Experimental Investigation of Sub-Cooled Flow Boiling Heat Transfer in Single Rectangular Metallic Micro-Channel

Qahtan A Al-Nakeeb, Ekhasl M Fayyadh and Moayed R Hasan
Mechanical Engineering Department, University of Technology, Baghdad, Iraq
Email: 20077@uotechnology.edu.iq

Abstract. In this study, an experimental investigation on the flow boiling heat transfer in single rectangular micro-channel was carried out. The micro-channel was formed by machining 300 µm wide x 700 µm deep groove (i.e. a hydraulic diameter 420 µm) into the surface of brass block having length of 60 mm. The working fluid was deionized water. The experiments were carried out with the mass flux range (300-600 kg/m².sec), and flux of heat (based on wall) 5.4 to 376.5 kW/m², inlet sub-cooling of 20 K and 1 atm. working pressure. The results of this study have shown that at heat fluxes with lower range, the local coefficient of heat transfer increased with increasing the flux of heat. This trend is reversed when heat fluxes exceed 64.2, 77.9, 93.8 and 116.7 kW/m² for mass fluxes 300, 400, 500 and 600 kg/m².sec respectively, where the heat transfer coefficient decreases as the heat flux increases and reach a constant value at the highest heat flux. The experimental results are compared to heat transfer correlations of minichannel and microscale, these correlations gave the results reasonably well.

Keywords. Microchannel, Single-phase flow, Two-phase flow, Sub-cooled flow boiling, Heat Transfer.

