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► **To cite this version:**

Bruno Agostini, André Bontemps. Vertical flow boiling of refrigerant R134a in small channels. International Journal of Heat and Fluid Flow, 2004, 28, pp.97-103. 10.1016/j.ijheatfluidflow.2004.08.003 . hal-00184231

**HAL Id: hal-00184231**

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Submitted on 3 Feb 2020

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# Vertical flow boiling of refrigerant R134a in small channels

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This article presents an experimental study of ascendant forced flow boiling in mini-channels with refrigerant R134a. A flat aluminium multi-port extruded tube composed of 11 parallel rectangular channels (3.28 mm × 1.47 mm) with hydraulic diameter of 2.01 mm was used. Mass flux ranged from 90 to 295 kg/m<sup>2</sup>s and heat flux from 6.0 to 31.6 kW/m<sup>2</sup>. Two working pressures, 405 and 608 kPa, were tested. Inlet subcooling varied from 1 to 17 K. Heat transfer was found to be greater than that previously reported in the literature for conventional tubes, while dry-out occurred at low qualities.

*Keywords:* Mini-channel; Compact heat exchanger; Heat transfer coefficient; Dry-out; Flow boiling

## 1. Introduction

The use of mini-channel heat exchangers (hydraulic diameter about 1 mm) in compact heat exchangers improves heat transfer coefficients, and thermal efficiency while requiring a lower fluid mass. They are widely used in condensers for automobile air-conditioning and are now being used in evaporators, then in other applications like domestic air-conditioning systems. However, more general use requires a better understanding of boiling heat transfer in confined spaces. In particular it is necessary to set out new correlations valid in mini-channels.

Existing correlations for forced flow boiling in conventional tubes such as those developed by Shah (1976); Liu and Winterton (1991); Steiner and Taborek (1992) or Kandlikar (1990) fail to predict the performances in mini-channels. Very recently Kandlikar (2004) developed a new correlation adapted to mini-channels which gives good results for low qualities but fails to

take dry-out into account. A few studies on boiling in mini-channels are available in literature and the experimental conditions are gathered in Table 1. Cornwell and Kew (1992, 1995) and Kew and Cornwell (1994, 1997a,b) analysed with a dimensionless confinement number <sup>1</sup>, the effects of the confinement on the heat transfer in mini-channels. Tran et al. (1993, 1996, 1997) and Wambsganss et al. (1993) published a very complete experimental study of forced flow boiling in mini-channels. They developed a new correlation taking into account the confinement number and found that nucleate boiling was a dominant phenomenon for  $\dot{q} < 8 \text{ kW/m}^2$  and  $\Delta T_{\text{sat}} > 3 \text{ K}$ . Aritomi et al. (1993) found nucleate boiling to be dominant in their confined annular geometry and showed that the confinement increased the heat transfer coefficient by  $D_h^{-0.75}$ . Oh et al. (1998) found that convective boiling and dry-out were the dominant phenomena during their experiments.

The motivation for the present work is therefore to get a more accurate vision of boiling in mini-channels.

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<sup>1</sup> The confinement number is the inverse of the Bond number square root.



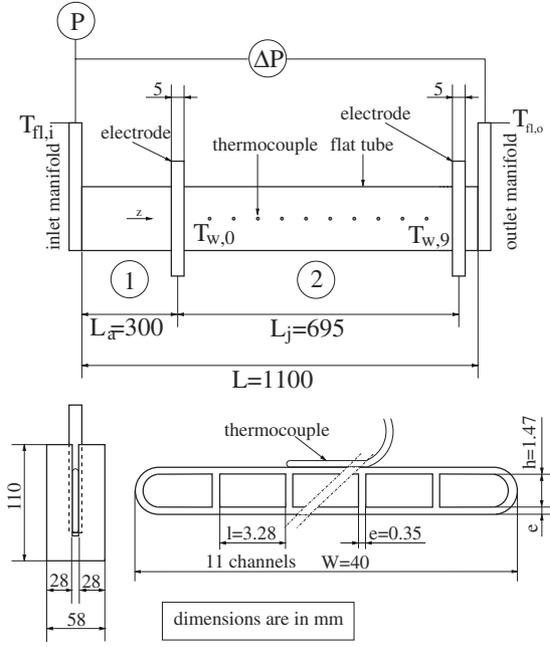


Fig. 1. Test section, dimensions in mm.

the length  $L_j$  of the tube (zone ②) is heated by the Joule effect with the passage of an electric current (up to 2800 A) from two brased electrodes through the tube wall. Upstream of the heated region there is an adiabatic zone of length  $L_a$  (zone ①) to ensure the flow was hydrodynamically developed. Experimental conditions are summarised in Table 2.

Fig. 1 shows the test section and instrumentation. Wall temperatures  $T_{w,j}$  ( $0 < j < 9$ ) on the tube external surface were measured with 0.5 mm diameter calibrated type E thermocouples. Fluid inlet and outlet mean temperatures (respectively  $T_{fl,i}$  and  $T_{fl,o}$ ) were measured with 1 mm diameter calibrated type K thermocouples. Calibration was carried out every 5 K between 268 and 333 K with a Rosemount 162-CE platinum thermometer. The thermocouples used for wall temperature measurements were equally spaced and fixed with aluminium

adhesive on the tube surface. Due to the high thermal conductivity of the aluminium and the low thickness of the tube walls (0.35 mm) the measured temperature is very close to the wall temperature in contact with the fluid (the difference was estimated at less than 0.01 K). The inlet fluid pressure was measured with a calibrated Rosemount type II absolute pressure sensor. Two differential pressure sensors calibrated from 0 to 7.6 kPa and 40.5 kPa measured the pressure loss through the test section. A Rosemount micro-motion coriolis flowmeter was used to measure the mass flux of R134a downstream of the pump. The heating voltage  $U$  and current  $I$  were measured directly through a HP 3421A multiplexer.

