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Convective boiling of pure and mixed refrigerants: An experimental study of the major parameters affecting heat transfer

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Abstract

An experimental study is carried out to investigate the characteristics of the evaporation heat transfer for different fluids. Namely, pure refrigerants fluids (R22 and R134a), azeotropic and quasi-azeotropic mixtures (R404A, R410A, R507) and zeotropic mixtures (R407C and R417A).

The test section is a smooth, horizontal, stainless steel tube (6 mm ID, 6 m length) uniformly heated by the Joule effect. The flow boiling characteristics of the refrigerant fluids are evaluated in 250 different operating conditions. Thus, a data-base of more than 2000 data points is produced.

The experimental tests are carried out varying: (i) the refrigerant mass fluxes within the range 200–1100 kg/m^2 s; (ii) the heat fluxes within the range 3.50–47.0 kW/m^2; (iii) the evaporating pressures within the range 3.00–12.0 bar.

In this study, the effect on measured heat transfer coefficient of vapour quality, mass flux, saturation temperature, imposed heat flux, thermo-physical properties are examined in detail.

Keywords: Flow boiling; R22; R407C; R410A; R404A; R507; R417A; R134a; Smooth tube

1. Introduction

In the present study, the evaporating heat transfer characteristics of the refrigerant fluid alternative to R22 are analyzed. R22 is a HCFC, widely used in refrigerant and air conditioning plants because of its excellent thermal properties. It is well known that in a short period of time, the partially halogenated HCFCs will be phase out, since the chlorine they contain can combine with ozone in stratosphere, thus depleting the ozone layer. Therefore, the research for a replacement of R22 has been intensified in recent years.

HFCs are a new family of substances which might substitute HCFCs. Indeed, they are harmless towards the ozone layer, because they do not contain chlorine.

In order to use these new refrigerants properly, their thermodynamic, thermo-physical, transport and heat transfer properties must be known. Specifically, the detailed heat transfer characteristics in condensation and boiling, together with the associated frictional pressure drops, are very important in order to avoid both under- and over-design of evaporators, boilers and other two-phase process equipment. The accurate prediction of heat transfer characteristics in condensers and evaporators is therefore a prerequisite to increase the cycle performance through the correct sizing of each plant component.

The importance of correctly predicting saturated flow boiling heat transfer coefficients has been recognized, as seen from a large number of analytical and experimental investigations conducted in the past years [1]. There are a large number of correlations available in literature to predict heat transfer in flow boiling [2–5]. These correlations can be classified in two categories. In the first category,

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Nomenclature

A	heat transfer area (m^2)
Bo	$\frac{q}{G\Delta h_{ev}}$ Boiling number
C_p	specific heat (kJ/kg K)
D	tube diameter (m)
d_b	bubble diameter (m)
g	gravitational acceleration (m/s^2)
G	mass flux (kg/s m^2)
h	heat transfer coefficient (W/m^2 K)
p	pressure (bar)
p_r	reduced pressure
k	thermal conductivity (W/m K)
n	power exponent
nq	power exponent
q	heat flux (W/m^2)
Q	heat transfer rate (W)
T	temperature (°C, K)
u	velocity (m/s)
x	vapour quality

Greek symbols

β	liquid contact angle
δ	thickness of annular liquid film (m)
Δh_{ev}	latent heat (J/kg)
ΔT	temperature glide (°C, K)
ε	void fraction
μ	dynamic viscosity (Pa s)
ρ	density (kg/m^3)
σ	surface tension (N/m)
τ	time characteristics (s)

Subscripts

b	bubble
cb	convective boiling
ev	evaporative
i	inner
l	liquid phase
nb	nucleate boiling
sat	saturation
v	vapour phase
w	inner wall

the correlations are developed by experimental investigators to represent their own data with a limited data base of operating conditions. These correlations can be used only in the same range of operating conditions and with the same boiling fluids.

In the second category, the correlations are developed on the basis of large data sets from different sources with a wide range of operating conditions and with different fluids. These correlations show a wide range of applicability [6].

Even in the presence of a large body of experimental data, there are still some remarkable insufficiencies. First, few correlations are developed with new refrigerant fluids, especially for R410A, R404A, R407C and R417A. Second, different functional forms have been adopted to account for the dependence of heat transfer coefficients on the relevant operating parameters. In many cases, the correlations are not able to predict correctly the dependence of the heat transfer from such crucial parameters as evaporating pressure and fluid properties. In particular, in flow boiling, simple correlation functions cannot fully replicate the complex relationship among the variables affecting the contribution of nucleate boiling and/or convective boiling to the heat transfer coefficient.

In literature, many studies deal with convective boiling of pure and mixed HFC fluids. Heat transfer coefficients of R134a have been widely investigated with different kinds of tube surfaces: smooth, micro fin, with perforated strip-type inserts [7–14]. The refrigerant mixture R410A has been investigated during convective boiling in internally grooved horizontal tubes [15], in smooth/micro-fin tubes [16], in horizontal annular finned ducts [17]. The zeotropic mixture R407C has been tested in horizontal smooth tubes [18,19], in horizontal small tubes [20], in enhanced surface tubing [21], and in micro-fin and smooth tubes [22,23]. Different pure and mixed refrigerant fluids have been tested in convective boiling in horizontal smooth tubes [24,25].

An experimental plant was set up at the University of Naples for evaluating the heat transfer characteristics of new refrigerant fluids in convective boiling. In the present paper, the local heat transfer coefficients are evaluated for pure refrigerant fluids R22 and R134a, for azeotropic and quasi-azeotropic mixtures R507, R404A and R410A, for zeotropic mixtures R407C and R417A.

R507, R404A and R410A are azeotropic and near azeotropic mixtures of R125/R143a (50/50% by weight), R125/R143a/R134a (44/52/4% by weight), R32/R125 (50/50% by weight). R407C and R417A are zeotropic mixture of R32/R125/R134a (23/25/52% by weight) and of R125/R134a/R600 (46.6/50/3.40% by weight). R407C has a temperature glide of about 6 °C, whereas R417A has a temperature glide of about 4 °C. The flow boiling characteristics of the above-mentioned refrigerant fluids are evaluated in 250 different operating conditions with a data-base of more than 2000 data points.

The experimental tests are carried out by varying refrigerant mass fluxes within the range from 200 to 1100 kg/m^2 s; heat fluxes within the range from 3.50 to 47.0 kW/m^2; evaporating pressures within the range from 3.00 to 12.0 bar.

The objective of the present experimental study are: (i) to develop an accurate flow boiling heat transfer database for several important new fluids and to provide data to the refrigeration industry for the design of high efficiency evaporators, (ii) to investigate in depth the influence of vapour quality, evaporating pressure, heat flux, refrigerant mass flux, refrigerant fluid thermo-physical properties on the flow boiling characteristics.

2. Experimental apparatus

A schematic view of the experimental apparatus is reported in Fig. 1. Indications on the location of the sensors are provided, as well.

The experimental apparatus consists of a main loop and two auxiliary ones.

A detailed description of main loop, of the test procedures and of data reduction is reported in [26].

In order to investigate a wide range of operating condition, the auxiliary circuit of the plant described in [26] was modified by introducing two auxiliary vapour compression plants, operating alternatively.

Along the main loop, a gear pump drives the liquid refrigerant coming from the boiler and by means of a hydrodynamic speed variator allows the refrigerant to flow at different flow rates.

The refrigerant mass flow rate is measured by a Coriolis effect mass flow meter.

The test section is a 6.0 m stainless steel horizontal tube with inside diameter 6.0 mm and wall thickness 1.0 mm, in which the refrigerant evaporates. The tube is electrically heated by Joule effect. A direct electrical current supplied by a feed current device circulates on the surface of the tube. The heat flux is changed by varying the direct electrical current. The heat flux supplied along the tube can be safely assumed to be constant, along the axis.

In order to calculate the heat transfer coefficient, the heat flux supplied to the refrigerant in the test section, as well as the temperatures of both the refrigerant and the tube wall are measured.

The heat transferred to the refrigerant at the test section can be calculated, since the values of the voltage supplied to the test section and of the current thus generated are both known. Voltage and current are monitored with a voltmeter and an amperometer, respectively.

