Experimental Investigation of Heat Transfer and Flow Characteristics in Different Inlet Subcooled Flow Boiling in Microchannel

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Abstract. The current experimental study considered the heat transfer and pressure drop for subcooled flow boiling of deionized water in a microchannel heat sink. The heat sink consisted of a single microchannel, 300μm wide and 300μm in height with a hydraulic diameter of 300μm. The heat sink was formed of oxygen-free cooper with 72mm length and 12mm width. Experimental operation conditions spanned the heat flux 78-800 kW/m², mass flux 1700 and 2100 kg/m².s and at 21, 31 K subcooled inlet temperature. Boiling heat transfer coefficient was measured and compared with existing correlations. Also, the experimental pressure drop was measured and compared with micro scale pressure drop correlations. The results show that higher mass flux leads to a higher boiling heat transfer coefficient and the dominant mechanism is convective boiling. Also, the experimental pressure drop decreases with increasing heat flux in a single phase region, while it increases in a two-phase region. The heat transfer coefficient increase with increasing the inlet subcooled. When compared the experimental results in the experimental condition range, it was found that an existing correlation provides satisfactory prediction of heat transfer coefficient and pressure drop.

1. Introduction

In recent years, flow boiling through microchannels has attracted lots of interest because in compact areas, it has potential for very elevated heat transfer rates. When the requirement is for smaller rates of coolant flow by using the latent heat of the coolant, higher heat fluxes can be dispersed significantly by flow boiling as compared with its single-phase counterpart [1]. The phase variation process happens at the fluid saturation temperature with higher temperature constancy through the microchannel heat sinks, and that is another characteristic of the convective boiling process.

In recent years, flow boiling heat transfer properties have been studied extensively in particular and efforts have been made to discover the heat transfer mechanisms at small scale. However, various heat transfer properties were adduced under various experimental conditions such as working liquid, channel geometry, the used mass flux and heat flux and saturation pressure, etc. In spite of that, there are yet numerous existing problems that limit the additional implementation of the micro-channel, e.g., high wall temperature at the start of boiling (O.N.B)[2], large wall-temperature differences between the channels and over the channel, as well as the instability of the two-phase flow.

Subcooled boiling is predominantly noticed in a micro-channel heat sink. This can maintain comparatively low wall temperature under highly subcooled conditions, and generate high heat transfer rates. When the hot of the surface is enough for the formation of bubbles but the bulk liquid temperature stays under its saturation value, subcooled flow boiling subsists. The onset of nucleate boiling, ONB, is called on the initial state of bubbles. According to the classical theory [3,4] as the bubbles that created...
at the wall proceed away from the developing saturation boundary layer, they will precipitate, but the heat transfer between the fluid and the wall will be impacted by the presence of these bubbles. During high levels of subcooling or low heat fluxes, just a few nucleation sites are active and part of the heat is transported by single-phase convection among patches of bubbles. This system is called partial nucleate boiling. Numerous nucleation sites are activated until fully-developed nucleate boiling as the heat flux is developed, when the surface becomes totally active for nucleation. Subsequently, the saturated nucleate boiling zone is entered when the saturation boundary layer develops and finally covers the whole channel as the bulk fluid is heated.

The effects of inlet temperature is one of the most important parameters affecting the efficiency of the microchannel heat sink, so many investigators have turned their attention to this, such as Peng and Wang [5] who studied flow boiling using deionized water in a rectangular micro channel for an evaporator application made from stainless steel. The channel had a 0.86 aspect ratio (W/H), 60mm length and 0.65 mm hydraulic diameter. The experimental setup was an open loop and the liquid subcooling ranged from 40K to 70K. In that study, although a fully developed nucleate boiling was noticed from the boiling curve, no bubbles were seen in the channel. The authors ascribed this phenomenon to the reality that the size of the channel was much less than the “minimum evaporating space” required for bubble development. Also, they did not describe any influence for inlet subcooling and mass flux.

