An experimental study of critical heat flux of flow boiling in minichannels at high reduced pressure

A V Belyaev, A V Dedov, A N Varava and A T Komov
National Research University "Moscow Power Engineering Institute"
Russia, 111250 Moscow, Krasnokazarmennaya, 14

Abstract. This paper presents an experimental setup and experimental data for critical heat flux. The hydraulic loop of the experimental setup allows it to maintain stable flow parameters at the inlet of the test section at pressures up to 2.7 MPa and temperatures up to 200 °С. Experiments of hydrodynamics and heat transfer were performed for R113 and RC318 in two vertical channels with diameters of 1.36 and 0.95 mm and lengths of 200 and 100 mm, respectively. The inlet pressure-to-critical pressure ratio (reduced pressure) was \( p_r = p/p_{cr} = 0.15 \div 0.9 \), the mass flux ranges were between 700 and 4800 kg/(m²s), and inlet temperature varied from 30 to 180 °C. The primary regimes were obtained for conditions that varied from highly subcooled flows to saturated flows. For each regime with fixed parameters, the maximum possible heating power value was applied, with the maximum limited by the maximum output of the power supply, the onset of dryout, or wall temperatures exceeding 350 °C. The influence of flow conditions (i.e., mass flow rate, pressure, inlet temperature, and the channel diameter) on the critical heat flux is presented.

Nomenclature

- \( q_{cr} \) – critical heat flux, W/m²
- \( d \) – diameter, m
- \( L \) – channel length, m
- \( G \) – mass flux, kg/(m²s)
- \( p \) – pressure, Pa
- \( T \) – temperature, K
- \( x \) – vapour quality
- \( c_p \) – specific heat, J/(kg·K)
- \( h_l \) – latent heat of evaporation, J/kg
- \( We = \frac{G^2d}{\sigma} \) – Weber number

Greek symbols

- \( \sigma \) – surface tension, N/m
- \( \rho \) – density, kg/m³

Subscripts

- \( calc \) – calculated
- \( exp \) – experimental
- \( sub \) – subcooled
- \( cr \) – critical
- \( in \) – inlet
- \( l \) – liquid
- \( g \) – gas
- \( s \) – saturated
- \( r \) – reduced

1. Introduction

An important trend in the development of new technologies is miniaturisation of technical objects, an effort that requires extensive background knowledge of hydrodynamics and heat transfer in single-phase convection and flow boiling in small diameter channels. The ability to accurately predict the pressure drops and heat transfer and the choice of minichannel geometry and working conditions are
important factors for design and selection of the optimal settings of heat exchangers. One of the
effective methods of heat transfer from heating surfaces used in various fields of technology is the
boiling of liquid. The heat flux density transferred from the heat exchange surface by boiling liquid is
limited by the critical heat flux (CHF), which when exceeded sharply decreases the heat transfer
coefficient.

A lot of studies have been devoted to CHF in minichannels. To generalize CHF data in
minichannels, correlations [1-4], in which boiling number Bo and channel geometry are taken into
account, appear to be the most common. Below there is an example of the formula from [2]:

\[ Bo = \frac{q_{ch}}{G_{h_{lg}}} \]

\[ Bo = 0.437 \left( \frac{\rho_\alpha}{\rho_r} \right)^{0.073} We^{-0.24} \left( \frac{L}{d} \right)^{-0.72}. \]

These formulas are mostly preferable for conditions at saturated-liquid boiling in the region of low
mass flow rates.

The analysis of above and other works available in the literature, devoted to the CHF in
minichannels, shows a practical lack of research in the field of high reduced pressures. There are
known studies for high reduced pressures, performed in "normal" channels [5-7]. The possibility of
using the approach [8], based on the conversion of the water CHF look-up table [9], requires
justification for such conversion of CHF values from 8 mm to 1 mm diameter channel.

