Vaporization inside a mini microfin tube: experimental results and modeling

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Abstract. This paper proposes a comparison among the common R134a and the extremely low GWP refrigerant R1234yf during vaporization inside a mini microfin tube. This microfin tube has an internal diameter of 2.4 mm, it has 40 fins, with a fin height of 0.12 mm. Due to the high heat transfer coefficients shown by this tube, this technology can lead to a refrigerant charge reduction. Tests were run in the Heat Transfer in Micro Geometries Lab of the Dipartimento di Ingegneria Industriale of the Università di Padova. Mass velocities range between 375 and 940 kg m\(^{-2}\) s\(^{-1}\), heat fluxes from 10 to 50 kW m\(^{-2}\), vapour qualities from 0.10 to 0.99, at a saturation temperature of 30 °C. The comparison among the two fluids is proposed at the same operating conditions, in order to highlight the heat transfer and pressure drop differences among the two refrigerants. In addition, two correlations are proposed to estimate the heat transfer coefficient and frictional pressure drop during refrigerant flow boiling inside mini microfin tubes. These correlations well predict the experimental values, and thus they can be used as a useful tool to design evaporators based on these mini microfin tubes.

1. Introduction
HFCs are greenhouse gases widely used for refrigeration and air conditioning applications. Even though they have a zero ozone depletion potential, they have a non-negligible global warming potential. For instance, R134a, which is one of the most used HFCs, has a GWP of approximately 1400. HFCs must be gradually phased out in order to reduce the emissions of global warming gases. R1234ze(E) and R1234yf have been proposed as replacements of the HFC 134a. R1234ze(E) and R1234yf are hydro fluoro olefins, they have a zero ozone depletion potential and a GWP lower than 1 [1]. Together with the use of low GWP refrigerants, the reduction of greenhouse gases could be achieved with the use of efficient systems, which lead to a refrigerant charge reduction.

Small microfin tubes were initially studied for CO\(_2\) applications, due to its high working pressure. Gao et al. [2] performed experiments on the flow boiling heat transfer of almost pure CO\(_2\) and CO\(_2\)-oil mixtures in horizontal smooth and microfin tubes. The smooth tube is a stainless steel tube with an inside diameter of 3 mm, whereas the microfin tube is a copper tube with a mean inside diameter of 3.04 mm. Experiments were carried out at mass velocities from 190 to 1300 kg m\(^{-2}\) s\(^{-1}\), heat fluxes from 5 to 30 kW m\(^{-2}\), at a saturation temperature of 10 °C, with oil circulation ratio from <0.01% to 0.72% wt. In case of almost pure CO\(_2\), the flow boiling mechanism was found to be dominated by nucleate boiling. Dang et al. [3] investigated the flow boiling heat transfer of CO\(_2\) inside a small microfin tube with a mean inner diameter of 2.0 mm. Mass flux ranges between 360 and 720
kg m$^{-2}$ s$^{-1}$, heat flux between 4.5 and 18 kW m$^{-2}$, at a saturation temperature of 15 °C. Experimental results indicated that the heat flux has a significant effect on the heat transfer coefficient, whereas the coefficient does not always increases with mass velocity: under certain conditions, the heat transfer coefficient at higher mass velocity was lower than that at lower mass velocities, indicating that the convective heat transfer has a suppression effect on nucleate boiling.

R1234yf has a normal boiling temperature approximately 3.15 K lower than that of R134a, thus it is a possible substitute of the common R134a, whose Global Warming Potential is about 1400. McLinden et al. [4] 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.

Up today, 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. [5] presented experimental results for local heat transfer coefficients, adiabatic frictional pressure gradients, and two-phase flow regimes with the low GWP R1234ze(E). The circular smooth tube has an inner diameter of 6 mm. Mass velocities were varied from 146 to 520 kg m$^{-2}$ s$^{-1}$, heat fluxes from 5.0 to 20.4 kW m$^{-2}$, at a saturation temperature from -2.9 °C to 12.1 °C. Compared to R134a, R1234ze(E) revealed similar values of local heat transfer coefficients, showing the same trend with vapour quality, but it showed an earlier inception of the dry out phenomenon. The experimental adiabatic frictional pressure drop data of R1234ze(E) result slightly higher than those obtained for R134a at the same operating conditions.

