The Effect of Mass Flow Rate and Heat Flux on Vertical Two-Phase Flow Regimes in a Small Diameter Tube

Ali K. Mohammed\textsuperscript{a}, Majid H. Majeed\textsuperscript{b}, Ahmed Q. Mohammed\textsuperscript{c}

\textsuperscript{a}Engineering Technical College- Baghdad, Middle technical university
\textsuperscript{b} alikadhimmohammed@gmail.com, \textsuperscript{b} Drmajidhm@gmail.com, \textsuperscript{c} dr.ahmed56@yahoo.com

Abstract: In recent years, there has been a rising concentration in investigating and researching the boiling of micro channel flow; since they afford high thermal effectiveness, small size, and low weight. In this paper, the boiling heat transfer was experimentally examined using water as the working fluid in small diameter tubes. Experiments for the flow model were performed using the same Pyrex glass tube test facility. The experiments of heat transfer were carried out using an internal diameter copper tube (10.922 mm). The range of other parameters was varied: mass flow rate (0.008 – 0.0214 kg/s); heat flux (17351 – 80529 W/m\textsuperscript{2}); it has been found that the average coefficient of heat transfer depends on the form of flow pattern. The average heat transfer coefficient increases by about 37% when the flow pattern varied from slug flow to churn flow at a constant mass flow rate of 0.008 kg/s. Besides, the heat transfer coefficient ($h$) in the two-phase flow, is 39%, which is more than that for single-phase flow, when the mass flow rate is 0.0214 kg/s, and $q''$ (heat flux) increases from 17351 W/m\textsuperscript{2} to 49105 W/m\textsuperscript{2}. Also, when mass flow rate rises from (0.008 kg/s) to (0.0166 kg/s), the pressure drop increases by about (15%) and increases by about (5%) when the mass flow rate rises from (0.0166 kg/s) to (0.0214 kg/s). The main regime of flow was a slug, churn, bubbly, dispersed bubbly, and confined bubbly flow.

Keywords: Heat transfer coefficient, Flow patterns, Mass flow rate, Micro channels and Heat flux.

1. Introduction
The technology regarding the improvement of heat transfer has become the main concentration in the modern field of thermal engineering applications. In specific, the economic concern of heat transfer augmentation is associated with pressure loss [1], [2], which usually occurs in the boiling of flow at high heat flux. Producers of boiling water reactors, heat exchangers, and technology extending from cooling of mini-scale machines to plant equipment for the chemical industry are examining various heat transfer enhancement methods for thermal transport. The heat transfer of supercritical water with two-phase in a nuclear reactor that flows through channels plays a significant role in efforts to increase the characteristics safely of thermal-hydraulic (Liu et al., 2019; Yin et al., 2019). The pressure loss and $h$ (heat transfer coefficient) rate change in each regime of two-phase flow [5]. Numerous regimes of heat transfer are connected with the process of heat transfer (Torfeh & Kouhikamali, 2015; Wijayanta et al., 2016).

Heat transfer is energy movement due to a temperature alteration. There are three physical mechanisms of heat transfer; convection, conduction and radiation. In practical problems all three modes can occur simultaneously. The mass transfer may occur because of heat transfer from phase to phase, such as boiling and condensation.