Qu and Mudawar [6] investigated mechanisms of convective boiling by the experimental study of 21 copper micro channel of 349 hydraulic diameter with 30 °C – 60 °C inlet temperature condition at a range of heat fluxes (400-2400 kW /m²) and 135-402 kg/m² mass fluxes. The tested substance was deionized water. They found the dependency of heat transfer coefficient on mass flux and vapor quality but independent of heat flux.

Galvis and Culham [7] tested flow boiling pressure drop and heat transfer in two dissimilar microchannel test sections, using deionized water as a substance. The sections had the same length and aspect ratio but different hydraulic diameter (217, 419 µm) with different inlet subcooling (50.7 °C , 54.2 K). The tests were conducted with a range of mass fluxes (340, 1373 kg/m²), and 31.7 –1414 kW/m² heat fluxes. The authors found that the nucleate flow boiling mechanism was the dominant heat transfer mechanism due to the effect of heat flux on the coefficient of heat transfer, while the effect of mass fluxes on the heat transfer coefficient was insignificant.

Krishnamurthy and Peles [8] investigated subcooled flow boiling of HFE-7000 in 222 µm hydraulic diameter, with mass flux and heat flux of range 350-827 kg/m², 100-1100 kW /m² respectively. Channels contained a single row of pin fins; significant heat transfer enhancement was found and it was observed that the local heat transfer coefficient during sub cooled boiling was higher than the corresponding single-phase flow.

Claudi Martín-Callizo et al. (2007) [9] studied subcooled flow boiling heat transfer in vertical cylindrical tubes of internal diameter 0.83, 1.22 and 1.70 mm for refrigerant R-134a, experimentally. They examined the impacts of the inlet subcooling, channel diameter, mass flux, heat flux and system pressure on the subcooled boiling heat transfer and found that the wall superheat at ONB was significantly greater than that predicted with correspondences for larger tubes. For early subcooled boiling, they found that a rise of the mass flux gave a rise in the heat transfer coefficient. Increases in the mass flux occurred in a small development of the heat transfer for fully developed subcooled boiling. Smaller channel diameter, higher system pressure and higher inlet subcooling led to better boiling heat transfer.

In the present study, experiments were conducted to investigate subcooled flow boiling heat transfer and flow characteristics along the copper microchannel by using deionized water as a working fluid. The microchannel dimension is 300µm hydraulic diameter and aspect ratio of 1. This test has been conducted for various experimental conditions, mass fluxes (1700, 2100 kg/m³), heat fluxes (range 78-800 kW/m²) and 21, 31K inlet subcooled temperature.
2. Experimental apparatus and data reduction

2.1 Experimental setup
The experimental facility consists of a liquid tank, sub-cooler, peristaltic pump, turbine flow meter, pre-heater, test section, and inline filters. Schematic diagram and a plate of the experimental set-up are displayed in Figure 1 and 2, sequentially. A chiller unit is used for cooling purpose in sub-cooler and the condenser. Substance was deionized water. Vigorous boiling for about one hour was used for

Figure 1. Schematic diagram of the experimental facility.
degassing of the substance in the fluid tank. In order to release gases that are non-condensable to the atmosphere, the top valves of the condenser were opened.

To separate all particles in the water, 7 µm filter was installed before the peristaltic pump in the suit. After that, water free gas was introduced into the microchannel by pumping, while a preheater was used to control the working fluid inlet temperature. The microchannel test section is designed and machined from oxygen free copper blocks; hence, the block of the micro-channel test section has dimensions of 12 mm width, 25 mm height and 72 mm length. Single microchannel with a length of 62 mm was cut into the top surface of the copper blocks between the 2 mm diameter inlet and outlet plenums using milling machine at a feed rate of 10 mm/min. The typical dimensions of the micro-channel are 300 µm width and 300 µm depth. These dimensions were measured utilizing an electron microscope and the actual values are 367 µm for width and 269 µm for height, as shown in Figure 3.

The surface roughness at the bottom of the microchannel test sections were measured with an AA3000 scanning probe microscope (atomic force microscope AFM contact mode) which has multi-analysis: granularity and roughness.

