

High heat flux flow boiling of refrigerant R236fa in parallel microchannels

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Abstract. This paper presents the results of an experimental study of the heat transfer during flow boiling of refrigerant R236fa in a horizontal microchannel heat sink. The experiments were performed using closed loop that re-circulates coolant. Microchannel heat exchanger that contains two microchannels with 2x0.4 mm cross-section was used as the test section. The dependence of average heat flux on wall superheat and critical heat flux were measured in the range of mass fluxes from 600 to 1600 kg/m²s and in the range of heat fluxes from 5 to 120 W/cm². For heat flux greater than 60 W/cm², nucleate boiling suppression has significant effect on the flow boiling heat transfer, and this leads to decrease of the heat transfer coefficient with heat flux grows.

1 Introduction

Studies of the processes of heat and mass transfer in microsystems with phase change are fast-developing field of science. This is due to the growth of industrial applications, which require the transfer of large heat fluxes in a limited volume, including compact vapor generators/condensers of cryogenic devices and microprocessor cooling systems [1]. During channel size reduction, a wide variety of phenomena arise which are not typical for conventional tubes. That confirmed by experimental data from [2-6] and other papers. In spite of a large number of published methods for predicting flow boiling heat transfer in microchannels, the application of these methods requires verification, and the determination of heat transfer coefficient for various heat transfer modes remains actual [7]. The objective of this study is experimentally establishing how the average heat flux correlate with wall superheat during flow boiling of refrigerant R236fa and determining the dependence of critical heat fluxes (CHF) from mass flux in a horizontal microchannel heat sink.

2 Experimental setup

The setup for investigation of flow boiling heat transfer and critical heat flux is shown in Fig. 1. Liquid refrigerant R236fa is supplied from the condenser through the filter and damper of pulsation to the flow controller Bronkhorst HI-TECH via the plunger pump.

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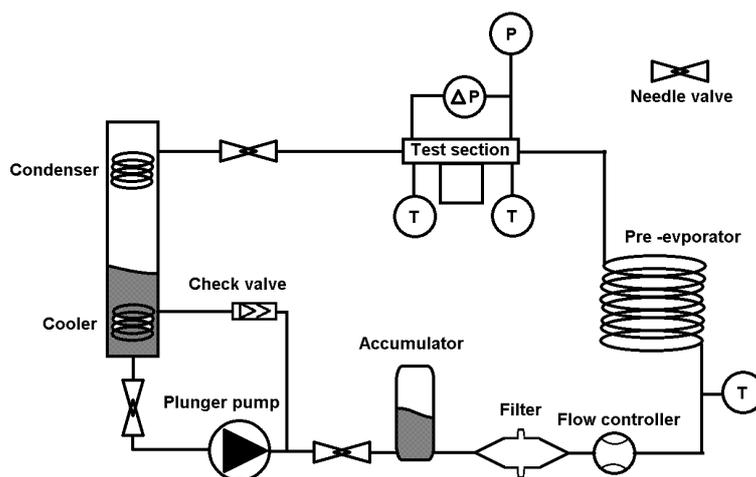


Fig. 1 Schematic of the experimental setup.

Then the refrigerant goes through the pre-evaporator to achieve the flow with desired vapor quality or temperature. After passing through test section it goes into the condenser.

A test section, which was used to investigate flow boiling heat transfer and CHF, is presented in Fig. 2. The oxygen-free copper block has two micro-channels having cross section $2\text{ mm} \times 0.4\text{ mm}$ milled with channel-to-channel spacing equals to 2 mm . The length of microchannels equals to 16 mm , the channel aspect ratio equals to 5. The surface of the microchannels was treated with sandpaper and the measured average surface roughness R_a equals to $0.6\text{ }\mu\text{m}$. The cross-section of top area of the copper block is $10\text{ mm} \times 16\text{ mm}$. The thermal conductivity of copper equals to 380 W/m K at 20°C . Cartridge heaters are mounted into copper block as it shown in Fig. 2 and total power of cartridges is 1200 W . Two rows of two thermocouples are placed into each sides of the copper block along the length of microchannels. The thermocouple rows are 4.5 mm apart from each other, and the top thermocouples are placed at 1 mm below the surface of the microchannels. The thermocouples from all rows are placed at the same locations along the length of the microchannels. The locations, as measured from the inlet of the microchannels and along

