EXPERIMENTAL ANALYSIS OF PRESSURE DROP IN SINGLE AND TWO PHASE IN MINI CHANNELS

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ABSTRACT

Evaporators with mini and micro channels are one of the main focuses in the design and development of equipment applied to compact refrigeration systems. The objective of this work is to investigate pressure drop of natural refrigerant, isobutane (R-600a), in the single-phase flow through two small tubes, with 1.0 mm and 2.6 mm of internal diameter. Also, the pressure drop was analyzed in the boiling flow in a 2.6 mm internal diameter tube. The experimental tests included mass velocities of 188, 240, 280 and 370 kg/(m²s), heat fluxes in the range from 0 to 134 kW/m² and boiling flow the saturation temperature of 22 ºC and vapor quality up to 0.8. It was possible to observe the significant influence of the diameter and mass velocity on the total pressure drop and the frictional pressure drop, respectively. The experimental frictional pressure drop in flow boiling in 2.6 mm of internal diameter was compared with four different correlations in literature.

Keywords: pressure drop, natural refrigerant, single phase, boiling, mini channel.

NOMENCLATURE

A	area, m²
D	diameter, m
e	absolute internal roughness, μm
f	friction factor
ID	internal diameter, m
G	mass velocity, kg/(m²s)
L	length, m
$\dot{m}$	mass flow rate, kg/s
p	pressure, kPa
P	electrical power supplied, W
PH	preheater
q"	heat flux, kW/m²
Re	Reynolds number
T	temperature, ºC
TS	test section
x	vapor quality

Greek symbols

α	void fraction
η	efficiency of thermal insulation
ρ	density, kg/m³
ν	specific volume, m³/kg

Subscripts

ac	acceleration
exp	experimental
f	frictional
G	gas
i	inlet
in	entrance
L	liquid
out	exit
pred	predict
sat	saturation
tot	total

INTRODUCTION

Compact heat exchangers should be able to dissipate high heat fluxes and at the same time to achieve better thermal performance and reliability with little pressure drop. These devices are employed in small size refrigeration systems, which find use in air conditioners applications, heat pumps, cooling of electronic devices and in advanced applications with microprocessors. In comparison with single-phase flow, flow boiling is a good option to be applied in mini-channels because of its extremely high heat transfer efficiency with small wall temperature rises. However, a penalty of flow boiling is the increased pressure drop and pressure fluctuation, which can limit the applicable range of flow boiling in such devices. Therefore, a comprehensive understanding of pressure drop in mini-channel is important for accurate design, performance evaluation and optimization of these heat exchangers.

Experimental results have been presented in recent years. Qi *et al*. (2007) described the characteristics of pressure drop in single phase in turbulent flow of liquid nitrogen in tubes of internal diameter ranging from 0.53 mm to 1.93 mm.

The results for boiling flow of different refrigerants have been reported by Greco and Vanoli (2006), in a horizontal tube of 6 mm ID. Pamitran *et al*. (2010) showed results for mini tube and R-22, R-134a, R-410, R-290 and R-744. Tran *et al*. (2000) analyzed the pressure drop for tubes of 2.92 and 2.46 mm *ID* using R-113, R-12 and R-134a. Saitoh *et al*. (2005) studied the pressure drop in the flow boiling analyzing the influence of the internal diameter in the test section. In general the results have showed that pressure drop depends on mass velocity, vapor quality and, also, heat flux.

The objective of the present experimental study is: (i) develop an accurate data base of flow boiling pressure drops in a mini tube, in order to gather new design data for heat exchanger design with natural fluid, R-600a; (ii) compare the measured data with available correlations.

