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ABSTRACT

This paper presents some new experimental data on R1234yf saturated flow boiling inside a 4 mm horizontal smooth tube: the effects of heat flux, refrigerant mass flux, and mean vapor quality are investigated separately to point out the different heat transfer mechanism contributions (i.e., nucleate boiling or/and forced convection boiling). The experimental tests were carried out at a saturation temperature equal to 10 °C, refrigerant mass flux from 200 to 600 kg m^{-2} s^{-1}, heat flux from 15 to 30 kW m^{-2}, and at increasing vapor quality up to incipient dryout. The measurements are here reported in terms of boiling heat transfer coefficient and frictional pressure drop. Furthermore, the R1234yf performance is compared against R1234ze(E) and R134a, since the substitution of R134a with low GWP refrigerants is one of the most important actual challenge for refrigeration and air conditioning, and R1234ze(E) and R1234yf seem to be very promising substitutes of it. Finally, the experimental heat transfer and frictional pressure drop data are used to assess some classical literature correlations.

1. INTRODUCTION

R1234yf, together with other low GWP molecules was pointed out to be an environmentally friendly substitute to R134a. During years, R1234yf has been proposed to be applied as R134a alternative in many applications, among them: automotive (Zilio et al., 2011), domestic refrigerators (Righetti et al., 2015), water heat pumps (Nawaz et al. 2017), and ORC systems (Yamada et al., 2012).

Despite the actual diffusion of this molecule as refrigerant, it can be stated that the literature lacks of experimental heat transfer data. This consideration can be drawn by analyzing the number of data published on R1234yf and R1234ze(E), another viable candidate to substitute R134a in similar technological applications. For instance, at the best authors' knowledge, less than 2500 data are now available on R1234yf flow boiling inside tubes, while more than 3700 are published on R1234ze(E).

The literature presents some works on R1234yf flow boiling inside small tubes and many of them compared R1234yf performance against R134a under the same working conditions. Among these latter, Saitoh et al. (2011), Anwar et al. (2015) and Sempértegui-Tapia and Ribatski (2017). All these papers are concordant in evaluating R1234yf heat transfer coefficients similar to those obtained with R134a and pressure drops slightly lower than R134a ones.

This paper presents some new experimental data collected in a horizontal copper smooth tube during R1234yf flow boiling. The data are going to be compared against R134a and R1234ze(E) since it is well-known that R134a is going to be phased out and that R1234yf and R1234ze(E) are two viable environmentally friendly alternatives to it.

2. EXPERIMENTAL MEASUREMENTS AND DATA REDUCTION

The experimental facility, shown in Fig. 1, consists of three different loops: one for refrigerant and two for the secondary fluids (water and water-glycol solution). In the first loop the refrigerant is pumped from the sub-cooler into the pre-evaporator, a brazed plate heat exchanger, where it is partially evaporated to achieve the set quality at the inlet of the tubular test section. The refrigerant goes through the test section where it is evaporated and then it comes back to a condenser and a sub-cooler, both brazed plate heat exchangers. A variable speed volumetric pump varies the refrigerant flow rate, whereas a bladder accumulator connected to a nitrogen bottle and a pressure regulator controls the operating pressure in the refrigerant loop. The second loop is able to supply a water-glycol flow at a constant temperature in the range of -10 to 30 °C with a stability within ±0.1 K used to feed the sub-cooler and the condenser, whereas the third loop supplies a refrigerated water flow at a constant temperature in the range of

3 °C to 30 °C with a stability within ±0.1 K used to feed the test section and the pre-evaporator. The test-section (Fig.2) is a double tube evaporator in which the refrigerant evaporates in the inner tube while the refrigerated water flows in the annulus. The test-section is subdivided into two different parts: a pre-section, 200 mm long, in which the refrigerant flow achieves a fully developed flow regime and the measurement section, 800 mm long, in which the heat transfer coefficient is measured. This arrangement is obtained using a single inner smooth tube, 4 mm in diameter, 1300 mm long and two separated cooling water jackets fed in series. The inner tube is instrumented with four copper-constantan thermocouples embedded in its wall to measure surface temperature. The thermocouples are inserted into two equidistant axial grooves, at the top and the bottom of the cross section, 100 mm from the inlet and outlet of the cooling water. Each groove is sealed with a copper wire fixed by epoxy. The main features of the different measuring devices are reported in Table 1.

