Capillary hydrodynamics and transport processes during phase change in microscale systems

V V Kuznetsov

1 Institute of Thermophysics Siberian Branch of Russian Academy of Sciences
1, Academician Lavrentyev Av., Novosibirsk, 630090, Russian Federation
2 Novosibirsk State University, 2, Pirogova Str., Novosibirsk, 630090, Russian Federation
E-mail: vladkuz@itp.nsc.ru

Abstract. The characteristics of two-phase gas-liquid flow and heat transfer during flow boiling and condensing in micro-scale heat exchangers are discussed in this paper. The results of numerical simulation of the evaporating liquid film flowing downward in rectangular minichannel of the two-phase compact heat exchanger are presented and the peculiarities of microscale heat transport in annular flow with phase changes are discussed. Presented model accounts the capillarity induced transverse flow of liquid and predicts the microscale heat transport processes when the nucleate boiling becomes suppressed. The simultaneous influence of the forced convection, nucleate boiling and liquid film evaporation during flow boiling in plate-fin heat exchangers is considered. The equation for prediction of the flow boiling heat transfer at low flux conditions is presented and verified using experimental data.

1. Introduction

Due to a rapid growth of the applications that require transport of the large amount of heat in the limited space, more and more attention has recently been paid to the development of micro-scale heat exchangers. These heat exchangers are characterized by extremely large ratio of surface area to volume and traditionally classified as compact heat exchangers. Two-phase compact heat exchangers offer ample opportunities for engineering applications such as high compact generators/condensers of power and cryogenic units and natural gas liquefaction units. The mechanisms of microscale heat transfer during flow boiling and condensation in mini/micro-channels were investigated, for example, in [1, 2] and are fairly inconsistent especially for flow boiling. It has been found in a number of studies that heat transfer during flow boiling in mini- and microchannels is determined by the heat flux and depends only weakly on the mass flux [3]. Experimental study of heat transfer during boiling and condensing of cryogenic fluids, hydrocarbons and refrigerants in the plate-fin heat exchangers with minichannels showed weak dependence of heat transfer coefficient on the heat flux as it shown in [4]. Liquid film flow, which provides large heat flux at small temperature difference, deserves special attention in the analysis of heat transport in microscale systems [5]. Published methods for predicting the heat transfer with flow boiling in micro-scale heat exchangers are based either on certain modifications of models that characterize boiling in conventional tubes [1] or on specially developed correlations [3], but their application in various heat-exchange regimes was hardly a success. In particular, this applies to cryogenic compact heat exchangers operated at low mass and heat fluxes.

The aim of the present study is revealing the mechanisms of fluid flow and microscale heat transfer for boiling in downward flow in the plate-fin heat exchanger with minichannels when the value of heat
flux is not sufficient to support active nucleate boiling. The characteristics of gas-liquid flow and heat transport during flow boiling of refrigerant R-21, resulting from the numerical studies, are discussed. This refrigerant has similar thermophysical properties as the cryogenic liquids at low temperatures. The results of numerical modelling of the liquid evaporation in rectangular minichannel are presented for annular flow. The equation for prediction of flow boiling heat transfer in plate-fin heat exchanger at low flux conditions that accounts simultaneously the influence of the forced convection, nucleate boiling and liquid evaporation is presented. The results for the calculation of heat transfer during flow boiling are compared with the experimental data.

2. Capillary hydrodynamics and heat transfer during evaporation in the annular flow

Identification of the characteristics of gas-liquid (vapor-liquid) flow and heat transport during flow boiling and condensing plays a key role in justification of the micro-scale or microstructured heat exchangers. In these heat exchangers the fluids flow in the channels with typical lateral dimensions below the capillary constant. The gas-liquid flow in these channels is characterized by considerable influence of the capillary forces on flow patterns and the gas bubbles are elongated due to the restriction of channel wall. In plate-fin heat exchangers, the form of the mini- and microchannels usually is closed to a rectangle. Three basic flow patterns such as transition flow with elongated bubbles, annular flow with disturbance waves and annular flow with liquid meniscus near the channel corners were observed in [6] during adiabatic experiments in plate-fin heat exchanger with downward flow of refrigerant R-21. For relatively small heat fluxes, liquid evaporation is one of the most promising heat transfer mechanisms. For this case, the heat transport from the wall in annular flow is determined by liquid film thickness distribution along the channel perimeter. The gas and liquid superficial velocities when the transition to annular flow occurs can be determined from the model of equivalent film thickness [7]. In this model, it is assumed that the transition to the annular flow occurs if the film thickness $\delta_{eb}$ in elongated bubble $\delta_{eb}$ or transition flow is equaled to the film thickness in the annular flow $\delta_{lim}$

