Saturated R134a Flow Boiling inside a 4.3 mm ID Microfin Tube

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Abstract

The refrigerant charge minimization in refrigerating and air conditioning systems represents a challenging issue due to the new environmental national and international regulations. The use of smaller smooth tubes, i.e. with outer diameter around 5 mm*, is becoming more and more common in many applications. More recently, also the microfin tubes have started to be reduced in size to cope with the continuously increasing demand of new, efficient, and compact heat exchangers for air conditioning and refrigeration equipment. This work investigates the performance of R134a during saturated flow boiling inside a microfin tube with internal diameter at the fin tip of 4.3* mm*. Boiling heat transfer coefficients, frictional pressure drops, and critical vapor qualities were measured at 30*°C *of saturation temperature, by varying the refrigerant mass velocity between 100* kg m^{-2} s^{-1} *and 800* kg m^{-2} s^{-1} *and the vapor quality from 0.1 to 0.95 at four different heat fluxes: 15* kW m^{-2}*, 30* kW m^{-2}*, 60* kW m^{-2}*, and 90* kW m^{-2}*. Moreover, the reliability of several models for flow boiling heat transfer and pressure drop estimations was assessed by comparing the experimental results with the calculations.*

Introduction

Microfin tubes have been deeply investigated and used in many technical applications (e.g., air conditioning and refrigeration systems) since their first introduction in 1977 by Fujie et al. (1977). In fact, they potentially have many advantages with respect to smooth tubes, mainly when applied during refrigerant phase change. Despite a pressure drop increase, they provide higher heat transfer coefficients and a delayed onset of dryout during the boiling process. Furthermore, the fins realized along the tube circumference should contribute to an easier and quicker transition to annular regime with a consequent increase of the heat transfer performance (Han and Lee, 2005, Doretti *et al.*, 2013). Beside, in the last years microfin tubes with relatively lower diameters have been investigated. A reduction in diameter leads to a reduction of the refrigerant charge, so these mini microfin tubes can provide more compact heat exchangers with a minimization of the whole system refrigerant hold-up, while maintaining high efficiencies. This latter feature could result attractive also to cope with the new and even more stringent national and international environmental regulations that limit the refrigerant charge inventory and its maximum GWP, e.g. the new European F-gas regulation (Regulation (EC) No 517/2014). The scientific literature is strongly motivated in continuously studying microfin tubes in terms of heat transfer performance to provide an updated experimental database including different geometries and refrigerants, and to identify and/or develop reliable heat transfer and pressure drop models. Despite that, in the last decade or so, just few research groups have been focusing their attention microfin tubes smaller than D=6 mm.

Baba et al. (2012) presented an experimental work on R1234ze(E), R32, and a zeotropic R1234ze(E)/R32 (50:50) by mass% mixture flow boiling heat transfer inside a 4.86 mm ID at the fin tip microfin tube. The mass velocity ranged from 150 kg m^{-2} s^{-1} to 400 kg m^{-2} s^{-1} at a saturation temperature of 10°C.

Kondou et al. (2013) tested several refrigerants during flow boiling inside a water heated microfin tube having 4.94 mm ID at the fin tip: R32, R1234ze(E), a R32/R1234ze(E) (20:80) by mass% mixture, and a R32/R1234ze(E) (50:50) by mass% mixture. The saturation temperature was fixed at 10°C, the heat flux was set at 10 kW m^{-2} and 15 kW m^{-2}, and mass velocity ranged from 150 kg m^{-2} s^{-1} to 400 kg m^{-2} s^{-1}. Furthermore, Kondou et al. (2014a) analyzed the heat transfer performance in terms of heat transfer and pressure drop during both the condensation and the vaporization processes inside the same horizontal microfin tube (4.94 mm ID at the fin tip) of four refrigerant mixtures: two R744/R32/R1234ze(E) mixtures with compositions: (4:43:53) and (9:29:62) by mass% and two R32/R1234ze(E) mixtures with compositions: (30:70) and (40:60) by mass%. The authors carried out the flow boiling experimental tests at a saturation temperature of 10°C, a heat flux equal to 10 kW m^{-2}, and mass velocity ranging from 150 kg m^{-}

^{2} s^{-1} to 600 kg m^{-2} s^{-1}. In addition, the same 4.94 mm ID at the fin tip microfin tube was investigated by Kondou et al. (2014b) during R1234ze(E), R1234ze(Z), and R134a condensation and vaporization. During flow boiling tests, the heat flux was fixed at 10 kW m^{-2}, the saturation temperature ranged from 0°C to 30°C, and the mass velocity was set at 150 kg m^{-2} s^{-1}, 200 kg m^{-2} s^{-1}, and 300 kg m^{-2} s^{-1}.

In a different laboratory, a 3.4 mm ID at the fin tip electrically heated microfin tube was investigated during flow boiling of several refrigerants at 30°C of saturation temperature, heat fluxes ranging from 10 kW m^{-2} to 50 kW m^{-2} and mass velocities from 190 kg m^{-2} s^{-1} to 940 kg m^{-2} s^{-1}. Mancin et al. (2015) tested R134a, Diani et al. (2014) presented R1234ze(E) data, and Diani et al. (2015) measured R1234yf heat transfer performance. The same approach was used to test a 2.4 mm ID at the fin tip electrically heated microfin tube during flow boiling of R1234ze(E) and R134a (Diani et al., 2016), and R1234yf (Diani and Rossetto, 2015) at 30°C of saturation temperature, with mass velocities ranging from 375 kg m^{-2} s^{-1} to 940 kg m^{-2} s^{-1}, and heat fluxes from 10 kW m^{-2} to 50 kW m^{-2}.

