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A STUDY ON BOILING HEAT TRANSFER CHARACTERISTICS OF REFRIGERANT MIXTURES IN A HORIZONTAL TUBE

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ABSTRACT

In the present study the evaporating heat-transfer characteristics of mixtures of refrigerant fluids were analysed. This paper presents experimental heat transfer coefficients obtained during the evaporation of the zeotropic mixtures: R407C (R32-R125-R134a 23-25-52% in weight) and R417A (R125-R134a-R600 46.6-50-3.4 % in weight) and of the near-azeotropic mixtures: R404A (R125/R143a/R134a 44/52/4 % by weight) and R410A (R32/R125 50/50 % by weight).

The test section was a smooth, horizontal, stainless steel tube (6 mm I.D., 6 m length) uniformly heated by Joule effect. The heat transfer characteristics were measured at an evaporation pressure of about 4.00 bar varying the refrigerant mass flux within the range 200-1100 kg/m^2s, with heat fluxes within the range 10-15 kW/m^2. The experimental data are discussed in terms of the heat transfer coefficients as a function of the vapour quality. The experimental results clearly show that the heat transfer coefficients increase with increasing the refrigerant mass flux with a power coefficient varies in the range 0.57 – 0.86. In the experimental tests R410A shows the higher values of the heat transfer coefficients. In the paper are reported also the pressure drops values for the same experimental data.

INTRODUCTION

The constant depletion of the ozone layer has determined many international agreements demanding a gradual phase-out of the halogenated fluids. As known the CFCs have been banned since 1996. Also the partially halogenated HCFCs can't be used for the manufacture of new equipment in all refrigerating and air conditioning applications from 1st January 2001; relatively to the existing equipment there will be a ban on the use of virgin HCFCs from 1st January 2010 and a ban on the use of all HCFCs, including recycled materials, from 1 January 2015 [1, 2]. The HFCs are synthetic refrigerant fluids entirely harmless towards the ozone layer since they do not contain chlorine. For this reason actually they are the most used substitutes of HCFCs. On the other hand there is a further key environmental issue that must be considered in the choice of the HCFCs alternatives: the global warming. Hence, it is essential to choice the refrigerant fluid in order to maximize the energy efficiency of the equipment. To maximize the performances of the refrigerating and air-conditioning plant operating with HFCs, a detailed analysis of each component is necessary. Among the components which influence the cycle efficiency, the heat exchangers (evaporator and condenser) are of particular importance. To optimise the design of evaporators and condensers for refrigeration and air conditioning applications, the accurate knowledge of heat transfer coefficients and pressure drops is fundamental.

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 [3, 4].

The objective of the present experimental study are to: (i) develop an accurate flow boiling heat transfer database for several important new fluids, (ii) provide data to the refrigeration industry for the design of high efficiency evaporators, (iii) compare the thermal performances of R404A, R410A, R407C, R417A and (iv) investigate the influence of the refrigerant mass flux on the flow boiling characteristics of the fluids.

An experimental plant has been set up at the University of Naples for evaluating the heat-transfer characteristics of pure refrigerants and refrigerant mixtures during convective boiling. In the present paper the local heat transfer coefficients of zeotropic mixtures R407C and R417A and of the azeotropic mixtures R404A and R410A are reported.

In table 1 are reported the characteristics of the refrigerant fluids tested. Specifically, the composition, the chlorine content (ODP), the direct global warming potential (GWP), and the temperature glide are reported. All the refrigerant fluids are non flammable and non toxic.

Table 1. The refrigerant fluids tested.