1. Introduction

The past several years have seen unprecedented increases in computer processor efficiency which has been induced, for the most part, by a restless pursuit of micro-miniaturization of components in the processor itself. Computer processors are not limited to the significant increase in heat dissipation per unit surface area and per unit volume, but also for these systems, heat dissipation rates have already risen to levels that could no longer be controlled using traditional cooling techniques. In recent years, many cooling schemes have been produced capitalizing on the merits of phase change to achieve the desired cooling effectiveness. These include thermosyphons for pool boiling and channel flow boiling. A special type of channel flow boiling systems is the two-phase micro-channel heat sinks. In such applications as rockets, avionics, and portable computers, these characteristics have made these heat sinks a prime contender for small, lightweight cooling systems. Finally, the realistic merits of two-phase micro-channel heat sinks have drawn significant attention to many issues relating to their characteristics of phase change [1]. Qu and Mudawar (2003) investigated flow boiling heat transfer in a multi microchannel heat sink using deionized water as a working fluid. The test part was composed of 21 rectangular microchannels measuring 231 µm width and 713 µm depth. The heat sink was built out of free oxygen copper. The experiment was conducted at mass flux with ranges of (135 - 402 kg/m².sec), inlet working fluid temperatures were 30 °C and 60 °C and an exit pressure 1.17 bar. It was found that the coefficient of heat transfer of saturation water flow boiling depends on mass flux and its weak function of heat flux at moderate as well as high heat flux. So, the dominate heat transfer for
deionized water flow through microchannels heat sink is forced convective boiling mechanism [2]. Liu and Garimella (2007) explored the water flow boiling heat transfer in two sizes of microchannels (275x636 and 406x1063 µm). The experiment conducted at inlet temperature of 67-95 °C, and range of mass fluxes with (221-1283 kg/m².sec). It was found that the degree of sub-cooling and velocity had a slight effect on the onset nucleation boiling in the boiling curve [3]. Studying the heat transfer characteristics for two phase heat sink at low temperature by using indirect refrigeration cooling system in rectangular microchannels was carried out by Mudawar and Lee (2008). The authors conducted the experiments at a range of the hydraulic diameter within the values:175.7, 200, 334, and 415.9 µm with a different range of working fluid temperature from -30 °C to 0 °C using a pre-cooled of the fluid working substance (Hydrofluoroether (HFE)). The microchannel was made of oxygen free copper blocks. It was found that increasing sub-cooled would cause the decreasing of the two phase pressure drop. Also, two phase cooling performance of microchannel increased when decreasing the hydraulic diameter as a result of increasing the wetted area and the mass velocity [4]. Additionally, Mudawar and Lee, (2008) extended their work to study the effect of decreasing working fluid temperature on the onset of boiling. It was found that the onset of boiling would be delayed simultaneously, it reduced the size of bubble and coalescence effects that lead to enhance the performance of cooling system [5]. Harirchian and Garimella (2009) studied the influence of mass flux (225-1420 kg/m². sec) and the microchannel sizes on the flow boiling patterns by using high speed camera. Multichannel was grooved on the silicon heat sink. 5 K of sub-cooled of Fluorinert F-77 at the inlet of the channels, seven different width from100 µm to 5850 µm, and all the depth is 400 µm are studied. The heat transfer data showed that flow regimes with microchannel width 400 µm and over were the same with nucleate boiling dominate over the range of heat fluxes, while in channels with narrower width, convective boiling dominates. Five flow patterns were recognized. These patterns were: bubbly, slug, churn, wispy-annular and annular flows. Bubble nucleation at the wall was suppressing at low heat flux with the smaller sizes of microchannel [6]. However, in a highly degree of sub-cooled, the heating microchannel surfaces collapsed the coalescing bubbles into many fine bubbles. This regime is called Microbubble Emission Boiling (MEB). Suzuki et. al. (2009) conducted the sub-cooled of water in the horizontal rectangular multichannel heat sink, which consisted of 11 channel having hydraulic diameter 155 µm. This heat sink was made of copper block. Maximum pressure in the channel was 25 mm Hg and the outlet channel was opening to atmosphere. At the higher heat flux, the microchannels were completely covered with large coalescing bubbles where the coalescing bubble covered a whole heating surface. The length of the heating unit and the amount of heating played a significant role creating a microbubble emission boiling through the channels [7]. Moreover, Galvis and Culham (2012) investigated the water flow boiling features in two unlike copper single microchannel. The length and aspect ratio remained the same, but with hydraulic diameters of 0.217 mm and 0.419 mm. The experiments were accomplished with a range of mass fluxes 350 - 1373 kg/m².s, and heat fluxes (31.7-1414) kW/m² at two different inlet sub cooling (50 and 70) K. It was reported that nucleate boiling was the dominated mechanism of heat transfer for a smaller microchannel also vice versa with the bigger microchannel where the convective boiling possibly would dominate over nucleate boiling mechanism [8]. The effect of mass and heat fluxes on the coefficient of heat transfer was studied by Mehmed et. al. (2016). The authors conducted an experiment in a single micro channel with 1mm width and 0.39 mm height which was made of the oxygen copper block surface. The operation parameters were inlet temperature of 89 °C, mass flux from 200 up to 800 kg/m².sec and heat flux with range of 56-865 kW/m². High speed and high resolution camera was used to capture flow pattern. At all mass fluxes, the unstable flow boiling were observed at boiling incipience. At low level of mass and heat fluxes, the heat transfer coefficient depended on the heat fluxes only. The visualization method proved that the bubbles nucleate were rapidly growth without departure from the channel surface. However, when elongated bubbles were formed, instability with great fluctuation were inevitable [9]. Yin et al. (2017) investigated the water flow boiling with high aspect ratio in microchannels having hydraulic diameter of 571 µm. The experiments were carried out at inlet temperature of 65 °C, ranges of mass flux 261-961 kg/m².sec, and range of heat flux 631-987 kW/m². It was observed that nucleate boiling dominated through the high aspect ratio microchannels i.e. the heat transfer coefficient generally depended on the heat flux
instead of the mass flux. The bubble confinement influences on the heat transfer through the microchannel; it was significant in small and moderate aspect ratio of microchannels. In contrast, the confinement bubbles did not occur at large aspect ratio of microchannel [10]. Mohammed and Fayyadh (2020) carried out an experiment in single microchannel heat sink to study the consequence of degree sub-cooled on flow boiling. The heat sink was made of oxygen free copper with hydraulic diameter of 0.3 mm. The operation conditions of experimental work for mass fluxes and heat fluxes from 78 to 800 kW/m², 1700 and 2100 (kg/m²sec) respectively for inlet degree of sub-cooled 21 K and 31 K. The results showed that there was an increase in mass flux which lead to increasing the heat transfer coefficient. Also, the convective flow boiling was the dominated mechanism. In addition, heat transfer coefficient increased g with increasing the degree of sub-cooled [11]. However, from the literature review of flow boiling it could be observe that the dominant heat transfer mechanism and the influence of heat fluxes and mass fluxes on the heat transfer flow boiling in microchannel are not obvious. There is a discrepancy on the dominant heat transfer mechanism. A group of researchers state that the nucleate boiling is a dominant heat transfer mechanism. In contrast, some researchers take the position that the convective boiling mechanism is the dominant heat transfer mechanisms in microchannels. To understand the characteristics and mechanism(s) of flow-boiling heat transfer in microchannels, the above review shows that further research is still needed. Therefore, one of the key aims of the present research is to study the effect of mass fluxes (300-600 kg/m².sec), and heat fluxes (5.4- 376.5 kW/m²) on the flow boiling heat transfer in single microchannel with hydraulic diameter of $D_h=420$ μm at degree of sub-cooled 20K of de-ionized water as working fluid. At the same time, validated correlations are necessary for the widespread use of multichannel heat sinks in cooling high heat flux devices. The results of the current experimental analysis have also been used to test proposed correlations that predict heat transfer rates.

