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EXPERIMENTAL DETERMINATION OF THE BOILING HEAT TRANSFER COEFFICIENTS OF R-134a AND R-417A ON A SMOOTH COPPER TUBE

Pardiñas A.A., Fernández-Seara J.*, Piñeiro-Pontevedra C., and Bastos S.
*Author for correspondence
Área de Máquinas y Motores Térmicos,
Universidad de Vigo,
Vigo, 36310,
Spain,
E-mail: jseara@uvigo.es

ABSTRACT

Flooded evaporators are widely used as compact cooling units so as to cool liquids. They consist of a shell-and-tube heat exchanger, with the fluid to cool flowing inside the tube bundle and a refrigerant that evaporates over those tubes. Pool boiling on the external surface of the tubes is a very complex process and, therefore, an experimental analysis is required in order to determine the boiling heat transfer coefficients (HTCs). Copper and copper alloys tubes are commonly employed in such heat exchangers, due to their high thermal conductivity and low cost when compared to other materials. On the other hand, refrigeration and air conditioning sectors are undergoing significant changes caused mainly by the necessity of replacing existing refrigerants with more environmentally-friendly ones. This paper reports the work carried out to determine pool boiling HTCs of two HFCs refrigerants, R-134a and R-417A blend on a smooth copper tube with and outer diameter of 18.87 mm, at two saturation temperatures of 10 ºC and 7 ºC. This work has been done experimentally by means of a test rig specifically designed and prepared for the study of these processes. The experimental setup and data acquisition are described, the experimental procedure is explained, the data reduction methodology is detailed and the results are presented and discussed.

INTRODUCTION

Nowadays, the air conditioning and refrigeration industry is undergoing significant changes as it continues to replace the ozone depleting substances (ODSs) commonly used as refrigerants, such as chlorofluorocarbons (CFCs) and hydrochlorofluorcarbons (HCFCs). Hydrofluorocarbons (HFCs) are synthetic fluids entirely harmless to the ozone layer, since they do not contain chlorine. HFC-134a is the only pure HFC clearly established as a substitute for CFCs and HCFCs in several types of applications at high and medium temperature levels despite its high global warming potential (GWP = 1300, 100 years integrated time horizon). New refrigerant blends, such as R-417A, are becoming very important due to the possibility of using them in R-22 systems with only minor changes (drop-in refrigerants).

Flooded evaporators are widely used to produce chilled water in refrigeration systems with high cooling loads. These devices mainly consist of a shell-and-tube heat exchanger. The shell side is completely filled with liquid refrigerant and heating water circulates inside the tubes. The main advantage of such devices is the complete wetting of the outer surface of the tubes with liquid refrigerant, which brings about high boiling HTCs. Copper and copper alloys tubes are generally used with halogenated refrigerants due to its high thermal conductivity.

Nucleate boiling heat transfer on the outer surface of tubes is a complex mechanism which requires experimental research in order to determine its HTCs. These experimental data become very important to evaluate the evaporators performance in refrigeration systems or as a guideline for the evaporator design in new appliances. Previous work, both theoretical and experimental, has been mainly focused on traditional refrigerants and different smooth and enhanced tubes. Reference [1] states a study which includes 5 different refrigerants on smooth tube, low-finned tubes and enhanced tubes. References [2,3] study the performance of R-114 (with and without oil) on smooth and enhanced tubes as a single tube and as a tube bundle. Reference [4] broadens the refrigerant-surface database stating the HTCs obtained with R-134a and different material-coated tubes. Reference [5] studies boiling HTCs on smooth tube using 3 refrigerants and two saturation conditions. References [6,7] analyze experimentally nucleate boiling of 8 pure halogenated refrigerants on smooth tube and three enhanced tubes. Reference [8] includes the study of 5 refrigerants on smooth tubes made of different materials and several roughnesses. Reference [9] studies nucleate boiling with R-134a and 4 enhanced surfaces as well. Several authors have stated nucleate boiling heat transfer experimental correlations on smooth tube and for pure refrigerants. Those from Rohsenow [10], Gorenflo [11], Cooper [12] or Stephan-Abdelsalam [13] can be pointed out. According to [14] and from the design point of view, Gorenflo [11] and Cooper [12] are recommended, despite the fact that evaporators with smooth tubes and HCFCs and HFCs are not currently of industrial application.

