R245fa Flow Boiling inside a 4.2 mm ID Microfin Tube

G A Longo, S Mancin, G Righetti and C Zilio

Dept. of Management and Engineering, University of Padova, Str.lla S. Nicola 3, 36100, Vicenza, IT

E-mail: tony@gest.unipd.it

Abstract. This paper presents the R245fa flow boiling heat transfer and pressure drop measurements inside a mini microfin tube with internal diameter at the fin tip of 4.2 mm, having 40 fins, 0.15 mm high with a helix angle of 18°. The tube was brazed inside a copper plate and electrically heated from the bottom. Sixteen T-type thermocouples are located in the copper plate to monitor the wall temperature. The experimental measurements were carried out at constant mean saturation temperature of 30 °C, by varying the refrigerant mass velocity between 100 kg m⁻² s⁻¹ and 300 kg m⁻² s⁻¹, the vapour quality from 0.15 to 0.95, at two different heat fluxes: 30 and 60 kW m⁻². The experimental results are presented in terms of two-phase heat transfer coefficient, onset dryout vapour quality, and frictional pressure drop. Moreover, the experimental measurements are compared against the most updated models for boiling heat transfer coefficient and frictional pressure drop estimations available in the open literature for microfin tubes.

1. 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. [1]. 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 [2-3]. 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 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. [4] 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. [5] 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. [6] analysed 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. [7] during R1234ze(E), R1234ze(Z), and R134a condensation and vaporization. During flow boiling tests, the heat flux was set 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. [8] tested R134a, Diani et al. [9] presented R1234ze(E) data, and Diani et al. [10] 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 [11], and R1234yf [12] 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. [13] 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. [14] 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 R245fa flow boiling inside a new geometry microfin tube, having a fin tip diameter of 4.2 mm. Experimental measurements were collected at several mass velocities, from 100 kg m$^{-2}$s$^{-1}$ to 300 kg m$^{-2}$s$^{-1}$ at two different heat fluxes 30 kW m$^{-2}$ and 60 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.
2. Experimental setup and data reduction

As shown in Figure 1, the experimental test 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- and nano-geometries. The refrigerant is pumped through the circuit by means of a variable speed volumetric gear pump, then it is vaporized in a Brazed Plate Heat Exchanger (BPHE) fed with hot water to achieve the desired value of vapour quality. The hot water is supplied by a thermostatic bath; both water flow rate and water temperature can be independently set. The heat flow rate exchanged at the BPHE evaporator is accurately measured by means of a magnetic flow meter and a calibrated T-type thermopile; furthermore, preliminary tests were run to verify the heat balance between refrigerant and water sides, the results showed a misbalance always less than 2%. The refrigerant enters the microfin test tube at a known mass velocity and vapour quality and then it is vaporized by means of a calibrated Ni-Cr wire resistance.

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 electrical difference potential of the resistance wire inserted in the copper heater. The current can be calculated from the Ohm’s law. The fluid leaves the test section and enters in a post-condenser, a brazed plate heat exchanger fed with tap water, where it is fully condensed and subcooled. A damper connected to the compressed air line operates as pressure regulator to control the saturation conditions in the refrigerant loop.

![Figure 1. Experimental setup.](image)

| 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 (loop)    | ± 0.10% of the reading            |
| Magnetic volumetric flowmeter     | ± 0.2% of FS= 0.33 $10^{-3}$ m$^3$ s$^{-1}$ |
| Differential pressure transducer  | ± 0.075% of 0.3 MPa               |
| Absolute pressure transducers     | ± 0.065% of FS= 4 MPa             |
As shown in Figure 1, the 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. The refrigerant mass flow rate can be independently controlled by the gear pump and it is measured by means of a Coriolis effect flowmeter. No oil circulates in the refrigerant loop. Table 1 lists the values of uncertainty (coverage factor, $k=2$) of the instruments used in the experimental facility. The mini microfin tube was brazed inside a guide milled on the top surface of a copper plate, which is 200 mm long, 10 mm wide, and 20 mm high. 16 holes were drilled just 1 mm below the microfin tube, in order to locate as many T-type thermocouples to monitor the wall temperature distribution. Another guide was milled on the bottom side of the copper plate, to host a Nickel-Chrome wire resistance connected to a DC current generator, which supplies the heat flow rate needed to vaporize the refrigerant flowing inside the tube. In order to avoid the abrupt pressure drops due to flow contraction and expansion, a suitable smooth connection to the refrigerant circuit having the same fin tip diameter ($D=4.2$ mm) was designed and realized to join the test tube with inlet and outlet pipes. Pressure ports are located about 25 mm downstream and upstream of the copper plate, thus the length for pressure drop measurements is 250 mm. The test section is located inside an aluminum housing filled with 15 mm thick ceramic fiber blanket, to minimize the heat losses due to conduction to the ambient. Figure 2 and Table 2 summarise the main geometrical characteristics of the tested tube while Figure 3 shows a cross section of the tested microfin tube. Given the reported dimensions, the area enhancement with reference to the smooth tube having the same fin tip diameter is equal to 1.62.