The average surface roughness value was 0.011µm. Figure 4 shows the roughness values that were evaluated over sample areas of 2cm x 1cm. A cartridge heater of 250 W heating power was inserted to the copper block at the bottom, in a region parallel to the flow, which was placed in a drilled (8mm) hole within the copper block in order to supply the test section with the heating power. The local axial wall temperature was estimated in the 1 mm inner diameter and 6 mm depth hole at the portion of the block of copper. To estimate the local heat transfer coefficient along the channel, six holes were located over the axial side of the channel at equal distances of 12.4 mm with 1 mm from the bottom of the channel to accommodate K-type thermocouples. Also, from the inlet of the channel with a distance of 24.8 mm, two holes were located vertically and 5 mm below the axial holes with interval of 5 mm. In order to seal against leaks among microchannels and the top cover, on the top face of the microchannels, an O-ring slot was machined.
To reduce heat loss to the environment, the block was inserted into an insulated fiber glass sheet housing, then the assembled is inserted into the housing component which was designed and manufactured from stainless steel to accommodate the test section in the test loop. Between the stainless-steel cover plate and the upper surface of the copper block, a 4 mm thickness of polycarbonate transparent layer was sandwiched. Then two holes were drilled at two locations in the top cover polycarbonate to accommodate thermocouple, one of them at the inlet plenum while other was at the outlet plenum of the microchannel test section to accommodate differential pressure drop across the microchannel test sections.

Figure 5 presents the main part of the test section. During the present study, all thermocouples were calibrated with ± 0.5 K of uncertainty. A differential pressure transducer (26pctfT6D) was used to estimate the pressure drop. It was calibrated with an uncertainty of ±0.5 kPa. The data were listed after steady state condition for 5 min using a data acquisition system of type AT4532x.

![Figure 3. Measurements of surface roughness for microchannel](image1)

![Figure 4. Microscope picture of microchannel](image2)

![Figure 5: Microchannel constructions showing the main parts](image3)

2.2 Data reduction
2.2.1 Single-Phase flow data reduction

For single phase flow, the net pressure drop over the micro channel $\Delta P_{ch}$ is given by:

$$\Delta P_{ch} = \Delta P_{meas} - \Delta P_{loss}$$  \hfill (1)

$\Delta P_{meas}$ represents the total pressure drop among the inlet and exit plenums. The differential pressure sensor is used to measure it directly. $\Delta P_{loss}$ is the pressure loss due to the sudden enlargement and contraction and the inlet and outlet manifolds, it is described by Eq.2 below

$$\Delta P_{loss} = 2\left(\frac{1}{2}\rho_l V_p K_{90}\right) + \frac{1}{2}\left(\rho_l V_{ch}(K_c + K_e)\right)$$  \hfill (2)

In Eq. (2), $V_p$ represents plenum liquid velocity while $V_{ch}$ represents channel liquid velocity. The direction of entering and leaving flow in the channel is normal to the flow direction. Here, the loss coefficient according to 90° turns of the flow through inlet and outlet plenums is represented by $K_{90}$ and that is specified by Philips (1987) [10] as 1.2.

For the channel entry and exit, sequentially, $K_c$ is the sudden contraction loss coefficient and $K_e$ is the sudden enlargement loss coefficient. Tabular data introduced by Shah and London [11] can be used for their values by interpolating.