Nucleate boiling with refrigerant blends differs significantly from that using pure substances, since two new effects appear: mass transfer and mixture physical properties estimation. References [15,16] state that boiling HTCs decrease when using refrigerant blends compared to those employing pure refrigerants. Moreover, reference [17] notes that this decrease is higher for enhanced tubes than for smooth tubes. Reference [18] presents an analytical prediction in order to quantify the effect of using mixtures on HTCs which relates the convection coefficient of the refrigerant blend to that if the refrigerant were pure.

The main goal of this work has been to determine by means of experimentation the pool boiling heat transfer coefficients of R-134a and R-417A on smooth copper tubes of 18.87 mm of outer diameter and 1.5 m long, at saturation temperatures of 7 and 10 ºC. A description of the experimental facility is included, as well as the experimental procedure and data reduction. Results for both refrigerants are compared to experimental correlations, adapted and not depending on the refrigerant.

NOMENCLATURE

A	[m^2]	Area
β	[m/s]	Mass transfer coefficient
gtd	[K]	Glide
h	[W/m^{2}K]	Convection heat transfer coefficient
i	[J/kg]	Specific enthalpy
i_{lv}	[J/kg]	Specific latent heat
$\dot{m}$	[kg/s]	Mass flow
q	[W]	Heat flow
q/A	[W/m^2]	Heat flux
R	[K/W]	Thermal resistance
ρ	[kg/m^3]	Density
T	[K]	Temperature
ΔT_{LMTD}	[K]	Logarithmic mean temperature difference

Dimensionless numbers

B	[-]	Scaling factor
f	[-]	Friction factor
Nu	[-]	Nusselt number
Pr	[-]	Prandtl number
Re	[-]	Reynolds number

Subscripts

a	Adapted
cw	Cooling water
hw	Heating water
i	Inner
id	Ideal
inl	Inlet
l	Liquid
o	Outer
out	Outlet
ov	Overall
w	Wall

EXPERIMENTAL FACILITY

The layout of the experimental facility is shown in Figure 1. The facility consists of a boiling test section composed by a shell-and-tube evaporator connected to a heating water loop and a shell-and-tube condenser connected to a cooling water loop. The condenser and the evaporator consist of a 6 mm thick horizontal cylindrical body and blind flanges made of stainless steel (AISI-316L). The condenser has an external diameter of 168.3 mm and a total length of 1895 mm. The evaporator has an external diameter of 200 mm and a total length of 1530 mm. Both extremes of the tubes installed at the evaporator and at the condenser protrude from the tube sheets. A single tube placed at the lower part of the evaporator was used in the experiments, while three tubes were installed in a horizontal row at the middle of condenser. Both, the condenser and the evaporator, are equipped with six sight glasses for lighting up and observing condensation and boiling processes.