The outer-wall temperature of the heated tube is measured with 32, four-wire, 100 Ω platinum resistance thermometers mounted at eight measuring points. At each

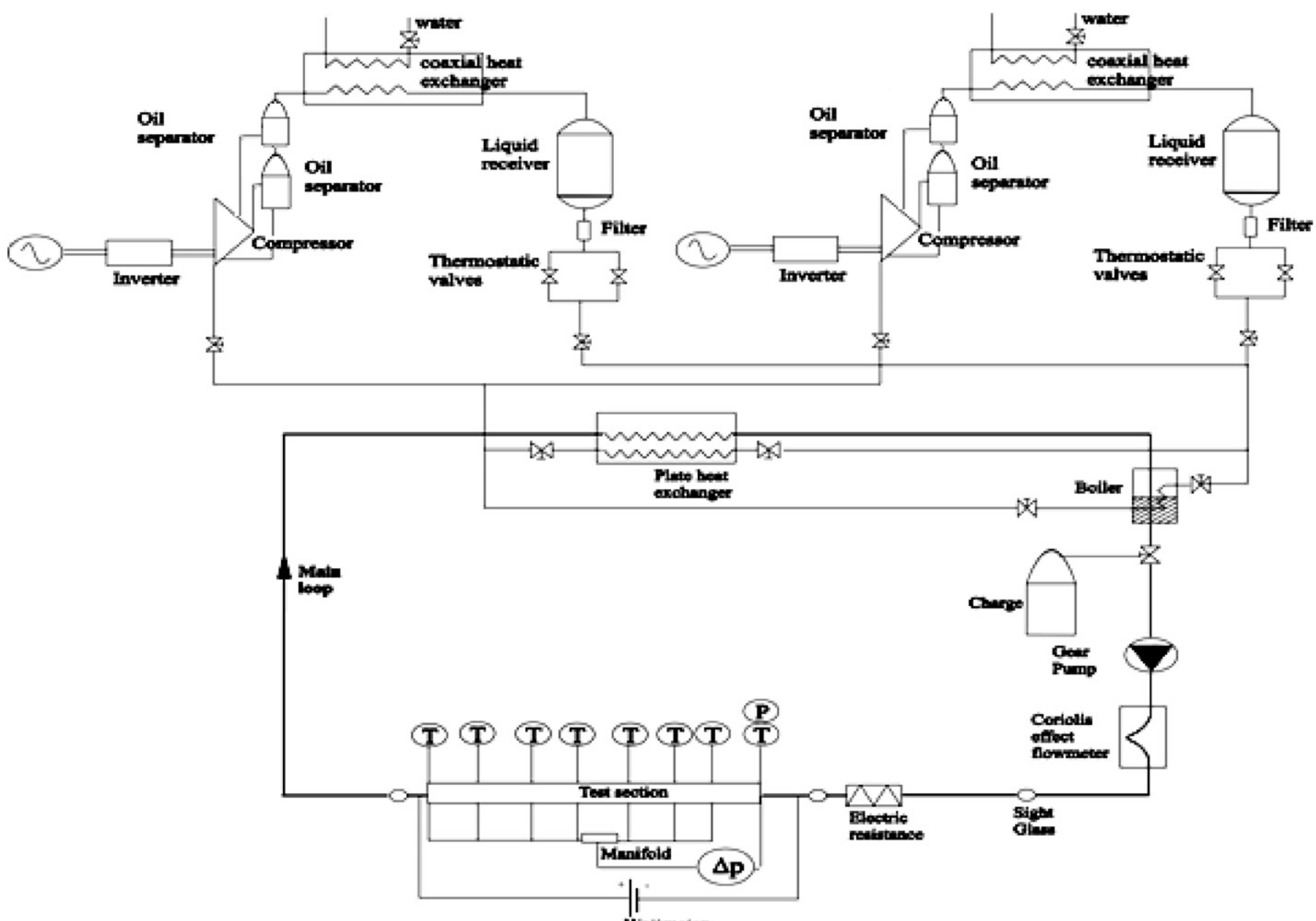

Fig. 1. The experimental apparatus.

Table 1
Measurement equipment

Variable	Device	Accuracy	Range
Temperature	Resistance thermometers Pt100	±0.03 °C	−50 to 100 °C
Pressure	Piezoelectric	0.1% F.S.	0–14 bar
Differential pressure	Piezoelectric	0.1% F.S.	0–1 bar
Mass flow rate	Coriolis effect	±0.2%	0–2 kg/min
Voltage	Voltmeter	±0.2%	0–30 V
Direct current	Amperometer	±0.2%	0–220 A

measuring point four resistance thermometers are fixed on the top, the bottom, and left and right sides of the tube, respectively.

Two four-wire, 100 Ω platinum resistance thermometers are inserted in the refrigerant flow at the inlet and the outlet of the test section.

A piezoelectric pressure transducer measures the absolute pressure at the inlet of the test section. Piezoelectric pressure difference gauges measures the pressure drop between the inlet pressure and the pressure in each measuring point of the test section.

Table 1 summarizes all the relevant characteristics of the plant instrumentation.

Plant insulation is provided by a 32 mm layer of cellular insulate at the heat exchangers and with a coaxial tube of armaflex insulate for tubes and tube fittings.

The refrigerant condenses in a plate heat exchanger that acts as the evaporator of two different vapour compression plants working alternatively. In both, R507 is the working fluid. The experimental apparatus has to be able to carry out tests in which it is possible to control independently the evaporating temperature, the mass flow rate of the circuiting refrigerant and the heat flux in the test section. To this aim, the components of the two auxiliary loops are sized differently, in order to work at a high range and at low range of evaporating temperatures, respectively. The compressor of each loop is connected to an inverter, which enables to vary the revolution speed. Each loop has two thermostatic expansion valves, in order to work under different operating conditions. To avoid that the lubricant oil might flows from one auxiliary loop to the other one, two oil separators in sequence have been inserted in each auxiliary loop.

3. Data reduction

The local heat transfer coefficient is defined as

$$h_{ev} = \frac{q}{(T_w - T_{sat})}, \tag{1}$$

where q is the inner wall heat flux based on the inside surface area of the tube, calculated on the basis of the total heat input. T_w is the local inner wall temperature and is calculated from the measured outside wall temperature by applying the one-dimensional, radial, steady-state heat conduction equation for a hollow cylinder. T_{sat} is the bulk saturation temperature. It is calculated rather than measured directly.

The saturation pressure at each wall temperature measurement position is estimated from the measured inlet pressure of the test section and the pressure drop measured in the subsection. For a zeotropic mixture the saturation temperature is a function of quality as well as of pressure. The quality in each section is evaluated by an energy balance from the inlet to the wall temperature measurement position. The measured temperature and pressure of the liquid at the inlet of the test section are used to determine the test section inlet enthalpy.

The data analysis of the thermo-physical properties of pure and mixed fluids is carried out by using the software REFPROP 7.0 [27].

Uncertainties in the experimental data were calculated using the analysis suggested by Moffat [28]. In the present study, the data were classified as single sample. The uncertainty of each measured quantity consisted mainly of the uncertainty of the measurement device, of the data acquisition system, of system–sensor interaction errors, of system disturbance errors. The experimental uncertainties evaluated with the error analysis are of ±1.7%, as regards the heat flux, and of ±1.6% for the refrigerant mass flux.

The error analysis for the heat transfer coefficient was carried out following the equation:

$$\frac{\partial h_{ev}}{h_{ev}} = \sqrt{\left(\frac{\partial Q}{Q}\right)^2 + \left(\frac{\partial A}{A}\right)^2 + \left(\frac{\partial T_w}{(T_w - T_{sat})}\right)^2 + \left(\frac{\partial T_{sat}}{(T_w - T_{sat})}\right)^2} \tag{2}$$

according to the uncertainty analysis. The error depends on the operating conditions and mainly on the accuracy of the wall temperature-difference. Therefore, it depends largely upon the accuracy of the measurements of the local pressure and of the wall temperature, and on the accuracy of the state-equation employed in evaluating the equilibrium temperature used in the computer program. In the operating condition of our tests, the errors in the heat transfer coefficients stemming from these uncertainties should range between 3.5% and 12.4%.

4. Experimental results

As already stated, in this work, the experiments are performed for seven pure and mixed refrigerants (R22, R134a, R404A, R410A, R507, R407C, R417A) varying the evaporating pressure, the heat flux, the refrigerant mass flux. The experimental conditions are summarized in Table 2.