$(h_{sp}(z))$ that is the local single-phase heat transfer coefficient as well as $(Nu)$ which represents the average Nusselt number are determined as:

$$h_{sp}(z) = \frac{q''}{T_w(z) - T_f(z)}$$  \hfill (3)

$$Nu = \frac{1}{L_0} \int_0^L h_{sp}(z) \, dh \, dz$$  \hfill (4)

The 1D heat conduction equation used for the correction of the channel wall temperature $(T_w(z))$ at axial location $z$ is presented as Eq. (5). The liquid thermal conductivity is represented by $k_l$ and $T_f(z)$ is determined using Eq. (6) below, considering uniform heat flux boundary conditions and depending on an energy balance. $(q'')$ that is the heat flux was described in Eq. (7).

$$T_w(z) = T_{tc}(z) - \frac{q'' \cdot t}{K_{Cu}}$$  \hfill (5)

$$T_f(z) = T_i + \frac{q'' \cdot wz}{m \cdot C_p}$$  \hfill (6)

$$q'' = \frac{P - Q_{loss}}{A_{ht}}$$  \hfill (7)

where: the local thermocouple reading is represented by $T_{tc}(z)$, the thermal conductivity of copper is represented by $k_{Cu}$ and the dimension from the base of microchannel to the thermocouple position represented by $(t)$ and equal to (1 mm). The temperature of inlet fluid and the specific heat of liquid are represented by $T_i$ and $C_p$ respectively. The applied electrical power symbolized by $P$, the heat transfer area is represented by $A_{ht}$ and it is described in Eq. (8). When there is no fluid inside the test section, an electrical power $(P)$ applies to it in order to evaluate $(Q_{loss})$, which is the heat loss from the test section. Subsequent to achieving steady state, the temperature difference among ambient and the bottom wall was listed for all heating power. After that, in order to achieve an equation to determine the heat loss $(Q_{loss})$ through single-phase and boiling tests, data obtained from the plotting of the applied power against this temperature difference.
A_{ht} = (2H + W)L \tag{8}

Where H, W and L are the high of the channel, the width and the length of it sequentially.

2.2.2 Two-Phase flow data reduction

The subcooled liquid state is the condition at which the fluid will enter the channel. After that, the channel is separated to two regions, single-phase and two-phase. The region that begins from the test section entrance to the position of zero thermodynamic quality is the single-phase region with length L_{sub}. Therefore, (L_{sat}) which represents the two-phase region length becomes

\[ L_{sat} = L - L_{sub} \tag{9} \]

Next equations are used for the calculation of (L_{sub}) which is the length of the single-phase region and it is determined iteratively.

\[ L_{sub} = \frac{m \cdot \rho_p (T_{sat} - T_i)}{q'' (2H+w)} \tag{10} \]

\[ P_{sat}L_{sub} = P_l - \frac{2f_{app} \cdot L_{sub}^2}{\rho_l D_h} \tag{11} \]

\[ f_{app} = \frac{3.44}{Re \sqrt{L_{sub}^2}} + \frac{k \cdot \frac{4 \cdot L_{sub}}{L_{sub}^2}}{Re \cdot (1 + C(L_{sub}^2 - 3.44))} \tag{12} \]

T_{sat} is the liquid saturation temperature while the apparent friction factor that was taken from Shah [12] is represented by f_{app}. Shah [12] introduces the (f \cdot f_d \cdot Re), K(\infty) and C constant values mentioned in Eq. (12) for rectangular channels. (L''_{sub}) that is, the dimensionless length in Eq. (12) can be calculated using Eq. (13). (f \cdot f_d \cdot Re) which is represents the fully-developed Poiseuille number is provided by Shah and London [11] in Eq. (14) as a function of channel aspect ratio (\beta).

\[ L''_{sub} = \frac{L_{sub}}{Re D_h} \tag{13} \]

\[ f \cdot f_d Re = 24(1 - 1.3553\beta + 1.9467\beta^2 - 1.7012\beta^3 + 0.9564\beta^4 - 0.2537\beta^5) \tag{14} \]

\( \beta \) is the aspect ratio =1. In the two-phase section, the local pressure was supposed to reduce linearly with z (the axial length) and it is able to be determined from:

\[ P_{sat}(z) = P_{sat}L_{sub} - \frac{z - L_{sub}}{L - L_{sub}} \Delta P_{tp} \tag{15} \]

The net pressure drop of the flow boiling part in the channel is measured from:

\[ \Delta P_{tp} = \Delta P_{ch} - \Delta P_{sp} \tag{16} \]