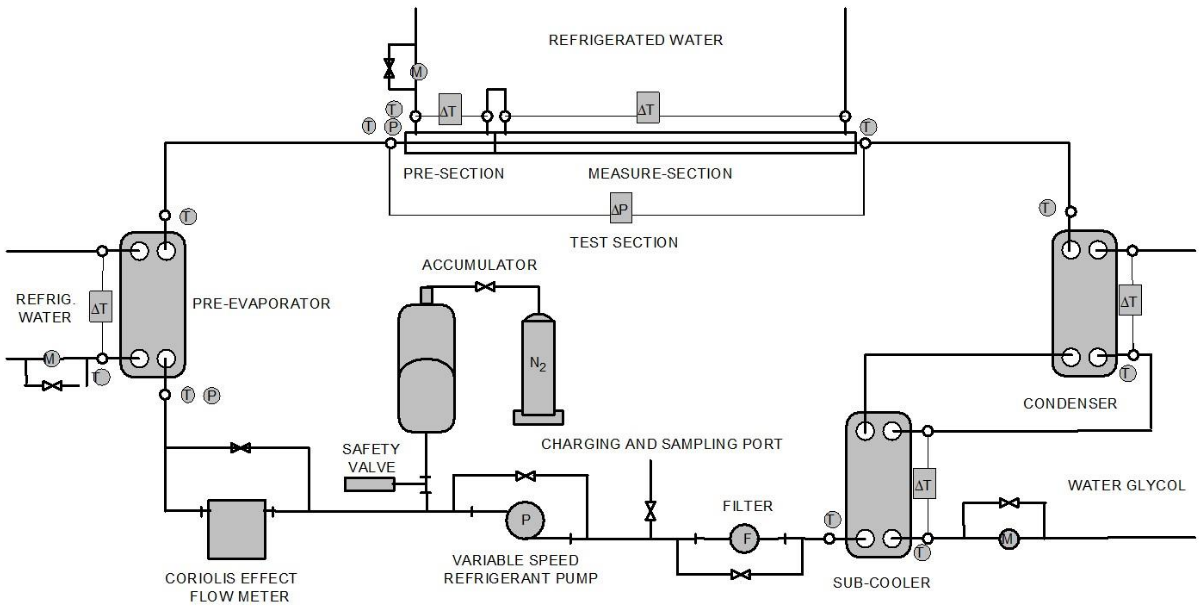


Figure 1. Schematic view of the experimental test rig.

The experimental results are reported in terms of boiling heat transfer coefficients *HTC* and frictional pressure drop Δp_f. The boiling heat transfer coefficient *HTC* is equal to the ratio between the heat flow rate *Q*, the heat transfer area *A* and the mean temperature difference ΔT:

$$HTC = Q / (A\ \Delta T) \qquad \text{Eq. (1)}$$

The heat flow rate Q is derived from a thermal balance on the water-side of the measurement section, the heat transfer area A of the measurement section is equal to the area of the inner surface of the test tube, and the mean temperature difference ΔT is equal to the difference between the average saturation temperature, derived from the average pressure on refrigerant side, and the arithmetical mean value of the reading of the four thermocouples embedded in the tube wall.

The frictional pressure drop on the refrigerant-side Δp_f is computed by subtracting the inlet / outlet local pressure drops Δp_c, and the momentum pressure drops Δp_a from the total pressure drop measured Δp_t:

$$\Delta p_f = \Delta p_t - \Delta p_c - \Delta p_a \qquad \text{Eq. (2)}$$

Being the test section horizontal, no gravity pressure drops Δp_g occur.

For more details on data reduction, please refer to Longo et al. (2017).

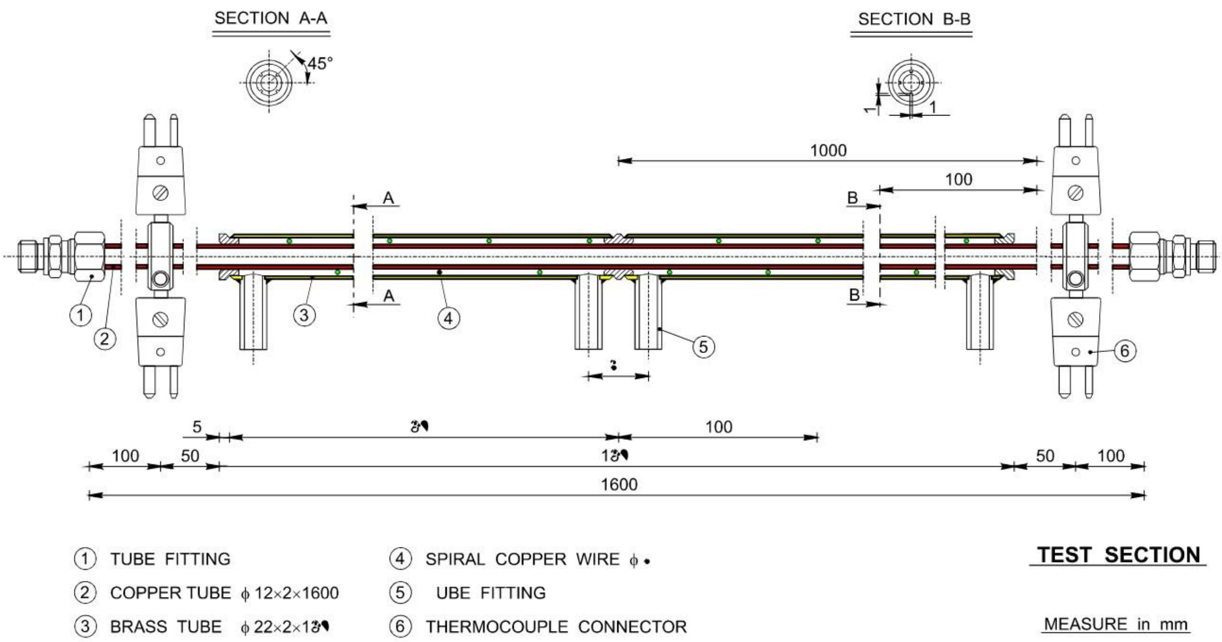


Figure 2. Schematic view of the test section.