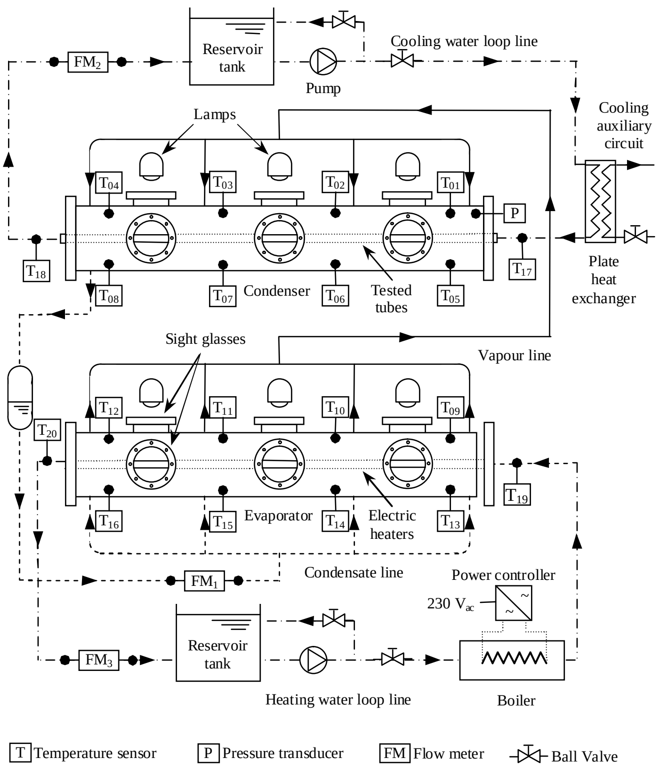

Figure 1 Experimental facility layout

The evaporator was filled with liquid refrigerant until the tube to be tested was flooded. The tested tube was internally heated by circulating hot water through it, as shown in Figure 1. A centrifugal pump circulates the heating water from a hot water reservoir tank through the tested tube and an electric boiler in a closed loop, so that the heat removed in the test section was restored in the boiler (heating water loop in Figure 1). A recirculation loop from the pump outlet to the hot water reservoir tank is used to control the water flow rate through the tested tubes. The hot water temperature at the test section is controlled by adjusting the heating power in the boiler using an electric power regulator and a PID controller.

The generated vapour is conducted through a vapour line from the upper part of the evaporator to the upper part of the condenser. The vapour is condensed on the tubes placed at the condenser by circulating cold water through them. A centrifugal pump forces the cooling water from a cold water reservoir tank through a plate heat exchanger and the tubes in a closed loop, so that the heat absorbed in the condenser is removed from the plate heat exchanger by means of a cooling auxiliary circuit filled with propylene glycol (25%). An electric control valve, controlled by a PID, adjusts the propylene glycol flow rate through the plate heat exchanger and allows the control of the cooling water temperature at the condenser. The condensate is split into four streams and returned by gravity to the bottom of the evaporator at four different points in order to guarantee an appropriate distribution of the liquid at the boiling test section (see Figure 1).

The experimental setup was equipped with a data acquisition system based on a 16-bits data acquisition card and a PC. The features of the measuring devices are shown in Table 1. The vapour temperature in the condenser is measured by means of eight sensors (sensors T01 – T08 in Figure 1). The vapour and liquid temperature in the evaporator are measured by using eight sensors, T09 – T12 for the vapour and T13 – T16 for the liquid, according to Figure 1. These sensors are A Pt100 inserted into 100 mm long and 3 mm diameter stainless steel pockets. The pressure in the condenser is measured by using a pressure transducer, denoted by P in Figure 1. The temperature, density and mass flow rate of the condensate returned to the evaporator are measured by means of a Coriolis flow-meter (FM1 in Figure 1). Two temperature sensors, T17 and T18, are employed to measure the inlet and outlet temperature of the cooling water at the condenser. Moreover, T19 and T20 are used to acquire the heating water temperatures at the evaporator. All of these temperature sensors are A Pt100 inserted into 50 mm long and 3 mm diameter stainless steel pockets. The cooling and heating water flow rates are measured by means of two electromagnetic flow-meters, FM2 and FM3, respectively. The calibration test of the data acquisition system shows that the average uncertainty in temperature measurement is within ±0.05 ºC of the operating range.

Table 1 Measuring device

Device	Type	Measuring range	Accuracy
Temp. sensors	RTD Pt100 A	0 / 100 ºC	0.15±0.002·T ºC
Pres. sensors	Danfoss MBS-5150	0 / 2600 kPa	±0.5%
Vol. flow meters	Electromagnetic flow meter Danfoss MAG3100/MAG6000	0 / 3500 m3/h	±0.25%
Mass flow rate density meter	Coriolis mass flow meter Micro Motion F-0.25/2700R	0 / 200 kg/h 600 / 1200 kg/m^3	±0.25% ±2 kg/m^3

In the present study a smooth copper tube was tested. The tested tube has 16.75/18.87 mm inner/outer diameters and an effective heat transfer length of 1.5 m. Before each experiment, special attention was paid in order to guarantee an effective removal of any non-condensable (i.e. air) from the condenser and evaporator sections.

EXPERIMENTAL PROCEDURE AND DATA REDUCTION

Experimental procedure

Experiments were conducted by varying the mean heating water temperature into the tested tube while keeping constant the refrigerant liquid saturation temperature in the evaporator test section and the heating water flow rate inside the tubes. The liquid temperature in the evaporator was fixed by regulating the heat delivered into the boiler through the electric power controller (see Figure 1). The heating water flow rate inside the tested tube was kept as high as possible (water velocity around 3.6 m/s and Reynolds numbers between 54000 and 64000) in order to increase the inner HTCs and to reduce the heating water temperature variation along the test section. Once the system was stabilized with the desired liquid saturation temperature and mean heating water temperature, the experimental data was recorded.

