An experimental investigation on condensation of R134a refrigerant in microchannel heat exchanger

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Abstract. This study presents an experimental investigation of heat transfer under condensation inside horizontal multi-microchannel system. R134a refrigerant was used. The copper heat exchanger contains 21 rectangular microchannels with 335x930 μm cross-section. Experiments were performed at two mass fluxes 481 to 830 kg/m²s, and vapor qualities ranged from 0.8 to 0.05. Obtained data set on heat transfer and pressure drop was compared with calculation by theoretical models.

1. Introduction
The dissipation of 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. And for design of a small scale pumped loop-cooling system the compact condensers with high effectiveness are needed [1, 2].

Micro-channels 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]. When reducing a channel size, a wide variety of phenomena arise, which are not typical for conventional tubes. The condensation mechanism 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 number of studies in the literature focused on studying condensation in multi-microchannel systems. Therefore, the aim of this work is to investigate experimentally the condensation efficiency of a multi-microchannel condenser.

2. Experimental equipment and methods
Figure 1 shows a schematic of the experimental set-up used in this study for investigation of condensation heat transfer under two-phase refrigerant flow in the multi-microchannel condenser. Liquid refrigerant 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 accuracy of 0.02 g/s. Then the refrigerant 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 335 μm wide and 930 μm deep micro-slots into the 20x40 mm top surface of the oxygen-free copper plate of the multi-microchannel condenser. The distance between channels equals to 650 μm (fin thickness) 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 elements and water heat exchangers was mounted. The thickness of the partition wall between microchannel plate and 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 stainless steel cover trough fluorooplast gasket.

Figure 2. Schematic of the test section.
During the test, temperatures of tin gaskets were measured in four cross-sections along the heat sink at distances of 5 mm, 15 mm, 25 mm, and 35 mm from the microchannels inlet, see fig.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 chamber were measured using the pressure probes and insulated thermocouples. The heat inflows to the test section were calibrated and did not exceed 0.19 W/K.

The heat transfer coefficient was calculated using internal wall temperatures $T_{w,i}$:

$$h_i = q_{w,i} / (T_{w,i} - T_{\text{sat},i})$$  \hspace{1cm} (1)$$

where $T_{\text{sat},i}$ is the 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 linear approximation of the measured input and output pressure, and the dependence of the saturation temperature on pressure for tested refrigerant. Local heat flux was determined from temperature difference $\Delta T_{w,i}$ through the partition wall as follows:

$$q_{w,i}^* = \left( \frac{\delta_{\text{st}}}{\lambda_{\text{st}}} + \frac{\delta_{\text{tin}}}{\lambda_{\text{tin}}} \right)^{-1} \Delta T_{w,i}, \quad q_{w,i}^* = q_{w,i}^* \frac{A_{\text{out}}}{A_{\text{in}}}$$  \hspace{1cm} (2)$$

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

$$x_0 = \left[ Q_{\text{coil}} - Q_{\text{lost,coil}} - m \cdot C_{p,\text{liq}} (T_{\text{sat,in}} - T_0) \right] / \left( m h_{fg} \right)$$  \hspace{1cm} (3)$$

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

$$x_L = x_0 + \left( \int_0^L W (q_w) dl - Q_{\text{lost,hs}} / L \right) / \left( m h_{fg} \right),$$  \hspace{1cm} (4)$$

where $L$ is the microchannel plate length, $Q_{\text{lost,hs}}$ is the heat loss, $W$ is the heat sink width, $<q_w>$ is the cross section averaged local heat flux supplied to heat sink, and $x_0$ is vapor quality at the heat sink inlet.

The heat removed from test section by Peltier elements $Q_{\text{coolP}}$ was calculated using mass flow of cooling water $m_{\text{water}}$, water heat capacity $C_{p,\text{water}}$, temperature difference of water between outlet and inlet of cooling water heat exchangers $\Delta T_{\text{water}}$, and electrical power of Peltier elements $U_{\text{IP}}$ as

$$Q_{\text{coolP}} = m_{\text{water}} C_{p,\text{water}} \Delta T_{\text{water}} + U_{\text{IP}} I_{\text{P}}$$  \hspace{1cm} (5)$$

and correspond to heat passed trough partition wall between the cooling block and microchannel plate.

