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CRITICAL HEAT FLUX FOR WATER FLOWING INSIDE A 0.38 MM TUBE

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Abstract: *This paper presents critical heat flux results for water flowing inside a horizontal tube with 0.38 mm internal diameter. The experimental campaign was performed for mass velocities ranging from 100 to 6441 kg/m^{2}s, heat fluxes up to 1.2 MW/m^2, exit pressure of 92 kPa, subcooling of 70 ^{o}C and heated length of 70 mm. The test section was heated by joule effect by applying direct current along the tube wall using a controlled power supply. The results were compared against predictions methods from literature. Critical heat flux increased with mass flux in a quite similar way to the results observed in macroscale literature. For this tube diameter the effect of two-phase flow instabilities were very important and the experimental critical heat flux results were half of the expected by prediction methods due the presence of flow instabilities.*

Keywords: : *critical heat flux, microchannel, water, flow boiling, heat transfer.*

1. INTRODUCTION

Critical heat flux (CHF) or burnout is generally related to a drastic decrease in the heat transfer coefficient during single phase or flow boiling implying in a very high temperature increment. When heat is dissipated from a device whose imposed parameter is the heat flux (as in electric/electronics components and nuclear reactors) exceeding the CHF may result in an irreversible damage of the thermal device. Consequently, CHF is the maximum operational heat flux that can be achieved under safe operation during convection. Due to such a fact, this topic has attracted great attention of the academic society dealing with heat transfer and also of the industrial sector involved with the dissipation of high heat flux densities. The development of compact heat exchangers has lead researchers to investigate CHF inside small tube diameters (Tibiriçá et al., 2012a,b).

Lowdermilk et al. (1958) was one of the first authors to publish CHF for smaller diameter tubes flowing with water. They performed experiments for vertical tubes between 1.3 – 4.7 mm, using water at saturation temperatures up to 165^{o}C, heated length between 64-955 mm, mass velocities of 27-3320 kg/m^{2}s, achieving a CHF up to 10.8 MW/m^2.

Vandervort et al. (1994) investigated CHF for water in vertical small diameter tubes. Their experiments were performed with metallic tubes having inside diameters ranging from 0.3 to 2.7 mm. Mass fluxes ranged from 5000 to 40 000 kg/m^{2}s and exit subcoolings from 40 to 130^{o}C. Exit pressures ranged from 0.2 to 2.2 MPa, and length-to diameter ratios ranged from 2 to 50. Over 200 CHF stable data points were obtained and a CHF up to 123 MW/m^2 was achieved CHF was shown to be an increasing function of both mass flux and subcooling, and an inverse function of diameter. CHF increased for length-to-diameter ratios less than 10, and decreased with increasing exit pressure.

Mudawar and Bower (1999) obtained CHF experimental results for water flowing in a vertical short small diameter tubes. The data included tube diameter from 0.406 to 2.54 mm, mass velocities from 5000 to 134000 kg/m^{2}s, inlet temperature between 18-70 ^{o}C, outlet pressure 2.5-172 bars and CHF values up to 273 MW/m^2. CHF increased with increasing mass velocity increasing subcooling decreasing tube diameter and decreasing heated length to diameter ratio. For a constant inlet temperature CHF increased with increasing pressure for pressures up to 29 bars remained fairly constant between 29 and 049 bars and decreased afterwards as the critical pressure was approached CHF was accompanied by physical burnout of the tube wall near the exit and tube material had little effect on the magnitude of CHF.

Sumith et al. (2003) studied the CHF in a 1.45 mm vertical tube flowing with water at mass velocities between 23-150 kg/m^{2}s, heated length of 100 mm, saturation temperature of 100 ^{o}C obtained critical heat fluxes between 145-723 kW/m^2. They could predict their experimental data with 20% average error using Katto (1980) correlation.

Zhang et al (2006) evaluated critical heat flux correlations against literature database for water in mini- channels. They concluded that the correlations of Hall and Mudawar (2000)] and Shah (1987) obtained good predictions for subcooled and saturated CHF respectively. Tibiriçá et al. (2008) performed a wide CHF comparison using different critical heat flux correlations and fluids and concluded that Katto and Ohno (1984) correlation worked well for saturated critical heat flux.

In this context, this work presents new critical heat flux experiments for water in a microchannel of 0.38 mm, analyzing the mass flow effect in the critical heat flux, since there is a lack of results for single channel micro-scale flow in horizontal orientation.