(\Delta P_{sp}) which is the pressure drop of the single-phase is able to be calculated depending on Eq. (10) and Eq. (12) that measured the single-phase region length (L_{sub}) and the apparent friction factor respectively. Local heat transfer coefficient of the two-phase was determined as:

\[ h_{tp}(z) = \frac{q''}{T_w(z) - T_{sat}(z)} \tag{17} \]

(T_{sat}(z)) temperature in Eq. (17) that is the local saturation is determined depending on the local pressure presented by Eq. (15). (x(z)) is the vapor quality and it was calculated by Eq. (18) and Eq. (19).
\[ i(z) = i_l + \frac{q''(2H+W)z}{m} \]  
(18)

\[ x(z) = \frac{i(z) - i_l}{i_L(z)}(z) \]  
(19)

Depending on the measured inlet temperature \( T_i \) and inlet pressure \( P_i \), the local inlet specific enthalpy \( i(z) \) is calculated. In Eq. (19), \( i_L(z) \) represents local liquid specific enthalpy while \( i_L(z) \) represents local enthalpy of vaporization.

3. Results and discussions

3.1 Single-Phase Results

The single phase experiments were carried out before the boiling experiments, in order to confirm the experimental system. The comparison between the calculated friction factor and the Shah and London [12] correlation for developing and fully developed flow is represented in Figure 6. The figure proves that there is a good contract with correlations. Figure 7 shows the experimental Nusselt number compared with the predictions from the correlations of Shah and London [11], Mirmanto [13] and Mehmed [14]. It is clear that the experimental values show a similar trend where the Nusselt number increases with the Reynolds number. However, the current experimental results agree very well with the experimental results of Shah and London [11] for developing laminar flow.

3.2 Two-Phase Results

3.2.1 Boiling curve

The effects of inlet liquid subcooling on the subcooled boiling curves at mass flux 1700 kg/m² and wall temperature located at \( z/l = 0.6 \) are illustrated in Figure 8. These results indicate that at mass flux of 1700 kg/m².s, the wall temperature of boiling increases by increase inlet subcooling, i.e., the boiling starting is triggered at \( (T_w-T_{sat}) \) 0.6 and 1.2 K where the subcooling are \( \Delta T_{sub}=21 \) and 31K respectively.

The effects of the liquid mass flux on the subcooled boiling curves at low and high mass flux (1700 and 2100 kg/m².s) and degree subcooled (31K) are illustrated in Figure 9.
Figure 6. Single phase results, Fanning friction factor versus Reynolds number.

Figure 7. Single phase results, Nusselt number versus Reynolds number.

The results indicate that for given boiling curve, at low imposed heat flux, the temperature of the heated wall is below the saturated temperature of deionized water, the wall superheated is negative ($T_\text{w} - T_\text{s}\text{at} < 0$), and the heat transfer in the microchannel is completely due to single phase liquid forced convection. As the imposed heat flux is raised gradually, the heated wall temperature increases slowly to exceed $T_\text{s}\text{at}$ at a certain $q$, i.e., the wall superheat ($T_\text{w} - T_\text{s}\text{at}$) is positive. When the positive wall superheat
reaches a certain critical level, a small increase in \( q'' \) causes boiling, this is apparent from the figures, in that there is a steep change in slope of the plots indicating the onset of nucleate boiling (ONB), which implies the transition from single phase region to flow boiling region.

Figure 8. Effect of inlet subcooled temperature and heat flux on the boiling curve at mass flux 1700 kg/m\(^2\).K

Figure 9. The impact of mass flux upon the boiling curve at 31K subcooled inlet temperature
3.2.2 Effect of heat flux on heat transfer

The effects of operation parameters such as heat flux, mass flux, and inlet of liquid subcooled and artificial are presented in Figures 10 and 11. The effect of heat flux on the local heat transfer coefficient is discussed for same operational conditions of mass flux and degree of subcooled temperature.

