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# BOILING HEAT TRANSFER IN MINI-CHANNELS: INFLUENCE OF THE HYDRAULIC DIAMETER

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## ABSTRACT

The present experimental study shows how the reduction of the hydraulic diameter affects the heat transfer coefficient for a boiling flow in mini-channels. Flat aluminium multiport extruded tubes composed of parallel rectangular channels with hydraulic diameters 0.77 and 2.01 mm were used with refrigerant R134a. It was shown that the reduction of the hydraulic diameter increased the heat transfer coefficient and favored dry-out, probably because of the thinning of the liquid layer due to the bubble confinement. However it was found that the quality averaged heat transfer coefficient was still greater than what would be obtained in macro-tubes.

## 1. INTRODUCTION

New environmental policies on global warming require that emissions of gases with significant greenhouse effect should be decreased. The use of mini-channel heat exchangers (hydraulic diameter about 1 mm) contributes to achieve this purpose thanks to higher heat transfer coefficients, thermal efficiency and a lower required fluid mass. They are widely used in condensers for automobile air-conditioning and will be used in evaporators and domestic air-conditioning. However, more general use requires a better understanding of boiling heat transfer in confined spaces. In particular it is necessary to outline the effects of the confinement on heat transfer in mini-channels.

In spite of numerous publications on the experimental study of heat transfer in boiling flows, still few concern the specific study of the influence of the hydraulic diameter on the boiling heat transfer coefficient. Consequently classical correlations of boiling heat transfer coefficient in macro-tubes take into account the effects of the hydraulic diameter rather poorly. The Steiner and Taborek (1992) correlation includes a  $D_h^{-0.4}$  term based on some experimental results for hydraulic diameters as low as 1 mm. Ishibashi and Nishikawa (1969) experimental results within a confined annular space showed that when the bubbles were confined, the heat transfer coefficient was proportional to  $D_h^{-0.67}$ . Aritomi *et al.* (1993) obtained  $D_h^{-0.75}$  for boiling flows in annular spaces with  $D_h < 4$  mm. Tran *et al.* (1997) proposed a correlation for flow boiling heat transfer in mini-channels with a  $D_h^{-1}$  for the confinement effect. Finally, Kew and Cornwell (1994) argued that the confinement should have a sizable influence on the heat transfer coefficient when a new dimensionless number, the confinement number defined as  $Co = (\sigma / (D_h^2 \cdot g \cdot (\rho_l - \rho_v)))^{0.5}$ , is greater than 0.5.

## 2. EXPERIMENTAL APPARATUS

### 2.1 The test loop and test section

The test loop includes a liquid pump (10 – 100 l/h) and a glycol-water mixture circuit for heat evacuation. Subcooled liquid enters the inlet manifold, then is vaporised in the test section and condensed in a heat exchanger by the water-glycol flow. The test section, shown in figure 1, consists of two different flat tubes. The inlet and outlet manifolds are 10 mm diameter tubes in a U pattern at 90 degrees. The manifold diameter is five to thirteen times higher than the channel hydraulic diameter to limit fluid distribution effects. The whole test section is thermally insulated with 40 mm thick wrapping foam. For heat transfer measurements the length  $L_j$  of the tube (zone ②) was heated thanks to Joule effect by passing an electric current (up to 2 800 A) from two brazed electrodes through the tube wall. Experimental conditions are summarized in table 1.

Figure 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 temperatures (respectively  $T_{fi,i}$  and  $T_{fi,o}$ ) 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. Thanks to the aluminium high thermal conductivity and the tube walls low thickness (0.35 mm) the measured temperature is equal to the wall temperature in contact with the fluid (the difference was estimated less than 0.01 K). The fluid