Refrigerant fluid	Composition % in weight	ODP	GWP	Temperature glide at O°C
R407C	R32/R125/R134a (23/25/52)	0	1650	6.12
R417A	R125/R134a/R600 (46.6/50/3.4)	0	1900	3.93
R404A	R125/R143a/R134a (44/52/4)	0	1730	0.512
R410A	R32/R125 (50/50)	0	3260	0.104

NOMENCLATURE

A	[m^2]	heat transfer area
ΔT	[°C, K]	temperature glide
h	[W/Km^2]	heat transfer coefficient
p	[bar]	pressure
q	[kW/m^2]	heat flux
T	[°C, K]	temperature

Subscripts

bubble	bubble
ev	evaporative
sat	saturation
w	inner wall

THE EXPERIMENTAL APPARATUS AND DATA REDUCTION

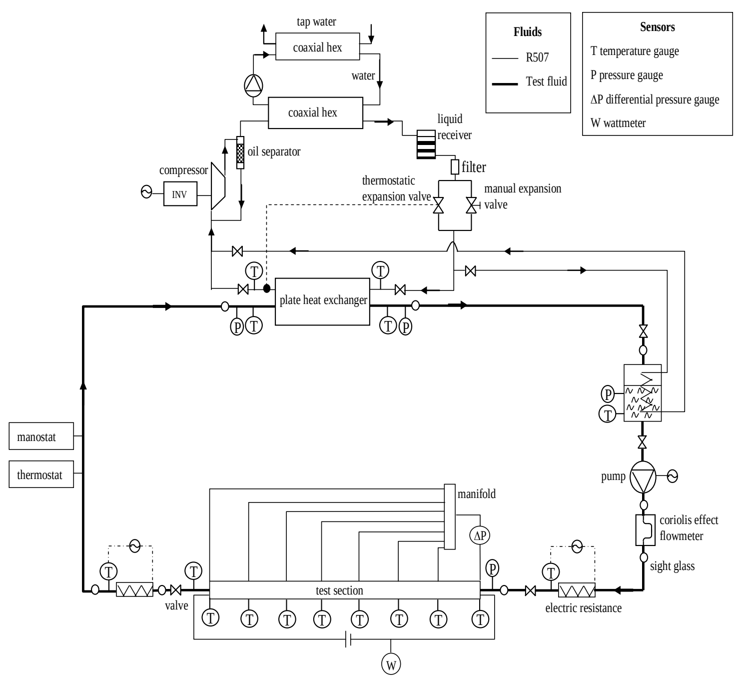

Fig. 1 The experimental apparatus.

A schematic view of the experimental apparatus is given in figure 1. This apparatus consists in two main loops: the test refrigerant loop and a secondary vapour compression loop for condensing the test refrigerant fluid.

The test refrigerant fluid evaporates in the test section. Its condensation is achieved in a plate type evaporator of the secondary vapour compression refrigerating plant.

Along the main loop, a gear pump drives the liquid refrigerant and by means of a hydrodynamic speed variator allows the refrigerant to flow at different flow rates. The refrigerant mass flux can be varied within the range 100-1000 kg/s m^2. Refrigerant mass flow-rate is measured by a Coriolis effect mass flow-meter, inserted after the gear pump, where sub cooled liquid conditions have been achieved. Since it could be affected by mechanical vibrations, it was mounted on a 5 kg steel plate separated from the plant.

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. It is possible to have the desired heat flux by varying the direct electrical current. The heat flux supplied along the tube can be safely considered constant. In the test section to obtain heat-transfer coefficients from the experiments, the heat flux supplied to the refrigerant and temperatures of the refrigerant and of the wall should be measured.

Heat-transferred to the refrigerant at the test section is measured knowing the values of the voltage supplied to the test section and of the consequent current.

The voltage and the current are monitored by means of a voltmeter and an amperometer, respectively. In the test section, the outer wall temperature of the heated tube is measured by means of 32 four-wire 100 Ω platinum resistance thermometers mounted in 8 stations. Each station had four resistance thermometers clamped on the top, bottom, and both sides of the tube. At the inlet and the outlet of the test section two four-wire 100 Ω platinum resistance thermometers are inserted in the refrigerant flow stream. Absolute pressure at the inlet of the test section is measured by a piezoelectric pressure transducer. A piezoelectric pressure difference gauge measures the pressure drop between the inlet and the outlet of the test evaporator, and the pressure drop in each measurement station.

The refrigerant condenses in a plate heat exchanger. This heat exchanger is the evaporator of a vapour compression plant, working as auxiliary. The secondary vapour compression loop working with R507, consists in: a semi hermetic bi-cylindrical compressor, a tap water plate condenser, a thermostatic valve, the already mentioned plate evaporator where the test refrigerant fluid works as secondary fluid.

