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Experimental Investigations On The Momentum Pressure Drop During Flow Boiling Of R134a

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Abstract. The article presents experimental investigations of the pressure drop during two-phase flow. Experiments were performed for both adiabatic and heated flow of R134a. Obtained flow patterns were compared with the literature. Obtained data is used to validate momentum pressure drop predictions, a set of graphs showing comparisons, for a representative set of experimental conditions, of the two-phase frictional pressure gradients for the adiabatic and diabatic flow. The model proposed in the article allows to predict both values and peak pressure drop with very good accuracy. Verification of the momentum pressure drop predictions for two-phase adiabatic flow showed that all correlations have good agreement with experimental data.

1. Introduction

Striving for energy efficiency is clearly visible in every part of modern society. Despite the use of alternate energy sources, major gains can be made by increasing the effectiveness of energy utilization, i.e. recovering low-grade waste heat [1], or improving process efficiency [2]. An essential step to assure efficient work of power and process apparatus is their proper design [3]. Modern devices and machines operate with high heat flux densities. Therefore, usage of heat transfer heightening technologies i.e. microjet technology [4,5], flow turbulization [6–8].

A wide range of those devices operates within boiling or condensation of the working fluid to take advantage of high heat transfer coefficients [9]. Two-phase flows are related to large rates of heat transfer because of the latent heat of vaporization and augmentation of the turbulence level between the liquid and the solid surface [10]. Calculating heat transfer coefficient and pressure drop is a challenging task, and has been pursued by numerous research for decades. In the case of diabatic flows, the total pressure drop is due to the conversion of kinetic and potential energy. Also, it is affected by the flow resistance at the wall channel. It has been shown that total pressure drop is the summary of acceleration, static and frictional pressure drop [11]. For two-phase flow acceleration and static pressure drop components are present, a careful selection of void fraction correlation becomes necessary for reliable prediction of these two pressure drops.

The frictional pressure drop of two-phase flow is a function of fluid properties and significantly depends on mass and volume fractions of the mixture components. It was observed that vapor quality and void fraction parameters are unceasingly shifting. This affects overall flow pattern and thus affects pressure drop of the system. Past studies aimed at predicting the frictional pressure drop, includes a large number of studies that rely on either the Homogeneous Equilibrium Model (HEM) or semi-empirical correlations [12]. Sometimes researchers have contradictory views concerning the same issue, such as the application of a given correlation in the calculation of the flow pressure drop or heat



transfer coefficient, because of a vast number of existing correlations [13]. This problem is even more pronounced in fluid mixtures flows [14].

The prediction of the two-phase pressure gradient is an essential step in the design of a variety of equipment in the power and process engineering. This study focuses on experimental values of the total, frictional and momentum pressure drop components and their prediction.

2. Experimental test facility

In order to obtain experimental values of momentum pressure drop component, the experimental facility is an R134a loop consisting of a reservoir of refrigerant, a pump, a conditioner, a flow meter, a heated test section, a visualization glass, an adiabatic section and a condenser, see figure 1. The view of the test facility is presented in figure 2.

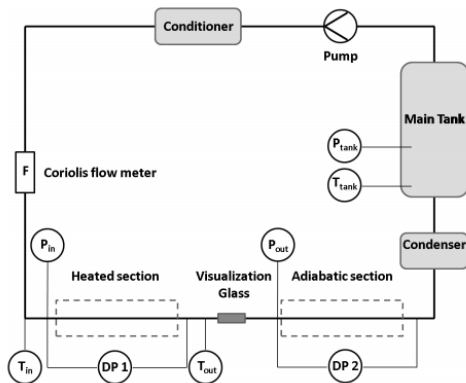


Figure 1. Schematic of the test facility.



Figure 2. The view of the test facility.

The fluid pressure in the loop is set by controlling the temperature in the main tank where the refrigerant is at saturation conditions. The fluid is circulated by a magnetically coupled gear pump. The conditioner is a shell and tube heat exchanger with glycol in the shell side which is used for adjusting the R134a inlet temperature. The fluid flow rate is measured by a Coriolis type mass flow meter. The heated division is made of stainless steel, as a seamless pipe. The tube is electrically heated by Joule effect with the use of a low voltage DC power supply. The section is thermally insulated with a thick layer of mineral wool, thus thermal losses are neglected. The adiabatic test section is a thermally insulated 1m long stainless steel pipe with an inner diameter of 5mm and 8mm outer diameter. The piping system at the supply and outlet of the sections is made with similar pipe dimensions, thus compression and expansions effects are eliminated. Flow visualization allowed for verification of the obtained vapor-liquid mixture patterns, as can be seen in figures 3 and 4