Heat flux was varied for every fixed mass flow rate in order to obtain a series of outlet vapour qualities between 0.2 and 1 with a step of approximately 0.05. Steady state values were monitored using a Hewlett Packard 3421A with a 30 min time lapse between each mass flow rate or heat flux change. Averaging was carried out after every 20 values and uncertainties were calculated according to the Kline and McClintock (1953) and Moffat (1982, 1985) methods. Uncertainties are reported in Table 2. The total electrical power  $P$  dissipated in the test section was calculated as the product of voltage and current. The variations of R134a thermophysical properties with temperature were calculated with the REFPROP 6 software.

The determination of the channel dimensions was carried out using scanning electron microscopy. The dimensions of several channels were measured and the mean value and standard deviation were calculated. The uncertainty was estimated as twice the standard deviation (95% of the measurements in this interval). The hydraulic diameter was calculated with the total flow area and wet perimeter measured from electron microscope images in order to take into account the effect of the first and last channels which are rounded. Roughness measurements were also carried-out. The dimensions are  $\bar{l} = 3.28 \pm 0.02$  mm,  $\bar{h} = 1.47 \pm 0.02$  mm and  $\bar{R}_a < 1 \mu\text{m}$ , yielding  $\bar{D}_h = 2.01 \pm 0.06$  mm.

Table 2  
Operating conditions and uncertainties

	Value	Error
$D_h$ (mm)	2.01	$\pm 3\%$
$L$ (mm)	1100	$\pm 0.4\%$
$\dot{m}$ (kg/m <sup>2</sup> s)	90–295	$\pm 1.7\text{--}8.6\%$
$\dot{q}$ (kW/m <sup>2</sup> )	6–31.6	$\pm 2\text{--}4.1\%$
$T_w, T_n$ (K)	276–308	$\pm 0.1\text{--}3$ K
$\Delta T_{sub,i}$ (K)	1–17	$\pm 0.1$ K
$p_i$ (kPa)	405, 608	$\pm 4\%$
$\Delta p$ (kPa)	9.5–37.5	$\pm 1.2\text{--}17\%$
$\alpha$ (kW/m <sup>2</sup> K)	0.8–10.3	$\pm 6\text{--}30\%$
$x_o$	0.26–1	$\pm 1\text{--}7\%$
$L_{TP}$ (mm)	596–689	$\pm 1\text{--}4.5$ mm
Flow	Ascendant	

### 3. Data reduction

The heat flux and the power dissipated between 0 and  $z$  are calculated respectively by

$$\dot{q} = \frac{U \cdot I}{S} \quad \text{and} \quad \dot{Q}(z) = U \cdot I \cdot \frac{z}{L_j}. \quad (1)$$

If the subcooled fluid receives the power  $\dot{Q}(z)$ , its temperature is calculated via the power balance:

$$T_{fl}(z) = T_{fl}(0) + \frac{\dot{Q}(z)}{\dot{M} \cdot \bar{c}_{p,l}} \quad \text{with} \quad \bar{c}_{p,l} = \frac{c_{p,l}(0) + c_{p,l}(z)}{2}. \quad (2)$$

This is an implicit equation for  $T_{fl}(z)$  ( $c_{p,l}(z)$  is a function of  $T_{fl}(z)$ ) and it is resolved iteratively. An iterative method is also used to calculate the tube length  $z_{boil}$  where bulk boiling starts, i.e. where the fluid bulk temperature is equal to the saturation temperature at the local pressure, as liquid pressure losses cannot be neglected in mini-channels. Initial conditions for the iteration are

$$z_{boil} = L_j,$$

$$\Delta p_{lo} = \frac{\dot{m}^2}{2\rho_l} \cdot 4f_1 \cdot \frac{(z_{boil} + L_a)}{D_h} + \rho_l \cdot g \cdot (z_{boil} + L_a), \quad (3)$$

$$T_{boil} = T_{fl}(z_{boil}) = T_{fl}(0) + \frac{\dot{Q}(z_{boil})}{\dot{M} \cdot c_{p,l}},$$

$$\Delta T = T_{boil} - T_{sat}(p_i - \Delta p_{lo}).$$

Iteration for  $z_{boil}$  is done using the dichotomy method until  $|\Delta T| < 10^{-6}$ . The two-phase flow length is  $L_{TP} = L_j - z_{boil}$ . A previous experimental campaign on liquid flows in this tube by Agostini et al. (2002) provided values for  $4f_1$ . Calculation of the local vapour quality, fluid pressure and temperature, was carried out using a power balance over the length  $dz$ ,

$$\delta \dot{Q} = \dot{M} \cdot [x \cdot c_{p,v}(T_{sat}) \cdot dT_{sat} + (1-x) \cdot c_{p,l}(T_{sat}) \cdot dT_{sat} + h_{lv}(T_{sat} + dT_{sat}) \cdot dx], \quad (4)$$

where  $c_p(T_{sat})$ ,  $h_{lv}(T_{sat})$  are  $c_p$ ,  $h_{lv}$  at  $T_{sat}$ . Here, the final term  $\dot{M} \cdot h_{lv}(T_{sat} + dT_{sat}) \cdot dx$  is the power required to vaporise a  $dx$  fraction of fluid,  $\dot{M} \cdot x \cdot c_{p,v}(T_{sat}) \cdot dT_{sat}$  and  $\dot{M} \cdot (1-x) \cdot c_{p,l}(T_{sat}) \cdot dT_{sat}$  are the sensible heat converted into latent heat so that the thermal equilibrium should be maintained as the saturation temperature decreases.