EXPERIMENTAL SET UP AND TESTS

An experimental apparatus was used to investigate the pressure drop in single-phase and boiling flow in a horizontal mini channel electrically heated. The experimental apparatus, according to Fig. 1, comprises a circuit which provides mass flow rate controlled to a wide range of flow conditions. The main part of the circuit has a preheater section, the test section and a visualization section. The secondary part consists of a refrigerant reservoir, condenser, and sub-cooler, with separate circuits using a solution of ethylene glycol and water as a secondary coolant. The sub-cooler is used to compensate any increase of temperature experienced by the refrigerant when passing the pump. The preheater establishes the inlet conditions (vapor quality) in the test section. Both consist of horizontal stainless steel tubes thermally insulated with 445 mm and 185 mm length, respectively, and 2.6 mm *ID*, heating by Joule effect controlled by power supply. Downstream the test section there is a visualization section constituted by a glass tube with 158 mm length and the same internal diameter of the test section.

The measurements of pressures and temperatures at the inlet and outlet of the preheater are held respectively by two absolute pressure transducers and thermocouples type E of 0.076 mm, in contact with the refrigerant, and three thermocouples in external wall. In the test section the pressure drop are measured by a differential pressure transducer and the refrigerant temperature by thermocouples in the inlet and exit of the section, besides the wall temperatures along the tube.

A frequency inverter is used to control the pump flow rate and a bypass line downstream the pump and parallel to the preheater and the test section is used for more precise adjustments of flow and is controlled by a needle-valve. The pressure transducers, thermocouples and the mass flow meter are connected to a data acquisition system controlled by a computer.

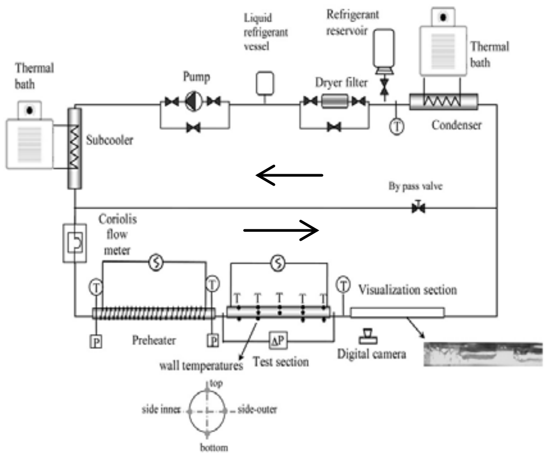

Figure 1. Schematic diagram of experimental apparatus.

All experiments were carried out with isobutane (R-600a) to measure and analyze the behavior of pressure drop in single-phase flow and in boiling. The single phase tests were carried out in diabatic and adiabatic conditions. Table 1 shows the single phase and boiling tests conditions. During boiling tests different heating powers in the preheater were used to establish the vapor quality at the entrance of the test section in each experiment.

Another test section with 1.0 mm *ID* and 148 mm in length was developed with the purpose of analyzing the effects of diameter on the pressure drop in adiabatic single-phase flow. In both sections, the absolute roughness, *e*, is 2.5 μm.

Table 1. Summary of test conditions using experimental apparatus with 2.6 mm *ID* tube.

Parameter	1.0 mm *DI*	2.6 mm *DI*
q” (kW/m²)	0	0, 28, 56, 100 e 134
G (kg/m²s)	440 to 740	188 to 590
T_{sat} (ºC) and p_{sat} (kPa)	-	22/323.50
Average *T* (ºC) and pressure (kPa) in single phase	8/290	8/290

Adiabatic single-phase tests

The total pressure drop in the preheater and test section during single phase tests were evaluated and they are the sum of frictional pressure drop (Δp_f) and the entrance and exit losses (Δp_{out-in}).

$$\Delta p_{tot} = \Delta p_f + \Delta p_{out-in} \quad (1)$$

In Eq. (1), the inlet and exit losses are due to the contraction or enlargement of the flow area and they are minor losses.

$$f_{exp} = \frac{2\Delta p_f \rho D_i}{G^2 L} \quad (2)$$

From the frictional pressure drop it was possible to calculate the experimental friction factor, f_{exp}, using the Eq. (2).