The heat transfer coefficients are evaluated in 250 different operating conditions.

The latest version of the flow pattern map of Kattan–Thome–Favrat [29,30] is used for the evaluation of two-phase flow regime.

From the maps, it can be seen that the flow pattern is intermittent from the inlet section to a quality between 20% and 40%, depending on test conditions. With increas-

Table 2
The operating conditions

Parameters	Range
Refrigerant fluid	R22, R134a, R404A, R507, R410A, R407C, R417A
Evaporatine pressure (bar)	2.90–12.4
Mass flux (kg/m^2 s)	200–1100
Heat flux (kW/m^2)	3.50–47.0

ing vapour content, an annular flow is achieved. At higher quality values (>70%) the liquid film in the upper portion of the tube disappears and the flow becomes annular with a partial dry-out. Only few data points corresponding to very low mass fluxes are in stratified wavy flow regime. Few data points corresponding to very high mass fluxes and evaporating pressures are in mist flow regime.

4.1. Influence of different parameters on heat transfer coefficients

In order to determine the characteristics and the mechanism of convective boiling, the influence of vapour quality, mass flux, heat flux, evaporating pressure, refrigerant fluid thermo-physical properties on heat transfer coefficients is investigated.

4.1.1. Effect of vapour quality on boiling heat transfer

From Figs. 2–4, local boiling heat transfer coefficients are reported for R404A, R507 and R410A to investigate the influence of vapour quality on heat transfer. Similar trends are observed in all the experiments with similar operating conditions. In Fig. 2 [26], the heat transfer coefficients of R404A are reported in two different operating conditions. Both are achieved with an almost constant refrigerant mass flux of 480 kg/m^2 s. The former corresponds to a heat flux of 12.0 kW/m^2, with an evaporating pressure of 4.30 bar. The latter to a heat flux of 24.4 kW/m^2, with an evaporating pressure of 7.40 bar. In Fig. 3 [31], the heat transfer coefficients of R507 are reported as a function of vapour quality corresponding to a low mass flux of 286 kg/m^2 s, a low evaporating pressure of 4.00 bar and a low heat flux of 11.2 kW/m^2. In Fig. 4, the heat transfer coefficients of R410A are reported with a refrigerant mass flux of 589 kg/m^2 s, a high value of evaporating pressure of 11.2 bar and a high heat flux of 29.2 kW/m^2.

In all the experimental tests performed, a different dependence of the heat transfer coefficients on vapour quality is apparent. Indeed, at low values of the evaporating pressure and heat flux, the heat transfer coefficient increases with increasing vapour quality. On the contrary,

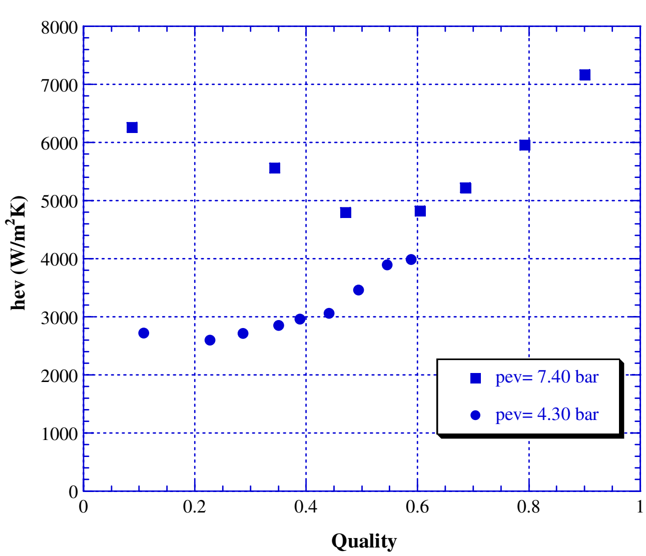

Fig. 2. Heat transfer coefficients of R404A at $G = 483$ kg/m^2 s, $p_{ev} = 7.40$ bar, $q = 24.4$ kW/m^2; and at $G = 474$ kg/m^2 s, $p_{ev} = 4.30$ bar, $q = 12.0$ kW/m^2.

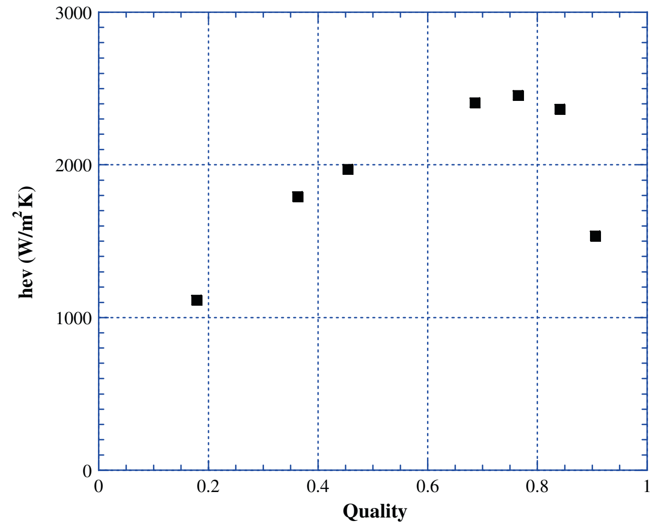

Fig. 3. Heat transfer coefficients of R507 at $G = 286$ kg/m^2 s, $p_{ev} = 3.99$ bar, $q = 11.2$ kW/m^2.

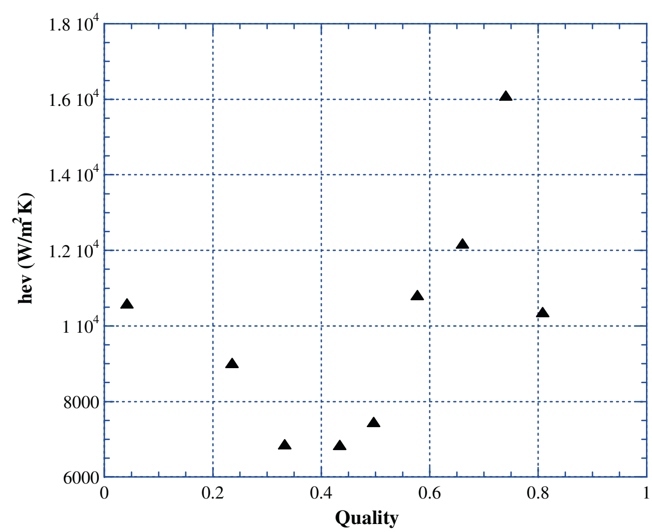

Fig. 4. Heat transfer coefficients of R410A at $G = 589$ kg/m^2 s, $p_{ev} = 11.6$ bar, $q = 29.2$ kW/m^2.

for high pressure and heat flux values, the heat transfer coefficient initially decreases with quality and then increases, presenting a local minimum in the vapour quality range between 20% and 50%. In both cases, the heat transfer coefficients suddenly drop when the liquid film disappears (for high vapour qualities), leaving the tube-wall partially or totally dry.

The heat transfer coefficient in flow boiling results from the interaction between nucleate boiling and liquid convection. At high heat fluxes and evaporating pressures, heat transfer is predominantly influenced by the nucleate boiling mechanism. The convective contribution to heat transfer predominates at low heat fluxes and evaporating pressures.

In the experimental tests where liquid convection is the main mechanism, the heat transfer coefficient increases with quality. Indeed, as the flow proceeds downstream and vaporization takes place, the void fraction increases, thus decreasing the density of the liquid–vapour mixture. As a result, the flow accelerates enhancing convective transport from the heated wall of the tube.

The ensuing increase in heat transfer coefficient proceeds until the liquid film disappears, leaving the tube-wall partially or totally dry. In this region, the heat transfer coefficient decreases because of the low thermal conductivity of the vapour.

In the experimental tests corresponding to higher values of heat flux and of evaporating pressure, there are two distinct heat transfer regions during evaporation. In the first one, occurring at low qualities, nucleate boiling dominates. In this region, heat transfer coefficients decrease as the effect of nucleate boiling diminishes. Indeed, as quality increases in annular flow, the effective wall superheat decreases due to a thinner liquid film (less thermal resistance) and to an enhanced convection caused by high vapour velocity. Thus the number of active nucleation sites decreases until a transition quality is reached. Beyond the transition quality, the effective wall superheat is below the threshold value required for bubble nucleation on the wall. The second region corresponds to convective evaporation. It is characterized by the increase of the heat transfer coefficients with quality. This increase proceeds until the liquid film disappears.

