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FLOW BOILING HEAT TRANSFER AND TWO-PHASE PRESSURE DROP OF ISOBUTANE IN A 1.1 MM DIAMETER TUBE.

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Abstract. *In the present paper, two-phase pressure drop and flow boiling heat transfer results of isobutane (R600a) in a circular channel with an internal diameter of 1.1 mm are presented. Experiments were performed for mass velocities ranging from 100 to 500 kg/m²s, heat fluxes up to 145 kW/m², saturation temperatures of 31 and 41°C, and vapor qualities from 0.05 to 0.91. The experimental data are carefully analyzed and discussed. Subsequently, the experimental data were compared against predictive methods from literature. For pressure drop, the methods by Sempértegui-Tapia and Ribatski (2016) and Del Col et al. (2013) provided accurate predictions of the results. For heat transfer coefficient, the methods of Kanizawa et al. (2015) and Liu and Winterton (1991) provided the best predictions. However, both methods mostly underpredicted the experimental data characterized by the predominance of convective effects.*

Keywords: *flow boiling, pressure drop, microchannel, isobutane.*

1. INTRODUCTION

During the last two decades, the Montreal and Kyoto Protocols have caused a great impact on the research in the refrigeration area due to the necessity of performing new experimental studies for "natural" refrigerants with no ozone depletion potential (ODP) and a global warming potential (GWP) less than 150, and thereby gradually eliminate the use of HFC refrigerants. In the last years, some experimental studies on two-phase heat transfer coefficient and pressure drop for natural refrigerants such as ammonia (R717), isobutane (R600a), propane (R290) and propylene (R1270) were performed. However, the majority of these studies were performed for conventional channels (hydraulic diameter higher than 3 mm) or heat sinks based on multi-microchannels. Therefore, there is still limited information about the capability of the predictive methods available in literature to predict the pressure drop and the heat transfer coefficient of the "natural" fluids.

In this sense, the present paper concerns an experimental investigation on two-phase pressure drop and the heat transfer coefficient of isobutane (R600a) in a horizontal micro-scale single channel. Experiments were performed for a tube with internal diameter of 1.1 mm, mass velocities ranging from 200 to 500 kg/m²s, heat fluxes up to 145 kW/m², saturation temperatures of 31 and 41°C and vapor qualities from 0.05 to 0.91. Additionally, the experimental data for pressure drop and heat transfer coefficient were compared against ten predictive methods from the literature. The methods were evaluated according to four criteria: the mean absolute error (MAE), the mean relative error (MRE), the parcel of data predicted within error bands of ±30% and ±20%.

2. EXPERIMENTS

2.1 Experimental apparatus

The experimental setup is comprised of refrigerant and water circuits. The water circuit is intended to condense and subcool the working fluid. The refrigerant circuit is schematically shown in Figure 1a. In the refrigerant circuit, the test fluid is driven by a self-lubricating oil-free micropump through the circuit. The liquid flow rate is set by a frequency inverter acting on the micropump. Downstream the micropump, the mass flow rate is measured with a Coriolis flow meter. Just upstream the pre-heater, the inlet conditions are determined by a thermocouple and an absolute pressure transducer.

The pre-heater and the test section are 200 and 150 mm long, respectively. The pre-heater and the test section are heated by applying direct DC current to their surface and both sections are thermally insulated. The heating power is supplied by independent DC power sources controlled from the data acquisitions system. A differential pressure transducer measures the pressure drop, Δp, from the pre-heater inlet to the test section outlet. A detailed scheme of the pre-heater and the test section is shown in Figure 1b indicating the position of the thermocouples used for the wall temperature measurements.

