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# Experimental Investigation of Pressure Drop during Two-Phase Flow of R1234yf in Smooth Horizontal Tubes with Internal Diameters of 3.2 mm to 8.0 mm

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**Abstract**

In this study, the two-phase pressure drop of R1234yf refrigerant (as an alternative to R134a) in a smooth horizontal tube is investigated. Tests are carried out and under several conditions of vapor quality (0% to 100%), mass velocity (200, 300 and 400 kg/m²s), heat flux (0, 7 and 14 kW/m²) and evaporation temperatures (20°C and 30°C). The internal diameters tested are 3.2 mm, 4.8 mm, 6.4 mm and 8.0 mm. A comparison of pressure drop between R1234yf and R134a is also carried out. The results demonstrate that the two-phase pressure gradient of R1234yf is approximately 20% lower than that of R134a. The experimental data is compared with 19 correlations from the literature. From this comparison, it is concluded that the experimental data are in accordance with the correlation proposed by Xu and Fang with a mean absolute error of 20% and with approximately 65% of the data predicted within a ± 30% error band.

**Keywords:** R1234yf; low GWP refrigerant; pressure drop; two-phase flow; smooth horizontal tubes; R134a

**Nomenclature**

| | |
|---|---|
| **Greek Symbols** | |
| $\alpha$ | void fraction [-] |
| $\Delta P$ | pressure drop [kPa] |
| $\eta$ | efficiency [-] |
| $\pi$ | evaporator inclination [rad] |
| $\sigma$ | surface tension [N.m$^{-1}$] |
| $\rho$ | density [kg.m$^{-3}$] |
| $\mu$ | dynamic viscosity [Pa.s] |
| **Latin Symbols** | |
| $Co$ | Confinement number [-] |
| $D$ | diameter [m] |
| $dP/dz$ | pressure gradient [kPa.m$^{-1}$] |
| $dx_{flash}$ | vapor quality variation by flashing [-] |
| $f$ | friction factor [-] |
| Fr | Froude number [-] |
| $G$ | mass velocity [kg.m$^{-2}$.s$^{-1}$] |
| $g$ | gravitational acceleration [m.s$^{-2}$] |
| $i$ | enthalpy [J.kg$^{-1}$] |
| $I$ | electrical current [A] |
| $L$ | length [m] |
| $P$ | pressure [kPa] |
| $Re$ | Reynolds number |
| La | Laplace number [-] |
| $\dot{m}$ | mass flow [kg.s$^{-1}$] |
| $n$ | number of samples |
| $\dot{Q}$ | heat rate [W] |
| $q"$ | heat flux [kW.m$^{-2}$] |
| $T_{sat}$ | saturation temperature [°C] |
| $V$ | electric voltage [V] |
| $v$ | specific volume [m³.kg$^{-1}$] |
| $x$ | vapor quality [-] |
| **subscripts** | |
| $exp$ | experimental |
| $f$ | friction pressure drop |
| $in$ | inlet |
| $m$ | momentum |
| $l$ | liquid |
| $lo$ | only liquid in the tube |
| $lv$ | difference between liquid and vapor |
| $out$ | outlet |
| $pred$ | predicted |
| $ph$ | pre-heater |
| $tp$ | two-phase mixture |
| $ts$ | test section |
| $v$ | vapor |
| $vo$ | only vapor in the tube |

## 1. Introduction

Hydrofluorocarbons (HFCs) do not deplete the ozone layer because their molecules contain no chlorine. However, they have a high global warming potential (GWP) (Lu et al., 2013), meaning that they contribute to the greenhouse effect. Due to concerns about climate change, use of fluids with high GWP has been restricted. The European Directive F-Gas established limits for GWP of less than 750 for residential heating systems and less than 150 for automotive air conditioning systems (Saitoh et al., 2011).

One of the most used HFCs in refrigeration systems is R134a (Mclinden et al., 2014), which has a GWP of 1430 (Wodzisz, 2015). The main alternatives for replacing this fluid are natural refrigerants such as ammonia and carbon dioxide; hydrocarbons (HC) such as R290 and R600a; HFCs with low GWP, such as R32 and R152a; and hydrofluoroolefins (HFOs), specifically R1234yf and R1234ze(E) (Spatz and Minor, 2008). Among all these options, R1234yf has GWP of 1 (BOC, 2015), and it has been proposed as the main substitute for R134a in automotive air conditioning systems because the thermophysical properties of the two refrigerants are similar (Cho and Park, 2016).

Promising results were found in the literature concerning the use of the R1234yf as substitute for the R134a in refrigeration systems results (Lee and Jung, 2012; Qi, 2013, 2015). Thus, R1234yf has been widely used in new cars in Europe since 2017. In addition, with the objective of finding new applications in which R1234yf can replace R134a, many works have been published about systems such as domestic refrigerators (Belman-Flores et al., 2017) and residential heat pumps (Botticella et al., 2017; Nawaz et al., 2017). These studies show that R1234yf provides satisfactory performance and can be used in systems that work with R134a without requiring further modifications.

With the need to replace refrigerants, it is essential to identify the heat transfer coefficient, void fraction and pressure drop in the two-phase flow of boiling and condensation, since the predictions of these values are indispensable for the optimization of refrigeration cycle components (Greco, 2008). For this reason, in the last eight years, heat transfer coefficient and pressure drop (Anwar et al., 2015; Choi et al., 2014; Del Col et al., 2013; Del Col et al., 2010; Illán-Gómez et al., 2015; Li et al., 2012; Lu et al., 2013; Mortada et al., 2012; Padilla et al., 2011; Saitoh et al., 2011; Sempértegui-Tapia and Ribatski, 2017a, b; Wang et al., 2012) and critical heat flux (Mastrullo et al., 2016; Mastrullo et al., 2017) have been evaluated under conditions of two-phase flow in smooth tubes. Most of the above-mentioned papers deal with mini and microchannels (internal diameter < 3mm, according to Kandlikar and Grande (2003)), which are typical in automotive air conditioning system applications.

For R1234yf in two-phase flow, pressure drop has been studied less than heat transfer coefficient. Table 1 summarizes the published literature on two-phase pressure drop in smooth and horizontal tubes. It indicates that there are few studies, the majority are partial or limited, and some of the test results conflict with each other, even under similar conditions. Hence, more studies are needed to clarify the mass transfer mechanism in this refrigerant, with test sections that measure the two-phase pressure drop in the absence of confounding effects such as single-phase flow and change of cross-sectional area in the tube.

Research continues into the use of R1234yf in other applications such as freezer, refrigerator and residential air conditioning, which commonly use macrochannels or conventional channels. In this work, experimental data on pressure drop in two-phase flow in conventional horizontal tubes (diameters > 3 mm) for R1234yf is presented.