![Figure 2. Microfin tube drawing.](image1)

![Figure 3. Cross section of the tested Microfin tube.](image2)

| Table 2. Microfin tube geometry |
|------------------------------|
| **Parameter** | **Value** |
| Outer diameter | 5 mm |
| Fin tip diameter, $D$ | 4.2 mm |
| Number of fins, $n$ | 40 |
| Fin height, $h$ | 0.15 mm |
| Apex angle, $\gamma$ | 42° |
| Helix angle, $\beta$ | 18° |
| Tube thickness, $t$ | 0.25 mm |
3. Data reduction

As described in the previous section, the subcooled liquid pumped by the volumetric gear pump is vaporized into a BPHE fed with hot water. Thus, the vapour quality at the inlet of the test section can be calculated from a thermal balance at the evaporator, as:

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

(1)

where \( \dot{m}_w \) is the water mass flow rate, \( c_{p,w} \) is the specific heat capacity of the water, \( t_{w,\text{in}} \) and \( t_{w,\text{out}} \) are the inlet and outlet water temperatures. The right-hand side term of eq. (1) reports the refrigerant side heat flow rate where \( \dot{m}_r \) is the refrigerant mass flow rate while \( J_{\text{in,TS}} \) and \( J_{\text{L,sub}} \) are the unknown specific enthalpy at the inlet of the test section and the specific enthalpy of the subcooled liquid entering the BPHE, respectively. Once calculated \( J_{\text{in,TS}} \), the vapour quality at the inlet of the test section can be estimated by:

\[ x_{\text{in,TS}} = \frac{J_{\text{in,TS}} - J_L}{J_V - J_L} \]  

(2)

where \( J_L \) and \( J_V \) 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. As already described, 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 electrical difference potential of the resistance wire inserted in the copper heater. Preliminary heat transfer measurements permitted to estimate the heat loss (\( q_{\text{loss}} \)) due to conduction through the test section as a function of the mean wall temperature. The tests were run under vacuum conditions on the refrigerant channel by supplying the power needed to maintain the mean wall temperature at a set value. The measurements were carried out by varying the mean wall temperature from 27 °C to 64 °C. The results show that the heat loss increases linearly as the mean wall temperature increases (R>0.99). In the tested range of wall temperature, the heat loss by conduction through the test section can be estimated by:

\[ |q_{\text{loss}}| = 0.1965 \cdot t_{\text{wall}} [\^\circ C] - 4.3574 [W] \]  

(3)

thus, the actual heat flow rate \( q_{\text{TS}} \) supplied to the sample is given by:

\[ q_{\text{TS}} = P_{EL} - |q_{\text{loss}}| = \Delta V \cdot I - |q_{\text{loss}}| \]  

(4)

It is worth underlying that the \( q_{\text{loss}} \) varied from 1.0% to 5.0% of the electrical power supplied. The mean vapour quality, \( x_{\text{mean}} \) is the average value between the inlet and outlet ones. The two-phase heat transfer coefficient \( HTC \), referred to the nominal area \( A \), can be defined as:

\[ HTC = \frac{q_{\text{TS}}}{A(\ell_{\text{wall}} - \ell_{\text{sat}})} = \frac{q_{\text{TS}}}{\pi D \cdot L(\ell_{\text{wall}} - \ell_{\text{sat}})} \]  

(5)

where \( \ell_{\text{wall}} \) and \( \ell_{\text{sat}} \) are defined by the followings:
The hydraulic performance of the microfin tube is given in terms of frictional pressure drop, which was calculated from the measured total pressure drop by subtracting the momentum and gravitational pressure gradients, as:

\[ \Delta p_f = \Delta p_t - \Delta p_c - \Delta p_a \]  