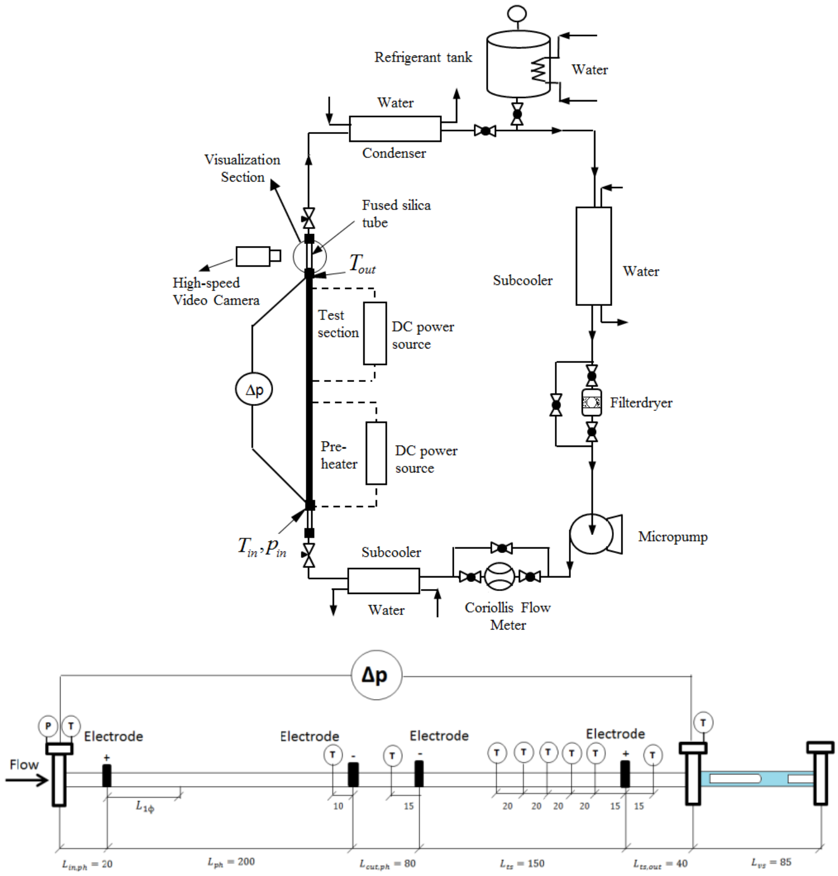

Figure 1. Schematic diagram of the: a) experimental apparatus, b) pre-heater and the test section.

2.2 Data reduction

The local vapor quality was determined through an energy balance over the pre-heater and the test section according to the following equation:

$$x(z) = \frac{1}{i_{LG}(z)}\left[\frac{P_{ph} + P_{ts}(z)}{\dot{M}} + \left(i_{L,in} - i_L(z)\right)\right] \tag{1}$$

where $i_{L,in}$ is the enthalpy of the liquid at the inlet of the pre-heater, $i_L(z)$ and $i_{LG}(z)$ are the enthalpy of the saturated liquid and the latent heat of vaporization corresponding to the local saturation at the position z, respectively.

The two-phase frictional pressure drop was estimated according to the following equation:

$$\Delta p_{ts} = \Delta p_{measured} - \Delta p_{1\phi} - \Delta p_{2\phi,ph} - \Delta p_{acc} \tag{2}$$

where $\Delta p_{measured}$ is the measured pressure drop by the differential pressure transducer, $\Delta p_{1\phi}$ is the single-phase pressure drop, $\Delta p_{2\phi,ph}$ is the two-phase pressure drop over the pre-heater length and Δp_{acc} is the accelerational pressure drop comprising the two-phase flow region of the pre-heater and the test section. Δp_{acc} was estimated using the method to predict the superficial void fraction recently proposed by Kanizawa and Ribatski (2015). The two-phase pressure drop along the pre-heater, was estimated through an iterative process as the sum of the pressure drop over 100 discrete elements comprising the beginning of the saturated region at the pre-heater, corresponding the end of the single-phase flow region ($x=0$), and the end of the pre-heater. The frictional pressure drop parcel along each discrete element was calculated according to the predictive method developed by Sempértegui-Tapia and Ribatski (2016).

The local heat transfer coefficients were calculated according to the Newton's cooling law as follows:

$$h(z) = \frac{\phi_{ts}}{T_w(z) - T_{sat}(z)} \tag{3}$$

where $T_w(z)$ is the inner wall temperature estimated according to Fourier's law based on the outer wall temperatures measurements. $T_{sat}(z)$ is the local saturation temperature of the refrigerant, which is estimated using the two-phase frictional pressure drop method developed by Sempértegui-Tapia and Ribatski (2016).