Table 1. Studies on two-phase pressure drop for R1234yf in smooth tubes.

| **Author** | $D$ (mm) | $G$ $(kg/m^2s)$ | $T_{sat}$ $(°C)$ | **Comments** |
|---|---|---|---|---|
| Del Col et al. (2010) | 0.96 | 400<br>600<br>800 | 40 | The focus of this study was on heat transfer during condensation, but the pressure drop in adiabatic two-phase flow was measured briefly. The test section was a copper 0.96mm diameter tube soldered between two stainless steel 0.76mm diameter tubes. Therefore, the pressure drop was the sum of the frictional pressure drop in the three tubes and the pressure losses due to change of cross-sectional area (from 0.76 to 0.96mm diameter and vice-versa). The results did not include a deep analysis. However, a graph of the results showed an increase in pressure drop as mass velocity increased. |
| Padilla et al. (2011) | 7.90<br>10.85 | 300<br>400<br>500<br>600<br>750 | 10<br>15<br>20 | The maximum frictional pressure drop in adiabatic two-phase flow occurs in the annular flow regime on the vapor quality from 77% to 92%. The experimental data was compared against selected correlations found in literature. The best accuracy was given by Müller-Steinhagen and Heck (1986) with around 90% of the data predicted within a ± 30% error band. |
| Saitoh et al. (2011) | 2 | 100<br>200<br>400 | 15 | This study focused on boiling heat transfer coefficient. The pressure drop during flow boiling with heat fluxes of 6, 12, and 24 kW/m² was measured. This study had few experimental data (< 15 points). The experimental results were not analysed and were only compared with the Chisholm (1967) correlation. Based on the qualitative analysis, the authors concluded that the measured pressure drops agreed well with the mentioned correlation. |
| Wang et al. (2012) | 4 | 100<br>200<br>300<br>400 | 45 | The authors studied the condensation heat transfer and pressure drop. Only 12 experimental measurements about pressure drop were reported. A comparison with three correlations was presented. The Huang et al. (2010) and Haraguchi et al. (1994) correlations (developed for condensation flow) had average deviations of 38.2% and 27.3%, respectively. The Chisholm (1967) correlation had an average deviation of 29.8%. |
| Lu et al. (2013) | 3.9 | 200 | 10 | The main objective was to evaluate the performance of convective boiling heat transfer. However, the adiabatic two-phase pressure drop was also measured |

| Author | $D$ (mm) | $G$ $(kg/m^2s)$ | $T_{sat}$ $(°C)$ | Comments |
|---|---|---|---|---|
| | | 300<br>400<br>500 | | for R1234yf and R134a. The pressure drops for R134a were approximately 5–15% higher than for R1234yf. The effects of mass velocity and vapor quality were reported, but were not analysed. |
| Anwar et al. (2015) | 1.6 | 400<br>500 | 27<br>32 | The work highlighted flow boiling heat transfer. However, data on the flow boiling pressure drop was collected and included single-phase, two-phase and end effects (contraction at inlet and expansion at outlet). The heat flux used was reported. The results showed that the frictional pressure drop decreased with increasing saturation temperature and decreasing mass velocity and vapor quality. High values were reported for frictional pressure drop for R134a compared to R1234yf. The experimental data was compared with three correlations. The Grönnerud (1979) and Mishima and Hibiki (1996) correlations had the best average deviations at 25.55% and 28.2% respectively. |
| Sempértegui-Tapia and Ribatski (2017b) | 1.1 | 200<br>400 | 31<br>41 | The flow boiling pressure drop data was collected for the refrigerants R134a, R1234ze(E), R1234yf and R600a by a differential pressure sensor included single-phase and two-phase pressure drops. 10% of all experimental data was obtained for R1234yf. The results revealed that the pressure gradient peak moves to lower vapor qualities with increasing mass velocity and decreasing saturation temperature. The experimental data was compared with ten known correlations. Friedel (1979) correlation had the best average deviation (19.9%) for R1234yf. A new predictive method was developed for data collected (microchannels) with average deviation of 10.7% for R1234yf. |
| Yang et al. (2018) | 4 | 200<br>400<br>800<br>1200 | 15 | The two-phase pressure drop during flow boiling with heat fluxes between 10 to 50 kW/m² was measured. The experimental data was compared with two correlations. The Zhang and Webb (2001) correlation performed better than the Friedel (1979) correlation, with 82% and 44% of the data within the range of ± 20%, respectively. |

## 2. Experimental method

### 2.1. Experimental setup

Figure 1 presents a schematic of the experimental setup used to measure the pressure drop of R1234yf. The refrigerant loop is made of a copper tube with an internal diameter of 4.8 mm and contains a self-lubricating oil-free gear micropump, which delivers subcooled refrigerant to the heater. The refrigerant is preheated and partially evaporated in the heater to the desired vapor quality. The fluid passes through the test section and is condensed and subcooled in the condenser. Thus, the refrigerant returns to the micropump as a subcooled liquid. To reduce the heat loss to surroundings, the entire device was insulated. The refrigerant flow rate was controlled by adjusting the frequency of the magnetic gear pump. The vapor quality at the inlet of the test section was adjusted by the amount of heat supplied to the refrigerant in the pre-heater. The pressure was controlled in the condenser by the temperature of the water loop, which can be chosen by the operator. When the performance of the condenser was poor, the pressure was controlled by the amount of refrigerant in the circuit using a sub-tank with a water bath. The temperature of the two water loops that are shown in Fig. 1 were controlled by an auxiliary refrigeration system with a PID controller. More details on the experimental apparatus are found in Garcia et al. (2017).

Thermocouples and pressure transducers were installed in various places in the refrigerant loop to ascertain the fluid conditions. The accuracy of the T-type thermocouples was within ± 0.5°C (calibrated by Laboratório de Metrologia ECIL). The pressure was measured using manometers (Novus, model NP460) with an accuracy of ±0.02 bar (2.0 kPa). The facility also included a turbine-type flow meter (Kobold, model DPM-1110) with accuracy of 2%. The electrical input power in the pre-heater was measured using a voltmeter and amperemeter with accuracy of ± 1.6%.

The propagation of uncertainty was computed for mass velocity, mass flux and vapor quality. The method for determining these propagations is described by Taylor and Kuyatt (1994) and values were calculated with EES software. Uncertainty values of ± 3.0%, ± 2.8% and ± 3.3% were observed for mass velocities of 200, 300 and 400 kg/m²s, respectively. The uncertainty values for heat fluxes of 7 and 14 kW/m² were ± 2.0% and ± 1.7%, respectively. The uncertainty for vapor quality was more varied, at ± 8% for $x = 0.1$ and decreasing to ± 5% for $x = 1$.

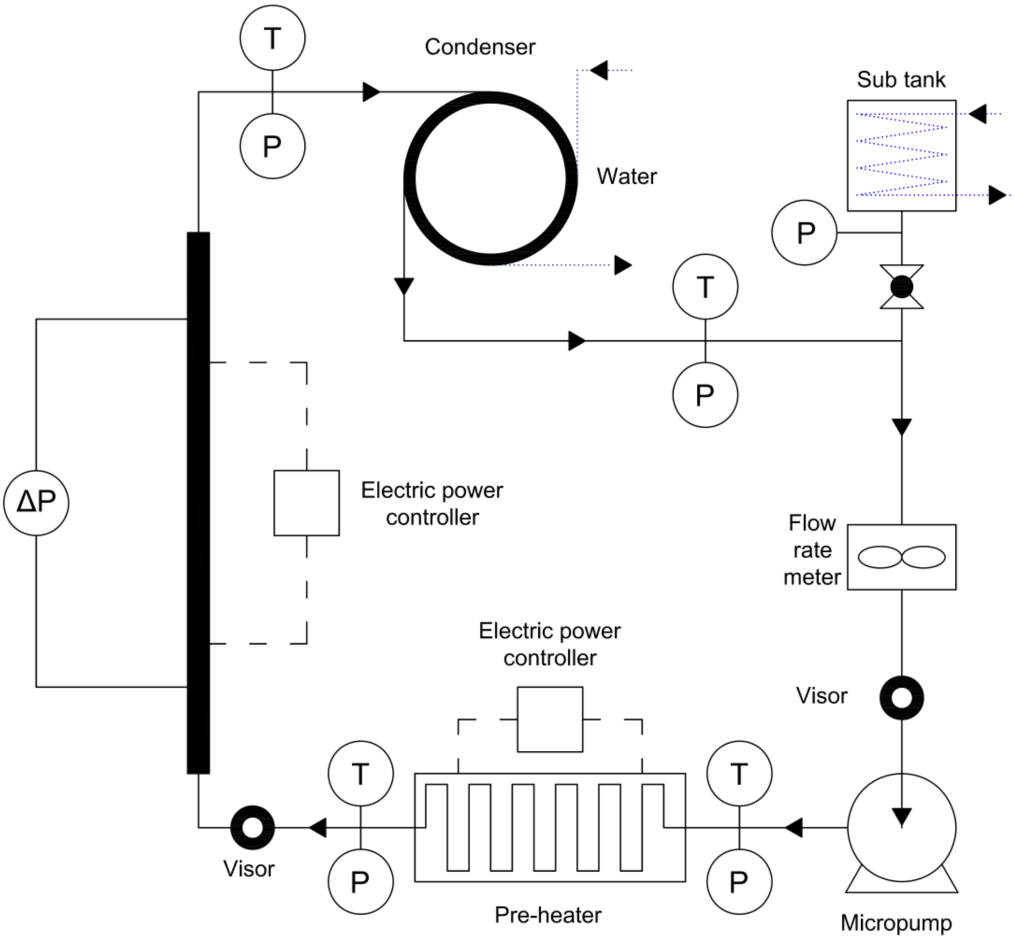


Figure 1. Schematic of experimental setup.

