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ABSTRACT

In the recent years much attention has been paid to the use of fluorinated propene isomers for the substitution of high-GWP refrigerants in refrigeration and air conditioning applications. However, the HFOs (hydrofluoroolefins) cannot cover all the applications due to their thermodynamic properties. For the purpose to obtain a fluid that brings together the good heat transfer characteristics of HFCs, such as R410A, and the environmental friendly behavior of HFOs, in the latest literature it was attempted to blend R1234ze(E) with another refrigerant, R32. R32 has relatively low GWP and excellent thermodynamic characteristics. Therefore, a zeotropic mixture of R1234ze(E) and R32 can be used in the field of air-conditioning due to its mild impact on the environment.
In this paper, a mixture of R1234ze(E) and R32 (0.5/0.5 by mass) is investigated. In particular the frictional pressure gradient and the local heat transfer coefficients during flow boiling and condensation of this mixture in a single microchannel with 0.96 mm diameter are measured. Tests are carried out in the experimental apparatus available at the Two Phase Heat Transfer Lab of the University of Padova.
The heat transfer coefficients are compared against predicting models available in the literature. The new experimental data are also compared to heat transfer data of pure R1234ze(E) and R32. This allows to analyze the heat transfer penalization due to the mass transfer resistance of this zeotropic mixture. Pressure drop data are also used to assess predicting pressure gradient correlations.

1. INTRODUCTION

The worldwide alert about global warming has led to an increasing interest in new HVAC (heating, ventilation and air conditioning) technologies with low environmental impact. When considering this impact, both an indirect effect due to the energy consumption, and the consequent carbon dioxide emissions caused by the electricity production process, and a direct effect due to leakages of refrigerant must be taken into account. The introduction of microchannels in the field of enhanced heat transfer and in the refrigeration and air conditioning applications is surely one attempt to respond to these needs. Microchannels allow to develop compact elements which work with reduced refrigerant charge minimizing the problems of release of potentially hazardous fluids in the atmosphere and to reach high heat transfer performance. Beside achieving high heat transfer rates in compact heat exchangers, there is an increasing interest in refrigerants possessing low global warming potential (GWP). The Kyoto Protocol of the United Nations Framework Convention on Climate Change (1997) placed hydrofluorocarbons (HCFs) among the six categories of greenhouse gases because of their large values of global warming potential (GWP) and called for their phase-out. Afterwards, in 2012, the European Commission proposed to cut F-gas emissions by two-thirds by 2030. The search for alternatives primarily focuses on the use of natural refrigerants (hydrocarbons, ammonia, carbon dioxide) and new synthetic refrigerants having low GWP. However, natural refrigerants are often flammable or toxic and few single-component low-GWP refrigerants are well developed so far. Hence, for most applications an alternative to high GWP synthetic refrigerants would rely on refrigerant mixtures. In Koyama *et al.* (2010) it was attempted to blend R1234ze(E) into an other refrigerant, R32, to improve the COP and capacity of heat pump cycles. As the result of their drop-in tests with R32/R1234ze(E) 0.5/0.5 by mass, they concluded that the tested binary

mixture achieved a superior COP at some operating conditions, and this binary mixture is the most promising candidate to replace R410A.

For a cycle evolving between two temperatures (T_1 and T_2) with $T_1 > T_2$ the maximum coefficient of performance is defined by the Carnot cycle as in Eq. 1. In a cycle operating with a zeotropic mixture, the heat exchange at the evaporator and at the condenser will occur with gliding temperatures. Such cycles are sometimes called as Lorenz-cycles. The coefficient of performance for a Carnot-cycle is still valid for reversible cycles with gliding temperatures provided that T_1 and T_2 are properly evaluated. In this case the temperature T_1 and T_2 can be replaced by the mean logarithmic absolute temperature of the working fluid. For example, in a refrigerating system, if the zeotropic mixture evolves in the condenser between the temperature T_{1in} and T_{1out} due to the temperature glide, T_1 will be the mean logarithmic temperature as reported in Eq. (1).

$$COP = \frac{T_2}{T_1 - T_2} \qquad\qquad T_1 = \frac{T_{1in} - T_{1out}}{\ln\left(T_{1in}/T_{1out}\right)} \qquad\qquad T_2 = \frac{T_{2out} - T_{2in}}{\ln\left(T_{2out}/T_{2in}\right)} \qquad (1)$$

In the literature, a limited number of pressure drop and heat transfer data for mixtures of HFOs and HFCs in small diameter channels ($D_h < 3$ mm) are available. Regarding the vaporization of the mixture R1234ze(E)/R32, Kondou *et al.* (2013) performed flow boiling tests in a horizontal microfin tube of 5.21 mm inner diameter at different mass compositions. The heat transfer coefficient and pressure drop are 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}. Hossain *et al.* (2013) measured the heat transfer coefficient of the mixture R1234ze(E)/R32 at 55/45% mass composition inside a 6 mm tube. However no data is available for microchannels.