In the present work, experimental studies of CHF have been carried out for R113 and RC318 in
two vertical channels with diameters of 1.36 and 0.95 mm and lengths of 200 and 100 mm,
respectively, in the range of reduced pressures \( p_r = \frac{p}{p_{cr}} \), mass flux \( G = 700 \div 4800 \)
kg/(m\(^2\)s) and inlet temperatures \( T_{in} = 30 \div 180 \degree C \).

2. Experimental setup description

The experimental setup is shown in Figure 1. The hydraulic loop of the experimental setup allows it to
maintain stable flow parameters at the inlet of the test section at pressures up to 2.7 MPa and
temperatures up to 200 \degree C. A multistage centrifugal pump is used to circulate the working fluid
(location 6 in Figure 1). The mass flow rate was measured using two flow metres with large and small
ranges (location 7). The flow metres were calibrated prior to testing, and the experimental dependence
of the output current signal on the flow was determined. The measurement error of the flow rate was
2.2%.

The working fluids in this study were R113 and RC318, which have critical temperatures of 214.3
\degree C and 115.2 \degree C and critical pressures of 3.41 MPa and 2.78 MPa, respectively. The saturation
temperature, specific heat and critical pressure of these refrigerants are considerably lower than those
of water, which allowed us to achieve the desired parameters with less energy.

Before entering the test section (location 10 in Figure 1), the working fluid was heated in a pre-
heater (location 8). Subsequently, the refrigerant was cooled via cooling water in a recuperative heat
exchanger (location 12). Circuit pressure was increased using a thermocompressor (location 1). The
pressure and pressure drops across the inlet and outlet of the test section were measured using a
pressure sensor, and the pressure measurement accuracy was 1%. The inlet and outlet temperatures
were measured with Chromel-Copel cable thermocouples with a cable diameter of 0.7 mm. Prior to
the experiments, calibration was performed as follows. Cold thermojunctones were placed in the
Dewar vessel. Hot thermojunctones were placed in a metrological ‘Fluke 9173’ dry-block thermostat,
which has an absolute error of temperature stabilisation of \( \pm 0.006 \degree C \). As a result, the individual
characteristics of the Chromel-Copel cable thermocouples were determined. The temperature
measurement accuracy of flow was 1%.

The test section was heated with alternating current. The electrical current strength was measured
using an LA 55-P current transducer. The measurement error of the electric power was 1%.
Figure 1. Experimental setup: 1) thermocompressor, 2) tank, 3) and 5) filters, 4) balloon with refrigerant, 6) multistage centrifugal pump, 7) flow metres, 8) pre-heater, 9) roughing-down pump, 10) test section, 11) current transducer, 12) recuperative heat exchanger, and 13) bypass line

The test section is shown in Figure 2. Two vertical stainless steel tubes were used with heated lengths of 200 and 100 mm, internal diameters of 1.36 and 0.95 mm, and external diameters of 1.60 and 1.23 mm, respectively. The tube was electrically insulated and hydraulically sealed using PTFE seals. Electrodes were soldered to the tube with silver. Inlet and outlet collectors were located on platforms made of kaprolon, which has low thermal conductivity, to minimise heat losses from the test section to the experimental setup.

Figure 2. Design of the test section for $d = 1.36$ mm

The design of the test section is temperature compensated, and the inlet collector has a vertical degree of freedom. The platform is mounted on two vertical metal rods on which it is able to slide. To
avoid vibration and create stability for the test tube, the platform of the inlet collector is pressed by a spring along the rods towards the tube.

Measurements of the wall temperatures are collected by six Chromel-Copel thermocouples. The wires (diameter 0.2 mm) were welded via lasers to the working area of the tube in six cross-sections (T1-T6, see Table 1) on opposite sides of the tube. This mounting method for the thermocouples created low thermal inertia for the sensors and allowed measurement of the average temperature of the wall along its perimeter. The inner wall temperatures were calculated using a correction for the wall conductivity. The temperature measurement error was 1%. The temperatures of the cold junctions of the thermocouples mounted in a metal box were measured using a thermistor.