Hossain et al. [6] experimentally measured the heat transfer coefficients during in-tube evaporation of R1234ze(E), R32, R410A, and a mixture of R1234ze(E)/R32 (55/45 mass %). The smooth tube has an inner diameter of 4.35 mm. The experiments have been carried out under the conditions of mass flux from 150 to 445 kg m$^{-2}$ s$^{-1}$, the saturation temperatures are 5 and 10 °C, over the vapour quality range 0.0-1.0. It was found that the experimental heat transfer coefficients of R1234ze(E) were lower than those of R1234ze(E)/R32, R410A and R32 by 11%, 56%, and 83%, at about 0.48 vapour quality and 300 kg m$^{-2}$ s$^{-1}$.

Really few works can be found in the open literature regarding low GWP refrigerants flow boiling inside small microfin tubes. Kondou et al. [7] experimentally studied the flow boiling of R1234yf(E), R32, and mixture of R1234yf(E)/R32 in a horizontal microfin tube of 5.21 mm inner diameter. The heat transfer coefficient and pressure drop were measured at a saturation temperature of 10 °C, heat fluxes of 10 and 15 kW m$^{-2}$, and mass velocities from 150 to 400 kg m$^{-2}$ s$^{-1}$. The heat transfer coefficients of R1234yf(E) were lower than those of R32, and the heat transfer coefficient of the mixture R1234yf(E)/R32 was even lower than that of R1234yf(E).

One of the few works about R1234yf flow boiling inside microfin tubes is that of Han et al. [8], who experimentally investigated the boiling heat transfer characteristics of R1234yf inside a microfin tube with an outer diameter of 7 mm, at mass fluxes between 100 and 400 kg m$^{-2}$ s$^{-1}$, heat fluxes of 4, 8 and 12 kW m$^{-2}$, saturation temperature of 5 °C and 15 °C, with four different oil concentrations: 0%, 1.5%, 3.0%, and 5.0%.

This paper explores the possible viable replacement of R134a with the low GWP R1234yf. Experimental results are given in terms of comparison between R134a and R1234yf heat transfer coefficient and frictional pressure drop. Furthermore, two empirical correlations are proposed and discussed for the estimation of the heat transfer coefficient and frictional pressure drop. The heat transfer model is an updated version of the model by Padovan et al. [9], and it is valid for vapour quality prior to the thermal crisis and for mass velocities from 200 to 940 kg m$^{-2}$ s$^{-1}$. The model for the estimation of the frictional pressure gradient uses a liquid only two-phase multiplier, which is a modified version of the multiplier suggested by Cavallini et al. [10] for minichannels. These correlations well predict the experimental values, and thus they can be used as a useful tool to design evaporators based on these mini microfin tubes.
2. Experimental set up and test section

The experimental tests were carried out in a facility built at the Heat Transfer in Micro Geometries laboratory of the Dipartimento di Ingegneria Industriale of the University of Padova. This experimental facility permits pressure drop and either flow boiling or condensation measurements to be performed. It has a maximum working pressure of 30 bar, and the mass flow rate can be varied up to 72 kg h\(^{-1}\). A schematic of the experimental set up is depicted in figure 1. The facility consists of three main loops: the refrigerant loop, the hot water loop, and the cold water loop. In the refrigerant loop, the fluid is pumped by means of a magnetic coupled gear pump. An oil free pump was chosen in order to avoid any possible oil contamination in the refrigerant loop. It is connected to an inverter, so that it is possible to vary the refrigerant mass flow rate by controlling an electronic display. The refrigerant is pumped through a Coriolis effect flow meter, which reads the refrigerant mass flow rate with an accuracy of ±0.10% of the reading, and then the fluid reaches an evaporator, at the exit of which the refrigerant is superheated. The evaporator is a brazed plate heat exchanger, where the refrigerant flows in counter current with hot water, which is supplied by the hot water loop. The hot water loop consists of an electric boiler, able to supply up to 5 kW: it is split into three electric resistances, two of which work in on-off mode, whereas the power of the third resistance can be modulated. A controlling valve to adjust the water flow rate, an expansion vessel to cope with water volume variations, and a magnetic flow meter with an accuracy of ±0.25% of the reading, complete the hot water loop. The water temperature difference across the heat exchanger is measured by means of a 4-junction T-type thermopile (accuracy ±0.03 K). The superheated vapour is then partially condensed in a pre-condenser, which is a tube-in-tube heat exchanger, where the refrigerant flows in the inner tube, and cold water flows in the annulus. The cold water is supplied by the cold water loop, where a stabilized chiller controls the water temperature at the inlet of the pre-condenser. The cold water flow rate can be controlled by means of a valve and it is measured with a magnetic flowmeter (accuracy ±0.25% of the reading). The water temperature across the pre-condenser is measured by means of 4-junction T-type thermopile (accuracy ±0.03 K). It is possible to adjust the heat flow rate by independently controlling the water flow rate and the water temperature, in order to set the vapour quality at the inlet of the test section at the desired value.