Table 1. Specification of the different measuring devices

Devices	Uncertainty (*k*=2)	Range
T-type thermocouples	0.1 K	-20/80 °C
T-type thermopiles	0.05 K	-20/80 °C
Abs. pressure transducers	0.075% f.s.	0/3.0 MPa
Diff. pressure transducers	0.075% f.s.	0/0.3 MPa
Coriolis effect flow meters	0.1%	0/300 kg h^{-1}
Magnetic flow meters	0.15% f.s.	100/1200 l h^{-1}
Data logger	± 2.7 μV	0 / 100 mV

3. ANALYSYS OF THE EXPERIMENTAL RESULTS

A new set of 34 experimental tests collected during R1234yf flow boiling at a saturation temperature of 10 °C inside a 4 mm horizontal smooth copper tube was carried out.

The dominant heat transfer regimes in flow boiling inside smooth tubes are nucleate boiling and forced convection boiling. In nucleate boiling, the heat transfer coefficients show a great sensitivity to heat flux, whereas in convection boiling they depend mainly on mass flux and vapor quality. Therefore, the experimental tests were carried out in order to separate the contribution of heat flux, refrigerant mass flux and mean vapor quality. First, at constant heat flux (q = 20 kW m^{-2}), four different refrigerant mass fluxes (G = 200, 300, 400, and 600 kg m^{-2} s^{-1}) were applied at increasing mean vapor quality up to incipient dryout. Then, at constant refrigerant mass flux (G = 400 kg m^{-2} s^{-1}), four different heat fluxes (q = 15, 20, 25, and 30 kW m^{-2}) were applied at increasing mean vapor quality up to incipient dryout.

A detailed error analysis performed in accordance with Kline and McClintock (1953) indicates an overall uncertainty within ±7% for the refrigerant heat transfer coefficient measurements and within ±9%, for the total pressure drop measurements.

Fig. 3 shows the boiling heat transfer coefficient *HTC* against mean vapor quality at constant heat flux (q = 20 kW m^{-2}) and four different refrigerant mass fluxes (G = 200, 300, 400, and 600 kg m^{-2} s^{-1}). At low refrigerant mass fluxes, the heat transfer coefficients show a reduced sensitivity to vapor quality, that points out the predominant effect of nucleate boiling with respect to convection boiling. While, by increasing the mass flux, the heat transfer coefficients have a positive slope versus vapor quality and the slope increases with refrigerant mass flux, indicating an increasing effect of forced convection boiling mechanism.

Fig. 4 presents the boiling heat transfer coefficient *HTC* against mean vapor quality at constant refrigerant mass flux (G=400 kg m^{-2} s^{-1}) and four different refrigerant heat fluxes (q=15, 20, 25, and 30 kW m^{-2}). Again, heat transfer coefficients have a positive slope versus vapor quality but here the slope decreases with heat flux, confirming the presence of forced convection boiling mechanism.

Fig. 5 plots the boiling frictional pressure drop, evaluated accordingly Eq. (2), against refrigerant mass flux at a fixed heat flux equal to 20 kW m^{-2}. As expected, frictional pressure drop increases with mass flux and vapor quality.

The experimental data collected were then compared against different heat transfer and frictional pressure drop correlations for flow boiling inside smooth tubes. The equation by Kim and Mudawar (2014) for pre-dryout saturated flow boiling heat transfer in mini/micro-channels shows the best performance with a mean relative deviation of 3.2% and a mean absolute deviation of 6.4%. Furthermore, the correlation by Friedel (1979) predicts the experimental frictional pressure drop data with a mean relative deviation of -5.6% and a mean absolute deviation of 17.4%.

Fig. 6 shows the deviation between the experimental heat transfer coefficient data and the calculated ones by the Kim and Mudawar (2014) correlation. It predicts very well the experimental data in trend and magnitude. Furthermore, Fig. 7 shows the deviation between the experimental and the calculated frictional pressure drop data by the Friedel (1979) correlation, highlighting a good predicting capability.