Wu et al. (2013) investigated R22 and R410A flow boiling inside one smooth tube and five microfin tubes having the same outer diameter (5 mm) and different number of fins and helix angles. The investigated mass velocities ranged from 100 to 620 kg m^{-2} s^{-1}, at a saturation temperature of 6 °C. They found that microfin tubes are more efficient at low mass velocities, since the heat transfer coefficient per unit pressure drop decreases with mass velocity.

He et al. (2016) investigated R410A and a near azeotropic R290/R32 (68:32) by mass% mixture during flow boiling inside three different microfin tubes, having 4.3 mm, 6.1 mm, and 8.48 mm diameter at the fin tip, respectively. The saturation temperature was kept equal to 7 °C, 9 °C, and 11 °C, the mass velocity ranged from 50 kg m^{-2} s^{-1} to 250 kg m^{-2} s^{-1}, and the heat flux from 10 kW m^{-2} to 30 kW m^{-2}. The authors measured heat transfer coefficients also inside smooth tubes with 4.0 and 6.0 mm respectively. The HTC measured during flow boiling inside the microfin tubes having similar fin tip diameter were consistently higher than the ones inside smooth tubes at the same working conditions.

Despite the few recent works listed above, the available experimental data set in different microfin geometries having small diameter is still rather limited. So new sets of data obtained on different microfin tubes are indeed very helpful to increase the knowledge on the subject and to properly assess the classical correlations proposed during the years.

This paper focuses on R134a flow boiling inside a new geometry microfin tube, having a fin tip diameter of 4.3 mm. Experimental measurements were collected at several mass velocities, from 100 kg m^{-2} s^{-1} to 800 kg m^{-2} s^{-1}, heat fluxes from 15 kW m^{-2} to 90 kW m^{-2}, and by keeping the mean saturation temperature equal to 30°C. The new data sets permit to investigate the vapor quality, heat flux, and mass velocity effects on heat transfer coefficient and pressure drop. Furthermore, several literature correlations were assessed against the experimental database to test the suitability of the most common models also in this microfin geometry. Finally, the performance of the tested microfin tube was compared against that of an equivalent smooth tube.

Experimental apparatus and test section

Figure 1 reports a schematic of the experimental facility, which consists of three circuits: the green lines refer to the refrigerant loop, the blue ones to the cold water loop, the red ones to the hot water loop. The set up was meant to perform flow boiling heat transfer and pressure drop measurements of pure refrigerants and refrigerants mixtures inside structured geometries.

Starting from the refrigerant circuit (green lines), the subcooled liquid is circulated by a variable speed volumetric gear pump, it flows through a Coriolis effect mass flow meter, and then it is partially vaporized in a Brazed Plate Heat Exchanger (BPHE) fed with hot water (red lines).

The two-phase mixture leaves the BPHE evaporator and it reaches the test section at known mass velocity and vapor quality. There, it is vaporized by the power generated by a calibrated Ni-Cr wire resistance. The electrical power is supplied by a DC power generator rated up to 900 W. The electrical power supplied to the microfin tube is estimated by means of a calibrated reference resistance (shunt) used to measure the electrical current and by the measurement of the effective electrical difference potential of the resistance wire located in the copper heater. Finally, the refrigerant is condensed and subcooled by the tap water (blue lines) flowing in a dedicated post-condenser, another BPHE. The hot water loop (red lines) consists of a thermostatic bath, which permits to set both the water temperature and the water flow rate at the evaporator inlet. A magnetic flow meter and a calibrated T-type thermopile allow for an accurate estimation of the heat exchanged in the BPHE evaporator.

Tests were run to verify the heat balance between refrigerant and water sides, the results showed a misbalance always less than 2%. The refrigerant saturation pressure in the loop is controlled by the amount of refrigerant charge and by a damper connected to a compressed air line, which operates as pressure regulator. As highlighted in the schematic reported in Figure 1, refrigerant pressure and temperature are measured at several locations throughout the circuit to know the refrigerant properties at the inlet and outlet of each heat exchanger.

No oil circulates in the refrigerant loop. Table 1 lists the values of maximum uncertainty (k=2) of the instruments used in the experimental facility.

Figure 2 shows a drawing of the test copper plate; the microfin tube was brazed inside the guide milled on the top surface of the plate, which is 8 mm deep. As clearly shown, 16 holes, equally spaced, were drilled just 1 mm below the microfin tube, in order to locate as many T-type thermocouples to monitor the wall temperature distribution. Besides, as depicted in the additional section B of the test section, the distance between the thermocouples' junctions and the tube wall is 0.5 mm. Another guide was milled on the bottom side of the copper plate, to host a Nickel-Chrome wire resistance connected to the DC current generator.

Particular attention was dedicated to the design of the fittings used to connect the inlet and outlet pipes and the test section. In fact, a suitable smooth connection to the refrigerant circuit having the same fin tip diameter (D=4.3 mm) was realized to avoid any additional abrupt pressure drops. Pressure ports are located around 25 mm downstream and upstream of the copper plate, thus the length for pressure drop measurements is 250 mm. The test section assembly was then located inside an aluminum housing filled with 15 mm thick ceramic fiber blanket, to reduce the heat losses due to conduction to the surroundings.