$$\delta_{eb} = \delta_{lim}$$

This means that the gas velocity in a channel becomes sufficiently high to move all liquid in the film under the action of shear stress and the formation of liquid plugs is not occurs. For elongated bubble flow, the residual film thickness $\delta_{eb}$ can be determined using the “Taylor” law from [8] as follows:

$$\delta_{eb}/D = 0.67Ca^{2/3}\left[1 + 3.35Ca^{2/3}\right]$$

(2)

where $D$ is the tube diameter, $Ca = U_b \mu_{liq}/\sigma$ is the capillary number defined by bubble velocity $U_b$, liquid viscosity $\mu_{liq}$ and surface tension $\sigma$. For rectangular channel, the hydraulic channel diameter $D_h$ can be used in the equation (2) instead of $D$ as it was shown in [9]. Corresponding film thickness in the annular flow $\delta_{lim}$ can be calculated using the annular flow model of Asali et al. [10] without liquid entrainment.

Let’s consider the gas-liquid flow and heat transfer during liquid evaporation in the rectangular minichannel of the plate-fin heat exchanger. In annular flow, the capillary forces are crucial for liquid flow distribution and move liquid into the corners of the channel forming a thin film on the walls. The model of the creeping flow for annular flow in rectangular channel [11] is based on the allocation of liquid flow in two zones of the flow: flow in a corner of the channel bounded by meniscus and thin film flow on the channel wall; and matching solutions for these zones with the conditions of their conjugation. In approximation of a thin layer there is a small parameter $\varepsilon = \delta_{eb}/a \ll 1$, where $\delta_{eb}$ is the initial film thickness and $a$ is the half width of the long side of the channel. For this approximation, the Navier-Stokes equation for the downward flow can be reduced to the dimensionless equation for the local thickness of curved film with a last term which takes into account the evaporation of liquid

$$(m^1 + 1.5 m^2) + (m^m m^m) = \frac{3}{\varepsilon} \left( Ga\frac{m^m}{m^m} - \frac{G\delta_{eb}}{m^m} \right)$$

(3)

Here $\gamma = 1 - (d \rho_{gas}/dx)/\rho_{liq} g$, $\kappa = \tau/\rho_{liq} ga$, $Bo = \rho_{liq} ga^2/\sigma$, $Ga = A\gamma/((6m^2\sigma)$, the dimensionless film
thickness is scaled via initial film thickness $m=\delta_0$, longitudinal $x$ and transverse $y$ coordinates are scaled via $a \delta_0/\epsilon$ and $a$ respectively, $\tau$ is shear stress at the interface, $A_0$ is the Gamaker's constant, $G_0 = \lambda_{liq} \Delta T_0 / (\delta_0 \rho_0)$ and $\Theta_{a,i}=(T_{w,i}-T_{sat})/T_a$ are determined by temperature of the inner wall $T_{w,i}$, saturation temperature $T_{sat}$ and characteristic temperature $T_0 = H \cdot a^2 / \lambda_{liq}$ for the volume heat density $H$ in the wall area, $h_{tg}$ is the latent heat of vaporization.

Equation (3) is solved together with the equations for liquid flow in the meniscus and total liquid flow rate conservation. The boundary conditions are the symmetry conditions at the center of the channel and conditions $m=0$ and $m_y=1/(r \epsilon)$ at the point of conjunction of interface solutions for the film and the meniscus, where $r=R/a$. The singularity due to the fact that thickness of the film streams toward zero in the vicinity of the contact line is eliminated accounting the disjoining pressure. The inspection of the rivulet half-width for the current liquid flow is required to calculate the shape of the interface. When the rivulet half-width becomes less than the distance from the center of the channel to the meniscus, the film rupture is supposed and new configuration of the flow consisting of rivulet, menisci and dry spots between them is established. When the interface shape is obtained, the Fourier equations for liquid and wall areas are solved together to determine the local heat transfer coefficient.