Triplett et al. (1999) measured pressure drop and void fraction in mini-channels with air–water adiabatic flows and showed that the homogeneous model best predicted their results in the slug, churn and slug-annular flow patterns. According to their flow maps and our experimental conditions such patterns should be encountered in the present study. Thus the homogeneous model was used to calculate the local pressure inside the tube in the two-phase zone. In the homogeneous model the pressure drop gradient with acceleration, frictional and gravity contributions is

$$-\left(\frac{dp}{dz}\right) = \frac{\dot{m}^2 \cdot (v_v - v_l) \cdot \frac{dx}{dz} + \rho_{TP} \cdot g + \frac{\dot{m}^2}{2\rho_{TP} \cdot D_h} \cdot \left(4f_{TP} + \xi \cdot \frac{D_h}{L_{TP}}\right)}{1 + \dot{m}^2 \cdot x \cdot \frac{dv_v}{dp}} \quad (5)$$

with

$$\rho_{TP} = \frac{1}{x \cdot v_v + (1-x) \cdot v_l}, \quad (6)$$

$dv_v/dp$  was evaluated at  $1.3 \times 10^{-7} \text{ m}^4 \text{ s}^2/\text{kg}^2$  with REFPROP 6 and  $4f_{TP} + \xi \cdot D_h/L_{TP}$  is adjusted so that

$$\Delta p_{lo} + \int_{z_{boil}}^{L_j} \frac{dp}{dz} \cdot dz = \Delta p_{mes}. \quad (7)$$

However the two-phase pressure drop and the saturation temperature depend upon the quality and so the fluid pressure, temperature and quality have to be calculated simultaneously. The two-phase length  $L_{TP}$  was divided into 100 sections and the three quantities were calculated iteratively from  $z = z_{boil}$  to  $z = L_j$ . Eqs. (4) and (5) were discretized, iterations started with  $z = z_{boil}$ ,  $x_0 = 0$ ,  $p_0 = p_i - \Delta p_{lo}$ ,  $T_{sat}^0 = T_{sat}(z_{boil})$  and were repeated until the end of the tube was reached. Finally the local heat transfer coefficient was calculated by

$$\alpha(z) = \frac{\dot{q}}{T_w(z) - T_{sat}(z)}. \quad (8)$$

#### 4. Experimental results

The total pressure drop through the test section was measured. Since subcooled fluid enters the tube, it includes the pressure drop of the single-phase liquid flow and the two-phase flow pressure drop. It is well known that wild oscillations of pressure can occur during convective boiling in channels. However, as explained in Section 2, a time averaging method was used in order to reduce these oscillations. For a given mass flow rate and heat flux, measurements were performed every 20 s during about 7 min, so that about 20 measurements were recorded, then average values and standard deviations were calculated. Finally the pressure drop uncertainties reported in Table 2 take into account calibration errors and time oscillations errors by the mean of the standard deviation.

Significant mal-distribution of coolant fluid can occur in multi-channels systems. In order to avoid this effect only subcooled liquid is injected in the inlet manifold. Furthermore the engineering rule that the manifolds diameter should be at last five times higher than the channel hydraulic diameter to equalise the fluid distribution was used. This does not ensure that mal-distribution is totally suppressed but should reduce it within acceptable limits. However even if mal-distribution occurs it will not affect the inlet and outlet measurements which are performed out of the manifolds and it should not affect the local temperature measurements because of the averaging of wall temperatures across the  $N$  channels due to the aluminium very high thermal conductivity.

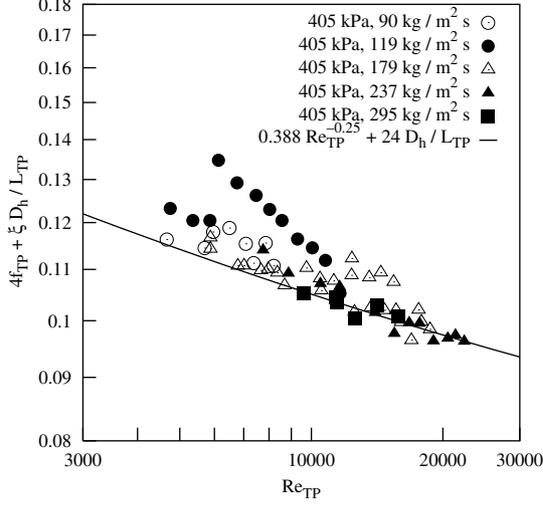


Fig. 2.  $4f_{TP} + \xi \cdot D_h / L_{TP}$  versus  $Re_{TP}$ .

Fig. 2 shows  $4f_{TP} + \xi \cdot D_h / L_{TP}$  calculated with Eq. (7) as a function of the two-phase Reynolds number  $Re_{TP}$  defined by McAdams (1942):

$$Re_{TP} = \frac{\dot{m} \cdot D_h}{\mu_{TP}} \quad (9)$$

$$\frac{1}{\mu_{TP}} = \frac{1 - (x_i + x_o)/2}{\mu_l} + \frac{(x_i + x_o)/2}{\mu_v}. \quad (10)$$

In a previous experimental campaign on liquid flows in this tube by Agostini et al. (2002) it was found that  $4f_l + \xi \cdot D_h / L \approx 0.388 Re^{-0.25} + 21 D_h / L$ . Close results should be obtained replacing the liquid friction factor and Reynolds number by the two-phase friction factor and Reynolds number. Indeed, in Fig. 2, adjusting the singular pressure loss coefficient the trend becomes  $0.388 Re_{TP}^{-0.25} + 24 D_h / L_{TP}$  and is close to measurements.

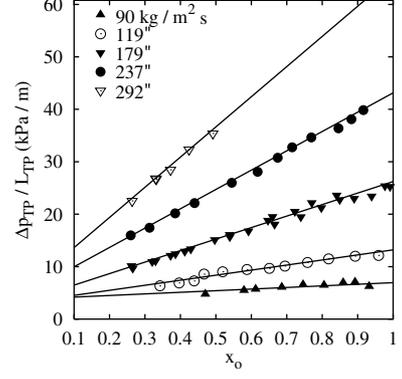


Fig. 3.  $\Delta p_{TP} / L_{TP}$  versus  $x_o$ .