Considering the tested mini microfin tube, has n=54 fins, which are h=0.12 mm high, while the helix angle β is 27°, and the apex one γ is around 11°. Given the reported dimensions, the area enhancement with reference to the smooth tube having the same fin tip diameter is equal to 1.87. Figure 3 reports two photos of the tested microfin tube where one can clearly observe a cross section of the microfins (a) and the helical fins (b).

Data reduction

The two-phase heat transfer coefficient *HTC*, referred to the nominal area *A*, can be defined as:

$$HTC = \frac{q_{TS}}{A \cdot (\bar{t}_{wall}\text{-}\bar{t}_{sat})} = \frac{q_{TS}}{\pi \cdot D \cdot L \cdot (\bar{t}_{wall}\text{-}\bar{t}_{sat})} \tag{1}$$

where, the nominal area *A* is the area of an equivalent smooth tube having the inner diameter equal to the tube fin tip of the microfin tube, *D*. $\bar{t}_{wall}$ is the average value of the 16 measured wall temperatures $t_{wall,i}$ as:

$$\bar{t}_{wall} = \frac{1}{16}\sum_{i=1}^{16} t_{wall,i} \tag{2}$$

while the average value of the saturation temperature $\bar{t}_{sat}$ is calculated from the measured values of the pressure:

$$\bar{t}_{sat} = \frac{t_{sat,in}(p_{sat,in}) + t_{sat,out}(p_{sat,out})}{2} \tag{3}$$

Finally, the actual heat flow rate exchanged in the test section, q_{TS}, is calculated from the electrical one, P_{EL}, by subtracting the heat losses to the surroundings.

Preliminary tests under vacuum conditions inside the refrigerant channel permitted to evaluate the heat loss (q_{loss}) to the environment. This parameter was found to be a linear function of the mean wall temperature. The measurements were run by varying the mean wall temperature from 28°C to 63°C. In this wall temperature range, the heat loss can be estimated by:

$$|q_{loss}| = 0.2006 \cdot \bar{t}_{wall}\ [°\text{C}] - 4.6698\ \ [\text{W}] \tag{4}$$

The actual heat flow rate q_{TS} supplied to the sample is given by:

$$q_{TS} = P_{EL}\text{-}|q_{loss}| = \Delta V \cdot I\text{-}|q_{loss}| \tag{5}$$

The refrigerant enters the test section at an imposed vapor quality, this value can be evaluated from a thermal balance at the brazed plate heat exchanger, which acts as an evaporator:

$$q_{evap} = \dot{m}_w \cdot c_{p,w} \cdot \left(t_{w,in}\text{-}t_{w,out}\right) = \dot{m}_r \cdot \left(J_{in,TS}\text{-}J_{L,sub}\right) \tag{6}$$

where $\dot{m}_w$ is the water mass flow rate, $c_{p,w}$ is the specific heat capacity of the water, and $t_{w,in}$ and $t_{w,out}$ are the inlet and outlet water temperatures. The right-hand side term of eq. (6) reports the refrigerant side heat flow rate where $J_{in,TS}$ and $J_{L,sub}$ are specific enthalpies at the inlet of the test section and of the subcooled liquid entering the BPHE, respectively. Once calculated $J_{in,TS}$, the vapor quality at the inlet of the test section can be estimated by:

$$x_{in,TS} = \frac{J_{in,TS}\text{-}J_L}{J_V\text{-}J_L} \tag{7}$$

where J_L and J_V are the specific enthalpies of the saturated liquid and vapor, respectively, evaluated at the refrigerant saturation pressure measured at the inlet of the test section.

It is worth underlying that q_{loss} varied from 2.5% to 4% of the electrical power supplied. The specific enthalpy at the outlet of the test section can be calculated from:

$$q_{TS} = \dot{m}_r \cdot \left(J_{out,TS}\text{-}J_{in,TS}\right) \tag{8}$$

The outlet vapor quality $x_{out,TS}$ is given by:

$$x_{out,TS} = \frac{J_{out,TS} - J_L}{J_V - J_L} \tag{9}$$

where J_L and J_V are the specific enthalpies of the saturated liquid and vapor, respectively, evaluated at the saturation pressure of the refrigerant measured at the outlet of the test section. The mean vapor quality, x_{mean} is the average value between the inlet and outlet ones.

The frictional pressure drop exploited during the two-phase flow inside the microfin tube was calculated from the measured total pressure drop by subtracting the momentum and the gravitational pressure gradients, as:

$$\Delta p_{\mathrm{f}} = \Delta p_{\mathrm{t}} - \Delta p_{\mathrm{c}} - \Delta p_{\mathrm{a}} \tag{10}$$

The momentum pressure drops are estimated by the homogeneous model for two-phase flow as follows:

$$\Delta p_{\mathrm{a}} = G^2(v_{\mathrm{V}} - v_{\mathrm{L}})\ |\Delta x| \tag{11}$$

where v_L and v_V are the specific volumes of liquid and vapor phases, whereas $|\Delta x|$ is the absolute value of the vapor quality change through the whole test section. The gravitational contribution Δp_c was kept equal to 0 Pa, because the microfin tube is horizontal. All the thermodynamic and transport were estimated by RefProp v9.1 (Lemmon *et al.*, 2013).