Figure 10. Dimensionless distance from the inlet and low heat flux effectiveness on the local heat transfer coefficient at mass flux of 1700 kg/m$^2$.s and subcooled inlet temperature of 31K

Figure 11. The impact of dimensionless distance from entry and high heat flux upon the Local heat transfer coefficient at a mass flux of 1750 kg/m$^2$.s and subcooled entry temperature of 31K
However, to determine the effect of heat flux on heat transfer coefficient clearly, the heat flux values are distinguished into the form of low, moderate, and high heat flux input. The water entered the channel in a subcooled condition, and was progressively heated up along the test section. So, the flow boiling is divided into two regions (Qu and Mudawar, 2003): subcooled and saturated. In the subcooled region, the heat flux range is low and moderate, while in the saturated region, the heat flux input is high.

Figure 10 shows the heat transfer coefficient is increased with increasing heat flux in the single-phase region, because the thermal boundary layer is not fully developed. Also, the heat transfer coefficient increased along the microchannel test sections’ length for the fixed heat flux because the effect wall temperature of microchannel increased in the axial direction due to the axial heat conduction effect. Figure 11 shows that for the same point, increasing heat flux led to decreasing local heat transfer coefficient. The same results were found by other researchers, such as Liu et al. [15], and Lee and Mudawar [16], who reported that the reason for the decrease of two-phase heat transfer (h_{tp}) coefficient was partial dry-out.

Figure 12 shows the variation of local heat transfer coefficient with heat flux for two mass fluxes at the position of z/l = 0.6. The figure is divided into two regions; subcooled flow boiling region and saturated flow boiling region. The two-phase region starts at the boiling incipience, where the peak value for heat transfer coefficient is reached and then declines sharply. The figure also shows that the heat transfer coefficient increases by increasing mass flux for both the subcooled region and saturated region at constant heat flux. Also the boiling incipience is delayed for the higher mass flux and needs higher heat flux for boiling incipience.

To fully understand the heat transfer mechanism, the local heat transfer coefficient with mass flux 1700 kg/m$^2$.s and 2100 kg/m$^2$.s is shown in Figure 12 at degree of subcooled 31K. This figure presents the situation at the thermocouple, near the microchannel exit (Z/L = 0.8), because that is the highest value of saturated flow boiling and the highest quality of vapor. The figure is divided into two regions subcooled flow boiling region and saturated flow boiling region. The two-phase region starts at the boiling incipience, where the peak value for heat transfer coefficient is reached, and then declines sharply. The figure also shows that the heat transfer coefficient increases by increasing mass flux for both subcooled and saturated regions at constant heat flux. And the boiling incipience is delayed for the higher mass flux and need higher heat flux for boiling incipience.

### 3.2.3 Effect of mass flux on heat transfer

Figure 12 shows the variation of local heat transfer coefficient with heat flux for two mass fluxes at the position of z/l = 0.6. The figure is divided into two regions, subcooled flow boiling region and saturated flow boiling region. The two-phase region starts at the boiling incipience, where the peak value for heat transfer coefficient is reached, and then declines sharply. The figure also shows that the heat transfer coefficient increases by increasing mass flux for both subcooled region and saturated region at constant heat flux. And the boiling incipience is delayed for the higher mass flux and needs higher heat flux for boiling incipience.

To fully understand the heat transfer mechanism, the local heat transfer coefficient with mass flux 1700 kg/m$^2$.s and 2100 kg/m$^2$.s will be illustrated in Fig. 12 at degree of subcooled 31K. These measurements are taken at thermocouple near the exit of the microchannel (Z/L = 0.8), that relates to the greatest amount of saturated boiling and vapor quality. The figure is divided into two regions, subcooled flow boiling region and saturated flow boiling region. The two-phase region starts at the boiling incipience where the peak value for heat transfer coefficient is reached and then decline sharply. The figure also shows that the heat transfer coefficient increase by increasing mass flux for both subcooled and saturated region at constant heat flux. And the boiling incipience is delayed for the higher mass flux and needs higher heat flux for boiling incipience.
Figure 12. Mass flux effectiveness upon the local heat transfer coefficient at subcooled inlet temperature of 31 K.