In order to analyse the influence of operating parameters on the evaporative heat transfer characteristics, the experimental apparatus must be able to carry out tests in which it is possible to control independently the evaporation temperature, the mass flow rate of the circuiting refrigerant and the heat flux inside the test section. In the experimental apparatus built the test temperature depends on the evaporating temperature of the secondary fluid because the condensation of the test fluids happened with the heat transfer to secondary fluid in the plate evaporator. In order to obtain different evaporating temperature at fixed evaporation power the compressor motor of the secondary plant is equipped with an inverter that enables its speed to be suitably varied. When the compressor number of revolution varies the thermostatic valve might operate under severely different conditions; indeed when the number of revolutions is low occurs that the refrigerant pressure across the thermostatic valve is too small to permit lamination of the refrigerant fluid. In this circumstance the orifice of the thermostatic valve is too large, and so a hand regulation valve has been inserted in the secondary loop to have a correct

lamination. Using all systems it can be varied the evaporating temperature within -15÷20°C.
The location of the temperature, pressure and mass flow rate measure devices are indicated in Fig. 1. Table 2 summarizes all characteristics of the plant instrumentation.

Table 2. Plant instrumentation.

Variable	Device	Accuracy	Range
Temperature	Resistance thermometers Pt100	± 0.03°C	-50-100 °C
Pressure	Piezoelectric	± 1.4%	0-14 bar
Differential pressure	Piezoelectric	± 0.1%	0-1 bar
Mass Flow-rate	Coriolis effect	± 0.2 %.	0-0.033 kg/s
Voltage	Voltmeter	± 0.2 %	0-30 V
Direct Current	Amperometer	± 0.2 %	0-220A

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. In the test section, the transferred heat (evaluated through the supplied voltage and the ensuing current) and those evaluated by means of an energy balance on the refrigerant fluid agree to within $\pm$ 3% maximum error (less then 1 % average error).

All tests are carried at steady-state conditions. Temperature and pressure values in key points of the plant were continuously monitored, in order to check the achievement of steady-state conditions. The latter are assumed to hold when the deviations of the controlled values from their corresponding mean values fall within a predetermined interval. At this stage, the test started and the logging of data with 0.1 Hz acquisition frequency was performed on all channels for 1000 s. For each channel, the 100 samples recorded were averaged. Each sample was checked against the corresponding mean value and it was rejected if it did not lie within the fixed range. If more than 5% of the samples were rejected, the whole test is discarded. Each test was iterated three times, in order to check reproducibility.

A personal computer connected with a data acquisition system, consisting of a controller, a 48 channel scanner, and a multimeter was used to record the measurement data.
The local heat transfer coefficient is defined as:

$$h_{ev} = \frac{q}{(T_W - T_{sat})}$$

where:
• q is the inner wall heat flux based on the inside surface area of the tube;
• T_w is the local inner wall temperature;
• T_{sat} is the saturation temperature.
The inner wall temperature of the test section, T_w, is estimated from the measured outside wall temperature by applying the one-dimensional, radial, steady-state heat conduction equation for a hollow cylinder assuming uniform heat generation within the tube wall and an adiabatic condition on the outside of the tube.

The evaporating fluid temperature T_{sat} is calculated rather than directly measured. The 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 sub-section. For a zeotropic mixture the saturation temperature is a function of quality as well as pressure. The quality in each section is evaluated by an energy balance from the inlet to wall temperature measurement position. The saturation temperature is obtained from the saturated pressure and quality from the evaporation curve of the mixture.
For the data analysis the thermophysical properties of pure and mixed fluids are evaluated using the software REFPROP 7.0 [5].
Under the employed operating condition, the accuracy of the heat transfer coefficients varies between 2.52% and 7.72%. Uncertainty in the experimental data was calculated using the simple sample analysis suggested by Moffat [6].

EXPERIMENTAL RESULTS

In the present paper the heat transfer coefficients and pressure drops were measured varying the refrigerant mass flux, keeping the evaporating pressure almost constant. The tests conditions are performed in the evaporating pressure range 3.60 - 4.40 bar, varying the refrigerant mass flux in the range 200 - 1100 kg/m^{2}s, and the heat flux in the range 10 - 15 kW/m^2. The evaporating pressure at the inlet of the test section, the corresponding saturation temperature, the temperature glide and the heat flux for each fluid are summarized in Table 3.