Fig. 3 shows the two-phase pressure drop ( $\Delta p_{mes} - \Delta p_{lo}) / L_{TP}$  as a function of the outlet quality  $x_o$ . The solid lines represent the modelled pressure gradient adjusted with Eq. (7). As shown in Fig. 3, the present measured pressure gradient is linear with  $x_o$ . This is characteristic of preponderant frictional pressure losses. Indeed integration of Eq. (5) for uniform longitudinal heating and constant thermophysical properties and friction factor leads to

$$\begin{aligned} \frac{\Delta p_{TP}}{L_{TP}} &= \frac{\dot{m}^2 \cdot (v_v - v_l)}{L_{TP}} \cdot \left( f_{TP} \cdot \frac{L_{TP}}{D_h} + \frac{\xi}{4} + 1 \right) \cdot x_o \\ &+ \frac{\left( 4f_{TP} + \xi \cdot \frac{D_h}{L_{TP}} \right) \cdot \dot{m}^2}{2\rho_l \cdot D_h} + \frac{g}{(v_v - v_l) \cdot x_o} \\ &\cdot \ln \left( 1 + x_o \cdot \frac{v_v - v_l}{v_l} \right), \end{aligned} \quad (11)$$

where the frictional part of the two-phase flow pressure drop is linear with the outlet quality.

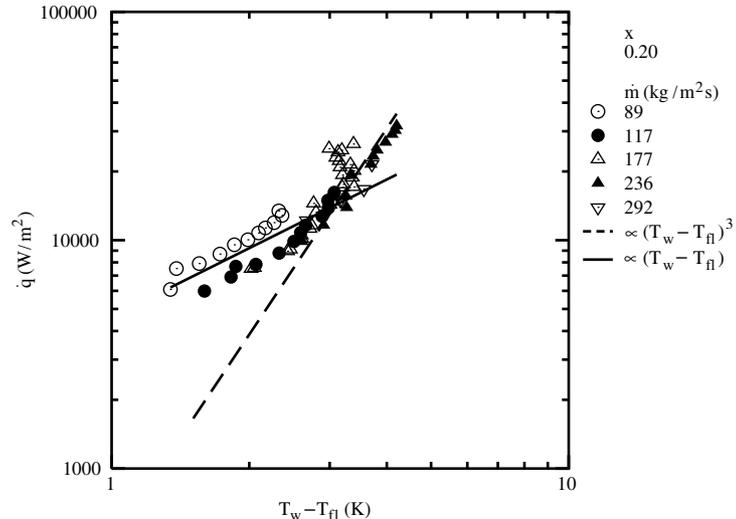


Fig. 4.  $T_w - T_{sat}$  versus  $\dot{q}$  for  $x = 0.2$  ( $p_i = 405$  kPa).

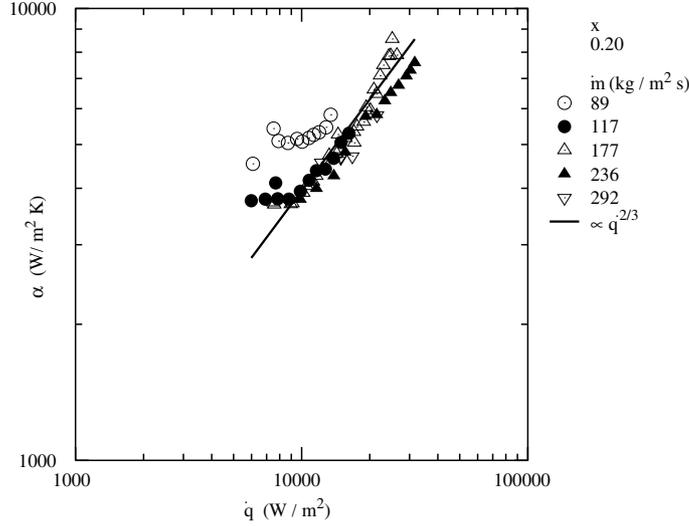


Fig. 5.  $\alpha$  versus  $\dot{q}$  for  $x = 0.2$  ( $p_i = 405$  kPa).

Figs. 4 and 5 exhibit two trends. For  $T_w - T_{sat} \lesssim 3$  K and  $\dot{q} \lesssim 14$  kW/m<sup>2</sup> (this frontier is visible when  $0.1 \leq x \leq 0.3$ ),  $\dot{q}$  is proportional to  $T_w - T_{sat}$ . Thus  $\alpha$  is independent of  $\dot{q}$ , as seen in Fig. 5, and moreover decreases with  $\dot{m}$ . This region may correspond to a convective boiling regime and, as will be further highlighted, the decrease with  $\dot{m}$  may be due to the occurrence of partial dry-out. For  $T_w - T_{sat} \gtrsim 3$  K and  $\dot{q} \gtrsim 14$  kW/m<sup>2</sup>,  $\dot{q}$  is proportional to  $(T_w - T_{sat})^3$ , therefore  $\alpha$  is proportional to  $\dot{q}^{2/3}$ , and the heat transfer coefficient depends only weakly on  $\dot{m}$ . This second region can be identified as a nucleate boiling regime. The transition point is comparable to that found by Tran et al. (1997).

Figs. 6 and 7 show typical results of the local heat transfer coefficient versus the local quality obtained for

$\dot{m} = 117$  kg/m<sup>2</sup> s and different heat fluxes. Three tendencies can be outlined.

For  $Bo \gtrsim 4.3 \times 10^{-4}$  and  $x \lesssim 0.4$  the heat transfer coefficient is weakly dependent on  $x$  and proportional to  $\dot{q}^{2/3}$  as shown in Fig. 5. Thus the nucleate boiling regime governs this region.

For  $Bo \gtrsim 4.3 \times 10^{-4}$  and  $x \gtrsim 0.4$  the heat transfer coefficient decreases with  $x$  but is still proportional to  $\dot{q}^{2/3}$ . This suggests that partial dry-out occurs because of slug bubble confinement thinning down the liquid layer thickness at the tube wall. This is confirmed in Figs. 8 and 9 where the wall temperature and the statistical uncertainty on  $T_w$  suddenly rise for  $x \gtrsim 0.4$ . Moreover the greater the mass flow rate, the more probable dry-out should be, because the liquid film is increasingly dragged from the wall.