2. Experiment apparatus and data reduction

2.1. Experiment setup
The experimental system is made up of de-ionized water reservoir, a micro gear-pump, a pre-heater, the test section, a sub-cooler, and in-line filter. The rotameter model (LZB-3WBF) is used for the experiment works, which calibrated with an accuracy of ±0.64 ml/min. The sub-cooler and the condenser use a chiller unit for cooling purposes. Figure 1 and 2 are represented the schematic diagram and experimental facility set-up respectively. For the degassing of deionized water in the liquid tank, intense boiling for around one hour was used. The top valve of the condenser was opened to relieve gases to the ambient, which are not condensable gasses. 5 μm filter was mounted prior to the gear pump at the site to remove all particles from the water. After that, through pumping, the degassed water was introduced into the test section and a pre-heater was used to regulate the inlet temperature of the working fluid. The test section consists of brass-block which has a single microchannel at the top surface, cartridge heaters, polycarbonate housing and top cover plate. Single rectangular microchannel was cut off into the upper-surface of brass-block by a milling-machine at a feeding rate of 10 (millimeters/minute). The dimensions of brass-block were as follows: 12mm width, 60mm length, and 60mm height. The nominal dimensions of the microchannel are 0.3 mm width, 0.7 mm depth. To supply a power of heat that used for heating the brass-block, two cartridge heaters each one with 100 Watt heating powers were inserted horizontally at its bottom, as seen in figure 3. Four thermocouples type-K were implanted vertically along the centerline of the brass-block with equidistant 14.7 mm to calculate the heat flux. To calculate the coefficient of local heat transfer along the length of the channel at axial direction, seven thermocouples type-K were implanted at 1 mm underneath the channel and distributed along the axial direction of the channel with a distance equal to 9.67 mm. All thermocouples diameter implanted at the plane center of the brass block are 0.5 mm. The accuracy of thermocouples calibration were about ± 0.5K. The brass block was place into the polycarbonate housing, and it was sealed using O-rings as seen in figure 3. The housing of polycarbonate comprises of the inlet manifold and outlet manifold, inlet -outlet sub channels. Thermocouples type K with diameter of 1 mm was introduced at inlet-out manifold to measure inlet-outlet temperature of fluid. It's calibrated with an accuracy of ±0.5 K.
The pressure of fluid at inlet-outlet manifolds was measured using absolute pressure transducers model (MPX5500DP). It's calibrated with an accuracy of ±4.27kPa %. Pressure drop across the test section was measured directly using differential pressure transducer (MPX4250DP). It was calibrated with an accuracy of ±2.34kPa. The dimensions of the inlet manifolds and outlet manifold are (20 mm wide, 10 mm height and 10 mm depth), while the dimensions of both inlet-outlet sub channels are (2mm wide, 0.7 mm depth and 7 mm length). To pass thermocouples wires from brass block through...
polycarbonate housing, a number of holes were drilled with diameter of (0.9 mm) in it. Layer of Polycarbonate having thickness 6 mm was sandwiched between the top polycarbonate cover plate and top-surface of the housing. To prevent any leakage from upper-surface of the housing, a layer of polycarbonate was locked by using the O-ring established in the housing at its top surface. As shown in Figure 4, a visualized window has been created at the top cover plate with the same size as the microchannel block, including the sub-channels. The data of temperature was recorded using data logger (Applent AT 4532x) reading. Bus-power multifunction DAQ USB (model NI USB-6009) was used as interface device to compile the pressure transduce signals (0-5 DC–Volt) from the pressure transduce gauges to the LabVIEW software. The data was read and recorded through NI USB-2009 and LabView software at time interval of 1 msec. The tests were carried out by keeping the flow rate unchanged and progressively increasing the supply power. The data was collected through 3 minutes and then summed to be used in the process of data reduction.

Figure 3. Test section design and construction.

Figure 4. Test section
2.2. Data reduction

2.2.1. Single phase data reduction. The net pressure drops for the single flow along the microchannel $\Delta P_{ch}$ is expressed by:

$$\Delta P_{ch} = \Delta P_m - \Delta P_{loss} \tag{1}$$

Where $\Delta P_m$ is the total measured pressure drop and $\Delta P_{loss}$ is the total pressure losses due to pressure drop in the inlet sub-channel $\Delta P_{sch,i}$, the outlet sub-channel $\Delta P_{sch,o}$, the sudden contraction $\Delta P_{sc}$ and sudden expansion $\Delta P_{se}$ which expressed below:

$$\Delta P_{loss} = \Delta P_{sch,i} + \Delta P_{sch,o} + \Delta P_{sc} + \Delta P_{se} \tag{2}$$

Due to the inlet sub-channel and the outlet sub-channel, the contribution of losses at the maximum Reynolds number ($Re = 2322$) are 0.42 percent and 0.42 percent, respectively. For a lower Reynolds number, these values are considered lower. So the pressure losses due to sudden contraction and expansion were calculated as given by the equation below:

$$\Delta P_{loss} = (K_{sc1} + K_{se2}) \frac{v^2_{sch,p}}{2} + (K_{sc2} + K_{se1}) \frac{v^2_{ch,p}}{2} \tag{3}$$