2. EXPERIMENTAL APPARATUS

The experimental facility was specially built for these experiments. It consists of an open loop system, with the exit of the test section opened to the atmosphere. Water was used and pumped from a syringe to the test section. As indicated by Vanderport (1994) and Cioncolini (2007) the effect of dissolved air in critical heat flux is small and this parameter was not evaluated in this experimental study. The test section consists of a circular 0.38 mm internal diameter tube, of stainless steel 304, 100 mm length. The experimental campaign was performed for mass velocities ranging from 100 to 6441 kg/m^2s, critical heat fluxes up to 1.2 MW/m^2, exit pressure of 92 kPa, subcooling of 70 ^oC and heated length of 70 mm. The test section was heated by joule effect by applying direct current along the tube wall using a controlled power supply. The heat flux over the tube surface was calculated dividing the applied electrical power by the internal tube surface area. Mass velocity was measured by measuring the volume of syringe over time and also by measuring the mass which left the test section in a precise scale. Single phase pressured drop was also used for measurement of the mass flow using a pressure drop section. The test section was thermally insulated from the surrounding by insulation foam, with 0.03 W/m.K thermal conductivity. Surface roughness measurements in the test section obtained by an optical surface profilometer indicated roughness average (RA) of 1.3 µm. Inlet and outlet pressure in the test section were measured with two calibrated pressure transducers.

The experimental procedure used for critical heat flux measurements were: *i*) fill the syringe with water; *ii*) impose pressure to the syringe to promote flow, *iii*) turn on the power supply; *iv*) increase power applied to the tube in small steps up to a point where the small thermocouples over the tube surface detect suddenly temperature increment. A data acquisition system saved all parameters (temperature, pressure, mass velocity) during the experimental procedure.

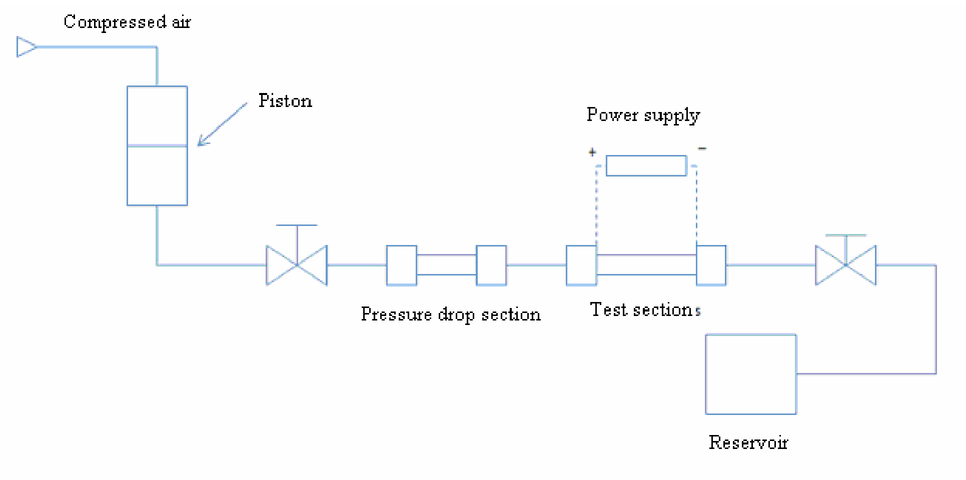

Figure 1. Schematic of the test facility.