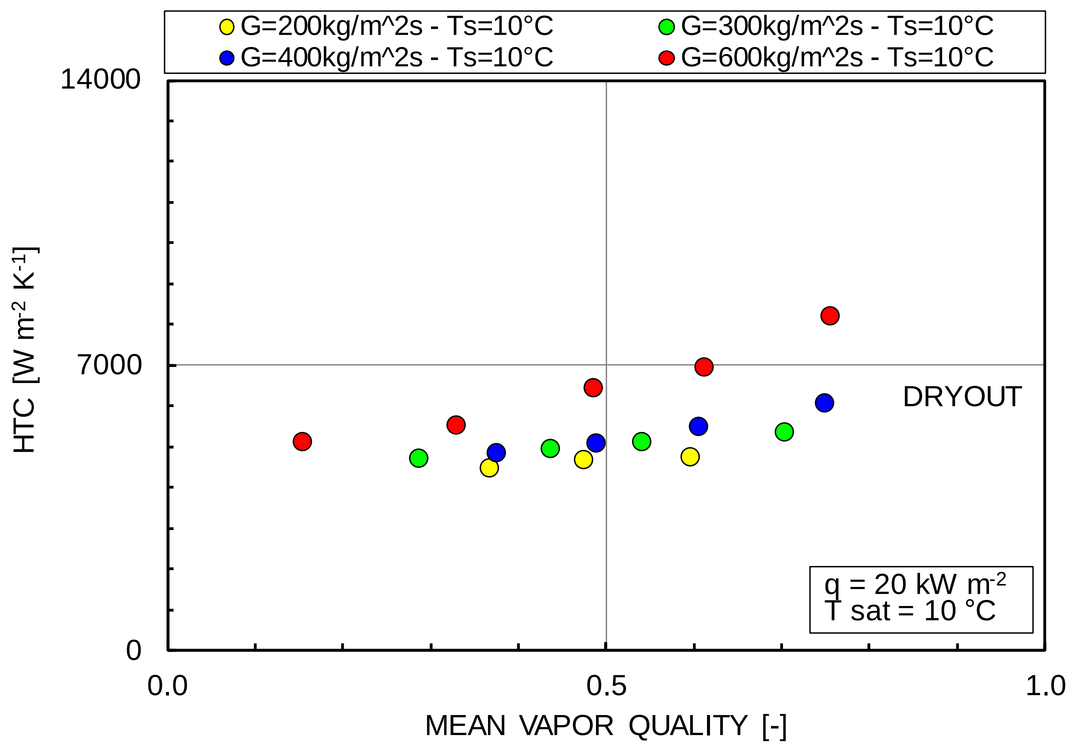


Figure 3. Boiling heat transfer coefficient vs. mean vapor quality and refrigerant mass flux at 10°C of saturation temperature and 20 kW m^{-2} of heat flux. Mass fluxes equal to 200, 300, 400, and 600 kg m^{-2} s^{-1}.

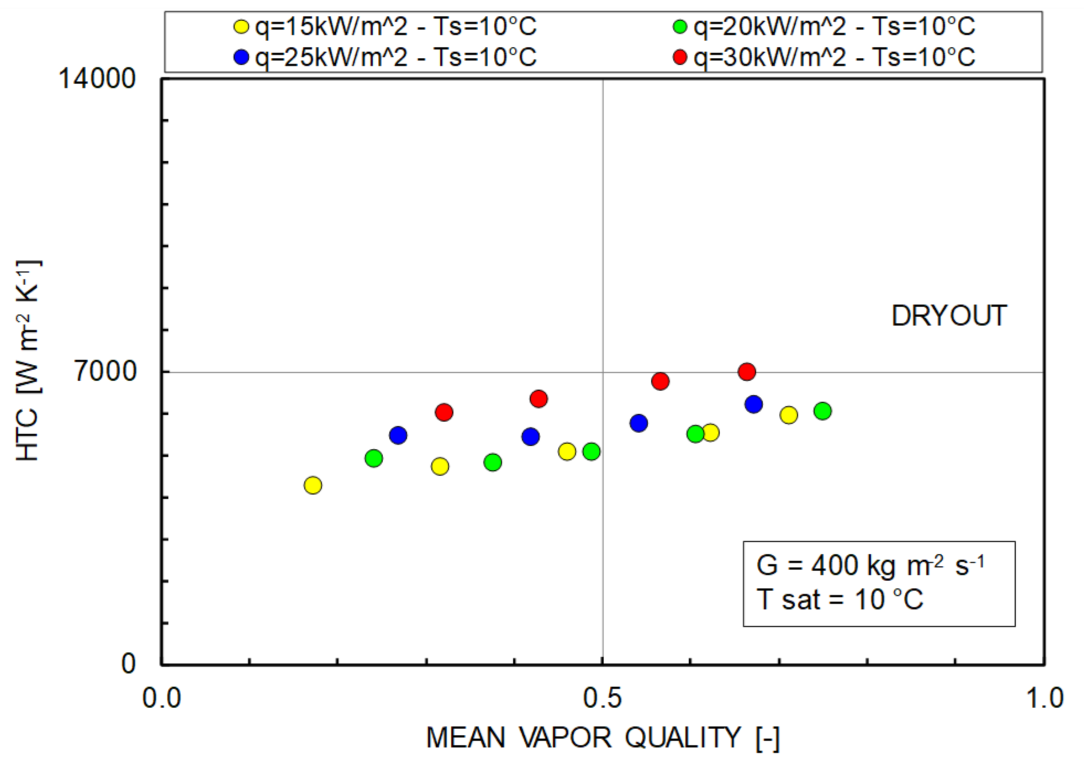


Figure 4. Boiling heat transfer coefficient vs. mean vapor quality and heat flux at 10°C of saturation temperature and 400 kg m^{-2} s^{-1} of refrigerant mass flux. Heat fluxes equal to 15, 20, 25, and 30 kW m^{-2}.