Table 1: Operating conditions and uncertainties

|                                 | tube 1    |             | tube 2    |             |
|---------------------------------|-----------|-------------|-----------|-------------|
| $\dot{m}$ (kg/m <sup>2</sup> s) | 90–295    | ± 1.7–8.6 % | 214–469   | ± 3.3–8.6 % |
| $\dot{q}$ (kW/m <sup>2</sup> )  | 6–31.6    | ± 2–4.1 %   | 2.8–19.5  | ± 3 %       |
| $T_w, T_H$ (K)                  | 276–308   | ± 0.2–3 K   | 281–307   | ± 0.2–2 K   |
| $\Delta T_{\text{sub},i}$ (K)   | 1–17      | ± 0.1 K     | 1–8       | ± 0.1 K     |
| $p_i$ (kPa)                     | 405       | ± 3.4 %     | 517       | ± 2.7 %     |
| $\Delta p$ (kPa)                | 9.5–37.5  | ± 1.2–17 %  | 153–1570  | ± 0.3–3.7 % |
| $\alpha$ (kW/m <sup>2</sup> K)  | 0.8–10.3  | ± 6–30 %    | 0.6–32.5  | ± 5–30 %    |
| $x_o$                           | 0.26–1    | ± 1–7 %     | 0.23–0.9  | ± 2–9 %     |
| $Co$                            | 0.46      |             | 1.15      |             |
| flow                            | ascendant |             | ascendant |             |

inlet pressure was measured with a calibrated Rosemount type II absolute pressure sensor. Three differential pressure sensors calibrated from 0 to 7.6, 40.5 and 182 kPa measure 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.

## 2.2 The channel geometry

The determination of the channel dimensions has been carried out using scanning electron microscopy. The dimensions of several channels were measured and the average 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 ( $S_{fl}$ ) and wet perimeter ( $P_{fl}$ ) measured from electron microscopy imaging in order to take into account the effect of the first and last channels, and the corners which are rounded, yielding  $D_h = 4S_{fl}/P_{fl}$ .

Table 2: Geometric dimensions measurements.

| tube | $\bar{b}$ [mm] | $\bar{a}$ [mm] | $\bar{D}_h$ [mm] | $\gamma$ | $N$ | $L$ [mm] | $W$ [mm] | $e$ [mm] |
|------|----------------|----------------|------------------|----------|-----|----------|----------|----------|
| 1    | 3.28           | 1.47           | 2.01             | 0.45     | 11  | 690      | 40       | 0.35     |
|      | ±0.02          | ±0.02          | ±0.06            | ±0.01    | -   | ±5       | -        | ±0.03    |
| 2    | 0.73           | 0.72           | 0.77             | 0.98     | 18  | 695      | 18       | 0.25     |
|      | ±0.01          | ±0.01          | ±0.054           | ±0.03    | -   | ±2.5     | -        | ±0.03    |

## 3. OPERATING CONDITIONS AND PROCEDURE

Steady state values were monitored thanks to the Hewlett Packard 3421A multiplexer with a 30 minutes time lapse between each change in mass flow rate or heat flux. An averaging occurred after every 20 values and uncertainties were calculated according to Kline and McClintock (1953) and Moffat (1982, 1985) methods, with twice the standard deviation. For every fixed mass flow rate, the heat flux was adjusted in order to obtain series of outlet vapor qualities between 0.2 and 1 with a step of 0.05 approximatively. At the same time the pressure was kept constant by adjusting the quantity of fluid in the test loop for each heat flux and mass flow rate variation. 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 6.01 software. Uncertainties are reported in table 1.

#### 4. DATA REDUCTION

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

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

The temperature of the subcooled fluid absorbing the power  $\dot{Q}(z)$  is calculated with a heat 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 equation is implicit with  $T_{fl}(z)$  because  $c_{p,l}(z)$  is a function of  $T_{fl}(z)$  so that it must be solved iteratively. The tube coordinate  $z_{boil}$  where bulk boiling starts is the position along the tube where the fluid bulk temperature is equal to the saturation temperature at the local pressure, since liquid pressure losses cannot be neglected in mini-channels. Therefore both the temperature and the pressure values must be iterated to determine  $z_{boil}$  because the single-phase pressure loss is a function of  $z_{boil}$ .

