Experimental study of critical heat flux at flow boiling of R125 in small diameter channel

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Abstract. This paper presents an experimental setup and experimental data for critical heat flux at high reduced pressures. Studies on critical heat flux during the forced flow of Freon R125 in a vertical channel with a diameter of 0.95 mm and length of 100 mm were performed. The hydraulic loop of the setup provides stable flow conditions at the inlet of the test section under pressures up to 2.5 MPa and temperatures up to 200 °C. The measured parameters were the mass flow rate, pressure and temperature at the inlet and outlet of the test section, the joule heating power and the wall temperature in the six cross-sections along the tube. An automated data-acquisition system was used during the experiments. A comparison between the experimental and calculated data is presented.

1. Introduction

In minichannels (0.2 < d ≤ 3 mm [1]), peculiarities may result from the characteristic scale of the phenomena taking place during flow boiling and from the linear scale of the channel. Investigations have been made into heat transfer in mini- and microchannels over many years [2]. Many experimental works are currently available in the literature; however, the data involved is mainly obtained from experiments involving low and moderate reduced pressures. Moreover, the calculation methods proposed by the authors are usually empirical in nature and are only suitable for describing cases that are relevant to those of the authors. That is why they cannot predict the results of experiments with conditions different from those under which they were obtained.

In the field of high reduced pressures based on the performed estimates of the ratios of the agreed scales, we can assume, that flow regimes in minichannels become similar to those in regular channels [3]. In this case, the heat transfer in minichannels can then be to apply calculation methods used for “normal” channels [4]. This work provides experimental confirmation of the hypothesis.

2. Description of the experimental setup

The experimental setup is shown in Figure 1. The hydraulic loop of the experimental setup allows for the maintenance of stable flow parameters at the inlet of the test section at pressures up to 2.5 MPa and temperatures up to 200 °C. A multistage centrifugal pump is used to circulate the working fluid (location 6). The mass flow rate was measured using flow meter (location 7). Previously, the flowmeters were calibrated and the experimental dependence of the output current signal from the flow was obtained. The measurement error for the flow rate was 2.2%. The working fluids were R125.
Before entering the test section (location 10 in Figure 1), the working fluid was heated in a pre-heater 1 (location 8). Pre-heater 2 (location 14) was used to create the necessary vapor quality at the entrance to the test section. 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. 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 meters, 8) pre-heater 1, 9) roughing-down pump, 10) test section, 11) current transducer, 12) recuperative heat exchanger, 13) bypass line, 14) pre-heater 2.](image)

The test section is shown in Figure 2. Vertical stainless steel tubes were used with heated lengths of 100 mm, internal diameters of 0.95 mm, and external diameters of 1.23 mm. 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 minimize heat losses from the test section to the experimental setup.

![Figure 2. Design of the test section for d = 0.95 mm.](image)
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 five cross-sections (T1-T5, 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%.

Before inlet the test section, the pre-heater was used to create the required vapor quality, made of a stainless tube with inner diameter of 1.36 mm and a heated length of 60 mm. The tube was heated by direct current from a laboratory power source. A Chromel-Copel thermocouple was soldered at the exit section on the pre-heater wall to monitor and prevent heat transfer crisis.

| Diameter (mm) | T1 | T2 | T3 | T4 | T5 |
|--------------|----|----|----|----|----|
| 0.95         | 35 | 50 | 65 | 80 | 95 |

### 3. Research results

#### 3.1 Primary data

In the course of experiments, following parameters were measured: mass flow rate, pressure and temperature at the input and output of the working area, electric power of heating, temperature of the wall at six \( T1 \) – \( T5 \) sections along the mini channel. Experiments were performed at the reduced pressure \( p_r = \frac{p}{p_{cr}} = 0.44 \pm 0.7 \), the mass flow rate was varied from 1800 to 5200 kg/(m\(^2\)s). At fixed parameters of the flow at the input to the working area, in steps, from the area of loads corresponding to one-phase convection, the input heat load was increased. When reaching stationary values at each step, the readings of sensors were recorded. The values were averaged, and protocols of primary measurements were formed.

