Analysis on Two-phase Heat Transfer Characteristics in Horizontal Mini-tubes

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Abstract. The two-phase heat transfer characteristics of degassed water flow boiling in horizontal mini-tubes are experimentally investigated in this paper. The inner diameters (d_i) of the tubes are 1.2mm, 1.6mm, and 2.0mm. Experiments are conducted with heat fluxes between 1.5kW/m² and 155.75kW/m² at mass fluxes of 21.19, 42.39, and 84.77kg/(m²s). The boiling curves of the three tubes are obtained. Incipient hysteresis is observed only in the tube with d_i=1.2mm. Two-phase dominant heat transfer mechanisms are analyzed in the tubes with d_i=1.2mm and d_i=2.0mm based on the curves of local heat transfer coefficient versus vapor quality and the visualization results. The trend of peak value of the local heat transfer coefficient versus heat flux at different mass fluxes in the tube with d_i=1.2mm is discussed in accordance with the thickness variation of the liquid film in annular flow.

Keywords: Two-phase, Heat Transfer Characteristics, Mini-tube.

1. Introduction

Flow boiling mechanisms in mini-tubes and micro-tubes have been extensively investigated both experimentally and theoretically in recent years for its practical application in various heat transfer equipment, such as the precooler in the air-breathing pre-cooled turbojet engine and the car engine cooling system. Several comprehensive discussions on this research topic were performed by [1-4].

Experimental investigations related to the heat transfer characteristics of flow boiling in mini-tubes have been performed by many researchers. Balacignesh et al. [5] conducted an experimental study on flow boiling heat transfer of n-pentane in a mini-tube with d_i=1.6mm. They found that the heat transfer coefficient is influenced by both heat flux and mass flux at higher vapor quality regions. Jiang et al. [6] presented an experimental and theoretical investigation on carbon dioxide flow boiling in a small-diameter tube and found that the heat transfer coefficient increases with increasing mass flux but decreases with the increase of tube’s inner diameter and saturation temperature in a low-medium vapor quality region. Qi et al. [7] experimentally studied the flow boiling heat transfer mechanism in vertical tubes with different diameters and found that both nucleate boiling and forced convection with
evaporation are dominant heat transfer mechanisms in the conventional tube \(d_i=6.88\text{mm}\) and the mini-tube \(d_i=2.15\text{mm}\) under the tested conditions. Zhang et al. [8] conducted experiments on liquid nitrogen flow boiling in a horizontal steel small-diameter tube and found that under the tested conditions, nucleate boiling is the dominant heat transfer mechanism.

In summary, plenty of experimental results related to flow boiling heat transfer mechanisms in mini-tubes have been obtained, but some are contradictory. Thus, more experimental investigations are needed to further explore the relevant mechanisms. As the tube size of most existing compact heat exchangers is classified as mini scale, this paper concentrates on the flow boiling heat transfer mechanisms in mini-tubes. The two-phase heat transfer characteristics of degassed water flow boiling in mini-tubes with \(d_i=1.2\text{mm}, 1.6\text{mm},\) and \(2.0\text{mm}\) are experimentally investigated in this article.

2. Experimental setup and data reduction

2.1. Experimental system

An integrated system (shown in Figure 1) consisting of a syringe pump, a DC power supply system, a data acquisition system, a test section, a high-speed camera, and a fluid reservoir was used in the present study. The experimental system was mounted in a \(1\times1\times1\ \text{m}^3\) plexiglass box. The ambient temperature inside the box was maintained at \(25^\circ\text{C}\) and the atmospheric pressure was 1 bar. A LEAD FLUID® TYD 01 syringe pump was used to push the degassed water into the flow loop at a predetermined flow rate. A DC power supply was clamped on both ends of the test section to heat the tube. Eight Omega® K-type thermocouples \(T_{w,1}-T_{w,8}\) were soldered onto the outer surface of the mini-tube to monitor the local surface temperatures. Two K-type sheathed thermocouples were used to measure the fluid temperatures at the inlet and the outlet of the tube. All thermocouples were calibrated before the experiments. Experimental data were recorded by a National Instruments® data acquisition system with a frequency of 100 Hz. A Photron FASTCAM® SA3 high-speed camera with a frame rate of 500 fps was used to record the flow patterns through a section of glass tube, and the visualization results were analyzed by the Image Pro.® software.