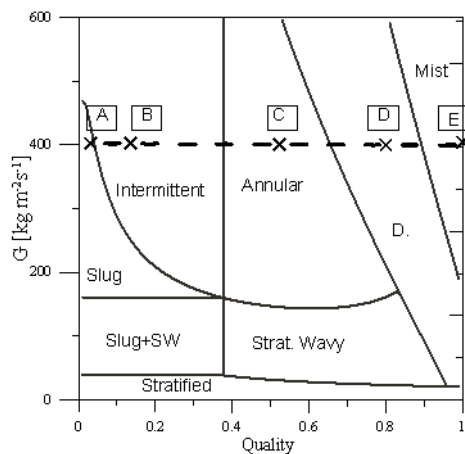


Figure 3. Flow pattern map

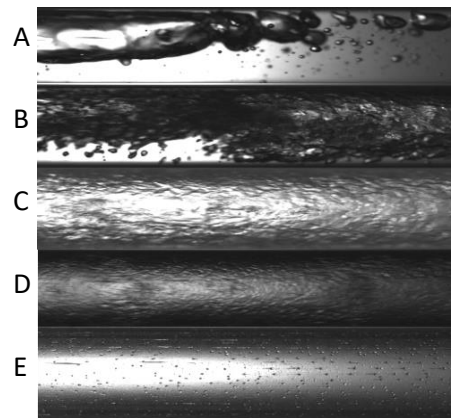


Figure 4. Obtained flow structures.

The measurements data are acquired by National Instruments CompactRIO data acquisition system. The signals from measuring devices were processed with the aid of the LabVIEW application. The temperatures, absolute pressures, pressure differences and mass flow rates were acquired at a frequency of 2 Hz. For every experimental point - about 100 data points were acquired. Additionally, in order to verify the steady-state conditions, data points were doubled after 15 minutes. The range of experimental conditions for the obtained database is presented in table 1.

The local heat flux as the function of generated Joule heat is assumed to be constant during the evaporation process along the length of the tube. Uncertainties were estimated according to the standard procedures described by NIST [15], the measuring device and overall measurement uncertainties are grouped in table 1.

Table 1. Experimental conditions and uncertainty of experiments

Parameters	Range	Uncertainty
x	0 ÷ 1	±5%
D [mm]	5	
L _h [m]	2	
L _{ad} [m]	1	
T _{sat} [°C]	19.4	
G [kg/m ² s]	150 ÷ 500	±0.2%
q̇ [kW/m ²]	0.1 ÷ 69	±3%

3. Data reduction

The total pressure drop of fluid Δp_{total} is the summation of acceleration, static and frictional pressure drop according to eq. (1) if acceleration and static pressure drop are present.

$$\Delta p_{total} = \Delta p_{mom} + \Delta p_{frict} + \Delta p_{grav} \quad (1)$$

where Δp_{grav} is the elevation head pressure drop and can be omitted in case of a horizontal tube, Δp_{mom} is the momentum pressure drop created by the acceleration of the flow in a heating/cooling process, and Δp_{frict} is the frictional pressure drop of two-phase flow. Most commonly ([16–19]) the momentum pressure is expressed by means of eq. (2).

$$\Delta p_{mom} = G^2 \left\{ \left[\frac{x^2}{\rho_v \varepsilon} + \frac{(1-x)^2}{\rho_l (1-\varepsilon)} \right]_{out} - \left[\frac{x^2}{\rho_v \varepsilon} + \frac{(1-x)^2}{\rho_l (1-\varepsilon)} \right]_{in} \right\} \quad (2)$$

Differences between void fraction correlations are minor, therefore in the further analysis, only three formulas were used. The author's previous experimental studies on pressure drop [20–22] have shown that the predictions given by various methods differ significantly, therefore a modification of Thome [11] approach based on flow pattern map was proposed.