### 2.2. Test section

A schematic of the test section is presented in Fig. 2. A tube of 0.5 m (> 50D) length is located upstream of the test section to achieve a fully-developed flow condition. A differential pressure transducer (Microcyber, model NCS-PT105) with an accuracy of 0.028 kPa is used to measure the pressure drop across the tube of 1m±1mm length. A tube of length 0.5 m is downstream of the test section. Inlet differential pressure taps of 0.5 mm diameter were installed in the wall of the tube to minimise the influence of this measurement on flow. The test section was made of copper tubes with internal diameters of 3.2 mm, 4.8 mm, 6.4 mm and 8.0 mm, which correspond to the conventional copper tubes of 3/16 in, 1/4 in, 5/16 in and 3/8 in commonly used in the refrigeration and air conditioning fields. The test section had an electrical resistance wrapped around the exterior surface, providing a maximum heat flux of 15 kW/m². In order to check the measurements made with the present experimental device, single-phase pressure drop measurements of R134a were carried out. The experimental single-phase pressure drop shown in the Fig. 3

was compared to the Fang et al. (2011) correlation for smooth tubes, producing a mean error less than ± 7% for 6000 < Re < 19000. Thus, the experimental device was validated.

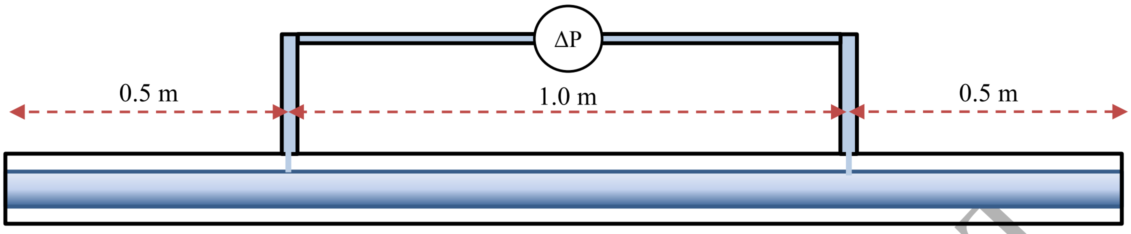


Figure 2. Test section schematic.

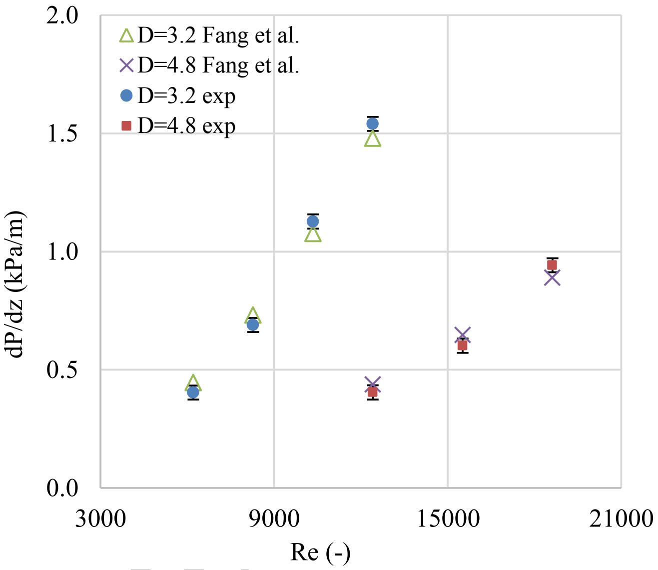


Figure 3. Single-phase pressure drop measurements.

### 2.3. Data reduction

The pressure gradient is defined as the ratio of the pressure drops in smooth horizontal tubes to the length, $L = 1\ m$:

$$\frac{dP}{dz} = \frac{\Delta P}{L} \tag{1}$$

The mass velocity is calculated as the ratio between the measured mass flow rate ($\dot{m}$) and the internal cross-sectional area of the tube (which is a function of internal diameter ($D$)), according to the following equation:

$$G = \frac{4\dot{m}}{\pi D^2} \tag{2}$$

The vapor quality at the outlet of the pre-heater is calculated from energy balance over the pre-heater according to the following equation:

$$x_{ph,out} = \frac{1}{i_{lv,ph}}\left(\frac{\dot{Q}_{ph}}{\dot{m}} - \left(i_{l,ph} - i_{ph,in}\right)\right) \tag{3}$$

where $i_{ph,in}$ is the enthalpy of the liquid at the inlet of the pre-heater, $i_l$ and $i_{lv}$ are the enthalpy of the saturated liquid and the latent heat of vaporization corresponding to the saturation temperature at the outlet of the pre-heater (inlet of the test section). The $\dot{Q}_{ph}$ is the heat rate in the pre-heater:

$$\dot{Q}_{ph} = \eta_{ph} V_{ph} I_{ph} \tag{4}$$

where $V_{ph}$ and $I_{ph}$ are the voltage and electrical current applied in the pre-heater. $\eta_{ph}$ is the efficiency between heat and electrical power in the pre-heater, which for the single-phase experimental tests has a value of 0.93.

In the experiments, the maximum pressure drop measured from the inlet to outlet of the pre-heater (where $x_{ph,out}$ is calculated) can reach up to 0.9 bar. This pressure drop is not negligible and provokes a so-called flashing (the increase of the vapor quality due to expansion) and a temperature difference. The flashing effect, $dx_{flash}$ may be calculated for all data using the relation developed by Revellin et al. (2009). The vapor quality at the inlet of the test section was determined as:

$$x_{ts,in} = x_{ph,out} + dx_{flash} \tag{5}$$

The majority of two-phase pressure drop values were directly obtained from the differential pressure transducers under adiabatic conditions; thus, $x_{ts,in} = x_{ts,out}$. However, diabatic conditions also were tested. In the test section, the applied heat flux is calculated by:

$$q" = \frac{\eta_{ts} V_{ts} I_{ts}}{\pi D L} \tag{6}$$

where $V_{ts}$ and $I_{st}$ are the voltage and electrical current applied in the test section. $\eta_{ts}$ is the efficiency between the heat rate and electrical power in the test section. Single-phase experimental tests determined that $\eta_{st} = 0.97$.

The vapor quality at the outlet of the test section is determined as:

$$x_{ts,out} = \frac{\eta_{ts} V_{ts} I_{ts}}{\dot{m}\, i_{lv,ts,in}} + x_{ts,in} \tag{7}$$

The average vapor quality along the test section is estimated by the following equation:

$$\bar{x}_{ts} = \frac{x_{ts,in} + x_{ts,out}}{2} \tag{8}$$

The property data for the refrigerant is calculated from EES (developed by F-CHART).