In order to calculate the vapor quality, the local pressure and the fluid temperature, a heat balance was used over the length  $dz$  giving

$$\delta \dot{Q} = \dot{M} \cdot [x \cdot c_{p,v}(T_{sat}) \cdot dT_{sat} + (1-x) \cdot c_{p,l}(T_{sat}) \cdot dT_{sat} + h_{lv}(T_{sat} + dT_{sat}) \cdot dx] \quad (3)$$

The measured total pressure drop through the test section includes the pressure drop of the single-phase liquid flow ( $\Delta p_{l0}$ ), since subcooled fluid enters the tube, and the two-phase flow pressure drop. Figure 2 shows the average pressure drop gradient  $(\Delta p_{mes} - \Delta p_{l0})/L_{TP}$ , with  $L_{TP} = L - z_{boil}$ , as a function of the outlet quality  $x_o$ .

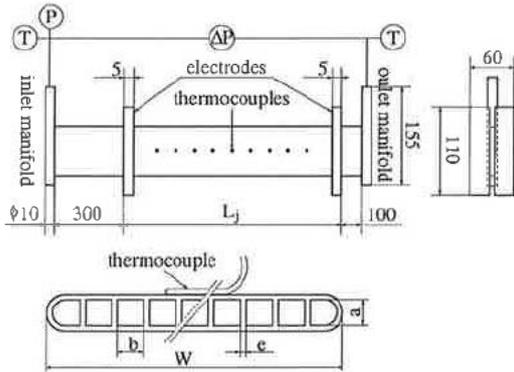


Figure 1: The test section.

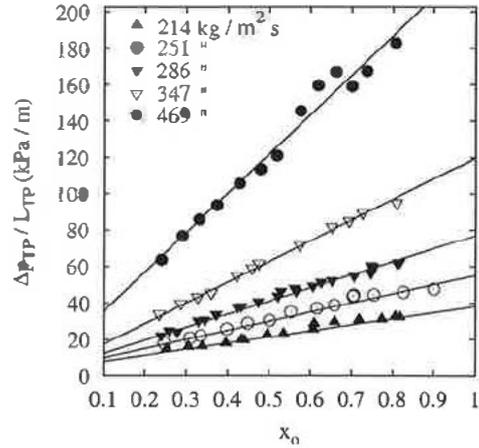


Figure 2: Measured average pressure drop gradient,  $D_h = 0.77$  mm

The homogeneous model was chosen to model the local pressure because this model predicts that  $(\Delta p_{mes} - \Delta p_{l0})/L_{TP}$  is linear with  $x_o$  when the gravity and acceleration pressure losses are negligible. This is shown on figure 2 where the lines represent the predictions of the homogeneous model.

The two-phase pressure drop and the saturation temperature depend on the vapor quality so that the local pressure, fluid temperature and quality must be calculated simultaneously with an iterative method. The details of these calculations are given by Agostini (2002).

Finally the heat transfer coefficient is calculated by

$$h(z) = \frac{\dot{q}}{T_w(z) - T_{sat}(z)} \quad (4)$$

## 5. EXPERIMENTAL RESULTS

Figures 3 and 4 show the local heat transfer coefficient as a function of the local quality for one given mass flow rate and several heat fluxes. The trends are the same for the other tested mass flow rates. The heat transfer coefficient does not depend on the heat flux given the uncertainties. For both tubes there exists a critical quality beyond which the heat transfer coefficient decreases sharply with increasing vapor quality and this critical quality is smaller for the smallest hydraulic diameter.

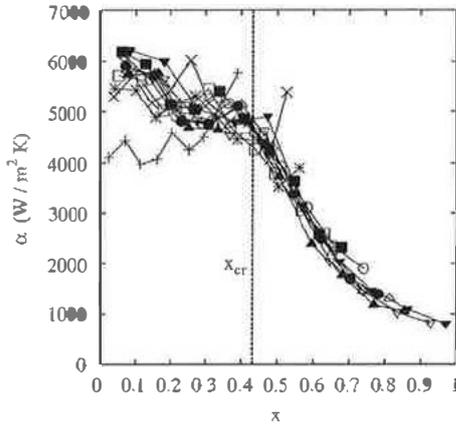


Figure 3: Local heat transfer coefficient versus local quality,  $D_h = 2.01$  mm