The two-phase frictional pressure drop for annular flow, usually is calculated as:

$$(\Delta p_{fric})_{annular} = 2 \cdot (f_i)_{annular} \cdot \frac{\rho_v \cdot u_v^2}{d_i} \quad (3)$$

The modification proposes to calculate the friction pressure drop from Eq.3 at the interface of the liquid-vapor mixture based on modified core flow density ($\rho_c = \rho_c$) due to entrainment. The velocity of the core gaseous flow is affected by the entrainment:

$$u_v = \frac{G \cdot x}{\rho_c \cdot \varepsilon} \quad (4)$$

The average density of gaseous core is calculated based on mass flow rate and the density of the vapor-droplet mixture at the core of the annular flow. It is calculated assuming homogeneous flow for the core flow as follows

$$\rho_c = (1 - \varepsilon_c)\rho_L + \varepsilon_c\rho_V \quad (5)$$

where ε_c is the fraction of the core flow with entrained droplets, :

$$\varepsilon_c = \frac{\varepsilon}{\varepsilon + E(1 - \varepsilon)} \quad (6)$$

The liquid entrainment fraction correlation by Tibiriçá [23] was used in this study:

$$E = \left(1 + 10^{6.8592} \left(\frac{\rho_V}{\rho_L} \right)^{0.6267} We_{jv}^{-1.1641} Re_L^{-0.3988} \right)^{-1} \quad (7)$$

where the superficial gas velocity Weber number is defined as:

$$We_{jv} = \frac{\rho_v \cdot j_v^2 \cdot D}{\sigma} \quad (8)$$

The superficial gas velocity in [23] is predicted as:

$$j_v = \frac{x \cdot G}{\rho_v} \quad (9)$$

and liquid Reynolds number :

$$Re_L = \frac{G \cdot D}{\mu_L} \quad (10)$$

Therefore friction factor calculated based on homogeneous vapor – droplet mixture flow in the core can be written as:

$$f_i = 0.67 \cdot \left[\frac{\delta}{d_i} \right]^{1.2} \cdot \left[\frac{(\rho_l - \rho_c) \cdot g \cdot \delta^2}{\sigma} \right]^{-0.4} \cdot \left[\frac{\mu_v}{\mu_l} \right]^{0.08} \cdot We_l^{-0.034} \quad (11)$$

where film thickness, Webber number and velocities of phases are calculated as follows:

$$\delta = \frac{d_i}{2} - \left[\left(\frac{d_i}{2} \right)^2 - \frac{(1 - \varepsilon_c) \cdot \pi \cdot d_i^2}{2 \cdot (2 \cdot \pi - \Theta_{dry})} \right]^{1/2} \quad (12)$$

$$u_l = \frac{G \cdot (1 - x)}{\rho_l \cdot (1 - \varepsilon)} \quad (13)$$

$$We_l = \frac{\rho_l \cdot u_l^2 \cdot d_i}{\sigma} \quad (14)$$

Literature observations state that slug and intermittent flow regimes are difficult to separate and have similar behavior. Therefore similar methodology was adopted for calculation pressure drop in these regions:

$$(\Delta p_{fric})_{slug+intermittent} = (\Delta p)_l \cdot \left(1 - \frac{\varepsilon_c}{\varepsilon_{IA}} \right)^{0.25} + (\Delta p)_{annular} \cdot \left(\frac{\varepsilon_c}{\varepsilon_{IA}} \right)^{0.25} \quad (15)$$

The frictional pressure gradient of a mist flow is obtained as:

$$(\Delta p_{fric})_{mist} = \frac{2 \cdot (f_i)_v \cdot G^2}{d_i \cdot \rho_h} \quad (16)$$

When dryout regime does exist, the following linear interpolation was used to capture the variation in frictional pressure gradient across the regime without introducing any jump in the value:

$$(\Delta p_{fric})_{dryout} = (\Delta p)_{x=x_{di}} - \frac{x - x_{di}}{x_{de} - x_{di}} \cdot [(\Delta p)_{x=x_{di}} - (\Delta p)_{x=x_{de}}] \quad (17)$$

where x_{di} and x_{de} are denoting quality values at the inception and completion of the dryout of the refrigerant respectively:

$$x_{di} = 0.58 \cdot \exp \left[0.52 - 0.000021 \cdot We_v^{0.96} \cdot Fr_v^{-0.02} \cdot \left(\frac{\rho_v}{\rho_l} \right)^{-0.08} \right] \quad (18)$$

$$x_{de} = 0.61 \cdot \exp \left[0.57 - 0.0000265 \cdot We_v^{0.94} \cdot Fr_v^{-0.02} \cdot \left(\frac{\rho_v}{\rho_l} \right)^{-0.08} \right] \quad (19)$$

$$We_v = \frac{G^2 \cdot d_i}{\rho_v \cdot \sigma} \quad (20)$$

$$Fr_v = \frac{G^2}{\rho_v \cdot (\rho_l - \rho_v) \cdot g \cdot d_i} \quad (21)$$

The obtained experimental pressure drop data were compared with the Darcy-Weisbach correlation, finding the friction coefficient according to the Haaland [24] equation, with pipe internal roughness height given by supplier lower than 0.03mm, having the best agreement. Therefore this correlation was used to predict single phase flow friction factors for vapor and liquid flows.