### 2.4. Experimental conditions

Table 2 summarizes the experimental conditions of the present work along with the corresponding uncertainties. The three mass velocities and two saturation temperatures were selected in accordance with the capacity of the experimental device to avoid instabilities in the micropump. The majority of pressure drop measurements were performed for R1234yf under adiabatic conditions. In addition, tests with heat fluxes of 7 and 14 kW/m² were carried out for R1234yf with a tube diameter of 4.8 mm, mass velocity of 300 kg/m²s and saturation temperature of 20°C to study the effect of diabatic conditions on the two-phase pressure drop. The final group of tests were performed with R134a refrigerant under adiabatic conditions, with a tube diameter of 4.8 mm, mass velocity of 300 kg/m²s and saturation temperature of 20°C, to compare the two-phase pressure drop between R1234yf and R134a.

**Table 2.** Experimental conditions evaluated in the present study.

| **Fluid** | $\boldsymbol{D\ (mm)}$<br>$\pm 0.05$ mm | $\boldsymbol{T_{sat}(°C)}$<br>$\pm 0.5°C$ | $\boldsymbol{G(kg/m^2s)}$<br>$\pm 3.3$ % | $\boldsymbol{q"(kW/m^2)}$<br>$\pm 2.0\%$ |
|---|---|---|---|---|
| R1234yf | 3.2<br>4.8<br>6.4<br>8.0 | 20<br>20, 30<br>20, 30<br>20, 30 | 200, 300, 400<br>200, 300, 400<br>200, 300<br>200, 300 | Adiabatic conditions |
| R1234yf | 4.8<br>6.4 | 20<br>20 | 300<br>200 | 7, 14<br>7 |
| R134a | 4.8 | 20 | 200, 300, 400 | Adiabatic conditions |

## 3. Two-phase pressure gradient prediction methods

The pressure gradient during convective boiling inside horizontal tubes is the sum of the frictional and momentum pressure drop:

$$\left(\frac{dp}{dz}\right)_{tp} = \left(\frac{dp}{dz}\right)_{f} + \left(\frac{dp}{dz}\right)_{m} \tag{9}$$

The momentum pressure gradient $\left(\frac{dp}{dz}\right)_{m}$ results from the change in momentum of both phases (it outcomes from the change of the mass and velocity in each phase caused by evaporation), which can be calculated by the following equation:

$$\left(\frac{dp}{dz}\right)_{m} = \frac{G^2}{L}\left\{\left[\frac{x^2}{\alpha\rho_v} + \frac{(1-x)^2}{(1-\alpha)\rho_l}\right]_{out} - \left[\frac{x^2}{\alpha\rho_v} + \frac{(1-x)^2}{(1-\alpha)\rho_l}\right]_{in}\right\} \tag{10}$$

where the mean void fraction $\alpha$ is calculated by Xu and Fang (2014) correlation (based on R134a data) for horizontal tubes.

$$\alpha = \left[1 + \left(1 + 2\mathrm{Fr}_{lo}{}^{-0.2}\alpha_h{}^{3.5}\right)\left(\frac{1-x}{x}\right)\left(\frac{\rho_v}{\rho_l}\right)\right]^{-1} \quad (11)$$

$$\mathrm{Fr}_{lo} = \frac{G^2}{gD\rho_l{}^2} \quad (12)$$

$$\alpha_h = \left[1 + \left(\frac{1-x}{x}\right)\left(\frac{\rho_v}{\rho_l}\right)\right]^{-1} \quad (13)$$

The frictional pressure gradient $\left(\frac{dp}{dz}\right)_f$ results from the shear stress between the flowing fluid and the tube wall, as well as from the shear stress between the liquid and vapor phases. Due to the complexity of two-phase flow, it is difficult to calculate the frictional pressure analytically. Therefore, it is necessary to develop prediction methods based on experimental results. Thus, the effect of friction is substantial in studies involving the two-phase pressure drop.

$\left(\frac{dp}{dz}\right)_f$ is calculated using three distinct approaches of prediction methods for smooth tubes: the homogeneous, two-phase multipliers, and phenomenological approaches. The correlations selected in this study are known to be valid in adiabatic or convective boiling conditions, horizontal and smooth tubes, and macro-scale diameters. They are presented in section 4.2.

### 3.1. Error of correlations

The correlations are evaluated by the relative deviation (MRD) and the mean absolute relative deviation (MARD).

$$MRD = \frac{1}{n}\sum_{i=1}^{n}\frac{dP/dz(i)_{pred} - dP/dz(i)_{exp}}{dP/dz(i)_{exp}} \quad (14)$$

$$MARD = \frac{1}{n}\sum_{i=1}^{n}\left|\frac{dP/dz(i)_{pred} - dP/dz(i)_{exp}}{dP/dz(i)_{exp}}\right| \quad (15)$$

$n$ is the data number, and the subscript $pred$ and $exp$ are predicted and experimental values, respectively.

## 4. Results and discussion

### 4.1. Two-phase pressure drop

An experimental investigation of two-phase pressure drop for the R1234yf refrigerant was performed. There were a total of 212 sets of the test data under adiabatic conditions and 36 under non-adiabatic conditions. Tests were performed to study the effect of mass velocity, diameter, evaporation temperature, heat flux, and quality on two-phase pressure drop for convective boiling of R1234yf. In addition, the same tests were carried out on the R134a refrigerant for comparison purposes.

The flow patterns in the present experiment are important to understand the effects of experimental conditions on two-phase pressure drop. The map proposed by Wojtan et al. (2005) was used to predict the flow patterns. Figure 4 presents the flow pattern map of R1234yf for minimum and maximum values of tube diameter. As can be seen in Fig. 4, the flow regimes observed at mass velocities of 200, 300 and 400 kg/m²s are slug, intermittent and annular, respectively. In Fig. 4 (a) and Fig. 4 (b), the influence of tube diameter on transition between slug and intermittent flows can be observed. In Fig. 4 (a), the saturation temperature has an influence on the transition line between intermittent and annular flows.

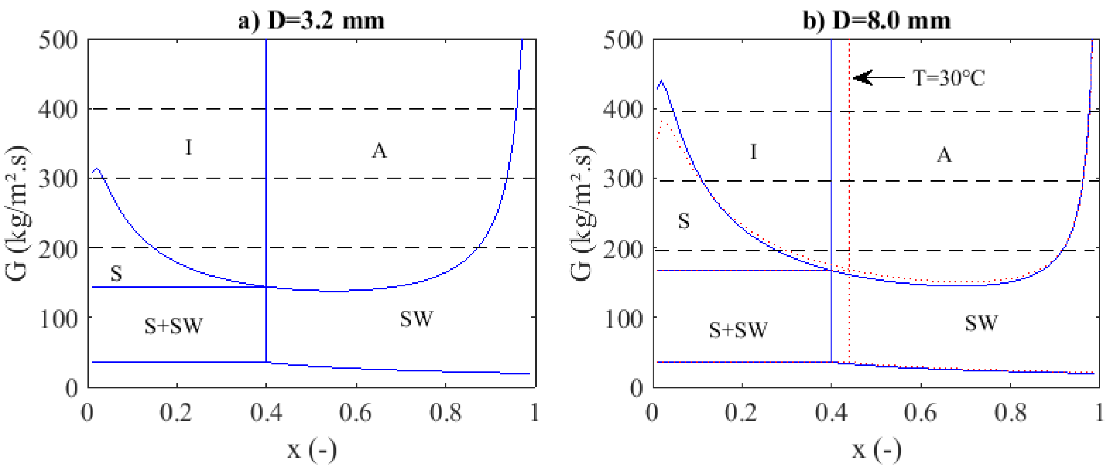


Figure 4. Flow pattern map of Wojtan et al. (2005) (I: intermittent, A: annular, S: slug, SW: stratified-wavy), $T_{sat} = 20°C$ and $q" = 0kW/m^2$.