### 3.2.4 Effect of inlet temperature on heat transfer coefficient

The subcooling $\Delta T_{sub}$ is defined as the difference between the saturation temperature $T_{sat}$ and the local bulk temperature $T_b$. Figure 13 shows the local two-phase heat transfer coefficients as a function of heat flux at two inlet subcooled temperature. The results show that the heat transfer coefficient is highly related on the degree of subcooling.

Figure 13. Effect of inlet temperature on the local heat transfer coefficient at mass flux of 1700 kg/m$^2$.s
As presented in Figure 13, the high inlet subcooled degree (low inlet temperature)\((T_{in} = 69 \, ^\circ C)\), \((\Delta T_{sub} =31)\) has higher heat transfer coefficient while the heat transfer coefficient has the lower value at \((T_{in} = 79 \, ^\circ C)\), \((\Delta T_{sub} =21)\). A higher heat flux is required to instigate boiling on the heated surface for a higher degree of subcooling. Whereas the higher heat flux means that the surface is more active for nucleation and more bubbles will be generated, which explains the higher heat transfer coefficient with lower degree of subcooled temperature \((\text{lower bulk liquid temperature})\).

### 3.2.5 Pressure drop

Two-phase pressure drops in microchannels are considered one of the most important parameters in microchannel design. Figures 14 and 15 indicate the data pressure drop inconstant with heat flux at two mass fluxes \((1700 \, \text{and} \, 2100 \, \text{kg/m}^2.\text{s})\) at a subcooled degree of 31 K. The figures are divided into two sections; subcooled region (single phase, all liquid section) near the inlet of the microchannel, where a deionized water flow under subcooled condition can be seen. In this section there is an increase of heat flux, the pressure drop shows a somewhat decreasing tendency due to decreasing flow resistance owing to the reduction of dynamic viscosity in the liquid phase with increasing temperature. However, the effect of heat flux on the pressure drop in the saturated region, which represents the second section, is different to the single-phase flow where the experimental pressure drop rises at the same conditions due to a mounting evaporation rate. Also, by likening the pressure drop with mass flux at different heat flux it seems that at higher mass flux the inertia force overcomes the evaporation momentum force, which increases the acceleration pressure drop. Also the frictional pressure drop component becomes higher at high mass fluxes.

The microchannel pressure drop in the microchannel is a crucial parameter in its design. Figures 14 and 15 show the pressure drop versus the wall heat flux for two mass fluxes \((1700 \, \text{and} \, 2100 \, \text{kg/m}^2.\text{s})\) and degree of subcooled 31 K. As shown in figures, the pressure drop with heat flux for a given mass flux in single phase is decreased with increasing wall heat flux. Increasing the wall heat flux (increasing the wall temperature) causes reduction in liquid viscosity (reduction of flow resistance) that lead to a decrease in pressure drop. The trend of pressure will change as boiling occurs. Here, increasing wall heat flux increases the pressure drop.

![Graph showing pressure drop vs. heat flux with two data points](image-url)
4. Comparison with exiting prediction methods

4.1 Micro-scale heat transfer correlation

The heat transfer coefficient, divined by two correlations, which are Lee and Mudawar [17] and Cooper [18], is compared with the calculated heat transfer coefficient in the present work. The mean absolute error percentage (MAEP) is used for evaluating the accuracy of each correlation as follows:

$$\text{MAEP} = \frac{1}{2} \sum \left| \frac{h_{\text{exp}} - h_{\text{pred}}}{h_{\text{exp}}} \right| \times 100$$  \hspace{1cm} (19)

A relationship in a rectangular microchannel, having a 0.35 mm hydraulic diameter for heat transfer of flow boiling, is suggested by Lee and Mudawar [1]. Data points for 318 heat transfers of water and refrigerant R134a are used for the correlation. The correlations of Lee and Mudawar [17] give good prognostications, having less than 20 % of mean absolute error as shown in Figure 16.