![Figure 1. Schematic of the experimental facility.](image-url)
The refrigerant evaporates in the test section, and is then condensed and subcooled in a post condenser, which is another brazed plate heat exchanger fed with tap water. The saturation conditions can be controlled with a damper, which is connected to the compressed air line. As highlighted in Figure 1, temperature and pressure measurements are carried out in order to know the refrigerant thermodynamic state throughout the circuit. Calibrated T-type thermocouples, with an accuracy of ±0.05 K, are used to measure the refrigerant temperatures at the inlet of the evaporator, at the inlet of the pre-condenser, at the inlet of the test section, and at the outlet of the post-condenser, whereas absolute pressure transducers, with an accuracy of ±1950 Pa, are used to monitor the refrigerant pressures at the inlet of the evaporator, at the inlet of the pre-condenser, and at the inlet of the test section. A differential pressure transducer (accuracy of ±25 Pa) is used to measure the total pressure drop across the test section.

A picture of the assembled test section is reported in figure 2. The microfin tube was brazed in a guide milled on the top surface of a copper plate, which is 225 mm long, 20 mm high, and 10 mm wide. Fifteen holes were drilled just 1 mm beneath the tube, in order to host as many T-type thermocouples (accuracy ±0.05 K) to monitor the wall temperature distribution. Another guide was milled on the bottom surface, where a Nickel-Chrome wire resistance was located. The wire resistance is connected to a DC power supplier which provides the heat flow rate needed to vaporize the refrigerant flowing inside the tube. The electric power can be calculated from two different voltage drop measurements: the first one is across the Ni-Cr wire resistance, and the second one is across a known reference resistance (shunt), which is in series with the Ni-Cr wire resistance and permits to calculate the current flowing in the circuit. The accuracy of the measurements of the electric power is ±0.13% of the reading. Pressure ports are located about 57.5 mm upstream and downstream of the copper plate, and thus the length for the pressure drop measurements is 340 mm. A smooth connection was designated to connect the microfin tube to the tubes of the facility in order to avoid any possible abrupt pressure losses. The microfin tube under investigation has an outer diameter OD of 3 mm, an inner diameter at the fin tip \( D \) of 2.4 mm, it has 40 fins along the circumference, and each fin is 0.12 mm high. The apex angle \( \gamma \) is 43°, whereas the helix angle \( \beta \) is 7°. The test section is located inside an aluminum housing, which contains a 30 mm thick rock wool to limit as much as possible the heat losses due to conduction to the ambient. Preliminary tests were run in order to evaluate these heat losses: tests were carried out in vacuum conditions by supplying the electric power needed to maintain the wall temperature at fixed values. Heat losses \( q_{\text{loss}} \) were found to have a linear dependence on the mean wall temperature, according to the following equation:

\[
q_{\text{loss}}[\text{W}] = 0.1235 \cdot \bar{T}_{\text{wall}}[\text{°C}] - 2.6114
\]

where \( \bar{T}_{\text{wall}} \) is the mean wall temperature. During these tests, the wall temperature was maintained between 25 °C and 60 °C, and the ambient temperature was approximately 20 °C.
3. Data analysis

The vapour quality at the inlet of the test section depends on the heat flow rate exchanged in the pre-condenser and on the thermodynamic conditions at the outlet of the evaporator, as:

\[ q_{pc} = m_{w,pc} \cdot c_{p,w} \cdot (t_{w,\text{out}} - t_{w,\text{in}}) = m_{\text{ref}} \cdot (h_v - h_{TS,\text{in}}) \]  