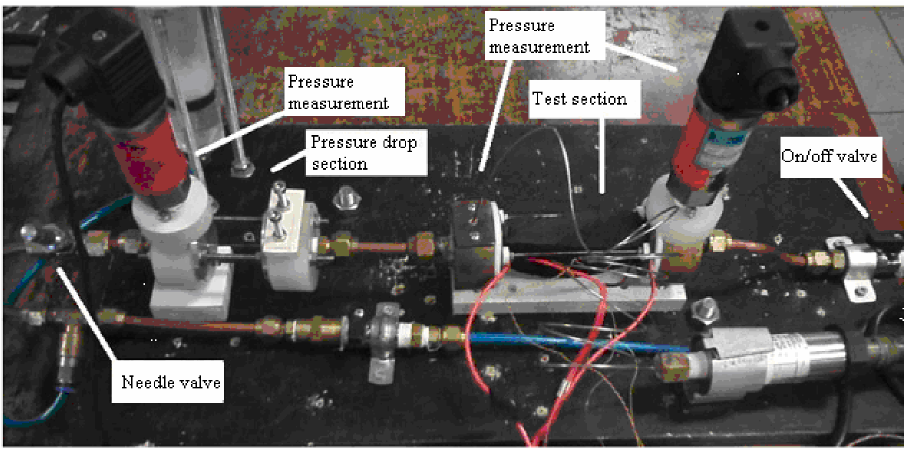

Figure 2 Photograph of the test facility

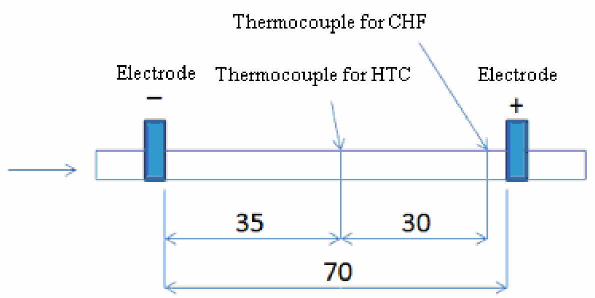

Figure 3. Test section details with thermocouple positions.

The mass velocity is the mass flow ratio divided by the internal cross section area of the tube:

$$G = \frac{4\dot{m}}{\pi {D_i}^2} \tag{1}$$

The heat flux is the ratio of the applied electrical power to the test section divided by the internal surface area of the test section. Heat losses were not account in this equation.

$$q = \frac{P}{A_s} \tag{2}$$

The heat transfer coefficient is given by the Newton law of cooling:

$$h = \frac{q}{\Delta T} \tag{3}$$

In a specific length z, along the test section the heat transfer coefficient is calculated as:

$$h(z) = \frac{q}{T_{pi}(z) - T_{sat}(z)} \tag{4}$$

3. SINGLE PHASE VALIDATION

From the presented data in Table 1, the Nusselt number for each experimental condition were calculated and compared against the prediction models. These results are relative to the thermocouple positioned in the middle of the test section.

Table 1. Experimental data for single phase flow

Ten (°C)	Reynolds	Fluxo de calor (W/m²)	Tcentro (°C)
21.5	303.2	78263	42.7
21.7	426.8	103047	44.8
21.4	535.5	127525	46.9
20.6	618	126606	43.7
20.4	815.2	155625	45.8
24.5	1448	201277	49.6
24.3	1870	192129	46.9
24.4	2163	220211	47.2
24.5	2385	239996	47.5
24.7	2809	293180	39.9
24.8	3314	338056	37.6

For developed laminar flow in circular channels the Nusselt number is 4.36. For laminar developing flow two correlations were used for prediction. The Shah and London (1978) correlation given by equation 5 and Churchil and Ozoe (1973) correlation given by Eq. 7.

Shah and London (1978) correlation:

$$Nu = 4{,}364 + 0{,}263.\,Gz^{0.506}.\,e^{-41/Gz} \tag{5}$$

where Gz is the Graetz number:

$$Gz = \frac{D}{L}.\,Re.\,Pr \tag{6}$$

Churchil and Ozoe (1973) correlation:

$$Nu = 4{,}364.\left(+ \left(\frac{\pi}{4}.\,Gz.\,\frac{1}{29{,}6}\right)^{2}\right)^{\frac{1}{6}}.\left(+ \left(\frac{\frac{\pi}{4}.\frac{Gz}{19{,}4}}{\left(1+\left(\frac{Pr}{0{,}0207}\right)^{\frac{2}{3}}\right)^{0.5}}.\left(1 + (\frac{\pi}{4}.\,Gz.\,\frac{1}{29{,}6})^{2}\right)^{\frac{1}{3}} \right)^{\frac{3}{2}}\right)^{\frac{1}{3}} \tag{7}$$

For developed turbulent flow the Gnielinski (1976) correlation, Eq. 8 was used.

$$Nu = \frac{\left(\frac{f}{8}\right).\,(Re - 1000).\,\mathrm{Pr}}{1 + 12{,}7.\left(\frac{f}{8}\right)^{0.5}.\left(Pr^{\frac{2}{3}} - 1\right)} \tag{8}$$

The results of these comparisons are presented in Figs 4 and 5. The only experimental point where the flow was laminar and developed was for a Reynolds number of 303.2. All other laminar results are for developing flow. This can be observed in Figs. 4 and 5, where the Nusselt number increases for laminar flow as Reynolds increases. The experimental laminar heat transfer coefficient were better predicted by Churcill and Ozoe (1973) correlation with mean absolute error of 7%. Shah and London (1978) correlation predicted the same data with a mean absolute error of 10.7%. For Reynolds number of 2809 the flow is still under laminar to turbulent transition and for Reynolds number of 3314 the experimental Nusselt number approaches the value calculated by the Gnielinsk (1976) correlation with a mean absolute error of 4.8 %.