In this work the condensation and vaporization heat transfer performance of the mixture R32/R1234ze(E) (0.5/0.5 by mass) was investigated inside a single circular cross section microchannel with an inner diameter of 0.96 mm. The experimental characterization of the R32/R1234ze(E) mixture as a refrigerant includes the measurement of the frictional pressure drop in adiabatic regime in a dedicated test section. The heat transfer data here presented are obtained in a test section where heat is transferred between the refrigerant and a secondary fluid flowing in a separated loop. During condensation tests, water is used as cooling fluid to subtract heat from the refrigerant; during evaporation experiments, the boiling process is governed by controlling the inlet temperature of the heating secondary fluid.

2. EXPERIMENTAL APPARATUS

Test runs have been performed at the Two Phase Heat Transfer Lab of the University of Padova. In the test rig, two thermal baths are employed: one provides distilled water used as secondary fluid for the heat transfer in the test section, the other serves brine at 5°C to the subcooler and to the auxiliary loop of the post-condenser, in which the working refrigerant is subcooled after exiting the measuring sector. The subcooled refrigerant passes through a filter drier before entering an independently controlled gear pump, which allows setting the mass flow measured by a Coriolis-effect mass flow meter. Before entering the test section, the working refrigerant can be either subcooled or heated up, vaporized and superheated. The refrigerant is finally sent through the test section for pressure drop measurement or through the test section for heat transfer investigation. Each test section is described below.

2.1 Test section for pressure drop measurements
During the pressure drop test runs, the refrigerant enters the pre-conditioning sector either as subcooled liquid or as superheated vapor. The thermodynamic state of the refrigerant at the inlet of the pre-conditioning sector is determined from temperature and pressure measurements. The pre-conditioning sector is a counter-flow heat exchanger in which flow rate and inlet temperature of distilled water are set to get the desired vapor quality of the refrigerant at the inlet of the measuring sector. The mass flow rate of the distilled water is measured by a Coriolis-effect mass flow meter, the water outlet temperature in the pre-conditioning sector is measured by a T-type thermocouple and the water temperature difference between inlet and outlet is measured by a copper constantan triple-junction thermopile. The refrigerant vapor quality at the inlet of the measuring sector is obtained from the energy balance in the pre-conditioning sector.

The measuring sector consists of a copper rod with a 0.96 mm diameter internal bore obtained by drawing and having an inner surface roughness *Ra* = 1.3 μm, as from the measurements performed following the EN ISO 4287 - 1998 standard. Two pressure ports were made directly on the copper rod and they are connected through pressure lines to pressure transducers for the measurement of the absolute pressure at the inlet and of the pressure drop through the measuring sector. The length of the measuring sector between the pressure ports is 0.22 m.

2.2 Test section for heat transfer coefficients measurements

The test section is completely described in Matkovic *et al.* (2009) and consists of two diabatic sectors, a 50 mm pre-conditioning sector and a 230 mm measuring sector and two stainless steel adiabatic sectors, 31 mm long and located at the ends of the measuring sector. The diabatic sectors are obtained from two segments of the same microchannel used for the pressure drop tests. Furthermore, the external surface of the microchannel underwent a machining process for the realization of the complex secondary fluid channel which is closed on top with plastic sheath. Both in pre-conditioning sector and in measuring sector, the distilled water and the refrigerant flow in countercurrent. Each loop is provided with a Coriolis-effect mass flow meter to measure the water flow rate. The inlet and outlet water temperatures are measured by thermocouples at the ends of each sector, while the water temperature differences across both sectors are measured by four and three junction copper-constantan thermopiles. In the measuring sector, to reach a compromise between measurement accuracy and feasible design, 15 thermocouples have been installed in the water path and 13 thermocouples are used to measure the wall temperature at different positions. At the inlet of the pre-conditioning sector, a pressure port was realized and a T-type thermocouple was places on the outer surface of the connection tubes, so the thermodynamic state of the refrigerant is determined by means of pressure and temperature measurements. When operating in condensation mode, the pre-conditioning sector works as a desuperheater. On the other hand, when operating in flow boiling mode, the pre-conditioning sector serves to control the subcooling degree of the refrigerant at the inlet of the measuring sector.

3. TWO-PHASE PRESSURE DROP

Two-phase frictional pressure drop tests have been performed during adiabatic flow of R32/R1234ze(E) mixture (0.5/0.5 by mass composition) at pressure of 18 bar corresponding to a dew temperature of 43.7°C and a bubble temperature of 36.3°C inside the circular cross section microchannel at mass velocities ranging from 200 kg m^{-2} s^{-1} to 600 kg m^{-2} s^{-1}.