2.3 Experimental validation and uncertainties

Experimental tests for single-phase flow were previously performed in order to assure the accuracy of the measurements and evaluate the heat losses. Figure 2a illustrates the single-phase pressure drop results. As shown in this figure, the laminar experimental data for pressure drop agree reasonably well with the theory of laminar flows. For turbulent flow, the experimental data were compared with the method of Blasius and reasonable agreement was found. Figure 2b illustrates the experimental single-phase heat transfer coefficient. As noted in the figure, the laminar theory underpredicts the experimental values. For turbulent flow, the correlation of Gnielinski (1976) agrees reasonably well with the experimental data.

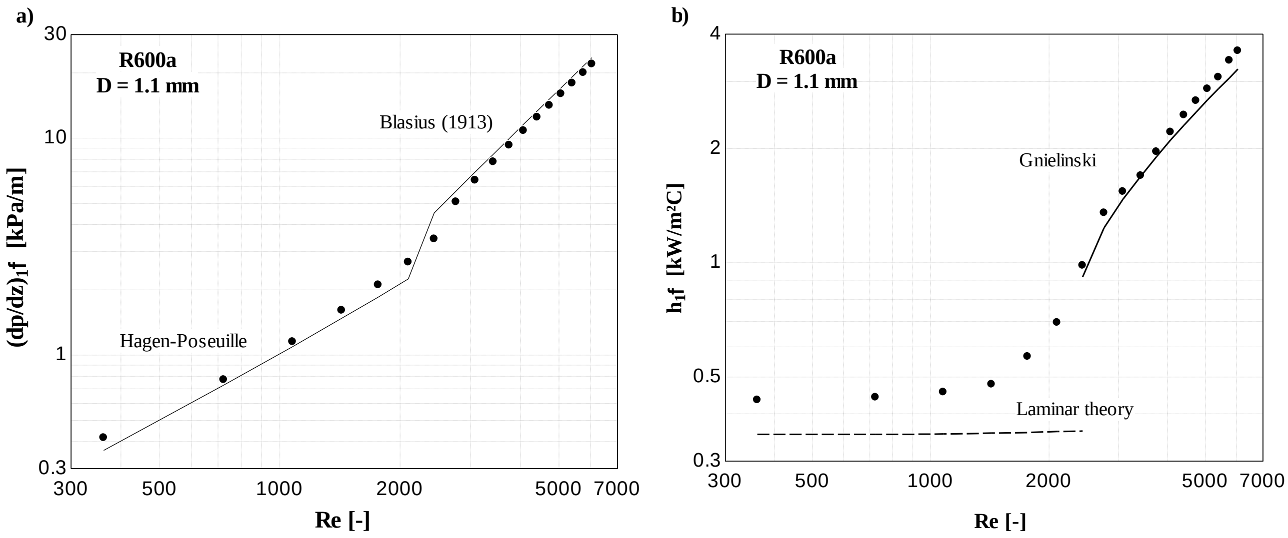

Figure 2. Single-phase flow results, a) pressure drop, b) heat transfer coefficient.

Temperature measurements were calibrated and the temperature uncertainty was evaluated according to the procedure suggested by Abernethy and Thompson (1973). Accounting for all instrument errors, uncertainties for the calculated parameter were estimated using the method of sequential perturbation according to Taylor and Kuyatt (1994). All the experimental uncertainties associated with the sensors and calculated parameters are listed in Table 1.

Table 1. Uncertainty of measured and calculated parameters

Parameter	Uncertainty	Parameter	Uncertainty
D	20 μm	T	0.15 °C
L_{ph}, L_{ts}	1 mm	x	< 5%
p_{in}	4.5 kPa	G	< 2%
Δp	150 Pa	$\left(dp/dz\right)_{2\phi}$	< 10%
P_{ts}, P_{ph}	0.8 %	$h_{2\phi}$	< 10%

3. RESULTS AND DISCUSSIONS

3.1 Frictional presure drop

Figure 3 illustrates the effect of mass velocity and saturation temperature on the frictional pressure drop gradient during adiabatic two-phase flow. As expected, the pressure drop increases with increasing the mass velocity. As observed in Figure 3b, the frictional pressure drop increases approximately 20% with decreasing saturation temperature from 31°C to 41°C.