Experiments were carried out for liquid saturation temperatures of 7 ºC and 10 ºC and logarithmic mean temperature differences at the test section from 4 to 8 ºC for R-134a and from 4 to 10 ºC for R-417A, providing heat fluxes from 8000 to 29600 for W/m^2 for R-134a and from 4900 to 33800 W/m^2 for R-417A, approximately.

Data reduction

The boiling HTCs on the smooth copper tube were obtained from the data measured in the experimental facility. The refrigerant and water properties were obtained using the REFPROP Database [19]. A detailed uncertainty analysis was carried out and the results are given hereafter where convenient. The uncertainty calculation procedure was based on the application of the general uncertainty propagation expression, according to the ISO Guide [20] to the calculated magnitudes.

The heat transfer rate was determined from energy balances on the heating water (q_{hw}) at the evaporator tubes and on the cooling water (q_{cw}) at the condenser tubes. The heating water energy balance provides equation (1), where $\dot{m}_{hw}$ stands for the heating water mass flow rate and $i_{hw,i}$ and $i_{hw,o}$ for the specific enthalpies at the entrance and exit of the evaporator tubes. The energy balance at the condenser was evaluated from the cooling water mass flow rate ($\dot{m}_{cw}$) and the inlet and outlet specific enthalpies at the condenser tubes ($i_{cw,inl}$ and $i_{cw,out}$), according to equation (2). The energy balance on the refrigerant was not considered in the analysis due to the formation of vapour bubbles in the condensate line during the current experiments, which provoked unstable measurements of the condensate flow rate returned to the evaporator.

$$q_{hw} = \dot{m}_{hw} \cdot \left(i_{hw,inl} - i_{hw,out}\right) \quad (1)$$

$$q_{cw} = \dot{m}_{cw} \cdot \left(i_{cw,inl} - i_{cw,out}\right) \quad (2)$$

Figure 2 compares the heat transfer rates obtained from equations (1) and (2). These results show considerable differences between the energy balance at the evaporator and condenser. Results of the uncertainty analysis on both energy balances are also included in Figure 2. These results revealed that the uncertainties in the determination of the heat transfer rate from the energy balance at the evaporator were higher than from the energy balance at the condenser due to the high water mass flow rates used inside the evaporator tubes during the experiments, which led to extremely low inlet-outlet water temperature differences (from 0.02 to 0.8 ºC). The mean typical uncertainties in the calculation of q from the energy balances at the evaporator and at the condenser were ±40.52% and ±3.47%, respectively. Taking into account these results, the heat transfer rates considered in the calculation procedure were those obtained from equation (2).

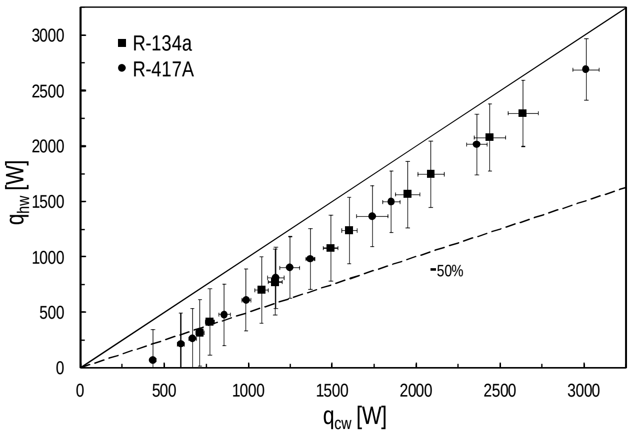

Figure 2 Comparison of the heat transfer from the energy balances at the condenser and evaporator

The overall thermal resistance (R_{ov}) was obtained from equation (3), where the logarithmic mean temperature difference (ΔT_{LMTD}) was determined as a function of the boiling temperature (T_l), according to equation (4).

$$R_{ov} = \frac{\Delta T_{LMTD}}{q} \tag{3}$$

$$\Delta T_{LMTD} = \frac{(T_l - T_{hw,inl}) - (T_l - T_{hw,out})}{\ln\left(\dfrac{T_l - T_{hw,inl}}{T_l - T_{hw,out}}\right)} \tag{4}$$

