Boiling Heat Transfer in a Micro-Channel Complex Geometry

R. SHAKIR
Department of Petroleum and Gas Engineering, College of Engineering, University of Thi-Qar, Iraq
shraed904@gmail.com

ABSTRACT

"This prediction demonstrates an inclusive investigation on" single-phase and two-phase pressure drop properties and flows boiling" instabilities in micro-channels to resume the concepts in the boiling heat transfer tests". The predictions of "single-phase and two-phase heat transfers" in the aluminum micro-channel heat sink have been investigated". "Different heat fluxes and different mass fluxes have been applied "(each mass flow rate under numerous heats fluxes) in an aluminum parallel channel test piece is tested". "It was built up from a piece of aluminum," twenty-five mm wide by twenty-five mm long and ten mm high"."R113 working fluid temperature was constant (25ºC) for all prediction tests; "Therefore the heat applies have been applied "in the ranges of (40-600 Watts) "The processes and iteration loops have reached for the prediction heat transfer properties "for three mass flow rates are (0.0125;0.015 and 0.0175 kg/s) "respectively;" and another heat transfer properties ."The aluminum micro-channels heat sink" with 0.5 mm channel height and 0.5 mm channel width "is heated via a wire electrical heater device". "The single-phase and two-phase flows are important parameters" for geometrical configuration of the aluminum micro-channels under variations heat applied". The purpose of this investigation is to explore the relationship between "prediction heat transfer coefficients and mini-scale heat sink geometrical configuration". Heat-transfer coefficients and pressure drops are investigated"." For sub-cooled and saturated boiling data acquired with Single-phase and boiling flows," The single-phase consequences are seen to be location-following, regular with a fully developed laminar flow. All the prediction boiling heat transfer coefficients are seen to have reasonably relied on as mass fraction and mass flux"." Nevertheless, some are seen to have relied on heat flux whereas others are not. The convective boiling consisting is seen to have a boiling heat-transfer coefficient that is moderately relied on mass flow rate, gas mass fraction, and heat applied"." The prediction work presented here provides one of the first investigations into how to get single-phase" and two-phase properties" data via computer programming .In the end, "the single-phase and two-phase" heat transfer coefficients are reported" for square aluminum micro-channels heat sink under" boiling tests".

**Keywords**: Prediction Heat Transfer Coefficient, Exit Quality, Single-Phase Flow, Two-Phase Flow, Prediction Fluid Temperature
INTRODUCTION

A lot of investigators have reported single-phase and two-phase flows used experiments and predictions for different working fluid such as water, R113…ets. Recently, researchers have shown an increased interest in micro-channel flows began with the investigation work on direct chip cooling with water via Tuckerman and Pease [1]. Over the past years, there has been a dramatic increase in a number of researchers including Li et al. [2], Celata et al. [3], and Steinke and Kandlikar[4]. Critically guessed the obtainable literature and seen explanations for the high deviations from traditional theory reported by some of the researcher. Zunaid, M.[5], studied heat transfer and pressure drop advantages of a straight rectangular and semi-cylindrical projections micro-channel heat sink. The test piece constructed of copper plate, The Water working fluid was utilized as cooling was built to flow through the micro-channels heat sink. Constant heat flux applied to the platform area of the heat sink . Reynolds number ranging between two hundred to one thousand and heat transfer increases with the employ of semi-cylindrical projections micro-channel heat sink those results shown. Özdemir, M.[6], reported an inclusive review on single-phase pressure drop instability in micro-channels to summary the contradictions in the literature. A wide range of literature studies including the past and present researchers was over surveyed crucially. The empirical conditions, channel geometrical parameters, heat flux, mass flux. As well as traditional and micro-scale pressure drop correlations are debated. This study has given that the continuum rule is usable for single-phase pressure drop applications with some significance. Several investigators have investigated the flow boiling phenomena Cornwell and Kew.[7]; Kandlikar et al.[8] and [9]; Kuznetsov and Vitovsky,[10]. Cuta et al.;[11]. This investigation systematically estimations single and two-phase pressure drop investigations in the micro-channels heat sink, sighting to supply collective understanding about the subject. The influence of various predictive conditions and channel geometries on single and two-phase pressure drop properties was shown from this study. Traditional and micro-channels heat sink pressure drop correlations were also studied. In the end, flow boiling tests instabilities in micro-channels heat sink were predicted critically.