#### 3.2 Critical heat fluxes

The experiment started from setting given values of pressure, flow rate and enthalpy of the medium. Then, the heat load at the experimental area was gradually increased, enthalpy increased during that, and thus, the vapor quality at the output from the tube \( \rho + \rho \omega \) remained constant. At a certain combination of the specific heat flow and the vapor quality at the output from the tube, the critical heat flux of the first kind was created. At that moment, all mode parameters were recorded, and, in order to prevent the destruction of the working area, the heat load was immediately decreased. The next experiment was carried out with a slightly higher value of enthalpy of the cooling medium at the input to the experimental area, and the procedure of the experiment was repeated.

At a certain value of medium enthalpy at the input and of the specific heat flow at the output section of the channel, possibly, according to some works [5], the limiting critical quality was reached, that is, the flow mode was changed to dispersed, and a microfilm was created. In the course of further increase of the heat load, the \( x_{out} \) also increased, and the limiting critical quality was reached at sections which were located further and further from the output end of the heated tube, and at the moment when the released amount of heat at the part of the tube from the transit to the output section become sufficient for vaporization of the microfilm, at the end of the tube the critical heat flux of the second kind was created.
During the further increase of the medium enthalpy at the input to the channel, the length of the microfilm increased, and, thus, the critical heat flux of the second kind at the output section of the tube was created also with gradually decreased value of \( q_{cr} \).

When, at the input to the channel, the vapor quality became higher than the limiting critical quality in the course of increase of the enthalpy at the input to the heated tube, the critical heat flux was created at the output section of the tube at the of steam which continued increasing. With the increase of the vapor quality in the tube, the thickness of the microfilm decreased, and, thus, the amount of heat necessary for drying up the microfilm also decreased.

It is known that with increasing pressure increases heat transfer during boiling. In contrast to the heat transfer, the effect of pressure on the critical heat flux is not always unambiguous. The data in tables [6] show that for a significant range of changes in reduced pressure \( (p_r = 0.2 \div 0.9) \) \( q_{cr} \) decreases with increasing pressure. The experimental data obtained in this work also show a decrease in \( q_{cr} \) with increasing pressure. Figure 3 shows a decrease in the critical heat flux value with an increase in the reduced pressure at approximately the same parameters of the coolant at the inlet of test section the effect of pressure on \( x_{lim} \).

![Figure 3](image)

**Figure 3.** Dependence of \( q_{cr} \) from \( x \) at the moment of creation of the critical heat flux for freon R125 and the effect of pressure on \( x_{lim} \).

Figure 4 shows the dependence of inlet vapor quality from output vapor quality; the horizontal part of the graph characterizes \( x_{lim}^0 \) and is reached at \( x_{in} \leq x_{lim} \). The inclined line covers the area in which the medium with the vapor quality of \( x_{in} > x_{lim} \) is supplied to the input of the heated part of the tube. The crossing point of the horizontal and the inclined lines define values of \( x_{lim} \) and \( x_{lim}^0 \), and thus the point of transit of the circular flow mode to the dispersed mode. Taking into consideration the significant number of experimental points which are approximated by two straight lines, \( x_{lim} \) and \( x_{lim}^0 \) were determined sufficiently reliably, which will allow to detect erroneous data in the future.
Figure 4. Dependence of $x_{in}$ from $x_{out}$ at the moment of creation of the critical heat flux for freon R125.

**Conclusions**

The paper presents data on heat transfer and critical heat flux during flow boiling under conditions of high reduced pressure in a small-diameter channel. The obtained heat transfer experimental results satisfy the presented calculation method, which is used to calculate heat transfer in common channels. The effect of pressure, mass flux on the critical heat flux and limiting critical quality is shown. $x_{lim}$ increases with increasing pressure.

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