Table 1. Coordinates of the cross-sections (mm)

| Diameter (mm) | T1  | T2  | T3  | T4  | T5  | T6  |
|--------------|-----|-----|-----|-----|-----|-----|
| 1.36         | 40  | 70  | 100 | 130 | 160 | 190 |
| 0.95         | 20  | 35  | 50  | 65  | 80  | 95  |

3. Analysis of primary data
The experiments were performed for 195 regimes, where the critical heat load with different flow parameters was determined in the range of reduced pressures \( p_r = p/p_{cr} = 0.15 \div 0.9 \), mass flux \( G = 700 \div 4800 \) kg/(m²s) and inlet temperatures \( T_{in} = 30 \div 180 \) °C. Each regime with fixed flow parameters got the maximum possible heat flux, which was determined either by the power source or by the onset of CHF, or when the wall temperature reached 350 °C. In the regimes with high pressure close to critical, it was not possible to determine the CHF value. Thus, 166 values of CHF were obtained in the above range of working fluid parameters.

The development process and fixation of the CHF were implemented by wall thermocouple readings at the outlet of the channel in the T6 section. Depending on values of inlet subcooling and coolant mass flux, the nature of CHF has its own peculiarities. Under flow parameters with high mass flux and high subcooling, it was observed a sharp wall temperature increase in the T6 section. In this case, the crisis occurs as a result of the transformation of subcooled nucleate boiling to a film boiling on the heating surface. It occurs generally at high heat flux density. Figure 3 provides an example of such crisis development with wall temperatures readings T1-T6 as a function of time at \( G = 1760 \) kg/(m²s), \( T_{in} = 160 \) °C, \( T_s = 171 \) °C, \( p_r = 0.52 \) for R113. Here we can see a pre-crisis gradual increasing of amplitude of wall temperature fluctuations with further sharp increase. In this moment, the heat load on the test section was rapidly reduced. The CHF value was taken before such high wall temperature fluctuations and sharp temperature increases.

Figure 3. Wall temperatures during the development of the heat transfer crisis
An example of crisis development at flow parameters $G = 3200 \text{ kg/(m}^2\text{s})$, $T_{\text{in}} = 155 \degree \text{C}$, $T_s = 188 \degree \text{C}$, $p_r = 0.68$ for R113 is presented in Figure 4. It is shown the values of wall temperatures corresponding to a number of consecutive values of $q$ ($q_1 < q_2 < q_3 < q_4$) increasing stepwise. Before the crisis onset, the wall temperature at first significantly decreases by $\sim 1 \degree \text{C}$ under $q_3$, and then increases sharply with the heat load increase to $q_4$. This pre-crisis phenomenon was described in [10] and referred to as ST regime. In this regime, the crisis of heat transfer may be preceded by the formation of a steam "conglomerate" near the wall with a thin liquid film in its base, which makes it possible to intensify the evaporation of the liquid and reduce the wall temperature. This steam "conglomerate" can occur when individual bubbles are merged near the wall. When the liquid film in its base has been completely evaporated, the wall surface is "steamed" that leads to the crisis of heat transfer, the wall temperature increases sharply. Further fluctuation of the wall temperature in Figure 4 characterizes the unstable film boiling. In this case, the value of the heat flux in the ST regime before a sharp increase of the wall temperature was taken as the CHF value.

![Figure 4](image-url)  

*Figure 4.* Wall temperatures during the development of the heat transfer crisis. $\square$ – Missed stationary time interval. Time scale for information.