![Figure 1](image1.png)  
**Figure 1.** Liquid interface for $x=0.2$ (a) and $x=0.6$ (b) for R-21 evaporation in the 1.05x6 mm channel with $q_w=1.7$ kW/m$^2$ and $G=50$ kg/m$^2$s.  

![Figure 2](image2.png)  
**Figure 2.** Dependence of average heat transfer coefficient on vapor quality for R-21 evaporation with $q_w=1.7$ kW/m$^2$, $G=50$ kg/m$^2$s.

Figure 1 shows the results of calculation of the liquid interface during R-21 evaporation in the 1.05x6 mm channel with supplied heat flux $q_w=1.7$ kW/m$^2$ and mass flux $G=50$ kg/m$^2$s. The liquid interface is presented for two distances from channel inlet that correspond to vapor quality of $x=0.2$ and $x=0.6$. Formation of dry spots and increasing local heat flux in the vicinity of contact line are typical for evaporation in minichannels. Heat flux in this area is limited by the roughness of the wall equaled to 7.5 $\mu$m. The condensation provokes alignment of the film thickness near the meniscus, and lowers the coefficient of heat transfer in comparison with the ruptured film which is typical for the evaporation. Figure 2 presents the dependence of average heat transfer coefficient on vapor quality for R-21 liquid film evaporation in the 1.05x6 mm rectangular channel with $q_w=1.7$ kW/m$^2$ and $G=50$ kg/m$^2$s. The calculation results show that the mechanism of heat transport during evaporation in micro-scale heat exchangers is conductive heat transfer through curved liquid film and in the vicinity of liquid-vapor-wall contact line. The heat transfer deterioration occurs for vapor quality higher then 0.7, when liquid in meniscus will be evaporated.

3. Flow boiling heat transfer in micro-scale heat exchangers at low flux conditions

The mechanism of flow boiling in plate-fin heat exchangers at low heat and mass fluxes is more complex then only evaporation. The reliable model for prediction of heat transfer should account for the...
determining mechanisms of heat transfer selected in [12] as follows: the nucleate boiling, nucleate boiling suppression, forced convection, evaporative heat transfer in the annular and elongated bubble flows. The physical basis of the approach proposed in this paper consists in integration of all these mechanisms in one general equation. To account the contribution of nucleate boiling, forced convection and liquid film evaporative heat transfer on the two-phase heat transfer coefficient \( h_{lp} \) it is suggested to use the method proposed in [12] as follows:

\[
 h_{lp}^2 = (h_{con} F)^2 + (h_{boil} \Psi_{sup} S)^2 + (E \varphi h_{ev})^2
\]  

(4)

Here \( h_{con} \), \( h_{boil} \) and \( h_{ev} \) are the forced convection, nucleate boiling and liquid film evaporative heat transfer coefficients, accordingly; \( F \) and \( S \) are the factors of forced convection enhancement and nucleate boiling suppression, proposed by Liu & Winterton [13]

\[
 F = \left( 1 + x \frac{\rho_{liq}}{\rho_{gas}} \left( \rho_{liq} / \rho_{gas} - 1 \right) \right)^{0.35}
\]

\[
 S = \left( 1 + 0.055 P_{Re}^{0.16} \right)^{-0.1}
\]

(5)

and \( \Psi_{sup} \) is the boiling suppression factor. Evaporation factor \( E \) corresponds to the relative cross-sectional areas along the flow occupied by the continuous gas core. For the elongated bubble flow and transition flow, evaporation factor can be estimated as vapor volume fraction for homogeneous flow model \( E = (1 + \rho_{gas} (1 - x) / \rho_{liq} x)^{1/2} \), and for annular flow, the gas core is continuous and \( E = 1 \). Parameter \( \varphi \) equals the part of the channel perimeter, where the liquid film exists.