In the above equation ($K_{sc1}$ and $K_{sc2}$) is the sudden contraction loss coefficients from the manifold to the sub-channel and from the sub-channel to the microchannel respectively, while ($K_{se1}$, and $K_{se2}$) is the sudden expansion loss coefficients from the microchannel to the sub-channel and from the subchannel to the manifold. The values of loss coefficients ($K_{sc1}, K_{sc2}, K_{se1}$ and $K_{se2}$) are determined as (0.5, 0.47, 0.72 and 0.81) respectively [12]. The heat flux added to the test section $q_b$ was calculated as:

$$q_b = \frac{P - Q_{loss}}{A_p} \tag{4}$$

$$P = IV \tag{5}$$

Where; $P$ is the heating power in watt, $I$ and $V$ are the electrical current and electrical voltage in amperes and volts unit, respectively, and $A_p$ is the base area of the heat sink calculated with width base $W$ and heated length ($L_{ch}$) of base block is given as $A_p = W \cdot L_{ch}$. Here, $Q_{loss}$ is the heat loss from the test section to the ambient, estimated by heating the test section without liquid pumping. When the system became steady, the power was documented. This procedure was repeated. Finally, the power loss correlated with temperature difference between the test section and the ambient for different values of power and system temperature. This method was adopted by many researchers, [13] and [11]. The local heat transfer coefficient ($h_{sp}(z)$) is determined as following:

$$h_{sp}(z) = \frac{q_b W}{(T_w(z) - T_{fcc}(z))(W_{ch} + 2H_{ch})} \tag{6}$$

Where; $W$ is the width of brass block, Also $W_{ch}$ and $H_{ch}$ are respectively, the width and height of the channel. The local thermocouples were inserted at distance ($\eta$) of 1 mm from the bottom of the microchannel. Thus, the temperature values of thermocouples ($T_{\eta}(z)$) were amended using the one-dimensional heat conduction formula to evaluated the value of inner surface of microchannel temperature ($T_{wa}(z)$), as expressed by equation below:

$$T_w(z) = T_{wa}(z) - \frac{q_{\eta} \bar{h}}{K_s} \tag{7}$$

By assuming the uniform boundary condition the locally fluid temperature ($T_f(z)$) can be calculated by equation below:

$$T_f(z) = T_i + \frac{q_b W z}{m C_p} \tag{8}$$

Where; ($T_i, C_p$, and $\dot{m}$) are inlet fluid temperature, specific heat of the fluid, mass flow rate and $z$ is the axial location of the channel length. The average Nusselt number is given by:
\[
\frac{\Delta P_{tp}}{\Delta P_{sp}} = \frac{\Delta P_{ch} - \Delta P_{sp}}{\Delta P_{sp}}
\]

2.2.2. Two-phase flow data reduction. Sub-cooled deionized water is supplied into the microchannel. So, water maintains a single phase liquid state until thermodynamic equilibrium quality approaches to zero. Microchannel length can be separated into two areas: an upstream inlet sub-cooled region (single-phase region) and downstream saturated region (two-phase region); the zero thermodynamic quality location represented the dividing point between the two regions. Therefore, microchannel pressure drop is divided into single-phase pressure drop region, \(\Delta P_{sp}\), and a two-phase pressure drop region, \(\Delta P_{tp}\). Thus, the net two-phase pressure drop along the microchannel is evaluated as:

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

The single-phase pressure drop along the single-phase part can be determined from:

\[
\Delta P_{sp} = \frac{2f_{app}G^2}{\rho D_{h}} L_{sp}
\]

\[
L_{sp} = \frac{mC_p(T_{sat} - T_i)}{q_i W}
\]

\(T_{sat}\) can be calculated by using the inlet pressure, assuming that pressure drops across the sub-cooled region is small; this is practical for a slight pressure drop in the channel. For developing flow, apparent friction factor \((f_{app})\) can be calculated using the following equation by Shah and London [14]:

\[
f_{app} = \frac{3.44}{Re(L)^{1/2}} + \frac{(f_{FD}Re) + K(\infty)}{4Re(L)^{1/2}}
\]

Where; \(L^*\) is the dimensionless axial distance \((L_{sp}/ReD_h)\), while \((K(\infty))\) and \((C)\) are constant, it depends on the aspect ratio. For the geometry used in this work, the values are 1.7784x10^{-4}, and 1.1962. Also, Poiseuille number \((f_{FD} Re)\) for fully developing single phase fluid flow was formulated by Shah and London [14]:

\[
f_{FD} Re = 24(1 - 1.353534\beta + 1.9467\beta^2 - 1.7012\beta^3 + 0.9653\beta^4 - 0.2537\beta^5)
\]

