Numerical study on condensation of R134a refrigerant in microchannel

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Abstract. The action of capillary forces during flow condensation of refrigerant R134a in rectangular microchannel is considered numerically using the annular flow model. It was shown that significant accumulation of the liquid occurs near the channel corners and causes the heat transfer enhancement. The heat transfer related to the condensation is taken into account assuming the interface temperature is at saturation. The numerical method is validated against experiments and well predicts the flow patterns inside the microchannel and average heat transfer coefficients.

1. Introduction
Due to a rapid growth of the applications that require transfer of the large amount of heat from small areas with uniform surface temperature, more and more attention has recently been paid to design of the microchannel cooling systems. Realization of two-phase flow boiling in these systems requires development of high effective compact condensers based on microchannel technology [1]. Compact condensers have microchannels with different cross sections having rectangular, square, and circular shape. The experiments with horizontal rectangular and square channels are reported in [2, 3] for condensation of refrigerants R134a, R32, R1234ze, and R410A in multiport tubes. The correlations are proposed in [2] to predict the heat transfer coefficients for uniform film thickness in annular flow. A heat transfer model for condensation heat transfer in rectangular minichannels is developed in [3] considering the effects of vapor shear stress and surface tension. Nevertheless, the action of capillary forces in small size channels requires more detailed consideration, since they can cause significant redistribution of the liquid flow and the heat transfer enhancement. Therefore, the aim of this study is numerical investigation of the characteristics of liquid distribution and local heat transfer during condensation of refrigerant R134a in rectangular microchannel using the model of annular flow.

2. The model formulation
One of the important features of vapor-liquid flow in the rectangular microchannel is liquid accumulation near the channel corners due to capillary forces action. It happens because of curved interface and reduced pressure in the liquid in this area. The condensation in microchannels starts from the high vapor quality and the main flow pattern is the annular flow. The model of the evaporation in rectangular minichannel was proposed in [4] for annular flow. This model can be used also for prediction of the heat transfer during condensation in horizontal microchannel if interface waves are absent. The model is based on the allocation of liquid flow in two zones consist in the flow in a corner.
of the channel bounded by meniscus and film flow on the channel walls. To obtain the resultant interface shape, these solutions match together with account of the conjugation conditions. For thin film area the small parameter exists: \( \varepsilon = \delta_0/a \ll 1 \), where \( \delta_0 \) is the initial film thickness and \( a \) is the half width of the long side of the channel. Using this small parameter and neglecting the gravity term for flow in microchannel, the Navier-Stokes equations and interface boundary conditions can be reduced for film flow area to the equation for prediction of the evolution of interface shape during condensation as follows:

\[
\left( m^i + 1.5 \frac{K}{\varepsilon} \right) \frac{\partial \xi}{\partial x} + \left( m^i m_{\gamma y} \right) \frac{\partial \eta}{\partial y} = \frac{3}{\varepsilon^2} \left( \frac{\gamma}{m} \right) - \frac{G_0 \Theta_{w,i}}{m^e}.
\]

Here \( \gamma = 1 - \left( \frac{dp_{\text{gas}}}{dx} \right) / \rho_{\text{liq}} g \), \( \kappa = \tau / \rho_{\text{liq}} ga \), \( Bo = \rho_{\text{liq}} ga^2 / \sigma \), \( Ga = A_0 \left( 6 \pi a^2 \sigma \right) \), the dimensionless film thickness is scaled via initial film thickness \( m = \delta / \delta_0 \), longitudinal \( x \) and transverse \( y \) dimensionless coordinates are scaled via \( a Bo \varepsilon \) and \( a \) respectively, \( \tau \) is shear stress at the interface, \( A_0 \) is the Gamaker's constant, \( G_0 = \lambda_{\text{liq}} T_{\text{sat}} / (h_{\text{fg}} \sigma a) \) and \( \Theta_{w,i} = (T_{w,i} - T_{\text{sat}}) / T^* \) are determined by temperature of the inner wall \( T_{w,i} \), saturation temperature \( T_{\text{sat}} \) and characteristic temperature which is \( T^* = (T_{w,e} - T_{\text{sat}}) / T^* \) for constant temperature of the external wall \( T_{w,e} \) and \( h_{\text{fg}} \) is the latent heat of vaporization. The shear stress in the interface is selected according to [5]. The disjoining pressure should be account for the walls with a very low roughness, which provides the last terms in the equation (1).

