International J. of Math. Sci. & Engg. Appls. (IJMSEA)
ISSN 0973-9424, Vol. 10 No. I (April, 2016), pp. 257-265

FLOW BOILING HEAT-TRANSFER IN PLATE MICRO-CHANNEL HEAT SINK

R. S. H. AL-KHAFAJY
College of Engineering / Mechanical Engineering Department
Thi-Qar University, Iraq.

Abstract

Heat-transfer coefficients are investigated for an aluminum test section including plate micro-channel is reported. The plate micro-channel was fifty mm long and fifty mm width also had a glass top plate to permit visual observations. The data were produced whilst boiling R113 at atmospheric pressure. The heat flux range was 25–252.5 Watts and the mass flux range was 2.5 – 12.3 g/s. An electrical heating theory was heated from below the test section that is normally supported with the boundary condition of constant heat flux. Nevertheless, Heat-transfer coefficients are investigated for boiling stream, with saturated boiling and sub cooled boiling data acquired. All of the measured boiling heat transfer coefficients are presented to be rationally independent of vapor quality and mass flux. Nevertheless; some are presented to be independent of heat flux whilst others are not. This is true of the sub cooled boiling data and saturated boiling data, which are all substantially over the magnitudes associated with nucleate boiling. The boiling data thereafter have a nucleate and convective component. The convective boiling component is presented to have a heat transfer coefficient that is rationally independent of mass flux, liquid sub cooling, and vapor quality and heat flux.

1. Introduction

In the early years, electronic devices, such as micro-processors and lasers, have been rising in power and reducing in size. This has guided to an increasing intensity in heat generation that needs to be removed during normal operation. Removing heat is becoming increasingly hard. It thence appears likely that new methods will be needed in the not too distant future. One possibility is to use a boiling fluid as the coolant, this has guided to a plenty of research into boiling in small diameter channels and mini and micro-channels.

For boiling in complex geometries, many studies have been investigated at the macro-scale. For example, for tube bundles, Shrage et al [1] , Dalwati et al [2, 3] and Feenstra et al [4] have reported void fraction distributions, while Ishihara et al [5] , suggested a technique for the frictional two-phase multiplier. Fewer studies have been investigated at the micro-scale. A review of various surface enhancement techniques, including drilled cavities, re-entrant cavities, alumna sprayed particles and microstructures, has been investigated by Honda and Wei [6] for nucleate boiling heat transfer. The key issues identified were the temperature overshoot required to initiate boiling, the effect of liquid sub cooling and the critical heat flux. The overshoot temperature is seen to decrease with increased roughness of the microstructure. Nevertheless, the most effective method of reducing temperature overshoot was gasification of the liquid. This could permit boiling below the saturation temperature and had little effect on the performance at high, or close critical, heat flux. Honda and Wei [6] also investigated that the heat transfer and critical heat flux were found to improve with enhanced area, but not in direct proportion. Critical heat flux was found to increase with increasing sub cooling. The most effective enhancement method was found to be pin fins. This gave enhanced heat transfer because vapor trapped between the fins provided additional nucleation sites. Additionally, these spaces retained vapor for longer, giving enhanced heat transfer. The optimum spacing of the fins depended on liquid sub cooling.

2. Experimental setup

In the Figure 1 Schematically, the flow loop is presented. Before to running each single phase flow exam series, the working fluid was degassed by vigorous boiling for closely three hours to force any dissolved gases to run away from the system to the ambient. During this interval the vent valve above the condenser was periodically opened to permit dissolved gases goes to elopement to the atmosphere. This furthermore set the test pressure to close atmospheric. After degassing the liquid, as it was observed that no gas or air bubble coming out of the liquid inside the test piece before to boiling, flow boiling exams were executed. Exams were conducted by setting the wanted liquid mass flow rate and inlet temperature. Mass flow rate was adjusting by the by-pass valve and modify by the throttling valve located before the filters. The pre-heater was linked to a controller. With respect to the exam's mass flow rate, the controller was adjusting to the wanted applied heat to the fluid flow which was passing the pre-heater, to supply the required inlet temperature. Simultaneously the test section heater was adjusting to the required heat flux to the test piece. The liquid was distributed through the flow loop until the wanted entry temperature was obtained. This took three hours approximately. Steady state conditions were obtained when the fluid outlet, heater and the aluminum housing temperatures were seen to be stable. This took half an hour approximately. All of the wanted readings were achieved before the heat flux was re-set to the next required value and the process iterated. During the exams to maintain the system pressure close the atmospheric pressure, the vent valve above the condenser was periodically open and a balloon was connected to it to block the vapor to elopement from the system. High speed videos were taken of the boiling occurring on the test piece by a Kodak micro motion 1000 camera. The camera was set to 240 frames /s at a resolution of 720 by 480 pixels.