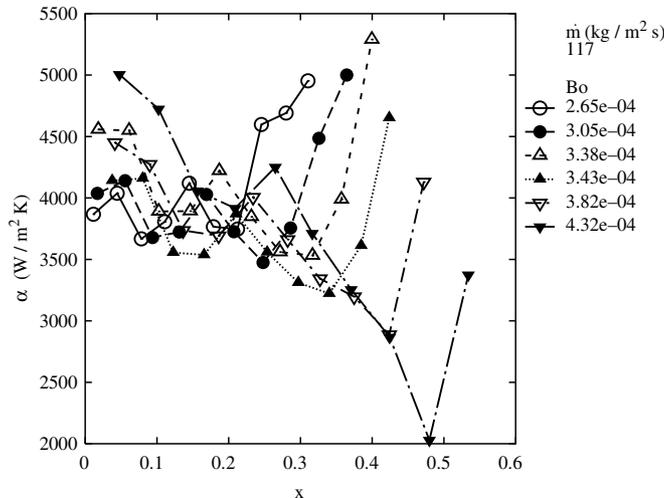


Fig. 6.  $\alpha$  versus  $x$  for  $Bo < 4.3 \times 10^{-4}$  ( $p_i = 405$  kPa).

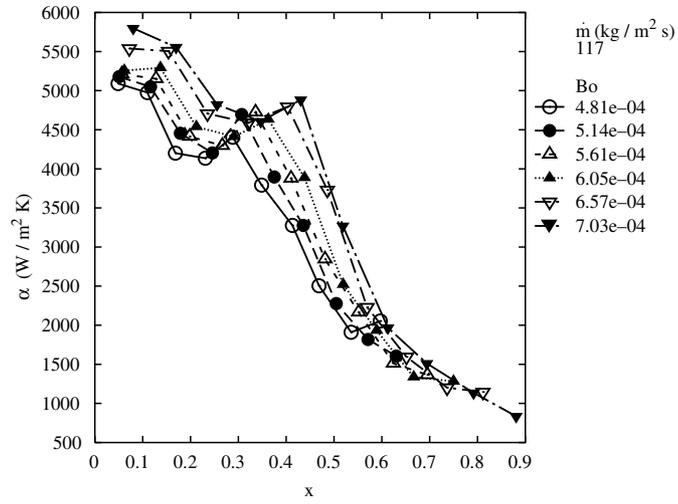


Fig. 7.  $\alpha$  versus  $x$  for  $Bo > 4.3 \times 10^{-4}$  ( $p_i = 405$  kPa).

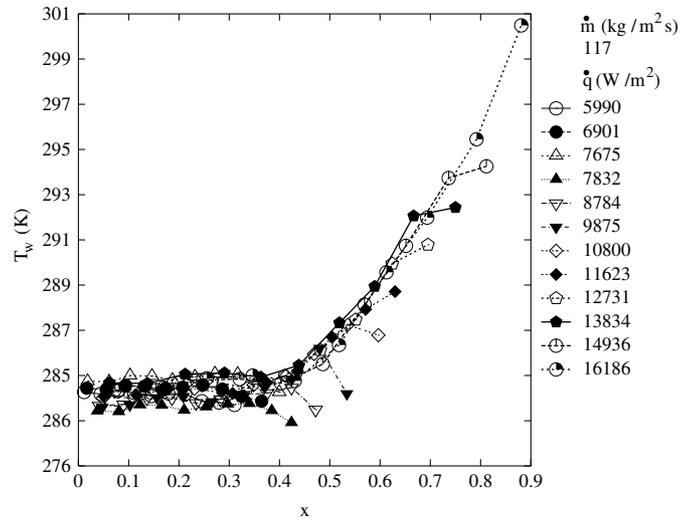


Fig. 8.  $T_w$  versus  $x$  ( $p_i = 405$  kPa).

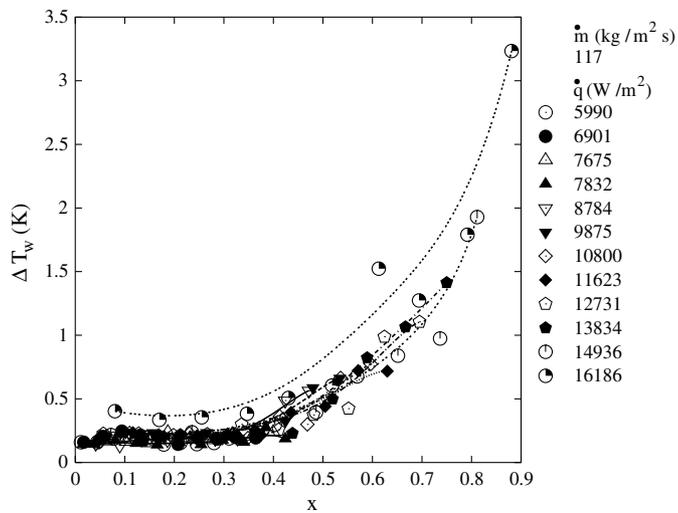


Fig. 9. Uncertainty on  $T_w$  versus  $x$  ( $p_i = 405$  kPa).

Table 3  
Transition product  $Bo \cdot (1-x)$

$\dot{m}$ (kg/m <sup>2</sup> s)	89			117					177			
$\dot{q}$ (kW/m <sup>2</sup> )	6.1	7.5		6.0	6.9	7.7	8.8	9.9	10.8	7.5	8.9	10.2
$Bo \cdot (1-x) \times 10^4$	2.52	2.22		2.16	2.34	2.31	2.25	2.31	2.26	1.9	2.00	2.12

For  $Bo \lesssim 4.3 \times 10^{-4}$  the heat transfer coefficient is weakly dependent on  $x$  and proportional to  $\dot{q}^{2/3}$  for low qualities. It then starts increasing with vapour quality when  $x$  is greater than a transition value (see Fig. 6). This transition value is all the greater since the heat flux is high for a given mass flow rate. This behaviour may correspond to competition between convective boiling and a dry-out regime where partial dry-out and regeneration of the liquid layer occur. Table 3 shows that this transition occurs for a constant value of the product  $Bo \cdot (1-x)$ .