(2)

where \( m_{w,pc} \) is the water mass flow rate flowing in the pre-condenser, \( c_{p,w} \) is the water specific heat at constant pressure, and \( t_{w,\text{out}} \) and \( t_{w,\text{in}} \) are the water temperatures at the outlet and inlet of the pre-condenser, respectively. On the right side of equation 2, \( m_{\text{ref}} \) is the refrigerant mass flow rate measured by the Coriolis effect flowmeter, and \( h_v \) and \( h_{TS,\text{in}} \) are the refrigerant specific enthalpies at the outlet of the evaporator (which corresponds to the enthalpy at the inlet of the pre-condenser), and at the inlet of the test section (which corresponds to the enthalpy at the outlet of the pre-condenser). From this thermal balance, it is possible to calculate the refrigerant enthalpy at the inlet of the test section as:

\[ h_{TS,\text{in}} = h_v - \frac{m_{w,pc} \cdot c_{p,w} \cdot (t_{w,\text{out}} - t_{w,\text{in}})}{m_{\text{ref}}} \]  

(3)

and thus the inlet vapour quality is given by:

\[ x_{\text{in}} = \frac{h_{TS,\text{in}} - h_L}{h_v - h_L} \]  

(4)

with \( h_v \) and \( h_L \) the specific enthalpies of the saturated liquid and vapour, respectively, calculated from the measured values of refrigerant inlet pressure. By controlling the water flow rate at the pre-condenser and the water temperature at the inlet of the pre-condenser, it is possible to set the desired vapour quality \( x_{\text{in}} \) at the inlet of the test section. All the thermo-physical properties are calculated using Refprop 9.1 [11].

The net heat flow rate exchanged by the microfin tube is calculated as:

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

(5)

where \( \Delta V \) and \( I \) are the voltage and current in the electric wire resistance. Therefore, the refrigerant specific enthalpy at the outlet of the test section can be calculated with the following equation:

\[ h_{TS,\text{out}} = h_{TS,\text{in}} + \frac{q_{TS}}{m_{\text{ref}}} \]  

(6)

and the vapour quality as:

\[ x_{\text{out}} = \frac{h_{TS,\text{out}} - h_L}{h_v - h_L} \]  

(7)

where, in this case, the refrigerant properties are calculated from the knowledge of the outlet pressure. The mean vapour quality can now be calculated. From the error propagation analysis, the mean, maximum, and minimum uncertainties on the mean vapour quality are 0.027, 0.029, and 0.025, respectively.

The thermal performance of the microfin tube is given in terms of heat transfer coefficient \( HTC \), referred to the area of an equivalent smooth tube with the same inner diameter at the fin tip \( A_D \), as:

\[ HTC = \frac{q_{TS}}{A_D \cdot (\bar{t}_{\text{wall}} - \bar{t}_{\text{sat}})} \]  

(8)

where \( \bar{t}_{\text{wall}} \) is the average value of the fifteen temperatures measured by the fifteen thermocouples located 1 mm under the microfin tube, and \( \bar{t}_{\text{sat}} \) is the mean saturation temperature. The mean,
maximum, and minimum uncertainties on the heat transfer coefficient are 1.8%, 5.4%, and 0.7%, respectively. The hydraulic performance of the microfin tube is given in terms of frictional pressure gradient, which was calculated from the measured total pressure gradient by subtracting the momentum pressure gradient, as:

$$\left(\frac{dp}{dz}\right)_f = \left(\frac{dp}{dz}\right)_{tot} - \left(\frac{dp}{dz}\right)_a \quad (9)$$

The model of Rouhani and Axelsson [12] was taken into account to calculate the void fraction, needed to calculate the momentum pressure drop. The gravitational contribution was not considered since the microfin tube is horizontally located.

4. Experimental results

This paragraph proposes a comparison among the thermal and hydraulic performances of R1234yf and R134a during vaporization inside a mini microfin tube. The saturation temperature at the inlet of the test section was kept constant and equal to 30 °C, which is suitable for high temperature industrial heat pumps and for electronics cooling applications. The mass velocity is referred to the cross sectional area of an equivalent smooth tube having an internal diameter equal to the diameter at the fin tip of the microfin tube under investigation, whereas the heat flux is referred to the outer surface area of the mini microfin tube. The vapour quality was varied from 0.1 to 1, the mass velocity was varied from 375 to 940 kg m$^{-2}$ s$^{-1}$, and the heat flux from 10 to 50 kW m$^{-2}$: in these operating conditions, the quality variation from the inlet to the outlet of the test section is between 0.04 and 0.39.