Table 3. The operating conditions

Refrigerant fluid	G (kg/m^{2}s)	p (bar)	Tbubble (°C)	ΔT (°C)	q (kW/m^2)
R407C	199	3.63	-13.0	6.42	9.90
	344	3.56	-13.6	6.43	13.1
	507	3.95	-10.7	6.38	14.6
	700	3.50	-14.0	6.44	14.1
	1100	3.70	-12.5	6.41	15.1
R417A	202	3.10	-6.11	4.09	9.89
	347	3.60	-7.70	4.14	11.9
	512	3.90	-5.35	4.08	14.3
	697	3.90	-5.35	4.08	14.7
	1109	4.01	-4.60	4.05	14.9
R410A	363	4.20	-18.7	0.091	13.4
	574	4.30	-18.1	0.092	13.4
	784	4.00	-20.0	0.091	14.4
	1068	4.40	-17.5	0.092	14.8
R404A	289	3.75	-14.5	0.578	10.7
	477	4.20	-11.3	0.563	13.6
	793	4.15	-11.6	0.565	14.8
	1078	4.19	-11.4	0.564	15.0

Since the heat transfer process depends upon the flow regime, to calculate the heat transfer coefficients, one must first estimate which flow pattern is present.
In the present paper the evaluation is made with the latest version of the Kattan-Thome-Favrat map [7.8].
An intermittent flow pattern was observed from the inlet section to the part of the test section with vapour qualities between 20 –

30 %, depending on the test conditions. . The intermittent flow covers both plug and slug flow regimes (it is essentially a stratified–wavy flow pattern with large amplitude waves that wash the tube top). With increasing vapour qualities, an annular flow was achieved. At higher vapour qualities (>80%) the liquid film in the upper part of the tube disappears and the flow became annular with a partial dry-out.
Figure 2, figure 3, figure 4 and figure 5 show R417A, R407C, R414A and R410A experimental coefficients during evaporation as a function of the vapour quality varying the refrigerant mass flux at a fixed evaporating pressure and with little variation of the heat flux.

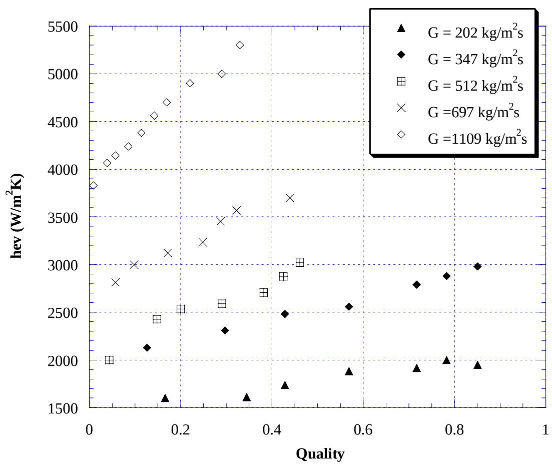

Fig. 2 Heat transfer coefficients of R417A as a function of vapour quality for different values of refrigerant mass flux at an almost constant pressure of 3.80 bar.

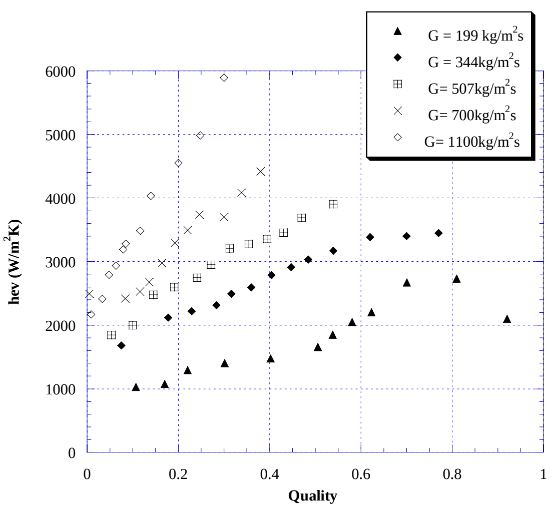

Fig. 3 Heat transfer coefficients of R407C as a function of vapour quality for different values of refrigerant mass flux at an almost constant pressure of 3.70 bar.