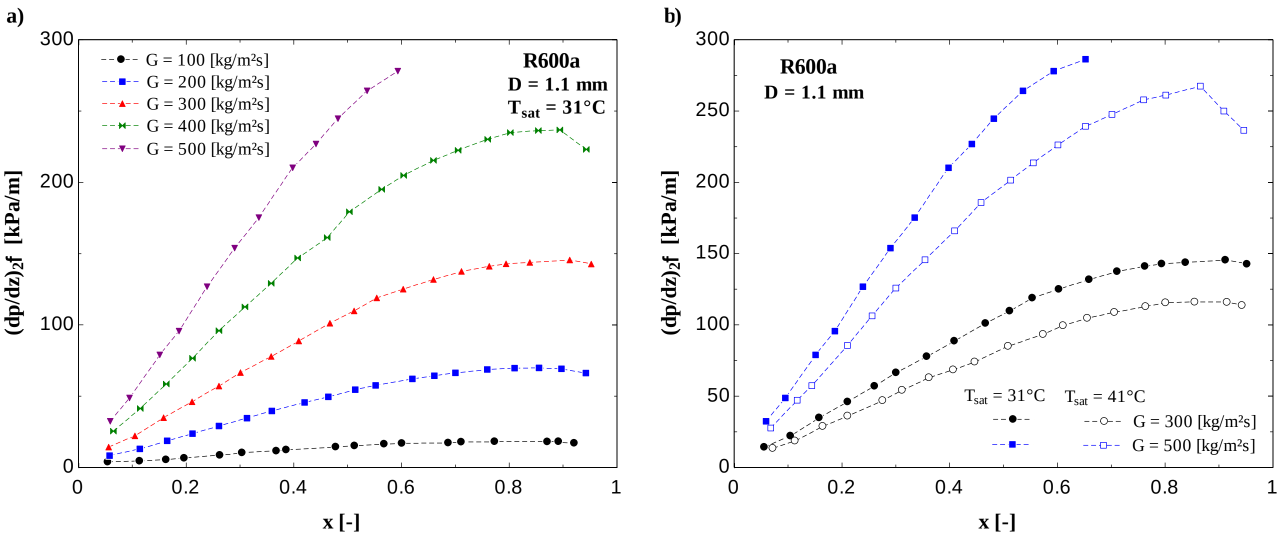

Figure 3. a) Effect of the mass velocity on the frictional pressure drop, b) Effect of the saturation temperature on the frictional pressure drop.

3.2 Heat transfer coefficient

According to Figure 4, the heat transfer coefficient increases with heat flux independently of the saturation temperature and mass velocity range (not shown). It can also be observed that the nucleate boiling effect increases with increasing the heat flux. For a saturation temperature of 31°C, convective effects seems to be predominant over nucleate boiling effect. On the other hand, for a saturation temperature of 41°C, the predominance of nucleate boiling effects becomes prominent under high heat flux conditions.