A limited number of publications reported data of thermophysical properties for the mixture R32/R1234ze(E). Among them Akasaka (2013) emphasizes that the models proposed are designed not for scientific use but for engineering use and they are based on limited experimental data. For the R32/R1234ze(E) the model proposed by Akasaka (2013) is here implemented in REFPROP (Lemmon *et al.*, 2013) to determine the mixture properties.

The vapor quality for each experimental point, assuming an equilibrium state, is calculated from the pressure, enthalpy and mixture composition as proposed in Kondou *et al.* (2013):

$$x(z) = f\left(p(z), h(z), X_{R1234ze} \right) \tag{2}$$

where $X_{R1234ze}$ is the mass fraction of R1234ze(E) and *h(z)* is substituted with $h_{in,MS}$, the specific enthalpy of the refrigerant at the inlet of the measuring sector that results from the energy balance in the pre-conditioning sector according to Eq. 3:

$$h_{in,MS} = h_{in,PS} - \frac{\dot{m}_{water} \cdot c_{water} \cdot \Delta T_{water,PS}}{\dot{m}_{ref}} \tag{3}$$

The specific enthalpy of the refrigerant at the inlet of the pre-conditioning sector $h_{in,PS}$ is calculated from the measured temperature and pressure at inlet. When performing test runs with vapor quality lower than 0.5, the refrigerant enters the pre-conditioning sector as subcooled liquid at 23-26°C, therefore in the pre-conditioning sector heat exchanger vaporization occurs. On the other hand, to get experimental points with vapor quality higher than 0.5, the fluid enters the pre-conditioning sector as superheated vapor at 53-57°C and a partial condensation occurs.

The experimental pressure gradient for mixture R32/R1234ze(E) is plotted against vapor quality in Figure 1. In Figure 2, the frictional pressure gradient calculated using the Del Col *et al.* (2013a) model is plotted along the experimental data. The model was validated for pressure drop inside microchannels during the flow of halogenated fluids, mixtures and carbon dioxide and in the present comparison the effect of surface roughness is accounted for. The present mixture two-phase pressure drop is satisfactorily predicted by the Del Col *et al.* (2013a) model.
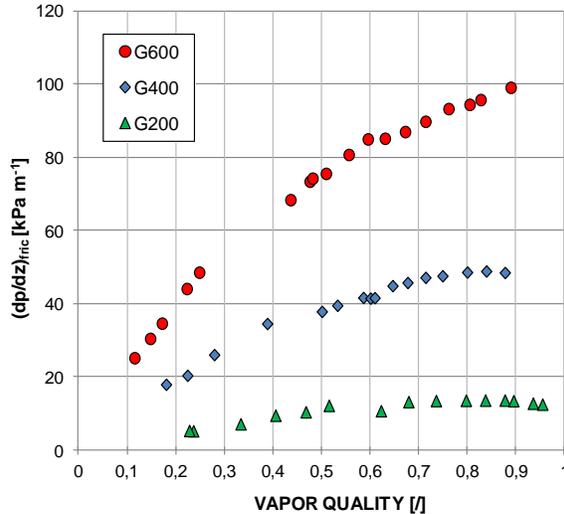
Figure 1: Experimental frictional pressure gradient versus vapor quality for the refrigerant mixture
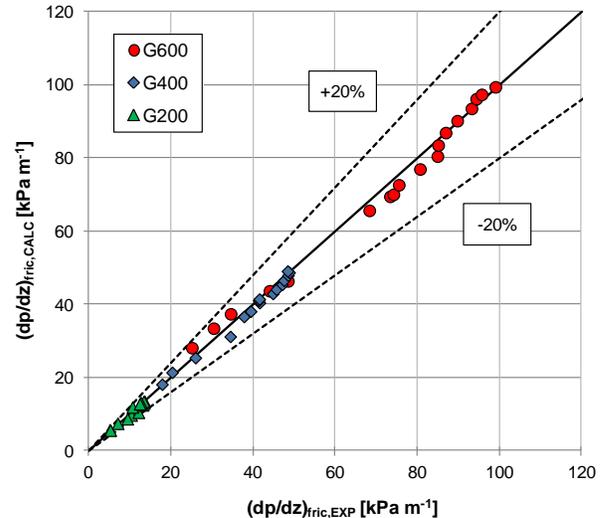
Figure 2: Experimental and calculated frictional pressure gradient using the Del Col *et al.* (2013a) model