The advantage of using vaporization in such a system is that the fluid encounters small temperature changes as it is circulated, but it still transfers a large quantity of heat (Gómez et al., 2015; Markal et al., 2016). The heat transfer process is affected by the flow pattern regime, and identifying the dominant mechanism of heat transfer is essential to establish a more accurate model for boiling flow.
Numerous investigators have exposed the mechanism of heat transfer dominant for boiling in micro-channels, and relatively few of them are flow pattern dependent (Boye et al., 2015; Park & Hrnjak, 2009; Ong & Thome, 2009; Meulen & Pieter, 2012; Thome & Consolini, 2010). Additionally, it is influenced by the size of a channel as detected by [13], who establishes that in conventional tubes the nucleate boiling heat transfer is principal while decrease of the tube diameter of the convective boiling heat transfer becomes dominant. According to some other investigators (Ong & Thome, 2009; Meulen & Pieter, 2012; Thome & Consolini, 2010). It has been found that the annular flow pattern is typically observed at moderate and high vapor levels. The evaporative process happens after the liquid reaches the boiling temperature at the right pressure condition. Therefore, the mixture of fluid may be liquid, liquid, and vapor or exactly vapor. These conditions depend on the amount of heat supply. For example, when single-phase water flows through a tube and is a substance to heat flux, two-phase flow occurs. Water vapor (steam) then begins to form along the tube, producing a system of two-phase flow. A two-phase method is used in the petroleum industry at risers, wells, and pipelines (Meulen & Pieter, 2012). Determining the flow regime and its boundaries are fundamental to selecting the correct prediction approach. Subsequently, numerous researches have been carried out recently to establish a map of flow regimes for liquid boiling in micro-channels (Harirchian et al., 2010; Karayiannis et al., 2010; Saisorn et al., 2008). Conversely, most of these maps of flow patterns were developed using an adiabatic flow database. This paper reports the experimental results of the flow patterns and the mechanism of boiling heat transfer in micro-diameter tubes was discussed. The main objective of the paper was to define the boiling heat transfer mechanism and flow patterns in small to micro-diameter tubes under varying heat flux and mass flow rate. Hence, there is a quick growth of practical applications in engineering for micro-systems, micro-devices, and manufacturing is producing enlarged attention by the community of heat transfer into the investigation in the transport phenomena in micro channels.

2. Materials and methods

The experimental configuration is constructed of a vertical copper tube (1400mm) long with an outside diameter of (12.7mm). The wall thickness of the tube is (0.889mm). The general layout of the major components of the present investigation apparatus is presented in figure 1. Water is a working fluid, and before each experiment, the water was preheated to a specified temperature in a storage tank. The system of measurement included thermocouples of chromel-alumel (Type K AWG-30) to measure the temperature. A pressure gauge was used to determine the pressure at the bottom and top of the tube. A glass tube was installed downstream investigation area of the heat transfer for analysis of flow patterns. A digital camera was applied to capture the two-phase flow patterns. All instruments used including (thermocouples and pressure gauge) were calibrated carefully in the Iraqi central inspection and quality control agency. The ambient heat loss in the experiment section was calculated dependent on the difference of temperature between the insulation's internal and external surfaces and allowed for in all our tests. In this study, the principal variable parameters are the mass flow rate with different values including (0.008, 0.0166, and 0.0214kg/s) and heat flux values (17351, 25680, 49105, 64423, and 80529W/m²). Moreover, the purpose of running the experiments is to examination the two-phase flow and flow regimes for instance slug, dispersed bubbly, churn, bubbly, and confined bubbly flow on the h and wall temperature. Instead, a comparison is made between the results of the present experimental as well as the results of the correlation equation obtained by Dittus-Boelter, which is distinct from that of Dittus-Boelter [20].

\[
Nu = 0.023 \ Re^{0.8} \ Pr^{0.4}
\]  

(1)

There is a good agreement between the correlation built by Dittus-Boelter and the experimental results when the boiling is not found. The heat transfer coefficient \(h\) for a single-phase flow shows a reasonable covenant with the Dittus – Boelter correlation. The \(h\) at each point of the thermocouple was calculated from the equation below (Bergman & Incropera, 2011):
where $T_s$ is the temperature locally within the wall, $T_f$ it is a local temperature of the fluid. $q''$ is the heat flux of the inner wall to the fluid. The temperature of fluid was evaluated in a single-phase flow from heat flux and inlet fluid temperature. Via the two-phase flow, it was expected from fluid pressure, calculated by considering a linear decrease in pressure. $T_s$ was considered dependent on the outside surface temperature, measured by the thermal resistance of the wall tube, thermocouples, and heat flux.

$$h = \frac{q''}{(T_s - T_f)}$$

(2)