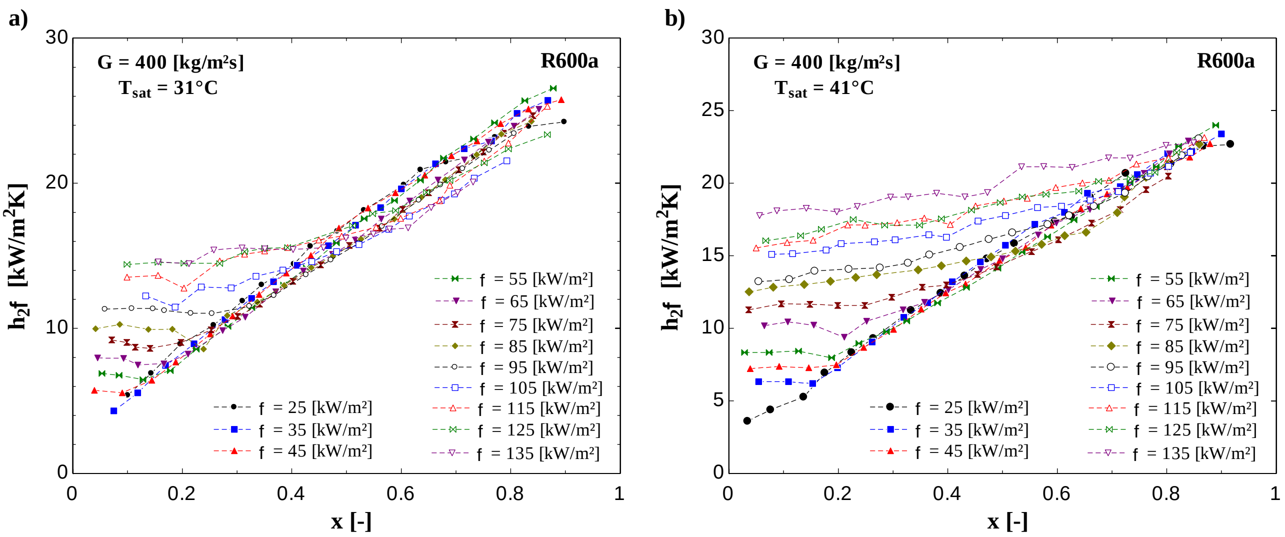

Figure 4. Effect of the heat flux on the heat transfer coefficient.

Figure 5a illustrates the effect of mass velocity and the saturation temperature on the heat transfer coefficient. According to this figure, the HTC increases with increasing mass velocity at intermediary and high vapor qualities. For low vapor qualities, the nucleate boiling effect is dominant; hence, it can be observed an almost constant heat transfer coefficient with increasing the mass velocity. According to Figure 5b, the heat transfer coefficient increases with increasing saturation temperature at low vapor qualities, while decreases for high vapor qualities. The vapor quality of this transitional behavior of the HTC with the saturation temperature seems related to the heat flux. The higher the heat flux, the sooner the transition happens.

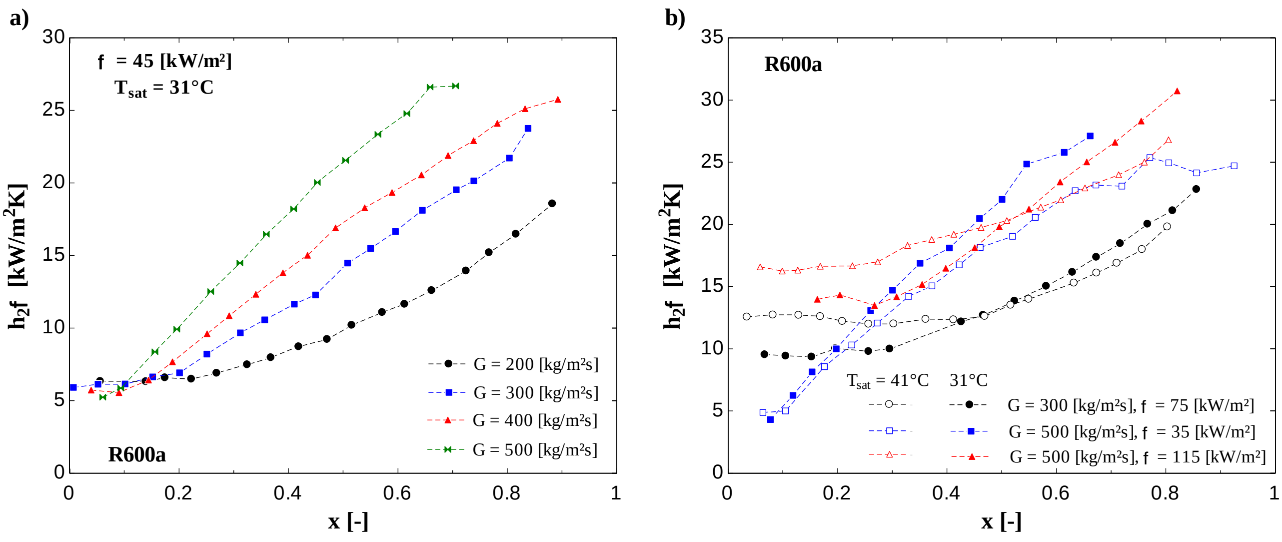

Figure 5. a) Effect of the mass velocity on the HTC, b) Effect of the saturation temperature on the HTC.