Equation (1) is solved numerically together with the equations for liquid flow in the meniscus and total liquid flow conservation. The boundary conditions are the symmetry conditions at the center of the channel and conditions \( m_y = 0 \) and \( m_{\gamma y} = 1 / (r g) \) at the point of conjunction of interface solutions for the liquid film and the meniscus, where \( r = R / a \). Solving Poisson's equation for the liquid flow in a corner of the channel bounded by meniscus of the given curvature, the dependence of the flow rate on radius curvature of meniscus \( R_m \), its configuration, contact angle and shear stress at the interface was obtained in the form of polynomials in [4]. When interface shape is determined, the Fourier equations for liquid and wall area are solved together to define the temperature field and evaporation rate in assumption of heat conduction through the liquid layer.

3. Results and Discussion
A numerical study of the heat transfer during R-134a condensation in microchannel plate with rectangular microchannels having the cross-section of 335×930 \( \mu \text{m} \) were carried out by the finite difference method. The calculations were done to compare the calculation results with experimental data [6]. The variation of the interface shape for vapor quality \( x_v = 0.772 \) (a) and \( x_v = 0.48 \) at mass flux of 481 kg/m²s and static pressure of 0.74 MPa is shown in figure 1. The calculations were done for the quarter of the channel at uniform initial liquid film thickness and initial vapor quality equals to 0.995. As seen, the preferential accumulation of the liquid in a short side of the channel occurs.

\[ \text{Figure 1. Interface shape for vapor quality } x_v = 0.772 \text{ (a) and } x_v = 0.48 \text{ (b) during condensation with mass flux of 481 kg/m}^2 \text{s.} \]
Simultaneously, very thin liquid film is formed near the meniscus area and deformed liquid film exists on the long side of the microchannel. The distribution of the local heat transfer coefficient along the channel perimeter for vapor quality \( x_v = 0.772 \) (a) and \( x_v = 0.48 \) (b) shown in Figure 2 for the static pressure equals to 0.74 MPa, mass flux equals to 481 kg/m\(^2\)s, surface roughness equals to 2.5 \( \mu \)m. The results of numerical calculation show that non-uniform liquid film thickness produces high value of the heat transfer coefficient near the meniscus. The comparison of the dependence of the averaged local heat transfer coefficient (frequent dotted line) and the effective heat transfer coefficient (solid line) on mass vapor quality for condensation of non-uniform liquid film is shown in Figure 3.
The calculation of heat transfer coefficients for uniform liquid film according to [7] is shown as dashed line. The calculations were performed for mass flux of 481 kg/m²s and static pressure of 0.74 MPa. The effective heat transfer coefficient was determined from heat flux through the base of microchannel plate using the fin efficiency model. Heat transfer coefficients for uniform film were determined using the model of turbulent liquid film [7] with account of interface shear stress from [5] in the regime of ripple wave.

The experimental data from [6] are shown as the rhombs in figure 3. These data were obtained using the fin efficiency model. As seen, for vapor quality higher than 0.4, the numerical results correspond to the experimental data. The calculations for less vapor qualities do not make sense, since the transition to flow with elongated bubbles occurs. The calculations according to uniform film model over predict the data but can be used for estimation of the averaged heat transfer coefficient. Nevertheless, this model does not account the fundamental features of the flow such as existence of meniscus near the channel corners. The above results of numerical calculations show that non-uniform flow of condensate along the perimeter of the microchannel due to predominant influence of the capillary forces causes considerable enhancement of heat transfer near the channel corners, this compensates heat transfer deterioration in the meniscus area.

4. Conclusions
The action of capillary forces in rectangular microchannel requires more detailed consideration, since they can redistribute the liquid in channel cross-section. The numerical calculations according to the annular flow model show significant accumulation of the liquid near the channel corners during condensation and heat transfer enhancement in this area. The numerical method is validated against experiments and well predicts the average heat transfer coefficients. The calculations according to turbulent film model with uniform film thickness over predict the experimental data but can be used for estimation of the averaged heat transfer coefficient. The results of numerical calculations show that predominant influence of the capillary forces causes the formation of non-uniform flow of condensate in the microchannel cross-section. This produces considerable enhancement of the heat transfer near the channel corners and deterioration of heat transfer in the meniscus area.

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