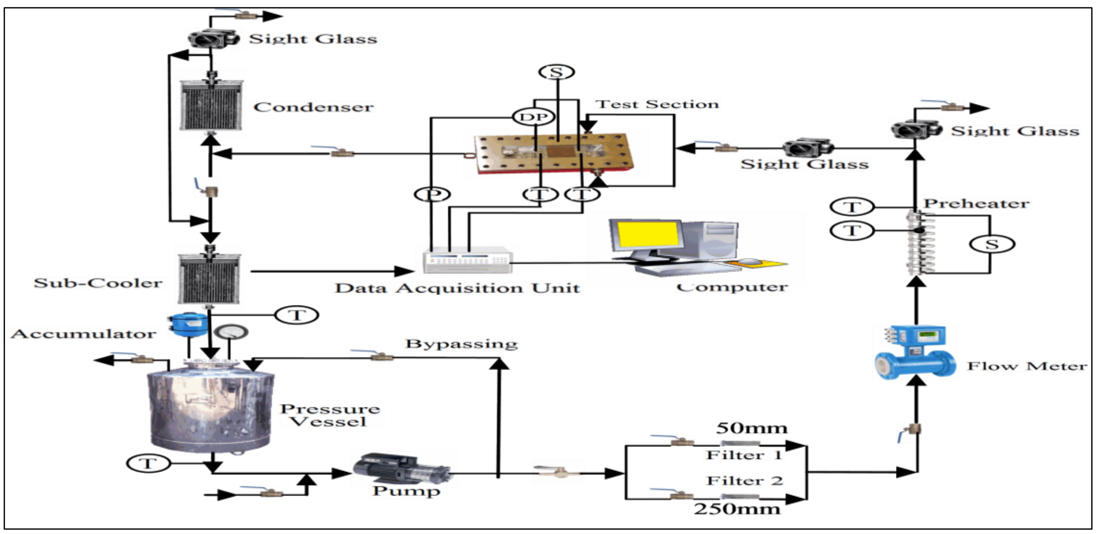

Fig. 1: Schematic of flow loop[7]

Fig. 2 .the aluminum test section is shown. Liquid entered the inlet plenum of the test-section through the two inlet ports, set at 90° to the direction of stream in the aluminum test piece. The plenum chamber dimensions were set to reduce the liquid velocity to close to zero before it.

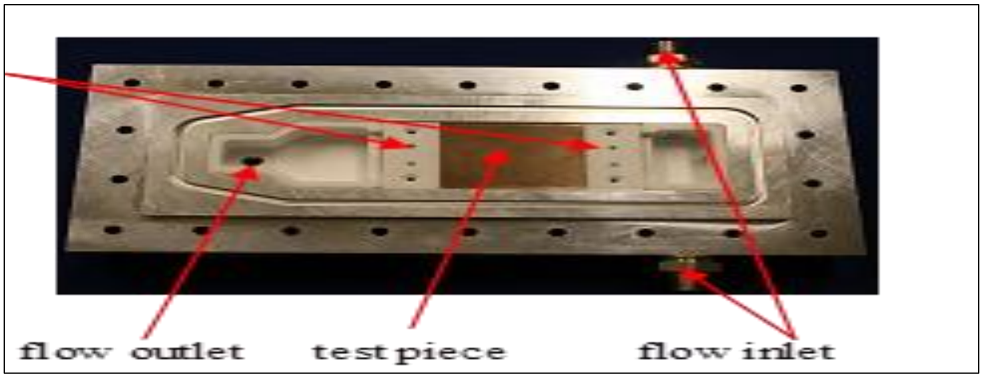

Fig.2. Test section[7]