4. CONDENSATION TESTS

4.1 Data reduction
The local heat flux is calculated from the slope of the water temperature profile along the test section:

$$q' = \frac{\dot{m}_{water}\, c_{water}}{\pi\, d} \frac{dT_{water}(z)}{dz} \tag{4}$$

where dT_{water}/dz is the derivative of the polynomial equation interpolating the measured water temperature along z. The heat flow rate transferred to the secondary fluid up to a certain position z is obtained integrating the local heat flux from the refrigerant inlet to the position z:

$$q(z) = \pi\, d \int_0^z q'(z)\, dz \tag{5}$$

During a condensation test run the mixture enters the pre-conditioning sector as superheated vapor: temperature and pressure are measured and the enthalpy evaluated for the vapor. In the pre-conditioning sector the refrigerant is cooled down and the Eq. (3) can be solved with a thermal balance on the water side and the refrigerant enthalpy at the inlet of the measuring sector is determined. Along the measuring sector, using Eqs. (4-5) the heat flow rate and then the refrigerant enthalpy at the position z can be calculated as reported in Eq. (6):

$$h(z) = h_{in,\,MS} - \frac{q(z)}{\dot{m}_{ref}} \tag{6}$$

The refrigerant vapor quality and temperature, assuming an equilibrium state, are evaluated using respectively Eq. (2) and Eq. (7):

$$T_{ref}(z) = f\left(p(z), h(z), X_{R1234ze}\right) \tag{7}$$

where *p(z)* is obtained from the pressure profile along the microchannel calculated by implementing the Del Col *et al.* (2013a) two-phase pressure gradient correlation. The calculated pressure gradient is then corrected by an empirical coefficient to match the experimental total pressure drop to the value measured by the differential pressure transducer.

The local heat transfer coefficient inside the microchannel is obtained as:

$$\alpha(z) = \frac{q'(z)}{T_{ref}(z) - T_w(z)} \tag{8}$$

The wall temperature is measured by means of thermocouples embedded in the copper tube. Since the wall thermocouples are installed 0.5 mm far from the internal tube surface, a correction is required for the wall temperature measurement due to the distance between the thermocouple and the inner tube surface.

4.2 Experimental results and comparison against the model by Cavallini et al. (2006)

The condensation test runs are performed with a mass velocity ranging from 200 kg m^{-2} s^{-1} to 800 kg m^{-2} s^{-1} at a pressure of 17.7 bar. In Figure 3 the heat transfer coefficient for the R32/R1234ze(E) mixture (0.46/0.54 by mass composition) is reported versus vapor quality. The heat transfer coefficients display the usual trend that one would expect for condensation inside plain tubes: the heat transfer coefficient decreases as the condensation proceeds and the vapour quality decreases in the channel. Results show that the heat transfer coefficient increases with mass velocity although at low mass velocity and vapor quality the heat transfer coefficient does not vary significantly.

Cavallini *et al.* (2006) proposed a method to predict condensation heat transfer coefficients inside horizontal smooth tubes with internal diameter greater than 3 mm. The model was validated against data for HCFCs, HFCs, HCs, carbon dioxide, ammonia and water. As reported in Matkovic *et al.* (2009) and in Del Col *et al.* (2013b) the model was in good agreement with experimental data for pure R32 and pure R1234ze(E). In case of mixture, models for pure vapor condensation can in principle be applied when complete mixing, both in the liquid and in the vapor phase, is reached and overall equilibrium is maintained provided that the additional resistance due to the mass transfer is properly accounted for. So the Cavallini *et al.* (2006) model can be applied to the present mixture using the Bell and Ghaly (1973) and Silver correction (1947) to account for the mass transfer thermal resistance:

$$\alpha_m = \left(\frac{1}{\alpha} + \frac{\partial q_{sg}/\partial q_t}{\alpha_g}\right)^{-1} \tag{9}$$

where α is the condensate film heat transfer coefficient calculated with the model for pure fluids, $\partial q_{sg}/\partial q_t$ is the ratio of sensible to total heat duty, α_g is the heat transfer coefficient of the vapor phase flowing alone in the tube and calculated with the Dittus-Boelter equation. The ratio of the sensible to total heat duty, if the ratio of d*T/*d*h* remains approximately constant during the condensation process it can be expressed with good approximation as reported in Eq. (10).

$$\frac{\partial q_{sg}}{\partial q_t} \approx x c_{p,g}\left(\frac{dT}{dh}\right) \approx x c_{p,g} \frac{\Delta T_{GL}}{\Delta h_m} \tag{10}$$