For the sake of clarity, it has to be specified that the mass velocity G and the heat flux HF are referred to the cross sectional area and to the heat transfer area, respectively, of an equivalent smooth tube having an internal diameter equal to the diameter at the fin tip of the microfin tube under investigation. A detailed error analysis was performed in accordance with Kline and McClintock (1953) using the values of the instruments uncertainties (k=2) listed in Table 1; the uncertainty (k=2) on the two-phase heat transfer coefficient showed a mean value of ±2.1% and a maximum value of ±3.8%, while the uncertainty on the vapor quality was ±0.03. The pressure drops showed a mean uncertainty of ±8%.

Experimental Results

This section presents the experimental results collected during vaporization of the R134a inside the mini microfin tube at a mean saturation temperature of 30°C. This value can be considered suitable for tap water heat pumps, civil and industrial driers, and for electronics cooling applications, in which R134a is the most common used refrigerant. The effects of the most important operating test conditions were investigated: mean vapor quality x_{mean}, mass velocity G, and heat flux HF.

The mean vapor quality x_{mean} was varied from 0.1 to 0.95, the mass velocity from 100 kg m^{-2} s^{-1} to 800 kg m^{-2} s^{-1}, and the heat flux from 15 kW m^{-2} to 90 kW m^{-2}. At these operating conditions, the vapor quality change through the test section varied from 0.02 to 0.32. The value of Δx=0.32 was considered the maximum acceptable to allow a proper comparison among the collected data; thus, when increasing the heat flux, one or more refrigerant mass velocities were not collected because they would have presented a higher vapor quality change.

Figure 4 shows the effect of mass velocity G on the heat transfer coefficient as a function of the mean vapor quality x_{mean} at the four investigated heat fluxes: HF=15 kW m^{-2} (a), HF=30 kW m^{-2} (b), HF=60 kW m^{-2} (c), and HF=90 kW m^{-2} (d). At these operating test conditions, the vapor quality change between inlet and outlet of the test section vary from 0.02 and 0.16 at HF=15 kW m^{-2}, from 0.04 to 0.16 at HF=30 kW m^{-2}, from 0.08 to 0.32 at HF=60 kW m^{-2}, and from 0.12 to 0.25 at HF=90 kW m^{-2}.

Starting from the lowest heat flux, HF=15 kW m^{-2} (Figure 4a) and from the lowest mass velocity, G=100 kg m^{-2} s^{-1}, one can see that the heat transfer coefficient remains almost constant at around 8000 W m^{-2} K^{-1} up to a mean vapor quality of 0.5, meaning that the nucleate boiling seems to control the phase change process, whereas at higher vapor qualities it increases, and thus, also the two-phase forced convection starts to play a relevant role in the flow boiling phenomenon.

When increasing the mass velocity (Figure 4a), the plateau at low vapor quality, where the heat transfer coefficient remains constant, quickly disappears, and the heat transfer coefficient increases almost linearly with the vapor quality, meaning that the two-phase forced convection is becoming the most affecting phase-change mechanism. It is worth pointing out that for x_{mean}<0.3, all the investigated mass velocities show similar values of heat transfer coefficient. Furthermore, at x_{mean}>0.65, the values of heat transfer coefficient measured at G=200 kg m^{-2} s^{-1} and G=400 kg m^{-2} s^{-1} are greater than those obtained at higher mass velocities. Moreover, at 15 kW m^{-2}, the effect of the mass velocity is only noticeable when passing from 100 kg m^{-2} s^{-1} to 200 kg m^{-2} s^{-1}, where the heat transfer coefficients increase. This can be linked to a particular effect induced by the helical microfins that might be emphasized at these operating test conditions. Furthermore, this behavior can be also linked to the fact that when increasing the mass velocity over 400 kg m^{-2} s^{-1}, the two-phase pressure drops also increase, which imply a saturation temperature drop with a consequent penalization of the heat transfer. This behavior was also found by Mancin *et al.* (2015) during R134a flow boiling inside another mini microfin tube an internal diameter at the fin tip of 3.4 mm and by and Jige et al. (2016) for R32 boiling inside a 2.6 mm microfin tube. The onset of the dryout was only observed at G=100 kg m^{-2} s^{-1} and G=200 kg m^{-2} s^{-1} and it occurred at around x_{mean}=0.83 and x_{mean}=0.91, respectively.

At HF=30 kW m^{-2}, only the data taken at G=100 kg m^{-2} s^{-1} show a well-defined plateau where the heat transfer coefficient can be considered fairly constant; at all the other mass velocities, the heat transfer coefficient increases with the vapor quality but the slope of the experimental trends is lower as compared to the data collected at HF=15 kW m^{-2}. Moreover, at low vapor quality, the heat transfer coefficients are almost the same at all the investigated mass velocities. At high vapor quality, the forced convection remains the dominant heat transfer mechanism and, again, G=400 kg m^{-2} s^{-1} displays the highest heat transfer coefficients. The onset of the dryout was only observed at G=100 kg m^{-2} s^{-1} and G=200 kg m^{-2} s^{-1} and it occurred at around x_{mean}=0.67 and x_{mean}=0.77, respectively.