The plate channel test piece is shown in Fig.3.The plate channels were constructed from a piece of copper, 50 mm wide by 50 mm long and 6 mm high. The channels were formed by cutting slots 1 mm wide and 1 mm deep in the longitudinal direction. The slots were 1 mm apart. Three holes, 0.6 mm in diameter by 12.5 mm long, were drilled into the test piece at the inlet and outlet ends. The holes were located 2.5 mm from the top of the boiling surface and 11, 25 and 39 mm from an edge. These holes allowed six sheathed K-type thermocouples, 0.5 mm in diameter, to be located below the boiling surface. All thermocouples were calibrated in a water bath and were accurate to ±0.1 K..

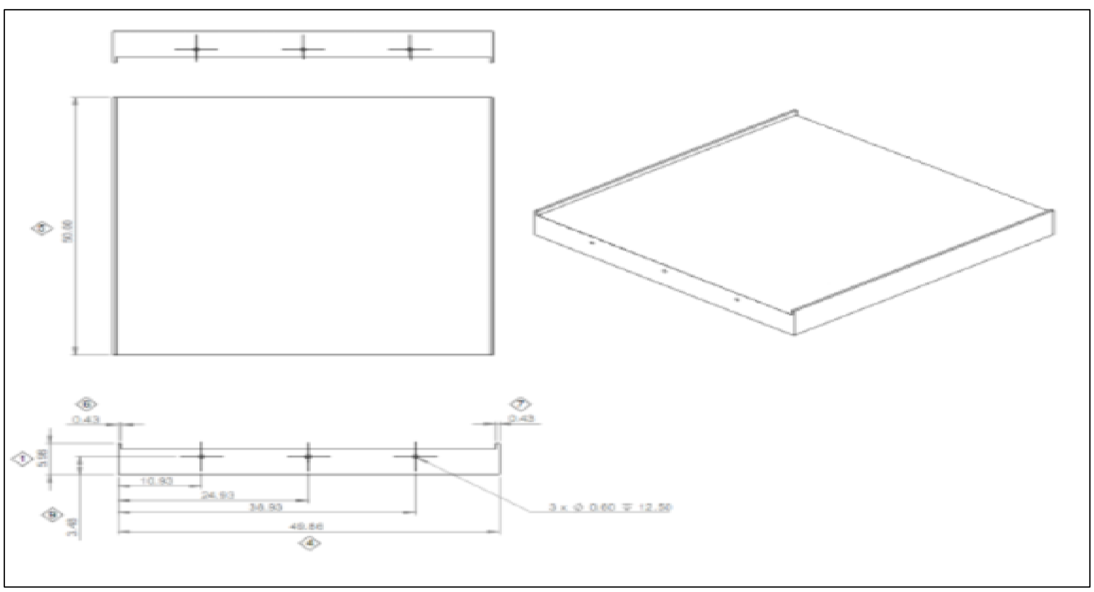

Fig.3. plate channel test piece[7]

3. Data Processing

The heat flow from the heater to the test piece is proportional to the difference between the heater temperature and the test-piece surface temperature. Data taken when the liquid outlet temperature was below the saturation temperature, deduced from the outlet fluid pressure, were used to establish this relationship. The effective heat flux was determined from the ratio of increase in sensible heat of the liquid to the base area of the test-piece, and were correlated by

$$q_E = 1.9\,(T_h - T_w) - 1.5 \qquad (1)$$

The working fluid, which had a liquid specific heat capacity of was delivered to the test piece in a sub-cooled state. The test-piece was therefore divided into two lengths, a single phase length and a boiling length .The fluid which had a saturation temperature of approached the test piece, of width *W*, at a temperature of T_{in} and at a mass flow rate of *M*. The length of the test piece in single-phase flow was therefore estimated from

$$L_{sub} = \frac{MC_P(T_{sat} - T_{in})}{q_E W} \qquad (2)$$

Since the test piece had a length *L*, the boiling length was estimated from

$$L_{sat} = L - L_{sub} \qquad (3)$$

A linear pressure distribution was assumed across the test piece. This was deduced from the measured inlet pressure, P_{in}, and the measured pressure drop, Δp. Thus,

$$P = P_{in} - \Delta P \frac{Z}{L} \qquad (4)$$

The wall temperature was obtained by averaging the readings from the three wall temperatures near the test piece inlet and outlet to obtain T_{tc}, which was then corrected for depth from the plate surface, H_{tc}, through the one-dimensional heat conduction equation, i.e.