In the above equations \(\beta\) is the channel aspect ratio. At the two phase region, the heat transfer coefficient can be calculating by the equation below which base on the heat balance:

\[
h_{tp}(z) = \frac{q_i W}{(T_w(z) - T_{sat}(z))(W_{ch}+2H_{ch})}
\]

Where; \(T_{sat}(z)\) defines as the locally fluid saturation temperature, and it can be evaluated base on the local pressure which flows in the channel. The thermodynamic vapor quality at any axial position \(x(z)\) was calculated as given by equation below:

\[
x(z) = \frac{(i(z)-i_l(z))}{i_{lg}(z)}
\]

Where \(i_l(z)\), and \(i_{lg}(z)\) are the liquid specific enthalpy and the latent heat of vaporization respectively which it computed at the local pressure in the microchannel. The equation above covers the concept of vapor quality to sub-cooled liquid flow; in which case, it will assume negative value. The enthalpy of the fluid, \(i(z)\), at \(z\) location along the microchannel is specified by energy balance as shown below:

\[
i(z) = i_i + \frac{q_i W}{m}
\]

with \(i_i\), the specific enthalpy at the entrance to the channel. To asss fluid property in equation (16) and to estimate the local temperature of the two phase flow, an expression for the pressure gradient
must be assumed. In this study, the local pressure through the channel is calculated by assuming the linearly pressure drop with axial length of the micro-channel as reported by [15] and [11].

\[
P_{sp(z)} = P_t - \left( \frac{z}{L_{sp}} \right) \Delta P_{sp}
\]

\[
P_{tp}(z) = P_{sat}(L_{sp}) - \left( \frac{z-L_{sp}}{L_{ch}-L_{sp}} \right) \Delta P_{tp}
\]

Experimental data were attained at 1 atm. Pressure of the system along a range of parameters as following: mass fluxes range is \( G = 300-600 \) kg/m\(^2\)sec and heat fluxes range is \( q^* = 5.4-376.5 \) kW/m\(^2\). The propagated analysis of uncertainty was performed using the approach explained in Coleman and Steel [16] and the values were 10.2–15.7%, 11.4–15.9% and 2.2–50% respectively for the values of heat fluxes, heat transfer coefficient and fanning friction factor. The higher value of the uncertainty of the friction factor up to (50%), as a consequence of the errors in the flow and pressure drop measuring instruments through the lowest values of Reynolds number.

3. Results and discussion

3.1. Single-phase experimental validation

Single-phase experiments were performed to verify the experimental system prior to the boiling experiments. Figure (5-a) shows the comparison of experimental result for friction factor with the correlation of Shah and London [14] for developing flow Eq. (13) and fully developed flow (\( f_{FD} = C/Re \)) as viewed in equation 14. The constant C is 16.3935 for the geometry used in the present study. The figure proves that there is a good agreement with the researchers' correlations, and the deviation is within the experimental uncertainty. Also, Figure (5-b) depicts the comparison of the experimental result for the Nusselt number (\( Nu_{exp} = h.Dh/k_f \)) with the predictions from the correlation of Shah and London [14] defined below by Equations 23 and 24, and it is determined experimental results of Mirmanto [17] and Mehmed [9]. It is clear that the experimental values prove a comparable trend, i.e. the values of Nusselt number growths with Reynolds number, but the values are lower than the prediction from the correlation and previous result. Fanning friction factor for fully developed turbulent flow established by Blasius as:

\[
f = 0.079 Re^{-0.25}
\]

Also, the fanning friction factor for developing turbulent flow was formulated by Phillis [18]:

\[
f_{app tur b.} = \left( 0.0929 + \frac{1.0161D_h}{L} \right) Re^* \left( -0.268 - \frac{0.3195D_h}{L} \right)
\]

Where \( Re^* \) is equivalent Reynolds number which can be evaluated by equation below:

\[
Re^* = Re \left( 2/3 + \frac{11}{24} \beta (2 - \beta) \right)
\]

Nu\_number formulated by Shah and London 1978 for a fully developed and developing laminar flow is illustrated in equation 23 and 24 respectively:

\[
Nu = 8.235(1 - 10.6044 \beta + 61.1755 \beta^2 - 155.1803 \beta^3 + 176.9203 \beta^4 - 72.9236 \beta^5)
\]

\[
Nu = 0.775 L^*_{t}^{-1/3} (f Re)^{1/3}_{FD}
\]

Where \( L^* \) is a dimensionless channel length can be represented by the equation:

\[
L^*_t = \frac{L}{Re^* Pr^* D_h}
\]

Also, Nusselt number for developing laminar flow correlated by Mirmanto [17] is:

\[
Nu = Re^{0.283} Pr^{-0.513} L^*_t^{-0.309}
\]
3.2. Two-Phase results and discussion