As reported in Figure 4c, when increasing the heat flux to HF=60 kW m^{-2} slightly different results can be highlighted. The two-phase heat transfer coefficient trends do not indicate any noticeable effect of the mass velocity: hence, the heat transfer mechanism seems to be controlled by the nucleate boiling. For x_{mean} < 0.5 the heat transfer coefficient (being around 11000 W m^{-2} K^{-1}), is almost constant with vapor quality at all the investigated mass velocities. As the vapor quality increases, the heat transfer coefficient slightly increases showing an even more reduced slope as compared to HF=30 kW m^{-2}. The dryout phenomenon was only observed at G=200 kg m^{-2} s^{-1} and the mean vapor quality at the dryout onset is around x_{mean}=0.75.

Finally, Figure 4d presents the heat transfer coefficients collected at HF=90 kW m^{-2}, the heat transfer is controlled by the nucleate boiling, the heat transfer coefficients are almost constant at around 13000 W m^{-2} K^{-1} up to x_{mean}=0.65, the effect of the vapor quality is still present but further weakens as compared to lower heat fluxes. There is no noticeable effect of the mass velocity. The dry out was only observed at G=400 kg m^{-2} s^{-1} and the mean vapor quality at the onset of the dryout is around x_{mean}=0.77.

In order to further analyze the boiling behavior of this microfin tube, the measurements can be presented to show the effects of the heat flux on the flow boiling; Figure 5 reports the heat transfer coefficient plotted against the mean vapor quality at four different mass velocity: 200 kg m^{-2} s^{-1} (a), 400 kg m^{-2} s^{-1} (b), 600 kg m^{-2} s^{-1} (c), and 800 kg m^{-2} s^{-1}.

Figure 5a reports the data collected at G=200 kg m^{-2} s^{-1}, for heat fluxes lower than 30 kW m^{-2}, the heat transfer coefficient increases with vapor quality and the values measured at x_{mean}<0.6 are almost the same. When increasing the vapor quality, the experimental measurements collected at HF=15 kW m^{-2} show higher heat transfer coefficients compared to those measured at HF=30 kW m^{-2}. When increasing the heat flux, a constant value for the heat transfer coefficient is observed up to the dryout onset, meaning that the nucleate boiling is the dominant heat transfer mechanism. Considering the results depicted in Figure 5b, it can be stated that at heat flux lower than 30 kW m^{-2}, there is no noticeable effect of this parameter on the boiling heat transfer. In fact, for vapor qualities lower than 0.4, the heat transfer coefficients are almost the same; then, for x_{mean}>0.4, the heat transfer coefficient increases and those measured at HF=15 kW m^{-2} become even slightly higher than those collected at HF=30 kW m^{-2}. It is worth highlighting that when increasing the heat flux, the plateau where the heat transfer coefficient can be considered almost constant is extended to higher vapor qualities (i.e. to x_{mean}=0.5 and x_{mean}=0.65 for HF=60 kW m^{-2} and HF=90 kW m^{-2}, where the heat transfer coefficients are around 11400 W m^{-2} K^{-1} and 12900 W m^{-2} K^{-1}, respectively).

At higher vapor qualities, the heat transfer coefficient slightly increases. Moreover, at x_{mean}<0.6, for a given vapor quality, the heat transfer coefficient increases as the heat flux increases, especially for HF>30 kW m^{-2}. At these operating conditions, the nucleate boiling can be considered the prevailing phase change mechanism. When the vapor quality becomes higher than 0.65, the heat transfer coefficient profiles converge exhibiting almost the same values. Finally, the dryout was only observed at *HF*=90 kW m^{-2} confirming the interesting capabilities of the mini microfin tube in delaying the onset of dryout; this feature is particularly suitable for electronics cooling application where the dryout event and the consequent sharp surface temperature increase must be avoided.

The results observed at low heat fluxes in Figures 5a and 5b can be explained considering that, at these operating conditions, the dominant heat transfer mechanism is the forced convection, which might be also positively influenced by the turbulence induced by the helical microfins. Similar considerations can be drawn when considering the results plotted in Figures 5c and 5d: there is not any noticeable effect of the heat flux up to *HF*=30 kW m^{-2}. The heat transfer coefficients measured at *HF*=15 kW m^{-2} and *HF*=30 kW m^{-2} are similar and they increase with vapor quality. The two-phase forced convection seems to control the boiling process. When comparing the data measured at higher heat fluxes with that for *G*=200 kg m^{-2} s^{-1} (Figure 5a) and *G*=400 kg m^{-2} s^{-1} (Figure 5b), it can be stated that due to the high mass velocity, the plateau where the heat flux can be considered almost constant slightly recedes to lower vapor quality, meaning that the nucleate boiling is quickly overcome by the convective boiling heat transfer mechanism. In this case, the heat transfer coefficients measured at *HF*=90 kW m^{-2} are higher than those measured at lower heat fluxes. This confirms what highlighted before and also stated by Mancin *et al.* (2015) and Jige *et al.* (2016), i.e. it seems that there is a mass velocity range, in which the favorable characteristics of the helical microfin tube are even more effective leading to very high boiling heat transfer performance. On the basis of the present work and of Mancin *et al.* (2015) and Jige *et al.* (2016), it appears that this mass velocity range depends on the tube geometry and the refrigerant type.

The experimental frictional pressure gradients are plotted in Figure 6 as a function of the mean vapor quality. As expected, it was found that the frictional pressure gradients did not depend from the imposed heat flux; thus, for the sake of clarity, only the data relative to HF=60 kW m^{-2} was plotted. As described before, the homogeneous model was considered to estimate the momentum pressure drops, which were subtracted from the total measured pressure drops.