The comparison between the experimental data and the Cavallini *et al.* (2006) model with the Silver-Bell-Ghaly correction (SBG) is reported in Figure 4. The model is able to predict well the experimental data: the absolute deviation e_{AB} is about 5% and the standard deviation σ_N is 4%.
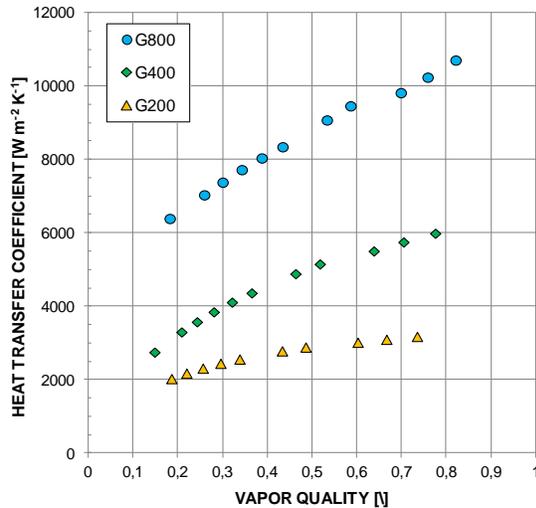
Figure 3: Experimental local heat transfer coefficient for the zeotropic refrigerant mixture R32/R1234ze(E).
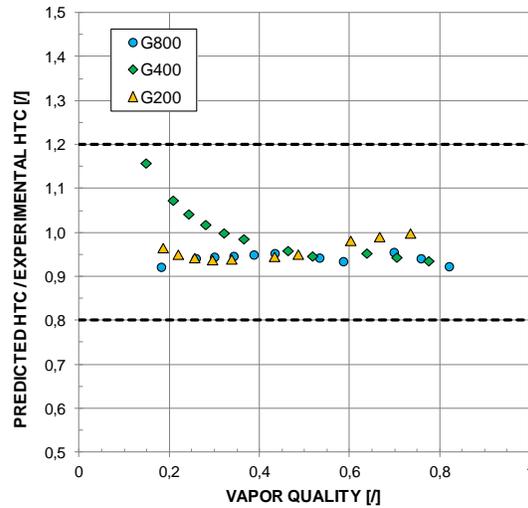
Figure 4: Comparison between the experimental heat transfer coefficient and prediction using the Cavallini *et al.* (2006) model with the SBG correction.

5. FLOW BOILING TESTS

5.1 Data reduction

During flow boiling test runs, the heat transfer coefficient is not measured by imposing the heat flux; instead, the boiling process is governed by controlling the inlet temperature of the heating secondary fluid. Since the heat flux is not directly fixed, it must be obtained indirectly, from the slope of the secondary fluid temperature profile, as reported in Eq. (4). In the calculation, two different independent procedures have been adopted for the determination of the fitting polynomial degree of the water temperature and thus the calculation of the heat flux. The first criterion, dubbed as physical, is based on the assumption that all the calculated values by the polynomial interpolation of the water temperatures should be below ±0.05 K with respect to the experimental values. Beside this, a statistical method based on the choice of the simplest fitting polynomial has been used. This statistic approach uses the R^2 statistic parameter, and adjusts it on the basis of the residual degrees of freedom: this is the reason of the name R^2_{adj} (Rawlings *et al.* 1998).

The local heat transfer coefficient $\alpha(z)$ inside the microchannel is obtained as the ratio of heat flux to temperature difference:

$$\alpha(z) = \frac{q'(z)}{T_w(z) - T_{ref}(z)} \tag{11}$$

The refrigerant temperature and vapor quality are calculated, as for the condensation test, using Eqs. (2 and 7).
The refrigerant enthalpy used in Eqs. (2 and 7) at the position of the *i-th* wall thermocouple is obtained from a heat balance between the mixture and the water that flows in counter-flow:

$$h(z) = h_{sub} + \frac{\dot{m}_{water}}{\dot{m}_{ref}} c_{water} \left(T_{water,out} - T_{water}(z) \right) \tag{12}$$

where the enthalpy of the subcooled refrigerant is determined from the measured inlet pressure and temperature.
Since the refrigerant pressure is measured at inlet and outlet of the test section, a linear variation is usually assumed in the channel. However, during vaporization, the vapor quality changes along the microchannel and therefore the pressure gradient varies in the measuring sector; for this reason, a linear interpolation of the saturation pressure between inlet and outlet may lead to errors in the determination of the heat transfer coefficient. As for condensation

test in the present work the pressure profile along the microchannel is calculated by implementing the Del Col *et al.* (2013a) two-phase pressure gradient correlation.

5.2 Flow boiling results

Flow boiling test runs have been performed with the mixture R32/R1234ze(E) at 0.5/0.5 by mass composition, at mass velocity ranging between 300 and 600 kg m^{-2} s^{-1} and at a pressure of 14 bar. As previously mentioned, in this work the heat flux is not electrically imposed: the controlled parameters are the inlet fluid temperatures and the mass flow rate of the secondary fluid. As a consequence, for each test run, not only the heat transfer coefficient and the vapor quality vary along the channel but also the heat flux is a variable. Therefore, with the present technique, it is impossible to get the heat transfer coefficient variation with vapor quality at fixed heat flux as reported in literature using electric heating. In order to explore the effects of heat flux and vapor quality on the heat transfer coefficient, several tests were made at constant refrigerant mass velocity and varying the inlet water conditions (mass flux and temperature).

