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To cite this article: A Diani et al 2014 J. Phys.: Conf. Ser. 547 012026

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# R1234yf vs. R134a Flow Boiling Heat Transfer Inside a 3.4 mm ID Microfin Tube

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Abstract. The refrigerant charge minimization as well as the use of eco-friendly fluids can be considered two of the most important targets for these applications to cope with the new environmental challenges. This paper compares the R1234yf and R134a flow boiling heat transfer and pressure drop measurements inside a small microfin tube with internal diameter at the fin tip of 3.4 mm. This study is carried out in an experimental facility built at the Dipartimento di Ingegneria Industriale of the University of Padova especially designed to study both single and two phase heat transfer processes. The microfin tube is brazed inside a copper plate and electrically heated from the bottom. Several T-type thermocouples are inserted in the wall to measure the temperature distribution during the phase change process. In particular, the experimental measurements were carried out at constant saturation temperature of 30 °C, by varying the refrigerant mass velocity between 190 kg m<sup>-2</sup> s<sup>-1</sup> and 940 kg m<sup>-2</sup> s<sup>-1</sup>, the vapour quality from 0.2 to 0.99, at different imposed heat fluxes. The two refrigerants are compared considering the values of the two-phase heat transfer coefficient and pressure drop.

#### **1. Introduction**

The use of synthetic refrigerants with a non-negligible Global Warming Potential or, on the contrary, of natural but flammable or toxic natural fluids calls for the charge minimization of the refrigerating and air conditioning equipment. The refrigerant charge minimization as well as the use of eco-friendly fluids can therefore be considered two of the most important targets for these applications to cope with the new environmental challenges. Traditional microfin tubes are also widely used in air and water heat exchangers for heat pump and refrigerating applications during condensation or evaporation. The possible downsizing of microfin tubes can lead to more efficient and compact heat exchangers and thus to a reduction of the refrigerant charge of the systems. Furthermore, over the last several years, much research and development effort has been focused on potential refrigerants possessing low GWPs. Among the fluorinated propene isomers which have normal boiling point temperature data published in the public domain, several have low GWPs and normal boiling temperatures relatively close to R134a; among them, R1234yf has as a normal boiling temperature approximately 3.4°C lower than that of R134a, with a GWP<1. Recently, Domanski et al. [1] and McLinden et al. [2] have performed a detailed thermodynamic analysis of refrigerants, studying the performance limits of the vapour compression cycle and testing the possible low-GWP refrigerant candidates suitable for use in common types of refrigeration and air conditioning equipment. In particular, Domanski et al. [1] suggested that the critical temperature of a refrigerant can be considered the most dominant parameter

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influencing the tradeoff between the Coefficient of Performance (COP) and the volumetric capacity  $(Q_{vol})$ . McLinden et al. [2] carried out a screening of the possible refrigerants, eliminating those toxic or unstable and focusing on those that presented critical temperatures between 300 K and 400 K; they analysed 62 refrigerants. Two of those were R1234ze(E) and R1234yf, which have recently been matter of research and investigation of the scientific community because they present values of low-GWP and normal boiling temperature close to that of R134a, the traditional fluid commonly used in refrigeration and air conditioning equipment.

The R1234yf has as a normal boiling temperature approximately 3.15 K lower than that of R134a, whereas that of R1234ze(E) is 7.3 K lower than that of R134a; these new refrigerants have a GWP<1. These two fluids are candidates to substitute the traditional R134a in several applications: from the automotive air conditioning to the high temperature heat pump.

Fukuda et al. [3] thermodynamically, experimentally, and numerically analyzed the feasibility of R1234ze(E) and R1234ze(Z) for high temperature heat pumps, demonstrating that these new low-GWP fluids are capable of being a potential refrigerant in high-temperature heat pump systems for industrial purposes, rather than typical air conditioners or refrigeration systems. However, up today only few works experimentally investigated the heat transfer capabilities of these refrigerants during single and two-phase flow inside conventional and enhanced tubes.