![Figure 1. Photograph of the experimental system.](image1)

![Figure 2. Schematic drawing of the test section.](image2)

The schematic drawing of the test section is shown in Figure 2. Stainless mini-tubes with three
different diameters were selected. The inner surface roughness of the tubes was measured by Talysurf PGI 1240®. The ratio of the average roughness to the inner diameter in each mini-tube was found to be less than 1%. Thus, the effect of inner surface roughness can be neglected. A section of glass tube was connected to the exit of the mini-tube by adapters to visualize the flow patterns in the mini-tube. Copper clamps were used to connect to the DC power supply and maintain the test section in a horizontal position. The mini-tube was covered with asbestos for thermal insulation. The geometric parameters of the three test sections are listed in Table 1.

Table 1. Geometry parameters of the three test sections.

| d_i (mm) | d_o (mm) | l_{mt} (mm) | l_{p,1} (mm) | l_{p,2} (mm) | l_{gt} (mm) |
|---------|---------|-------------|--------------|--------------|-------------|
| 1.2     | 3.0     | 240.5       | 52.5         | 52.7         | 50.0        |
| 1.6     | 3.0     | 245.0       | 57.8         | 56.7         | 50.0        |
| 2.0     | 4.0     | 250.0       | 37.6         | 38.6         | 50.0        |

Prior to the experiment, the heat loss of the test section was estimated. The estimation method was described in [9]. By fitting the experimental data, three polynomial relations were obtained to calculate the heat loss in mini-tubes with $d_i=1.2$, 1.6, and 2.0 mm, given by Equation (1–3):

$$Q_{\text{loss}} = -0.07 + 0.02 \times (T_{w,\text{avg}} - T_{\text{state}}) + 7.39 \times 10^{-5} \times (T_{w,\text{avg}} - T_{\text{state}})^2$$ \hspace{1cm} (1)

$$Q_{\text{loss}} = 0.007 + 0.02 \times (T_{w,\text{avg}} - T_{\text{state}}) + 7.47 \times 10^{-5} \times (T_{w,\text{avg}} - T_{\text{state}})^2$$ \hspace{1cm} (2)

$$Q_{\text{loss}} = 0.06 + 0.02 \times (T_{w,\text{avg}} - T_{\text{state}}) + 1.89 \times 10^{-4} \times (T_{w,\text{avg}} - T_{\text{state}})^2$$ \hspace{1cm} (3)

The experiments were conducted with heat fluxes ranging from 1.5 kW/m² to 155.75 kW/m² at mass fluxes of 21.19, 42.39, and 84.77 kg/(m²s). Experimental data were recorded for 100 s after all wall temperatures were in a steady state. At each mass flux, the DC power supplied to the test section was increased incrementally until the wall temperature near the exit of the mini-tube started to increase rapidly. Then, the experiment was terminated.

2.2. Data reduction

**Effective heat flux.** The effective heating power was calculated using Equation (4):

$$Q_{\text{eff}} = Q_{\text{tot}} - Q_{\text{loss}}$$ \hspace{1cm} (4)

Where $Q_{\text{tot}}=UI$, $U$ and $I$ represent the output voltage and current of the DC power supply, respectively.

The effective heat flux was calculated using Equation (5):

$$q_{\text{eff}} = \frac{Q_{\text{eff}}}{A_i}$$ \hspace{1cm} (5)

Where $A_i=\pi dL$, $L$ is the length of the mini-tube.

**Vapor quality.** The local vapor quality was calculated using Equation (6):

$$x = \frac{q_{\text{eff}} \pi d}{A_i} \left( z - z_0 \right)$$ \hspace{1cm} (6)

Where $G$ is the mass flux, $h_{fg}$ is the phase-change latent heat of degassed water. The subcooled length $z_0$ was calculated from the energy balance according to Equation (7):

$$z_0 = \frac{A_i G C_p (T_s - T_u)}{Q_{\text{eff}}} L$$ \hspace{1cm} (7)