$$T_w = T_{tc} - \frac{q_E H_{tc}}{K_c} \qquad (5)$$

the heat-transfer coefficient was found from

$$h = \frac{q_E}{(T_w - T_f)} \qquad (6)$$

The local gas-mass fraction, *x*, was estimated from

$$x = \frac{q_E W(Z - L_{sub})}{M h_{fg}} \qquad (7)$$

4. Experimental results

In Fig.4 .shows the two-phase heat transfer coefficient with the empirical database. The inlet and outlet two-phase heat transfer coefficient are plotted for measured and prediction against experiment mass flow rates for 0.0025, 0.005, 0.0075, 0.01 and 0.0123 kg/s; besides, the predicting accuracy of this experiments is also improved for two-phase flow in pipes. Therefore, the empirical succeeds in accurately predicting the two-phase heat transfer coefficient of plate micro-channel heat sink.

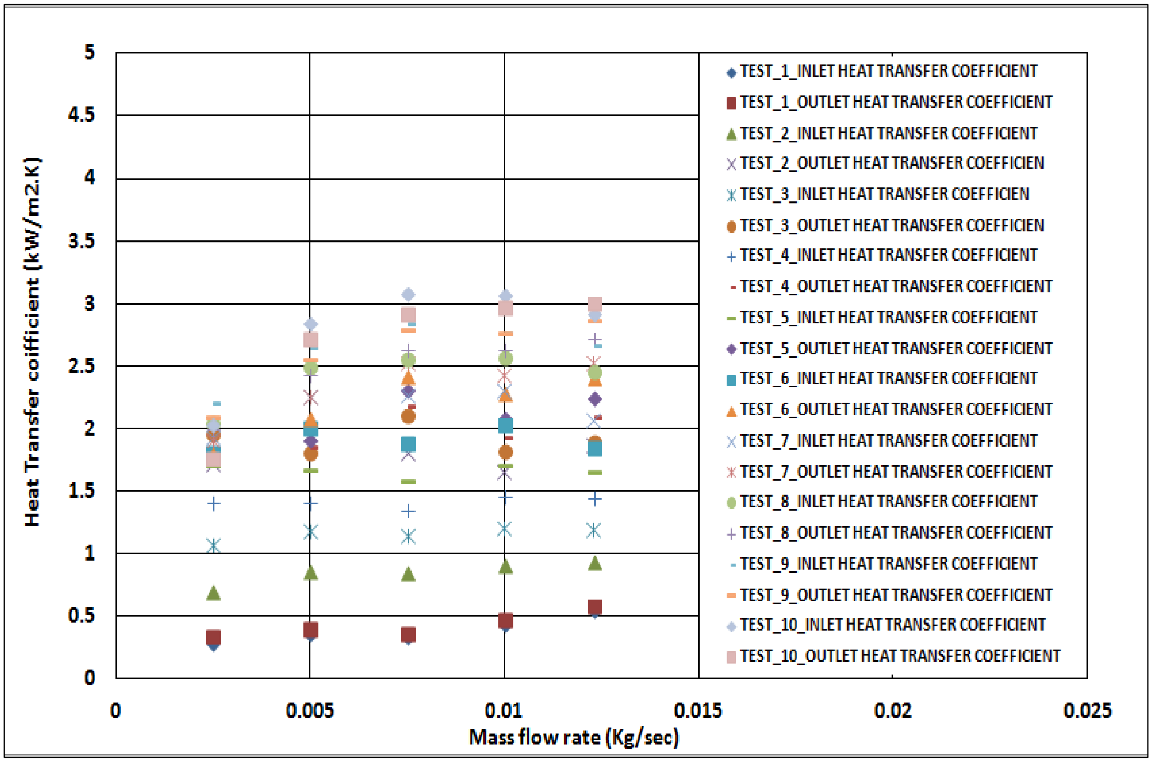

Fig.4. Variation of two-phase heat transfer coefficient with mass flux.