The results obtained with respect to f_{exp} were compared with the correlations proposed by Blasius (1913) and Petukhov (1970) for smooth tubes, and Colebrook (1939) that consider the roughness.

$$f = 0.316\,Re^{-0.25} \quad (Re < 2\times10^4) \quad (3)$$

$$f = [0.79\ln(Re) - 1.64)^{-2}] \quad (3000 \le Re \le 5\times10^6) \quad (4)$$

$$f = \left\{-1.8\log\left[\frac{6.9}{Re} + \left(\frac{e}{3.7D}\right)^{1.11}\right]\right\}^{-2} \quad (\frac{e}{D} > 1\times10^{-6}) \quad (5)$$

These correlations are given by Eqs. (3), (4) and (5), respectively. This comparison procedure has the purpose of ensure the validation of the experimental apparatus.

Diabatic single-phase tests

Before running boiling experiments, single phase tests were performed to evaluate the heat losses in the preheater and test section for different conditions of heat flux and mass flow. Assuming that the efficiency of the heat transfer process is the rate of heat transferred to the fluid divided by the electrical power input.

$$\eta = \frac{\dot{m}(i_{out} - i_{in})}{P} \quad (6)$$

As derived from Eq.(6), the efficiency of both pre-heater and test section could be determined for a set of operation conditions, where η is the efficiency of thermal , $\dot{m}$ is the mass flow rate, i_{out} e i_{in} are the exit and inlet enthalpies, respectively, and P is the electrical power supplied. The enthalpies were determined using the measured values of pressure and temperature.

Flow boiling tests

In the flow boiling, the total pressure drop also includes the momentum pressure drop, ΔP_{ac}.

$$\Delta p_{tot} = \Delta p_f + \Delta p_{out-in} + \Delta p_{ac} \quad (7)$$

The momentum pressure drop, indicated in Eq. (7), is caused by acceleration of both phases, is a result of the refrigerant specific volume enhancement in evaporation.

$$\Delta p_{ac} = G^2\left\{\left[\frac{(1-x)^2}{\rho_L(1-\alpha)} + \frac{x^2}{\alpha\rho_G}\right]_{out} - \left[\frac{(1-x)^2}{\rho_L(1-\alpha)} + \frac{x^2}{\alpha\rho_G}\right]_{in}\right\} \quad (8)$$

The momentum pressure drop component is calculated by Eq. (8), where G is the mass velocity, x is the vapor quality, ρ_L and ρ_G are liquid and vapor densities, respectively.

$$\alpha = \frac{\nu_G x}{\nu_G x + \nu_L(1-x)} \quad (9)$$

As described in Eq. (9), the void fraction α is calculated according homogenous model in function of specific volume of gas ν_G and of liquid ν_L.

RESULTS AND DISCUSSION

The pressure drop of the refrigerant in adiabatic flow was determined in the pre-heater and in the test section, with 2.6 mm ID. Fig. 2 shows the variation of the pressure drop relative to the length of each section (L_{PH}=445 mm and L_{TS}=185 mm) with Reynolds number, Re. The pressure drop increases significantly with increasing the mass velocity, and therefore with the Re number.

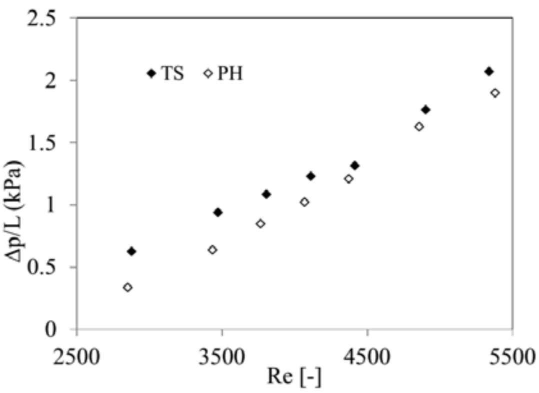

Figure 2. Single phase flow results for pressure drop variation with flow regime.

Table 2 shows the results of pressure drop in test section and experimental friction factors obtained from different mass velocities, G, in the single phase tests.

Table 2. Results on pressure drops and experimental friction factors obtained in the single phase tests in tube of 2.6 mm ID.

