by more than 50% according to many reports.

Problems arise because flow resistance due to friction in two-phase flow is greater than in the corresponding case of a single phase flow at the same flow rate. The two-phase flow multiplier is defined as a ratio of friction pressure drop in the two-phase flow, $\left(\frac{dp}{dz}\right)_{TP}$ to the friction pressure drop in the flow of either liquid of vapor $\left(\frac{dp}{dz}\right)_{0}$, as in eq.:

$$\Phi = \frac{\left(\frac{dp}{dz}\right)_{TP}}{\left(\frac{dp}{dz}\right)_{0}} \tag{1}$$

Two principal types of models were used in developing frictional pressure drop models namely: homogeneous and separated flow model. In the first, the flow of both phases is assumed to be in equilibrium, and the gas and liquid velocities are assumed equal (slip ratio $s = 1$). The frictional pressure drop is computed as if the flow were a single-phase flow, except for introducing modifiers to the properties inside the single-phase friction coefficient.

The homogeneous frictional pressure drop can be calculated as:

$$\Delta p_{frict} = \frac{2 \cdot f_{tp} \cdot L \cdot \dot{m}_{total}^2}{d_i \cdot \rho_{tp}} \tag{2}$$

where the two-phase friction factor may be expressed in terms of the Reynolds number by the Blasius equation:

$$f_{tp} = \frac{0.079}{Re^{0.25}} \tag{3}$$

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Nomenclature

A	area (m^2)
C	parameter in Lockhart-Martinelli correlation (–)
c_1, c_2, c_3	correlation constants
C_c	coefficient of contraction (–)
C_p	specific heat (J/kg K)
f	friction factor (–)
Con	confinement number (–)
D, d	diameter (m)
$\left(\frac{dp}{dz}\right)$	frictional pressure drop (–)
Fr	Froude number
G	mass flux (kg/m^2 s)
g	acceleration due to gravity (m/s^2)
h_{lv}	specific enthalpy of vaporization (kJ/kg)
j	superficial velocity (–)
L	channel length (m)
I	current (A)
M	molecular weight (kg/mol)
MAD	mean absolute deviation, MAD $\left[\frac{1}{N}\right]\left[\frac{\Sigma\lvert X_{pred}-X_{exp}\rvert}{X_{exp}}\right]$ (%)
$\dot{m}$	mass flow of refrigerant (kg/s)
ΔP	pressure drop (Pa)
p_{kr}	critical pressure (Pa)
$\dot{q}$	heat flux density (W/m^2)
$\dot{Q}$	heat flux (W)
Re	Reynolds number (–)
t	friction (N)
w, u	velocity (m/s)
We	Weber number (–)
x	quality (–)
X^2	Lockhart-Martinelli parameter (–)

Greek symbols

λ	thermal conductivity (W/m K)
μ	viscosity (Pa s)
ε	void fraction (–)
ρ	density (kg/m^3)
φ	void fraction
σ	surface tension (N/m)
Φ	two-phase multiplier
Ψ	dimensionless number in Lee and Lee correlation ()

Superscripts

a	acceleration
+	non-dimensional
cb	convective boiling
con	contraction
exp	expansion
f	forced flow
h	hydraulic
cr	critical
l	liquid
lo	liquid only
PB	pool boiling
sat	saturation
TP	two-phase flow
TPB	two-phase boiling
v, g	saturated vapor
vo, go	vapor only

Because flow properties are assumed homogeneous, Reynolds number is calculated as:

$$Re = \frac{\dot{m}_{total} \cdot d_i}{\mu_{TP}} \tag{4}$$

The two-phase viscosity for calculating the Reynolds number in literature sometimes is assumed simply as the viscosity of the liquid phase or as a quality averaged viscosity [1]:

$$\mu_{TP} = x \cdot \mu_v + (1 - x) \cdot \mu_l \tag{5}$$

The two-phase viscosity proposed by [2] is based on the mass averaged value of reciprocals as follows:

$$\mu_{TP} = \left(\frac{x}{\mu_v} + \frac{1 - x}{\mu_l}\right)^{-1} \tag{6}$$

Similarly, two-phase density can be calculated as:

$$\rho_{TP} = \rho_l \cdot (1 - \varepsilon_H) + \rho_v \cdot \varepsilon_H \tag{7}$$

And the homogeneous void fraction is determined as a function of quality:

$$\varepsilon_H = \frac{1}{1 + \left(\frac{u_v}{u_l} \cdot \frac{(1-x)}{x} \cdot \frac{\rho_v}{\rho_l}\right)} \tag{8}$$

In the separated flow model, the two phases are considered separate and therefore their velocities may differ. The correlations developed for conventional size tubes were based on a vast number of experimental data. One of the most commonly used models for two-phase flow pressure drop was developed by Lockhart-Martinelli [3]. That empirical correlation has been derived based on a large number of data for two-phase flow of air–water mixture, gasoline, naphtha, and oils. The experiments were carried out for a wide range of channel diameters from 1.5 to 26 mm. Friedel [4] developed the own correlation based on a database of 25,000 points for adiabatic flow through channels with d > 1 mm. Another correlation due to Müller-Steinhagen and Heck [5] proposed an empirical interpolation between all liquid and all vapor flow regimes. This correlation was developed based on the experimental data bank for a number of most commonly used refrigerants. In the literature review compiled by Ould-Didi et al. [6] this correlation is recommended especially for predictions of the two-phase pressure drop of refrigerants. All of the mentioned above methods are applicable to the whole range of vapor qualities.

As mentioned above, in the case of small diameter channels there are other correlations advised for use for two-phase pressure drop calculations, which have been primarily developed on the basis of Lockhart-Martinelli and Chisholm [7] models. Their major modification is based on the fact of inclusion of the surface tension effect into modeling. Amongst the most acknowledged correlations for flow boiling in mini-channels and small diameter channels are those due to Mishima and Hibiki [8], Tran et al. [9] and Zhang and Webb [10].

Mishima and Hibiki [8] investigated various flow structures, void fraction and mass fluxes of air–water mixtures in vertical tubes of diameters ranging from 1.05 mm to 3.9 mm and tube lengths from the range 210–1000 mm. This correlation is a modified form of the model proposed by Lockhart and Martinelli [3] correlation in which a new C value is suggested by incorporating the effect of channel diameter.

Another well-established correlation in literature for small diameter channels is the one due to Tran et al. [9], who modified the Chisholm correlation [7] on the basis of measurements of adiabatic pressure drop of R134a, R12 and R113 in two circular channels of internal diameters 2.46 and 2.92 mm and one square