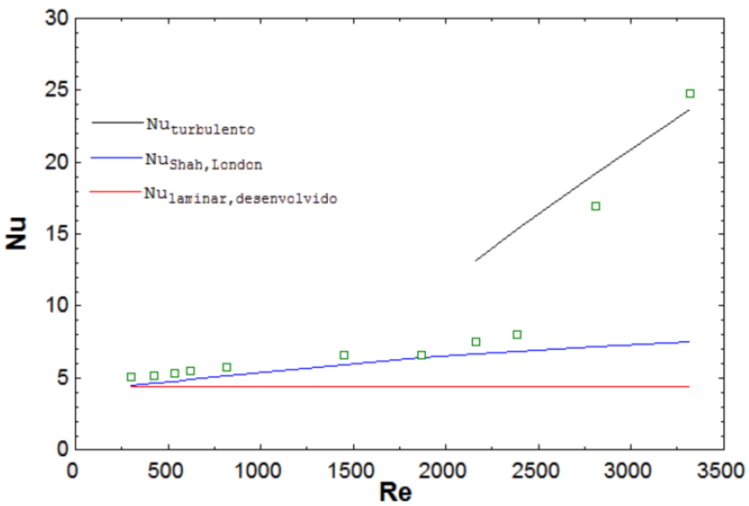

Figure 4. Single-phase experimental results. Nusselt vs. Reynolds number. Shah and London (1978) correlation.

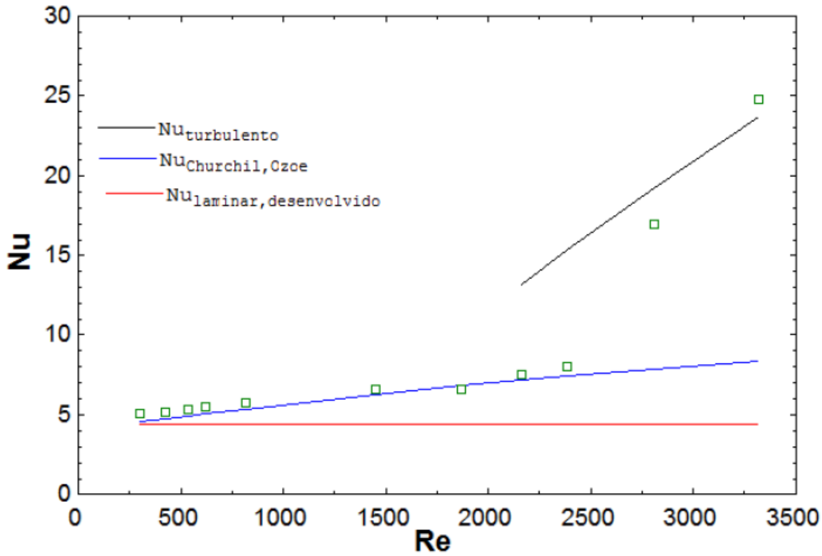

Figure 5. Single phase experimental results. Nusselt vs. Reynolds number. Churchil and Ozoe (1973) correlation.

4. RESULTS

A problem found during the experiments was related to the flow stability during two-phase flow. The presence of mass flow and temperature oscillations reduced the maximum critical heat flux obtained in the experiments. Figure 6 presents some of the oscillations occurred during the experiments for G of 170 kg/m^{2}s. In this graph it is plotted the surface temperature at the end of the test section vs. time during a critical heat flux experiment.

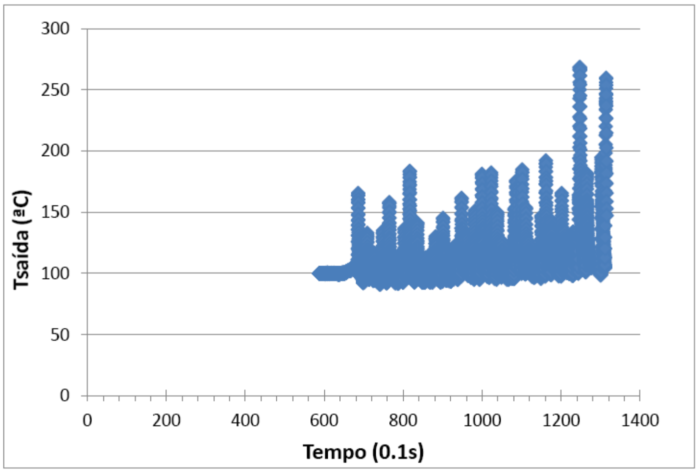

Figure 6. Experimental surface temperature at the exit of the microchannel vs time.

