Capillary hydrodynamics and heat transfer in two-phase gas-liquid microscale systems

V V Kuznetsov
Kutateladze Institute of Thermophysics SB RAS, 1 Lavrentyev Ave., Novosibirsk, 630090, Russia
E-mail: vladkuz@itp.nsc.ru

Abstract. This paper aims at discussing the hydrodynamics and heat transfer in gas-liquid microscale systems. A new approach for the determination of the flow pattern, based on measuring the statistical characteristics of gas-liquid flow is proposed that can be used for flow pattern map design. The characteristics of heat transfer during upflow boiling in microchannel heat exchangers that operate at low wall superheat are discussed. A novel method for prediction of the flow boiling heat transfer which accounts the heat transfer enhancement due to formation of the thin curved liquid film and surface waves is presented. The characteristics of heat transfer during subcooled flow boiling in microchannel heat sink designed for high heat flux removal are discussed and method for heat transfer prediction is proposed.

1. Introduction
One of the most promising areas of the equipment development in various fields of technology is the use of microscale channel systems, which significantly increases energy transfer efficiency and reduces the overall dimensions. The absence of reliable methods for predicting heat transfer processes in these systems significantly prevents the development of ultra-compact devices for various applications in such areas as distributed, hydrogen and solar energy [1]. The key role in developing the equipment with microscale channels is played by fundamentals of hydrodynamics and heat transfer in microchannels and nanochannels [2]. Microchannels provide an efficient way to remove the heat from concentrated heat sources because they have a large surface area per unit fluid flow volume. The channel cross-section may have different shape, but typically, it is similar to rectangle. During boiling, gas-liquid flow arises in very small channels and the capillary forces strongly influence the flow pattern and heat transfer. It causes difficulties in selection of reliable equations for predicting the flow boiling heat transfer. Despite the fact that this problem was addressed in many studies, the flow boiling in microchannels is still understood incompletely [3]. The determination of the conditions under which the influence of the scale on heat transfer performance during flow boiling becomes decisive is considered in this paper. The characteristics of capillary hydrodynamics of the gas-liquid flows in rectangular microchannels are discussed, and the methods for predicting heat transfer during upflow boiling at low wall superheat in compact heat exchangers and subcooled flow boiling at high heat fluxes in microchannel heat exchanger are developed.

2. Two-phase gas-liquid flow in microchannels
The review on two-phase gas–liquid flow characteristics in microchannels [4] shows that the flow behavior in the microscale channels is significantly different from those in conventional channels. For
this case, the existing approaches do not give an exact criterion for the transition from macro to microscale behavior and provide only an approximate estimation of the size at which this transition occurs. One way to demonstrate the conditions under which microscale behavior becomes essential is considering the motion of a long gas bubble in the microchannel. The Navier-Stokes equations for steady state laminar liquid flow in the system of coordinates associated with the bubble nose are as follows

\[ \rho_{\text{liq}} \mathbf{u} \cdot \nabla \mathbf{u} = -\nabla p_{\text{liq}} + \mu_{\text{liq}} \nabla^2 \mathbf{u} + \rho_{\text{liq}} \mathbf{g}, \quad \nabla \cdot \mathbf{u} = 0. \]  

Here \( \mathbf{u} \) is the liquid velocity, \( p \) is the pressure and \( \mu_{\text{liq}} \) is the liquid viscosity. The conjugation conditions at the interface include the equality of shear and normal stresses at the interface as follows

\[ p_{\text{liq}} = p_{\text{gas}} - \sigma \kappa, \quad \kappa = R_{1}^{-1} + R_{2}^{-1}, \quad e_{i} \tau_{n} = 0, \]  

where \( e \) is the stress tensor in the liquid, \( \mathbf{n} \) is the unit normal vector directed to the gas, \( \tau \) is the unit tangent vector, \( \kappa \) is interface curvature and \( R_{i} \) is the principal radius of curvature. With account of (2), the equation (1) can be reduced to dimensionless variables assuming a constant gas pressure in the bubble

\[ \text{We} \bar{u} \cdot \nabla \bar{u} = -\nabla \bar{\kappa} + \text{Ca} \nabla^2 \bar{u} + \text{Bo} (g / \sigma). \]  