Grauso et al. [4] studied the heat transfer and pressure drop during evaporation of R1234ze(E) in a circular smooth tube of 3 mm OD; the authors measured the two-phase heat transfer coefficient at different mass velocity from 146 to 520 kg m<sup>-2</sup> s<sup>-1</sup>, and vapour quality, saturation temperature between -2.9 °C and 12.1 °C ( $p_{red} = 0.05$ -0.09), imposing two different heat fluxes: 5 and 20 kW m<sup>-2</sup>, provided by electrical heating.

Hossain et al. [5] experimentally investigated the evaporation of R1234ze(E), R32, R410A, and a mixture of R1234ze(E) and R32 inside a horizontal smooth tube. The experiments were carried out under the conditions of mass flux varying from 150 to 445 kg m<sup>-2</sup> s<sup>-1</sup>, the saturation temperature were 5 °C and 10 °C (R1234ze(E)  $p_{red} = 0.07 - 0.09$ ; R32  $p_{red} = 0.19$ ; R410A  $p_{red} = 0.22$ ; R1234ze(E)/R32 mixture  $p_{red} = 0.15 - 0.18$ ), over vapour quality from 0 to 1. Refrigerant flowed inside a smooth copper tube with an inner diameter of 4.35 mm and an outer diameter of 6.35 mm, whereas heating water flowed through an annular space in counter current. With this technique, authors achieved heat fluxes up to 70 kW m<sup>-2</sup>. The authors found that the heat transfer coefficients exploited by pure R1234ze(E) are lower than those measured for R1234ze(E)/R32 mixture, R410A, and R32.

Del Col et al. [6] investigated the R1234yf flow boiling heat transfer in a 1 mm diameter circular microchannel. During tests, the heat was provided to the boiling fluid by using a secondary fluid. Flow boiling tests were carried out at 31 °C of saturation temperature ( $p_{red} = 0.24$ ), mass fluxes ranging between 200 and 600 kg m<sup>-2</sup> s<sup>-1</sup>, and heat fluxes from 10 to 130 kW m<sup>-2</sup>. Authors compared the results with those measured for R134a, and no significant differences between the flow boiling performance of R1234yf and R134a were reported.

Lu et al. [7] investigated the influences of heat flux and mass flux on the two-phase convective boiling heat transfer performance for refrigerants R1234yf and R134a in a 3.9 mm smooth diameter tube. The horizontal test section is a double-pipe heat exchanger: water flows countercurrently in the test annulus, while refrigerant is evaporated inside the test tube. Tests were performed with a saturation temperature of 10 °C ( $p_{red} = 0.13$ ), with a mass velocity ranging from 200 to 500 kg m<sup>-2</sup> s<sup>-1</sup>, and heat flux from 5 to 20 kW m<sup>-2</sup>. A noticeable deterioration of the heat transfer coefficient for R1234yf was encountered whereas the pressure drops for R134a are about 5–15% higher than those of R1234yf.

Only one work relative to flow boiling heat transfer of new low-GWP refrigerants inside microfin tubes can be found in the open literature; Kondou et al. [8] experimentally investigated the flow boiling of R32, R1234ze(E) and R32/R1234ze(E) non-azeotropic mixtures in a horizontal microfin tube of 5.2 mm inner diameter at 10 °C of saturation temperature (R32  $p_{red} = 0.19$ , R1234ze(E)  $p_{red} = 0.09$ , R32/R1234ze(E)  $0.2/0.8 p_{red} = 0.100$ ; R32/R1234ze(E)  $0.5/0.5 p_{red} = 0.179$ ), with heat fluxes of 10 and 15 kW m<sup>-2</sup> with a water heating, and mass velocities from 150 to 400 kg m<sup>-2</sup> s<sup>-1</sup>. The heat

transfer coefficients of R1234ze(E) are lower than those of R32 but they are greater than those obtained for R32/R1234ze(E) mixtures.

However, a few of other works studied the flow boiling heat transfer inside microfin tubes with an internal diameter lower than 5 mm, among those: Mancin et al. [9-11], Gao et al. [12-13], Dang et al. [14], and Wu et al. [15].