With the inlet temperature increase and mass flux decrease, the crisis phenomena proceed more smoothly, and zones with a stable film boiling appear. In this case, there was a spread of the crisis against the flow and film boiling down to T2 section. Figure 5 shows wall temperatures T1-T6 as a function of time for a similar regime at $G = 1100 \text{ kg/(m}^2\text{s})$, $T_{\text{in}} = 180 \degree \text{C}$, $T_s = 180 \degree \text{C}$, $p_r = 0.59$ for R113. As a result of heat load increase after the crisis, the wall temperature was increasing till it reached the stationary values. The CHF value was taken before the wall temperature increase. Figure 5 shows the boiling of saturated liquid in sections T1-T4 and the stable film boiling in sections T5-T6, as well as pre-crisis temperature fluctuation in T4 section.
In some regimes with a pressure close to the critical one, it was not possible to determine the CHF value based on the indications of the wall thermocouples. Figure 6 shows an example of the wall temperature values as a function of the heat flux density in sections T1-T6 for the regime with close to critical reduced pressure $p_r = 0.9$, mass flux $G = 3300 \text{ kg/(m}^2\text{s})$ and subcooling at the inlet $\Delta T_{\text{sub}} \approx 47 ^\circ\text{C}$. Wall temperature parameters at $q<200 \text{ kW/m}^2$ allow defining convective heat transfer region, after which usually with increasing $q$ the wall temperature stabilizes by reason of bubble boiling and then increases sharply when $q$ reaches CHF. In this case, there are no accurate areas of changes in the heat transfer modes, and the main criterion for the indication of crisis of heat transfer is a sharp wall temperature increase with a small increase of heat flux value.

**Figure 5.** Wall temperatures during film boiling

**Figure 6.** The wall temperature in sections T1-T6 as a function of the heat flux density in the regime with a pressure close to the critical one
Figure 7 shows a characteristic graph of inlet temperature influence on CHF via mass flux for RC318. The maximum CHF value was achieved at high mass flux and a significant subcooling at the inlet. In the region of high mass flux $G \geq 2000 \text{ kg/(m}^2\text{s})$, the inclination of the curves decreases, the rate of increase in CHF decreases with increasing mass flux. A similar change of the inclination was observed earlier in the boiling of subcooled liquid in [7,11-14]. With a decrease in subcooling, the influence of mass flux on CHF is reduced.

![Figure 7. Inlet temperature influence on CHF via mass flux](image)

Figure 8 shows the dependence of CHF on the inlet temperature at different mass flux for RC318. It is seen that when mass flux increases and the inlet temperature decreases, the CHF begins to increase. The mass flux decrease leads to the less influence of the inlet temperature on the CHF. The inclination of the curve for $G = 3600 \text{ kg/(m}^2\text{s})$ is approximately twice as large as for $G = 850 \text{ kg/(m}^2\text{s})$.

![Figure 8. Mass flux effect on CHF as a function of inlet temperature](image)
Figure 9 shows an effect of the channel diameter on CHF via inlet vapour quality at different mass fluxes for R113. At approximately same mass fluxes, larger CHF values correspond to a smaller diameter.

![Graph showing effect of diameter on CHF](image)

**Figure 9.** Effect of the diameter on CHF via inlet vapour quality at different mass fluxes

### 4. Conclusions

This paper presents an experimental setup and experimental data for critical heat flux of R113 and RC318 in two vertical channels with diameters of 1.36 and 0.95 mm and lengths of 200 and 100 mm, respectively, under different combinations of reduced pressure, liquid subcooling and mass flux and heat load. These governing parameters were varied within the following ranges: $p_r = 0.15$–0.9, mass flux $G = 700$–4800 kg/(m$^2$s), inlet fluid temperature $T_{in} = 30$–180 °C (maximal value of CHF was $q_{cr} = 907$ kW/m$^2$).

The analyses examined the influence of the various flow parameters on CHF of subcooled flow boiling in the channels with small diameters. For the full range of flow parameters, univocal influences of mass flow rate, pressure, diameter and fluid subcooling on CHF were observed. For a fixed inlet subcooling, the CHF increases with the mass flux and the increasing rate becomes smaller at the high mass flux. For a fixed mass flux, the CHF increases linearly with increasing the inlet subcooling. Decrease of diameter enhances CHF.