$$\left| \int_0^L W <q_w> dl \right| / Q_{\text{coolP}} < 0.05$$  \hspace{1cm} (6)$$
3. Result and Discussion

3.1. Pressure drop

Total pressure drop ($\Delta P_{\text{tot}}$) measured in tested heat exchanger consist of: friction pressure drop inside microchannels ($\Delta P_{f,m}$), friction pressure drop inside inlet ($\Delta P_{f,\text{in}}$), and outlet ($\Delta P_{f,\text{out}}$), chambers, deceleration pressure drop ($\Delta P_{d}$), pressure drop at the channel inlet (contraction pressure drop $\Delta P_{c}$) and outlet (expansion pressure drop $\Delta P_{e}$). It can be calculated as

$$\Delta P_{\text{tot}} = \Delta P_{f,m} + \Delta P_{f,\text{in}} + \Delta P_{f,\text{out}} + \Delta P_{d} + \Delta P_{c} + \Delta P_{e}$$ (7)

Pressure drop caused by abrupt flow area changes (contraction pressure drop and expansion pressure drop) was calculated as proposed in [3]. Vapor quality changing influencing the pressure drop (deceleration pressure drop) was calculated by Zivi correlation [4]. And friction pressure gradient inside microchannels $\left( \frac{dP}{dz} \right)_{f,m}$ for annular flow regimes can be calculated as for ascending flow without entrainment by [5] taking into account the fact that contribution of gravity for horizontal flow can be neglected. The transition conditions to the annular flow pattern can be calculated as proposed in [6]. Vapor and liquid velocities changed along microchannel due to condensation, and thus calculating the friction pressure drop requires integrating the calculated friction pressure gradient over the length of microchannel

$$\Delta P_{f,m} = \int_{0}^{L} \left( \frac{dP}{dz} \right)_{f,m} dz$$ (8)

Friction pressure drop inside inlet and outlet chambers can be calculated by using Lockhart-Martinelly model from [7].

Comparison of experimental total pressure drops and calculated by equation (7) is presented in figure 3. Experimental data well correspond to calculation.

![Figure 3](image_url)

**Figure 3.** Total pressure drop vs. vapor quality for: (a) mass flux 481 kg/m²s; (b) mass flux 830 kg/m²s. Points are experimental data; line corresponds to calculation by equation (7).
3.2. Heat transfer

Figure 4a,b, shows the dependence of heat transfer coefficient on vapor quality at mass fluxes of 481 kg/m²s and 830 kg/m²s. Increase in mass flux causes the increase in heat transfer coefficient. This fact results from shear stress increasing on interphase boundary and a thinner liquid film under large vapor velocity. As far as measured total pressure drop well corresponds to calculations by equation (7), the shear stress (\( \tau \)) and film thickness (\( \delta_f \)) can be calculated by [5]. This calculation was done on the assumption of uniform film distribution along the channel perimeter. Heat transfer coefficient can be calculated as

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

where \( \lambda \) is the liquid heat conductivity. The calculation by equation (9) is shown in figure 4 by dotted lines as Calc 1. This calculation is done supposing that temperature gradient is linear in the uniform laminar liquid film. Calculation by Eq. (9) is less than experimental data. On the other hand, the vapor flow in the channel is turbulent and the temperature gradient in liquid turbulent gas driven film can be nonlinear. The model of condensation heat transfer in horizontal conventional tubes given in [8] takes into account this effect and can be used for calculations. While calculating heat transfer coefficient by [8], the shear stress should be calculated by [5]. The calculation by [8] is presented in figure 4 by solid line as Calc 2. This calculation corresponds to experimental data better than Calc 1, but overpredicts experimental data.

4. Conclusion

This study concerns phenomena associated with condensation in microchannels. Pressure drop and heat transfer coefficients under condensation of refrigerant R134a in multi-microchannel system were measured and compared with calculations according to theoretical models. Pressure drop well corresponds to considered models. Mean absolute error of calculation of heat transfer coefficient using liquid film thickness by [5] is 45%. It is proposed to calculate condensation heat transfer coefficient based on model [8] taking into account shear stress from [5]. Mean absolute error of the calculation for the proposed model from the measured values was 18%.
5. Acknowledgement

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6. References

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