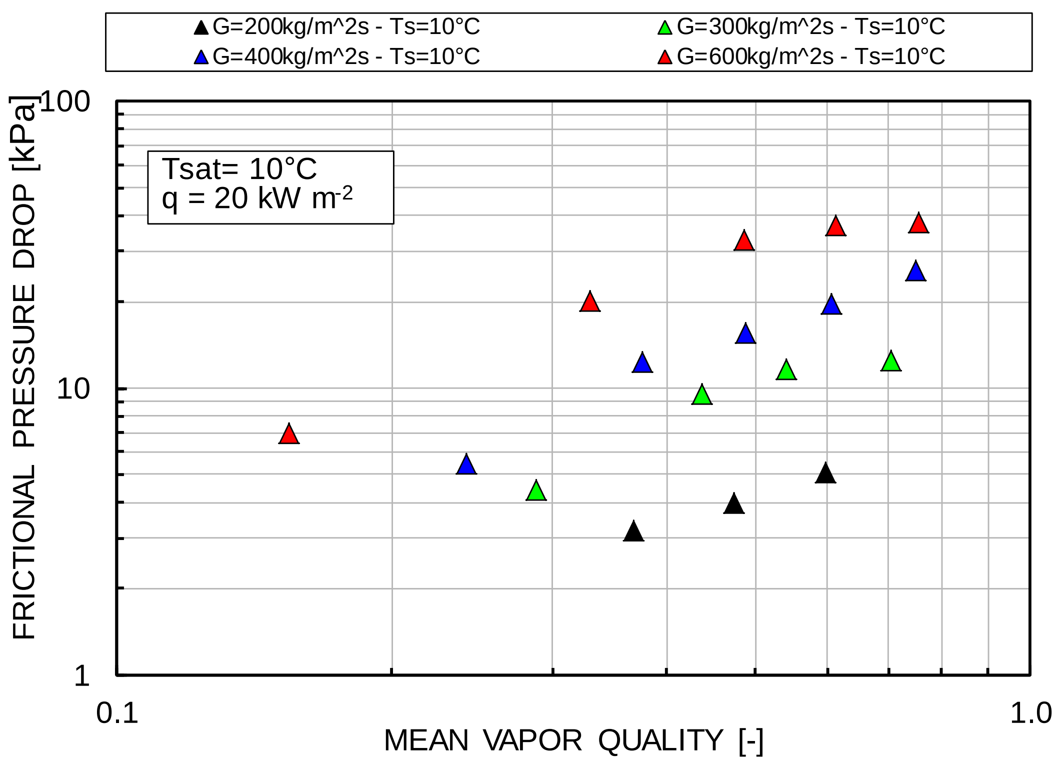


Figure 5. Frictional pressure drop vs. mean vapor quality at 10 °C saturation temperature. Mass fluxes equal to 200, 300, 400, and 600 kg m^{-2} s^{-1}.

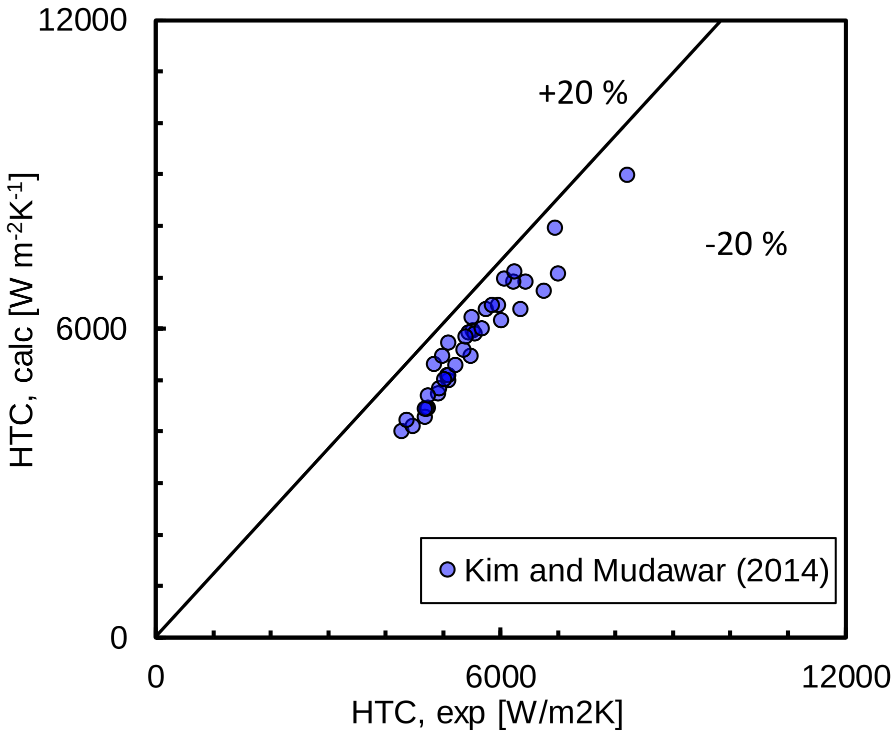


Figure 6. Comparison between experimental and calculated saturated boiling heat transfer coefficient by Kim and Mudawar (2014) correlation.