(7)

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

\[ \Delta p_a = G^2 (v_V - v_L) |\Delta x| \]  

(8)

where \( G \) is the refrigerant mass flux, \( v_L \) and \( v_V \) are the specific volume of liquid and vapour phase, \( |\Delta x| \) is the absolute value of the vapour quality change through the whole test section. The gravitational contribution \( \Delta p_c \) is equal to 0 Pa because the microfin tube is horizontally located. Thermodynamic and transport properties are estimated from RefProp v9.1 [15]. A detailed error analysis was performed in accordance with Kline and McClintock [16] using the values of the uncertainty of the instruments listed in Table 1; it was estimated that the uncertainty (\( k=2 \)) on the two-phase heat transfer coefficient showed a maximum value of \( \pm 8.4\% \), while the uncertainty on the vapour quality was \( \pm 0.03 \). The pressure drops showed a mean uncertainty of around 9%.

4. Experimental results

This section presents the experimental results collected during vaporization of the R245fa inside the mini microfin tube. The data were collected at constant, mean saturation temperature of 30 °C. Table 3 lists the main thermo-physical properties of the R245fa at this reference saturation temperature. This refrigerant can be classified as a low pressure fluid; its critical pressure is 36.51 bar and thus, at this operating conditions, it has a relatively low reduced pressure, around 0.049.

| Properties                              | Value |
|-----------------------------------------|-------|
| Saturation pressure, \( p_{sat} \) [bar]| 1.78  |
| Density, \( \rho \) [kg m\(^{-3}\)]     | Liquid 1324.9 |
|                                          | Vapour 10.2  |
| Thermal Conductivity, \( \lambda \) [mW m\(^{-1}\) K\(^{-1}\)] | Liquid 86.53 |
|                                          | Vapour 13.27 |
| Dynamic viscosity, \( \mu \) [\( \mu \)Pa s] | Liquid 382.3 |
|                                          | Vapour 10.4  |
| Surface tension [mN m\(^{-1}\)]         | 13.40  |
| Latent heat of vaporization [kJ kg\(^{-1}\)] | 187.33 |

Table 3. Thermo-physical properties of the R245fa at \( t_{sat}= 30\degree \) C. (Lemmon et al. [15])
Figure 4. Flow boiling heat transfer coefficient vs. vapour quality as a function of the mass velocity at constant heat flux of 30 kW m$^{-2}$. G expressed in [kg m$^{-2}$ s$^{-1}$]

Furthermore, it has a relevant difference between the liquid and vapour densities ($\rho_L/\rho_v=132$), and a relatively high surface tension (13.4 mN m$^{-1}$), if compared against others HFCs and HFOs refrigerants. It is well known that actual flow boiling heat transfer inside a tube is the result of the combination of the two main heat transfer mechanisms: nucleate boiling and forced convective boiling. Different scenarios can occur as a function of the actual operating test conditions: mass velocity, vapour quality, heat flux, and thermo-physical properties, and reduced pressure.

In particular, mass velocity, vapour density, and vapour quality mainly affect the forced convective contribution while heat flux and reduced pressure mainly affect the nucleate boiling. Considering the thermo-physical properties reported in Table 3, this fluid exhibits low vapour density and high surface tension, which might promote the forced convective boiling rather than the nucleate boiling which is also limited by the very low reduced pressure. These considerations are confirmed by the experimental results plotted in Figure 4 and 5 where the effects of the mass velocity at the two investigated heat fluxes of (HF) 30 kW m$^{-2}$ and 60 kW m$^{-2}$, respectively, are showed.