#### 4.1.1. Effect of mass velocity

Figure 5 shows the effects of mass velocity variation on the two-phase pressure gradient at the saturation temperature of 20°C under adiabatic conditions. For all tube diameters, it is explicitly observed that greater mass velocities cause greater pressure drops. In slug flow, the effect of mass velocity is lower than in the other flow patterns. In intermittent and annular flows (see Fig. 4 as reference), an increase of mass velocity elevates the slip ratio between liquid and vapor velocities, which subsequently increases the interfacial friction; as a result, the pressure drop rises for the same vapor quality. It is important to note that the increase in the (approximate) slope of the pressure drop curves (while x < 0.80) is proportional to the increase in the mass velocity. A pressure gradient peak occurs at a quality of approximately 0.80, known as critical quality; this phenomenon becomes more evident with increasing mass velocity

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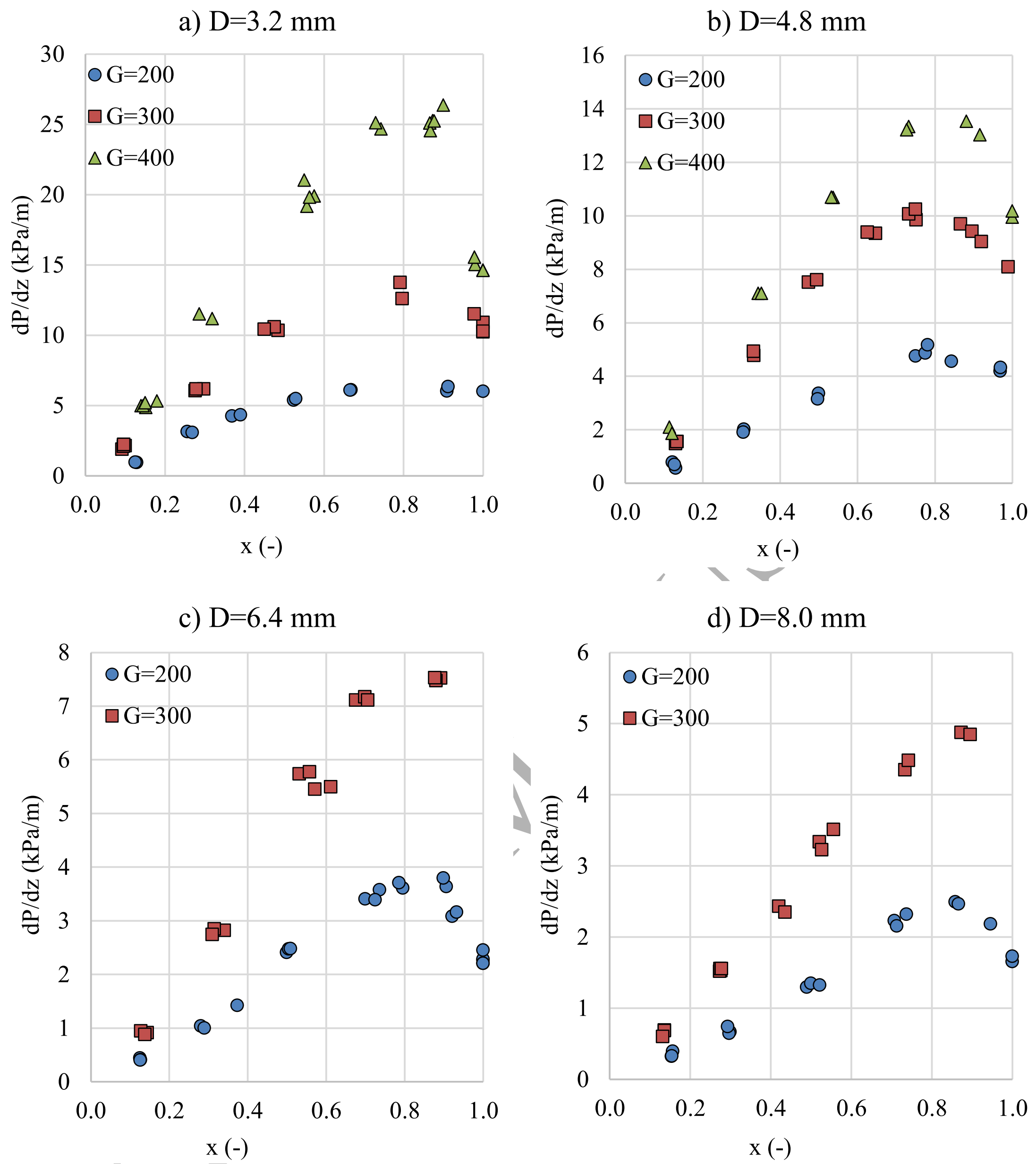


Figure 5. Effect of the mass velocity $G$ (kg/m²s) on the two-phase pressure gradient of R1234yf with $T_{sat} = 20°C$.

#### 4.1.2. Effect of tube diameter

Figure 6 demonstrates the effect of tube diameter on the two-phase pressure gradient for the R1234yf refrigerant with no heat flux in the test section, a saturation temperature of 20°C, and tube diameters greater than 3 mm. As tube diameter decreases, the increase of pressure gradient is higher. Thus, from 4.8 mm to 3.2 mm, 28% increase in pressure gradient occurs, at a vapor quality of 0.8 and mass velocity of 300 kg/m²s; this increase is more than that observed with other changes of diameter in the same conditions. This can be explained by the transition from macrochannels to minichannels. Li and Wu (2010) define this transition as the point where the behavior of the flow begins to deviate from conventional predictions due to the reduction of the diameter. Kandlikar (2010) writes that for larger tube diameters the gravitational forces are dominant; however, when diameter decreases, they are overcome by the inertia and surface tension forces. This means that a quick transition of slug/annular regimes arises at low vapor qualities, which increases the pressure drop. The parameter used to distinguish the macro-to-micro-scale transition is the Confinement number $\left(Co = \sqrt{\sigma/(D^2 g(\rho_l - \rho_v)}\right)$. Ong and Thome (2011) proposed a lower boundary for macro-scale at $Co < 0.34$ (based on visualization of R134a). This critical diameter varies depending on the fluid; for R1234yf, it is estimated to be 2.19 and 2.36 mm for 30 and 20°C, respectively. In conclusion, Fig. 6 shows a sharp increase in pressure gradient with diameters close to macro-to-micro-scale transition, but it should be further studied to understand the full scope of the effects.

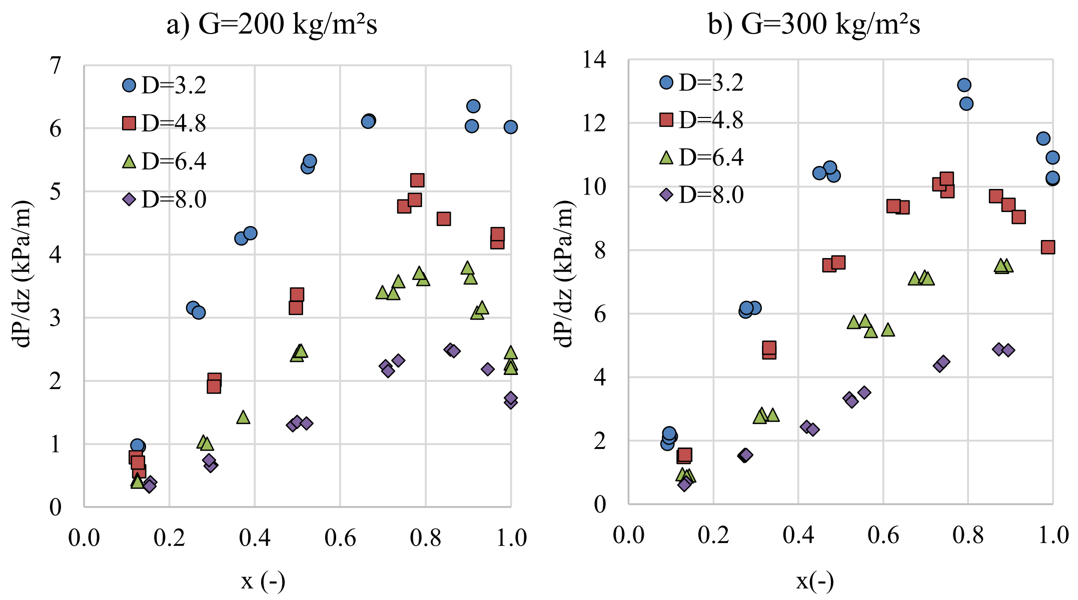


Figure 6. Effect of the tube diameter $D$ (mm) on the two-phase pressure gradient of R1234yf, $T_{sat} = 20°C$.

