Fundamental issues related to flow boiling and condensation in microchannels – experimental challenges and opportunities

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Abstract. In recent years, application of the microchannels has received considerable attention due to their capability for thermal management of microelectronic equipment, high power devices and development of compact evaporators. The present paper focuses at a discussion of the heat transfer during flow boiling and condensation in microchannels of the heat sink to identify the methods to improve heat transfer performance and select the physically based models for heat transfer prediction. The main purpose of this study is to discuss the mechanisms of heat transfer in microchannels at different aspect ratio for the three-sided heating case using the experimental data were obtained for refrigerant R134a and dielectric fluid.

1. Introduction
The ability of cooling system to remove sufficient heat is important for the design of the advanced electronic components, high-performance computer chips and high heat-load optical systems. Nowadays, the advanced electronic components generate heat fluxes higher then 100 W/cm², while some future electronic components, such as high-power laser and electronic radar systems, have been projected to generate heat fluxes over 1000 W/cm² [1]. To dissipate large amount of heat, flow boiling can be organized inside the multiple parallel microchannels named as microchannel heat sink [2]. Microchannel heat sink that uses refrigerants or water as a coolant allows absorbing high heat fluxes due to the latent heat of vaporization and maintains the desired temperature control. Considerable limitation of two-phase cooling is an existence of critical heat flux (CHF) refers to the maximum heat flux just before the boiling crisis occurs. The CHF can be increased in case of surface (subcooled) boiling [3] when a sudden increase of surface temperature will occur at higher heat flux than for saturated boiling. The use of water for electronic equipment cooling is limited by the high boiling point and more promising is the use of refrigerants or dielectric fluids, such as FC-72, that have a lower boiling point.

It is well known that the flow boiling heat transfer coefficient increases for refrigerants with increasing heat flux [4, 5], but in some experiments it was shown that the heat transfer coefficient depends on the vapor quality, showing growth [6] or reduction [7] when the quality is increased. The difficulties in selection of the reliable correlations for predicting the flow boiling heat transfer constrain the development of two-phase microchannel heat sinks and micro tube evaporators. This problem was discussed in many studies, e.g. [5], however, the flow boiling heat transfer in microchannels remains understood incompletely.

It is obvious that the design of a small scale two-phase cooling systems needs to development of the
microchannel condensers with high efficiency [8, 9]. Microchannels increase the shear stress exerted upon the film interface, that greatly decreases the film thickness and resulting in very high condensation heat transfer coefficients [8]. The mechanism of condensation in microchannels depends on the relative importance of the surface tension, gravity and shear forces, which, in turn, depend on several parameters, such as vapor quality, mass flux, fluid properties and channel geometry [9].

The objective of this paper is the experimental consideration of the mechanisms of flow boiling and condensation heat transfer in microchannels with different aspect ratio for the three-sided heating case. Existing models for predicting the heat transfer coefficient inside rectangular microchannels were verified using the experimental data obtained for refrigerant R134a and dielectric fluid perfluorohexane.

2. Experimental equipment and methods

Experiments were performed in the closed loop presented in detail in [10]. The working liquid is supplied from the condenser through the filter and damper of pulsation to the flow controller via the plunger pump. The desired inlet vapor quality is obtained when the fluid goes through the pre-evaporator placed before the test section. Two experimental test sections were used to investigate the flow boiling and condensation heat transfer phenomena in the microchannel heat sink. The heat sink with the channels having a low aspect ratio is shown in figure 1a. The oxygen-free copper microchannel plate and heating block are soldered by tin to the stainless steel shell. There are twenty-one microchannels on the plate. The cross-section of each microchannel is 0.335 µm × 0.930 µm and the spacing between the channels is 0.650 µm, the aspect ratio α=a/b is 0.36, where a is the width and b is the height of the channel. The size of the microchannel plate is 20 × 40 × 2.5 mm. Two layers of the thermocouples are placed along the microchannels into the solder on both sides of the partition sheet made from stainless steel. Their locations, measured from the inlet of the microchannels, are 5 mm, 15 mm, 25 mm, and 35 mm. After mounting into the stainless steel shell, the microchannel plate is covered by thin sheet of copper and stainless steel plate. The inlet and outlet temperature and pressure are measured inside inlet and outlet chambers.