4.1.2. Effect of mass flux on boiling heat transfer

In Figs. 5 and 6 [26], local boiling heat transfer coefficients are reported for R407C and R410A, in order to illustrate the influence of mass flux on heat transfer. The data reported for each refrigerant fluid corresponds to experiments with an almost constant heat flux and evaporating pressure, at different refrigerant mass fluxes.

Similar trends are observed for the other refrigerant fluids.

The data reported in the figures correspond to a low evaporating pressure of about 4.00 bar.

The experimental data clearly show that the heat transfer coefficients increase with increasing the refrigerant mass flux. Indeed, increasing the refrigerant mass flux increase

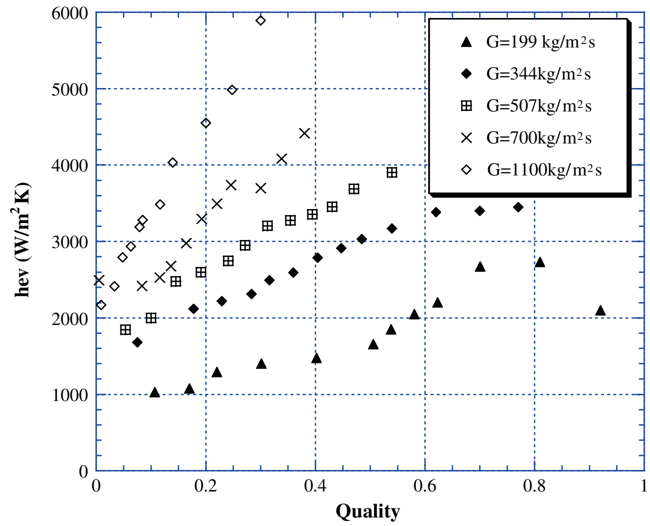

Fig. 5. Heat transfer coefficients of R407C varying the refrigerant mass flux at, $p_{ev} = 3.60$ bar and $q = 7.67$ kW/m^2.

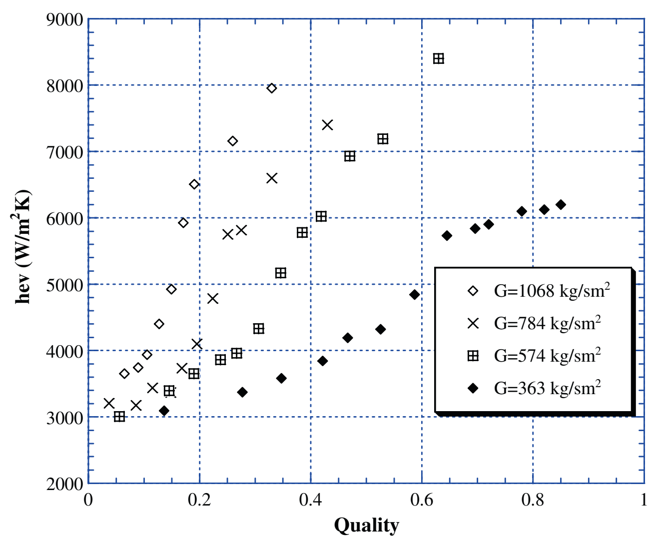

Fig. 6. Heat transfer coefficients of R410A varying the refrigerant mass flux at, $p_{ev} = 4.90$ bar and $q = 14.2$ kW/m^2.

the fluid velocity, thus enhancing convective boiling. In a very low quality region, corresponding to bubbly, plug and slug flow regimes, where nucleate boiling is dominant, the influence of the refrigerant mass flux became weaker and the heat transfer coefficients tend to merge together.

In literature, many correlations are available for the evaluation of the heat transfer coefficient in flow boiling for pure and mixed refrigerants. In most of them, the convective contribution to the heat transfer is proportional to the power 0.80 of the mass flux, because they follow the Dittus–Boelter type correlation. The mass flux has a negligible effect on nucleate boiling. Indeed, by increasing the mass flux, the interfacial shear stress is also increased. The bubbles on the heated surface are removed from the wall under the influence of shear stress and do not attain the larger diameters seen in pool boiling. This effect leads to slight reductions in the nucleate boiling component with increasing shear stress.

In the present study, the influence of mass flux on the heat transfer coefficient in flow boiling was investigated. Fig. 7 reports the exponent n, that describes the dependency of the heat transfer coefficient on the refrigerant mass flux G, as a function of the vapour quality, within the range 200–1100 kg/m^2 s, with an almost constant evaporating pressure of about 4.00 bar. This exponent was obtained as a result of curve fitting of our experimental data. By inspection of Fig. 7, it can be seen to vary within the range 0.52–0.83, except for some data points corresponding to very low vapour quality. The exponent n slightly increases with increasing vapour quality, as a consequence of the progressive increase in the relative relevance of the convective boiling contribution. Indeed, the data points reported in this figure correspond to a low evaporating pressure and therefore the influence of convective boiling is predominant. Increasing the evaporating pressure and the heat fluxes decreases the exponent n because of the increasing relative importance of nucleate boiling.

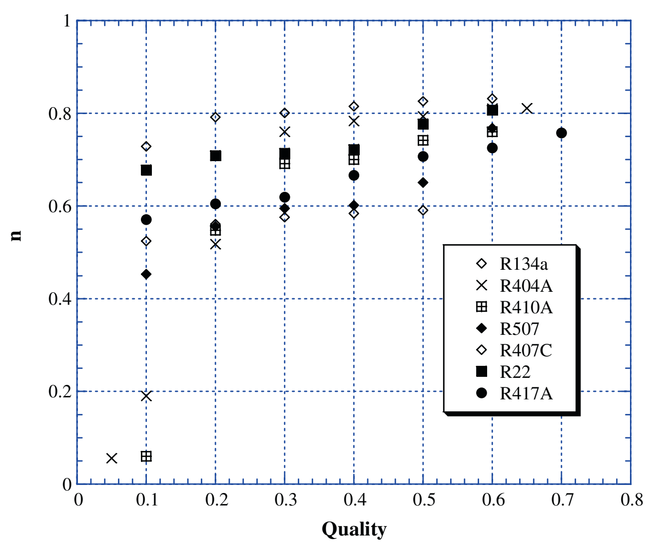

Fig. 7. Values of $n(G)$ power exponent as a function of vapour quality at an almost constant, $p_{ev} = 4.00$ bar and $q = 15.0$ kW/m^2.

In Fig. 8, R134a heat transfer coefficients are evaluated as a function of mass flux by varying the vapour quality at an almost constant evaporating pressure of 3.10 bar and at a heat flux of 14.0 kW/m^2.

In Fig. 9, R404A heat transfer coefficients are evaluated as a function of mass flux by varying the vapour quality at an almost constant evaporating pressure of 4.13 bar and at a heat flux of 11.5 kW/m^2.

In these two figures at any given vapour quality, the dependence of the heat transfer coefficients on mass flux it is clearly shown.

4.1.3. Effect of heat flux on boiling heat transfer

The heat flux has a negligible effect on the convective contribution to heat transfer, while strongly affecting nucleate boiling. Indeed, as heat flux increases, bubble departure frequency rapidly increases, and additional cavities of smaller sizes are activated on the heated surface. As a consequence, the nucleate boiling contribution increases.

Fig. 10 shows the heat transfer coefficients of R407C as a function of vapour quality at an almost constant mass flux of 200 kg/m^2 s and an evaporating pressure of 3.50 bar. The heat flux remains in the range 3.98–7.96 kW/m^2.

Fig. 11 shows the heat transfer coefficients of R407C as a function of vapour quality at an almost constant mass flux of 200 kg/m^2 s and an evaporating pressure of 6.00 bar. The heat flux remains in the range 4.10–7.96 kW/m^2.