As seen in figures, the heat transfers coefficients increases increasing the vapour quality. Indeed, the convective contribution to heat transfer predominates at low heat-fluxes and low evaporating pressures. During the experimental tests where liquid convection was the main heat transfer mechanism, convective evaporation being predominant. Under these conditions, the heat transfer coefficients increased with vapour quality. Indeed, as the flow proceeded downstream and vaporization occurred, the void fraction increased, and the density of the liquid-vapour mixture decreased. Thus, the flow accelerates enhancing convective transport from the heated wall of the tube.
In the annular flow regime, heat transfer is dominated by evaporation at the interface between the liquid film at the tube wall and the vapour at the tube core.
As vaporization proceeds, the thickness of the liquid film at the tube wall decreases together with its thermal resistance, thereby enhancing the heat-transfer effectiveness. The ensuing increase of the 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.
The dependence of the heat transfer coefficients on the vapour quality increases increasing the refrigerant mass flux at fixed evaporating pressure. Indeed, at low refrigerant mass fluxes the nucleate boiling contribution to the heat transfer coefficient is stronger and therefore the heat transfer coefficient is less sensitive to vapour quality.

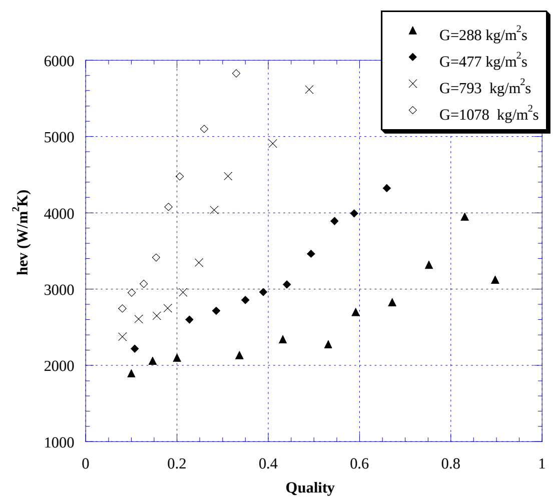

Fig. 4 Heat transfer coefficients of R404A as a function of vapour quality for different values of refrigerant mass flux at an almost constant pressure of 4.10 bar.

In this paper the attention has been focused on the effect of mass flow rate on the heat transfer coefficient during evaporation of the refrigerant mixtures. The experimental data clearly show that the heat transfer coefficients increase with increasing the refrigerant mass flux. Indeed, increasing the refrigerant mass flux increase the fluid velocity, enhancing the convective boiling. In a very low quality region, corresponding to 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.

R410A show the higher values of heat transfer coefficients. R410A vapour density is the lower. Therefore, the lower average density of the vapour-liquid mixture of R410A leads to the higher velocity at any given refrigerant mass flux.
Furthermore, the liquid conductivity of R410A is the higher. Therefore, the convective contribution to heat transfer of R410A is the higher.
From figure 6 to figure 9, graphs show R417A, R407C, R414A and R410A experimental pressure drop values as a function of vapour varying the refrigerant mass flux at a fixed evaporating pressure. As seen in the figures, the pressure drop increases with the quality.

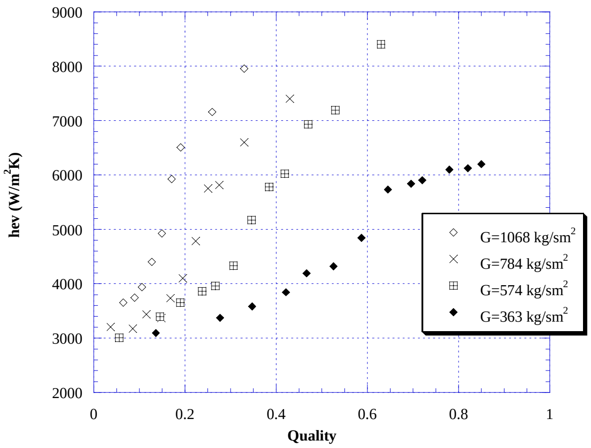

Fig. 5 Heat transfer coefficients of R410A as a function of vapour quality for different values of refrigerant mass flux at an almost constant pressure of 4.20 bar.