Recently, Mancin et al. [9-11] have experimentally investigated the flow boiling heat transfer of R134a inside a mini microfin tube, with an internal diameter at the fin tip of 3.4 mm, electrically heated. The collected data permitted to study the effects of refrigerant mass velocity, vapour quality, and heat flux on the liquid-vapour phase change process at constant saturation temperature of 30 °C ( $p_{red} = 0.19$ ).

Gao et al. [12-13] conducted experiments on flow boiling of CO<sub>2</sub> and oil mixtures in electrically heated horizontal smooth and microfin tubes. The microfin was a copper tube with an inner diameter of 3.04 mm. Experiments were carried out at mass velocities from 190 to 1300 kg m<sup>-2</sup> s<sup>-1</sup>, at a saturation temperature of 10 °C ( $p_{red} = 0.61$ ), heat fluxes from 5 to 30 kW m<sup>-2</sup>, and an oil circulation ratio from <0.01 to 0.72 wt%.

Dang et al. [14] investigated the flow boiling of CO<sub>2</sub> inside a horizontal internally-grooved tube, with an internal diameter of 2.0 mm at a saturation temperature of 15 °C ( $p_{red} = 0.67$ ), the heat flux ranged between 4.5 to 18 kW m<sup>-2</sup>, and the mass velocity from 360 to 720 kg m<sup>-2</sup> s<sup>-1</sup>. The test section was placed horizontally, and the copper test tube was heated by directly supplying DC current to it. The heat transfer coefficient for the grooved tube was 1.9 - 2.3 times higher than that for the smooth tube, and the dryout quality was much higher, ranging between 0.90 and 0.95, while the pressure drops are 1.5 - 2.7 higher.

Wu et al. [15] performed experiments during flow boiling of R22 and R410A inside one smooth tube and five microfin tubes with the same outer diameter of 5 mm, the mass velocity was varied from 100 to 620 kg m<sup>-2</sup> s<sup>-1</sup>, the heat flux from 5 to 31 kW m<sup>-2</sup>, at a saturation temperature of around 6 °C (R22  $p_{red} = 0.12$ ; R410A  $p_{red} = 0.20$ ). The test section was a 2 m long straight, horizontal tube-in-tube heat exchanger, where the refrigerant flows in counter current with heating water. They also developed a new general semi-empirical model based on present data and recent data from literature. This model is applicable for intermittent and annular flow patterns.

This paper presents the experimental results of R1234yf flow boiling heat transfer inside a 3.4 mm ID microfin tube electrically heated; the effects of mass flow rate, vapour quality, and heat flux at constant saturation temperature of 30°C (R1234yf  $p_{red} = 0.23$ ; R134a  $p_{red} = 0.19$ ) are presented and analysed. Furthermore, the experimental measurements for R1234yf are compared with those obtained for R134a in the same tube at the same operating test conditions [9-10].

#### 2. Experimental setup and data reduction

The experimental setup is located at the Heat Transfer in Micro-geometries Lab (HTMg-Lab) at the Dipartimento di Ingegneria Industriale of the University of Padova. As shown in Fig. 1, the experimental facility consists of three loops: refrigerant, cooling water and hot water loops. The rig was designed for heat transfer and pressure drop measurements and flow visualization during either vaporization or condensation of pure refrigerants and refrigerants mixtures inside structured micro-geometries. The facility has a maximum working pressure of 3 MPa, while refrigerant mass fluxes can be varied up to  $G=400 \text{ kg m}^{-2} \text{ s}^{-1}$  in a section of 50 mm<sup>2</sup>.

In the first loop the refrigerant is pumped through the circuit by means of a magnetically coupled gear pump, it is vaporized and superheated in a brazed plate heat exchanger fed with hot water. Superheated vapour then partially condenses in a pre-condenser fed with cold water to achieve the set quality at the inlet of the test section. The refrigerant enters the test section at a known mass velocity and vapour quality and then it is vaporized by means of a calibrated Ni-Cr wire resistance.

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Journal of Physics: Conference Series 547 (2014) 012026

doi:10.1088/1742-6596/547/1/012026



Figure 1. Schematic of the experimental setup.