At the pressures very close to the critical point ($p_r \approx 0.9$) the wall temperatures increase monotonously and a sharp rise of wall temperature no longer occurs which implies that the CHF phenomenon no longer exists.

For high reduced pressure ($p_r > 0.4$) qualitative differences of CHF data in minichannel from the results obtained in conventional tubes was not observed.

### References

[1] W. Zhang, T. Hibiki, K. Mishima, Y. Mi, Correlation of critical heat flux for flow boiling of water in mini-channels, Int. J. Heat Mass Transf. 49 (5–6) (2006) 1058–1072

[2] L. Wojtan, R. Revellin, J.R. Thome, Investigation of saturated critical heat flux in a single, uniformly heated microchannel, Exp. Therm. Fluid Sci. 30 (8) (2006) 765–774
[3] C.L. Ong, J.R. Thome, Macro-to-microchannel transition in two-phase flow: Part 2 – Flow boiling heat transfer and critical heat flux, Exp. Therm. Fluid Sci. 35 (6) (2011) 873–886

[4] 4. Zahid Anwar, Bjorn E. Palm, Rahmatollah Khodabandeh, Dryout characteristics of natural and synthetic refrigerants in single vertical mini-channels, Experimental Thermal and Fluid Science 68 (2015) 257–267

[5] A.P. Ornatskiy, A.M. Kichigin, Issledovaniye zavisimosti kriticheskoy teplovoi nagruzki ot vesovoy skorosti, nedogreva i davleniya, Teploenergetika 2 (1961) 75–79

[6] A.P. Ornatskiy, Kriticheskiye teplovyye nagruzki i teplootdacha pri vynuzhdennom dvizhenii vody v trubakh v oblasti sverkhvysokikh davleniy (175–220 bar), Teploenergetika 3 (1963) 66–69

[7] Luzhi Tan, Changnian Chen, Xiaoming Dong, Zhiqiang Gong, Mingtao Wang, Experimental study on CHF of R134a flow boiling in a horizontal helically-coiled tube near the critical pressure, Experimental Thermal and Fluid Science 82 (2017) 472–481

[8] I.L. Pioro, D.C. Groeneveld, S.C. Cheng, S. Doerffer, A.Z. Vasic, Yu. V. Antoshko, Comparison of CHF measurements in R-134a cooled tubes and the water CHF look-up table, International Journal of Heat and Mass Transfer 44 (2001) 73±88

[9] D.C. Groeneveld, L.K.H. Leung, P.L. Kirillov, et al., The 1995 look-up table for critical heat flux in tubes, Nuclear Engineering and Design 163 (1996) 1±23

[10] A.V. Belyaev, A.N. Varava, A.V. Dedov, A.T. Komov, An experimental study of flow boiling in minichannels at high reduced Pressure, International Journal of Heat and Mass Transfer 110 (2017) 360–373

[11] A.V. Dedov, A.T. Komov, A.N. Varava, V.V. Yagov, Hydrodynamics and heat transfer in swirl flow under conditions of one-side heating. Part 2: Boiling heat transfer. Critical heat fluxes, International Journal of Heat and Mass Transfer 53 (21–22) (2010) 4966–4975

[12] Syed Waseem Akhtar, Sang-Ki Moon, Se-Young Chun, Sung-Deok Hong, Won-Pil Baek, Modeling capability of R134a for a critical heat flux of water in a vertical 5_5 rod bundle geometry, Int. J. Heat Mass Transf. 49 (2006) 1299–1309

[13] C.L. Ong, J.R. Thome, Macro-to-microchannel transition in two-phase flow: Part 1 – Two-phase flow patterns and film thickness measurements, Experimental Thermal and Fluid Science 35 (2011) 37–47.

[14] Macro-to-microchannel transition in two-phase flow: Part 2 – Flow boiling heat transfer and critical heat flux, C.L. Ong, J.R. Thome, Experimental Thermal and Fluid Science 35 (2011) 873–886.

Acknowledgments
This work was supported by RSF Grant 16-19-10457.