3. Results and discussions

3.1. Flow patterns

Interfacial distribution forms are called flow regimes, or flow patterns. The identification of the flow regimes is still largely dependent on observation visual. Whereas arguments still exist about the description of flow patterns, most scientists accepted that their findings should be grouped into four major patterns flow: stratified, bubble, intermittent, and annular flow. Every significant group can be divided into subcategories. [22] distinct five characteristic flow regimes in their vertical upward flow regime map, i.e., bubbly, dispersed bubble, annular, slug, and churn flow. In this study, five types of patterns are observed. These types are a slug, churn, dispersed bubbly, bubbly, and confined bubbly flow. Figure 2 illustrates the flow patterns at constant mass flow rate of (0.008kg/s) and different values of heat flux of (17351, 25680, 49105, 64423 and 80529W/m²) respectively. Figures 2a and 2b show that there is no bubble formation along the tube because heat flux is small. Figure(2c) shows that the increase of heat flux causes a rise in the tube wall temperature, which produces the boiling of liquid (water) (the temperature above the saturated temperature), which helps the formation of so-called the slug flow pattern, as shown that the bubbles enlarge. Figure 2e illustrations the development of flow pattern to churn pattern due to increase in $q''$ (heat flux) effect in a rise in the tube wall temperature which produces the boiling of liquid (water) (the temperature above the saturated temperature) with the formation of so the called the churn flow pattern according to (Huo et al., 2004).

Figure 1. Schematic diagram of the test apparatus and the experimental structure.

1 Water Storage Tank  2 Immersion Heater  3 Pump
4 Test Tube  5 Difference Pressure  6 AC-Source
7 Observation Section  8 Camera  9 Separator
Figure 2. Flow patterns at a \( \dot{m} \) (mass flow rate) and variable \( q'' \) (heat flux) of \((17351, 25680, 49105, 64423, \) and \(80529\) W/m\(^2\)) respectively. Flow from left to right, and the flow patterns from top: Water Flow, Water Flow, Slug Flow, Churn Flow, Churn Flow.

Figure 3 illustrates the flow regimes at a constant mass flow rate \((0.0166\, \text{kg/s})\) and different values of heat flux of \((17351, 25680, 49105, 64423, \) and \(80529\) W/m\(^2\)) respectively. Figures 3a and 3b show there is no bubble formation along the tube because heat flux is small. Figure 3c shows as heat increases, and the tube wall temperature is increasing which produces the boiling of liquid and forming a pattern called the dispersed bubbly flow pattern. Figure 3d shows the bubbly flow regime, which results from increasing the tube wall temperature, which causes water to boil. Figure 3e shows increasing heat flux aims a rise in the tube wall temperature, which produces the boiling of liquid with the formation of these-called churn flow patterns.

Figure 4 illustrates the flow regimes at a \((0.0214\, \text{kg/s})\) mass flow rate and different values of heat flux of \((17351, 25680, 49105.2254, 64423, \) and \(80529\) W/m\(^2\)) respectively. Figures 4a to 4c shows, there is no bubble formation along the tube because heat flux is small. Figure 4d shows increasing \(q''\) due to an increase in the tube wall temperature, which causes the liquid to boil with the formation of the so-called dispersed bubbly flow pattern. Figure 4e shows increasing heat flux causes a rise in the temperature of the tube wall that produces the boiling of liquid with the formation of the so-called confined bubbly flow pattern. When the value of the mass flow rate is constant, the flow pattern shifts from no bubble formation along the tube to churn pattern depending on the heat flux as illustrated in Figures 2, 3, and 4. When the value of mass flow rate is \((0.008\, \text{kg/s})\), and heat flux is \((40105\, \text{W/m}^2)\) slug flow pattern is observed, if heat flux increases to \((64423\, \text{W/m}^2)\) churn flow pattern will be observed). When the \(q''\) is constant, the flow pattern moves from slug, churn, dispersed bubbly, bubbly, and confined bubbly flow depending on the \( \dot{m} \) (mass flow rate). For example, when heat flux is \((64423\, \text{W/m}^2)\) and the \( \dot{m} \) mass flow rate is \((0.0214\, \text{kg/s})\) dispersed bubbly flow pattern is observed if mass flow rate decreases to \((0.0166\, \text{kg/s})\) bubbly flow regime will be detected.
Figure 3. Flow patterns at a (0.0166kg/s) of $\dot{m}$ (mass flow rate) and variable $q''$ (heat flux) of $(17351, 25680, 49105, 64423 \text{ and } 80529 \text{ W/m}^2)$ respectively.