G [kg/(m²s)]	Re [-]	Δp_{TS} [kPa]	f_{exp} [-]
196	2774	0.116	0.04898
272	3762	0.207	0.04567
367	5130	0.318	0.03836
590	6873	0.880	0.03409

The comparison of the experimental friction factors in the test section of 2.6 mm ID with the correlations is shown in Fig. 3. It is possible to observe that the predicted friction factors from Petukov (Eqs. 4), and Colebrook (Eq.5) are closer to the measured data than the ones predicted by Blasius (Eq. 3), mainly for Reynolds numbers less than 4.000.

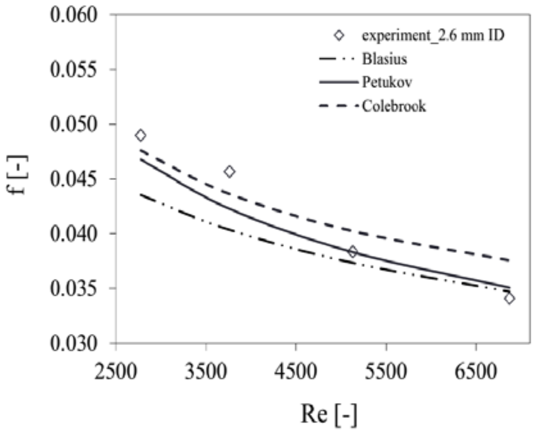

Figure 3. Experimental friction factor compared with predicted factors by correlations.

The effect of internal diameter on the total pressure drop for adiabatic single-phase tests, was analyzed for four different flow regimes. Figure 4 shows the results of total pressure drop as a function of Re number. It is observed a greater influence of the diameter with the increase of flow rate during the tests. An increment up to 9 times was observed for tube of 1 mm diameter comparing to the one of 2.6 mm. Such results are in accordance with the studies presented by Qi *et al.* (2006), Yang and Lin (2007).

Flow boiling tests

In the boiling tests it was evaluated the variation of pressure drop in the test section with the vapor quality for different mass velocities and heat fluxes.

The frictional pressure drop was calculated from Eq. (7) and represents 74.5% of total pressure drop, whereas the momentum pressure drop (acceleration) represents 24.5%, and the entrance and exit losses only 1%.

In Fig. 5 is shown the results of frictional pressure drop for constant heat flux of 34 kW/m². It is possible to observe that with increasing mass velocity, G, there is an increase in the pressure drop during boiling tests. It can be related to the vapor velocity increasing, and the effects caused by acceleration losses. Similar trends were presented by Pamitran et al. (2010) using five different refrigerants in their experiments.

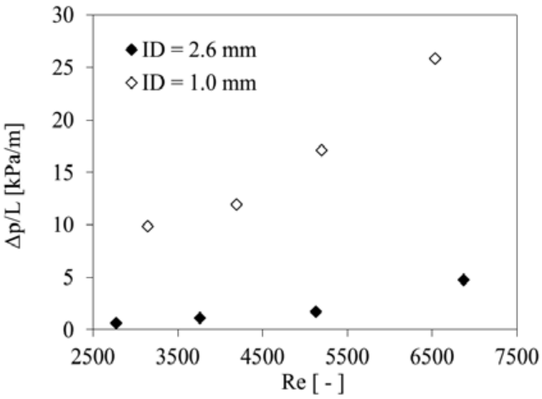

Figure 4. Effect of tube diameter on total pressure drop.

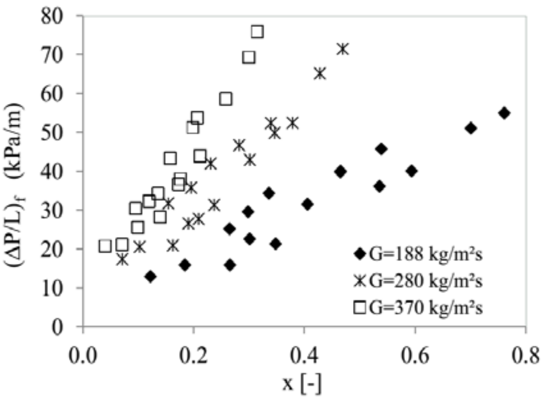

Figure 5. Effect of mass velocity on frictional pressure drop as a function of vapor quality.