MATERIALS & METHODS

The mathematical models of the estimate information are adjusted within the following steps. The single-phase and two-phase heat transfer coefficients for flow is defined depended on the liquid temperature and wall temperature conforming to the system pressure as a reference temperature. As indicated in Table 1, R113 fluid was supplied into the heat sink for all test conditions.

| Liquid temperature (°C) | Prandtl Number (-) | Dynamic viscosity (Pas) | Thermal conductivity (W/mK) | Liquid Density (kg/m³) |
|-------------------------|-------------------|-------------------------|-----------------------------|-----------------------|
| 10.00                   | 10.037            | 0.000821                | 0.076606                    | 1597.95               |
| 15.00                   | 9.519             | 0.000764                | 0.075714                    | 1586.54               |
| 20.00                   | 9.044             | 0.000712                | 0.074827                    | 1574.8                |
| 25.00                   | 8.609             | 0.000665                | 0.073944                    | 1553.23               |
| 30.00                   | 8.208             | 0.000623                | 0.073067                    | 1551.34               |
| 35.00                   | 7.838             | 0.000584                | 0.072194                    | 1539.4                |
| 40.00                   | 7.495             | 0.000549                | 0.071327                    | 1527.41               |
| 45.00                   | 7.178             | 0.000517                | 0.070464                    | 1520.22               |
| 50.00                   | 6.883             | 0.000487                | 0.069608                    | 1512.25               |

Since only three sides of the micro-channels were heated in the present work, the approach used in Lee et al. [12],Is adopted here to predict the Nusselt number in single-phase flow based on results in the literature that apply to all four sides of the micro-channels being heated can be got from:
In which \((NUx)\), \((NUf d_2)\), and \((NUf d_4)\), were given to be obtained respectively, by Ref.[13]. As defined all single-phase heat transfer coefficients should be obtained from:

\[
h_s = \frac{Nu_kP}{D_n}
\]  
(2)

and all two-phase heat transfer coefficients should be gained from:

\[
h_f = \frac{q_B/A_B}{T_w-T_f}
\]  
(3)

The planning of the test section contains the plate sort heaters. The rear face of the test section is insulted; within the bottom of the test section there’s electrical heater and top of the test section there is R113 working fluid Figure 1. Shows the design can solely produce a uniform heat flux distribution at the solid-fluid interface. Therefore; Some preparation needed the heat conductivity through the wall to be introduced into the analysis. The dominant wall conduction influence was so imagined to be two-dimensional, with the one-dimensional perpendicular to the fluid flow and also the other parallel there to it, the heat conduction equation is

\[
\delta^2 T / \delta y^2 + \delta^2 T / \delta z^2 = 0
\]  
(4)

Where \((T)\) is the temperature in the aluminium wall and \((y)\) and \((z)\) are the ordinate perpendicular and parallel to the R113 working fluid respectively; then the Heat conduction Equation (4) was formed via sub dividing the area into square cells. Cells have \((0.5 \text{ mm})\) in width and \((0.5 \text{ mm})\) in high. For each cell, an energy balance can be obtained:

\[
T_{ij} = \delta y^2 (T_{i+1,j} + T_{i-1,j}) + \delta z^2 (T_{i,j+1} + T_{i,j-1}) / 2 (\delta y^2 + \delta z^2)
\]  
(5)

**Figure 1.** Aluminium test piece wall conduction

Where \((\delta z)\) and \((\delta y)\), the cell dimensions the equation (5) and the different of it wanted to perform the boundary conditions; iteratively were solved until the temperature in each cell was the same as the former guess to within an estimated error of \((0.001)\). The local wall temperature was consequently acquired via averaging the data from the relevant thermocouples, to get, which was then corrected for depth from the plate surface; through the one-dimensional heat conduction equation i.e.

\[
T_W = T_{th} - \frac{q_B L_{th}}{k_A}
\]  
(6)

In which \((k_A)\) Aluminium thermal conductivity (W/m.k) and the local heat transfer coefficients were obtained by separating the flow filed into cells Thus; A typical two-dimensional unit cells at the location of
a thermo-couple situated inside the test piece for the mini-channel test piece for a fluid. The fluid temperature was given by:

\[ T_F = T_i + \frac{q_B W Z}{M_F c_p} \]  

(7)

The exit quality is computed from energy balance and the power supplied to the channel is used in heating the fluid up to the saturation temperature and also latent heat during phase change. The exit quality is thus obtained by:

\[ x_e = \frac{q_B A - M_F c_p (T_{sat} - T_{in})}{M_F h_{fg}} \]  

(8)

The hydraulic diameter of the channel in the presence of heat transfer analysis given by:

\[ D_h = \frac{4 \cdot A_h}{\pi} \]  

(9)

The Reynolds number for single-phase and two-phase regions can be found respectively by:

\[ Re_F = \frac{(G(1 - x)D_h)}{\mu_F} \]  