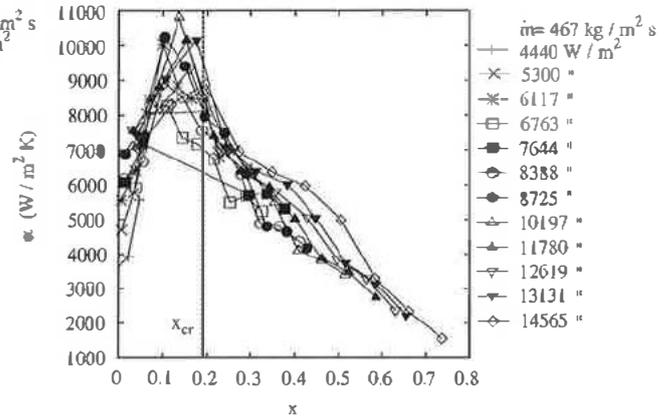


Figure 4: Local heat transfer coefficient versus local quality,  $D_h = 0.77$  mm

This critical quality was defined as the intersection of lines fitted by linear regressions when the heat transfer coefficient is plotted as a function of the local quality. This method led to  $x_{cr} = 0.43 \pm 14\%$  for tube 1 and  $x_{cr} = 0.19 \pm 26\%$  for tube 2. The critical quality was found to be independent of the heat flux and mass flow rate in the tested range. It doesn't mean that there is no such dependence but rather that if  $x_{cr}$  depends on  $\dot{q}$  and  $\dot{m}$ , this variation does not appear because of the uncertainties.

The physical interpretation of this critical quality is easier when studying the local wall temperatures and their statistical uncertainties as a function of the local quality. The trends are the same for both tubes. In the case of tube 1 for example, for qualities lesser than the critical quality both the local wall temperatures and their uncertainties are constant as expected. Furthermore  $\Delta T_w$  is about 0.2–0.3 K which is an acceptable uncertainty for temperature measurements with a boiling flow. On the contrary beyond  $x_{cr}$ ,  $T_w$  and  $\Delta T_w$  increase steadily with  $x$  to the point that the uncertainty is multiplied by 10 at high qualities.

These phenomena are consistent with a partial dry-out of the liquid layer at the internal wall. The following interpretation is proposed. As the vapor quality increases for a given heat flux, the thickness of the liquid layer decreases so that dry-out occurs more easily. Below a critical thickness, corresponding to a critical quality, a periodic dry-out alternates with a regeneration of the liquid layer. Since the wall is periodically in contact with liquid or gas, the heat transfer undergoes important fluctuations, because of the bad heat transfer with gas. These fluctuations are all the more important that the vapor quality is high, because a high quality means more frequent dry-outs. Finally these heat transfer coefficient fluctuations cause the time averaged wall temperature to increase and fluctuate with quality, so that  $\Delta T_w$  increases with  $x$ .

Cornwell and Kew (1992, 1995) proposed that a new flow configuration, the confined bubble regime, is dominant in mini-channels where the bubbles are confined between the channels walls. In such a regime, these confined bubbles thin the liquid layer, all the more that the confinement is important. Furthermore the liquid layer should be regenerated between two successive passing bubbles. This model could explain why the critical quality decreases with the hydraulic diameter because of the liquid layer breakdown under the confined bubbles.

In order to compare the heat transfer coefficients in tubes 1 and 2, data points must be carefully selected. For a proper comparison, measurements with very close heat fluxes, mass flow rates, vapor qualities and boiling regimes for both tubes must be used. Ideally the working pressure should also be the same, but pressure has no significant

influence on the heat transfer coefficient in the convective boiling regime and the pressure difference between tube 1 and 2 is not that important ( $\pm 12\%$  of the average pressure). Furthermore Agostini *et al.* (2002) measurements on tube 1 showed that between 417 and 604 kPa the pressure had no sizable influence on the heat transfer coefficient, even in the nucleate boiling regime. Among about 750 data points available for each tube, only 8 had such characteristics.