**Local heat transfer coefficient.** The local heat transfer coefficient was calculated using Equation (8):
According to the formula for cylinder-wall one-dimensional thermal conductivity, the inner wall temperature of the mini-tube was calculated using Equation (9):

\[
h = \frac{q_{\text{eff}}}{T_{w,i} - T_i}
\]

\[
T_{w,i} = T_{w,o} + \frac{q_v}{4k_w} \left[ \frac{d_o^2}{2} - \frac{d_i^2}{2} \right] + \left( \frac{q_v}{k_w} \frac{d_o}{2} + \frac{q_v}{2k_w} \frac{d_i}{2} \right) \ln \frac{d_o}{d_i}
\]

Where \(k_w\) is the thermal conductivity of the tube, \(T_{w,o}\) is the outer wall temperature, \(q_v\) is the effective heating power of unit volume:

\[
q_v = \frac{Q_{\text{eff}}}{4\pi(d_o^2 - d_i^2)L}
\]

When \(z < z_0\), according to energy balance, the liquid temperature was calculated using Equation (11):

\[
T_i = T_{in} + \frac{q_{\text{eff}}\pi d_i}{AGC_p} z
\]

When \(z > z_0\), the liquid temperature reached saturation temperature \(T_{\text{sat}}\).

Experimental uncertainties were determined according to the standard uncertainty analysis of Taylor [10] and summarized in Table 2.

| Parameter                        | Maximum uncertainty (%) |
|----------------------------------|-------------------------|
| Mini-tube inner diameter         | 0.83                    |
| Pump mass flow rate              | 0.50                    |
| Mass flux                        | 1.73                    |
| Voltage                          | 0.50                    |
| Current                          | 0.50                    |
| Effective heat flux              | 5.04                    |
| Local heat transfer coefficient  | 7.4                     |

3. Results and Discussion

3.1. Boiling curves

Figure 3(a) gives the boiling curves of the three tubes at \(G = 21.19 \text{ kg/} \text{m}^2\text{s}\) measured by \(T_{w,8}\). In the single-phase flow region, the wall superheat \(\Delta T_{\text{sup}}\) increases with increasing \(q_{\text{eff}}\) until the onset of nucleate boiling (ONB). Furthermore, heat dissipation increases with increasing \(d_i\) at the same \(\Delta T_{\text{sup}}\). For the tube with \(d_i = 1.2\text{mm}\), \(\Delta T_{\text{sup}}\) decreases suddenly after ONB, which is called incipient hysteresis. Incipient hysteresis is not observed remarkably in the tubes with \(d_i = 1.6\text{mm}\) and \(d_i = 2.0\text{mm}\). In the tube with \(d_i = 1.2\text{mm}\), the generated bubbles swiftly develop into annular flow due to the confinement of the inner wall. The forced convective evaporation between the thin liquid film and the wall might result in the rapid decrease of \(\Delta T_{\text{sup}}\). This phenomenon vanishes as the bubble confinement weakens with the increase of \(d_i\).

After the ONB, the slope of the boiling curve increases rapidly. In the tubes with \(d_i = 1.2\text{mm}\) and \(d_i = 1.6\text{mm}, \Delta T_{\text{sup}}\) shows minor dependence on \(q_{\text{eff}}\). In the tube with \(d_i = 2.0\text{mm}\), however, \(\Delta T_{\text{sup}}\) increases slowly with the increase of \(q_{\text{eff}}\). These phenomena can be attributed to the dominant heat transfer mechanisms of convective evaporation and nucleate boiling, respectively, which are illustrated by the visualization results in the subsequent section. Besides, at the same \(q_{\text{eff}}\), the tube with smaller \(d_i\) needs less \(\Delta T_{\text{sup}}\) and possesses stronger heat dissipation ability. \(\Delta T_{\text{sup}}\) of the tube with \(d_i = 1.2\text{mm}\) increases...
rapidly with the further increasing $q_{\text{eff}}$. The corresponding $q_{\text{eff}}$ is the critical heat flux (CHF), which indicates the deterioration of heat transfer. The tubes with $d_i=1.6\text{mm}$ and $d_i=2.0\text{mm}$ perform normally under high $q_{\text{eff}}$, which demonstrates that the increase of $d_i$ leads to a larger CHF.