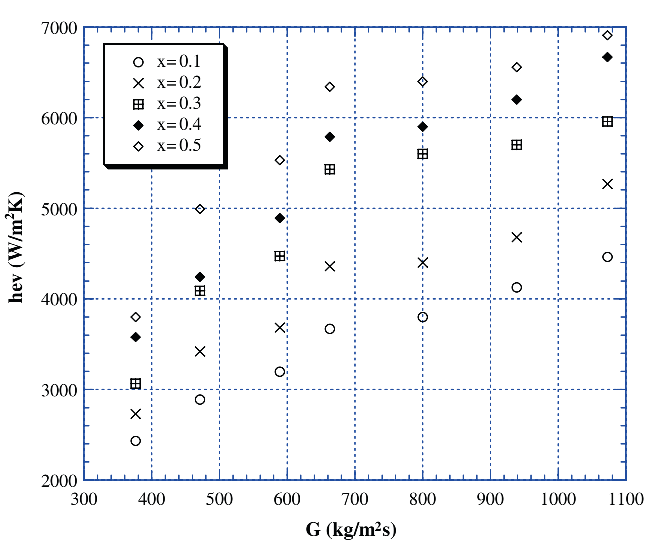

Fig. 8. Heat transfer coefficients of R134a as a function of mass flux varying vapour quality at $p_{ev} = 3.10$ bar and $q = 14.0$ kW/m^2.

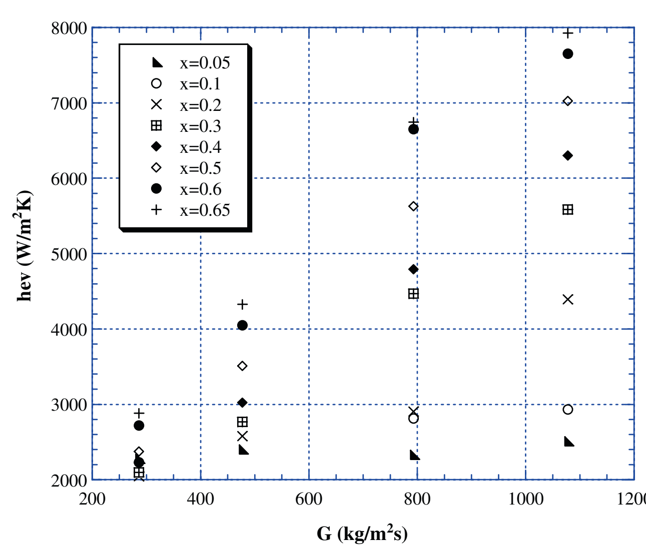

Fig. 9. Heat transfer coefficients of R404A as a function of mass flux varying vapour quality at $p_{ev} = 4.13$ bar and $q = 11.5$ kW/m^2.

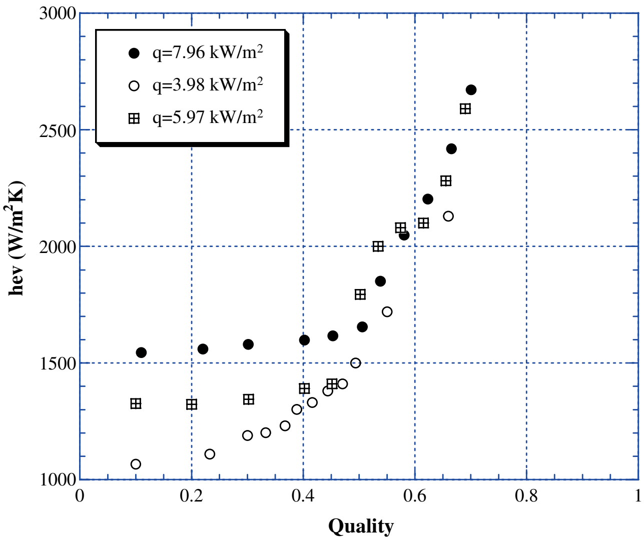

Fig. 10. Heat transfer coefficients of R407C as a function of vapour quality varying heat flux at $p_{ev} = 3.50$ bar and $G = 200$ kg/m^2 s.

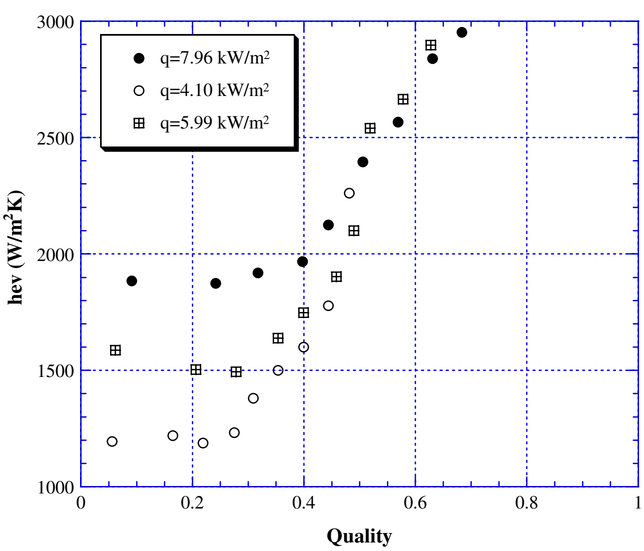

Fig. 11. Heat transfer coefficients of R407C as a function of vapour quality varying heat flux at $p_{ev} = 6.00$ bar and $G = 200$ kg/m^2 s.

From the experimental data, it is clear that the heat transfer coefficients depend on the heat flux, increasing with the latter, only in the region where nucleate boiling is dominant. After a transition quality (in the range 0.40–0.50), in the convective boiling region, the heat transfer coefficients merge together and the heat flux has a negligible effect.

In the nucleate boiling region the dependence of heat transfer coefficient on heat flux can be expressed as

$$h_{ev} \propto (aq^{nq}), \tag{3}$$

where a and nq are constants. The values of nq obtained from many experimental results vary from 0.5 to 0.8. Webb and Pais [32] showed that the slope of h_{nb} vs. q curve is not the same for different refrigerants. Gorenflo [33] found that nq is not a constant but is a function of the reduced pressure, Kandlikar [34] found it to be 0.7, Gungor [35] 0.86, Lazarek [36] 0.714, Klimenko [37] 0.6, Cooper [38] 0.67.

From the best exponential fit of our experimental data in the nucleate boiling region, we found that nq varies in the range 0.53–0.74, for heat flux varying between 4.00 and 25.0 kW/m^2. From the experimental results, it is apparent that nq is not a constant. Indeed, it varies with the evaporating pressure (and therefore with the reduced pressure) in the range 3.00–12.0 bar. As regards the reduced pressure, for the different fluids this interval corresponds to a range of variation between 0.06 and 0.323.

Indeed, the exponent nq increases with increasing reduced pressure in the nucleate boiling region. This is a consequence of the increasing importance of the nucleate boiling increasing with the pressure. The exponent nq has the following expression:

$$nq = 0.80 - (0.30 \times 10^{-0.8pr}). \tag{4}$$

From the reported figures it is apparent that, in the nucleate boiling region, the heat transfer coefficients slightly depend on vapour quality and strongly on heat flux. Beyond a transition quality, the influence of heat flux becomes negligible and the heat transfer coefficient starts to increase strongly with the vapour quality. Based on experimental results, a criterion can be found to characterize the transition between nucleate and convective boiling. The transition quality can be identified from the ratio between the time characteristics of nucleate boiling and that of convective boiling. The latter can be evaluated as the ratio between the thickness of the liquid film and liquid velocity:

$$\tau_{cb} = \frac{\delta}{u_l}. \tag{5}$$

If a simplified annular flow structure is assumed, i.e. an annular liquid ring of uniform thickness $\delta \ll D$ without interfacial waves and without liquid entrainment in the central vapour core, the thickness of the liquid ring is

$$\delta = \frac{D(1-\varepsilon)}{4}, \tag{6}$$

where ε is the cross sectional void fraction of the vapour.