As illustrated on figures 5 and 6 the heat transfer coefficient is higher in tube 2 before any dry-out occurs. The enhancement ratio is constant for the 8 data points given the uncertainties. The average enhancement ratio is  $\alpha_{T2}/\alpha_{T1} = 1.74 \pm 0.31$ , that is  $(D_{h,T2}/D_{h,T1})^{-0.6 \pm 0.1}$  which is, given the uncertainties, close to the values proposed by Ishibashi and Nishikawa (1969) and Aritomi *et al.* (1993). It is not clear how Tran *et al.* (1997) established the term  $D_h^{-1}$  because the range of tested hydraulic diameters, 2.4-2.92 mm, is too small to build a correlation. It is probable from their articles that it comes from an analogy with single-phase flow classical theory and has no experimental basis. On the contrary the  $D_h^{-0.4}$  term proposed by the Steiner and Taborek (1992) is based on some experimental data. Nevertheless the data concerning refrigerant fluids (R11 and R113) were performed for hydraulic diameters from 7 to 20 mm only. The lowest hydraulic diameters (from 1 to 5 mm) were tested with Helium I only so that this  $D_h^{-0.4}$  factor is difficult to compare with the present results.

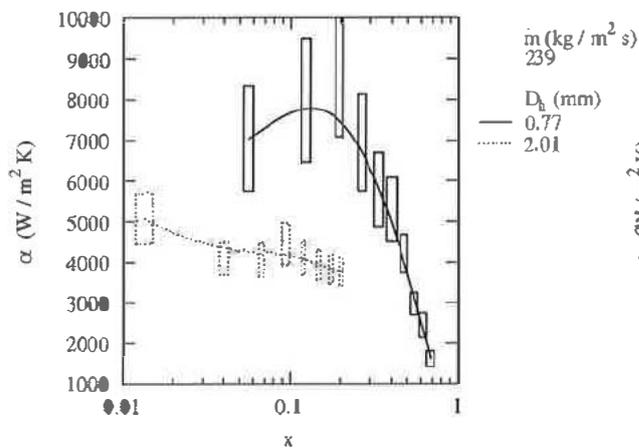


Figure 5: Influence of the confinement on the heat transfer coefficient,  $\dot{m} = 239 \text{ kg/m}^2 \text{ s}$

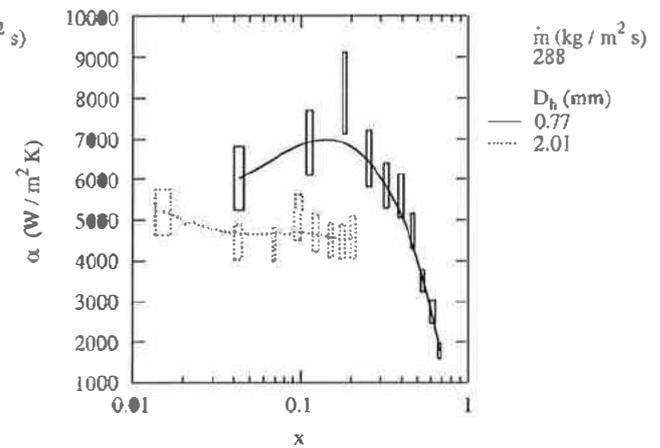


Figure 6: Influence of the confinement on the heat transfer coefficient,  $\dot{m} = 288 \text{ kg/m}^2 \text{ s}$

Although the confinement increases the heat transfer coefficient, it was shown that dry-out had the opposite effect and was dominant in mini-channels. Thus it is possible that the average heat transfer coefficient over the boiling length is not increased compared to what would be measured in a macro-tube. In order to answer this question this average experimental heat transfer coefficient divided by that predicted by Shah (1982) in a 10 mm diameter tube was represented as a function of the tested heat fluxes on figures 7 and 8. The conclusion is that even at high heat fluxes, that is high vapor qualities and important dry-out, the average heat transfer coefficient in mini-channels remains higher than what would be obtained in a macro-tube. Thus the dominant dry-out observed in mini-channels does not completely compensate for the enhancement effect due to the confinement.