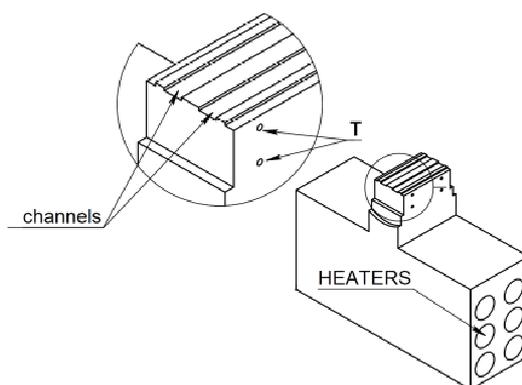


Fig. 2 Schematic of the test section used to flow boiling heat transfer measurement.

their length are 3 mm and 13 mm. While mounting, microchannels were covered by polished stainless steel plate and closed by the stainless steel shell trough fluoroplastic gasket. The inlet and outlet temperature and pressure are measuring inside inlet and outlet chambers, which are separated from copper block trough thermal insulating plate. The inlet and outlet chambers have a cylindrical shape with a diameter of 8 mm and a length of 6 mm. The entrance of the microchannels is located in the middle of the inlet chamber. The experiments were carried out with the horizontal orientation of microchannels. The test section is thermo insulated and the heat losses from the test section were calibrated and did not exceed 0.19 W/K. External local heat flux at the top of the copper block was determined from the measured temperature gradient $\nabla T_{w,i}$ between the thermocouples rows in the copper. The dependence of copper thermal conductivity k_{Cu} from temperature was taking into account. The average heat flux to the inner wall of microchannels q_w was determined as follows

$$q_w = \frac{A_{hs}}{A_{in}} \frac{1}{N} \sum_{i=1}^N k_{Cu} \nabla T_{w,i} \quad (1)$$

Here, A_{hs} is the cross-section area of microchannel plate and A_{in} is the inner area of microchannels. Accounting the measured temperature gradient in copper block, the microchannels surface temperatures were determined and averaged wall temperature was obtained.

3 Results and discussion

The experimental data on the dependence of average heat flux from the average wall superheat for three mass fluxes at saturation temperature 30 and 53 °C and inlet liquid temperature of 25 °C are presented in Fig.3. The calculations according to pool boiling heat transfer model [8] (solid line) and single-phase convection models (dashed lines) are presented in Fig. 3 also. For calculation of laminar single-phase convection under condition of three- side heating for the microchannel with aspect ratio equals to 5, the data from [9] was used taking into account thermal developing flow according to [10]. For calculations