In this work, the fouling resistances were neglected since the tubes were cleaned prior to each experiment. Consequently, the overall thermal resistance was expressed as the sum of the partial thermal resistances corresponding to the inner convection, the conduction through the tube wall (R_w) and the outer convection, according to equation (5). In this equation A_i and A_o are the inner and outer areas of the tube and h_i and h_o stands for the inner convection and outer boiling HTCs.

$$R_{ov} = \frac{1}{A_i \cdot h_i} + R_w + \frac{1}{A_o \cdot h_o} \tag{5}$$

The water side convection coefficient was calculated from the Petukhov (1970) correlation (equation 6), where the friction factor (f) was obtained according to equation (7), proposed by the same author for plain tubes. The inner heat transfer coefficients were also determined experimentally by means of a modified Wilson plot technique similar to that proposed by [22] at the condensation section and the friction factors were experimentally evaluated in a 6 m long tube. The experimental results agree with those provided by equations (6) and (7) within ±3% and ±0.5%, respectively. In equations (6) and (7), the Nusselt number (Nu_i) and the heating water Reynolds (Re_{hw}) and Prandtl (Pr_{hw}) numbers were evaluated at the bulk temperature.

$$Nu_i = \frac{(f/8) \cdot Re_{hw} \cdot Pr_{hw}}{1.07 + 12.7 \cdot (f/8)^{1/2} \cdot \left(Pr_{hw}^{2/3} - 1\right)} \tag{6}$$

$$f = (0.79 \cdot ln(Re_{hw}) - 1.68)^{-2} \tag{7}$$

Finally, equation (8) allowed the determination of the pool boiling coefficient, h_o.

$$h_o = \frac{1}{A_o \cdot \left[R_{ov} - \dfrac{1}{A_i \cdot h_i} - R_w\right]} \tag{8}$$

RESULTS AND DISCUSSION

Experimental pool boiling heat transfer coefficients

Experimental pool boiling HTCs (h_o) for R-134a and R-417A over smooth copper tube as a function of the heat flux (q/A_o) for liquid saturation temperatures of 7 and 10 ºC are shown in Figure 3.

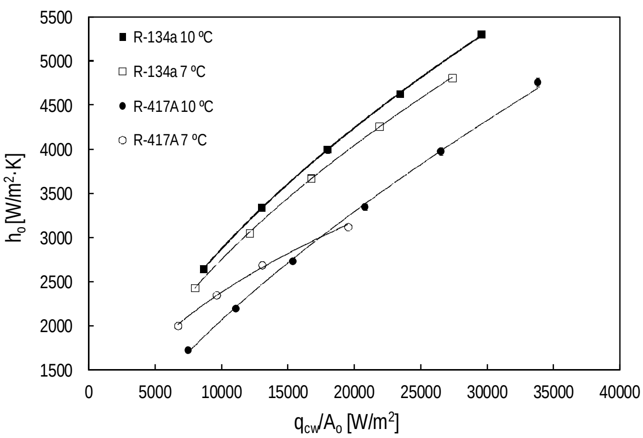

Figure 3 Experimental pool boiling heat transfer coefficients for R-134a and R-417A vs. heat flux at saturation temperatures of 7 ºC and 10 ºC

A clear dependence between heat flux and HTCs can be observed for both refrigerants under these conditions, the coefficients increasing rapidly with heat flux. However, this dependence is not linear between both variables, since the slope of the curves decreases slightly with rising heat flux. Concerning the difference in HTCs between both refrigerants, it can be seen that those obtained for R-134a are higher than those for R-417A, which agrees with several works previously mentioned [15-18]. For R-134a the HTCs obtained range from 2432 W/m^{2}K to 4809 W/m^{2}K (7990 W/m^2 ≤ q/Ao ≤ 27385 W/m^2) when the saturation temperature is 7 ºC and from 2644

W/m^{2}K to 5302 W/m^{2}K (8648 W/m^2 $\leq$ q/Ao $\leq$ 29611 W/m^2) when the saturation temperature is 10 ºC. In contrast, for R-417A the HTCs determined vary between 2002 W/m^{2}K and 3123 W/m^2·K (6722 W/m^2 $\leq$ q/Ao $\leq$ 19529 W/m^2) with a saturation temperature of 7 ºC and between 1352 W/m^{2}K and 4760 W/m^{2}K (4855 W/m^2 $\leq$ q/Ao $\leq$ 33819 W/m^2) with a saturation temperature of 10 ºC. The standard uncertainty bands obtained for the measured HTCs are also included in Figure 3 and range from 0.73% to 1.94%.