The effect of heat flux on the heat transfer coefficient can be better understood in Figure 5, where the local heat transfer coefficient has been reported at constant mass velocities and limited vapor quality range around 0.3. At a constant vapor quality, the heat transfer coefficient increases with heat flux. The present heat transfer coefficients are dependent on heat flux even if at the higher heat flux the heat transfer coefficient seems to become less dependent on the heat flux. By processing data at constant heat flux, it is possible to get information on the effect of vapor quality. Figure 6 shows the experimental trend of local heat transfer coefficient for the mixture R32/R1234ze(E) at different mass velocities and vapor quality, for a heat flux of about 100 kW m^{-2}. The heat transfer coefficient of the refrigerant mixture decreases with vapor quality (Figure 6). In Figure 5 and Figure 6 a mass velocity effect is shown. The heat transfer coefficient increases with the mass velocity at the higher heat flux.
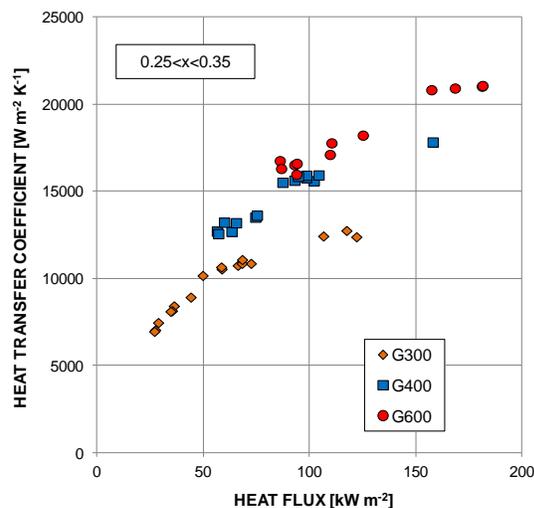
Figure 5: Local heat transfer coefficient versus heat flux at given vapor quality
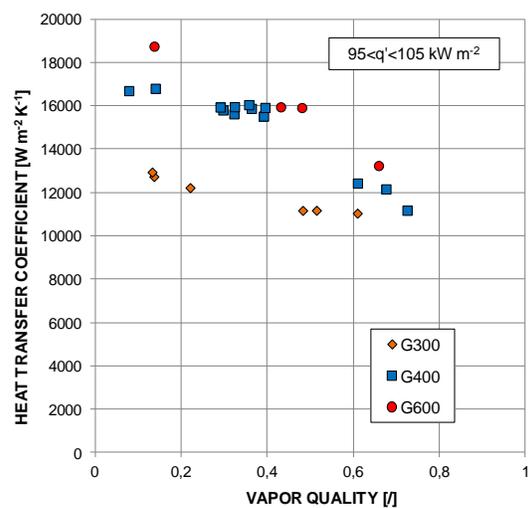
Figure 6: Local heat transfer coefficient versus vapor quality during vaporization at constant heat flux

## 5.3 Comparison against the model by Sun and Mishima (2009)

The correlation by Sun and Mishima (2009) is based upon the Lazarek and Black (1982) model and takes the effect of Weber number into account. An improvement in the prediction of heat transfer coefficients for pure fluids with respect to the Lazarek and Black (1982) model has been found by Del Col *et al.* (2013c, 2013d). As done for condensation heat transfer correlation, during vaporization of a zeotropic mixture an adequate model or correction to take into account the additional mass transfer resistance must be considered. To take into account the sensible heating of the vapor phase, as reported by Cavallini *et al.* (1998), in Eq. (9) α is replace by α_f which is the heat transfer coefficient pertinent to the liquid film. This can be calculated with an appropriate flow boiling correlation utilizing the physical properties of the mixture correcting the nucleate boiling component for mass diffusion effect as suggested by Thome (1996). The correction factor F_c (Eq. 13) was analytically developed by Thome (1989).

$$F_c = \left[1 + \left(\frac{\alpha_{fb,id}\ \Delta T_{GL}}{q'} \right) \left(1 - e^{-\left(\frac{Bq'}{\rho_l \Delta h_m \beta_l} \right)} \right) \right] \tag{13}$$