The fluid leaves the test section and enters in a post-condenser, a brazed plate heat exchanger, where it is fully condensed and subcooled. The subcooled liquid passes through a drier filter and then is sent back to the boiler by a pump. A damper connected to the compressed air line operates as pressure regulator to control the saturation condition in the refrigerant loop. As shown in Fig. 1, the refrigerant pressure and temperature are measured in several locations throughout the circuit to know the refrigerant properties at the inlet and outlet of each heat exchanger. The refrigerant mass flow rate can be independently controlled by the gear pump and it is measured by means of a Coriolis effect flowmeter. The inlet vapour quality to the test section is determined by the heat extracted in the precondenser, which can be controlled by varying water temperature and flow rate. The cold water loop consists of a chiller with thermostatic control connected to the precondenser. The hot water circuit consists of a pump, an electrical heater and a controlling valve; it permits to set both the water flow rate and the inlet water temperature. Water flow rates in the precondenser and boiler sections are measured by means of magnetic type flow meters, while the water temperature differences are measured using 4-junction T-type thermopiles. Table 1 lists the values of accuracy of the instruments implemented in the experimental facility.

| Transducer                                      | Accuracy                    |
|---|-----------------------------|
| T-type thermocouples                            | $\pm 0.05 \text{ K}$        |
| T-type thermopiles                              | $\pm 0.03 \text{ K}$        |
| Electric power                                  | $\pm 0.13\%$ of the reading |
| Coriolis mass flowmeter (refrigerant loop)      | $\pm 0.10\%$ of the reading |
| Magnetic volumetric flowmeters                  | $\pm0.25\%$ of the reading  |
| Differential pressure transducer (test section) | ± 25 Pa                     |
| Absolute pressure transducers                   | ± 1950 Pa                   |

Table 1. Accuracy of the implemented instruments.



Figure 2. Photo of the test section.

As shown in Fig. 2, the tested microfin tube is brazed inside a groove milled in a 20 mm thick, 10 mm wide, and 300 mm long copper plate. The pressure taps are located around 50 mm upstream and downstream of the heated tube; a smooth connection was designed and manufactured in order to prevent any possible abrupt pressure loss. The test tube has a heated length of 300 mm whereas the total length for the pressure drop measurement is 410 mm. The microfin tube is heated from the bottom by means of a calibrated Ni-Cr wire resistance inserted in a 2 mm deep groove milled on the bottom face of the copper plate. The heat is supplied by means of stabilized DC power supplier, which is able to supply up to 900 W. The instrumented test section is located inside an aluminum housing filled up with a 30 mm thick layer of rock wool to limit as much as possible the heat loss.

According to the nomenclature described in Fig. 3, the microfin tube has an OD of 4.0 mm, an ID at the fin's tip of D=3.4 mm, it has 40 fins with a fin's height of h=0.12 mm, the helix angle  $\beta=18^{\circ}$ . Twenty 1 mm ID holes were drilled along the centreline of the copper plate just 1 mm under the tested microfin tube. Twenty calibrated T-type thermocouples are inserted in those 5 mm deep holes to measure the wall temperature distribution during the heat transfer process. Figure 3 also reports a photo of the cross sectional area of the mini microfin tube tested where several fins are clearly visible.

Preliminary heat transfer measurements permitted to estimate the heat loss  $(q_{loss})$  due to conduction through the test section as a function of the mean wall temperature. The tests were run under vacuum conditions by supplying the power needed to maintain the mean wall temperature at a fixed value. The measurements were carried out by varying the wall temperature from around 30 °C to 60 °C, at different ambient temperature from 21 °C to 24 °C. In this range, there is not any appreciable effect of the ambient temperature on the actual heat loss. The relationship between the heat loss and the wall temperature is linear; in this way, the actual value of heat supplied to the sample can be evaluated. The heat loss was never higher than 4% of the electrical power when the ambient temperature is kept constant within 21 °C and 24 °C.