The results show that, at constant mass velocity, the frictional pressure gradient increases with vapor quality. Furthermore, at constant vapor quality, the frictional pressure gradient increases as the mass velocity increases; similar results were also found by Mancin *et al.* (2015), Diani *et al.* (2014, 2015, and 2016).

Models' Assessment

This section presents the comparison between the values obtained by applying different models for the estimation of flow boiling heat transfer coefficients and two-phase frictional pressure gradients and those experimentally measured. The models proposed by Hamilton *et al.* (2008), Padovan *et al.* (2011), Wu *et al.* (2013), Diani *et al.* (2014), and the recent one by Rollman and Splider (2016) were chosen for the flow boiling heat transfer coefficients assessment. The relative and absolute deviations exhibited by the selected models are listed in Table 3.

The best prediction capabilities were highlighted by the model proposed by Padovan *et al.* (2011) which showed a relative deviation of -6.4% and an absolute deviation of 20.4%. This model is an updated version of that proposed by Cavallini *et al.* (1999) to account for the capillary effects at very low refrigerant mass velocity. The recent models by Rollman and Spindler (2016) and by Diani *et al.* (2014) are even accurate, despite their relative and absolute deviations are higher if compared to those of Padovan *et al.* (2011).

Figure 7 presents the comparisons between the experimental and calculated flow boiling heat transfer coefficients obtained from Padovan *et al.* (2011) model, by subdividing the data as a function of the applied heat flux, *HF*.

The diagram permits to analyze the effect of the imposed heat flux on the prediction capabilities of the Padovan *et al.* (2011) model. In fact, the model estimates almost all the experimental measurements within ±30%; moreover, only the data points collected at 15 kW m^{-2} are slightly underestimated whereas, for higher heat fluxes, the experimental flow boiling heat transfer coefficients are fairly estimated.

The models proposed by Haraguchi *et al.* (1993), Kedzierski and Goncalves (1999), Cavallini *et al.* (2000), Goto *et al.* (2001), Newell and Shah (2001), Oliver *et al.* (2004), Bandarra Filho *et al.* (2004), Han and Lee (2005), Afroz and Miyara (2011), Diani *et al.* (2014), and Rollman and Splinder (2016) were then selected to be compared against the experimental two-phase frictional pressure gradients. The models were computed following the integral method as suggested by Mauro *et al.* (2007). The values of relative and absolute deviations of the applied models for frictional pressure drops are listed in Table 4, whereas Figure 8 reports the comparison between experimental and calculated frictional pressure gradients obtained by the four cited best models.

As an outcome, a few of the selected models fairly predict the experimental frictional pressure gradients; when combining the two statistical parameters, i.e. relative and absolute deviations, the best predictions are exhibited by the models proposed by Kedzierski and Goncalves (1999) and by Cavallini *et al.* (2000), which were developed for traditional macro microfin tubes. The model by Diani *et al.* (2014), which represents an updated version of the model proposed by Cavallini et al. (2000) to account for the effects of the tube size, and the recent procedure proposed by Rollman and Splinder (2016) also show good predictions.

Comparison with a reference smooth tube

The proper use of an enhanced surface passes through the analysis of its actual heat transfer enhancement compared to the unavoidable pressure loss penalization. This consideration can also be applied to microfin tubes and it becomes even more important when considering the flow boiling heat transfer. As previously highlighted, the prevailing heat transfer mechanism depends on the operating conditions: mass velocity, vapor quality, reduced pressure, and heat flux. For this reason, for each combination of the mentioned parameters, the actual performance of the tube might be enhanced or penalized by the presence of the micro-fins.

In order to evaluate the effective performance of this microfin tube, it is possible to compare the measured flow boiling heat transfer coefficients and frictional pressure gradients with those calculated for an equivalent smooth tube having the inner diameter equal to the fin tip diameter of the microfin tube, i.e. D=4.3 mm. The selection of the proper models to be applied to calculate the smooth tube reference values represents the key issue of this procedure.

Recently, the present authors (Longo *et al.*, 2016) measured flow boiling heat transfer coefficients and frictional pressure drops of R134a inside a 4 mm internal diameter horizontal smooth tube. The flow boiling measurements were conducted by varying the refrigerant mass velocity from 200 kg m^{-2} s^{-1} to 600 kg m^{-2} s^{-1}, the vapor quality from 0.11 to 0.96, the heat flux from 10 W m^{-2} to 30 W m^{-2}, at three different mean saturation temperatures of 10°C, 15°C, and 20°C, respectively. The authors conducted a models' assessment and the results demonstrated that the procedures proposed by Kim and Mudawar (2014) for flow boiling heat transfer coefficient and by Friedel (1979) for frictional pressure gradients exploited the best agreements when compared to the experimental database.

Accordingly, these two models were selected to estimate heat transfer coefficients and frictional pressure gradients during R134a flow boiling inside a reference smooth tube with an inner diameter *D* equal to 4.3 mm at the Same Operating Conditions (S.O.C.) of the microfin experimental data presented in this paper.

Then, the calculated values were used to define two different parameters, as done also by Diani *et al.* (2016), the Enhancement Factor EF and Pressure Drop Ratio PDR, as:

$$\mathrm{EF} = \left(\frac{HTC_{Microfin}}{HTC_{smooth}}\right)_{S.O.C.} \tag{12}$$

and

$$\mathrm{PDR} = \left[\frac{\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{f,Microfin}}{\left(\frac{\mathrm{d}p}{\mathrm{d}z}\right)_{f,smooth}}\right]_{S.O.C.} \tag{13}$$

According to eqs. 14 and 15, for given operating test conditions, the higher the EF and the lower the PDR, the better the performance of the microfin tube as compared to the smooth one. Another important consideration to be pointed out is related to the area enhancement caused by the presence of microfins on the inner tube surface. For the tested tube this area enhancement is equal to 1.87. This means that an EF less than 1.87 does not represent an optimal solution, and that only EF values higher than 1.87 can be considered acceptable.