Electric heating of the channel allows us to assume constant heat flux density along the heated channel length, therefore a linear change in local thermodynamic vapor quality of the flow can be assumed. Therefore the pressure drop along the heated channel can be recreated by using friction pressure drop values at "local" vapor qualities at heated channel length.

Rearranging eq.(1) the momentum pressure drop can be calculated by subtracting frictional pressure drop from measured values of total pressure drop in heated section. This operation for an infinite number of points can be expressed as an integral in form of eq.(22). The accuracy of this method highly depends on polynomial interpolation of the data.

For presented data, the best fit error was below 1%, which corresponds to total momentum pressure drop of 1.15%

$$\Delta P_{ad,decomposition} = \int_0^L \frac{d}{dz} \Delta P_{ad}(x) dz \quad (22)$$

The pressure drop component due to acceleration varies with outlet vapor quality and void fraction. Literature models of a void fraction are created with the assumption that the same flow regimes are obtained at the same local liquid and vapor flow rates. As explained earlier proper selection of void fraction model becomes crucial when evaluating the influence of momentum and elevation pressure drop components. In this study for calculating momentum pressure drop homogeneous:

$$\varepsilon = \frac{1}{1 + \left(\frac{1-x}{x} \right) \cdot \left(\frac{\rho_v}{\rho_l} \right)} \quad (23)$$

and Cioncolini [25] void fractions correlations were used:

$$\varepsilon = \frac{h \cdot x^n}{1 + (h-1)x^n} \quad (24)$$

Where: $h = -2.139 + 3.129(\rho_v \cdot \rho_l^{-1})^{-0.2186}$, $n = 0.3487 + 0.6513(\rho_v \cdot \rho_l^{-1})^{0.5150}$.

4. Experimental results

Adiabatic pressure drops as indicated in the description of the test section can be measured after preparation of vapor-liquid mixture in heated section. Varying the inlet subcooling temperature of working fluid will influence the amount of heat necessary to obtain same vapor quality. Figure 5 shows the experimental two-phase pressure drop of the 5 mm tube as a function of mass flux and heat flux of R134a at different subcooling temperatures, at a saturation temperature of 19.4 °C.



Figs. 6 to 8 shows the experimental adiabatic frictional pressure drops, plotted versus values predicted with the presented model. The pressure drop increases exponentially with increasing mass flux. It can also be seen that the two-phase pressure drop is increasing for higher exit vapor quality with a maximum around vapor qualities around $x=0.8-0.9$, as reported in the literature. The proposed model modification with entrainment effect allows predicting the peak pressure drop values. As can be clearly seen proposed modification provides very good data representation for selected flow range.

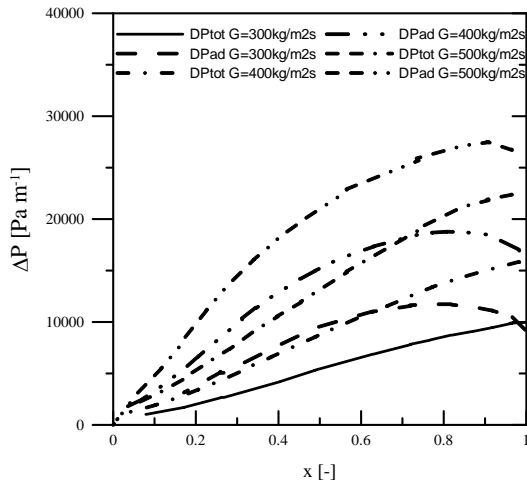


Figure 5. Experimental values of the total and adiabatic pressure drop of R134a versus exit vapor quality of the 5 mm tube at $T_{sat}=19.4\text{ °C}$

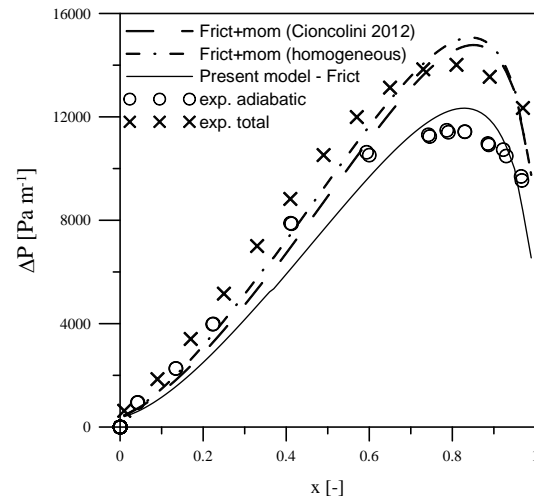


Figure 6. Experimental values of the total and adiabatic pressure drop of R134a versus predictions for $G=300\text{ kg/m}^2\text{s}$