In literature are available many correlations for the evaluation of the heat transfer coefficient in flow boiling for pure and mixed refrigerants. In most of the available correlation the heat transfer is proportional to the mass flux to the power of 0.80. In the present study the power coefficient varies in the range 0.57 – 0.86, except for some data points corresponding to a very low quality region.
The experimental data allow a comparison between R417A, R407C, R410A, R404A and R410A heat transfer coefficients obtained with the same boundary conditions. It can be shown that:

(i) the near azeotropic mixtures heat transfer coefficients are higher than those of the zeotropic mixture. The difference decreases increasing the refrigerant mass flux;

(ii) R410A heat transfer coefficients are higher than those of R404A, R407C and R417A. The difference with R404A ranges from -20 to -40 %, with R407C from-25 to -60 %, with R417A from -27 to -64 %.

During evaporation in the global heat transfer process, both nucleate boiling and liquid convection may be active heat transfer mechanisms. At low mass fluxes, nucleate boiling is increasingly important. The difference between azeotropic and zeotropic mixtures heat transfer coefficient is more marked corresponding to the runs carried out with lower refrigerant mass flux. Indeed, in this region where the influence of nucleate boiling is stronger, the heat transfer coefficient of the zeotropic mixtures (R407C and R417A) is 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 a near azeotropic mixture with negligible temperature glide.
At higher mass fluxes the difference between the heat transfer coefficients decreases. In this region the convective boiling contribution to the heat transfer coefficient is increasingly important.

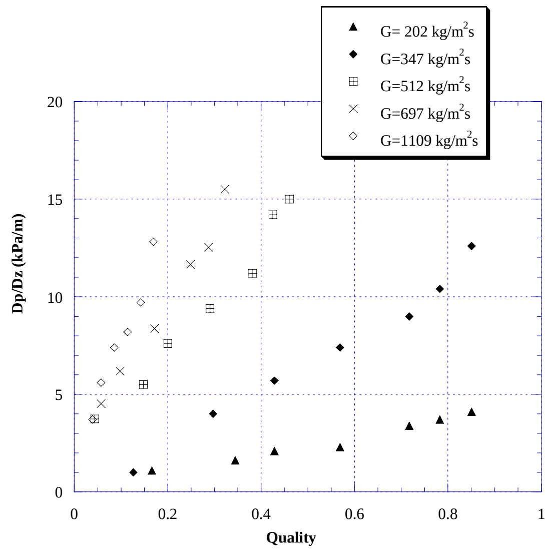

Fig. 6 Pressure drops of R417A as a function of vapour quality for different values of refrigerant mass flux at an almost constant pressure of 3.80 bar.

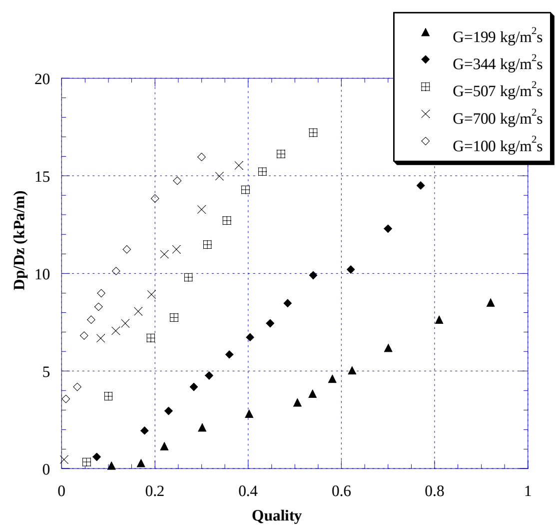

Fig. 7 Pressure drops of R407C as a function of vapour quality for different values of refrigerant mass flux at an almost constant pressure of 3.70 bar.