## 6. CONCLUSIONS

The influence of the hydraulic diameter on the heat transfer coefficient in mini-channels was experimentally studied. It was found that partial dry-out occurred in mini-channels at lower vapor qualities than in macro-tubes. When the hydraulic diameter decreases from 2.01 to 0.77 mm, the critical quality decreases from 0.43 to 0.19. It was also experimentally shown that when the hydraulic diameter decreases from 2.01 to 0.77 mm, the heat transfer coefficient is multiplied by 1.74 approximately. However this enhancement is somehow balanced by the early occurrence of dry-out. Nevertheless the heat transfer coefficient averaged on the whole boiling length remains higher than what would be obtained in a macro-tube. As a conclusion the decrease of the hydraulic diameter is an effective means of

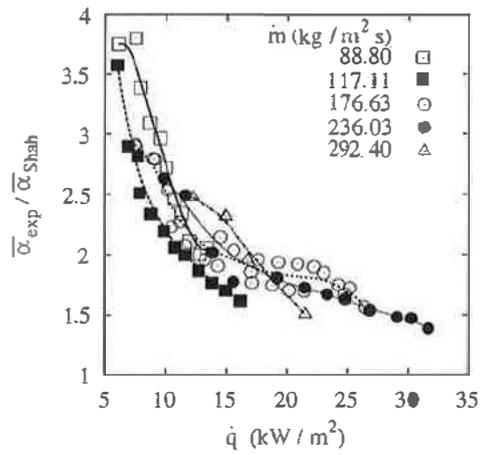


Figure 7: Comparison of the average heat transfer coefficient and that predicted by the Shah (1982) correlation,  $D_h = 2.01$  mm

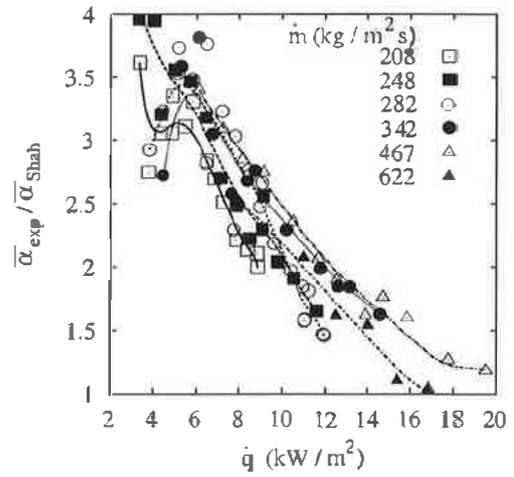


Figure 8: Comparison of the average heat transfer coefficient and that predicted by the Shah (1982) correlation,  $D_h = 0.77$  mm

increasing the heat transfer coefficient in boiling flows. A more thorough study of geometric effects should allow a better understanding of the early dry-out in mini-channels.

## 7. NOMENCLATURE

### Latin letters

|                            |                             |                       |
|----------------------------|-----------------------------|-----------------------|
| $a$                        | channel height              | (m)                   |
| $b$                        | channel width               | (m)                   |
| $c_p$                      | specific heat capacity      | (J/kg K)              |
| $D_h$                      | hydraulic diameter          | (m)                   |
| $g$                        | gravity acceleration        | (m/s <sup>2</sup> )   |
| $h$                        | specific enthalpy           | (J/kg)                |
| $h_{lv}$                   | latent heat of vaporization | (J/kg)                |
| $L_j$                      | tube heated length          | (m)                   |
| $L_a$                      | adiabatic tube length       | (m)                   |
| $\dot{M}$                  | mass flow rate              | (kg/s)                |
| $\dot{m}$                  | mass flux                   | (kg/m <sup>2</sup> s) |
| $N$                        | number of channels          |                       |
| $P_{\text{fl}}$            | total wet perimeter         | (m)                   |
| $p$                        | pressure                    | (bar)                 |
| $\Delta p$                 | pressure loss               | (bar)                 |
| $\dot{Q}$                  | thermal power               | (W)                   |
| $\dot{q}$                  | heat flux                   | (W/m <sup>2</sup> )   |
| $S_{\text{fl}}$            | total wet area              | (m <sup>2</sup> )     |
| $\nu$                      | specific volume             | (m <sup>3</sup> /kg)  |
| $x$                        | quality                     |                       |
| $z$                        | coordinate                  | (m)                   |
| $S$                        | internal heated surface     | (m <sup>2</sup> )     |
| $T$                        | temperature                 | (K)                   |
| $\Delta T_{\text{sub,in}}$ | inlet subcooling            | (K)                   |
| $W$                        | test section width          | (m)                   |