**Figure 3.** Drawing of a typical microfin tube and photo of the cross section of the tested tube.

As described before, the subcooled liquid is pumped to the boiler where it is vaporized and superheated; the refrigerant temperature and pressure are measured at both inlet and outlet of the heat exchanger. Preliminary tests were run to verify the heat balance at both pre-condenser and evaporator; the misbalance was always less than 2%. The vapour quality at the inlet of test section depends on the refrigerant conditions at the inlet of the precondenser and on the heat flow rate exchanged in the tube-in-tube heat exchanger and it can be obtained from a thermal balance on the cooling water side as given by:

$$q_{pc} = \dot{m}_{w,pc} \cdot c_{p,w} \cdot \left( t_{w,pc,out} - t_{w,pc,in} \right) = \dot{m}_{ref} \left( h_{vs} - h_{TSin} \right)$$
(1)

where  $\dot{m}_{w,pc}$  is the water mass flow rate at the precondenser,  $c_{p,w}$  the water specific heat at constant pressure,  $t_{w,pc,out}$  and  $t_{w,pc,in}$  the water temperatures at the outlet and inlet of the precondenser, respectively. Considering the right-hand side of eq. (1),  $\dot{m}_{ref}$  is the refrigerant mass flow rate, while  $h_{vs}$  is the enthalpy of the superheated gas at the inlet of the precondenser, and  $h_{TS,in}$  the enthalpy of the refrigerant at the inlet of the test section. The vapour quality at the inlet of the test section  $(x_{in})$  can be calculated from the heat balance, as:

$$x_{in} = \frac{h_{TSin} - h_L}{h_V - h_L} \tag{2}$$

where  $h_L$  and  $h_V$  are the specific enthalpies of the saturated liquid and vapour, respectively, evaluated at the saturation pressure of the refrigerant measured at the inlet of the test section. All the thermophysical properties of the refrigerant were estimated using Refprop ver. 9.1 [16] database. The electrical power supplied to the sample is indirectly measured by means of a calibrated reference resistance (shunt) and by the measurement of the effective EDP (Electrical Difference Potential) of the resistance wire inserted in the copper heater. The current can be calculated from the Ohm's law. From preliminary measurements, the heat loss through the test section can be estimated by:

$$q_{loss}[W] = 0.1121 \cdot \overline{t}_{wall}[^{\circ}C] - 2.4042$$
 (3)

where  $\bar{t}_{wall}$  is the mean wall temperature; thus, the actual heat flow rate supplied to the tested microfin tube is given by:

$$q_{\text{TS}} = P_{EL} - q_{loss} = \Delta V \cdot I - q_{loss} \tag{4}$$

The two-phase heat transfer coefficient HTC, referred to the area of the smooth tube with the same inner diameter of that at the fin tip,  $A_D$ , can now be defined as:

$$HTC = \frac{q_{TS}}{A_D \cdot (\overline{t}_{wall} - \overline{t}_{sat})}$$
(5)

where  $\bar{t}_{wall}$  and  $\bar{t}_{sat}$  are the average values of the wall and saturation temperatures, respectively. Their values are given by:

$$\bar{t}_{wall} = \frac{1}{20} \sum_{i=1}^{20} t_{wall,i} \qquad \bar{t}_{sat} = \frac{t_{sat,in}(p_{sat,in}) + t_{sat,out}(p_{sat,out})}{2}$$
(6)

From the error propagation analysis, it was estimated that the mean uncertainty on the two-phase heat transfer coefficient is  $\pm 2.5\%$ , whereas the vapour quality has an uncertainty of  $\pm 0.035$ .

The frictional pressure gradient is obtained from the measured value of the total pressure gradient by subtracting the momentum pressure gradient and neglecting the gravitational term, as given by:

$$\left(-\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{f} = \left(-\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{tot} - \left(-\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{M}$$
(7)

The model proposed by Rouhani and Axelsson [17] was used to estimate the void fraction values to account for the momentum pressure gradient.

