Two-phase compact heat exchangers: a review and future applications for high heat flux removal

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Abstract. The compact flow heat exchangers are important components of the equipment for power generation, chemical processing, and thermal environment control. This paper presents the flow pattern characterization and heat transfer performance for two-phase compact and ultra-compact heat exchangers. The scaling analysis is presented in this paper to identify the effect of various forces. The heat transfer enhancement mechanisms during flow boiling and condensation in compact heat exchangers are discussed using experimental data for refrigerant R-21. The mechanisms of subcooled and saturated flow boiling in ultra-compact heat exchangers based on microchannels are discussed using experimental data obtained for water.

1. Introduction

Two-phase compact heat exchangers are widely used in the power engineering, chemical, and food industries [1]. These heat exchangers with preferably rectangular channels are characterized by the specific surface area above 1000 m²/m³ and can be used as evaporators or condensers, to control the temperature of the process fluid and solid surface, recover or reject heat and so on. For example, cryogenic plate-fin exchanger has about 10% of the volume of an equivalent shell-and-tube exchanger and they can be plain, perforated or interrupted, such as offset strip fins that have the largest heat transfer efficiency. The important application has ultra-compact two-phase heat exchangers based on microchannels for electronic cooling because vapor/liquid phase change allows us to optimize the cooling strategy for the next generation of heat load systems. These heat exchangers have the potential to meet the large heat dissipation demands of high power electronics and computing systems. Currently, most of the advanced electronic components already generate heat fluxes higher than 100 W/cm², while future power electronic components, such as high-power laser and electronic radar systems, have been projected to generate heat fluxes over 1000 W/cm² [2]. To dissipate heat from planner surfaces, flow boiling can be achieved inside multiple parallel channels that are formed in a high conductivity substrate, named as a microchannel heat sink [3]. The two-phase microchannel heat exchangers also can be used for many small head load applications including condensers of heat pumps and air-conditioning systems [4].

This paper considers the flow pattern characterization and heat transfer performance in two-phase compact and ultra-compact heat exchangers based on experimental observations and numerical studies. To further explore the scale effects on flow boiling, the scaling analysis is presented in this paper to identify the effect of various forces. The heat transfer enhancement mechanisms in compact heat
exchangers, as well as their advantages and limitations, are discussed using the data for refrigerant R-21. The mechanisms of subcooled and saturated flow boiling in ultra-compact heat exchangers based on microchannels are discussed using experimental data obtained for water.

2. Channel size characterization for compact heat exchangers with rectangular channels
Existing definitions of the criterion for the transition to microscale [5] give only a rough estimation of the channel dimension when this transition occurs. In order to show the influence of the channel size on flow patterns, the elongated bubble flow in a capillary is considered. The Navier-Stokes equations for the steady-state laminar liquid flow in the system of coordinates associated with the head of a bubble

\[ \rho_{\text{liq}} \mathbf{u} \cdot \nabla \mathbf{u} = -\nabla p_{\text{liq}} + \mu_{\text{liq}} \nabla^2 \mathbf{u} + \rho_{\text{liq}} \mathbf{g}, \quad \nabla \cdot \mathbf{u} = 0. \quad (1) \]

and conjugation conditions at the gas-liquid interface

\[ p_{\text{liq}} = p_{\text{gas}} - \sigma \kappa, \quad \kappa = R_1^{-1} + R_2^{-1}, \quad \sigma \tau \mathbf{r} \cdot \mathbf{n} = 0 \quad (2) \]

are reduced to dimensionless equation assuming the constant gas pressure in the bubble as follows

\[ We \tilde{\mathbf{u}} \cdot \nabla \tilde{\mathbf{u}} = -\nabla \tilde{\kappa} + Ca \nabla^2 \tilde{\mathbf{u}} + Bo (\mathbf{g}/g). \quad (3) \]

Here \( \mathbf{u} \) is the liquid velocity, \( p \) is the pressure, \( \mu_{\text{liq}} \) is the viscosity, \( \mathbf{e} \) is the stress tensor in the liquid, \( \mathbf{n} \) is unit normal vector directed to the gas phase, \( \mathbf{r} \) is a unit tangent vector, \( \kappa \) is interface curvature and \( R_i \) is principal radii of curvature. The dimensionless variables are scaled by bubble velocity \( u_b \), the longitudinal, transverse coordinates and curvature are scaled by radius of the channel \( D_b/2 \). The dimensionless numbers are defined as follows: \( We = \rho_{\text{liq}} u_b^2 D_b/(2 \sigma) \), \( Ca = u_b \mu_{\text{liq}} / \sigma \), \( Bo = \rho_{\text{liq}} g D_b^2/(4 \sigma) \).

