Experimental investigation of heat transfer during flow boiling with ozone-safe refrigerants in microchannels

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Abstract. The results of an experimental investigation of heat transfer during subcooled flow boiling with ozone-safe refrigerants R236fa and R141b in horizontal microchannels are presented. The experiments are performed in a closed loop that re-circulates the refrigerant. The working section is represented by two microchannels with a 2x0.36 mm cross-section separated by the wall with a thickness of 2 mm. The dependences of the heat flux and pressure drop on wall superheat are measured in the range of heat fluxes from 5 to 120 W/cm

1. Introduction

The development of high power electronic devices needs the creation of the compact systems for high heat flux removing. These systems are typically based on microchannels and referred to as the microchannel heat sink. Heat fluxes for computer microprocessors already exceed now 100 W/cm², so the traditional cooling methods are ineffective [1]. To solve this problem, microchannel two-phase cooling, which uses the latent heat of the phase transition to remove large amounts of heat at a constant temperature of the coolant, is promising. As it was mentioned in [2-4], a variety of new phenomena arise when flow boiling occurs in the microchannels. The application of microchannels for high heat flux removing necessitates the experimental study of the subcooled flow boiling in special design of microchannel heat sink [5] intended to suppress the premature heat transfer crisis [6]. Therefore, experimental investigation of subcooled flow boiling heat transfer in the microchannel heat sink remains actual. The objective of this study is the experimental determination of the dependence of the heat flux and pressure drop on wall superheat in a wide range of heat fluxes for moderate and low reduced pressures using refrigerants R236fa and R141b accordingly. The experiments were performed using a microchannel heat sink described in [5], which allowed suppressing the development of premature heat transfer crisis.

2. Experimental setup

The experimental equipment for studying flow boiling heat transfer in microchannel heat sink was described in detail in [5]. The experiments were performed in a closed circuit where the refrigerant was supplied from the condenser tank by a plunger pump through a pulsation dampener, a thermostat, a filter, a flow meter, and a pre-evaporator to achieve the flow with the desired vapor quality or initial temperature. The working section is a copper block of 16 mm long with two microchannels of 2 mm wide and 0.36 mm deep, separated by a wall of 2 mm thick. The working section is thermally
insulated with paronite gaskets and the slits of 640 microns are cut in the gaskets. The ratio of the cross-section area of the distribution chamber to the passage section of the microchannels is 34.9, it makes it possible to achieve complete condensation of vapor discharged into the distribution chamber when nucleate boiling occurs. The length of microchannels equals 16 mm, the aspect ratio $a=a/b$ is 5.56, where $a$ is the width and $b$ is the height of the channel. The surface of the microchannels is treated with sandpaper, and the measured average surface roughness $R_a$ equals 0.67 µm. The cross-section of a top area of the copper block is 10×16 mm. Cartridge heaters are mounted into the copper block as is shown in figure 1 and the maximum power supplied to the microchannel heat sink is 1200 W. Eight thermocouples with a diameter of 0.5 mm are installed in a copper block at a distance of 1.2 mm and 5 mm from the inner surface of the channels as it is shown in figure 1.

![Figure 1. Schematic of the test section used for the study of subcooled flow boiling heat transfer.](image)

An unheated polished stainless steel plate closes the microchannels. Fluoroplastic gaskets are used for sealing. The inlet and outlet temperature and pressure are measuring inside the inlet and outlet distribution chambers. The experiments were carried out with the horizontal orientation of microchannels. The working section was thermally insulated and the heat losses from the test section were calibrated and did not exceed 0.19 W/K. The temperature of the internal surface of microchannels $T_w$ was determined from the measured local temperature gradients $\nabla T_w$ in the copper

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

(1)

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

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

(2)

here, $U$ and $I$ are electrical supply and current from heating cartridges and $A_{in}$ is the inner area of microchannels. The experiments were performed in the range of mass flow rates from 600 to 1600 kg/m’s and heat fluxes from 5 to 120 W/cm² under initial subcooling from 5 to 30 °C.
3. Results and discussion