Here, the velocity is nondimensionalized by velocity of the bubble \( u_{b} \), the coordinates are nondimensionalized by hydraulic radius of the channel \( 0.5D_{h} \), the dimensionless numbers \( \text{We} = \rho_{f} u_{b}^2 D_{h} / (2\sigma) \), \( \text{Ca} = \rho_{f} \mu_{f} / \sigma \), \( \text{Bo} = \rho_{f} g D_{h}^2 / (4\sigma) \) are determined by physical properties of the liquid and hydraulic diameter of the channel. One can see that these numbers depend on the channel transverse size and bubble velocity in different way. Therefore, the transition to microscale behavior occurs when the channel size and bubble velocity correspond to \( \text{Bo} \) and \( \text{We} \) much lower than unity, showing the transition to creeping flow in (3).

The experimental setup for studying the characteristics of the gas-liquid flow in rectangular microchannel is described in [5]. To measure local characteristics of the gas-liquid flow, the double laser scanning method and video recording were used. Figure 1 shows the shape of the elongated bubble during nitrogen-water flow in the microchannel with a transverse size of 270 x 360 \( \mu \)m at \( J_{\text{liq}} = 0.07 \text{ m/s}, J_{\text{gas}} = 0.062 \text{ m/s} \) and the shape of the bubble in nanochannel with a transverse size of 50 nm [6]. For a case of microchannel, the dimensionless numbers \( \text{Bo} = 3.18 \cdot 10^{-3}, \text{We} = 3.66 \cdot 10^{-5} \) and

![Figure 1](image_url)
Ca=1.81·10^{-3} are much less than unity and (3) predicts the gas-liquid interface with constant curvature accompanied by extremely thin liquid film, as it is seen in figure 1 (a). The same conclusion can be drawn for the shape of nanobubble shown in figure 1 (b).

When increasing the gas superficial velocity, We number grows and the bubbles interaction arises. As a result, the interphase instability produces in microchannels the new flow patterns such as the transition flow, and annular flow with waves [5]. The boundaries between these flows are blurred and the statistical characteristics of the flow should be applied to determine the boundary between the flow patterns. According to data of [5], the transition from periodic flow with elongated bubbles to transition flow corresponds to a sharp increase in dispersion for the period between the gas bubbles. The transition to annular flow corresponds to the disappearance of liquid plugs that can be registered as a vanishing signal from photodiode at the corresponding level.

3. Heat transfer during evaporation in rectangular microchannels

The problem of identifying the characteristics of heat transfer during flow boiling at small wall superheat is very important for design of two-phase compact heat exchangers for natural gas liquefaction systems and cryogenic equipment. Two-phase compact heat exchangers are manufactured on the basis of microchannels, providing for the transferred heat energy between coolant flows at very low temperature difference. The experimental data [7] obtained for upflow evaporation and condensation in the element of a plate-fin heat exchanger with a density of 800 fins per meter show a significant difference in the value of the heat transfer coefficient during evaporation and condensation at wall superheat (subcooling) less than one degree. Typical dependence of the heat transfer coefficient on equilibrium mass vapor quality is shown in figure 2 for mass flux G = 50 kg/m²s. It was obtained in [7] that heat transfer coefficient is weakly dependent on the heat flux and vapor quality that is not typical for flow boiling in the conventional channels.

![Figure 2](image-url)

**Figure 2.** Experimental data on the dependence of heat transfer coefficient on mass vapor quality for G = 50 kg/m²s [7] compared with numerical solution of (4).

To explain these observations, the mathematical model of heat transfer during annular flow in the rectangular microchannel [8] was applied. Key feature of this model is the separation of the liquid flow in the corner of the channel bounded by interphase meniscus and the film flow on the channel wall. The equations of motion are supplemented by the kinematic condition, boundary conditions, and the integral equation of mass conservation for the film-meniscus system. For film flow without waves there is small parameter $\varepsilon = \delta_0/a << 1$, where $\delta_0$ is the initial thickness of liquid film and $a$ is half the width of the long side of the channel, therefore the Navier-Stokes equations are reduced to as follows

$$
(m^3 + 1.5 \frac{K}{\varepsilon} m^2)_{,x} + (m^3 m_{yy})_{,y} = \frac{3}{\varepsilon^2} (Ga \frac{m_y}{m})_{,y} - \frac{G_0 \varepsilon^2}{m_\varepsilon}
$$