## 3. R1234yf vs. R134a

This section compares the experimental results obtained during flow boiling heat transfer of R1234vf and R134a at 30 °C of saturation temperature at the inlet of the mini microfin tube. Figures 4 and 5 report the data collected for R1234vf and R134a, respectively, at HF= 10 kW m<sup>-2</sup>; the heat flux is referred to the outer heat transfer area of the microfin tube having a diameter equal to 4 mm. At a glance, the two fluids show different behaviours; starting from the R1234vf (Figure 4), it clearly appears that the heat transfer coefficient is only weakly affected by the refrigerant mass velocity while it increases as the vapour quality increases. In fact, at low vapour quality the heat transfer coefficients are similar at all the investigated mass velocities. It is interesting to highlight that the highest heat transfer coefficients are exhibited by G=375 kg m<sup>-2</sup> s<sup>-1</sup> and, at high vapour quality, the heat transfer coefficient appears to slightly decrease as the mass velocity increases. It seems that the nucleate boiling controls the phase-change process limiting the two-phase forced convection especially at the high mass velocity. At G=190 kg m<sup>-2</sup> s<sup>-1</sup> the dryout occurred at a critical vapour quality around  $x_{cr}=0.97$ ; at higher mass velocity the dryout was not observed. The R134a (Figure 5) exploits a different behaviour: considering the lowest mass velocity, G=190 kg m<sup>-2</sup> s<sup>-1</sup>, the heat transfer coefficient remains almost constant at around 6100 W m<sup>-2</sup> K<sup>-1</sup> up to  $x_{mean}$ =0.5, this might mean that the phase change process is controlled by the nucleate boiling; then, it increases with vapour quality. When increasing the mass velocity, the plateau at low vapour quality, where the heat transfer coefficient remains constant disappears, and the heat transfer coefficient increases almost linearly with the vapour quality meaning that the two phase forced convection is affecting in the phase change process. It is worthy to point out that, at  $x_{mean} > 0.65$ , the values of the heat transfer coefficient measured at G=375 kg m<sup>-2</sup> s<sup>-1</sup> are greater than those obtained at higher mass velocities; this can be linked to a particular effect, due to the presence of the helical micro-fins that might be enhanced at this operating test condition. Furthermore, at this heat flux, no experimental evidence of the onset of dryout was observed at any mass velocity.



**Figure 4.** R1234yf flow boiling heat transfer coefficient at  $10 \text{ kW m}^{-2}$ .



**Figure 5.** R134a flow boiling heat transfer coefficient at  $10 \text{ kW m}^{-2}$ .

| Property   | R134a  | R1234yf |
|--|--------|---------|
| $p_{sat}$ [bar]  | 7.70   | 7.84    |
| $p_{red}$ [-]  | 0.19   | 0.23    |
| $\rho_L [\mathrm{kg}\mathrm{m}^{-3}]$                    | 1188   | 1073    |
| $\rho_V [\mathrm{kg m}^{-3}]$                            | 37.5   | 43.7    |
| $h_{LV}$ [kJ kg <sup>-1</sup> ]                          | 173    | 141     |
| $\lambda_L [\mathrm{W} \mathrm{m}^{-1} \mathrm{K}^{-1}]$ | 0.079  | 0.062   |
| $\mu_L$ [µPa s]  | 183    | 145     |
| $\sigma$ [mN m <sup>-1</sup> ]                           | 7.38   | 5.56    |
| $(dp/dT)_{sat}$ [MPa K <sup>-1</sup> ]                   | 0.0221 | 0.0212  |

Table 2. Major thermophysical properties of R1234yf and R134a at 30 °C of saturation temperature. Data from Refprop 9.1 [16]

The results described before can be discussed considering the properties of the two refrigerants at 30 °C of saturation temperature, as listed in Table 2. The reduced pressure and vapour density of the R1234yf are 22% and 17%, respectively, higher than those of R134a while its thermal conductivity is some 22% lower than that of the more traditional fluid. As a result, it might be expected that R1234yf shows greater attitude to the nucleate boiling by virtue of the higher reduced pressure. On the other hand, the two-phase forced convection would be better exploited by the R134a, which has lower vapour density that, for a given mass velocity, means higher vapour velocity. The lower thermal conductivity of the R1234yf as compared to that of R134a might also explain its lower values of heat transfer coefficient.