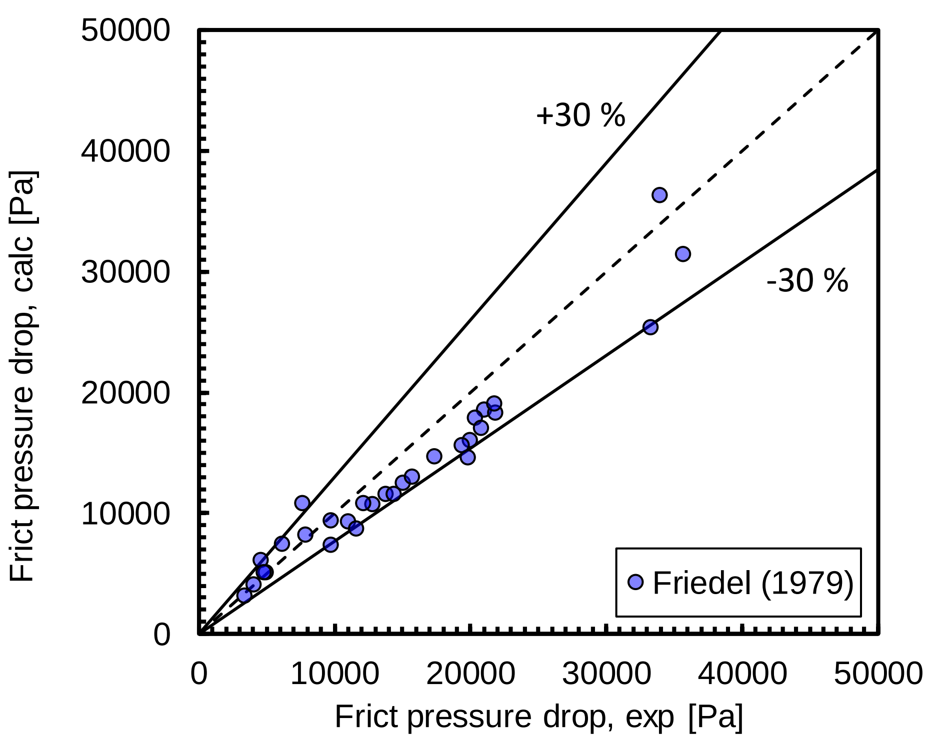


Figure 7. Comparison between experimental and calculated frictional pressure drops by Friedel (1979) correlation.

Since in these last years, R1234yf has been proposed together with R1234ze(E) as low GWP substitute to R134a, it was decided to compare the performance of these three fluids. All the data were collected under similar working conditions in the same test rig and the R1234ze(E) and R134a data were already presented by Longo et al. (2016).

Fig. 8 reports heat transfer coefficients of the three fluids at 10 °C of saturation temperature, heat flux 20 kW m^{-2}, and refrigerant mass flux equal to 600, and 200 kg m^{-2} s^{-1}, respectively. Accordingly, Fig. 9 presents a comparison between R1234yf, R1234ze(E) and R134a frictional pressure drop under the same working conditions (saturation temperature equal to 10 °C, mass flux equal to 600 and 200 kg m^{-2} s^{-1}, and heat flux equal to 20 kW m^{-2}).

At a first sight, all the fluids have similar heat transfer performance when compared under similar working conditions, as already highlighted by many other Authors (see Introduction). So it can be concluded that R1234yf together with R1234ze(E) is a viable environmentally friendly alternative to R134a, as already proposed in the literature.

By deeply analyzing the graphs, at high mass velocities (G=600 kg m^{-2} s^{-1}) for each refrigerant, heat transfer is influenced by convection boiling mechanism. In this case, as expected, convective boiling contribution is stronger during R1234ze(E) flow boiling, since it has the lowest pressure among the tested fluids. So, in some particular conditions (i.e., at high vapor qualities and high mass velocities) it outperforms the other fluids (+20% than R134a and +45% than R1234yf). This result is in accordance with what found by Sempértegui-Tapia and Ribatski (2017).

On the other hand, it is well-know that the nucleate boiling mechanism is stronger at high pressure. So, R1234yf should be the favorite at low vapor qualities and low mass velocities. Under these latest working conditions, it is possible to observe that R1234yf performs similarly to R134a, while R1234ze(E) shows the lowest heat transfer coefficients (approximately -20%).