The total pressure drop is mainly caused by the frictional pressure drop. It involves not only the transfer of momentum between fluid and wall, but also the transfer of momentum between the individual phases. This contribution increases with the vapour quality to a maximum between 70-90% and then decreases to the pressure drop of the sub critical vapour flow. 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 increasing the pressure drop.
The figures clearly show that at fixed pressure, pressure drop increases significantly with the increase of the mass flux. For a given quality, the pressure gradient is approximately proportional to $G^{1.4 \div 1..8}$ for all refrigerant fluids.
The experimental data allow the comparison of the pressure gradients for the tested fluids at equal pressure and refrigerant mass-flux. It can be shown that the pressure drop of the refrigerant fluids are similar. R417A and R407C pressure drops are slightly higher. This is a direct consequence of the greater liquid viscosity.

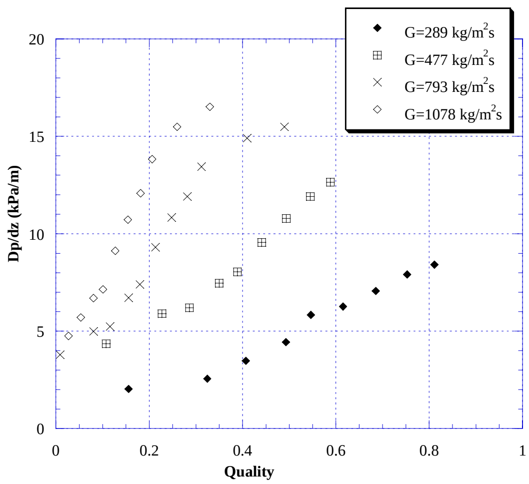

Fig. 8 Pressure drops of R404A as a function of vapour quality for different values of refrigerant mass flux at an almost constant pressure of 4.10 bar.

CONCLUSIONS

In the present paper attention has been focused on two-phase heat transfer characteristics of the zeotropic mixtures R407C and R417A and of the near azeotropic mixtures R404A and R410A. The test section is a smooth, horizontal, stainless steel tube (6 mm I.D., 6 m length) uniformly heated by Joule effect. The experimental data were taken in the evaporating pressure range 3.60 - 4.40 bar, varying the refrigerant mass flux in the range 200 - 1100 kg/m^2s, and the heat flux in the range 10 - 15 kW/m^2.
On the basis of the present experimental study, the following conclusions can be drawn:
(1) the heat transfer coefficients increase with increasing the refrigerant mass flux. In a very low quality region, the influence of the refrigerant mass flux became weaker and the heat transfer coefficients tend to merge together.
(2) The heat transfer is proportional to the mass flux with a power coefficient varies in the range 0.57 – 0.86.
(3) The heat transfer coefficient of the zeotropic mixtures is always lower than that of the quasi-azeotropic mixtures.
(4) R410A show the higher values of the heat transfer coefficients.
(5) The pressure drops are proportional to the mass flux with a power coefficient varies in the range 1.4 – 1.8.
(6) All the refrigerant fluids tested show similar pressure drops at constant pressure of about 4.00 bar.

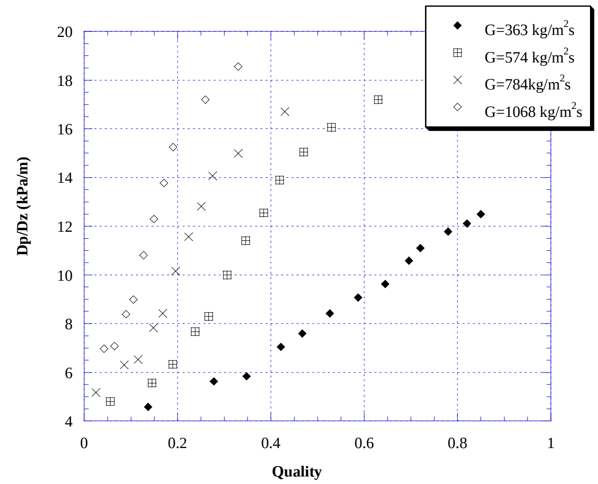

Fig. 9 Pressure drops of R410A as a function of vapour quality for different values of refrigerant mass flux at an almost constant pressure of 4.20 bar.

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