Heat is supplied to thermo-insulated test section from the heat cartridges mounted in the copper block as shown in figure 1a. In the case of flow condensation, heat cartridge is replaced by the Peltier module and external water heat exchanger. External local heat flux that passes into microchannel plate is determined from temperature difference \( \Delta T_{\text{w,i}} \) across the partition sheet. The heat flux on the wall of the microchannel \( q_{\text{w,i}} \) (internal heat flux) is calculated accounting the difference in the internal and base surface area of the microchannel plate. The internal wall temperatures \( T_{\text{w,i}} \) is determined accounting the local heat flux and measured temperature \( T_{\text{m,i}} \). The saturation temperature related to \( i\)-th thermocouple is calculated using linear approximation of the measured input and output pressure, and accounting the dependence of the saturation temperature on pressure for working liquid.

The heat sink with two microchannels having a high aspect ratio is shown in figure 1b. The
microchannels with cross section 2 mm × 0.36 mm were milled on the top of the copper block made from oxygen free copper with channel-to-channel spacing equal to 2 mm and covered by a thin nickel layer using electrochemical treatment. The length of micro-channels is 16 mm, the surface roughness Ra is 0.67 μm, the aspect ratio of the channels is 5.56. Heat cartridges are mounted into the copper block as is shown in figure 1b. Two layers of four thermocouples are placed into the copper block along the length of microchannels. The thermocouples locations, measured from the inlet of the microchannels, are 3 mm and 13 mm. The thermocouple layers are placed at distance 4.5 mm from each other and the top layer is placed at 1.5 mm below the surface of the microchannels. The microchannels are covered by a polished stainless steel plate. The inlet and outlet chambers have a cylindrical shape with a diameter of 8 mm and a length of 6 mm. The external local heat flux to the microchannel plate is determined from the measured temperature gradient in the copper. The inner wall temperature is calculated using a linear approximation of the temperature gradient in the copper block. The vapor quality at the inlet was determined considering heat production in the pre-evaporator $Q_{col}$.

3. Flow boiling heat transfer

Experimental data on local heat transfer coefficients for R-134a flow boiling in a microchannel heat sink with microchannels having a low aspect ratio (figure 1a) were obtained in the range of internal heat flux from 20 to 500 kW/m$^2$ and mass flux $G$ from 300 to 600 kg/m$^2$s. The local thermodynamic vapor quality was varied from -0.02 to 1.2 and static pressure from 6 to 16 bars. The heat transfer coefficient was determined accounting the fin efficiency that exceeded 0.98. The experimental data show a considerable effect of heat flux on the magnitude of the heat transfer coefficient. It shows that nucleate boiling is the dominant mechanism for heat transfer. Nevertheless, it was obtained that the pool boiling equation cannot be used for predicting flow boiling heat transfer. The comparison of experimental data for saturated flow boiling of R-134a in copper microchannel heat sink with prediction according to the equation for pool boiling on cooper surface [11] is shown in figure 2a. The effect of vapor quality $x$ on the heat transfer coefficient is presented for the average mass flux of 380 kg/m$^2$s, saturation temperature 24.2 °C and heat flux on internal channel surface of 22.3 kW/m$^2$. Points show experimental data, the calculation results according to [11] 

$$h_{w boil} = 100q_w^mP_r^{0.45}(-\log(p_r))^{0.8}K_u^{0.2}M^{-0.5}$$

Figure 2. Heat transfer coefficient vs. vapor quality for heat flux of 22.3 kW/m$^2$ (a), and vs. heat flux for $x < 0.3$ and $G = 550$ kg/m$^2$s (b). Points show experimental data, solid lines show the calculation results according to [10], dotted and dashed lines according to [11] and [12].
are shown as a dotted line, were $R_a$ is roughness, $m = 0.9 - 0.3p_r^{3/2}$ and $p_r = p/p_{cr}$ is the reduced pressure, the dashed line corresponds to the calculations according to [12]. As is seen, the pool boiling equation considerably underpredicts the experimental data, especially at high vapor quality. The reason for the increase of heat transfer coefficient in compare with pool boiling model is the importance of the evaporation mechanism of heat transfer because in microchannels the liquid film becomes very thin due to high interfacial shear stress. It should be noted that increasing the heat flux for the same mass flux causes the reduction of the heat transfer due to nucleate boiling suppression.