Table 1 reports the major thermo physical properties of R1234yf and R134a. As it can be seen, the two fluids exhibit approximately the same properties, which make R1234yf a possible replacement of the more common R134a. R134a shows a slightly lower vapour density, thus at constant mass velocity it might exploit a better performance in case of two-phase forced convection, whereas the liquid densities are similar. Furthermore, reduced pressure of R1234yf is 22% higher than that of R134a, but the latent heat of vaporization is 18% lower: the two opposite trends might balance the nucleate boiling performances of the two fluids.

Table 1. Major thermo-physical properties of R1234yf and R134a. Data from Refprop 9.1 [12].

| Property | R1234yf [bar] | R134a [bar] |
|----------|--------------|-------------|
| p_{sat}  | 7.84         | 7.70        |
| p_{red}  | 0.232        | 0.190       |
| \rho_v   | 1073         | 1188        |
| \rho_L   | 43.7         | 37.5        |
| c_{p,L}  | 1417         | 1447        |
| c_{p,V}  | 1086         | 1066        |
| h_{L,V}  | 141          | 173         |
| \lambda_L | 0.062        | 0.079       |
| \lambda_V | 0.014        | 0.014       |
| \mu_L   | 145          | 183         |
| \mu_V   | 11.3         | 11.9        |
| Pr_L    | 3.32         | 3.35        |
| Pr_v    | 0.86         | 0.88        |
| \sigma  | 5.56         | 7.38        |
| (dT/dp)_{sat} | 4.7         | 4.5        |
Figure 3. Effect of mass velocity on the heat transfer coefficient. Data at 10 kW m$^{-2}$.

Figure 3 compares the heat transfer coefficient of R1234yf and R134a. Heat transfer coefficients are plotted against the mean vapour quality as a function of mass velocity with an imposed heat flux of 10 kW m$^{-2}$. Generally speaking, the heat transfer coefficient always increases as vapour quality increases, meaning that two-phase forced convection mainly controls the phase change process in these operating test conditions. It is interesting to point out that the highest heat transfer coefficient is shown at the lowest tested mass velocity, i.e. 375 kg m$^{-2}$ s$^{-1}$, for both fluids: this can be linked to the particular geometry of the microfin tube, which enhances the heat transfer mechanism especially at low mass velocities. The higher thermal performances of R134a become more and more evident when the vapour quality increases. This is due to the fact that the contribution of two-phase forced convection increases with increasing vapour quality, i.e. on vapour velocity. Since R134a has a lower vapour density (see table 1), it exploits higher vapour velocity at constant mass velocity, and thus it shows higher heat transfer coefficients than those of R1234yf in these operating test conditions.

Figure 4 collects the same results but with an imposed heat flux of 50 kW m$^{-2}$. First of all, at low vapour qualities, the heat transfer coefficients are higher than the former case for both fluids, since in this case the heat flux is higher and the contribution of nucleate boiling is stronger, which enhances the heat transfer coefficients. At low vapour qualities, the two fluids show similar values of heat transfer coefficient, due to their thermo-physical characteristics which lead to comparable performances during the nucleate boiling mechanism. When the vapour quality increases, R134a shows slightly higher values of the heat transfer coefficient of R134a, since it has better convective characteristics. In addition, the heat transfer coefficients at high vapour quality are 1.4 times than those at low vapour quality in the case of R134a, whereas they are 1.25 times in the case of R1234yf. A start of the dryout phenomenon can be observed for both refrigerants at mean vapour qualities higher than 0.8.

Figure 5 shows the frictional pressure gradient for R1234yf and R134a as a function of the mean vapour quality with an imposed heat flux of 10 kW m$^{-2}$. The model of Rouhani and Axelsson [12] was considered to estimate the void fraction to calculate the momentum pressure drops, which were subtracted from the total measured pressure drops. Generally speaking, the results show that, at constant mass velocity, the frictional pressure gradient increases with vapour quality, reaching a maximum value after which it slightly decreases. At constant vapour quality, the frictional pressure gradient increases as the mass velocity increases. R1234yf and R134a show similar values of frictional pressure gradient at low vapour qualities, whereas R1234yf shows slightly lower frictional pressure gradients than those of R134a at higher vapour quality, and the difference increases with increasing the vapour quality: this behaviour can be explained considering the fact that R134a and R1234yf show similar value of liquid density, whereas the vapour density of R134a is lower, which leads to higher vapour velocity at constant mass velocity with consequent higher pressure drops.
5. Empirical modelling