In this figure, the flow changes to two-phase after 62 seconds and from this point there huge peaks of temperature above a minimum lever near 100 ^{o}C, which would be presented during a stable flow . Although the presence of oscillations, the heat flux could be increased up to a level where these peaks reached extremely high temperatures, near 300 ^{o}C, values that could damage the experimental facility if present for a long time. In this manner the experiments were performed, increasing the heat flux up to a temperature level of 300 ^{o}C. Figure 7 shows an adjustment curve by minimum square root method for the surface temperature at the exit of the microchannel. These adjusted curves were used to obtain the temperature value of the surface temperature for plotting the boiling curves showed in Figs. 8, 9 and 10.

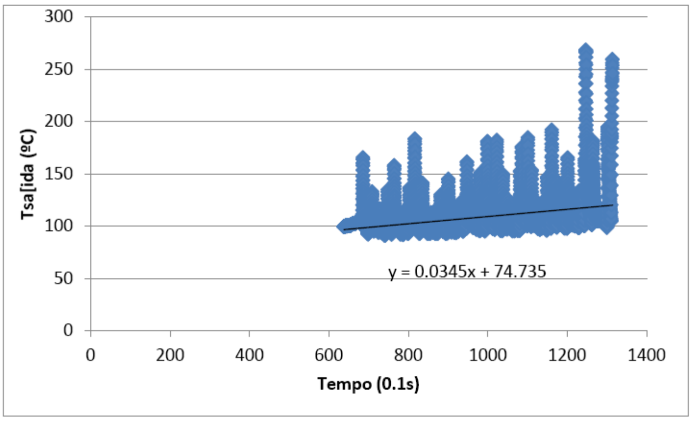

Figure 7.Adjusted curve for surface wall temperature at the exit of the test section.

Table 2 present values for critical heat flux calculated by the Zhang et al. (2006) and Katto and Ohno (1984) correlations for the same conditions of Figs. 8 to 10. Due the instabilities presented in the flow, the critical heat fluxes obtained in the experiments were less than half of the predicted by these correlations.

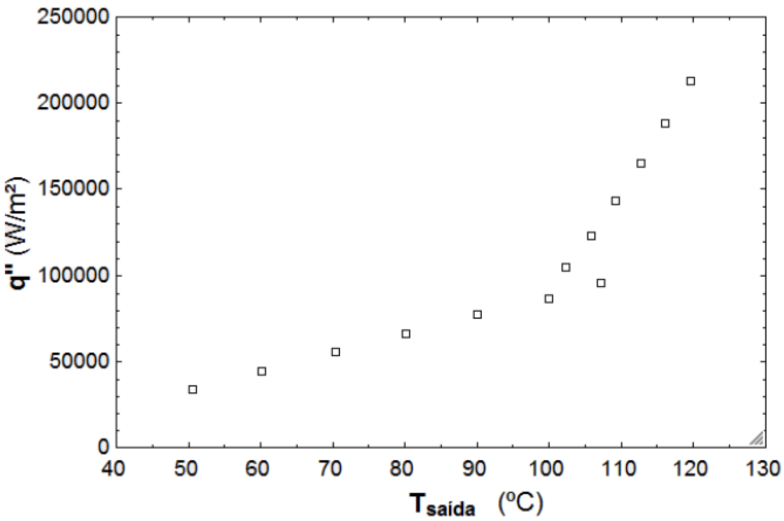

Figure 8. Heat flux vs. external surface temperature at the microchannel exit for G of 170 kg/m².s.