The data were obtained show that reliable approach for predicting the flow boiling heat transfer in microchannels should account for the determining mechanisms such as nucleate boiling suppression occurs when the liquid film thickness becomes sufficiently small, forced convection and evaporation of a thin liquid film. The physical basis of the approach proposed in [10] consists in accounting all these mechanisms in one general equation. The proposed method has the privilege over simple summing of corresponding terms because it allows clearly distinguishing the contribution of each term under the conditions when it is predominant. To determine the heat transfer coefficient for the evaporating wavy-turbulent liquid film, the model of [13] was used in this paper with the calculation of the shear stress according to [14]. The comparison of the calculation results on dependence of heat transfer coefficient from heat flux with experimental data obtained for $G = 550 \text{ kg/m}^2\text{s}$, $p_r = 0.22$ and vapor quality less then 0.3 is presented on figure 2b. One can see that model [10] better suits the experimental data both for low and high heat fluxes in compare with pool boiling model. It happens so because the proposed model takes into account both the nucleate boiling suppression and thin-film evaporation.

Measurements of the dependence of internal averaged heat flux on wall superheat during R-134a flow boiling in the microchannels having a high aspect ratio (figure 1b) are presented in figure 3a. The data were obtained for saturated and subcooled conditions at the inlet in the range of internal heat flux from 50 to 500 kW/m$^2$ and mass flux of 840 kg/m$^2$s at static pressure 7.3 and 7.35 bar accordingly. For flow boiling of refrigerant R-134a, the visible impact of heat flux on the magnitude of wall superheat was observed. It was obtained that the equation [11] for pool boiling heat transfer cannot be used for prediction of the flow boiling heat transfer in saturated conditions. Points in figure 3a shows the experimental data, dotted line show the result of calculations according to [11]. As is seen, good agreement with experimental data has the model [10] accounted for both the nucleate boiling suppression and evaporation of liquid film. It is also seen that in a subcooled region, the experimental data are above the calculation results that shows enhancement of heat transfer due to liquid film.
evaporation. It happens because after nucleation the produced vapor remains as the bubbles attached to the surface while growing and collapsing. In the region of saturated boiling, the experimental data are below the calculations due to increasing the level of wall temperature fluctuations that produced unsteady conditions inside the microchannels, see figure 3b.

4. Flow condensation heat transfer

Measurements of the local heat transfer coefficients during flow condensation of dielectric fluid perfluorohexane in the microchannels having a low aspect ratio (figure 1a) are shown in figure 4a. During condensation, the heat was removed from the wall of the test section using Peltier modules that allow varying local wall subcooling. The data were obtained in the range of wall subcooling from 1 to 8 °C and mass flux from 100 to 200 kg/m²s. The local vapor quality was varied from 0.1 to 0.92 and static pressure from 1 to 1.2 bars. The heat transfer coefficient was determined accounting the fin efficiency that exceeded 0.98. As is seen, the condensation heat transfer coefficient is highest near the channel inlet and decreases along the microchannels. Increasing the mass velocity increases the condensation heat transfer coefficient due to thinning the liquid film when the vapor interfacial shear stress is increased. For flow condensation of perfluorohexane, the impact of wall subcooling on the magnitude of the heat transfer coefficient was not observed. It shows a considerable influence of the capillary forces on local film thickness that was discussed in [15]. The comparison of experimental data with predictions according to the model [8] for the studied range of mass flux and local vapour quality is shown in figure 4b. One can see that calculation results are in good agreement with the experimental data. The deviation of experimental data from calculations is caused by the transition to elongated bubble flow.

5. Conclusions

The presented results show that saturated and subcooled flow boiling of refrigerant R134a in microchannels with low and high aspect ratio for the three-sided heating case is characterized by the visual impact of heat flux on the magnitude of heat transfer coefficient. Nevertheless, the pool boiling equation cannot be used for prediction of the flow boiling heat transfer. The reason for the increase of heat transfer coefficient in comparison with the pool boiling model is the importance of the evaporation mechanism of heat transfer because the liquid film becomes very thin at high interfacial shear stress. Therefore, the reliable approach for predicting the flow boiling heat transfer should account for the determining mechanisms such as nucleate boiling suppression occurs when the liquid film thickness
becomes sufficiently small, forced convection and evaporation of a thin liquid film. Comparison of the experimental data with the calculations according to Kuznetsov and Shamirzaev model [10] that accounts these mechanisms shows better suits the experimental data both for low and high heat fluxes.

The experimental data on flow condensation of dielectric fluid perfluorohexane in microchannels at low aspect ratio for the three-sided cooling case shows that the impact of wall subcooling on the magnitude of the heat transfer coefficient is not observed. It was obtained than increasing the mass velocity enhances the condensation heat transfer due to thinning the liquid film when the vapor interfacial shear stress is increased. The comparison of the experimental data with prediction according to the Kim and Mudawar [8] model shows that the calculation results are in good agreement with the experimental data. The deviation of experimental data from calculations occurs at the transition to elongated bubble flow.

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