From the Rohsenow (1952) and Kew and Cornwell (1994) analysis, an inertial characteristic time for the liquid layer can be expressed as:

$$\tau_{cv} = \frac{\delta(x)}{u_1} = \frac{\delta(x) \cdot \rho_l}{\dot{m} \cdot (1-x)}, \quad (12)$$

where  $\delta(x)$  is a liquid layer thickness depending on the void fraction and the geometry. The characteristic time for bubbles leaving the wall is

$$\tau_b = \frac{D_b}{u_b}. \quad (13)$$

The bubble diameter is calculated with the Kutaleladze (1981) equation

$$D_b = c_b \cdot \theta \cdot \sqrt{\frac{\sigma}{g \cdot (\rho_l - \rho_g)}}. \quad (14)$$

Rohsenow (1952) defined a bubble Reynolds number based on the velocity of a stream of bubbles leaving a wall and showed that it could be expressed as

$$Re_b = \frac{u_b \cdot D_b}{\nu_l} = c_R \cdot \theta \cdot \frac{\dot{q}}{\mu_l \cdot h_{lv}} \cdot \sqrt{\frac{\sigma}{g \cdot (\rho_l - \rho_g)}}, \quad (15)$$

whence

$$\tau_b = \frac{1}{Re_b} \cdot \frac{D_b^2 \cdot \rho_l}{\mu_l} = \frac{(c_b \cdot \theta)^2}{c_R \cdot \theta} \cdot \frac{h_{lv}}{\dot{q}} \cdot \sqrt{\frac{\sigma}{g \cdot (\rho_l - \rho_g)}} \cdot \rho_l. \quad (16)$$

The ratio of these two characteristic times corresponds to the comparison of convective effects in the liquid layer and bubble dynamics at the wall. This is written:

$$\frac{\tau_{cv}}{\tau_b} = \frac{c_R \cdot \theta \cdot \sqrt{g \cdot (\rho_l - \rho_g)}}{(c_b \cdot \theta)^2 \cdot \sqrt{\sigma}} \cdot Bo \cdot f(x). \quad (17)$$

$c_b$ ,  $c_R$ ,  $\theta$ ,  $\rho_l$ ,  $\rho_g$  and  $\sigma$  depend on the geometry, fluid properties and tube wall characteristics only. The function  $f$  is defined as  $f(x) = (1-x)/\delta(x)$ . Thus  $\tau_{cv}/\tau_b$  is proportional to  $Bo$ . Thus, the boiling number is the appropriate dimensionless number to study the transition between nucleate and convective boiling. In this study  $f(x)$  was found to be close to  $(1-x)$ . The exact expression for  $f(x)$  is difficult to obtain because  $\delta(x)$  depends on the flow pattern and geometrical effects.

Fig. 10 shows that pressure has no sizable influence on the heat transfer coefficient although the nucleate boiling regime is dominant. According to the Cooper correlation for nucleate pool boiling used by Liu and Winterton (1991),  $\alpha$  in the nucleate boiling regime is given by

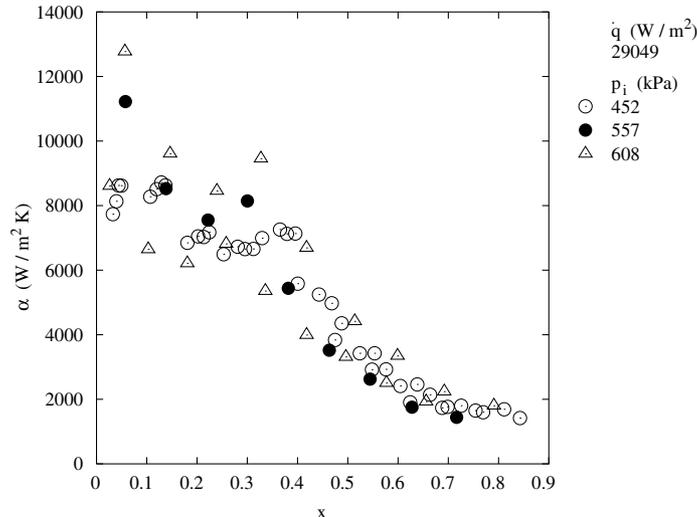


Fig. 10. Influence of pressure ( $\dot{m} = 219 \text{ kg/m}^2\text{s}$ ).

$$\alpha = 55 \cdot p_r^{0.12} \cdot \dot{q}^{0.67} \cdot (-\log_{10}(p_r))^{-0.55} \cdot \tilde{M}^{-0.5}, \quad (18)$$

$\alpha$  should increase by 17% from 4 to 6 bar, which is within the error margins for high heat transfer coefficients but not for lower values for which the uncertainty is  $\pm 6\%$ . A similar trend was observed by Ishibashi and Nishikawa (1969) whose measurements showed that  $\alpha$  was proportional to  $p_r^{0.4}$  in an isolated bubble regime commonly encountered in conventional tubes. It was found to be proportional to  $p_r^{-0.353}$  in a confined bubble regime characteristic of flows in mini-channels.

The following equation was proposed by Rohsenow (1952) to predict the heat transfer coefficient for pool boiling:

$$\frac{c_{p,l} \cdot (T_w - T_{fl})}{h_{lv}} = C_{sf} \cdot \left( \frac{\dot{q}}{\mu_l \cdot h_{lv}} \cdot \sqrt{\frac{\sigma}{g \cdot (\rho_l - \rho_g)}} \right)^{1/3} \cdot Pr_l^{1.7}. \quad (19)$$

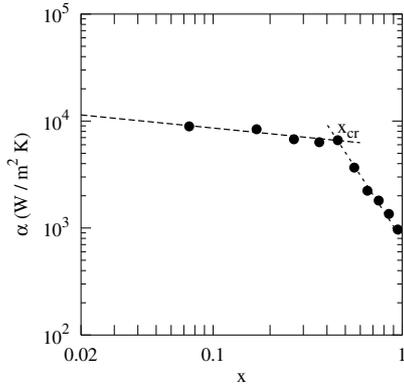


Fig. 11. Determination of the critical quality.