Diani et al. [13] recently proposed two empirical equations to estimate the heat transfer coefficient and frictional pressure drop during refrigerant flow boiling inside a mini microfin tube. The models were developed from experimental measurements carried out during R134a and R1234ze(E) flow boiling inside a mini microfin tube with an inner diameter at the fin tip of 3.4 mm, with a fin height of 0.12 mm. The heat transfer model is an updated version of the original one by Cavallini et al. [14], also revised in Padovan et al. [9]. The model is valid for vapour qualities prior to the onset of the dryout and for mass velocities ranging between 200 and 940 kg m$^{-2}$ s$^{-1}$.

The comparison of the calculated heat transfer coefficient against the experimental counterpart is shown in figures 6 and 7. The model slightly underestimates the heat transfer coefficient, with relative, absolute, and standard deviations of -13.7%, 14.1%, and 8.8%, respectively, for R1234yf, and -13.7%, 14.1%, and 10.5%, respectively, for R134a. More in details, the model calculates the heat transfer coefficient as the sum of the contribution of nucleate boiling and of two-phase forced convection. The nucleate boiling heat transfer coefficient can be calculated using the model proposed by Cooper [15] considering a suppression factor, whereas the convective heat transfer coefficient is a revised version of the two-phase heat transfer coefficient.
proposed by Cavallini et al. [14]. The effect of heat flux on the heat transfer coefficient calculated with the model is proposed in figure 8 for a case study at 565 kg m$^{-2}$ s$^{-1}$. As expected, the nucleate boiling contribution decreases with increasing the vapour quality, whereas the two-phase forced convection contribution increases with increasing the vapour quality. It is interesting to point out that there is a strong effect of heat flux on the heat transfer coefficient especially at low vapour qualities, being 26% even at the lowest heat flux of 10 kW m$^{-2}$. Figure 9 shows the effect of mass velocity on the calculated R1234yf heat transfer coefficient with a heat flux of 50 kW m$^{-2}$. As it appears, both the nucleate boiling and two-phase forced convection contributions are highly affected by vapour qualities, but are not affected by mass velocity. This behaviour is confirmed by the experimental results collected in figure 4.

Figures 10 and 11 compare the calculated frictional pressure drops against the experimental ones. The model suggests calculating the only two-phase multiplier as proposed by Cavallini et al. [10] for minichannels. The model underestimates the experimental values with a relative and standard deviations of -26.4% and 6.2% for R1234yf, and of -23.8% and 6.2% for R134a. The underestimation can be related to the ratio between fin height to tube diameter, which was 0.035 for the tested tube from which the correlation was developed, whereas it is 0.050 for the case under investigation.

Figure 8. Effect of heat flux with the model by Diani et al. [13] for R1234yf.

Figure 9. Effect of mass velocity with the model by Diani et al. [13] for R1234yf.

Figure 10. Comparison between experimental and calculated pressure gradients with the model of Diani et al. [13] for R1234yf.

Figure 11. Comparison between experimental and calculated pressure gradients with the model of Diani et al. [13] for R134a.
6. Conclusions
This paper compares the behaviour of R1234yf and R134a during flow boiling inside a mini microfin tube with an inner diameter at the fin tip of 2.4 mm. The microfin tube was electrically heated from the bottom, and the heat flux ranged between 10 and 50 kW m\(^{-2}\). Mass velocity was varied between 375 and 940 kg m\(^{-2}\) s\(^{-1}\), and the saturation temperature at the inlet of the test section was kept constant and equal to 30 °C. At low heat flux, R134a shows higher heat transfer coefficients than those of R1234yf, due to its better convective thermo-physical properties, whereas they have comparable performance at high heat fluxes. On the other hand, R1234yf shows lower frictional pressure gradient. Therefore, R1234yf is a possible candidate as replacement of the common R134a. The experimental results of heat transfer coefficient and frictional pressure drop were compared against the values predicted by the correlation of Diani et al. The model underestimates on average the heat transfer coefficient of -13.7%, and the frictional pressure drop of -25%. The model also permits to discern the effects of nucleate boiling and two-phase forced convection during the calculation of the heat transfer coefficient.

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