(10)

\[ Re_v = \frac{(G x D_h)}{\mu_v} \]  

(11)

For the geometry and boundary conditions considered. Friction factor can be calculated by Ref. [13].

\[ f = 2 f_{ laminar} P \frac{\rho V^2}{D_h} \]  

(13)

The friction and acceleration pressure gradients in Equations (14) and (15) respectively are calculated as follows:

\[ \left( \frac{DP_F}{DZ} \right)_F = 2 f_{ laminar} P \frac{G^2(1-x)^2}{D_h \rho_F} \]  

(14)

\[ \left( \frac{DP_v}{DZ} \right)_v = 2 f_v G^2 x^2 \frac{V^2}{D_h \rho_v} \]  

(15)

\[ X^2 = \left( \frac{DP_F}{DZ} \right)_F \left( \frac{DP_v}{DZ} \right)_v \]  

(16)

The two-phase multiplier is calculated with:

\[ \phi_{F1p} = 1 + \frac{12(1 - e^{315D_h})}{X} + \frac{1}{X} \]  

(17)

The two-phase pressure drop per unit length is found by:

\[ \Delta P_{F1p} = \left( \frac{DP_F}{DZ} \right)_F \phi_{F1p}^2 \]  

(18)

The acceleration pressure drop is found from:

\[ \Delta P_a = G^2 V_x e \]  

(19)
The total pressure drop single-phase pressure drop, two-phase pressure drop and acceleration pressure drop can be found using:

\[ \Delta P_T = \Delta P_{sp} + \Delta P_{tp} + \Delta P_a \]  

(20)

**THE RESEARCH PREDICTION SET UP**

The Rig consists of three substantial parts, the flow loop design, aluminum test section and aluminum test piece.

**FLOW LOOP**

Suppose that a tendency to design a new technique for Figure 2. Schematically illustrated, the flow loop is given. Before running the boiling tests series, the R113 working fluid was degassed by strong boiling for nearly one to three hours to force any dissolved gases to run away from the system to the ambient. Throughout this interval, the vent valve on top of the pre-heater was sporadically opened to permit a dissolved gas goes to flight to the atmosphere. This furthermore set the test pressure to close at morphemically. After degassing R113 working fluid, because it was appeared, that no gas or air bubble coming out of the fluid inside the test piece before to tests boiling. Fluid R113 was pulled from the accumulator by the pump so Valves within the by-pass and main lines allowable the required mass rate of flow to be set. Fluid R113 passed from the pump to a filter. The coarse filter was used primarily to block large debris. The finer filter was used throughout testing. The tests were conducted by setting the needed R113 mass rate of flow and inlet temperature \( (M_F \text{ and } T_{in}) \). Mass flow was modifying by flow meter then the by-pass valve and adjusts by the dominant valve situated before goes to a parallel micro-channel heat sink. The preheater was connected to a controller unit. With respect to the test’s mass flow, the controller was adjusting to the needed applied heat to the R113 fluid flow that was passing the preheater, to produce the required inlet temperature to micro-channels heat sink into the aluminum house. At the same time; the test section heater was modifying to the required heat applied to the aluminum test piece. The R113 working fluid was distributed through the flow loop system till the required Steady-state conditions were got. Steady-state conditions were got this proceeding took two-three hours roughly so the Outlet heater temperature and also the aluminum housing temperature had to be compelled to be stable. These proceedings took (30-70) minutes roughly. All of the specified readings were completed before the heat applied was re-set to successive desired value and also the method iterated throughout the tests to preserve the system pressure close to the atmospheric pressure.
ALUMINUM TEST SECTION

Figures (3 and 4) have process view of the aluminum test section utilized in this study. The aluminum check section consists of three parts specifically the aluminum housing, the top cover, and therefore the aluminum micro channels heat sink. Aluminum housing includes of the top housing, the bottom housing, and therefore the base, all of that are manufactured from aluminum. The top housing holds the aluminum micro channels heat sink. It’s the inlet plenum and outlet plenum, the pressure port and temperature port to collect the respective sensors”. There are two pressure ports, one at the inlet plenum and other one at the outlet plenum.
Figure 3. Test housing

Figure 4. Test section

ALUMINUM TEST PIECE

Figure 5. Shows an aluminum test piece, from this figure has acquired all details. There are six ports for thermocouple, one every at the inlet plenum and outlet plenum and four below the channel surface of the heat sink. The thermocouples used to measure the inlet, outlet and wall temperatures. The thermocouples wall temperature had (1.6 mm) probe diameter and were force-fitted into two holes from vertical direction drilled (8 mm) below the channel surface and from the parallel direction they were drilled (10 mm) into the test piece at the inlet end and (10 mm) long at the outlet end. They were used to measure the heat sink’s stream temperature distribution. These holes allowable four sheathed k-type thermocouples and all thermocouples were calibrated in an exceedingly water bath and were accurate to (±0.5°C) approximately. A groove is cut out on the surface of the top housing to line an O-ring. Therefore, aluminum micro-channel heat sink has footprint (25 mm x 25 mm).