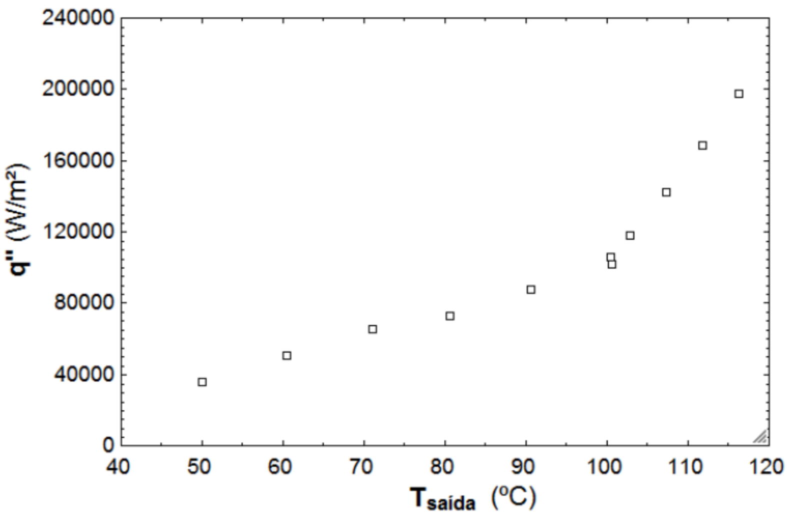

Figure 9. Heat flux vs. external surface temperature at the microchannel exit for G of 220 kg/m².s.

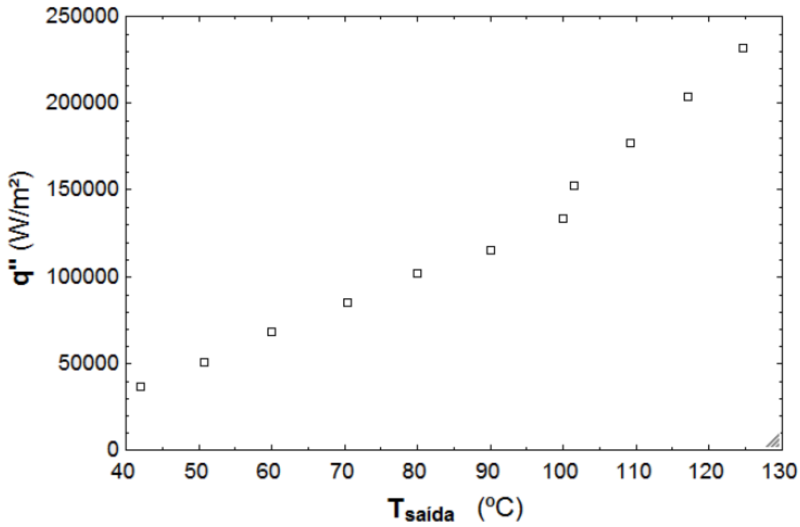

Figure 10. Heat flux vs. external surface temperature at the microchannel exit for G of 280 kg/m².s.

Table 2. Critical heat flux values predicted by correlations fo Zhang et al.(2006) e Katto e Ohno (1984) for the experimental conditions of Figs. 8 to 10.

L (m)	D (m)	Tsat (°C)	x-entrada	G (kg/m².s)	FCC-Zhang et al. (W/m²)	FCC-Katto e Ohno (W/m²)
0.07	0.00038	100	0	170	361403	607080
0.07	0.00038	100	0	220	467571	768405
0.07	0.00038	100	0	280	594841	957896

Aiming to demonstrate that very high heat flux can be obtained using microchannel flow, Table 3 shows a result obtained in this work with heat flux above 1.2 MW/m^2, for single phase flow at a mass velocity of 6441 kg/m2.s. Comparison with single phase turbulent correlation showed a difference around 20%. A pressured drop around 4 bar was necessary to obtain this flow. Higher values could be easily obtained with this experimental facility.

Table 3. Experimental results for MegaWatt/m^2 heat flux during single phase flow.

Ten (°C)	Tcentro (°C)	G (kg/m². s)	q" (MW/m²)	Nu-medido	Nu-calculado
21	74	6441	1,215	20.26	24.21

5. CONCLUSIONS

An experimental facility for the study of critical heat flux in microchannels was developed. Single phase and flow boiling heat transfer experiments were performed for water. The conclusions are the follows:

- Single-phase heat transfer conventional prediction methods performed very well for flow inside microchannels and can be used as designed tools in microscale applications.
- Flow boiling of water in single microscale channels presented high level of instability.
- Experimental critical heat flux values around half of the predicted values by the correlations was obtained in the current experimental campaign.
- Heat flux of 1.215 MW/m^2 could be achieved safely during single phase flow. Higher values could be easily achieved using this type of microchannel flow facility.

6. ACKNOWLEDGEMENTS

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8. RESPONSIBILITY NOTICE

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