#### 4.1.3. Effect of saturation temperature

Figure 7 shows effect of variation in evaporation temperature variation on the two-phase pressure gradient for the R1234yf fluid under adiabatic conditions and constant mass velocity for many diameters. The increase in evaporation temperature results in a decrease in the pressure gradient, especially for qualities greater than 0.4. This can be justified by the decrease in liquid/vapor ratios of density and viscosity, approximately 27% and 15% respectively, with increasing saturation temperature. Low density and viscosity ratios cause a decrease of slip ratio (vapor/liquid velocities) and decrease the frictional pressure drop as a result.

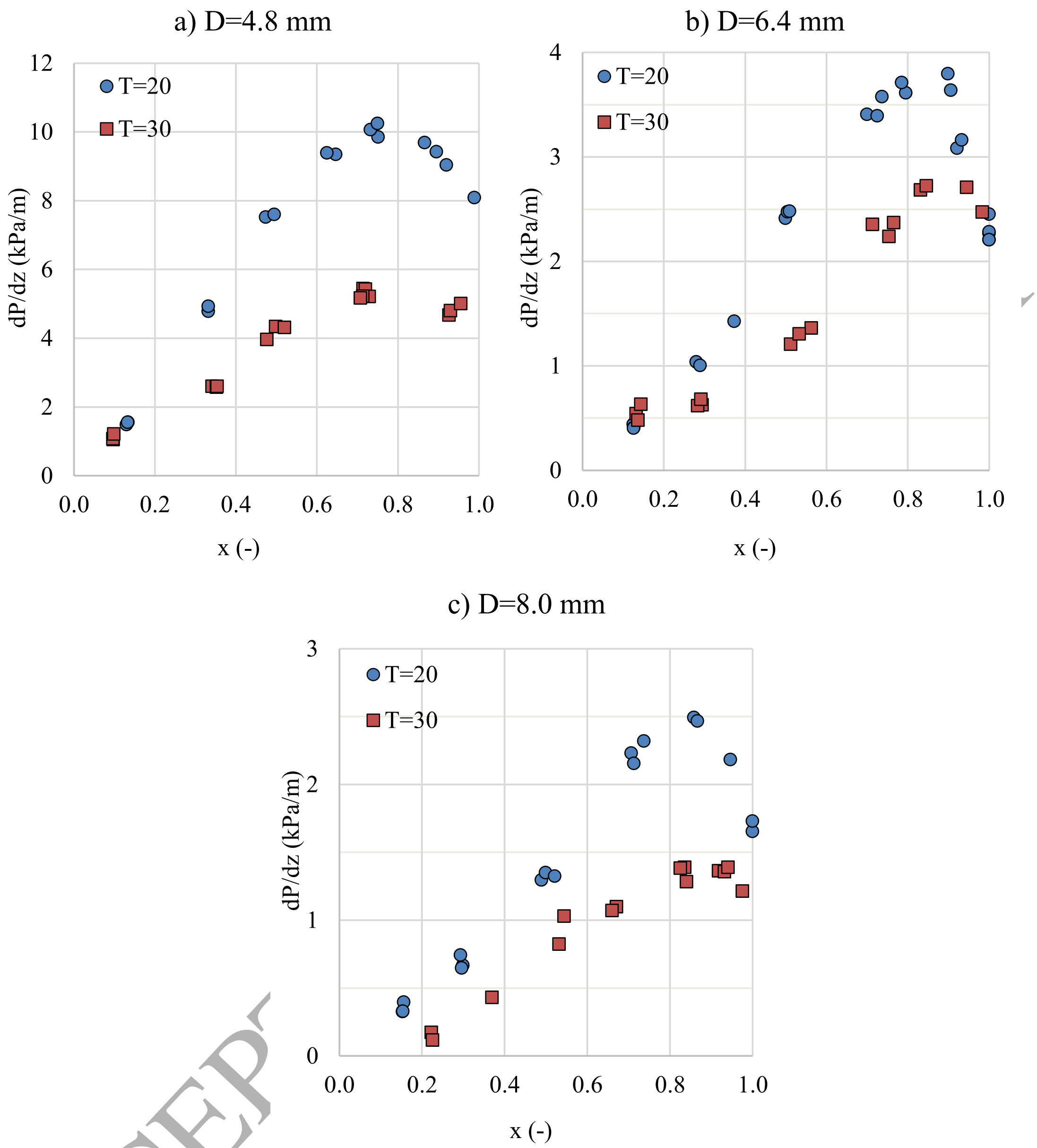


Figure 7. Effect of the saturation temperature $T_{sat}$ (°C) on the two-phase pressure gradient of R1234yf, $G = 300\ kg/m^2s$.

#### 4.1.4. Effect of heat flux

Figure 8 shows the pressure gradient measurement when the test section receives a heat flux. It is observed that the heat flux does not influences the frictional pressure gradient.

Under adiabatic flow conditions with constant quality in the test section, the pressure gradient remains constant along the tube; the pressure drop measured by the test section sensor is known as the local pressure gradient, which occurs only due to the friction effect. However, when there is heat flux in the test section, the quality increases with the position. Two-phase pressure drop is the integral of these gradients varying with the quality, and it is the result of friction and momentum effects (Mauro et al., 2007). Figure 8 (a) illustrates the local pressure gradient (in adiabatic conditions) and the integral pressure gradient (in heat fluxes of 7 and 14 kW/m²). Figure 7 (b) depicts the frictional pressure gradient calculated by Eq. (9). The momentum pressure gradient is estimated by Eq. (10) to be approximately 7% of total pressure gradient measured. The results in Fig. 8 (b) demonstrate that the existence of low heat fluxes (7 and 10kW/m²) does not significantly affect the frictional pressure gradient.