Figure 3(b) illustrates the effect of $G$ on the boiling curve of the tube with $d_i=1.2\text{mm}$ at $T_w, 8$. Before the occurrence of ONB, the increase of $G$ leads to the increase of $q_{\text{eff}}$ at a certain $\Delta T_{\text{sup}}$. The $q_{\text{eff}}$ needed for the occurrence of ONB increases with increasing $G$. After the ONB, the boiling curves at three mass fluxes almost converge into one line. As $q_{\text{eff}}$ subsequently increases, the boiling curves for $G=21.19\text{kg/ (m}^2\text{s)}$, $42.39\text{kg/ (m}^2\text{s)}$, and $84.77\text{kg/ (m}^2\text{s)}$ reach CHF one after the other. Thus, the increase of $G$ enhances the resistance to the deterioration of heat transfer.

3.2. Local heat transfer coefficient

Figure 4(a) illustrates the local heat transfer coefficient $h$ as a function of vapor quality $x$ in the tube with $d_i=1.2\text{mm}$ at $G=21.19\text{kg/ (m}^2\text{s)}$. With $q_{\text{eff}}$ ranging from $25.09\text{kW/m}^2$ to $31.13\text{kW/m}^2$, $h$ increases with the increase of $x$ due to nucleate boiling when $x<0.1$. Then, $h$ increases rapidly and attains its peak value as highlighted in Figure 4(a). The corresponding outlet flow pattern visualization is shown in Figure 4(b). The bubbles develop and form annular flow along the axial direction of the tube. The convective evaporation of thin liquid film between the vapor core and the wall is the dominant heat transfer mechanism. The peak value of $h$ increases with increasing $q_{\text{eff}}$ ranging from $25.09\text{kW/m}^2$ to $31.13\text{kW/m}^2$. This trend occurs since the thermal resistance diminishes as the thickness of liquid film decreases with the increase of $q_{\text{eff}}$. Moreover, with $q_{\text{eff}}$ ranging from $45.19\text{kW/m}^2$ to $46.66\text{kW/m}^2$, $h$ decreases with the increase of $x$. As depicted in Figure 4(c), partial dry-out appears in the liquid film of the annular flow, indicating the occurrence of heat transfer deterioration.
Figure 4. (a) Local heat transfer coefficient versus vapor quality at $G=21.19$ kg/(m$^2$s) in the tube with $d_i=1.2$ mm; (b) Visualization of the outlet flow pattern; (c) Visualization of the partial dry-out.

Figure 5. (a) Local heat transfer coefficient versus vapor quality at $G=84.77$ kg/(m$^2$s) in the tube with $d_i=1.2$ mm; (b) Visualization of the outlet flow pattern; (c) Visualization of the partial dry-out.

Figure 5(a) depicts the curves of $h$ versus $x$ at $G=84.77$ kg/(m$^2$s) in the tube with $d_i=1.2$ mm. When $q_{\text{eff}}$ varies from 57.60 kW/m$^2$ to 74.34 kW/m$^2$, the trend of $h$ versus $x$ is similar to that at $G=21.19$ kg/(m$^2$s). The peak value of $h$ decreases with the increase of $q_{\text{eff}}$ at $G=84.77$ kg/(m$^2$s) given that entrainment and sedimentation of liquid drops occur as $G$ increases. As $q_{\text{eff}}$ increases, even more entrainment and sedimentation lead to the increase of liquid-film thickness in annular flow, leading to the decrease of the peak value of $h$. When $q_{\text{eff}}$ ranges from 90.18 kW/m$^2$ to 110.55 kW/m$^2$, $h$ shows negligible dependence on $x$ and remains almost constant. However, at $q_{\text{eff}}=155.75$ kW/m$^2$, $h$ decreases with the increase of $x$ due to the partial dry-out shown in Figure 5(c).