The heat flux to be used in Eq. (13) is the local nucleate boiling heat flux but, as a first approximation, the total heat flux can be used. In the correlation of Sun and Mishima (2009) F_c was applied as reported in Eq. (14)

$$\alpha_f = \frac{6 \mathrm{Re}_{lo}^{1.05} \left(Bo \cdot F_c \right)^{0.54}}{We_l^{0.191} \left(\dfrac{\rho_l}{\rho_g} \right)^{0.142}} \frac{\lambda_l}{d} \tag{14}$$

In the database considered in the following comparison the heat flux ranges between 20 and 180 kW m^{-2} and vapor quality between 0.1 and 0.8. Only data points characterized by vaporization prior to onset of dryout are considered. Considering the present mixture data, the Sun and Mishima (2009) corrected correlation underestimates the experimental data and is able to predict almost all the data points in the ±30% error band.
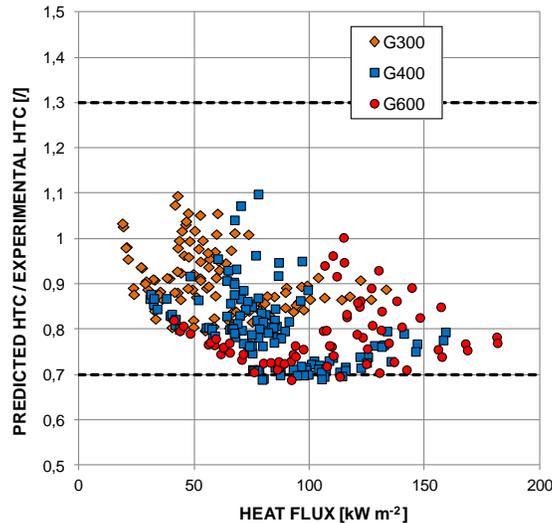

Figure 7: Prediction of the vaporization heat transfer coefficient with the Sun and Mishima (2009) model corrected for zeotropic mixture.

6. DEGRADATION OF THE HEAT TRANSFER COEFFICIENT

Figure 8 reports the heat transfer coefficient of the mixture and the pure components at G = 400 kg m^{-2} s^{-1}, during vaporization (vapor quality equal to 0.4 and 0.2) and condensation (vapor quality equal to 0.74 and 0.46).

During vaporization the present refrigerant mixture displays lower heat transfer coefficients than those of the single pure fluids. The heat transfer degradation with reference to an ideal behavior as a pure fluid is about 50% at the two values of vapor quality. This degradation is mostly caused by the bubble temperature increase at the bubble interface or at the liquid-vapor interface due to the mole fraction gradient at the interface, created by the volatility difference of the two pure components. During the evaporation process the liquid becomes richer in R1234ze(E) (less volatile component) and the vapor becomes richer of R32 (the more volatile). The heat transfer coefficient reduction is caused by the diffusion of the more volatile component to the interface and the concentration of the less volatile component at the heated surface is not favorable to bubble nucleation. Beside this, a part of the heat transfer coefficient reduction is due to the added heat flow rate needed to heat liquid and vapor to the boiling temperature,

constantly increasing along the tube. Similar results displaying heat transfer coefficients during vaporization of the R32/R1234ze(E) lower than those of the single components, are reported in Kondou *et al.* (2013).
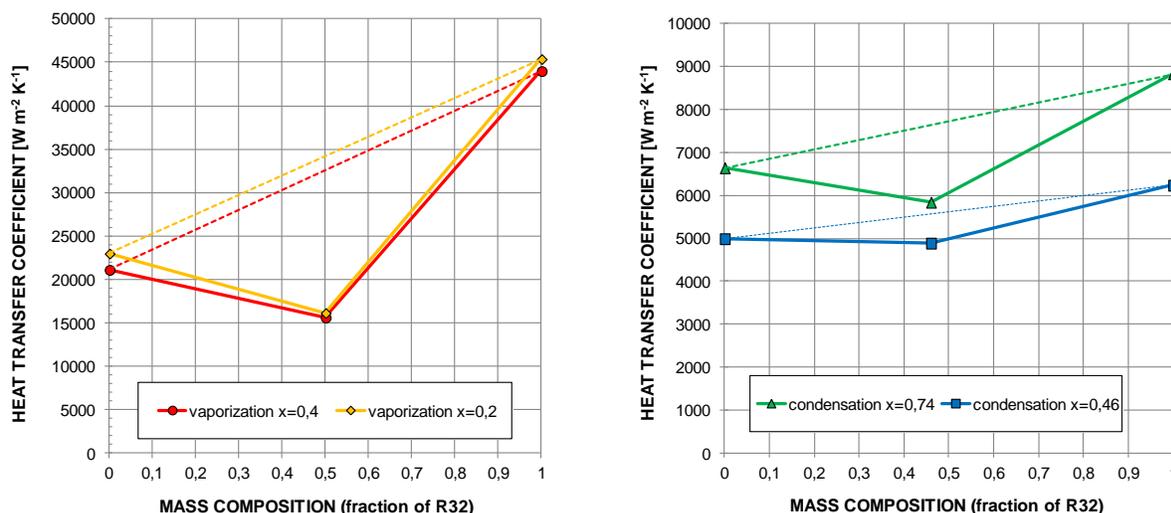
Figure 8: Experimental local heat transfer coefficient during vaporization (left) and condensation (right) for the zeotropic refrigerant mixture R32/R1234ze(E) and its pure components at G=400 kg m^{-2} s^{-1}. Dashed lines refer to an ideal mixing rule. A mass composition equal to 0 corresponds to pure R1234ze(E).