Table 4

Percentage of the present data predicted by various correlations in a  $\pm X\%$  range

Correlation	$\pm 20\%$	$\pm 30\%$
Eq. (20)	83%	95%
Tran et al. (1997)	43%	54%
Kandlikar (2004)	30%	41%
Liu and Winterton (1991)	14%	21%
Shah (1976)	13%	20%
Oh et al. (1998)	5%	7%
Steiner and Taborek (1992)	6%	17%

This equation shows the  $(T_p - T_{fl})^3$  dependence of  $\dot{q}$ . The constant  $C_{sf}$  depends on the nature of the fluid and channel surface. From the 44 data points of the present study in the nucleate boiling regime,  $C_{sf}$  could be estimated for the R134a/extruded aluminium couple as  $0.0034 \pm 15\%$ .

Most of the present data points belong to the nucleate boiling regime (with or without dry-out) so that it has been possible to correlate the heat transfer coefficient in this region with  $\dot{m}$ ,  $\dot{q}$  and  $x$ . First the critical quality above which dry-out occurs when  $Bo \gtrsim 4.3 \times 10^{-4}$  was determined. It was found that  $x_{cr}$  was equal to  $0.43 \pm 0.05$  regardless of the value of  $\dot{q}$  and  $\dot{m}$ . This does not mean that  $x_{cr}$  does not depend on  $\dot{q}$  or  $\dot{m}$  but simply that such a variation is less than the uncertainty. Fig. 11 shows that this value was determined by intersecting two lines issued from a linear regression for every data set. Finally the following expressions are valid for  $90 < \dot{m} < 295 \text{ kg/m}^2 \text{ s}$ ,  $6 < \dot{q} < 31.6 \text{ kW/m}^2$  and  $Bo \gtrsim 4.3 \times 10^{-4}$ :

$$\alpha = 28 \cdot \dot{q}^{2/3} \cdot \dot{m}^{-0.26} \cdot x^{-0.10} \quad \text{if } x < 0.43, \quad (20)$$

$$\alpha = 28 \cdot \dot{q}^{2/3} \cdot \dot{m}^{-0.64} \cdot x^{-2.08} \quad \text{if } x > 0.43.$$

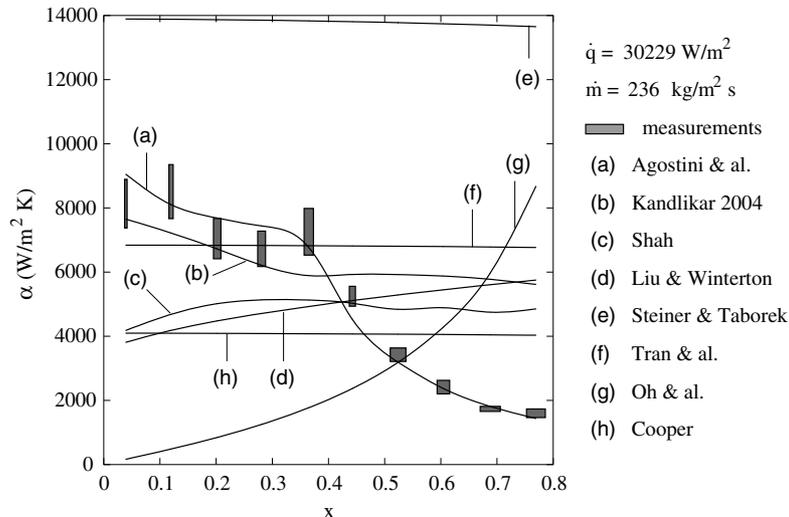


Fig. 12. Comparison of measurements with literature correlations.

Eq. (20), obtained by linear least squares fitting over 715 data points, predicts 83% of our data in the  $\pm 20\%$  range and 95% of our data in the  $\pm 30\%$  range. Table 2 summarises the uncertainties on  $\alpha$ ,  $x$  and  $L_{TP}$ . Table 4 compares the ability of various correlations to predict the present data as shown in Fig. 12. The Tran et al. (1997) and Kandlikar (2004) correlations, proposed for mini-channels predicts the present data rather well in the pure nucleate boiling regime but fails as soon as dry-out occurs. The Steiner and Taborek (1992) correlation over-predicts the present data since it includes a  $D_h^{-0.4}$  diameter correction term which is not well fitted for such small diameters. On the contrary the Shah (1976); Liu and Winterton (1991) correlations under-predict the present data because they do not take into account any confinement phenomenon as suggested by Cornwell and Kew (1992, 1995).

In his new correlation adapted for mini-channels Kandlikar (2004) highlights that conventional correlations were built using turbulent flow heat transfer correlations whereas the liquid flow is most likely laminar in mini-channels. Thus he advises to use Shah and London (1978) or Gnielinski (1976) correlations in his new correlation. However he notices that using the classical Dittus–Boelter turbulent correlation gives the best agreement with the Tran et al. (1997) measurements even with a liquid Reynolds number around 2000. Indeed our own measurements on Fig. 12 are best predicted when using the Dittus–Boelter expression together with the Kandlikar (2004) correlation as noticed by Kandlikar himself. The conclusion of this apparent paradox is that even with liquid Reynolds numbers smaller than 2000 the liquid layer may be very disrupted by growing and detaching bubbles at the wall so that heat transfer through the liquid layer is enhanced compared to a laminar flow.

The measured heat transfer coefficient in the nucleate boiling regime with no dry-out was  $2.75 \pm 0.75$  times higher than that predicted by the previous Kandlikar (1990) correlation for a 10mm diameter tube with the same mass and heat fluxes, giving  $\alpha \propto D_h^{-0.63 \pm 0.17}$ . Ishibashi and Nishikawa (1969) measurements in a confined annular space showed that  $\alpha \propto D_h^{-0.67}$  in the confined bubble regime and  $\alpha \propto D_h^{-0.13}$  in the isolated bubble regime. Aritomi et al. (1993) obtained  $\alpha \propto D_h^{-0.75}$  for  $D_h < 4\text{mm}$  and no  $D_h$  dependence for  $D_h > 4\text{mm}$  in the nucleate boiling regime. The correlation suggested by Tran et al. (1997)

$$Nu = 770 \cdot (Bo \cdot Re_{lo} \cdot Co)^{0.62} \cdot \left( \frac{\rho_v}{\rho_l} \right)^{0.297} \quad (21)$$

gives  $\alpha \propto D_h^{-1}$ . Thus the hydraulic diameter dependence of the two-phase heat transfer coefficient deduced from the present experimental results is supported by literature studies on flow boiling in mini-channels.