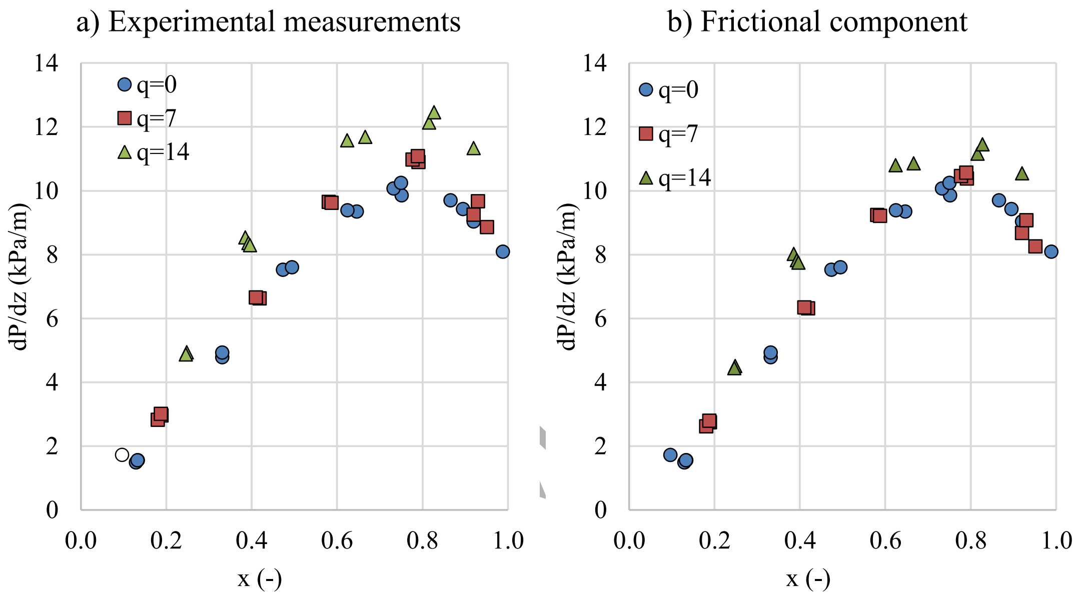


Figure 8. Effect of heat flux $q''$ (kW/m²) on the two-phase pressure gradient of R1234yf, $T_{sat} = 20°C$, $G = 300kg/m^2s$, $D = 4.8\,mm$.

#### 4.1.5. Effect of vapor quality

Figures 5-8 illustrate the effect of vapor quality on pressure gradient for different flow conditions. The pressure gradients increase linearly with the vapor quality until values are close to 0.8. For vapor quality greater than 0.8, the pressure gradients decrease. This behavior was observed by Greco and Vanoli (2006) and Garcia et al. (2017) for R407C and by Padilla et al. (2011) for R1234yf, and it is mainly attributed to the flow pattern change. According to Fig. 4 (the flow maps), the flow regime is generally slug or intermittent at low qualities (< 0.4). These two flow regimes were considered together and their respective frictional two-phase pressure drops follow similar trends, influenced mainly by the friction between the liquid and the wall of the tube (Quiben and Thome, 2007). In addition, as the liquid continually evaporates, the convergence of bubbles leads to an increasing pressure to the liquid film, which also increases the frictional pressure gradients(Wang et al., 2014). For higher quality, the flow tends to become annular and the liquid completely envelops the wall of the tube, while the vapor phase flows at the greatest velocity through the center. Thus, there are pressure drop stemming both from friction between liquid and wall and friction between liquid and vapor phases. The slip ratio between both phases increases with vapor quality, consequently increasing the pressure gradient (Quibén et al., 2009). As can be observed in the previous figures, the maximum pressure gradient occurs around x = 0.8. For quality greater than 0.8, the liquid layer begins to decrease until it dries completely. This reduces the interfacial friction, and the pressure gradient decreases to the value of single-phase vapor flow.

#### 4.1.6. Comparison between R1234yf and R134a

Figure 9 represents a comparison between the two-phase pressure gradients for R134a and R1234yf under adiabatic conditions at a constant evaporation temperature of 20°C and a tube diameter of 4.8 mm. R1234yf demonstrates a lower pressure drop over the entire quality range. The higher pressure drop of R134a is associated with the slight difference in thermophysical properties, as shown in Table 5. The vapor density of R134a is approximately 18% lower than that of R1234yf, which implies a higher vapor velocity for R134a (Lu et al., 2013). Other properties also may contribute to the lower pressure drop of R1234yf. For instance, at a saturation temperature of 20°C, the liquid viscosity of R1234yf is approximately 25% lower than that for R134a (Del Col et al., 2010). Fig. 9 illustrates that the difference is greater the mass velocity increases. For a flow rate of 400 kg/m² and quality of 0.75, the two-phase pressure gradient of R1234yf is 20% lower than that of R134a.

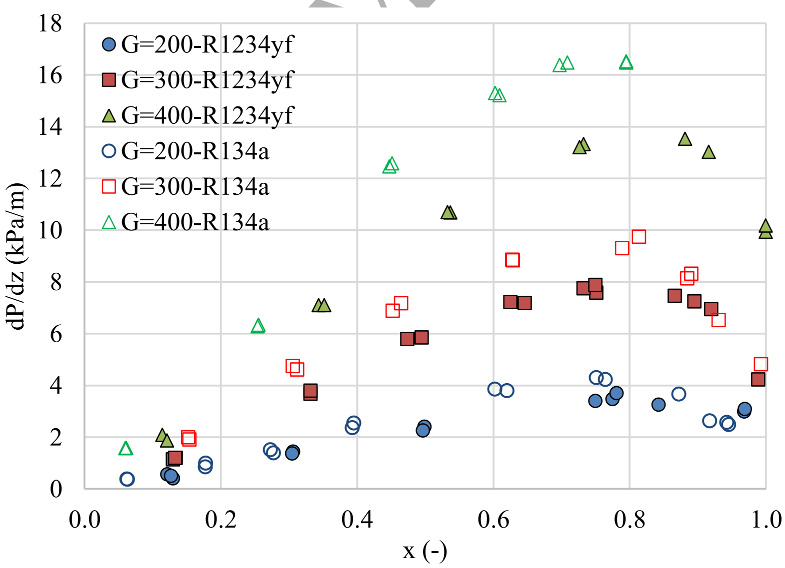


Figure 9. Two-phase pressure gradient of R1234yf and R134a, $T_{sat} = 20°C, D = 4.8\ mm$.

Table 5. Comparison of the physical properties of R134a and R1234yf at $T_{sat} = 20°C$.

| Property | R134a | R1234yf | Variation (%) (R1234yf-R134a)/R134a |
|---|---|---|---|
| $P_{sat}$ (kPa) | 572.1 | 591.7 | 3.44 |
| $P_r$ (-) | 0.141 | 0.175 | 24.11 |
| $\rho_l$ (kg/m³) | 1225 | 1110 | -9.39 |
| $\rho_v$ (kg/m³) | 27.80 | 32.84 | 18.13 |
| $\mu_l$ (μPa.s) | 207.4 | 154.4 | -25.55 |
| $\mu_v$ (μPa.s) | 11.5 | 12.3 | 6.96 |

### 4.2. Comparison of experimental results with existing correlations

Based on 212 sets of the test data in adiabatic conditions for R1234yf, the mean relative deviation (MRD) and mean absolute relative deviation (MARD) were calculated for the 19 two-phase pressure gradient correlations.

Table 6 presents the MARD and MRD statistical parameters, in which four of the best correlations have been bolded. It is observed that the correlation proposed by Xu and Fang (2012) is the best, with MARD equal to 20.4% and MRD equal to 4%. After that, correlations developed by Müller-Steinhagen and Heck (1986), Chawla (1967) and Wang et al. (1997) with MARD errors between 22.0% and 25.5% yield a satisfactory results. A second group of correlations with MARD less than 30.5% are Friedel (1979), Quiben and Thome (2007), Grönnerud (1979), Cicchitti et al. (1960) and Sun and Mishima (2009). The correlations that provided the worst estimates are Chisholm (1967), Tran et al. (2000) and Chisholm (1973). In this group, MARD was greater than 68.0% and MRD greater than -55.5%, which indicates that the estimated values are high when compared with the experimental database.

The model based on two-phase flow patterns of Quiben and Thome (2007) provided an acceptable result. The methods based on the homogeneous approach performed poorly. For the methods related to two-phase multipliers, the approaches had diverse results. The best and the worst predictions were obtained by methods based on two-phase multipliers.