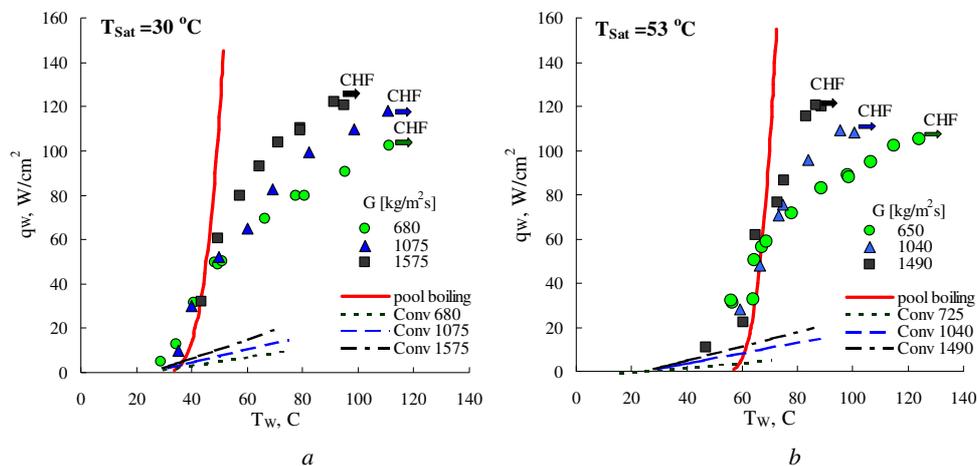


Fig. 3. The dependence of heat fluxes from wall superheat under saturation temperature: *a* correspond to 30°C and *b* to 53°C, inlet temperature equals to 25°C.

of convective turbulent heat transfer, the Gnielinski equation was used taking into account thermal developing flow and asymmetry heat flux around the channel cross section according to [11]. It was taken into account also that Reynolds number for single-phase flow increases along the channel because the liquid viscosity is dependent from temperature.

The experimental data in Fig. 3 correspond to convective heat transfer calculations only for wall temperature lower than saturation temperature of the refrigerant. For developed flow boiling, the experimental data correspond to pool boiling calculations [8] but suppression of nucleate boiling is observed when heat flux higher than 60 W/cm^2 . As is seen from data presented in the Fig.3, this suppression depends both from mass flux and initial subcooling.

The experimental data on the dependence of CHF from mass flux presented in Fig. 4 show the low dependence of CHF from initial subcooling and static pressure. As is seen, the calculations according to the model [12] presented as dashed lines underpredict the experimental data. In spite of absences of the influence of the initial subcooling on CHF in [12], this model shows a stronger dependence of CHF from saturation temperature than observed in the experiments.

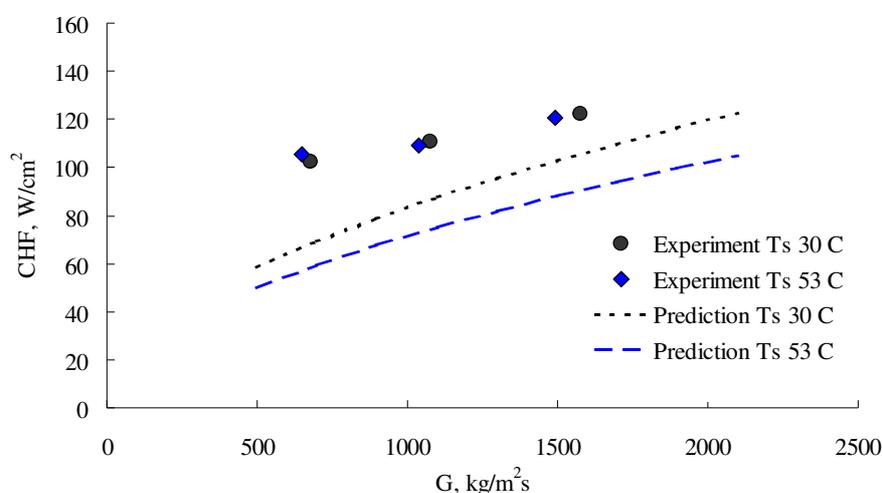


Fig. 4. The dependence of CHF from mass flux under inlet temperature is 25 C for different saturation temperatures.

4 Conclusion

The average heat transfer coefficients and critical heat flux were measured in the range of mass fluxes from 600 to 1600 $\text{kg/m}^2\text{s}$ and in the range of heat fluxes from 5 to 120 W/cm^2 . It was shown that for heat flux greater than 60 W/cm^2 , nucleate boiling suppression has significant effect on heat transfer, and this leads to a decrease of the heat transfer coefficient with heat flux grows. New CHF data have been obtained for short three side heated microchannels with large aspect ratio. A weak dependence of the CHF from the initial subcooling and static pressure was observed.

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