Experimental study of heat transfer under flow boiling ozone-safe refrigerants in microchannels

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Abstract. This paper devoted to experimental measurement of heat transfer and critical heat fluxes into horizontal microchannels under flow boiling of refrigerant 141b. The copper experimental test section contains two microchannels with a 2x0.36 mm cross-section. Average heat fluxes versus wall superheat in the range of mass fluxes from 600 to 2100 kg/m²s and heat fluxes up to 140 W/cm² are presented. A slight deterioration in heat transfer relative to pool boiling with heat fluxes close to critical was observed.

1. Introduction
The heat transfer in microsystems with phase transformations is widely investigated. This is due to the increase in industrial applications requiring large heat fluxes and mass fluxes in a limited space as well as with cooling electronic equipment. [1]. Heat transfer during liquid-vapor flow in channels smaller than capillary constant significantly different from heat transfer in conventional tubes. The experimental data from [2-6] and other papers confirmed that. There are many published methods for calculations of heat transfer coefficient in microchannels, but for the application the verification requires. Experimental investigation of the flow boiling in microchannels for different refrigerants is still relevant [7]. The goal of this study is the determination of the dependence of heat flux on the wall superheat and the critical heat flux (CHF) during flow boiling of ozone-safe refrigerant R141b in microchannels with a high aspect ratio experimentally.

2. Experimental equipment and measurements
A scheme of the experimental flow loop is shown in figure 1. The mass flow controller Bronkhorst HI-TECH is used to set the flow rate. In the presented series of experiments, subcooled liquid was supplied to the entrance to the experimental section. A heater (pre-evaporator) installed in front of the measuring section is used to maintain the fluid temperature constant. Copper coils with cooling water in the outlet reservoir serve to condense and cool the stream leaving the experimental section.

The microchannel system used for experiments is shown in figure 2. Two channels with a cross-section of 2 mm × 0.36 mm made at the top of the copper block, the aspect ratio α=a/b is 5.56, where a is the width and b is the height of the channel. The distance between channels equal to 2 mm. The channel length equals to 16 mm. Surface roughness of milled channels Rₐ equaled 0.67 μm. Six cartridges with a maximum power of 200 watts each heat the copper block. Wall temperature measured by thermocouples at 3 mm from edges of channels in each channel. Two thermocouples measure temperature in each point at 1 mm and 5.5 mm below the surface of the microchannels.
Figure 1. Schematic of the experimental loop.

A stainless steel lid covers the copper block with channels and is sealed on top with a fluoroplastic gasket. Inlet and outlet chamber separated from copper block trough thermal insulating plate. Temperature and pressure inside the inlet and outlet chamber are also measured. The experiments were carried out with the horizontal orientation of microchannels. Copper block with channels is thermoinsulated and the heat losses were measured and calibrated. The internal average surface temperature of the microchannels $T_W$ was calculated from the measured local temperature gradients $\nabla T_w$ between the thermocouples in the copper as follows

$$T_W = \frac{1}{N} \sum_{i=1}^{N} (T_{m,i} - \delta_i \cdot \nabla T_{W,i})$$

(1)

here $T_{m,i}$ is the measured temperature in the top of the thermocouple at $i$ position, $\delta_i$ is the distance from the top thermocouple to the inner surface of microchannel. The heat flux to the inner wall of channels was calculated with taking into account heat losses $Q_{loss}$ as follows

$$q = \frac{U \cdot I - Q_{loss}}{A_{in}}$$

(2)

Figure 2. Schematic of the test section.
here. \( U \) and \( I \) are electrical supply and current from heating cartridges and \( A_{in} \) is the inner area of microchannels.

3. Results and discussion

The measured dependence of average heat flux versus average wall superheat during flow boiling of refrigerant R141b at mass fluxes of 710, 1330 and 2050 kg/m\(^2\)s are shown in figure 3. The calculations according to the model of pool boiling heat transfer (solid line) and single-phase convection models (dashed lines) are also presented in figure 3. The experimental data were obtained under averaged saturated temperature of 51 \(^{\circ}\)C and inlet liquid temperature of 22-25 \(^{\circ}\)C. The pool boiling heat transfer equation for R141b was selected as the Cooper correlation, which is in good agreement with experimental data [8]. Data from [9] was used to calculate heat transfer under laminar convection with three- side heating in the channel with an aspect ratio of 5.56. Influence of thermal developing flow on average heat transfer as in [10] was taken into account.

Under turbulent flow condition, the heat transfer coefficient calculated by the Gnielinski correlation. The influence of thermal developing flow and asymmetry heat flux around the channel perimeter on average heat transfer was taking into account as in [11]. The effect of changes in the viscosity of the fluid with temperature along the channel on the Reynolds number was also taken into account.

The data under low wall superheat correspond well to calculated convective heat transfer. For developed nucleate boiling of R141b, the data of measurements correspond to pool boiling up to the threshold value of heat flux determined by the mass flux. When the heat flux reaches a threshold value, boiling suppression is observed. For mass flux equal to 1330 kg/m\(^2\)s this value is about 80 W/cm\(^2\). The nucleate boiling suppression depends on the mass flux. The experimentally measured CHF versus the mass flux are shown in figure 4 for subcooled flow boiling with an inlet temperature of 22-25 \(^{\circ}\)C and saturation temperatures of 39 and 51 \(^{\circ}\)C. As it is seen, the low dependence of CHF on static pressure and initial subcooling was observed. The solid and dotted lines in figure 4 correspond to the calculation of CHF according to the model [12]. The calculations overpredict the experimental data for all studied mass fluxes considerably and show low dependence of the CHF on saturation temperature similar to the experimental observations.

![Figure 3. The dependence of heat flux vs. wall superheat.](image-url)
Conclusions
The boiling curves and CHF for flow boiling of ozone-safe refrigerant R141b in microchannel heat sink were measured in the range of mass fluxes from 600 to 2100 kg/m²s and under heat fluxes up to 140 W/cm². It was found that for developed nucleate boiling of R141b, the data of measurements correspond to pool boiling heat transfer. When the heat flux reaches a threshold value, boiling suppression is observed. The nucleate boiling suppression depends on the mass flux. New data on critical heat flux have been measured for short three side heated microchannels with a high aspect ratio. A weak influence of initial subcooling and static pressure on the CHF was observed.

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