When reducing the diameter of the channel, the order of magnitude of the Bond number becomes much less than unity and gravity forces are negligible. For flow with \( Bo \ll 1 \) and \( We \geq 1 \) the inertial forces remain significant and flow characteristics are different from that for the conventional channel, this case can be considered as the flow in microchannel. At further reducing diameter of the channel, the Weber number becomes much less than unity \( We \ll 1 \), this case can be considered as the flow in microchannel. The dependence of the Weber number and Bond number on bubble velocity for refrigerant R-21 flow is in the channel with \( D_b = 200 \mu m \) is shown in figure 1. As is seen, the same channel can be considered as minichannel of microchannel depending on flow conditions.

For studying the nitrogen-water flow in a microchannel with cross-section of 269x362 \( \mu m \) the setup described in [6] was used. It was obtained that the typical flow patterns are periodic flow with elongated bubbles, transition flow, and annular flow. The boundaries between the flow patterns are blurred and can be characterized by a change in the statistical characteristics of the flow. Figure 2 shows the shape of the elongated bubble for a flow with \( J_{\text{liq}} = 0.064 \) m/s, \( J_{\text{gas}} = 0.068 \) m/s (\( We=0.058 \)).

3. Heat transfer during flow boiling and condensation in plate-fin heat exchangers
The problem of prediction of the heat transfer in two-phase compact heat exchangers with the low-temperature difference between the streams has acquired recently special attention in connection with the development of advanced systems for liquefying natural gas, cryogenic separation, and heat pumps. These heat exchangers are manufactured on the basis of minichannels, that allows considerably enhanced the heat and mass transfer due to capillary forces action in small size non-circular channels.

Compact evaporators and condensers are able to transfer the heat between the streams at a very low temperature difference. Experimental study of heat transfer during flow boiling in plate fin evaporators was performed in [7, 8]. The model based on conduction and convection through the film flowing down
We, Bo
G (kg/m²s)
We 0.2 mm
Bo 0.2 mm

Figure 1. Dependence of the Weber number and Bond number on bubble velocity for $D_h = 200 \mu m$, $p_s = 0.236$ MPa, vapor quality $x = 0.05$.

Along the walls of the channels was proposed in [7] to calculate heat transfer with evaporation.

The experimental test section for studying heat transfer during boiling and condensation of R-21 in a plate fin heat exchanger with 800 fins per meter and fins height of 6.1 mm is shown in figure 3a. This refrigerant was selected because its thermal properties at room temperature are very similar to nitrogen properties at cryogenic temperatures. The experiments were performed using closed loop that recirculate refrigerant presented in [9]. The experiments showed a significant difference in the heat transfer coefficient behavior during boiling and condensation for temperature difference near 1 °C. For flow condensation, the heat transfer coefficient growth with increasing of vapor quality. In contrary, for flow boiling, the deterioration of heat transfer was observed for quality higher than 0.8. Figure 3b shows the dependence of the heat transfer coefficient on vapor quality during flow boiling R-21 with wall superheat near 1.5 °C, mass flux $G = 43$ kg/m²s and static pressure of 0.236 MPa.

To explain this result, the mathematical model of the heat transfer during flow evaporation and condensation in rectangular microchannel proposed in [10] was used. The model is based on the coupling of liquid flow near the channel corner, limited by the interfacial meniscus, and in the film on a

Figure 3. Scheme of test section (a) and dependence of heat transfer coefficient on vapor quality for wall superheat near 1.5 °C, mass flux of 43 kg/m²s and pressure of 0.236 MPa (b). The lines show the calculations for uniform and non-uniform film.
The governing equations were supplemented by the kinematic condition, boundary conditions, and the integral equation of mass conservation for the film-meniscus system. For down flow with $e = \delta_0/a << 1$, where $\delta_0$ is initial thickness of the liquid film and $a$ is half the width of the long side of the channel, the Navier-Stokes equations are reduced to the equation

$$\left(\frac{m^3}{\varepsilon} + 1.5 \frac{K}{\varepsilon} \right) \frac{\partial \Theta}{\partial y} + (m^3 \frac{m_{yy}}{m^2}) \frac{\partial \Theta}{\partial x} = \frac{3}{\varepsilon} \left( \frac{m^2}{m^2} - \frac{G_o \Theta_{i,i}}{m^2} \right)$$

