BEHAVIOUR OF REFRIGERANT MIXTURES IN HEAT EXCHANGERS

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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 a direct effect due to leakages of refrigerant must be taken into account. There is an increasing interest in refrigerants possessing low global warming potential (GWP). After the Kyoto Protocol, 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. For many applications an alternative to high GWP synthetic refrigerants would rely on refrigerant mixtures.

McLinden et al. (2014) studied several HFOs recognizing the potential of R1234yf and R1234ze(E). However, these refrigerants are not suitable as drop-in refrigerants for R410A and R404A. In order to guarantee high system COPs and mitigate the environmental impact, refrigerant mixtures of HFOs and HFCs have been proposed. Condensation of binary mixtures of HFCs and HFOs have been recently investigated, although a limited number of heat transfer data for mixtures of HFOs and HFCs is available. Hossain et al. (2013) studied the heat transfer of the mixture R1234ze(E)/R32 at 55/45% mass composition inside a 6 mm tube. Del Col et al. (2015) studied condensation of R32/R1234ze(E) at different mass compositions inside a 0.96 mm diameter channel. In Azzolin et al. (2016) the heat transfer of the same mixture during flow boiling has been investigated.



Fig. 1 Temperature-composition diagram for the R32/R1234ze(E) mixture at 14 bar saturation pressure

More recently the heat transfer with ternary mixtures has been reported in the literature. For example, Azzolin et al. (2019) presents the condensation heat transfer coefficients of ternary low GWP zeotropic mixtures R455A (R32, R1234yf and R744 at 21.5/75.5/3.0% by mass composition) and R452B (R32, R1234yf and R125 at 67.0/26.0/7.0% by mass composition).

The temperature-composition diagram for the R32/R1234ze(E) mixture is reported in Fig. 1 at 14 bar pressure. For the 0.50/0.50 mixture at 14 bar pressure, the dew point temperature is 26.2°C and the bubble point temperature is 33.9°C, with a temperature glide of 7.7 K. Similar diagrams can also be plotted for the other mixtures discussed in this paper, but in the case of ternary mixtures, the temperature-composition diagram must be represented using surfaces.

All these mixtures display a temperature difference between dew and bubble temperature at given pressure, and therefore we call them zeotropic. A summary of the mixtures considered in this paper, together with their main pure components, is reported in Table 1.

Each mixture is characterized by a different value of the zeotropic glide. This will affect the temperature profiles in the heat exchangers (evaporator and condenser). In some cases it may lead to a better matching of the two profiles, in other cases it may cause additional problems (for example when the temperature glide is very high). An example of possible configurations can be seen in Fig. 2.

Fluid	R32/R1234ze(E) at 0.5 /0.5 by mass	R455A	R452B	R32	R1234yf	R1234ze(E)
GWP _{100 years}	339	146	676	677	<1	<1
p _{sat} [bar]	18.0	17.6	22.85	24.78	10.18	7.66
T _{bub} [°C]	36.3	35.06	39.40	-	-	-
Glide [K]	7.45	9.81	1.12	-	-	-

Table 1. Properties of mixtures and their pure components at 40°C mean saturation temperature



Fig. 2 Examples of temperature profiles in counter-flow heat exchangers using zeotropic mixtures: evaporator (left) and condenser (right).

2 EXPERIMENTAL APPARATUS

The test runs have been performed at the Two Phase Heat Transfer Lab of the University of Padova. In the test rig 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 or condensed. The refrigerant is finally sent through the test section for heat transfer investigation.

The measuring sector has been made from a copper rod with a 0.96 mm internal diameter having an inner surface roughness $Ra = 1.3 \mu m$.

Both flow boiling and condensation heat transfer has been measured inside this test section, with the aim of studying the heat transfer performance of such mixtures compared to their pure fluid components.

3 TWO-PHASE HEAT TRANSFER

Fig. 3 reports the heat transfer coefficient of the mixture R32/R1234ze(E) (0.5 /0.5 by mass) and its pure components at G = 400 kg m⁻² s⁻¹, during vaporization at a pressure of 14 bar.

As shown in Fig. 3, the present refrigerant mixture displays lower heat transfer coefficients as compared to the pure components R1234ze(E) and R32. The heat transfer degradation between the values pertaining to the pure fluids is about 50% for the operating conditions reported on the graph. This degradation is partially caused by the bubble point temperature increase at the liquid-vapor interface due to the mole fraction gradient at the interface. During the evaporation process the liquid becomes richer in R1234ze(E) (less volatile component) and the vapor becomes richer in R32 (more volatile component). Part of the heat transfer coefficient reduction is caused by the diffusion of the more volatile component to the interface and the concentration of the least volatile component at the heated surface is not favorable to the vaporization process. Beside this, some additional heat flow rate is needed to heat the liquid and the vapor to the boiling temperature, constantly increasing along the tube.

Due to this degradation of the heat transfer coefficient, models developed for pure fluids cannot be directly applied to vaporization of the present mixture.





Fig. 3 Local heat transfer coefficient versus vapor quality during flow boiling of the pure fluids R32 and R1234ze(E) and their 50/50% mixture at G = 400 kg $m^{-2}s^{-1}$. Data from Azzolin et al. (2016).

Fig. 4 Local heat transfer coefficients during condensation of R455A and R452B compared to those of the pure components R32 and R1234yf, at vapor quality x = 0.6. Data from Azzolin et al. (2019).

Figure 4 reports the heat transfer coefficients during condensation of R455A and R452B in the channel compared to those of the pure components R32 and R1234yf. The data of the pure fluids are taken from Matkovic et al. (2009) and Del Col et al. (2010). The heat transfer coefficients of R455A are comparable to those of R1234yf, but significantly lower than those of R32 and R452B. The degradation of the heat transfer coefficient of zeotropic mixtures, which is more evident for R455A, is related to the additional mass transfer resistance caused by the different volatility of the component of the mixture. At 0.6 vapor quality, R455A and R452B present heat transfer coefficients which are respectively 44% and 27% lower than those of R32 at G = 800 kg m⁻² s⁻¹. It is interesting to notice that increasing vapor quality causes higher degradation of the condensation heat transfer coefficient of the mixtures compared to the pure components.

SUMMARY

In this work the thermal performance during condensation and vaporization inside a 0.96 mm diameter channel of zeotropic mixtures has been presented. During condensation and vaporization tests a degradation of the heat transfer coefficient due to the additional mass transfer resistance can be seen. In the case of a binary mixture, the penalization can be estimated by comparing the measured heat transfer coefficients versus an ideal linear behavior between the values pertaining to the pure components at the same operating conditions. The heat transfer degradation during flow boiling of the mixture R32/R1234ze(E) in the tested conditions is shown to be about 50%.

Condensation data for the ternary mixtures R452B and R455A have been presented. R452B presents higher heat transfer coefficients compared to R455A due to the higher liquid thermal conductivity and the lower temperature glide. The heat transfer coefficients of R455A are comparable to those of R1234yf, but lower than those of R32 and R452B. The heat transfer degradation depends on the process (condensation or flow boiling) but also on the characteristics of the fluids, in particular on the temperature glide.

The condensation and vaporization processes of zeotropic mixtures are not isothermal and continuous changes in the composition of liquid and vapor phases lead to the buildup of mass transfer resistances, therefore models developed for pure fluids need to be corrected. When designing heat transfer equipment for zeotropic mixtures, although the varying saturation temperature may be beneficial to the matching of temperature profiles between the two fluids, it must be reminded that the poorer heat transfer characteristics may require additional heat transfer area.

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