To account the effect of thermo-physical properties of the working fluid and channel dimensions on forced convection heat transfer, the suitable correlation for heat transfer coefficient is used depending on all-liquid flow Reynolds number. For the laminar all-liquid flow, corresponding convective heat transfer coefficient \( h_{con} \) should be calculated accounting the channel geometry and developing flow conditions. For the rectangular channel, the correlating equations of Shah & London [14] for thermally developing laminar flow is used as follows:

\[
 h_{con} = \frac{\lambda_{liq}}{D} \frac{Nu_x}{Nu_{dev} + 6.68} \left( 10^3 z^* \right)^{0.56} e^{-41z^*}
\]

(6)

Here \( z^* = (z / D) / Re_{Sliq} \), \( z \) is the distance from the channel inlet, \( Nu_{dev} \) is the thermally developed heat transfer coefficient. For the rectangular channel, \( Nu_{dev} \) is determined as follows

\[
 Nu_{dev} = 8.235 \left( 1 - 2.042 \gamma + 3.085 \gamma^2 - 2.4765 \gamma^3 + 1.0578 \gamma^4 - 0.1861 \gamma^5 \right), \quad \text{where} \ \gamma = \text{the channel aspect ratio}
\]

(14)

The correct selection of nucleate boiling model \( h_{boil} \) in the equation (4) is very important for reliable heat transfer prediction, especially for the case of dominant nucleate boiling. For nucleate boiling in plate-fin heat exchanger, the Cooper equation [15] is used

\[
 h_{boil} = 55 M^{0.5} P_r^{0.12} \left[ - \log(p_r) \right]^{0.55} R_p^{0.2} \ln(p_r) q_w^{0.666}
\]

(7)

Here \( R_p \) is the surface average roughness and \( p_r = p / p_{cr} \) is the reduced pressure. Nucleate boiling suppression for a thin liquid film is considered in [12] using boiling suppression factor \( \Psi_{sup} \) as the multiplier for nucleate boiling suppression term \( S \) in the equation (4). For rectangular channels, it is necessary to take into account the presence of nucleate boiling suppression in the meniscus area as follows:

\[
 \Psi_{sup} = \left( \alpha_m + (1 - \alpha_m) \Psi_{sup0} \right)
\]

(8)

where \( \alpha_m = 6 R_m / (2a + b) \), \( R_m \) is the meniscus radius for the channel corner, \( b \) and \( a \) are the length of short and long sides of the channel, accordingly.
Figure 3. The dependence of heat transfer coefficient on vapor quality for $G=30$ kg/m$^2$s and $G=50$ kg/m$^2$s. Points are the data [6] and lines are calculation results for $q_w=1.7$ kW/m$^2$.

$$\Psi_{sup} = \tanh^2\left(2.5 \times 10^{-3} e_{sup}^2\right), \quad \theta_{sup} = \left(y_{lv}/d_{lw}\right)^{0.6}/\left(B_o x W e_{lw}^{-0.08} Pr_{lv}^{1/3}\right)$$ is the boiling suppression parameter introduced in [12]. $B_o = q_w/(G \times h_{lv})$ is the Boiling number based on the local liquid flow rate.

In a mini- and microchannels, the liquid film becomes very thin due to the high surface area, and the contribution of evaporation can exceed the contribution of nucleate boiling. For the elongated bubble flow, liquid surface is controlled by capillary forces providing a thin liquid film inside a bubble. In this particular case, the heat transfer coefficient for evaporating liquid film can be calculated as follows:

$$h_e = \left(\lambda_{lw}/\delta_{0h}\right), \quad \delta_{0h}/D_h = 0.67C_d^{2/3}/\left(1 + 3.35C_d^{2/3}\right)$$

(9)

Here, liquid film thickness $\delta_{0h}$ is determined in [8]. To determine the heat transfer coefficient for the evaporating wavy film in annular flow, the model of [16] is used with calculation of the shear stress according to model [10] without entrainment.

The solid line in figure 3 shows the results of calculation of heat transfer coefficient according to the equations (4)–(9) for mass fluxes of $G=30$ kg/m$^2$s and $G=50$ kg/m$^2$s, and heat flux of 1.7 kW/m$^2$ in comparison with experimental data [6] for boiling of R-21 in downward flow in plate-fin heat exchanger. The channels in this heat exchanger have not strictly rectangular cross-section, therefore, the arithmetic mean of the Nusselt numbers for rectangular $Nu_{dev/rec}$ and sine $Nu_{dev/sin}$ ducts from [14]

$$\bar{Nu}_{dev} = (Nu_{dev/rec} + Nu_{dev/sin})/2$$ is used to perform calculations. The calculation results are in an agreement with experimental data up to vapor quality of 0.6 where heat transfer deterioration occurs. The better agreement with the data is observed when heat transfer model is completed by the results of numerical simulation of evaporative heat transfer with liquid film rupture presented in figure 2 instead of uniform film thickness model. Figure 4 presents the values of forced convection, nucleate boiling and evaporation terms in equation (4) during R-21 flow boiling with $G=50$ kg/m$^2$s and $q_w=1.7$ kW/m$^2$. As can be seen in figure 4, the nucleate boiling is not determining mechanism of heat transfer in plate-fin heat exchanger at low flux conditions.

