Experimental study of condensation of dielectric liquid in microchannel heat exchanger

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Abstract. This study presents the results of an experimental investigation of the heat transfer under condensation of perfluorohexane inside the horizontal multi-microchannel system. The copper heat exchanger contains 21 rectangular microchannels with 335x930 μm cross-section. The heat from the condensing flow is rejected to a flow of water by Peltier modules. Experiments are performed in a range of mass fluxes from 90 to 200 kg/m²s and vapor qualities from 1 to 0. Obtained data set on heat transfer is compared with theoretical predictions.

1. Introduction
Removing high heat fluxes from small areas with uniform surface temperature is one of the greatest challenges in the thermal design of semiconductor electronic devices. Two-phase flow boiling in microchannels is one of the promising cooling techniques that can dissipate high heat fluxes at nearly uniform surface temperature. For the design of a small scale pumped loop-cooling system the compact condensers with high effectiveness are needed [1, 2].

The dielectric fluids are widely used for cooling electronic components but they have low thermal conductivity, and high intensity of heat transfer is needed. Microchannels greatly increase vapor velocity and, therefore, the shear stress exerted upon the film interface. This greatly decreases the film thickness, resulting in very high condensation heat transfer coefficients [1]. At channel size reduction, a wide variety of phenomena arise which are not typical of conventional tubes. The condensation mechanism 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 [2]. There are limited numbers of studies in the literature focused on the investigation of condensation in multi-microchannel systems. Therefore, the aim of this work is to investigate experimentally heat transfer during flow condensation of dielectric liquid perfluorohexane in a multi-microchannel condenser.

2. Experimental equipment and measurements
Figure 1 shows a schematic of the experimental set-up used in this study for investigation of condensation heat transfer in two-phase perfluorohexane flow in the multi-microchannel condenser. The liquid is supplied from the cooler through the filter and flow controller Bronkhorst HI-TECH to the temperature control system via the pump. The liquid flow rate was measured with an accuracy of 0.02 g/s. Then the perfluorohexane goes through the pre-evaporator to achieve the flow with desired vapor quality. After passing through to the tested multi-microchannel condenser it goes to the condenser.
Figure 1. Schematic of the experimental setup.

Figure 2 shows a schematic of the experimental test section. Twenty-one microchannels were made by precise milling of micro-slots with 335 μm width and 930 μm depth into the top surface of the oxygen-free copper plate of the multi-microchannel condenser with the size of 20x40 mm. The distance between channels equals 650 mm at total plate thickness of 2.5 mm. The microchannel plate was mounted in the stainless steel shell. At the outer side of the shell, the cooling copper block with two Peltier modules and water heat exchangers was mounted. The thickness of the partition wall between the microchannel plate and the cooling block equaled 2.51 mm. Tin gaskets with 0.5 mm thickness and thermocouples inside were placed between the microchannel plate and partition wall as far as between the cooling block and the partition wall. Gaskets were melted during the assembly process to reduce the contact thermal resistance. After mounting, the microchannel plate was closed by the copper cap and sealed by the stainless steel cover.

Figure 2. Schematic of the test section.
During experiments, the temperatures of tin gaskets were measured in four cross-sections along the length of heat sink at distances of 5 mm, 15 mm, 25 mm, and 35 mm from the microchannels inlet, see figure 2. Due to small fin height and high thermal conductivity of copper, the temperature of the internal channel wall was almost uniform in the cross-section. In each cross-section, two thermocouples were placed along the width of the microchannel plate. Pressures and temperatures in the inlet and outlet chambers were measured using the pressure probes and insulated thermocouples. External heat inflow to the test section was calibrated and did not exceed 0.19 W/K.

The local heat transfer coefficient \( h_i \) was determined using the wall superheat as follows:

\[
h_i = q_{w,i} / (T_{w,i} - T_{sat,i})
\]

where \( T_{w,i} \) and \( T_{sat,i} \) are the wall and saturation temperature related to the \( i \)-th thermocouple and \( q_{w,i} \) is the local heat flux from the inner wall to the flow. The saturation temperature related to the \( i \)-th thermocouple was determined using a linear approximation of the measured input and output pressure, and the dependence of the saturation temperature on pressure for the dielectric fluid. Local heat flux was determined from temperature difference \( \Delta T_{w,i} \) through the partition wall as follows:

\[
q^*_{w,i} = \left( \frac{\delta_{st} + \delta_{tin}}{\lambda_{st} + \lambda_{tin}} \right)^{-1} \Delta T_{w,i}, \quad q_{w,i} = q^*_{w,i} \frac{A_{out}}{A_{in}}
\]