Figure 9 reports the EF plotted against the vapor quality as a function of the mass velocity; the figure is subdivided in four diagrams reporting the data collected at the four different heat fluxes: (a) *HF*=15 kW m^{-2}, (b) *HF*=30 kW m^{-2}, (c) *HF*=60 kW m^{-2}, and (d) *HF*=90 kW m^{-2}.

The results plotted in Figure 9a highlight that at *HF*=15 kW m^{-2}, the EF is always greater than the area enhancement value (i.e., 1.87) and it increases with the vapor quality; moreover, as already described, at 200 kg m^{-2} s^{-1} and 400 kg m^{-2} s^{-1} the values of the EF are greater than those measured at higher mass velocities. When increasing the heat flux up to 30 kW m^{-2} (Figure 9b), the EF is less affected by the mass velocity, and it is lower than the area enhancement for vapor quality less than 0.8. Only at higher vapor qualities, the *EF* becomes equal and sometimes slightly higher than 1.87.

At heat fluxes equal to 60 kW m^{-2} and 90 kW m^{-2} (Figures 9c and 9d, respectively) the EF is almost constant being around 1 and it seems to slightly increase for vapor qualities greater than 0.5; in any case, it remains remarkably below the area enhancement, 1.87.

Generally speaking, when only the heat transfer performance is considered, the microfin tubes should be used for low heat fluxes and low mass velocities, when the forced convective boiling is enhanced by the presence of the micro-fins.

Figure 10 reports the Pressure Drop Ratio (PDR) calculated as defined in eq. 15; it is worth underlying that it has been chosen to report only the data collected at 60 kW m^{-2} because, as already stated, the frictional pressure gradients do not depend on the imposed heat flux. As expected, the PDR values are always greater than 1, with values ranging between 1.15 to 1.7. The lowest values are shown by the lowest mass velocity, *G*=200 kg m^{-2} s^{-1} while the highest are obtained at *G*=400 kg m^{-2} s^{-1}. For mass velocities up to 600 kg m^{-2} s^{-1}, the estimated trends present a maximum at a mean vapor quality of around 0.5, while at *G*=800 kg m^{-2} s^{-1} the PDR is almost constant, being around 1.3-1.4.

When comparing the results plotted in Figure 9 and 10, it clearly appears that the maximum enhancement is achieved at low heat fluxes where, with the present geometrical configuration, the microfins enhance the forced convective boiling heat transfer mechanism rather than the nucleate boiling one. Similar results were also found by Wu et al. (2013) and Diani et al. (2016).

Conclusions

This paper presents experimental heat transfer coefficients and pressure drops measured during flow boiling inside a mini microfin tube with an internal diameter at the fin tip of 4.3 mm. Tests were run at a constant mean saturation temperature of 30°C, by varying the vapor quality from 0.1 to 0.95, the mass velocity from 100 kg m^{-2} s^{-1} to 800 kg m^{-2} s^{-1}, and the heat flux from 15 kW m^{-2} to 90 kW m^{-2}. The results confirm that the heat transfer process is controlled by the two mechanisms that govern the flow boiling phenomenon, i.e. nucleate boiling and two-phase forced convection, and that the prevailing one depends upon the actual operating test conditions.

In general, it can be stated that at low heat fluxes, the heat transfer coefficient is highly affected by the vapor quality, meaning that the convective boiling dominates the flow boiling phenomenon. A different situation occurs at high heat fluxes, where the heat transfer coefficient is negligibly affected by mass velocity and just weakly affected by vapor quality, meaning that the phase change process is mainly controlled by nucleate boiling. The two-phase frictional pressure drops were also measured. They increase with both mass velocity and vapor quality.

Several models were selected, implemented, and then compared against the experimental measurements of boiling heat transfer coefficient and frictional pressure gradient; the models proposed by Padovan *et al.* (2011) and by Kedzierski and Goncalves (1999) can be suggested for the boiling heat transfer coefficient and frictional pressure drop estimations, respectively.

Finally, the comparison between the performance of the microfin tube and that of an equivalent smooth tube highlighted that the enhanced solution is worth of interest especially at low heat fluxes and mass velocities.