## 5. Conclusions

Forced flow boiling heat transfer in mini-channels in similar conditions as encountered in automobile air conditioners has been studied. Higher heat transfer coefficients than in conventional tubes are achieved but dry-out occurs as soon as  $x \gtrsim 0.4$  thus dramatically decreasing performances. These observations support literature studies which predict that bubble confinement leads to higher heat transfer coefficients and dry-out at medium qualities in mini-channels. The homogeneous model was used to calculate the local pressure and predict the saturation temperature. This choice was supported by some literature observations but there is a lack of experimental data in this field. Nucleate boiling was found to be the dominant mechanism for  $\dot{q} > 14\text{kW/m}^2$  and  $\Delta T_{\text{sat}} > 3\text{K}$  which is not far from the conclusions of Tran et al. (1997). The transition from nucleate boiling to supposed convective boiling occurred for  $Bo \cdot (1 - x) \approx 2.2 \times 10^{-4}$  regardless of the heat and mass flux. A 0.77mm hydraulic diameter multi-tube is being tested to verify these conclusions and outline the effects of the confinement on heat transfer.

## References

- Agostini, B., Watel, B., Bontemps, A., Thonon, B., 2002. Friction factor and heat transfer coefficient of R134a liquid flow in mini-channels. *Applied Thermal Engineering* 22 (16), 1821–1834.
- Aritomi, M., Miyata, T., Horiguchi, M., Sudi, S., 1993. Thermohydraulics of boiling two-phase flow in high conversion light water reactors (thermohydraulics at low velocities). *International Journal of Multiphase Flow* 19 (1), 51–63.
- Cornwell, K., Kew, P., 1992. Boiling in small parallel channels. In: *Proceedings of the International Conference on Energy Efficiency in Process Technology*. Elsevier Applied Science, pp. 624–638.
- Cornwell, K., Kew, P., 1995. Evaporation in micro-channel heat exchangers. In: *Proceedings of the 4th U.K. National Conference on Heat Transfer*. ImechE, pp. 289–294.
- Gnielinski, V., 1976. New equations for heat and mass transfer in turbulent pipe and channel flow. *International Chemical Engineering* 16 (2), 359–368.
- Ishibashi, E., Nishikawa, K., 1969. Saturated boiling heat transfer in narrow spaces. *International Journal of Heat and Mass Transfer* 12, 863–894.
- Kandlikar, S., 1990. A general correlation for saturated two-phase flow boiling heat transfer inside horizontal and vertical tubes. *Journal of Heat Transfer* 112 (February), 219–228.
- Kandlikar, S., 2004. An extension of the flow boiling correlation to transition, laminar, and deep laminar flows in minichannels and microchannels. *Heat Transfer Engineering* 25 (3), 86–93.
- Kew, P., Cornwell, K., 1994. Confined bubble flow and boiling in narrow spaces 10th International Heat Transfer Conference, vol. 7. Taylor & Francis, pp. 473–478.
- Kew, P., Cornwell, K., 1997. Correlations for the prediction of boiling heat transfer in small diameter channels. *Applied Thermal Engineering* 17 (8–10), 705–715.
- Kline, S., McClintock, F., 1953. Describing uncertainties in single-sample experiments. *Mechanical Engineering* (January), 3–8.
- Kutaleladze, S., 1981. Principal equations of thermodynamics of nucleate boiling. *Heat Transfer Sovietic Research* 13 (3), 1–14.

- Liu, Z., Winterton, R., 1991. A general correlation for saturated and subcooled flow boiling in tubes and annuli, based on a nucleate pool boiling equation. *International Journal of Heat and Mass Transfer* 34 (11), 2759–2766.
- McAdams, W., 1942. Vaporization inside horizontal tubes—II—benzene-oil mixtures. *Transactions of the ASME* 64, 193.
- Moffat, R., 1982. Contributions to the theory of single-sample uncertainty analysis. *Journal of Fluids Engineering* 104 (June), 250–261.
- Moffat, R., 1985. Using uncertainty analysis in the planning of an experiment. *Journal of Fluids Engineering* 107 (June), 173–182.
- Oh, H., Katsuta, M., Shibata, K., 1998. Heat transfer characteristics of R134a in a capillary tube heat exchanger. In: *Proceedings of 11th IHTC*, vol. 6. pp. 131–136.
- Rohsenow, W., 1952. A method of correlation heat-transfer data for surface boiling of liquids. *Transactions of the ASME* 74, 969–976.
- Shah, M., 1976. A new correlation for heat transfer during boiling flow through pipes. *Transactions of the ASHRAE* 82, 66–86.
- Shah, R., London, A., 1978. *Laminar flow forced convection in ducts*. Academic Press.
- Steiner, D., Taborek, J., 1992. Flow boiling heat transfer in vertical tubes correlated by an asymptotic model. *Heat Transfer Engineering* 13 (2), 43–69.
- Tran, T., Wambsganss, M., France, D., Jendrzejczyk, J., 1993. Boiling heat transfer in a small horizontal, rectangular channel. *Heat Transfer—Atlanta 1993*, vol. 89. AIChE, pp. 253–261.
- Tran, T., Wambsganss, M., France, D., 1996. Small circular and rectangular channel boiling with two refrigerants. *International Journal of Multiphase Flow* 22 (3), 485–498.
- Tran, T., Wambsganss, M., Chyu, M., France, D., 1997. A correlation for nucleate flow boiling in small channels. In: Shah, R.K. (Ed.), *Compact Heat Exchangers for the Process Industries*. Begell House, pp. 353–363.
- Triplett, K., Ghiaasiaan, S., Abdel-Khalik, S., LeMouel, A., McCord, B., 1999. Gas liquid two-phase flow in microchannels. Part I: two-phase flow patterns. *International Journal of Multiphase Flow* 25, 377–394.
- Wambsganss, M., France, D., Jendrzejczyk, J., Tran, T., 1993. Boiling heat transfer in a horizontal small-diameter tube. *Journal of Heat Transfer* 115 (November), 963–972.