4. ASSESEMENT OF PREDICTIVE METHODS

4.1 Frictional pressure drop

Table 2 provides the statistical parameters resulting from the comparison between ten frictional pressure drop predictive methods from literature and the experimental database gathered in the present study.

Table 2. Statistical parameters resulting from the comparison of the experimental data with pressure drop methods.*

Author(s)	MAE	MRE	$\eta_{30\%}$	$\eta_{20\%}$
Hom.–Cicchitti et al. (1960)	28.1%	-28.1%	52.2%	16,7%
Friedel (1979)	21.3%	-10.3%	85.5%	43.3%
Müller-Steinhagen and Heck (1986)	**20.1%**	-20.0%	**98.9%**	41.4%
Cioncolini et al. (2009)	49.1%	-48.6%	25.8%	10.8%
Sun and Mishima (2009a)	29.5%	29.5%	39.2%	21.0%
Zhang et al. (2001)	56.4%	-56.4%	0.5%	0.0%
Li and Wu (2011)	20.4%	**1.3%**	79%	**68.8%**
Kim and Mudawar (2012)	26.6%	-26.0	60.8%	14.5%
Del Col et al. (2013)	**11.3%**	**-5.0%**	**97.0%**	**94.1%**
Sempértegui-Tapia and Ribatski (2016)	**9.3%**	**-6.0%**	**99.0%**	**97.8%**

***Bold numbers indicate the three best results for each statistical parameter.**

As observed in the table, the method developed by Sempértegui-Tapia and Ribatski (2016), which is based on the method of Müller-Steinhagen and Heck (1986), provided the lowest MAE (9.3%) and the highest parcel of the data predicted within error bands of ±30% and ±20%. The method of Del Col et al. (2013) also provided accurate results providing a MAE and $\eta_{30\%}$ of 11.3% and 97%, respectively. Müller-Steinhagen and Heck (1986) and Friedel (1979) presents reasonable agreement with the present database, predicting 98.9% and 85.5% of the data within an error band of ±30%, respectively. However, the parcel of the data predicted within the ±20% drop off more than 50% for both methods. Li and Wu (2011) worked relatively well for the present database, obtaining a MAE of 20.4%. Sun and Mishima (2009a) and Kim and Mudawar (2012) didn't work well for the present experimental database. This result is explained by the fact that neither Sun and Mishima (2009a) nor Kim and Mudawar (2012) comprises R600a in the database used to develop their methods. The homogeneous model with the two-phase viscosity given by Cicchitti et al. (1960), Cioncolini et al. (2009) and Zhang et al. (2001) provided MAE's higher than 28% and $\eta_{30\%}$ less than 60%.

Figure 6a illustrates a comparisons between the experimental database and the four methods that provided the best predictions. According to this figure, the methods of Sempértegui-Tapia and Ribatski (2016) and Del Col et al. (2013) predicted almost all the data for pressure drop gradients less than 80 kPa/m within an error band of ±20%. For pressure drop gradients higher than 80 kPa/m, although providing reasonable predictions, the methods tend to underpredict the experimental results. Friedel (1979) overpredicts the experimental data for pressure drop gradients less than 20 kPa/m and underpredicts for higher values. Müller-Steinhagen and Heck (1986) underpredicts the experimental data independent of the pressure gradient range, resulting a MRE of -20%, as shown in Table 2. Figure 7b displays the evolution of the pressure drop gradient with the quality vapor. According to the figure, the method by Sempértegui-Tapia and Ribatski (2016) captures the experimental trend of the pressure drop gradient.