4. Conclusions

Identification of the characteristics of gas-liquid flows and heat transport during flow boiling and condensing plays a key role in justification of the micro-scale heat exchangers. The gas-liquid (vapor-liquid) flow in these exchangers is characterized by considerable influence of the capillary forces on flow patterns. In rectangular channels, the gas bubbles are elongated due to the restriction of channel
wall and a liquid is accumulated near the channel corners both for elongated bubble, transition and annular flows forming thin curved liquid film on the walls. For relatively small heat and mass fluxes, the evaporation of liquid is one of the most promising heat transfer mechanisms for micro-scale heat exchangers and the heat transport from the wall is determined by liquid film thickness distribution along the channel perimeter. The calculations according to model of liquid film evaporation in annular flow show that increasing of local heat flux in the vicinity of contact line for the case of dry spot formation is typical for evaporation in minichannels. In contrast, the condensation provokes alignment of the film thickness near the meniscus, and lowers the coefficient of heat transfer. Nevertheless, the mechanism of flow boiling in plate-fin heat exchangers at low fluxes is more complex than only evaporation. The reliable model for prediction of heat transfer should account for the determining mechanisms of heat transfer including nucleate boiling heat transfer, nucleate boiling suppression, forced convection heat transfer and evaporative heat transfer for elongated bubble, transition and annular flows. It is confirmed by comparison of the results of calculation according to the equation that units of all these mechanisms in one general equation with experimental data. The better agreement with experimental data is observed when heat transfer model is completed by the results of numerical simulation of the evaporative heat transfer with liquid film rupture instead of uniform liquid film thickness model.

Acknowledgments
This work performed in the Kutateladze Institute of Thermophysics SB RAS by a grant from the Russian Science Foundation (project No 14-49-00010).

References
[1] Kandlikar S G 2010 Heat Transfer Eng. 31 159
[2] Kim S M and Mudawar I 2012 Int. J. Heat Mass Transfer 55 984
[3] Bertsch S S, Groll E A and Garimella S V 2009 Int. J. Multiphase Flow 35 142
[4] Kuznetsov V V, Safonov S A and Shamirzaev A S 2015 Tech. Phys. Lett. 41 1124
[5] Pavlenko A N, Li X, Li H, Gao X, Volodin O A, Surtaev A S and Serdyukov V S 2015 Tech. Phys. Lett. 41 774
[6] Kuznetsov V V, Dimov S V and Shamirzaev A S 2017 J. Eng. Thermophys. 26 353
[7] Kuznetsov V V, Kozulin I A and Shamirzaev A S 2012 J. Phys. Conf. Ser. 395 012093
[8] Aussillous P and Quere D 2000 Phys. Fluids 12 2367
[9] Bartkus G V and Kuznetsov V V 2016 MATEC Web Conf. 84 00005.
[10] Asali J C, Hanratty T J and Andreussi P 1985 AIChE J. 31 886
[11] Kuznetsov V V and Safonov S A 2013 Fluid flow and heat transfer with phase change in minichannels and microchannels Heat Pipes and Solid Sorption Transformations: Fundamentals and Practical Applications, eds. L L Vasiliev and S Kakac (Boca Raton: CRC Press) pp 465–496
[12] Kuznetsov V V and Shamirzaev A S 2016 Heat Transf. Eng. 37 1105
[13] Liu Z and Winterton R H S 1991 Int. J. Heat Mass Transfer 34 2759
[14] Shah R K and London A L 1978 Laminar flow forced convection in ducts Advances in Heat Transfer (New York: Academic Press) Suppl. 1
[15] Cooper M G 1984 Advances in heat transfer 16 157
[16] Butterworth D 1974 Int. J. Multiphas. Flow 1 671