Figure 4. Flow patterns at a (0.0214kg/s) of $\dot{m}$ (mass flow rate) and variable $q''$ (heat flux) of $(17351, 25680, 49105, 64423 \text{ and } 80529 \text{ W/m}^2)$ respectively.
3.2. Heat transfer

The water mass flow rate in the heating tube can vary with different quantities. In this research, it is changed by three quantities to study its effect on another parameter. When raising the mass flow rate, the temperature distribution along the heating tube is decreased. Figure 5 indicate the variation of tube wall temperature (°C) with the axial distance (m) for three values of mass flow rate (0.008, 0.0166, and 0.0214kg/s) respectively at different values of heat flux (17351W/m², 49105 W/m² and 80529 W/m²) respectively. The reduction in mass flow rate at a constant heat flux tends to increase tube wall temperature; this is due to the conservation of energy principle. This causes the tube wall temperature to increase. Besides, the figures show that the temperature of the tube wall rises with axial distance created by the accumulated heat or energy with distance.

Figure 5. Wall temperature difference with a constant \( q'' \) (heat flux) equal to (A) = 17351W/m², (B) = 49105W/m², and (C) = 80529W/m².

Figure 6 indicates the effect of the \( \dot{m} \) (mass flow rate) on the \( h \) with a long tube from the beginning of the test region. These figures show that the rise in the \( \dot{m} \) tends to raise the \( h \), due to the increase in Reynolds number with the fluid velocity increases. It is distinguished the heat transfer is high at the start length of the test region because of the big difference in temperatures between the water and the tube wall and thin boundary layer. Then the \( h \) is reduced since temperature variations between the wall and fluid decreased. It was also found by (Na & Chung, 2011). Moreover, it was illustrious that the \( h \) rises suddenly at points where boiling is initiated with heat flux increase. Until a specific limit, then the \( h \) is decreased at the saturated section. In general, it is increasing the mass flow rate leading to a rise in the \( h \) (heat transfer coefficient). Conversely, the main aim of the reduction in the heat transfer coefficient is that the conduction heat transfer in the liquid thin film providing evaporative latent heat is proportional inversely to the thin film thickness according to (Na & Chung, 2011).

Figure 7 displays the impact of mass flow rate on the typical \( h \) at different values of heat flux (17351, 25680, 49105, 64423and 80529W/m²), respectively. From this figure, one can see other parameters that affect the Nusselt number, such as Prandtl number, flow patterns. The \( h \) in the churn flow is more
than in bubbly flow, and dispersed flow is more than bubbly flow at a \( (64423 \text{W/m}^2) \) and altered values of \( \dot{m} \) (0.008, 0.0166 and 0.0214kg/s) respectively. The temperature distribution along the tube wall at the test region is investigated with altered heat fluxes.

![Figure 6. Surface heat transfer coefficient (h) at a constant \( q'' \) (heat flux) equal (A) = 17351W/m², (B) = 49105W/m² and (C) = 80529W/m².](image)

**Figure 6.** Surface heat transfer coefficient (\( h \)) at a constant \( q'' \) (heat flux) equal (A) = 17351W/m², (B) = 49105W/m² and (C) = 80529W/m²

![Figure 7. Average heat transfer coefficient (\( \dot{h} \)) with a variable mass flow rate.](image)

**Figure 7.** Average heat transfer coefficient (\( \dot{h} \)) with a variable mass flow rate.

Figure 8 illustrates the variation of tube wall temperature with the axial distance (m) for five values of heat flux of (17351, 25680, 49105, 64423 and 80529W/m²) at different mass flow rate values of (0.008, 0.0166 and 0.0214kg/s) respectively. It can be shown that the increase in heat flux appears to rise tube wall temperature because of the rise in heating energy (\( q'' \)). It is to be noted, also, that the tube wall temperature is increased with tube length from the beginning of heating due to the heat accumulation along the tube length.
Figure 8. Wall temperature difference with a constant $\dot{m}$ (mass flow rate) equal to (A) $= 0.008$ kg/s, (B) $= 0.0166$ kg/s and (C) $= 0.0214$ kg/s.