The liquid velocity can be defined as

$$u_l = \frac{G(1-x)}{\rho_l(1-\varepsilon)}. \tag{7}$$

Therefore, the characteristic time of convective boiling is defined as

$$\tau_{cb} = \frac{D(1-\varepsilon)^2\rho_l}{G(1-x)}. \tag{8}$$

The time characteristic of nucleate boiling can be evaluated as the ratio between the bubble diameter and the bubble velocity

$$\tau_{nb} = \frac{d_b}{u_b}. \tag{9}$$

From the analyses of Fritz [39] and Wark [40] the bubble diameter just breaking off from a heated surface was found to be

$$d_b = 0.0146\beta\left(\frac{2\sigma}{g(\rho_l-\rho_v)}\right)^{0.5}. \tag{10}$$

From the Rohsenow [41] analysis, the vapour superficial velocity, i.e. the velocity of a bubble leaving the heated wall, is defined as

$$u_b = \frac{q}{\rho_v \Delta h_{ev}}. \tag{11}$$

Thus, the ratio between the two characteristic times can be estimated as

$$\frac{\tau_{cb}}{\tau_{nb}} = \frac{D(1-\varepsilon)^2\rho_l q}{4G(1-x)\rho_v\Delta h_{ev} d_b} = \frac{Bo(1-\varepsilon)^2}{(1-x)}A \tag{12}$$

The transition quality depends on the constant A, which is a function of tube geometry, of fluid properties and of tube wall characteristics; on the boiling number, on the vapour quality, on the void fraction, that is a function of the vapour quality, as well. Therefore, the boiling number is the appropriate dimensionless number to account for the transition between convective and nucleate boiling. In evaluating the transition quality, the correct tool to be adopted is difficult to determine, because of the different void fraction expressions available, that depends on flow pattern and on tube and channel geometries. In the experimental data base, this transition occurs for an almost constant value of the product

$$\mathrm{Bo}\left(\frac{1-x}{x}\right)^2. \tag{13}$$

4.1.4. Effect of evaporating pressure on boiling heat transfer

Figs. 12–14 show the influence of the evaporating pressure on the heat transfer coefficient.

Fig. 12 refers to R410A, at a fixed refrigerant mass flux of 380 kg/m^2 s, at a heat flux of about 17.0 kW/m^2, by varying the evaporating pressure.

Fig. 13 shows the heat transfer coefficient of R407C at a fixed refrigerant mass flux of 200 kg/m^2 s, at a heat flux of about 8.80 kW/m^2, by varying the evaporating pressure.

Fig. 14 shows the heat transfer coefficient of R417A at a fixed refrigerant mass flux of 200 kg/m^2 s, at a heat flux of about 8.00 kW/m^2, by varying the evaporating pressure.

The experimental results indicate that, at a given mass flux, the heat transfer coefficients increase with increasing pressure. The experimental tests reported correspond to low refrigerant mass fluxes, where the nucleate boiling contribution is always important. With increasing vapour quality, the influence of the evaporation temperature became weaker, along with the increase of the relative importance of the convective contribution to global heat transfer.

With increasing pressure, the contribution of nucleate boiling to the heat transfer coefficients increases, mainly due to the corresponding decrease in the average bubble departure diameter for any given wall superheat. This effect induces the activation of additional new small cavities on the heated surface, thus increasing the number of active nucleation sites. Furthermore, under nucleate boiling, the product of bubble departure frequency and diameter is constant. At increased pressure, bubble frequency increases with decreasing bubble departure diameter, thus contributing to the increase of the heat transfer coefficient.

Assuming spherical nuclei, the bubble departure diameter for a given wall superheat follows from the well-known equilibrium conditions for a small bubble (Thomson equation) and the linearized Clausius–Clapeyron equation and

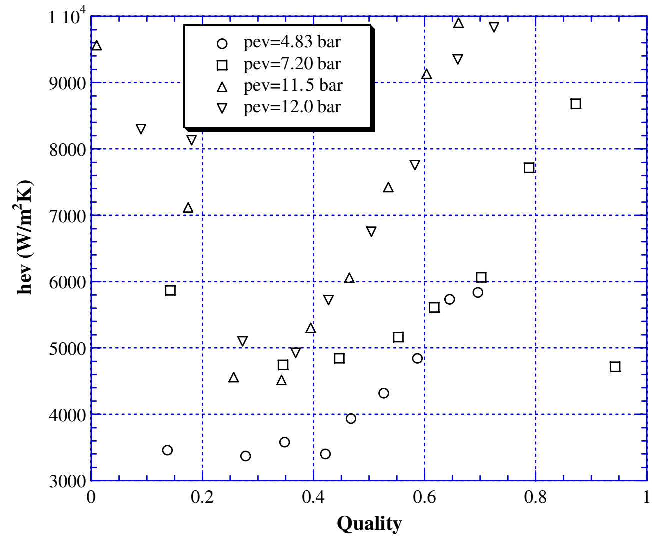

Fig. 12. Heat transfer coefficients of R410A as a function of vapour quality varying evaporating pressure at $q = 17.0$ kW/m^2 and $G = 380$ kg/m^2 s.

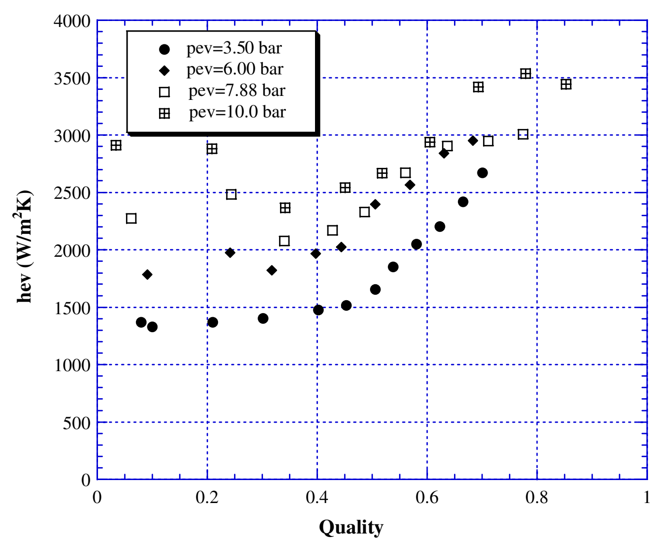

Fig. 13. Heat transfer coefficients of R407C as a function of vapour quality varying evaporating pressure at $q = 8.75$ kW/m^2 and $G = 200$ kg/m^2 s.

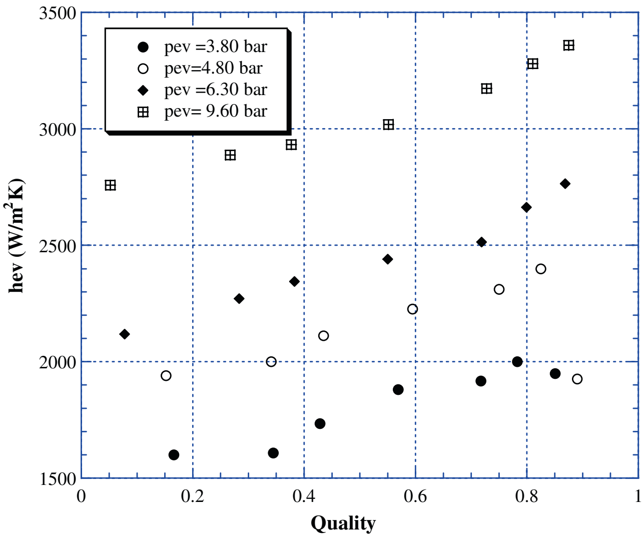

Fig. 14. Heat transfer coefficients of R417A as a function of vapour quality varying evaporating pressure at $q = 8.09$ kW/m^2 and G = 200 kg/m^2 s.

depends on physical properties (surface tension, enthalpy of vaporization, vapour-phase density). With increasing pressure, the vapour density strongly increased and the surface tension decreases.

An increase in pressure would also cause the decrease of the ratio ρ_l/ρ_v to, with an ensuing decrease in interfacial shear stress. This leads to a larger effective film thickness on the tube wall, and the suppression effect of nucleate boiling is reduced.

The effect of pressure on convective boiling contribution mainly results from changes in fluid properties. Indeed, the vapour density increases with increasing pressure. Therefore, the higher average density of the vapour–liquid mixture leads a lower velocity at any given refrigerant mass flux. The liquid conductivity decreases, but the liquid specific heat increases and the liquid viscosity decreases. Both effects lead to a negligible effect of the pressure on convective contribution.

Therefore, the increase of the evaporating pressure has on opposite effect on the nucleate and on the convective contribution. Indeed, the former increases and the latter slightly decreases.