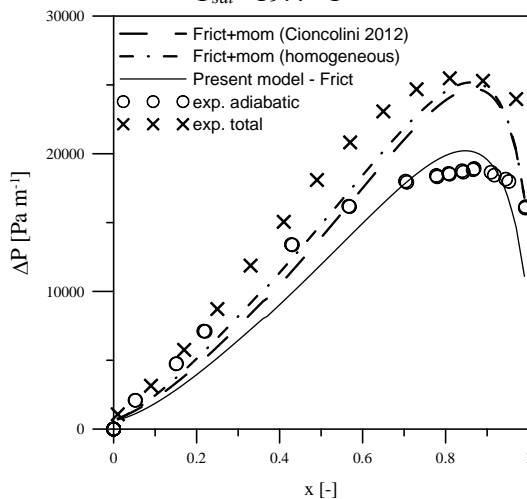


Figure 7. Experimental values of the total and adiabatic pressure drop of R134a versus predictions for $G=400\text{ kg/m}^2\text{s}$

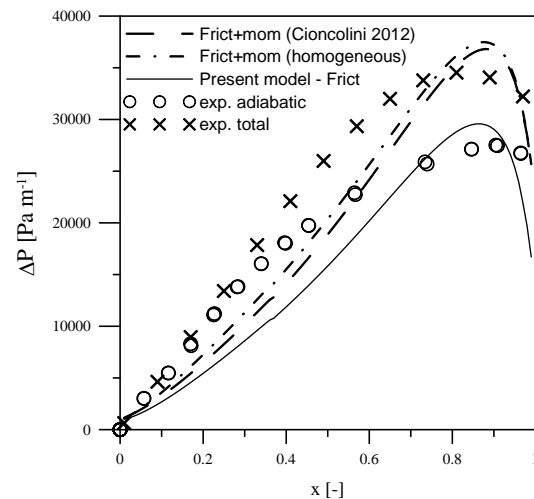


Figure 8. Experimental values of the total and adiabatic pressure drop of R134a versus predictions for $G=500\text{ kg/m}^2\text{s}$

From the analysis of data presented in figures 6 to 8, a good consistency with the proposed friction pressure drop model and momentum is visible for all of the cases. The influence of various void fraction correlations is more pronounced in higher velocities., as the mass flux is crucial scaling parameter in eq. (2).



5. Conclusions

The experiments in annular flow range were performed to validate void fraction correlations for specific applications of flow boiling momentum pressure drop. An experimental study was undertaken in order to obtain accurate total and frictional pressure drop values R134a in a smooth horizontal 5mm tube, at a saturation temperature of 19.4C. Based on collected experimental data, from a comparison of the adiabatic frictional pressure drop and total pressure drop, momentum pressure drop was calculated. The experimental campaign acquired over 500 experimental data points. Presented data along with data reduction procedure was used to obtain the momentum pressure drop values during flow boiling. Verification of the momentum pressure drop predictions showed that void fraction correlations predict experimental data in the range of $\pm 30\%$. Proposed modification of the flow pattern based model allows for very good agreement with own experimental data. Presented procedure allowed for prediction of the peak pressure drop in both adiabatic and diabatic flow.

Nomenclature

D	diameter, [m]
E	entrainment fraction [-]
f	friction factor, [-]
Fr	Froude number [-]
Ft	Froude rate, [-]
G	mass flux, [kg/m ² s]
g	acceleration due to gravity [m/s ²]
h	enthalpy [kJ/kg]
L	length [m]
ΔP	pressure drop [Pa]
\dot{Q}	heat flux [kW]
\dot{q}	heat flux density [kW/m ²]
Re	Reynolds number [-]
We	Webber number [-]
x	quality [-]
X_{tt}	Lockhart-Martinelli parameter [-]

Greek symbols

ε	void fraction [-]
μ	viscosity [Pas]
ρ	density [kg/m ³]

Superscripts

ad	adiabatic
A	Annular
c	core
D	dryout
fric	frictional
h	heated
in	inlet
I	Intermittent
l	liquid
mom	momentum
out	outlet
sat	saturation
sub	subcooling
static	gravitational
v	vapor
w	wall

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