Fig. 6 the effect of heat flux in pressure drop is pointed out. Despite the expectation that the heat flux would not affect the frictional pressure drop component, the experimental results show small variations in frictional pressure drop with heat flux. The trend of the current experimental results is similar to that shown by Pamitran et al. (2010). However, a high dependence of frictional pressure drop on mass velocity is possible to be verified in the same figure. This trend is due to the dependence of pressure drop on the volumetric flow rate, which is directly increased by the increase of vapor quality and mass velocity. Saisorn et al. (2010) and Ould Didi et al. (2002) also observed similar results in their work.

The experimental data of frictional pressure drop during boiling in 2.6 mm ID tube were compared with some correlations.

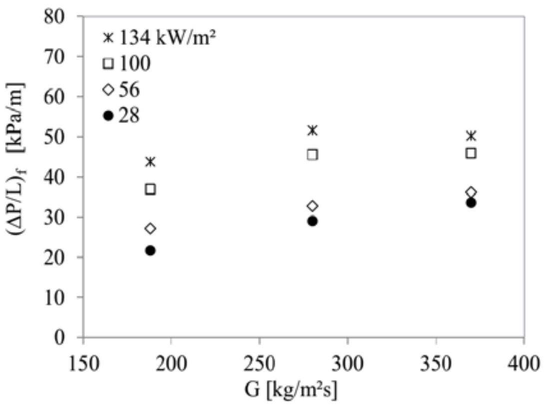

Figure 6. Effect of heat flux on pressure drop for different mass velocities.

The pressure drop homogenous models of Tran et al. (2000) and Zhang and Webb (2001) for prediction frictional pressure drop in mini and micro channels were evaluated. The correlations proposed for macro channels by Friedel (1979) and Müller-Steinhagen and Heck (1986) were also considered in this evaluation. The adjustment is shown in Fig. 7 for an error range of 35%, and the first two models proposed for mini channels can represent the data to within an error of 35% (more than 80% of points are in the range). The macro-scale models fail to predict mini scale pressure drop response because of the distinct flow behavior which happens in small diameter tubes.

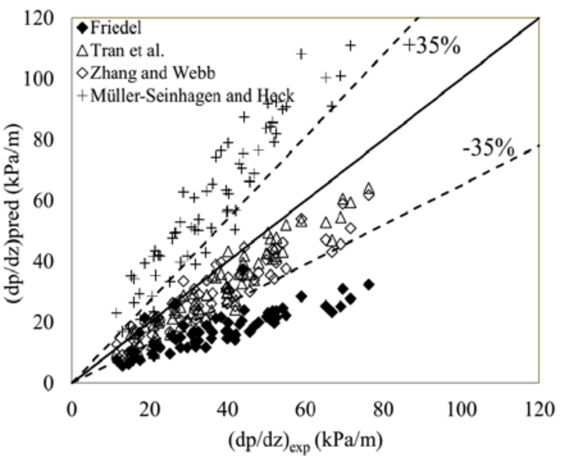

Figure 7. Comparison between the experimental frictional pressure drop results and the prediction methods for boiling.

CONCLUSIONS

Experimental results for the flow boiling of R-600a in a horizontal mini channels under the variation of mass velocity, heat flux and vapor quality were presented.

In single phase tests, it was observed the influence of the diameter on total pressure drop. An increment up to 9 times was observed for 1mm ID tube comparing to 2.6 mm ID tube.

The classical correlations for single phase friction factor were in agreement with experimental data for tube of 2.6mm internal diameter.

The frictional pressure drop in boiling increases with vapor quality and mass velocity and some influence of heat flux was detected.

For two-phase pressure drop, the Tran et al. and Zhang-Webb mini-channels models were the best fitted to the experimental data.

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