During condensation, the heat transfer coefficient for pure R1234ze(E), R1234ze(E)/R32 at 46/54% composition and pure R32 are reported at G = 400 kg m^{-2} s^{-1} and two different vapor qualities. The figure shows that the heat transfer coefficients for the mixture are always lower than the values calculated by an ideal linear behavior from the values pertaining to the pure components at the same working conditions. The penalization at 400 kg m^{-2} s^{-1} and x = 0.74 is about 23% and at x = 0.46 equal to 12%.

7. CONCLUSIONS

In this work the thermal performance during condensation and vaporization inside a 0.96 mm diameter microchannel of the refrigerant mixture R32/R1234ze(E) is investigated. During condensation test runs, data have been collected at a pressure of 17.7 bar and mass velocities ranging between 800 kg m^{-2} s^{-1} and 200 kg m^{-2} s^{-1}. The penalization of the heat transfer coefficient can be estimated by comparing the measured heat transfer coefficients versus an ideal linear behavior of the mixture between the values pertaining to the pure components at the same flow conditions. At 400 kg m^{-2} s^{-1} mass velocity and x=0.74, such penalization is about 23% and at x=0.46 it is around 12%. The present experimental heat transfer coefficients are well predicted by the Cavallini *et al.* (2006) model coupled with the Silver-Bell-Ghaly correction to account for the additional mass transfer resistance.

During flow boiling tests a secondary fluid is used to transfer heat to the vaporizing mixture instead of fixing the heat flux by electric heating. The heat transfer coefficient at a pressure of 14 bar is measured. It decreases with vapor quality and is dependent on the heat flux but such dependence diminished at higher heat flux. If the heat transfer coefficients for the refrigerant mixture are compared with those of the pure fluid components one can see the effect of the additional mass transfer resistance. The present data have been compared with the Sun and Mishima (2009) correlation, corrected to account for the effect of the mass transfer resistance and the sensible heating of the vapor phase. The correlation corrected for zeotropic mixtures predicts almost all the data in the ±30% error band.
To complete the performance analysis of this refrigerant mixture frictional pressure drop during adiabatic two-phase flow has also been measured. A good agreement between the experimental data and those calculated with the Del Col *et al.* (2013a) model has been found.

NOMENCLATURE

B	scaling factor assumed to be 1		Bo	Boiling number (-)
c	specific heat (J kg^{-1} K^{-1})		COP	coefficient of performance (-)
d	hydraulic diameter (m)		F_c	mixture correction factor (-)
G	mass velocity (kg m^{-2} s^{-1})		h	specific enthalpy (J kg^{-1})
$\dot{m}$	mass flow rate (kg s^{-1})		p	pressure (Pa)
q'	heat flux (W m^{-2})		q	heat flow rate (W)
Ra	surface roughness (μm)		T	temperature (K)
Re	Reynolds number (-)		We	Weber number (-)
x	thermodynamic vapor quality (-)		X	mass fraction (-)
z	axial position (m)			

Greek symbols

α	heat transfer coefficient (W m^{-2} K^{-1})
α_f	heat transfer coefficient pertinent to the liquid film (W m^{-2} K^{-1})
α_g	convective heat transfer coefficient of the vapor phase (W m^{-2} K^{-1})
$\alpha_{fb,id}$	ideal heat transfer coefficient calculated with pure refrigerant correlations (W m^{-2} K^{-1})
β_l	= 0.0003 (m s^{-1}) liquid-phase mass transfer coefficient
ΔT	temperature difference (K)
λ	thermal conductivity (W m^{-1}K^{-1})
ρ	density (kg m^{-3})

Subscripts

fb	flow boiling		*g*	vapor
GL	glide		*i-th*	i-th thermocouple position
id	ideal		*in*	inlet
l	liquid		*lo*	liquid only
m	mixture		*MS*	measuring sector
out	outlet		*p*	pressure
PS	pre-conditioning sector		*ref*	refrigerant
s	sensible		*sub*	subcooling
t	total		*w*	wall

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