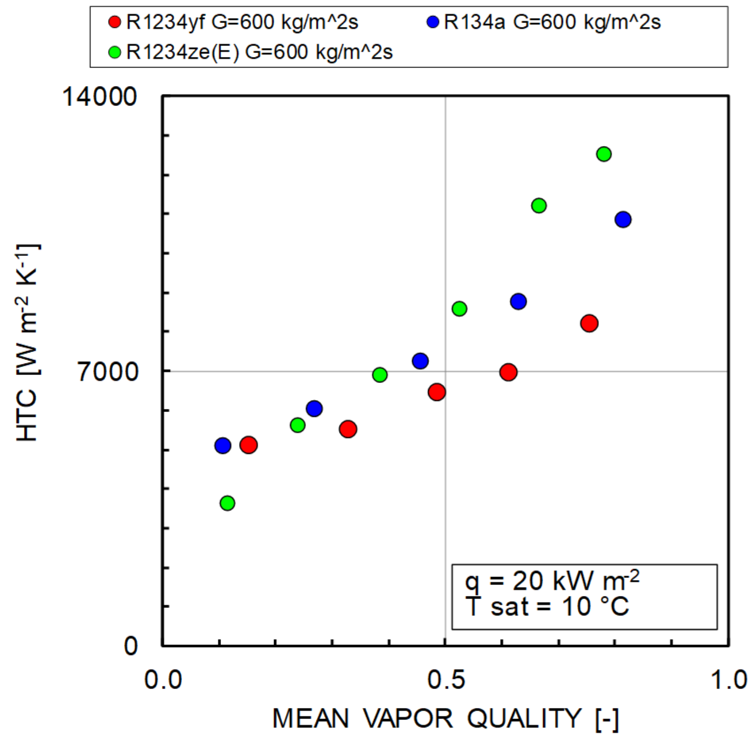
Finally, concerning frictional pressure drops, R1234yf presents the lowest pressure drop under all the investigated working conditions, up to -40% and -70% than R134a and R1234ze(E), respectively. This could be easily predicted due to the higher saturation pressure R1234yf has.

6. CONCLUSIONS

This paper presented some new experimental data collected during R1234yf flow boiling inside a 4 mm horizontal copper smooth tube. The effects of heat flux, refrigerant mass flux, and mean vapor quality have been evaluated separately. For each particular working condition, the influence of forced convection boiling and nucleate boiling contributions has been analyzed.

Following, some literature correlations were implemented and compared against the experimental data. The best heat transfer coefficient model was the recent model proposed by Kim and Mudawar (2014). The mean relative percentage deviation is - 3.2%, while the absolute percentage deviation is 6.4%. While the best frictional pressure drop model was the Friedel (1979) correlation, with a mean relative percentage deviation equal to -5.6% and a mean absolute percentage deviation of 17.4%.

Finally, since R134a is going to be phased out and some low GWP alternatives are urgently required, the R1234yf data here presented were compared against some other collected with R1234ze(E) and R134a in the same test rig. All the fluids have similar heat transfer performance when compared under similar working conditions, even if R1234ze(E) is more affected by the convective boiling contribution and for this reason in some cases it outperforms the other fluids. On the other hand, R1234yf presents a stronger nucleate boiling contribution that makes it performing similar to R134a at low vapor qualities and low mass velocities. Furthermore, R1234yf presents the lowest pressure drop under all the investigated working conditions, up to -40% and -70% than R134a and R1234ze(E), respectively.