The effect of the saturation temperature on the values obtained varies significantly between one refrigerant and another. For R-134a, HTCs increase by rising saturation temperature. However, for the R-417A blend and the lowest heat fluxes tested**,** the effect is the opposite. Both curves meet with increasing heat flux, as can be seen in Figure 3.

Comparison with experimental correlations

Trying to investigate further on experimental results, four well-know experimental correlations for pool boiling of pure refrigerants on smooth tubes have been selected: Rohsenow [10], with tube roughness of 0.5 µm, Gorenflo [11] and a correction of Cooper's correlation [12], proposed by himself, both recommending a 0.4 µm roughness and Stephan-Abdelsalam [13], which assumes a roughness of 1 µm. Therefore, all correlations require the precognition of the surface roughness but, unfortunately, this parameter is usually unknown since it is difficult to measure and can be affected by corrosion, fouling or oxidation, which are processes difficult to quantify. The experimental pool boiling HTCs on the smooth copper tube for R-134a and those obtained from the experimental correlations at the same operating conditions are compared in Figure 4.

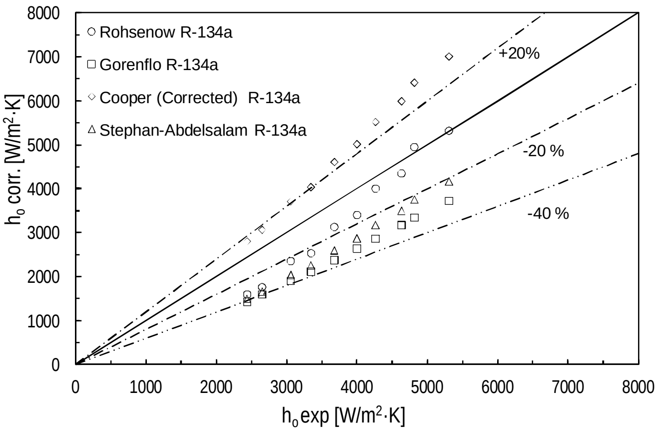

Figure 4 Experimental heat transfer coefficients for pool boiling of R-134a vs. model predictions

The Rohsenow correlation [10] predicts the experimental data within an error band between +3% and -34%, with a mean deviation of -15%. Despite being one of the first general correlations, it provides the most accurate predictions for the boiling HTCs amongst the models considered in the study. The correction of the correlation proposed by Cooper [12] provides predictions with an error band from +16% to +33% and a mean deviation of +25%. The Gorenflo [11] correlation underpredicts the present experimental data over the whole range of heat fluxes considered in the experiments, ranging the error band in the predictions of the boiling HTCs from -30% to 41%, with a mean deviation of -35%. Finally, from the comparison with the correlation proposed by Stephan-Abdelsalam [13] error bands between -21% and -38% and a standard deviation of -29% exist.

Experimental pool boiling heat transfer coefficients for R-417A cannot be directly compared to experimental correlations obtained for pure refrigerants, since mass transfer effects, which exist in pool boiling of refrigerant blends, are not considered. These effects are analytically studied in [18], where equation (9) is stated. By means of this equation, pure refrigerant predictions can be adapted in order to use them with refrigerant blends. In equation (9) $h_{o,a}$ stands for the predicted pool boiling HTC after the adaptation. $h_{o,id}$ stands for the predicted HTC considering the blend as a pure refrigerant. *q/A* represents the heat flux and *gtd* the glide of the refrigerant blend. Moreover, *B* stands for an scaling factor, assumed by 1.0 in [18], ρ_l for the refrigerant blend liquid density, i_{lv} for the refrigerant blend latent heat and β_l for the mass transfer coefficient in the liquid phase and assumed as 0.0003 m/s.

$$h_{o,a} \big/ h_{o,id} = \left[1 + \left(h_{o,id} \big/ q/A \right) \cdot gtd \cdot \left[1 - \exp\left(\frac{-B \cdot q/A}{\rho_l \cdot i_{lv} \cdot \beta_l} \right) \right] \right]^{-1} \quad (9)$$

The predictions adapted by means of equation (9) are exactly those previously included in the comparison study for R-134a and the result of the comparison among experimental HTCs and predicted values is shown in Figure 5. The adapted Rohsenow [10] correlation predicts the experimental data within an error band between +12% and -48%, with a mean deviation of -7%. Once again, it provides the most accurate predictions for the boiling HTCs among the chosen models. The adaptation of the correlation by Cooper [12] (corrected by the author) provides predictions with an error band from -5% to -45% and a mean deviation of -19%. The Gorenflo [11] correlation adapted by means of equation (9) predicts the experimental data with an error band that ranges from -30% to -57%, with a mean deviation of -38%. Finally, from the comparison with the Stephan-Abdelsalam [13] adapted correlation, error bands between -21% and -38% and a standard deviation of -29% exist.