### Greek letters

|                |                           |                       |
|----------------|---------------------------|-----------------------|
| $\alpha$       | heat transfer coefficient | (W/m <sup>2</sup> K)  |
| $\gamma = a/b$ | aspect ratio              |                       |
| $\rho$         | density                   | (kg/m <sup>3</sup> )  |
| $\mu$          | dynamic viscosity         | (N s/m <sup>2</sup> ) |
| $\sigma$       | surface tension           | (N/m)                 |

### Subscripts

|      |                 |
|------|-----------------|
| boil | boiling point   |
| cr   | critical        |
| in   | inlet           |
| fl   | fluid           |
| l    | liquid          |
| lo   | all liquid flow |
| mes  | measured        |
| out  | outlet          |
| sat  | saturation      |
| TP   | two-phase flow  |
| v    | vapor           |
| w    | wall            |

### Dimensionless numbers

|      |                    |
|------|--------------------|
| $Co$ | confinement number |
| $4f$ | friction factor    |

## REFERENCES

- Agostini, B., 2002. Étude expérimentale de l'ébullition en convection forcée de fluide réfrigérant dans des mini-canaux. Ph.D. thesis, Université Joseph Fourier - Grenoble I.
- Agostini, B., Watel, B., Bontemps, A., Thonon, B., 2002. Ascendant flow boiling of refrigerant R134a in small channels. *Submitted to International Journal of Heat and Fluid Flow*.
- 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*, vol. 19, no. 1: pp. 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. A., 1995. Evaporation in micro-channel heat exchangers. In *Proceedings of the 4<sup>th</sup> U.K. National Conference on Heat Transfer*. ImechE, pp. 289–294.
- Ishibashi, E., Nishikawa, K., 1969. Saturated boiling heat transfer in narrow spaces. *International Journal of Heat and Mass Transfer*, vol. 12: pp. 863–894.

- Kew, P. A., Cornwell, K., 1994. Confined bubble flow and boiling in narrow spaces. In *10<sup>th</sup> International Heat Transfer Conference*. Taylor & Francis, vol. 7, pp. 473–478.
- Kline, S., McClintock, F., 1953. Describing uncertainties in single-sample experiments. *Mechanical Engineering*: pp. 3–8.
- Moffat, R., 1982. Contributions to the theory of single-sample uncertainty analysis. *Journal of Fluids Engineering*, vol. 104: pp. 250–261.
- Moffat, R., 1985. Using uncertainty analysis in the planning of an experiment. *Journal of Fluids Engineering*, vol. 107: pp. 173–182.
- Shah, M., 1982. Chart correlation for saturated boiling heat transfer : equations and further study. *ASHRAE Transactions*, vol. 88: pp. 185–196.
- Steiner, D., Taborek, J., 1992. Flow boiling heat transfer in vertical tubes correlated by an asymptotic model. *Heat Transfer Engineering*, vol. 13, no. 2: pp. 43–69.
- Tran, T., Wambsganss, M., Chyu, M., France, D., 1997. A correlation for nucleate flow boiling in small channels. In R. K. Shah, editor, *Compact Heat Exchangers for the Process Industries*. Begell House, inc., pp. 353–363.

#### TRANSFERTS DE CHALEUR EN ÉBULLITION DANS LES MINI-CANAUX : INFLUENCE DU DIAMÈTRE HYDRAULIQUE

**RÉSUMÉ** : Cette étude expérimentale montre comment la réduction du diamètre hydraulique influe sur le coefficient d'échange de chaleur lors de l'ébullition en convection forcée d'un fluide réfrigérant dans des mini-canaux. Des tubes plats à canaux parallèles de diamètres hydrauliques 2,01 et 0,77 ont été étudiés avec du réfrigérant R134a. Nos mesures ont mis en évidence que la réduction du diamètre hydraulique augmentait le coefficient d'échange de chaleur et favorisait l'assèchement, probablement à cause de l'amincissement de la couche liquide dû au confinement des bulles. Cependant nous avons montré que le coefficient d'échange moyenné le long du tube demeure supérieur à celui qui serait obtenu dans des macro-tubes.