Here, \( q^*_{w,i} \) is the external local heat flux, \( \lambda_{st} \) and \( \lambda_{tin} \) are thermal conductivities of stainless steel and tin, respectively, \( \delta_{st} \) and \( \delta_{tin} \) are the thickness of the partition wall and tin gasket, \( A_{out} \) and \( A_{in} \) are the external and internal area of the microchannel plate. The vapor quality at the microchannel plate inlet was determined considering heat production in the pre-evaporator \( Q_{coil} \) and the liquid temperature at the pre-evaporator inlet \( T_0 \) as follows:

\[
x_0 = \left[ Q_{coil} - Q_{lost,cooil} - m \cdot C_{p,liq} (T_{sat,in} - T_0) \right] (m h_{fg})
\]

where \( m \) is the mass flow rate, \( h_{fg} \) is the latent heat of vaporization, and \( C_{p,liq} \) is the specific heat of liquid. The vapor quality along the microchannel plate was calculated as follows:

\[
x_i = x_0 + \left( \int_0^L W(q_w) dl - Q_{lost,he,i} / L \right) (m h_{fg})
\]

where \( L \) is the microchannel plate length, \( Q_{lost,he} \) is the heat loss, \( W \) is the width of heat sink, \( q_w \) is the averaged heat flux supplied to the heat sink, and \( x_0 \) is vapor quality at the heat sink inlet.

The heat removed from test section by Peltier elements \( Q_{cool} \) was calculated using the mass flow of cooling water \( m_{water} \), water heat capacity \( C_{p,water} \), temperature difference water between outlet and inlet of cooling water heat exchangers \( \Delta T_{water} \) and electrical power of Peltier elements \( U_p I_p \) as

\[
Q_{cool} = m_{water} C_{p,water} \Delta T_{water} - U_p I_p
\]

and corresponded to the heat passed through the partition wall between the cooling block and microchannel plate. Experiments were performed in the range of mass fluxes from 90 to 200 kg/m²s and vapor qualities from 1 to 0. The inlet pressure for the heat exchanger was maintained about 1 bar.
3. Results and discussion

The pre-evaporator and Peltier modules allow achieving the vapor qualities independent of wall subcooling in the heat exchanger. Figure 3 shows variation of the measured local heat transfer coefficient with thermodynamic equilibrium quality in the saturated region for two mass fluxes and wall subcooling. The condensation heat transfer coefficient was the highest near the inlet and it decreases along the microchannel. Increasing the mass velocity increases the condensation heat transfer coefficient by thinning the liquid film due to the increased interfacial vapor shear stress. The use of Peltier modules allowed us to obtain data at different wall subcooling without changing the temperature and flow of coolant water. The average wall subcooling for the presented data is shown in figure 3. Dark symbols show the data obtained with the wall subcooling of about 8 degrees Celsius. Light symbols show data obtained at lower wall subcooling. It was found that the local heat transfer coefficient does not depend on the wall subcooling and is determined by the mass flux and local vapor quality.

Figure 4. Variation of condensation heat transfer coefficient with thermodynamic equilibrium quality:
1 – averaged along heat exchanger experimental data; 2 – local experimental data; 3 – calculation by [3]; 4 – calculation by [4]; 5 – calculation by [4].
Variation of the condensation heat transfer coefficient with thermodynamic equilibrium quality was compared with the calculated value according to existing models. Figure 4 shows this comparison for mass flux of 120 kg/m²s. The local value of the measured heat transfer coefficient marked by symbol 1 and the heat transfer coefficient averaged along the heat exchanger length vs. average vapor quality marked by symbol 2 are shown in figure 4. Some approaches to predict the condensation heat transfer coefficient was considered. Simple approximation, which considers annular two-phase flow with uniform liquid film thickness on the perimeter of channels, is presented as line 3 in figure 4. Heat transfer coefficient for this case was calculated through thermal resistance of the liquid film

\[ h = \frac{\lambda_f}{\delta_f} \]

where \( \lambda_f \) is liquid conductivity and film thickness \( \delta_f \) determined by the model [3]. The calculation of heat transfer coefficient for condensation in annular flow for macro channels from [4] is shown in figure 4 as line 4. The model proposed in [5] for condensation of FC72 in parallel square microchannels with hydraulic diameter of 1 mm is shown in figure 4 as line 5. The model of [5] shows the best fit to the experimental data with vapor quality less than 0.7. The predictive accuracy of the correlations was measured by the mean absolute error, defined as

\[ MAE = \frac{1}{N} \sum \frac{|h_{\text{predicted}} - h_{\text{measured}}|}{h_{\text{measured}}} \times 100\% \]  

and estimated for simple approximation based on [3] as 26.8%, for the calculation by [4], it was estimated as 21%, and for the calculation by [5] it was 12.6%.

4. Summary
This study considers the phenomena associated with condensation of dielectric liquid perfluorohexane in microchannels. Heat transfer coefficients for condensation in the multi-microchannel system were measured and compared with theoretical models. The best predictive ability for experimental data was obtained for the model of Kim and Mudawar [5] with an overall MAE of 12.6%, but this model underestimates the measured local values for vapor quality higher than 0.7. The experimental data show independence of the local heat transfer coefficients on wall subcooling and their dependence on the mass flux and local vapor quality.

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