Figure 5. The micro-channel
RESULTS AND DISCUSSION

The overall predictions results are summarized in three figures consist of, First, prediction exit quality versus assumed heat flux, Second, prediction heat transfer coefficient versus R113 fluid temperature, Third, prediction heat transfer coefficient versus wall temperature and ,Forth, The prediction pressure drop which versus Wall temperature.

PREDICTION HEAT TRANSFER COEFFICIENT AND EXIT QUALITY

Figure 6. Is presents the analysis which plots prediction heat transfer coefficient versus prediction exit quality; relationship has been obtained that between. In Figure 6.The prediction heat transfer coefficients at the two flow rates are shown in Figure.6. Two regions of behavior are easily identified, corresponding to single-phase R113 fluid flow with a linear temperature increase with exit quality and to two-phase R113 flow with a much slighter increase in prediction heat transfer coefficient. As seen in Figure 6. for the single-phase R113 fluid flow zone; also, the prediction single-phase heat transfer coefficients are higher at greater flow rates but the variations are very small between the data results Finally, there are three significant regions single- phase and two-phase regions therefore from this relationships can be got a prediction the important factor for mini-scale heat sink geometry.

PREDICTION HEAT TRANSFER COEFFICIENT AND FLUID TEMPERATURE

Consider Figure 7. which plots between Prediction heat transfer coefficient against R113 fluid temperature, it can be clearly seen from Figure 7. The Prediction heat transfer coefficients at the two flow rates with heat applied series in range (40-600 watts) are shown in Figure 7. Two zones of behavior are easily identified, corresponding to single-phase fluid flow zone with a linear Prediction heat transfer coefficient increase with fluid temperature with a much smaller increase in Prediction single-phase heat transfer coefficient; and for two-phase flow zone with a power relationship are smaller linear increases in Prediction two-phase heat transfer coefficient. There is a continued increase in Prediction heat transfer coefficient with increasing heat flux as seen in Figure 7. Also, the Prediction heat transfer coefficients are higher at bigger flow rates for both phases.
The prediction heat transfer coefficient which against Fluid temperature

**PREDICTION HEAT TRANSFER COEFFICIENT AND WALL TEMPERATURE**

Heat transfer coefficients for both the single-phase fluid flow and two-phase flow at the two flow rates are presented in Figure 8. As expected, since the flow in single-phase flow zone is laminar at the entire two mass flow rate, heat transfer coefficients for single-phase flows remain relatively constant and approximately the same. For the two-phase flow, the heat transfer coefficients start to increase at a much lower wall temperature, whereas for the two-phase zone, two-phase heat transfer sustains a wide range of wall temperature and the power relationship between the results. It can be seen from the data that prediction heat transfer coefficients start to increase slowly at wall temperatures increasing. The boiling starting temperature for the R113 liquid is well limited (47 ºC), approximately.

**PREDICTION PRESSURE DROP AND WALL TEMPERATURE**

The predicted pressure drop between the inlet and outlet plenums demonstrates supports from the pressure drop cause of flow in the micro channels additionally the inlet and exit losses. In the pressure-drop consequences demonstrated in Figure 9, the inlet and exit loss ingredients are subtracted from the measured magnitudes according to the correlation of Liu and Garimella [14]. In Figure 9, the pressure drop measurements again indicate the two distinct regimes of single-phase flow and two-phase flow. For single-phase hydro-dynamically laminar fully developed flow in square channels, the friction factor can be found from Shah and London [13].