The results reported in Figures 6 and 7 confirmed the considerations drawn before; in particular, Figure 6 compares the heat transfer coefficients for R134a and R1234yf as a function of the mean vapour quality at HF=25 kW m<sup>-2</sup> and G=375 kg m<sup>-2</sup> s<sup>-1</sup>. At low vapour quality, the two fluids show similar values of the heat transfer coefficients. As the vaporization proceeds, the amount of liquid decreases and the two-phase forced convection becomes more and more important, thus the attitude of R134a to two-phase forced convection leads to higher heat transfer coefficients as compared to those measured for the R1234yf. The onset of the dryout occurred at almost the same value of vapour quality, around  $x_{cr}=0.89$ .



**Figure 6.** R1234yf vs. R134a at 25 kW m<sup>-2</sup> and G=375 kg  $m^{-2} s^{-1}$ 



**Figure 7.** R1234yf vs. R134a at 50 kW m<sup>-2</sup> and G=375 kg  $m^{-2} s^{-1}$ 

Figure 7 compares the behaviour of the two refrigerants at the same mass velocity G=375 kg m<sup>-2</sup> s<sup>-1</sup> but at higher heat flux, HF=50 kW m<sup>-2</sup>. The higher heat flux flattens the values of the heat transfer coefficients, which are almost constant up to around  $x_{mean}$ =0.5 and then they slightly increase with vapour quality. The two refrigerants show similar values of the heat transfer coefficient at low vapour qualities. For higher vapour qualities, R134a shows slightly higher values: in this case, the difference between the two fluids is lower than that at 25 kW m<sup>-2</sup>, since at 50 kW m<sup>-2</sup> the effect of nucleate boiling is stronger, and thus the better thermal performance of R1234yf in nucleate boiling seems to mitigate its worse performance in two-phase forced convection at high vapour qualities. The onset of the dryout occurred at lower vapour qualities for R1234yf ( $x_{cr}$ =0.81) as compared to R134a ( $x_{cr}$ =0.85), in any case earlier than those exhibited at lower heat flux.



**Figure 8.** R1234yf vs. R134a frictional pressure gradient at  $HF=10 \text{ kW m}^{-2}$ .

Figure 8 compares the frictional pressure gradients measured during flow boiling heat transfer of R1234yf and R134a inside the tested microfin tubes at HF=10 kW m<sup>-2</sup>. It clearly appears that, for given vapour quality and refrigerant mass velocity, the frictional pressure gradients shown by R1234yf are slightly lower, at least similar, to those measured for R134a. In particular, at low mass velocities, the pressure drops of the two fluids are almost the same whereas when G>375 kg m<sup>-2</sup> s<sup>-1</sup>, the values exhibited by the R1234yf are lower than those obtained for R134a. This behaviour might be explained considering that the dynamic viscosity of the liquid of the R1234yf is some 20% lower than that of R134a while the vapour density is around 20% higher leading to a somewhat lower pressure drop. The proposed comparison merely considers the heat transfer and fluid flow behaviours of the two refrigerant; however, for a given cooling capacity, keeping constant the evaporating and condensing temperatures, the R1234yf refrigerant mass flow rate needed is some 20% higher than that of R134a because the latent heat of the new refrigerant is as much lower. This means that for a comprehensive and detailed comparison of the two refrigerants, the different mass flow rate must be taken into account to obtain meaningful results using the data reported in this work.

## 4. Conclusions

This paper compares the flow boiling heat transfer and fluid flow behaviours of the R1234yf and R134a inside a mini microfin tube having an internal diameter at the fin tip of 3.4 mm. The experimental measurements were collected at 30  $^{\circ}$ C of saturation temperature, by varying the refrigerant mass velocity, the vapour quality, and the heat flux.

The two refrigerants show different results as a function of the operating test conditions due to their slightly different thermophysical properties which control their attitude to enhance either the nucleate boiling or the two-phase forced convection phase change mechanisms.

The presented results might also be useful to compare the performance of these two refrigerants in real air conditioning and refrigerating applications, which has to be optimized as a function of the selected fluid.

Finally, this paper highlights the interesting heat transfer features of this new mini microfin tube, which exhibits high heat transfer enhancement and large charge minimization capability as compared to the traditional microfin tubes.

## 5. Acknowledgment

The support of Wieland-Werke AG and Dr. Christoph Walther on this research activity is gratefully acknowledged. The support of University of Padova is gratefully acknowledged.

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