(4)
where $\gamma = 1 - \left(\frac{dp}{\rho g} \right)/\rho a$, $\kappa = -\tau/\rho a$, $Bo = \rho g a^2/\sigma$, $Ga = A_0/(6\pi^2\sigma)$. Here $m=\delta/\delta_0$ is the liquid film thickness, longitudinal $x$ and transverse $y$ coordinates are dimensionless on $aBo/\epsilon$ and $a$, $\tau$ is the shear stress on liquid surface, $G_0 = \lambda_0/T_0\nu_0/(h_g/\sigma a)$, $\Theta = (T_w_i - T_s)/T_i$, $A_0$ is the Hamaker constant, $h_g$ is the latent heat of evaporation, $T_w_i$ is the inner wall temperature and $T_s$ is the saturation temperature. The mass flow at the interface is determined by the model of conductive heat transfer in the film. After finding the interphase surface, the heat conduction equations for the liquid and the wall are solved together. A feature of this equation is the inclusion of disjoining pressure term for ultrathin liquid films, which becomes especially important for the flow in nanochannels. When the interface shape is obtained, the Fourier equations for liquid area and the wall are solved together and the local heat transfer coefficient is determined.

During solving this equation, the film rupture occurs [8] and new surface configuration consisting of rivulets, menisci and the dry spots between them is established. In original version of the model the waveless liquid film was assumed. Nevertheless, the numerous experimental data show that this occurs if the liquid $Re$ numbers are less then unity. In plate fin heat exchangers, the liquid $Re$ number can considerably exceed this value. For this case, at high gas velocity, the flow in liquid film and in the menisci becomes unstable that moves the liquid back from the meniscus to the film, leveling the surface of the liquid film. The calculation of flow stability in the meniscus in the film gives the increments of the perturbation growth and limit radius of the meniscus, which is stable for the given gas velocity. Upon reaching the limit of the radius, accumulation of liquid in the corners stops, and further liquid flow rate in the meniscus is determined by the evaporation rate and current velocity of vapor. Numerical calculations using this approach show that the rupture of the liquid film for the case presented in figure 2 occurs before film surface leveling, and the considerable evaporation in the micro region near gas-liquid-solid contact line maintains high value of heat transfer coefficient regardless of the vapor quality. The heat transfer during vapor condensation has fundamentally different character when the maximum condensate flow occurs in the region of the ultrathin liquid film equalizing the interphase surface. It explains the significant difference in the heat transfer coefficients behavior during upward evaporation and condensation in microchannels.

4. Heat transfer during subcooled flow boiling at high heat fluxes

The flow boiling of water and refrigerants in microchannel heat sink is the most effective for cooling of Solar PV panels and high power electronic due to high latent heat of vaporization. At the same time, there are some limitations caused by heat transfer deterioration at high vapor quality [9]. To avoid these limitations, the subcooled flow boiling characterized by high critical heat flux can be used. The experimental data on the dependence of average heat flux on the average wall temperature during flow boiling of water at mass fluxes of 3100 kg/m²s, $P = 111$ kPa and subcooling of 80 °C are presented in figure 3. These data were obtained using experimental equipment described in [9]. The heat sink with two microchannels having the cross section of 2 mm×0.36 mm and length of 16 mm is milled on the top of the copper block, made from oxygen free copper with channel-to-channel spacing equal to 2 mm. Heat cartridges are mounted into the copper block supporting heat to the inner surface of 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 to avoid premature heat transfer crises.

The calculations according to the equation [10] for pool boiling heat transfer are shown in figure 3 as the dotted line. As it is seen, the considerable nucleate boiling suppression occurs for heat flux higher than 3000 kW/cm² when saturated nucleate boiling arises. To develop the model for heat transfer prediction during subcooled flow boiling in microchannels, the approach [11] for the conventional tube is modified. As it is shown in figure 1, principal feature of the gas-liquid flow in microchannels is the extremely thin liquid film between interphase meniscuses. It causes complete evaporation of this film at high heat flux and the dry spots formation on the channel wall excluding the meniscus area. To predict the heat transfer coefficients for this case it is necessary to take into account that the
nucleate boiling can occur only in the meniscus near the short size of the channel. For rectangular channels of the microchannel heat sink, the size of this area divided by the channel perimeter is determined as $1-\alpha_m$, where

$$\alpha_m = 6R_m/(2a+b) \quad (5)$$

Here $b$ and $a$ are the length of the short and long sides of the channel cross-section, respectively.