The experiments were performed to make it possible to identify the areas of predominant influence of single-phase convection, subcooled flow boiling and transition to saturated flow boiling. The dependence of the heat flux on wall superheat during flow boiling with refrigerant R236fa at mass fluxes of 1490 kg/m²s is presented in figure 2 (a). The experimental data were obtained under the saturated temperature of 53°C, corresponding to reduced pressure $p/p_{cr} = 0.2$, and inlet liquid temperature of 25 °C. The predictions of the wall superheat according to pool boiling equation [7] (dotted line) and turbulent single-phase convection equation [8] (dashed lines) are presented also in figure 2 (a). As it is seen, for single-phase convection, the experimental data correspond well to the prediction according to [8] and nucleate boiling arising enhances the heat transfer considerably. The solid line in figure 2 (a) shows the prediction according to [9] for subcooled flow boiling when the forced convection enhancement factor is $F = 1$. For subcooled flow boiling, this correlation can be used for the estimation of the wall superheat. When saturated flow boiling occurs at the microchannel outlet, the heat transfer deterioration is observed similar to that discussed in [10] for saturated flow boiling.

![Figure 2](image_url)

**Figure 2.** The dependence of heat flux on wall superheat for R236fa - (a) and R141b - (b).

The dependence of the heat flux on wall superheat during flow boiling with refrigerant R141b at mass fluxes of 1330 kg/m²s is presented in figure 2 (b). The experimental data were obtained under a saturated temperature of 51°C corresponding to reduced pressure $p/p_{cr} = 0.044$, and inlet liquid temperature of 24°C. The predictions of the wall superheat according to pool boiling equation [11] (dotted line) and turbulent single-phase convection equation [8] (dashed lines) are presented also in figure 2 (b). As it is seen, subcooled flow boiling enhances the heat transfer considerably in comparison with forced convection and pool boiling. The pool boiling equation [11] for R141b was selected as the Cooper correlation that is in a good agreement with experimental data [12] if the multiplier coefficient is selected as 60. The solid line in figure 2 (b) shows the prediction according to [9] for subcooled flow boiling when the forced convection enhancement factor is $F = 1$. For subcooled flow boiling, this correlation can be used for estimation of the wall superheat of up to the critical heat flux. It shows that heat transfer deterioration during the transition to saturated flow boiling is not observed for the low reduced pressure.

The dependence of the pressure drop on the wall superheat for these conditions is shown in figure 3. As it is seen, the pressure drop for R236fa shown in figure 3 (a) increases rapidly when nucleate boiling arises at the wall superheat above ten degrees and increases gradually with increasing heat flux up to achieving the critical heat flux. For R141b, the incipience of nucleate boiling less affects the pressure drop during subcooled flow boiling, see figure 3 (b). The reason for this difference is
increasing vapor velocity at low reduced pressure causing a decrease in the effect of nucleate boiling on the pressure drop.

\[ T_{\text{Sat}}=53 \, ^\circ\text{C}, \quad T_{\text{inlet}}=25 \, ^\circ\text{C}, \quad G=1490 \, \text{kg/m}^2\text{s} \]

\[ T_{\text{Sat}}=51 \, ^\circ\text{C}, \quad T_{\text{inlet}}=24 \, ^\circ\text{C}, \quad G=1330 \, \text{kg/m}^2\text{s} \]

(a)  
(b)

**Figure 3.** The dependence of pressure drop on wall superheat for R236fa - (a) and R141b - (b).

4. Conclusions
   The boiling curves for subcooled flow boiling in the microchannel heat sink were measured for two ozone-safe refrigerants R236fa and R141b that have considerably different reduced pressures. It was shown that for refrigerant R236fa at moderate reduced pressure, the heat transfer deterioration is observed when saturated flow boiling occurs at the microchannel outlet. For R-141, at low reduced pressure, the heat transfer deterioration during the transition to saturated flow boiling is not observed. This result needs more detailed experimental study to find the reason for this phenomenon.

   The dependence of the pressure drop on the wall overheating shows a significant difference for flow boiling at low and moderate reduced pressures. For refrigerant R236fa, at moderate reduced pressure, the pressure drop increases rapidly when the nucleate boiling arises and increases gradually with increasing heat flux up to achieving the critical heat flux. For R141b, at low reduced pressure, the incipience of nucleate boiling affects the pressure drop during subcooled flow boiling in a less degree.

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