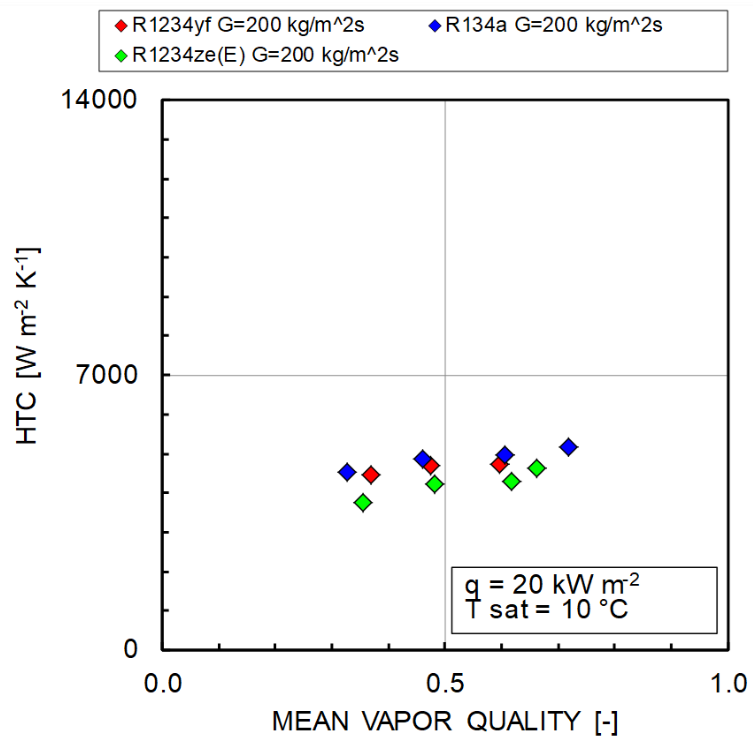
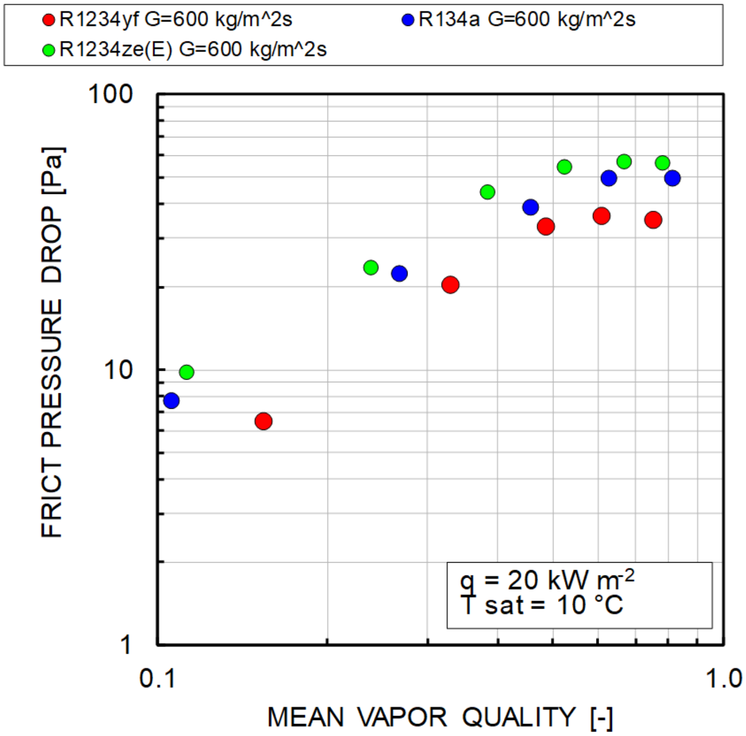


Figure 8. Heat Transfer Coefficient vs. mean vapor quality at 10 °C saturation temperature and 20 kW m^{-2} heat flux for R134a and two low GWP alternatives: R1234yf and R1234ze(E).
Refrigerant mass flux equal to 600 and 200 kg m^{-2} s^{-1}.



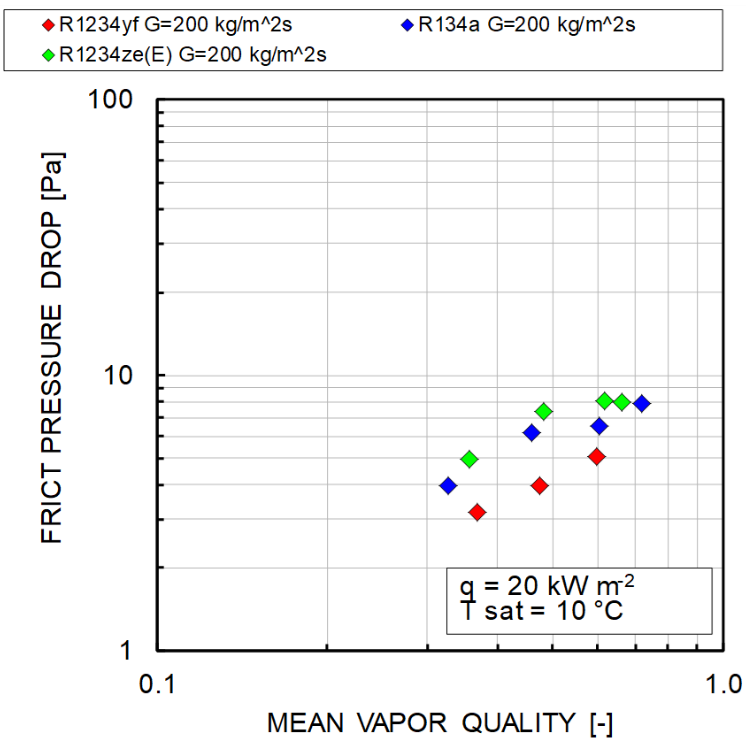


Figure 9. Frictional pressure drop vs. mean vapor quality at 10 °C saturation temperature and 20 kW m^{-2} heat flux for R134a and two low GWP alternatives: R1234yf and R1234ze(E).
Refrigerant mass flux equal to 600 and 200 kg m^{-2} s^{-1}.

NOMENCLATURE

A	heat transfer area	(m^2)
G	refrigerant mass flux	(kg $m^{-2}s^{-1}$)
h	heat transfer coefficient	(W m^{-2} K^{-1})
k	coverage factor	(-)
p	pressure	(Pa)
q	heat flux, $q = Q / A$	(Wm^{-2})
Q	heat flow rate	(W)
T	temperature	(°C)
X	vapor quality	(-)
D	difference	(-)

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