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Nomenclature

A	=	fin tip area	(m^2)	
BPHE	=	Brazed Plate Heat Exchanger		
c_p	=	specific heat capacity	(J kg^{-1} K^{-1})	
D	=	fin tip diameter	(m)	
(dp/dz)	=	pressure gradient	(Pa m^{-1})	
EF	=	enhancement Factor	(-)	
f	=	friction factor	(–)	
FS	=	Full Scale		
G	=	mass velocity	(kg m^{-2} s^{-1})	
GWP	=	global warming potential	(-)	
g	=	gravity acceleration	(m s^{-2})	
h	=	fin height	(m)	
HF	=	Heat Flux	(W m^{-2})	
HTC	=	Heat Transfer Coefficient	(W m^{-2} K^{-1})	
I	=	current	(A)	
ID	=	Inner Diameter	(m)	
J	=	specific enthalphy	(J kg^{-1})	
J_g	=	non-dimensional gas velocity	(–)	
J_g^T	=	transition gas velocity	(–)	
k	=	thermal conductivity	(W m^{-1} K^{-1})	
k	=	coverage factor	(–)	
L	=	tube length	(m)	
$\dot{m}$	=	mass flow rate	(kg s^{-1})	
n	=	fin number	(m)	
OD	=	Outer Diameter	(m)	
p	=	pressure	(Pa)	
PDR	=	pressure drop ratio	(-)	
q	=	heat flux	(W m^{-2})	
t	=	temperature	(°C)	
$\bar{t}$	=	average temperature	(°C)	$\bar{t}$
v	=	specific volume	(m^3 kg^{-1})	
V	=	voltage	(V)	
x	=	vapor quality	(–)	
X_{tt}	=	Martinelli parameter	(–)	

Greek symbols

β	=	helix angle	(°)
γ	=	apex angle	(°)
Δ	=	difference	(–)
μ	=	dynamic viscosity	(Pa s)
ρ	=	density	(kg m^{-3})

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a	=	momentum
c	=	gravitational
EL	=	electric

f	=	frictional
fin	=	fin
in	=	inlet
L	=	liquid
loss	=	loss
mean	=	mean
out	=	outlet
r	=	refrigerant
sat	=	saturation
S.O.C	=	Same Operating Conditions
sub	=	subcooled
t	=	total
tp	=	two phase
TS	=	Test Section
V	=	vapor
w	=	water
wall	=	wall

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Table 1. Instruments uncertainty.	
Transducer	**Uncertainty (*k*=2)**
T-type thermocouples	± 0.1 K
T-type thermopiles	± 0.05 K
Electric power	± 0.26% of the reading
Coriolis mass flowmeter (refrigerant loop)	± 0.10% of the reading
Magnetic volumetric flowmeter (hot water loop)	± 0.2% of FS= 0.33 10^{-3} m^3 s^{-1}
Differential pressure transducer (test section)	± 0.075% of 0.3 MPa
Absolute pressure transducers	± 0.065% of FS= 4 MPa

Table 2. Main tube characteristics.	
Parameter	**Nominal**
Outer Diameter, OD, (mm)	5.0
Inner Diameter at the fin tip, *D* (mm)	4.3
Tube Thickness, *t* (mm)	0.23
Number of fins, *n* (-)	54
Fin Height, *h* (mm)	0.12
Apex angle, γ (°)	11
Helix angle, β (°)	27

Table 3. Relative and absolute deviations of the selected models for flow boiling heat transfer coefficient calculation.		
Parameter	**Deviation**	
	Relative	**Absolute**
Hamilton *et al.* (2008)	-23.9	60.2
Padovan *et al.* (2011)	-6.4	20.4
Wu *et al.* (2013)	59.4	67.9
Diani *et al.* (2014)	24.4	25.7
Rollman and Spindler (2016)	16.4	32.7

Table 4. Relative and absolute deviations of the selected models for two-phase frictional pressure gradient calculation.		
Parameter	**Deviation**	
	Relative	**Absolute**
Haraguchi *et al.* (1993)	8.3	24.4
Kedzierski and Goncalves (1999)	-3.4	18.3
Cavallini *et al.* (2000)	-3.5	19.3
Goto *et al.* (2001)	28.9	36.6
Newell and Shah (2001)	-7.2	23.7
Bandarra Filho *et al.* (2004)	29.5	38.0
Oliver *et al.* (2004)	-27.2	31.6
Han and Lee (2005)	-18.5	36.9
Afroz and Miyara (2011)	-30.7	32.1
Diani *et al.* (2014)	-2.7	22.8
Rollman and Spindler (2016)	-2.4	20.3

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Figure 1: Schematic of the experimental setup.

Figure 2: Drawing of the copper plate with the locations of the thermocouples.

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Figure 4: Effect of mass velocity on heat transfer coefficient at four different heat fluxes: HF=15 kW m^{-2} (a), HF=30 kW m^{-2} (b), HF=60 kW m^{-2} (c), and HF=90 kW m^{-2} (d). G expressed in [kg m^{-2} s^{-1}].

Figure 5: Effect of heat flux on heat transfer coefficient at four different mass velocities: G=200 kg m^{-2} s^{-1} (a), G=400 kg m^{-2} s^{-1} (b), G=600 kg m^{-2} s^{-1} (c), and G=800 kg m^{-2} s^{-1} (d). HF expressed in [kW m^{-2}].

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Figure 7: Calculated vs. experimental heat transfer coefficients. Model by Padovan et al. (2011).

Figure 8: Calculated vs. experimental frictional pressure gradients. Models by: (a) Kedzierski and Goncalves (1999), (b) Cavallini et al. (2000), (c) Diani et al. (2014), and (d) Rollman and Splinder (2016).

Figure 9: Enhancement Factor plotted against the vapor quality as a function of the mass velocity at different heat fluxes: (a) HF=15 kW m^{-2}, (b) HF=30 kW m^{-2}, (c) HF=60 kW m^{-2}, and (d) HF=90 kW m^{-2}. G expressed in [kg m^{-2} s^{-1}].

Figure 10: Pressure Drop Ratio (PDR) plotted against the vapor quality as a function of the mass velocity. G expressed in [kg m^{-2} s^{-1}].