Table 6. MARD and MRD of two-phase pressure gradient correlations.

| Correlation | MARD | MRD | Correlation | MARD | MRD |
|---|---|---|---|---|---|
| **Xu and Fang (2012)** | **20.4** | **4.0** | McAdams et al. (1942) | 40.8 | 37.8 |
| **Müller-Steinhagen and Heck (1986)** | **22.1** | **9.7** | Dukler et al. (1964) | 43.7 | 41.2 |
| **Wang et al. (1997)** | **25.0** | **-16.4** | Jung and Radermacher (1989) | 48.5 | -39.1 |
| **Chawla (1967)** | **25.4** | **-10.3** | Yu et al. (2002) | 49.4 | 46.8 |
| Friedel (1979) | 27.8 | -6.7 | Li and Hibiki (2017) | 54.0 | -52.3 |
| Quiben and Thome (2007) | 28.9 | -18.5 | Mishima and Hibiki (1996) | 57.7 | -45.2 |
| Grönnerud (1979) | 29.3 | -19.9 | Chisholm (1967) | 68.4 | -55.6 |
| Cicchitti et al. (1960) | 30.1 | 22.1 | Tran et al. (2000) | 68.7 | -65.9 |
| Sun and Mishima (2009) | 30.3 | 22.4 | Chisholm (1973) | 72.9 | -61.6 |
| Awad and Muzychka (2010) | 37.1 | 32.4 | | | |

The analysis presented in Fig. 10 on the distribution of the absolute error percentage allowed a better understanding of the performance of the four best correlations. In general, the correlations developed by Xu and Fang (2012), Müller-Steinhagen and Heck (1986) and Chawla (1967) follow the same probability curve, and 65% of the data are estimated with a maximum error of ± 30%. The correlation proposed by Wang et al. (1997) estimates 55% of the data, within this range of error.

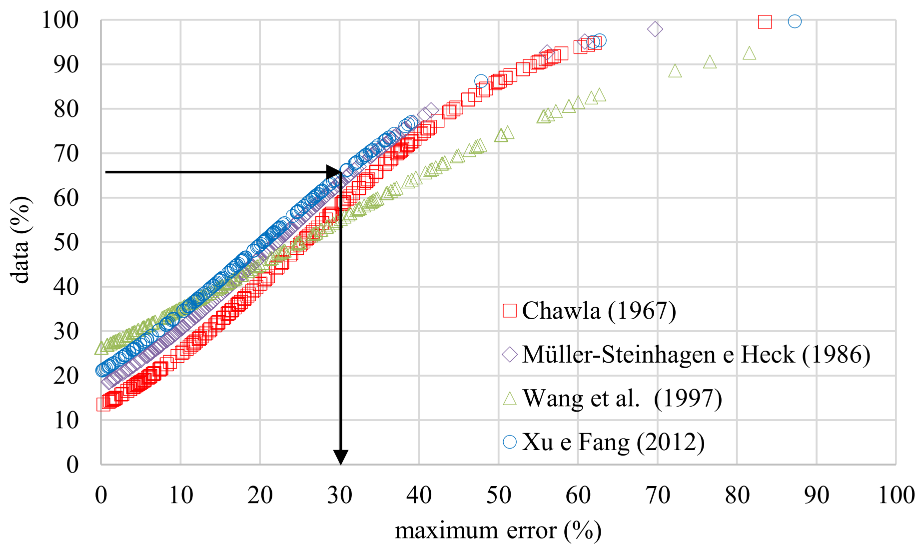


Figure 10. Distribution of data on the maximum relative error in the correlations.

Figure 11 represents the direct comparison between the experimental and predicted values using the correlation proposed by Xu and Fang (2012). It can be concluded that this correlation provides a smaller error for predicting the experimental two-phase pressure gradient in most cases in which the determination coefficient is equal to $R^2 = 0.95$. This successful performance is attributed to the fact that it was developed based similar experimental conditions similar to those of the present study. For example, their database was obtained from macrochannels and minichannels (a major portion of dataset was from tube diameters between 2 and 10 mm), and 40.7% and 16.2% of data were located in annular flow and intermittent flow, respectively. Additionally, the database involved refrigerants R134a, R22 and R410A, which contributed 27.8%, 22.0%, and 18.5%, respectively. Thus, R134a was the main fluid. This is important because Table 5 shows the similarity in thermophysical properties between R1234yf and R134a. Finally, the correlations developed by Xu and Fang (2012) and Müller-Steinhagen and Heck (1986) also were reported to be the best for their database.

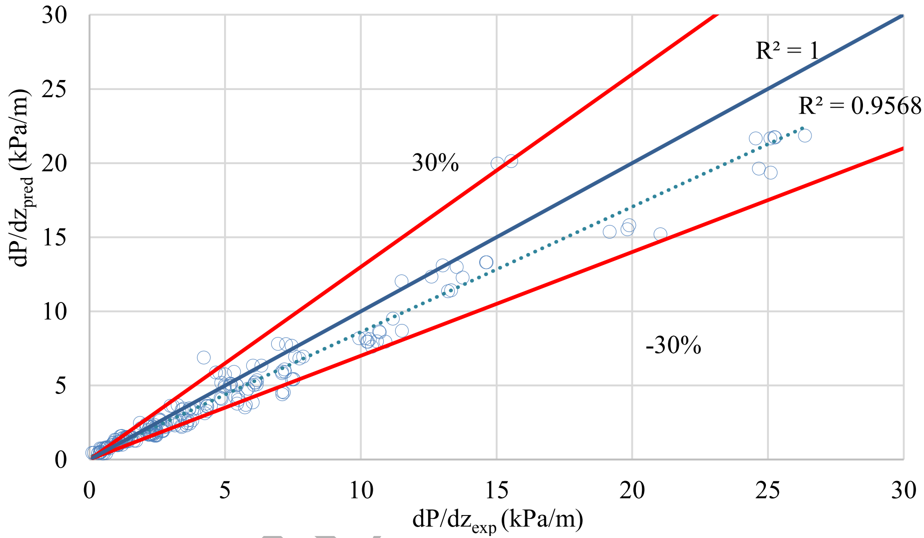


Figure 11. Experimental pressure gradient data compared to prediction of the Xu and Fang (2012) correlation.

## 5. Conclusion

An experimental database with 248 points for the two-phase pressure drop of R1234yf in macro-scale smooth and horizontal tubes was used. The test conditions included: four tube internal diameters (3.2 mm, 4.8 mm, 6.4 mm and 8.0 mm); three mass velocities (200 kg/m²s, 300 kg/m²s and 400 kg/m²s); two saturation temperatures (20°C and 30°C); adiabatic and diabatic conditions; and vapor qualities from 0 to 1.

The purpose of the current experimental investigation was to study the effects of quality, tube diameter, mass velocity, saturation temperature and heat flux on the two-phase flow of R1234yf. The results indicated that the two-phase pressure gradient was increased by an increase in mass velocity or a decrease in tube diameter or saturation temperature. The two-phase pressure gradient as a function of vapor quality presents a peak value. According to the flow pattern map of Wojtan et al. (2005), this peak value occurs in the annular flow regime. The experimental vapor quality corresponding to the maximum pressure gradient ranges from 0.75 to 0.95; therefore, it can be concluded that the two-phase pressure gradient for R134a is higher than that for R1234yf.

This work presents a performance evaluation of 19 pressure gradient correlations that were tested for R1234yf in adiabatic conditions. The Xu and Fang (2012) and Müller-Steinhagen and Heck (1986) correlations based on the two-phase multipliers approach provided the best predictions of the experimental database, with MARD values of less than 22% and around 65% of the data predicted within a ±30% error band. The correlations developed by Wang et al. (1997), Chawla (1967) and Friedel (1979) provided acceptable predictions (MARD < 27%). The Quiben and Thome (2007) correlation, a phenomenological approach, also achieved reasonable results. Utilizing correlations that are based on the homogeneous approach is not recommended.

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