At HF=30 kW m$^{-2}$ (Figure 4), it clearly appears that at all the investigated mass velocities, the heat transfer coefficient increases almost linearly with the vapour quality, meaning that the two-phase forced convection is mainly affecting in the phase-change process. Furthermore, the slope of the measured profiles increases when passing from 100 kg m$^{-2}$ s$^{-1}$ to 150 kg m$^{-2}$ s$^{-1}$ and then to 200 kg m$^{-2}$ s$^{-1}$. However, when further increasing mass velocity to 300 kg m$^{-2}$ s$^{-1}$, the enhancement decreases and, at high vapour qualities, i.e. $x_{\text{mean}}>0.6$, the values of the heat transfer coefficient are similar, even slightly lower, to those measured at 200 kg m$^{-2}$ s$^{-1}$. This behaviour might be due to the saturation temperature drop related to the high two-phase pressure drop, which increases with both mass velocity and vapour quality. The vapour quality at the onset of the dryout phenomenon occurs earlier at low mass velocity; in fact, it was observed at around $x_{\text{mean}}=0.72$, $x_{\text{mean}}=0.78$, $x_{\text{mean}}=0.86$, and $x_{\text{mean}}=0.92$ at G=100 kg m$^{-2}$ s$^{-1}$, G=150 kg m$^{-2}$ s$^{-1}$, G=200 kg m$^{-2}$ s$^{-1}$, and G=300 kg m$^{-2}$ s$^{-1}$, respectively.

As reported in Figure 5, when increasing the heat flux to HF=60 kW m$^{-2}$ slightly different results are found. The heat transfer coefficients measured at the three investigated mass velocities seem to be almost similar and they exhibit similar behaviour up to the respective vapour qualities at the onset of dryout.
In fact, there is not any noticeable effect of the mass velocity while there still remains a weak effect of the vapour quality since the heat transfer coefficient increases as the vapour quality increases but, for a given mass velocity, the slope of the experimental profile is lower as compared to that reported in Figure 4 at 30 kW m$^{-2}$. At these operating conditions, the nucleate boiling seems to be the most effective heat transfer mechanism. At all the investigated mass velocities, the onset of the dryout phenomenon occurs earlier as compared to the data collected at 30 kW m$^{-2}$; in fact, it was observed at $x_{\text{mean}}=0.50$, $x_{\text{mean}}=0.82$, and $x_{\text{mean}}=0.85$ at G=150 kg m$^{-2}$ s$^{-1}$, G=200 kg m$^{-2}$ s$^{-1}$, and G=300 kg m$^{-2}$ s$^{-1}$, respectively.

**Figure 5.** Flow boiling heat transfer coefficient vs. vapour quality as a function of the mass velocity at constant heat flux of 60 kW m$^{-2}$. G expressed in [kg m$^{-2}$ s$^{-1}$]

**Figure 6.** Frictional pressure gradient vs. vapour quality as a function of the mass velocity at constant heat flux of 30 kW m$^{-2}$. G expressed in [kg m$^{-2}$ s$^{-1}$]
The experimental frictional pressure gradients are plotted in Figure 6 as a function of the mean vapour quality; for the sake of clarity, only data points relative to an imposed heat flux of 30 kW m\(^{-2}\) are reported. The results show that the frictional pressure gradient increases with both the mass velocity and vapour quality.

![Figure 6](image_url)

**Figure 7.** Calculated vs. experimental heat transfer coefficients. HF expressed in [kW m\(^{-2}\)].

Figure 7 reports the model proposed by Diani et al. [9], which even if it slightly underestimated the experimental data, it still showed the acceptable prediction capabilities with relative and absolute deviations equal to -23.1\% and 23.1\%, respectively. Figure 8 presents the results obtained by applying the model proposed by Diani et al. [9], which presented a relative deviation of 26.7\% and an absolute deviation of 26.7\%.

![Figure 8](image_url)

**Figure 8.** Calculated vs. experimental frictional pressure gradients. HF expressed in [kW m\(^{-2}\)].
5. Conclusions

The paper presents experimental heat transfer coefficients and pressure drops measured during R245fa flow boiling inside a mini microfin tube with an inner diameter at the fin tip of 4.2 mm. Tests were run at a constant mean saturation temperature of 30 °C, by varying the vapour quality from 0.15 to 0.95, the mass velocity from 100 kg m\(^{-2}\) s\(^{-1}\) to 300 kg m\(^{-2}\) s\(^{-1}\), at two imposed heat fluxes, 30 kW m\(^{-2}\) and 60 kW m\(^{-2}\). The results confirm that the heat transfer process is controlled by the two main 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. The two-phase frictional pressure drops were also measured. They increase with both mass velocity and vapour quality. Finally, the results were compared against the models for flow boiling heat transfer coefficient and frictional pressure drop estimations proposed by Diani et al. [9], which demonstrated to be satisfactory consistent with the present data.

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7. References

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