The variance of $h$ with axial distance at specific values of heat flux (17351, 25680, 49105, 64423, and 80529 W/m²) and for different values of $\dot{m}$ (0.008, 0.0166 and 0.0214 kg/s) respectively are shown in Figure 9. It can be seen that there is a sudden reduction in the $h$ at the beginning of the region test, and the $h$ rises with heat flux. It was recognized that with a specific $\dot{m}$, there is a sudden rise in the $h$ at the boiling region. This effect is illustriously reduced when increasing the mass flow rate at a specific limit because of the bulk temperature of fluid drops lower than the saturated temperature. From Figure 10, it is noticed the $\text{Nu}$ (Nusselt number) increases with $\text{Re}$ (Reynolds number), this is due to the increase in a fluid disturbance with velocity (i.e., Reynolds number increase). From Figure 11, It is recognized that the rise of the pressure drop occurs when the $\dot{m}$ increases caused by the increase of friction resistance.
Figure 9. Surface heat transfer coefficient with a constant mass flow rate equal to (A)=0.008 kg/s, (B)= 0.0166 kg/s, and (C)= 0.0214 kg/s.

Figure 10. The correlation Nusselt number (Nu) versus Experimental (Nu).

Figure 11. Pressure drop at a variable mass flow rate.

4. Conclusions
The heat transfer of flow boiling results in small size tubes diameter were experimentally studied using water in a vertical copper tube. The experiment was carried out at a rate of mass flow varying from 0.008-0.0214kg/s, and heat flux from 17351-80529W/m². A digital camera observed and captured the regimes of two-phase flow. Moreover, the current experimental state five distinctive flow
patterns were detected for instance dispersed bubbly, bubbly, confined bubbly, chur, and slug flow. The following conclusions are made:

1. At the value (17351W/m²) of heat flux, mass flow rate rises from (0.008 to 0.0166kg/s), while decreasing the temperature of wall tube by 11% and 9% respectively when the value of mass flow rate rises from (0.0166 to 0.0214kg/s).
2. At the value (0.0166kg/s) of mass flow rate, the heat flux rises from (49105 to 64423W/m²), in contrast, the wall tube temperature increases by about (6%) and by approximately (13%) while increasing the heat flux value from (64423 to 80529W/m²).
3. The type of flow regime influences the average h. As well as, it was increased by about (37%) when the flow regime transfers from slug to churn flow at a (0.008kg/s) mass flow rate.
4. When the pattern of flow transitions from dispersed bubbly to confined bubbly flow at a (0.0214kg/s) mass flow rate, the rising of the average heat transfer coefficient is around by (17%).
5. In two-phase flow, the h is (39%), which is higher than the single-phase flow when heat flux increases from (17351W/m²) to (49105W/m²), and the mass flow rate is (0.0214kg/s).
6. When mass flow rate rises from (0.008kg/s) to (0.0166kg/s), the pressure drop increases by about (15%) and increases by about (5%) when the mass flow rate rises from (0.0166kg/s) to (0.0214kg/s).
7. When the boiling is not known, there is a strong agreement between the experimental findings and the Dittus-Boelter-built correlation.

Symbols
\( h \) : heat transfer coefficient (W/m²°C).
\( q \) : heat flux (W/m²).
\( T_s \) : surface temperature (°C).
\( T_f \) : fluid temperature (°C).

Reference

[1] B. K. Hardik and S. V. Prabhu, “Boiling pressure drop and local heat transfer distribution of water in horizontal straight tubes at low pressure,” Int. J. Therm. Sci., vol. 110, pp. 65–82, Dec. 2016.
[2] B. Ramesh and S. Gedupudi, “On the prediction of pressure drop in subcooled flow boiling of water,” Appl. Therm. Eng., vol. 155, pp. 386–396, Jun. 2019.
[3] J. Liu, Y. Jin, P. Zhao, Z. Ge, Y. Li, and Y. Wan, “Analysis of heat transfer of supercritical water by direct numerical simulation of heated upward pipe flows,” Int. J. Therm. Sci., vol. 138, pp. 206–218, Apr. 2019.
[4] L. Yin, P. Jiang, R. Xu, H. Hu, and L. Jia, “Heat transfer and pressure drop characteristics of water flow boiling in open microchannels,” Int. J. Heat Mass Transf., vol. 137, pp. 204–215, Jul. 2019.
[5] D. R. H. Beattie and P. B. Whalley, “A simple two-phase frictional pressure drop calculation method,” Int. J. Multiph. Flow, vol. 8, no. 1, pp. 83–87, Feb. 1982.
[6] S. Torfeh and R. Kouhikamali, “Numerical investigation of mist flow regime in a vertical tube,” Int. J. Therm. Sci., vol. 95, pp. 1–8, Sep. 2015.
[7] A. T. Wijayanta, T. Miyazaki, and S. Koyama, “Liquid–vapor phase distribution in horizontal headers with upward minichannel-branching conduits,” Exp. Therm. Fluid Sci., vol. 76, pp. 264–274, Sep. 2016.
[8] B. Markal, O. Aydin, and M. Avci, “Effect of aspect ratio on saturated flow boiling in microchannels,” Int. J. Heat Mass Transf., vol. 93, pp. 130–143, Feb. 2016.
[9] F. Illán-Gómez, A. López-Belchí, J. R. García-Cascales, and F. Vera-García, “Experimental two-phase heat transfer coefficient and frictional pressure drop inside mini-channels during
condensation with R1234yf and R134a,” Int. J. Refrig., vol. 51, pp. 12–23, Mar. 2015.