3.2.1. Boiling curve. The effect of mass flux on the boiling curve plotted corresponding to degree of sub-cooled 20 K and system pressure of 1 atm. can be shown in Figure 6. Data indicates that this curve was taken at steady state conditions and it shows the plot of the heat flux based on wall channel versus temperature difference between the wall temperature measured by the temperature sensor positioned near the microchannel outlet and saturation temperature. As shown, at low heat fluxes and different values of mass flux, the slopes of all boiling curves are constant indicating the single-phase of heat transfer state. As the value of heat flux increases, the system exhibits a sudden change in slope from its single-phase indicating the onset of nucleate boiling had commenced at that location. Also, figure shows that the onset of nucleate boiling is delay as an increasing of mass flux.

![Figure 6](https://example.com/figure6)
3.2.2. Results of Heat Transfer. The presence of local temperature sensors allows the local heat transfer coefficients to be computed. Figures 7(a) and figure 7(b) show the variation of the local heat transfer coefficient with heat flux and thermodynamic equilibrium respectively for the range of mass fluxes 300-600 kg/m²/sec at a degree of sub-cooled 20 K. These figures view the thermocouple location of \((Z/L) = 0.98\), which is related to the highest amount of saturated boiling and the larger vapor thermodynamic quality for at a certain input heat flux. However, water at the inlet of microchannel is at a sub-cooled condition then progressivity was heated up on the length of the test section. So, the flow boiling is separated in two areas [2]: Sub-cooled and saturated region. In sub-cooled quality \((x)\) less than 0, the heat flux range is within low and moderate values, while in saturated region \((x>0)\), the heat flux input value is as high as seen in the figures. The local heat transfer coefficient reveals a similar behavior with respect to the vapor quality and heat flux for any given mass flux. For low to moderate heat flux at sub-cooled region, \((x \text{ less than } 0)\) the values of local coefficient of heat transfer are small, then they are followed by a sudden increase in the local heat transfer coefficient until the vapor quality reaches zero \((x = 0)\) at wall heat fluxes of 64.2, 77.9, 93.8 and 116.7 kW/m² for mass fluxes of 300, 400, 500 and 600 kg/m².secs respectively. For the vapor quality which is greater than zero \((x > 0)\), saturated region, the local coefficient of heat transfer begins decreasing with an increase in the heat flux value and vapor thermodynamic quality. At higher heat fluxes, the local saturated heat transfer coefficient becomes largely insensitive to heat flux for the range tested. This trend does not support the nucleate boiling mechanism (heat transfer coefficient increases with the increase of heat flux). Furthermore, a trend of local heat transfer coefficient was reported by previous studies of flow boiling in microchannels which documented that the convective boiling was the prevailing heat transfer process. For example, Qu and Mudawer stated that the coefficient of heat transfer increased with increasing the value of mass flux, but it decreased when the vapor thermodynamic quality increased. It was stated that the convective boiling mechanism is the dominant heat transfer mechanism [2]. On other hand, Lee and Garimella found that the value of coefficient of heat transfer improved with heat flux at low and moderate heat flux inputs, while it reduced with heat flux at higher heat flux input value. The authors reported that at low and moderate heat flux input, nucleate boiling dominated the boiling process, while convective boiling became the dominant heat transfer mechanism at greater heat flux input [19]. However, they didn’t present any mass flux effect on the coefficient of heat transfer as indication. These facts lead to the conclusion that the nucleate and forced convection boiling is the dominant mechanism for present work. At the low mass flux, nucleate boiling starting earlier, in addition the higher heat transfer coefficient occurs at the higher mass fluxes, higher mass fluxes required a larger amount of heat for boiling incipience. The influence of the heat fluxes on the average heat transfer coefficient through the microchannel with range of mass fluxes can be shown in (figure 8). Generally, the higher values of average heat transfer coefficient occurs at a lower mass fluxes in the two-phase flow region.

**Figure 7.** Sub-cooled flow boiling heat transfer coefficient versus (a) heat fluxes and (b) exit vapor quality at 20 K inlet degree of sub-cooled with range of mass fluxes at the exit of microchannel.
Figure 8. The effect of heat fluxes on the Average heat transfer coefficient at 20 K inlet degree sub-cooled at range of mass flux.