For low mass fluxes, in the whole vapour quality range, the nucleate boiling contribution is important and therefore the heat transfer coefficients always increases with pressure. For higher mass fluxes, the relevance of the convective contribution increases. Therefore, in the low quality range, where nucleate boiling is dominant, the heat transfer coefficients increase with pressure; at higher vapour quality the coefficient tend to merge together because the opposite effects tend to cancel one another.

In the Cooper [42] correlation for nucleate boiling, the effect of pressure is evaluated as

$$h_{nb} \propto p_r^{0.12}(-\log p_r)^{-0.55}. \quad (14)$$

In all the other correlations, the heat transfer coefficient is proportional to the power of 0.2–0.4 of the reduced pressure, but in the proximity of the critical point, where the exponent is larger.

In our experimental tests, in the region dominated by nucleate boiling, the exponent ranges between 0.39 and 0.52, for all the refrigerant fluids tested. Correspondingly, the reduced pressure ranges between 0.06 and 0.323.

4.1.5. Effect of both evaporating pressure and heat flux on boiling heat transfer

The effect of the evaporating pressure and of the heat flux on boiling heat transfer was investigated, as well.

Fig. 15 reports the R404A heat transfer coefficients at a mass flux of 790 kg/m^2 s, varying both the evaporating pressure and the heat flux.

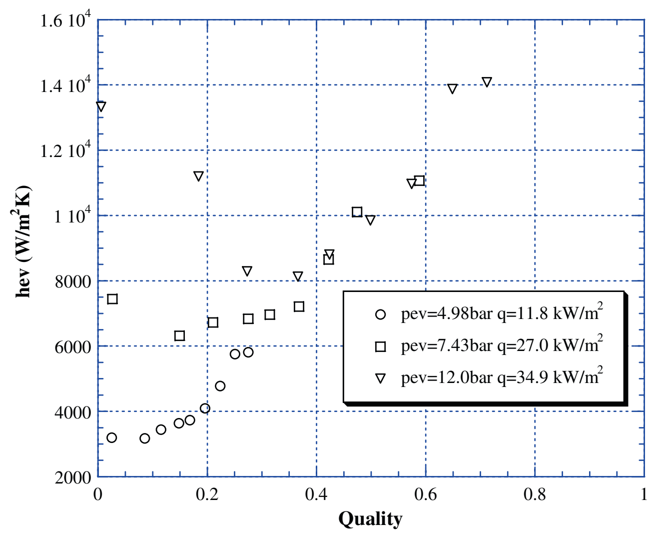

Fig. 15. Heat transfer coefficients of R404A as a function of vapour quality varying evaporating pressure and heat flux at $G = 790$ kg/m^2 s.

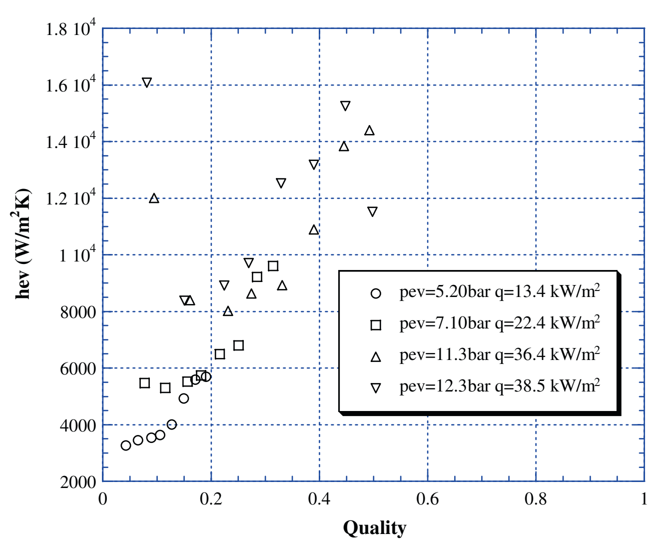

Fig. 16. Heat transfer coefficients of R410A as a function of vapour quality varying evaporating pressure and heat flux at $G = 1079$ kg/m^2 s.

Fig. 16 reports the R4010A heat transfer coefficients for a fixed mass flux of 1079 kg/m^2 s, varying both the evaporating pressure and the heat flux.

The experimental results indicate that, in the nucleate boiling region (vapour qualities between 0% and 30%) for any given refrigerant mass flux, the heat transfer coefficients increase with increasing pressure and heat flux. Indeed, increasing either leads to a corresponding increase in the nucleate boiling contribution to the heat transfer coefficient. In the region dominated by convective boiling (vapour qualities >30%), the heat transfer coefficients merge, since heat flux and pressure both have a negligible effect on convective boiling.

Increasing the mass flux reduces the region dominated by nucleate boiling dominate and the heat transfer coefficients merge at a lower vapour quality.

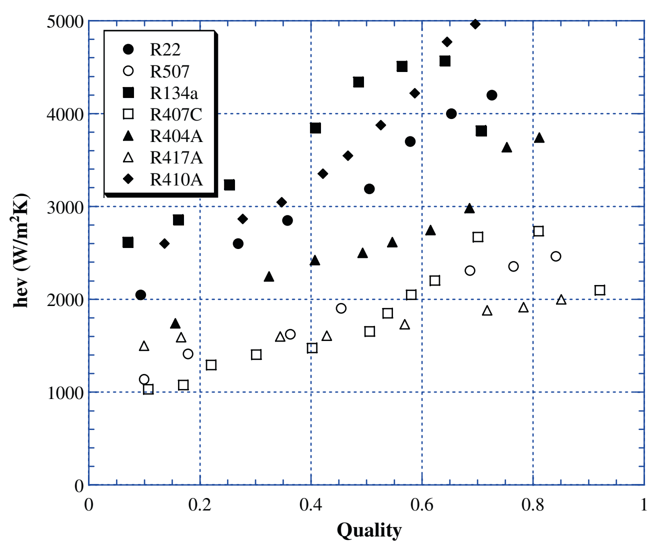

Fig. 17. Heat transfer coefficients as a function of vapour quality varying the refrigerant fluid at $G = 250$ kg/m^2 s, $q = 10.8$ kW/m^2, $p_{ev} = 4.00$ bar.

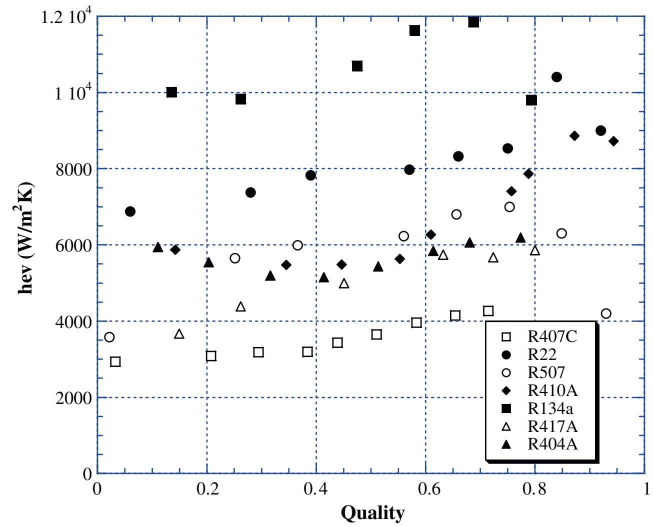

Fig. 18. Heat transfer coefficients as a function of vapour quality varying the refrigerant fluid at $G = 360$ kg/m^2 s, $q = 17.5$ kW/m^2, $p_{ev} = 7.40$ bar.

4.1.6. Effect of fluid properties on boiling heat transfer

The experimental data allow comparing the heat transfer coefficients of R134a, R507, R404A, R410A, R22, R407C and R417A at almost equal pressure, refrigerant mass-flux and heat flux.

Fig. 17 represents heat transfer coefficients of all the refrigerant fluids at a mass flux of 250 kg/m^2 s, a heat flux of about 10.8 kW/m^2, and an evaporating pressure of about 4.00 bar.

Fig. 18 represents heat transfer coefficients of all the refrigerant fluids at a mass flux of 360 kg/m^2 s, a heat flux of about 17.5 kW/m^2 and an evaporating pressure of about 7.40 bar.

Table 3 summarizes the relevant thermo-physical properties of the refrigerants fluids in the operating conditions adopted in the experiments reported.