where $\gamma = 1 - (dp/\rho \, dp)/\rho \, g$, $\kappa = v/\rho \, ga$, $B_o = \rho \, ga^2/\sigma$, $G_a = A_v/6 \, \sigma$, $m = \delta/\delta_0$ – film thickness, longitudinal $x$ and transverse $y$ coordinates are dimensionless on $a Bo/\varepsilon$ and $a$ accordingly, $v$ is shear stress on the surface, $\Theta_{i,i} = (T_{a,i} - T_s)/T^*$, $A_o$ is the Hamaker constant, $h_{fg}$ is the latent heat, $T_s$ is the internal wall temperature, $T_e$ is the saturation temperature and $T^* = T_{e,*} - T_s$ is defined by external wall temperature. A special feature of this equation is the consideration of the disjoining pressure in ultrathin liquid films, which becomes especially important for flow in nanochannels. The mass flow at the interphase surface is determined by the model of conductive heat transfer in a film. In the case of evaporation, the conditions for film rupture [10] were applied to predict the dry spot formation between rivulet and meniscus. After obtaining the configuration of the liquid surface, the Fourier equation for the mass velocity of 49.3 kg/m²/s and wall subcooling of 2 K in minichannel of 1.6x6.3 mm: (a) corresponds to refrigerant R-21 at $T_i = 293$ K, (b) corresponds to nitrogen at 95 K. The calculation shows the formation of thin film and an increase in the heat transfer coefficient in the vicinity of the channel corner during condensation and formation of the dry spot in this area during evaporation. The calculated dependence of the heat transfer coefficient averaged along the channel perimeter on vapor quality is shown in figure 3b as the dotted line. The evaporation model predicts well the deterioration of heat transfer coefficient at vapor quality higher than 0.8 as it was observed in experiments.

![Figure 4](image_url)

**Figure 4.** The interface shape during condensation at the mass velocity of 49.3 kg/m²/s and wall subcooling of 2 K: (a) corresponds to R-21 at $T_i = 293$ K, (b) corresponds to nitrogen at $T_i = 95$ K.

The alternative model of the heat transfer during flow boiling was proposed in [11]. This model takes into account the contribution of nucleate boiling $h_{boil}$, two-phase forced convection $h_{con}$ and evaporation of the liquid film $h_{ev}$. The proposed model has the privilege over simple summing of corresponding terms because it allows clearly distinguishing the contribution of each term under the conditions when it is predominant. A feature of this model is account for the nucleate boiling suppression in thin liquid film ($\Psi_{sup}$ factor), nucleate boiling near the channel corners, and evaporation in liquid film as follows

$$h^2 = (h_{con} F)^2 + (h_{boil} \Psi_{sup})^2 + (h_{ev} E)^2,$$

(5)
where F, S and E are the enhancement factors for corresponding terms. The heat transfer coefficient for forced convection was determined from [4] for three-side heating at T-constant. To determine the heat transfer coefficient for evaporating wavy-turbulent liquid film, the model [12] was used in this paper with the calculation of the shear stress according to [13]. The result of the calculation of the dependence of the heat transfer coefficient on vapor quality is shown in figure 3b as a solid line. As is seen this model reasonably predicts the heat transfer for quality less than 0.8 when liquid film rupture occurs.

4. Heat transfer during flow boiling in ultra-compact heat exchangers based on microchannels

A feature of the heat transfer in two-phase microchannel systems used to cool heat-stressed equipment is extremely high heat fluxes before the crisis arises. Despite the considerable efforts in this area, the data of experimental studies are quite contradictory. In a number of papers, it was found that heat transfer coefficients depend mainly on heat flux and pressure, and weakly depend on the flow rate and mass vapor content. In other works, it was noted that the contribution of nucleate boiling to heat transfer is not decisive and heat transfer coefficients increase with increasing mass flux and vapour quality. There is a general agreement that the heat transfer coefficient during flow boiling in microchannels increases for refrigerants with increasing heat flux [14], but in some experiments it was shown that the heat transfer coefficient depends on the vapor quality, showing growth [15] or reduction [16] when the quality increases. The difficulties in the selection of reliable equations for prediction of flow boiling heat transfer constrain the development of ultra-compact two-phase microchannel heat exchangers. Despite the fact that this problem was addressed in several studies, e.g. [14], the flow boiling heat transfer in microchannels is still understood incompletely.