4. A Comparison to the exiting prediction methods
The heat transfer coefficient resulting from tests realized for this study is compared to the heat transfer coefficient predicted by correlations which was taken from the open literature listed in the appendix. The experimental conditions which were conducted in these correlations were mass fluxes with (300 up to 600) kg/m$^2$.sec, and heat fluxes range from 5.4 to 376.5 kW/m$^2$ at a degree of inlet sub-cooling 20 K. The comparison between the experimental data and the selected correlation are based on the mean absolute error (MAE) defined by equation for heat transfer coefficient as given by:

$$MAE = \frac{1}{n} \sum \left| \frac{h_{exp} - h_{pred}}{h_{exp}} \right| * 100\%$$

(27)

However, present data are compared to heat transfer coefficient suggested by Kandlikar and Balasubramanian, (2004) [20], Sun and Mishima, (2009) [21], and Li and Wu, (2010) [22]. The comparison of the Kandlikar and Balasubramanian correlations with experimental results is depicted in (figure 9). In The first figure (9-a) established that the experimental data over predicted with a MAE value of 45.5%, at higher heat flux. This correlation was used to predict the heat transfer coefficient where the flow was dominated by nucleate boiling (i.e. the heat transfer coefficient increases with increases heat flux). The other correlations for Kandlikar and Balasubramanian were used when the flow was dominated by convective boiling which provides proper prediction within a MAE value of 21.1%, as shown in figure (9-b).

The correlation of Sun and Mishima [21] was intended for a small to microchannel with a diameter ranging from 0.21 to 6.05 mm for different liquid including water and based on 2505 flow boiling data sets. In this correlation, the convective boiling component was considered by introducing the liquid Webber number. As shown in figure 10, the correlation has validated by the data obtained in this study with a MAE value of 16.7%. Finally, the data of the present study is compared to the correlation of Li and Wu [22]. This correlation is based on the 3744 data point with twelve different liquids including de–ionized water at ranges of hydraulic diameters (0.148-3.25 mm). The correlation considered the effect of surface tension by using the Bond number. As demonstrated in figure 11 that the correlation deviates little from experimental data with MAE value of 12.8 %. It can be concluded from these correlations that the mechanism of the heat transfer in the current study is convective flow boiling.
Figure 9. Comparison of experimental heat transfer coefficient data with (a) nucleate and (b) convection mechanisms correlation of Kandlikar and Balasubramanian (2004), at 20 K inlet temperature sub-cooled.

Figure 10. Comparison of experimental coefficient of heat transfer data to the correlation of Sun and Mishima (2009), at 20 K inlet temperature sub-cooled.

Figure 11: Comparison of experimental coefficient of heat transfer data to the correlation of Li and Wu (2010), at 20 K inlet temperature sub-cooled.
5. Conclusion
The experiments of sub-cooled flow boiling in a brass single microchannel heat sink utilizing de-ionized water were achieved at 20 K degree sub-cooled with range of mass fluxes 300, 400, 500, and 600 kg/m² sec and wall heat fluxes 5.6-376.5 kW/m². Markedly, at low heat fluxes and different values of mass flux, the slopes of all boiling curves are constant, indicated the single-phase of heat transfer. As the heat flux raised up, exhibits a sudden change in slope from its single phase to two phase zone. At the same time, the onset of nucleate boiling delays as the mass flux increases. In addition, the heat transfer coefficient increases with increasing mass fluxes. The dominate mechanism of heat transfer is the nucleate and forced convection boiling for present work. The minichannel and microscale correlations could predict the experimental data reasonably well.