It can be shown that:

Table 3
The thermo-physical properties affecting heat transfer

Fluid	Bubble/ΔT (°C)	p_{ev} (bar)	ρ_v (kg/m^3)	Δ_{hev} (kJ/kg)	Cp_l (kJ/kg K)	k_l (mW/m K)	μ_l (μPas)	σ (N/m)	Φ (m^3/kg K)$^{0.4}$ (W/m K)$^{0.6}$
R22	−6.56	4.00	17.19	210	1.152	97.695	232	0.01269	7455
R134a	8.93	4.00	19.53	192	1.367	88.084	238	0.01029	7424
R410A	−20.0/0.087	4.00	15.10	249	1.438	122.69	226	0.01235	9435
R404A	−12.7/0.57	4.00	20.47	175	1.339	78.939	212	0.00906	7220
R507	−13.4	4.00	20.82	173	1.328	78.388	214	0.00898	7140
R407C	−10.3/6.36	4.00	16.11	228	1.383	105.89	244	0.01229	8247
R417A	−4.61/4.05	4.00	22.00	164	1.322	84.224	247	0.01052	7025
R22	12.8	7.40	31.29	194	1.209	88.991	188	0.00981	7815
R134a	28.6	7.40	36.03	174	1.440	79.585	186	0.00760	7871
R410A	−2.45/0.100	7.40	27.72	229	1.507	111.39	176	0.00946	10026
R404A	−6.25/0.485	7.40	37.88	158	1.418	71.549	164	0.00679	7718
R507	5.50	7.40	38.53	156	1.407	71.016	165	0.00671	7640
R407C	8.41/5.91	7.40	29.78	208	1.449	95.950	190	0.009375	8753
R417A	14.9/3.52	7.40	40.43	149	1.394	75.680	188	0.007803	7506

(i) In the experimental tests corresponding to the lower value of evaporating pressures and heat flux, the heat transfer coefficients of R134a are better than those pertaining to R22, by a mean factor of about +16%. R410A heat transfer coefficients are slightly better than those of R22, by a mean factor of 11%. R404A heat transfer coefficients are lower than those of R22, by a mean factor of −17%. R507, R407C and R417A heat transfer coefficients are consistently lower than those pertaining to R22 (with mean values of the deviations of −40%, −45% and −46%, respectively).
(ii) In the experimental tests corresponding to higher values of evaporating pressures and heat fluxes, R134a heat transfer coefficients are consistently better than those pertaining to R22, by a mean factor of about +29%. R410A, R507 and R404A heat transfer coefficients are similar and are lower than those pertaining to R22, with a mean value of −19%, −20% and −26%, respectively. R417A and R407C heat transfer coefficients are consistently lower than those of R22, with a mean value of −37% and −55%, respectively.

The heat transfer coefficient in convective boiling results from the interaction between nucleate boiling and liquid convection. According to the Dittus–Boelter, single-phase forced convection correlation, the convective contribution to heat-transfer coefficient is strongly affected by vapour density and is directly proportional to the liquid property combination Φ:

$$\Phi = \left(\frac{Cp_l}{\mu_l}\right)^{0.4} k_l^{0.6}. \tag{15}$$

Therefore, by increasing the value of Φ, the convective contribution to the heat transfer coefficient increases, as well. Furthermore, at any given refrigerant mass-flux, the decrease in vapour and liquid density results in a lower velocity of the refrigerant fluid.

For R134a the liquid property combination Φ is very close to R22, for R404A, R507 and R417A Φ is slightly lower, whereas for R410A and R407C is, respectively, about 28% and 11% greater.

The vapour density of R404A, R507 and R417A are consistently greater than that pertaining to R22.

The nucleate boiling contribution to the heat transfer coefficient is strongly affected by the heat flux, the vapour density, the superficial tension, the heat of vaporization.

The greater vapour density of R134a, together with a lower superficial tension and a greater enthalpy of vaporization, leads to a greater contribution of nucleate boiling, as compared to that of R22. Therefore, in the experimental tests corresponding to the lower value of the evaporating pressure, R134a heat transfer coefficients are slightly better than those pertaining to R22 because of the greater nucleate boiling contribution. The difference in the heat transfer coefficients increases with increasing pressure, since the relative importance of the nucleate boiling contribution increases.

In the experimental tests with low evaporating pressure, R410A heat transfer coefficients are better than those of R22 because of the more relevant convective contribution. For higher evaporating pressures and heat fluxes, i.e. for an increasing relative effect of nucleate boiling, the heat transfer coefficients of R410A become lower than that of R22, because of the lower nucleate boiling contribution (greater superficial tension, lower vapour density) of the latter.

The heat transfer coefficients of R404A and R507 are always lower than that of R22 because of the lower convective contribution than that pertaining to R22 (lower Φ values and higher vapour density). The thermal performance of both R404A and R507 increases with the pressure because of the greater nucleate boiling contribution.

Furthermore, R404A and R410A are almost azeotrope mixtures, with a gliding temperature difference less than 0.5 and 0.1 °C, respectively. Therefore, in the region dominated by nucleate boiling, a little mass transfer resistance can be observed.

The heat transfer coefficients of R407C and R417A are always consistently lower than those of R22. Indeed, the convective contribution of R407C is always higher than those or R22, but the potential for higher heat transfer coefficients is cancelled because of the temperature glide. Indeed, both fluids are zeotropic mixtures with a temperature glide of about 6 and 4 °C, respectively. Therefore, the nucleate boiling contribution to the global heat transfer coefficients is strongly reduced by diffusional limitation. The superposition of heat transfer and mass transfer phenomena associated with the evaporation of a zeotropic mixture decreases the heat transfer coefficient as compare to those pertaining to pure liquids. The heat transfer coefficients decreases with the difference in concentration more strongly at high pressure than at low pressures. This can be explained by the fact that the number of vapour bubbles being formed per surface unit increases with the pressure. The vapour bubbles are more densely packed, and there is less surface available for the mass flow rate in the liquid space. The delivery of low boiling components is thus more strongly impeded, this leads to a further reduction in the heat transfer.

5. Conclusions

An experimental plant has been set up at the University of Naples for evaluating the heat transfer characteristics of pure and mixed refrigerants during convective boiling. The local heat transfer coefficients of R22, R134a, R507, R404A, R410A, R407C and R417A were determined in a smooth horizontal tube during convective boiling.

The experimental tests are carried out varying: the refrigerant mass fluxes within the range 200–1100 kg/m^2 s; the heat fluxes within the range 3.50–47.0 kW/m^2; the evaporating pressures within the range 3.00–12.0 bar.

In this study, the effects of the refrigerant vapour quality, mass flux, saturation temperature, imposed heat flux and thermo-physical properties on the measured heat transfer coefficient were analyzed in detail.

The following conclusions can be drawn from the experimental evidence:

(1) In the experimental tests where liquid convection is the dominating mechanism, the heat transfer coefficient increases with quality. In the experimental tests at high evaporation pressures and heat fluxes there are two distinct heat transfer regions during evaporation and the heat transfer coefficients initially decrease with vapour quality and then after increase.
(2) The heat transfer coefficients always increase with mass flux. The exponent n that describes the influence of the refrigerant mass flux G on the heat transfer coefficients varies in the range 0.52–0.83, except for some data points corresponding to a very low-quality region.
(3) The heat transfer coefficients depend on the heat flux only in the region where nucleate boiling dominates. In this region, the exponent nq that describes the influence of the heat flux on the heat transfer coefficients varies in the range 0.53–0.74. From the experimental results it is apparent that nq varies with the evaporating pressure and therefore with the reduced pressure.
(4) The heat transfer coefficients increase with the evaporating pressure. For low mass fluxes, in all the vapour quality range explored, the heat transfer coefficients always increase with pressure. For higher mass fluxes, in the low quality range, the heat transfer coefficients increase with pressure; at higher vapour quality the coefficient tend to merge together. The exponent n that describes the influence of the evaporating pressure on the heat transfer coefficients in the nucleate boiling dominate region varies between 0.39 and 0.52.
(5) The heat transfer coefficients of R134a are higher than those of all the other refrigerant fluids. The difference increases with increasing evaporating pressure. The heat transfer coefficients of the zeotropic mixtures R407C and R417A are always lower because of the mass transfer resistances in nucleate boiling.

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