It is interesting to observe that each correlation follows a general trend except for three specific conditions (an example is indicated in Figure 5), which correspond to the HTCs achieved with saturation temperature of 7 ºC and the lowest heat fluxes. At these conditions, as mentioned before, the experimental HTCs obtained for R-417A are higher than those achieved in the same heat flux range but with a saturation temperature of 10 ºC, which is a behaviour different to the expected for pool boiling and pure refrigerants. Therefore, the predicted HTCs obtained by means of correlations, despite the fact that these correlations have been adapted using equation (9), underestimate the experimental HTCs.

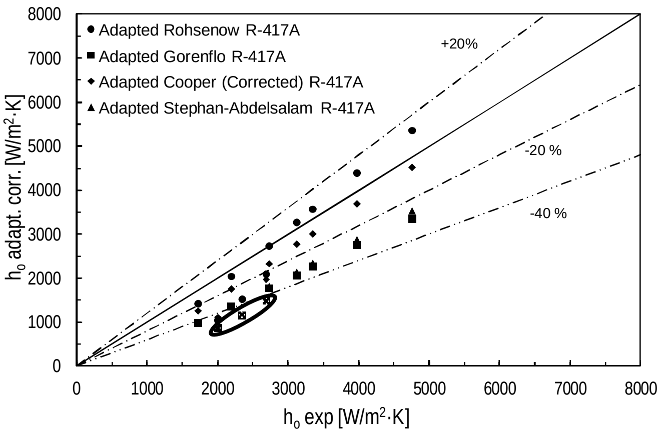

Figure 5 Experimental heat transfer coefficients for pool boiling of R-417A vs. adapted experimental correlations

From these comparisons it is concluded that the discrepancies are inside the actual differences between models. In addition, pool boiling is a complex process, very difficult to analyze, and becomes even more complicated when studying refrigerant blends. It is worthwhile pointing out again that the predicted boiling HTCs by the models have a strong dependence on the surface roughness of the tube, which was not experimentally measured. Therefore, the agreement between the calculated and experimental values of the boiling HTCs could be affected by the poor definition of the surface roughness of the tube, as stated in [8].

CONCLUSIONS

Experimental research was carried out to investigate the pool boiling heat transfer of R-134a and R-417A on a horizontal smooth copper tube of 18.87 mm outer diameter and 1.5 m long. Experiments were carried out for liquid saturation temperatures of 7 ºC and 10 ºC and logarithmic mean temperature differences at the test section from 4 to 8 ºC for R-134a and from 4 to 10 ºC for R-417A, providing heat fluxes from 8000 to 29600 for W/m^2 for R-134a and from 4900 to 33800 W/m^2 for R-417A, approximately. The experimental pool boiling HTCs were compared against the results provided by four well-known experimental correlations found in the literature for nucleate pool boiling. Based upon the experimental and numerical results, the following conclusions were drawn.

1. Experimental pool boiling HTCs are strongly influenced by heat flux, increasing rapidly with this variable.
2. The effect of saturation temperature is that expected for pool boiling of R-134a, increasing the HTCs with this temperature. When using R-417A this effect is not so clear and can be the opposite.
3. Pool boiling HTCs are higher for R-134a than for R-417A in the whole heat flux range studied.
4. From the comparison between experimental HTCs and predictions, discrepancies can be observed.
5. The experimental pool boiling HTCs of R-134a are best predicted by the experimental correlations developed by Rohsenow [10] and the correction of Cooper [12], with mean deviations of -15% and +25%, respectively.
6. When considering R-417A, the most accurate adapted experimental correlations, from those selected, with the pool boiling HTCs obtained, are once again that developed by Rohsenow [10] and the correction of Cooper [12], with mean deviations of -7% and -19%, respectively.

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