[10] G. Boye, J. Schmidt, and F. Beyrau, “Analysis of flow boiling heat transfer in narrow annular gaps applying the design of experiments method,” Adv. Mech. Eng., vol. 7, no. 6, p. 168781401558454, Jun. 2015.

[11] C. L. Ong and J. R. Thome, “Flow boiling heat transfer of R134a, R236fa and R245fa in a horizontal 1.030 mm circular channel,” Exp. Therm. Fluid Sci., vol. 33, no. 4, pp. 651–663, Apr. 2009.

[12] C. Y. Park and P. S. Hrnjak, “CO2 and R410A flow boiling heat transfer, pressure drop, and flow pattern at low temperatures in a horizontal smooth tube,” Int. J. Refrig., vol. 30, no. 1, pp. 166–178, Jan. 2007.

[13] H. J. Lee and S. Y. Lee, “Heat transfer correlation for boiling flows in small rectangular horizontal channels with low aspect ratios,” Int. J. Multiph. Flow, vol. 27, no. 12, pp. 2043–2062, Dec. 2001.

[14] G. Van der Meulen, “Churn-annular gas-liquid flows in large diameter vertical pipes,” 2012.

[15] J. R. Thome and L. Consolini, “Mechanisms of Boiling in Micro-Channels: Critical Assessment,” Heat Transf. Eng., vol. 31, no. 4, pp. 288–297, Apr. 2010.

[16] V. der Meulen and G. Pieter, “Churn-annular gas-liquid flows in large diameter vertical pipes,” 2012.

[17] T. Harirchian, S. G.-I. J. of H. and Mass, and undefined 2010, “A comprehensive flow regime map for microchannel flow boiling with quantitative transition criteria,” Elsevier.

[18] T. G. Karayiannis, D. Shiferaw, D. B. R. Kenning, and V. V. Wadekar, “Flow Patterns and Heat Transfer for Flow Boiling in Small to Micro Diameter Tubes,” Heat Transf. Eng., vol. 31, no. 4, pp. 257–275, Apr. 2010.

[19] S. Saisorn, S. W.-E. T. and F. Science, and undefined 2008, “Flow pattern, void fraction and pressure drop of two-phase air–water flow in a horizontal circular micro-channel,” Elsevier.

[20] B. X. Wang and X. F. Peng, “Experimental investigation on liquid forced-convection heat transfer through microchannels,” Int. J. Heat Mass Transf., vol. 37, pp. 73–82, Mar. 1994.

[21] T. L. Bergman and F. P. Incropera, Fundamentals of heat and mass transfer. Wiley, 2011.

[22] D. Barnea, “A unified model for predicting flow-pattern transitions for the whole range of pipe inclinations,” Int. J. Multiph. Flow, vol. 13, no. 1, pp. 1–12, Jan. 1987.

[23] X. Huo, L. Chen, Y. Tian, T. K.-A. T. Engineering, and undefined 2004, “Flow boiling and flow regimes in small diameter tubes,” Elsevier.

[24] Y. Whan Na and J. N. Chung, “Two-phase annular flow and evaporative heat transfer in a microchannel,” Int. J. Heat Fluid Flow, vol. 32, no. 2, pp. 440–450, Apr. 2011.