For calculation of the meniscus radius, it was assumed that $R_m=b/2$. During subcooled flow boiling, the wall superheat for forced convection, nucleate boiling and evaporation are different and wall heat flux is presented as follows

$$q_w^2 = \left( h_{con} F(T_w-T_{liq}) \right)^2 + \left( h_{boil} S \alpha_m (T_w-T_{sat}) \right)^2 \quad (6)$$

Here $h_{con}$, $h_{boil}$ and $h_{ev}$ are the forced convection, nucleate boiling and heat transfer coefficients of liquid film evaporation, respectively; $F=1$ and

$$S = \left( 1 + 0.055 F^{0.1} Re_{liq}^{0.16} \right)^{-1} \quad (7)$$

are the factors of forced convection enhancement and nucleate boiling suppression. The solid line in figure 3 represents the results of the calculations of subcooled flow boiling heat transfer according to (7) for water. For calculation of turbulent single-phase heat transfer, the correlation [12] was used with account of the difference of the liquid viscosities for wall and liquid temperatures. The results of calculations of the wall heat flux depending on wall temperature according to (6) are shown as a solid line in figure 3. As it seen, experimental data and calculations agree both for single phase and subcooled flow but heat transfer deterioration for saturated flow in experiment is a bit lower than that in prediction.
Conclusions
The presented results show strong impact of peculiarities of two-phase gas-liquid flow in microscale channels on the flow boiling heat transfer. A new approach has been presented for the prediction of upward flow boiling heat transfer in compact heat exchangers with the rectangular channels that operate at low wall superheat. This approach considers the effect of liquid film instability and wave formation on liquid distribution in the channel cross-section. The characteristics of heat transfer during subcooled flow boiling in microchannel heat sink designed for high heat flux removal shows the considerable nucleate boiling suppression for saturated boiling. The account of dry spots formation due to liquid film evaporation at high heat fluxes allows obtaining good agreement between experimental data and the calculation for single phase convection and subcooled flow boiling and improve their agreement for saturated flow boiling.

Acknowledgment
This work was performed in the Kutateladze Institute of Thermophysics SB RAS by a state contract with IT SB RAS,AAAA-A17-117022850026-8 (sections 2, 3) and by Russian Science Foundation, Project No 16-19-10519-C (section 4).

References
[1] Abramson A R, Tien C L 1999 J. Microscale Thermophys. Eng. 3 229–244
[2] Kuznetsov V V 2010 Proc. 14th Int. Heat Transfer Conference (Washington DC) IHTC14-22570
[3] Bertsch S S, Groll E A, Garimella S V 2009 Int. J. Multiphase Flow 35 142
[4] Saisorn S, Wongwises S 2008 Renew Sust Energ Rev. 12 824
[5] Kuznetsov V V, Kozulin I A, Shamirzaev A S 2012 J. Phys.: Conf. Ser. 395 012093
[6] Mirsaidov U, Mokkapati V R, Bhattacharya D, Andersen H, Bosman M, Özyilmaz B, Matsudaira P 2013 Lab on a Chip 13 2874
[7] Kuznetsov V V, Shamirzaev A S 2015 J. Eng. Thermophys. 24 357
[8] Kuznetsov V V, Safonov S A 2013 Heat Pipes and Solid Sorption Transformations: Fundamentals and Practical Applications ed. Vasiliev L L Kakac S CRC Press. 465–96
[9] Kuznetsov V V, Shamirzaev A S 2019 J. Phys. Conf. Ser. 1369 012043
[10] Yagov V V 2008 Thermal Engineering 55 245
[11] Liu Z, Winterton R H S 1991 Int. J. Heat Mass Transf. 34 2759
[12] Petukhov B S, Roizen L I 1964 High Temperature 2(1) 65