6. References:
[1] Yarin L P, Mosyak A and Hetsroni G 2009 Fluid Flow, Heat Transfer and Boiling in Micro-Channels (New York, NY: Springer)
[2] Qu W and Mudawar I 2003 Flow Boiling Heat Transfer in Two-Phase Micro-Channel Heat Sinks-I. Experimental Investigation and Assessment of Correlation Methods (Int. J Heat Mass Transf) vol 46 no 15 pp2755–2771
[3] Liu D and Garimella S V 2007 Flow Boiling Heat Transfer in Microchannels (J Heat Transfer) vol 129 no 10 pp 1321–1332
[4] Lee J and Mudawar I 2008 Fluid Flow and Heat Transfer Characteristics of Low Temperature Two-Phase Micro-Channel Heat Sinks-Part 2, Subcooled Boiling Pressure Drop and Heat Transfer (Int J Heat Mass Transf) vol 51 no 17–18 pp 4327–4341
[5] Lee J and Mudawar I 2008 Flow and Heat Transfer Characteristics of Low Temperature Two-Phase Micro-Channel Heat Sinks – Part 1: Experimental Methods And Flow Visualization Results (Int J. Heat Mass Transf) vol 51 no 17–18 pp 4315–4326
[6] Harirchian T and Garimella S V 2009 Effects of Channel Dimension, Heat Flux, and Mass Flux on Flow Boiling Regimes in Microchannels (Int j multiph flow) vol 35 no 4 pp 349–362
[7] Suzuki K, Nomura T, Hong C and Yuki K 2009 Subcooled Flow Boiling with Microbubble Emission in a Microchannel (ASME Second International Conference on Micro/Nanoscale Heat and Mass Transfer) vol 2
[8] Galvis E and Culham R 2012 Measurements and Flow Pattern Visualizations of Two-Phase Flow Boiling in Single Channel Microevaporators (Int j multiph flow) vol 42 pp 52–61.
[9] Ozdemir M R 2016 Flow Boiling Heat Transfer in a Rectangular Copper Microchannel (J Therm Eng) vol 2 no 3
[10] Yin L, Xu R, Jiang P, Cai H and Jia L 2017 Subcooled Flow Boiling of Water in a Large Aspect Ratio Microchannel (Int J Heat Mass Transf) vol 112 pp 1081–1089.
[11] Mohammed S A and Fayyadh E M 2020 Experimental Investigation of Heat Transfer and Flow Characteristics in Different Inlet Subcooled Flow Boiling in Microchannel (IOP Conf Ser Mater Sci Eng) vol 671
[12] Shaughnessy E J, Katz I M and Schaffer J P 2005 Introduction to Fluid Mechanics (New York, NY: Oxford University Press)
[13] Koşar A, Kuo C J and Peles Y 2005 Boiling Heat Transfer in Rectangular Microchannels with Reentrant Cavities (Int J Heat Mass Transf) vol 48 no 23–24 pp 4867–4886
[14] Shah R K and London A L 1978 Laminar Flow Forced Convection in Ducts (Advances in Heat Transfer. Academic Press, NY)
[15] Mirmanto M 2014 Heat Transfer Coefficient Calculated Using a Linear Pressure Gradient Assumption and Measurement for Flow Boiling in Microchannels (Int J Heat Mass Transf) vol 79 pp 269–278
[16] Coleman H W and Steele W G 2018 Experimentation, Validation, and Uncertainty Analysis for Engineers (Nashville, TN: John Wiley & Sons)
[17] Mirmanto M 2013 Single-Phase Flow and Flow Boiling of Water in Horizontal Rectangular Microchannels (Ph.D. Thesis, School of Engineering and Design, Brunel University London)
[18] Phillips R J 1987 *Forced Convection, Liqued Cooled, Microchannel Heat Sink* (Massachusetts Institute of Technology, Dept. of Mechanical Engineering, Massachusetts Institute of Technology)

[19] Lee P S and Garimella S V 2008 *Saturated Flow Boiling Heat Transfer and Pressure Drop in Silicon Microchannel Arrays* (Int J Heat Mass Transf) vol 51 no 3-4 pp 789–806

[20] Kandlikar S G and Balasubramanian P 2004 *An Extension of the Flow Boiling Correlation to Transition, Laminar, and Deep Laminar Flows in Minichannels and Microchannels* (Heat Trans Eng) vol 25 no 3 pp 86–93

[21] Sun L and Mishima K 2009 *Evaluation Analysis of Prediction Methods for Two-Phase Flow Pressure Drop in Mini-Channels* (Int j multiph flow) vol 35 no 1 pp 47–54

[22] Li W and Wu Z 2010 *A General Correlation for Adiabatic Two-Phase Pressure Drop in Micro/Micro-Channels* (Int J Heat Mass Transf) vol 53 no 13–14 pp 2732–2739

Appendix A

### Table A1. Flow boiling heat transfer correlations.

| Reference | Flow boiling heat transfer correlations | Applicability range |
|-----------|----------------------------------------|---------------------|
| Kandlikar and Balasubramanian (2004) | For Re<1600 h_{t_{p,NBD}} = 0.6683C_{0}^{-0.2}(1-x)^{0.8}h_{L} + 1058Bo^{-0.7}(1-x)^{0.8}F_{l}h_{L} h_{t_{p,CBD}} = 1.136e^{-0.9}(1-x)^{0.8}h_{L} + 667.2Bo^{-0.7}(1-x)^{0.8}F_{l}h_{L} h_{L} = \frac{h_{u.k}}{D_{h}} | Mass flux = from 13 to 8179 kg/m²s, Diameter = 4-32 mm. Water, R-11, R-12, R-22, R-13, R-134a, R-152a. Where: h_{t_{p,NBD}}: Two phase heat transfer coefficient for nucleate boiling dominate. h_{t_{p,CBD}}: Two phase heat transfer coefficient for convective boiling dominate. F_{l}: fluid surface parameter =1 for water |
| Sun and Mishima (2009) | h_{t_{p}} = \frac{6Re_{L}^{0.05}Bo^{0.54}K_{L}}{We_{L}^{0.191}(\rho_{L}/\rho_{g})^{0.142}D_{h}} | D_{h} = from 0.21- to 6.05 mm |
| Li and Wu (2010) | h_{t_{p}} = 334Bo^{0.3}(BdRe_{L}^{0.36})^{0.4}K_{L} \frac{K_{L}}{D_{h}} Bd = \frac{\alpha d_{h} \delta}{\sigma} | Founded on 3744 date. twelve different fluids water is including, D_{h} = from 0.148 to 3.25 (mm) |