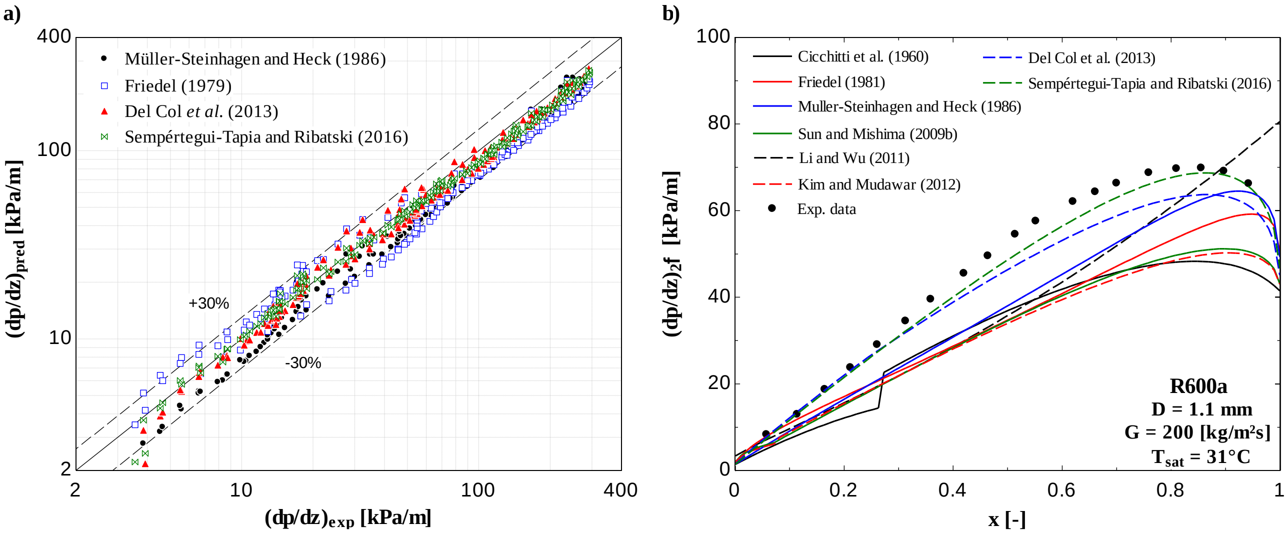

Figure 6. a) Comparison of the experimental database with the four best prediction methods, b) Comparison of frictional pressure drop trends according to predictive methods and the experimental data.

4.2 Heat transfer coefficient

Table 3 presents the statistical parameters resulting from the comparison of the experimental data with selected HTC predictive methods.

Table 3. Statistical parameters resulting from the comparison of the experimental data with the predictive methods.*

Author(s)	MAE	MRE	$\eta_{30\%}$	$\eta_{20\%}$
Liu and Winterton (1991)	**22.2%**	**-16.3%**	**69.4%**	**51.4%**
Tran et al. (1996)	55.7%	-55.7%	4.3%	1.0%
Kew and Cornwell (1997)	24.2%	**-13.6%**	**67.1%**	**53.4%**
Zhang et al. (2004)	31.4%	-27.9%	50.5%	29.9%
Thome et al. (2004)	41.2%	-36.0%	36.1%	25.2%
Saitoh et al. (2007)	31.5%	-26.0%	54.5%	38.9%
Sun and Mishima (2009b)	27.3%	-17.4%	59.1%	45.4%
Bertsch et al. (2009)	39.5%	-36.6%	39.0%	26.5%
Kim and Mudawar (2013a)	**23.8%**	-16.1%	62.9%	45.7%
Kanizawa et al. (2015)	**22.8%**	**-11.6%**	**65.7%**	**48.7%**

***Bold numbers indicate the three best results for each statistical parameter.**

According to Table 3, the methods of Kanizawa et al. (2015), Liu and Winterton (1991) and Kew and Corwell (1997) provided the best predictions of the experimental data, predicting 65.7, 69.4 and 67.1% within an error band of ±30%, respectively. It is important to mention that despite providing the best predictions, a $\eta_{30\%}$ around 65-70% is not satisfactory. The methods of Kim and Mudawar (2013a) and Sun and Mishima (2009b) provided values of $\eta_{30\%}$ around 60%. The methods of Tran et al. (1996), Zhan et al. (2004), Saitoh et al. (2007) and Bertsch et al. (2009) were unable of providing reasonable predictions of the experimental database.

Figure 7a illustrates comparisons between the experimental database and the four best prediction methods according to the results of Table 3. According to this figure, the methods of Kew and Cornwell (1997), Liu and Winterton (1991), Kim and Mudawar (2013a) and Kanizawa et al. (2015) mostly underpredict the experimental data, especially for HTC values higher than 10 kW/m²K.