The computed consequences from this process for single-phase flow at the three flow rates are also inclusive in Figure 9. It is indicated that the flow is supposed to be uniformly distributed through all the micro-channels in the heat sink and the pressure drop across any one single channel appears the total pressure drop. The computed consequences are viewed to be in satisfying agreement with another prediction and measurement correlations.
CONCLUSION

This is a significant judgment in the realization of single-phase and two-phase heat transfer coefficients in three mass flow rates with a range in heat fluxes. Other consequences were widely in line with the prediction consequences. It is exciting to note that there are of R133 on the single-phase flow and two-phase flow in micro-channels heat sink no large variations between the review consequences and prediction consequences. The effects predictable investigations were conducted in aluminum micro-channels over a wide range of heat applies were (40-600 Watts) and mass flow rates were (0.0125; 0.015 and 0.0175 kg/s). The heat transfer characteristics and instabilities were studied. The following consequences:

- The R113 working fluid profile was single-phase flow and two-phase flow at all mass flow rates in the test section.
- All the R113 fluid properties were depended on R113 fluid temperature.
- Adiabatic conditions at the four sides’ aluminum test section because the aluminum test section walls are deeply insulated.
- The flows were laminar fully developed for (0.0125; 0.015 and 0.0175 kg/s) mass flow rates in single-phase flows zones.
- The maximum R113 fluid temperature during the test series is (40.66 ºC).
- The heat transfer coefficients can be got from, R113 working fluid was divided into the heat sink in a single-phase, and two-phase zones, then their fluid location temperatures less than saturation temperatures for all prediction conditions.

CONFLICTS OF INTEREST

No conflict of interest was declared by the authors.

NOMENCLATURE

- $C_p$: Specific heat, kJ / kg.K
- $D_h$: Hydraulic diameter, m
- $h_s$: Single-phase heat transfer coefficient, W/m².k
- $h_T$: Two-phase heat transfer coefficient, W/m².k
- $k_F$: Thermal conductivity, W/m.K
- $L$: Micro-channel length, mm
- $Nu$: Nussle number
- $q_b$: Heat flux, W/m²
- $Re$: Reynolds number
- $Pr$: prandltls number
\( T \)  
Temperature, °C

\( V \)  
Flow velocity, \( m/s \)

\( W \)  
Micro-channel width, mm

\( Z \)  
Flow direction, mm

Greek symbols

\( \beta \)  
Aspect ratio of the square channel

\( \alpha \)  
Equation parameter

\( \gamma \)  
Equation parameter

\( \rho_F \)  
Density of a fluid, \( \text{kg/m}^3 \)

Subscripts

\( F \)  
Refers to fluid

\( in \)  
Refers to inlet

\( h \)  
Refers to heat

\( w \)  
Refers to well

REFERENCES

[1] Tuckerman DB, Pease RFW. High-performance heat sinking for VLSI. IEEE Electron Device Lett 1981;2:126–9.

[2] Li Z-X. Experimental study on flow characteristics of liquid in circular microtubes. Microscale Thermophys Eng 2003;7:253–65.

[3] Celata GP, Cumo M, Guglielmi M, Zummo G. Experimental investigation of hydraulic and single-phase heat transfer in 0.130-mm capillary tube. Microscale Thermophys Eng 2002;6:85–97.

[4] Steinke ME, Kandlikar SG. Single-phase liquid friction factors in microchannels. ASME 3rd Int. Conf. Microchannels Minichannels, American Society of Mechanical Engineers Digital Collection; 2005, p. 291–302.

[5] Zunaid M, Jindal A (6), Gakhar D, Sinha A. Numerical study of pressure drop and heat transfer in a straight rectangular and semi cylindrical projections microchannel heat sink. J Therm Eng 2017;3:1453–65.

[6] Özdemir MR. A review of single-phase and two-phase pressure drop characteristics and flow boiling instabilities in microchannels. J Therm Eng 2018;4:2451–63.

[7] Cornwell K, Kew PA. Boiling in small parallel channels. Energy Effic. Process Technol., Springer; 1993, p. 624–38.

[8] Kandlikar SG. Spiesman, 1997,“Effect of Surface Characteristics on Flow Boiling Heat Transfer,” Eng. Found. Conf. Convect. Pool Boil. Irsee, Ger. May, n.d., p. 18–25.

[9] Kandlikar SG. A theoretical model to predict pool boiling CHF incorporating effects of contact angle and orientation. J Heat Transf 2001;123:1071–9.

[10] Kuznetsov V V, Vitosky O V. Flow pattern of two-phase flow in vertical annuli and rectangular channel with narrow gap. Two-Phase Flow Model Exp 1995 1995.

[11] Cuta JM, McDonald CE, Shekarziz A. Forced convection heat transfer in parallel channel array microchannel heat exchanger. American Society of Mechanical Engineers, New York, NY (United States); 1996.

[12] Lee P-S, Garimella S V, Liu D. Investigation of heat transfer in rectangular microchannels. Int J Heat Mass Transf 2005;48:1688–704.

[13] Shah RK, London AL. Laminar flow forced convection in ducts (Advances in heat transfer, supplement 1) 1978.

[14] Liu D, Garimella S V. Investigation of liquid flow in microchannels. J Thermophys Heat Transf 2004;18:65–72.