Figure 6(a) shows the curves of $h$ versus $x$ at $G=21.19$ kg/(m$^2$s) in the tube with $d_i=2.0$ mm. When $q_{\text{eff}}$ from 23.94 kW/m$^2$ to 41.52 kW/m$^2$, $h$ increases with increasing $x$, but the peak value is only around 10 kW/m$^2$. As shown in Figure 6(b), the corresponding outlet flow pattern is slug flow, and the dominant heat transfer mechanism is nucleate boiling. With $q_{\text{eff}}$ increasing to 90.24 kW/m$^2$, $h$ decreases with the increase of $x$. Figure 6(c) illustrates that the corresponding outlet flow pattern is annular flow. However, the vapor phase between the liquid film and the wall results in the decrease of $h$. 
4. Conclusion

The heat transfer characteristics of degassed water flow boiling in mini-tubes with inner diameters of 1.2mm, 1.6mm and 2.0mm have been experimentally investigated. The experiments were conducted at mass fluxes of 21.19kg/(m²s), 42.39kg/(m²s) and 84.77kg/(m²s) with heat fluxes between 1.5kW/m² and 155.75kW/m². The following conclusions are drawn:

(1) The phenomenon of incipient hysteresis is observed in the tube with \(d_i=1.2\)mm. This might be attributed to the occurrence of liquid film evaporation in the annular flow, which is formed due to the development of the confined bubbles after the onset of boiling. The heat dissipation ability is strengthened with the decrease of the inner diameter of tube. Besides, the increase of \(G\) enhances the resistance to the deterioration of heat transfer in the tube with \(d_i=1.2\)mm.

(2) According to the curves of local heat transfer coefficient versus vapor quality and the flow pattern visualization results, the dominant heat transfer mechanisms are convective evaporation and nucleate boiling in the tubes with \(d_i=1.2\)mm and \(d_i=2.0\)mm, respectively. In the tube with \(d_i=1.2\)mm, the trend of peak value of the local heat transfer coefficient versus heat flux at different mass fluxes performs differently for the thickness variation of the liquid film in annular flow.

References

[1] S.G. Kandlikar. (2002) Fundamental issues related to flow boiling in minitubes and microtubes. Experimental Thermal and Fluid Science, 26, 389-407. https://doi.org/10.1016/S0894-1777(02)00150-4

[2] C. B. Tibirica, G. Ribatski. (2013) Flow boiling in micro-scale tubes-Synthesized literature review. International Journal of Refrigeration, 36, 301-324. https://doi.org/10.1016/j.ijrefrig.2012.11.019

[3] Yuan Wang, Zhen-guo Wang. (2014) An overview of liquid-vapour phase change, flow and heat transfer in mini-and micro-tubes. International Journal of thermal science, 86, 227-245. https://doi.org/10.1016/j.ijthermalsci.2014.07.005

[4] T.G. Karayiannis, M.M. Mahmoud. (2017) Flow boiling in microtubes: Fundamentals and applications. Applied Thermal Engineering, 115, 1372-1397. https://doi.org/10.1016/j.applthermaleng.2016.08.063

[5] S. Balavignesh, V. Umesh, B. Raja. (2013) Flow boiling heat transfer of n-pentane in a Mini-tube. Procedia Engineering, 64, 1524-1532. https://doi.org/10.1016/j.proeng.2013.09.234

[6] Jiang Linlin, Liu Jianhua, Zhang Liang, Liu Qi, Xu Xiaojin. (2017) Characteristics of heat transfer for CO2 flow boiling at low temperature in mini-tube. International Journal of Heat and Mass Transfer,108, 2120-2129. https://doi.org/10.1016/j.ijheatmasstransfer.2016.12.113
[7] Qi Lu, Deqi Chen, Chong Li, Xueqiang He. (2017) Experimental investigation on flow boiling heat transfer in conventional and mini vertical tubes. International Journal of Heat and Mass Transfer, 107, 225-243. https://doi.org/10.1016/j.ijheatmasstransfer.2016.11.020

[8] Qiaoyu Zhang, Jun Chen, Jiapeng Li, Jing Cao, Liang Chen, Yu Hou. (2017) Experimental study on saturated flow boiling heat transfer of nitrogen in a small-diameter horizontal heated tube. Experimental Thermal and Fluid Science, 86, 257-271. https://doi.org/10.1016/j.expthermflusci.2017.04.003

[9] Law M, Lee P-S, Balasubramanian K. (2014) Experimental investigation of flow boiling heat transfer in novel oblique-finned microtubes. International Journal of Heat and Mass Transfer, 76, 419–431. https://doi.org/10.1016/j.ijheatmasstransfer.2014.04.045

[10] J. R. Taylor. (1997) An introduction to analysis. 2nd Edition, University Science Books.