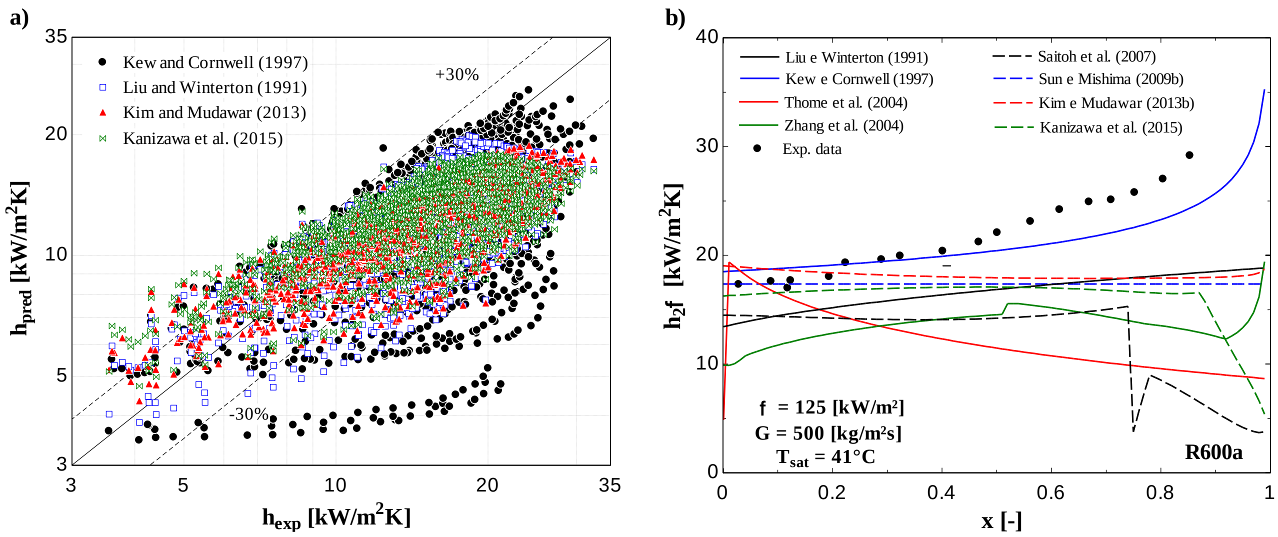

Figure 7. Comparisons of the experimental database and predictive methods from literature: a) the best four methods according to Table 3 (1991) and b) HTC trends.

Figure 7b displays the evolution of the HTC with vapor quality according to the predictive methods and the experimental data. According to this figure, the method of Kew and Corwell (1997) captures reasonably well the experimental trend. The methods by Kanizawa et al. (2015), Kim and Mudawar (2013a) and Sun e Mishima (2009b) capture relatively well the trend of the heat transfer coefficient for vapor qualities lower than 0.4, corresponding to a condition with the predominance of nucleate boiling effects. For conditions under the predominance of convective effects none of the methods is able of reasonably predict the HTC trend. It should also be noted that the dryout incipience predicted by Kim and Mudawar (2013b), indicated by the downward arrow in the Figure 7b, occurs for vapor qualities much lower than the values obtained in the present study.

5. CONCLUSIONS

The following remarks summarize the conclusions of the present investigation:

- The pressure drop gradient increases with increasing the mass velocity and decreases with increasing the saturation temperature;
- In general, the dominant heat transfer mechanism is mostly due to convective effects, however, under some heat flux conditions characterized by high saturation temperatures the nucleate boiling effect becomes dominant;
- The HTC increases with mass velocity at high vapor qualities. The heat transfer coefficient increases with increasing the saturation temperature at low vapor qualities and decreases for high vapor qualities;
- The methods developed by Sempértegui-Tapia and Ribatski (2016) and Del Col et al. (2013) provided good predictions of the frictional pressure drop data obtained in the present study;
- The methods developed by Liu and Winterton (1991) and Kanizawa et al. (2015) provided the best predictions of the present database. However, both methods mostly underpredict the experimental data for flows dominated by convective effects. None of the methods evaluated in this study provides satisfactory predictions of the HTC trends;

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