The effect of surface roughness was introduced by Cooper correlations [18] due to the importance of accurately accounting for the influence of surface roughness when correlating nucleate boiling data, the correlation predicts that there is suppression for the convective boiling enhancement factor. Fig. 17 depicts a comparison the experimental data with Cooper correlations [18]. The correlation indicate poor prediction with the experimental data the MAE is 33.7%.

4.2 Micro-scale pressure drop correlation

The experimental flow boiling pressure drop data for the Microchannel were compared to microscale pressure drop correlation. The comparison of the experimental flow boiling pressure drop data with these correlations are presented for two mass fluxes (1700 , 2100 kg/m$^2$.s) and two inlet subcooling degree (31, 21K) based on the mean absolute error (MAE) defined by Eq. (20).
\[
\text{MAE} = \frac{1}{N} \sum_{i=1}^{N} \left| \frac{(\Delta p_{\text{meas.}} - \Delta p_{\text{pred.}})}{\Delta p_{\text{meas.}}} \right| \times 100\% \tag{20}
\]

Figure 16. Comparison of experimental results with micro-scale correlation

Figure 17. Comparison of experimental results with micro-scale correlation

The experimental data of the current study was compared with two micro-scale pressure drop correlations; Qu and Mudawar [19] and Lee and Garimella [20].

Qu and Mudawar [19] modified the correlation of Mishima and Hibiki [21] by incorporating mass velocity term and the effects of channel size. Mishima and Hibiki [21] proposed correlation based on experimental data with $Dh$ (1.05-3.9) mm glass and aluminum vertical tubes and the use of air-water flow. Their data was well corroborated by Chisholm’s correlation with a modified Chisholm’s parameter (C) as a function of inner diameter. Qu and Mudawar [19] studied channels that had 0.35 mm hydraulic diameter and used water as working fluid. Figure 18 shows the global comparison with the experimental data. The correlation predicts the current experimental data, with MAE value about 24.4%. The correlation of Qu and Mudawar [19] with the experimental data may due to using deionized water as a test fluid, which is the same as the current study; also the subcooled inlet condition – the 0.35 mm hydraulic diameter is close to the hydraulic diameter of this study, at 0.3 mm.
The Lee and Garimella [20] correlation is used for the laminar liquid/laminar vapor regime and takes into account the effect of hydraulic diameter and mass flux. They modified the Chisholm parameter of the Lockhart-Martinelli [22] correlation. The microchannel depth was 400 μm in each case, while the width ranged from 102 μm to 997 μm. They used deionized water as a working fluid. Figure 19 shows the correlation predicts the experimental data with an MAE value of 27%.

5. CONCLUSIONS
Subcooled flow boiling experiments in a copper single microchannel heat sink, using deionized water, were performed for a mass flux range 1700–2100 kg/m².s and heat flux range 78–800 kW/m² and degree of subcooled degree of 31, 41 K.
The main conclusions drawn from this study are:
1. The correlations of fully developed flow and the conventional scale developing flow in the laminar regime were totally compatible with the experimental friction factor data.
The influence of mass flux on boiling curves was clear which shows that the mechanism is convective.

The results of heat transfer described that heat transfer coefficient based substantially on the heat flux where the two-phase heat transfer coefficient reached peak value at boiling incipience, while in the single-phase region the heat flux had little effect on heat transfer. The heat transfer coefficient was increased when the mass flux increased.

Inlet temperature had an obvious effect on the onset nucleate boiling heat flux, with increasing degree of subcooled (decreasing inlet temperature) the onset nucleate boiling heat flux increased. Also, the heat transfer coefficient increased with increasing subcooled fluid inlet temperature (decreasing inlet temperature).

The pressure drop across the microchannel in the single-phase region decreased with increasing heat flux, this trend is reverse in the two-phase region.

The heat transfer and pressure drop correlation predicted the experimental data well, depending on experimental conditions.

6. REFERENCES

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