For studying heat transfer during subcooled and saturated flow boiling of water in microchannel heat sink the experimental setup described in [17] was used. During the tests, water is supplied from the condenser through the filter and damper of pulsation to the flow controller via the plunger pump. Then it goes through the pre-evaporator to attain the flow with desired vapor quality or subcooling. The copper block has two horizontal microchannels with cross section 2 mm × 0.36 mm and channel-to-channel spacing of 2 mm. The length of micro-channels is 16 mm, surface roughness Rp is 1.5 µm and static pressure at the channel inlet is near 0.1 MPa. Heat cartridges and set of thermocouples are mounted into the copper block to supply heat and measure averaged heat flux and wall temperature.

The dependence of the heat flux on wall temperature for saturated flow boiling of water (x_{in}=0.1) is presented in figure 5a for G = 450 kg/m²s. The dotted line shows the calculation for pool boiling according to Yagov equation [18]

\[ q_{hool} = \frac{k_{liq} 2 \Delta T}{\nu_{liq} \sigma T_s} \left(1 + \sqrt{1 + 800B + 400B} \right) \]  

(6)

Here \( A = 3.43 \cdot 10^{-4} \), \( B = h_{f} \left( (\nu_{liq} \rho_{gas})^{0.5} \right) \left( \sigma \left( \lambda T_s \right)^{0.5} \right) \). \( h_{liq}, \nu_{liq} \) are the heat conductivity and kinematic viscosity of the liquid, \( h_{f} \) is the specific heat of vaporization, \( \rho_{gas} \) is the vapor density. As is seen, the experimental data are considerably below then the calculations. The solid line shows the calculations according to equation (5) when nucleate boiling suppression in the area of the liquid film was taken into account and \( A = 0.47 \cdot 10^{-4} \) as recommended in [19]. The heat transfer coefficient for forced convection was determined from [4] for three-side heating (microchannel aspect ratio is five). For this case, the calculations correspond well to the experimental data.

The dependence of the heat flux on wall temperature for subcooled flow boiling (\( T_{in} = 50 ^\circ C \)) is presented in figure 5b for G = 1170 kg/m²s. The dotted line shows the calculation for pool boiling according to [18]. At subcooling flow boiling, the wall superheat for forced convection, nucleate boiling and evaporation is different [20]. For this case the equation 5 can be rewritten as follows

\[ q_{w} = \left( h_{con} F(T_w - T_{liq}) \right)^2 + \left( h_{hool} \nu_{sup} S(T_w - T_{sat}) \right)^2 + \left( E \phi h_{x} (T_w - T_{sat}) \right)^2 \]  

(7)
Figure 5. The dependence of heat flux on wall temperature for saturated boiling of water at \( G = 450 \text{ kg/m}^2\text{s} \) (a) and subcooled boiling at \( G = 1170 \text{ kg/m}^2\text{s} \) (b). The dotted lines show the calculation for pool boiling equation (6), solid lines show the calculations for saturated boiling equation (5), and subcooled boiling equation (7) accordingly.

It is assumed that all functions in this equation are the same as for saturated boiling and correspond to vapor quality equals to zero. Therefore the factors in equation (12) are: \( F = 1.0 \), \( E = 0 \) and \( S < 1 \). Solving this equation for given wall heat flux one can obtain the wall temperature depending on heat flux in the subcooled boiling region. The solid line in figure 5b shows the calculations according to equation (7) when nucleate boiling suppression was taken into account in case of local saturated boiling and \( A=0.47 \cdot 10^{-4} \) in equation (6) as recommended in [19]. For this case, the calculations correspond well to the experimental data.

5. Conclusions
The results are presented show the high efficiency of the compact heat exchangers for operation with a low-temperature difference and ultra-compact heat exchangers for high heat flux removal which allows for the implementation of previously unavailable technologies. In minichannels, the high intensity of heat transfer is observed during flow condensation due to deformation of the liquid film surface by capillary forces. In contrary, during flow boiling, the deterioration of heat transfer was observed at high vapor quality due to the formation of the dry spots. To realize these possibilities, the methods for predicting phase change in mini- and microchannels was proposed that show not only the advantages of using the channels of small size but also the problems associated, for example, with the suppression of nucleate boiling in a thin liquid film.

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