In Fig.5 shows the two-phase heat transfer coefficient with the experimental database. The inlet and outlet two-phase heat transfer are plotted for measured against experiment fluid temperatures for different mass flow rate 0.0025, 0.005, 0.0075, 0.01 and 0.0123 kg/s; the inlet two-phase heat transfer coefficient bigger than the outlet heat transfer coefficient for all mass flow rates into all experiments .

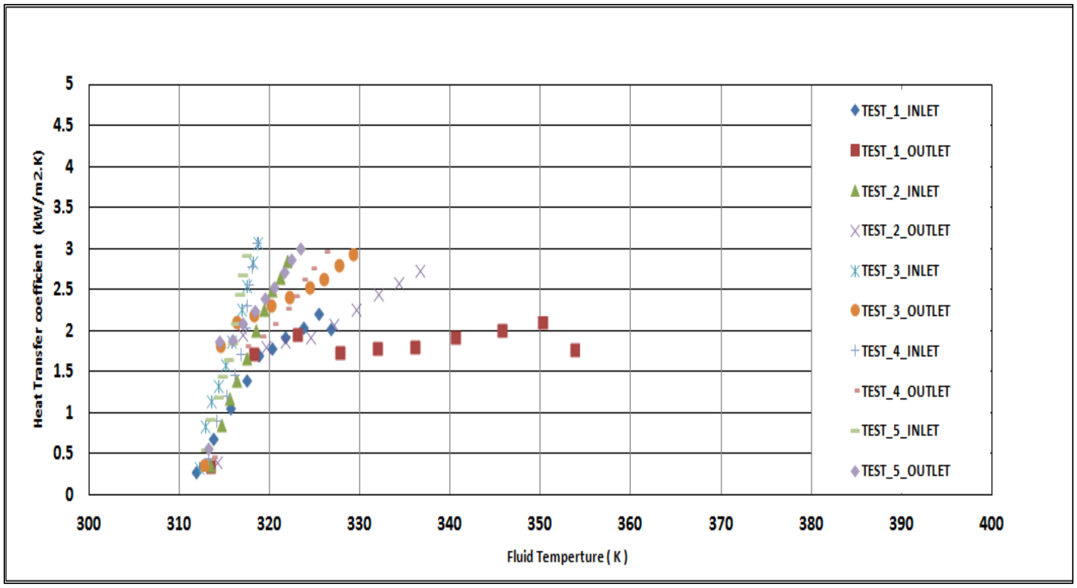

Fig.5. Variation of two-phase heat transfer coefficient with fluid temperature.

Conclusions

The two-phase tests were performed by supplying liquid R113 near atmospheric pressure to the test section with 2.2-5.1 K of sub cooling. Heat-transfer and data was obtained for heat loads in the range 25- 252.5 W. These were applied to each mass flow rate in steps of 25 W. This gave apparent heat fluxes in the range 10-101 kW/m^2. The five mass flow rates used in the two-phase tests were used for each heat flux. The plate surface at the lower mass fluxes, some of the larger heat fluxes were not possible, as damage to the electrical heater would have resulted from the excessive temperatures generated by the reduced heat-transfer area available .The experimental procedure for these tests led to the occurrence of three types of heat-transfer, single-phase, sub cooled boiling and saturated boiling. Single-phase heat-transfer was taken to have occurred when the fluid above the thermocouple was in a sub cooled state and the wall temperature was below the local saturation value. Sub cooled boiling heat-transfer was taken to have occurred when the fluid above the thermocouple was in a sub cooled state and the wall temperature was above the local saturation value. Saturated boiling heat-transfer was taken to have occurred when the fluid above the thermocouple was in a saturated state and the wall temperature was above the local saturation value.

Nomenclature

Cp specific heat at constant pressure

h_{fg} enthalpy of evaporation

K_C copper thermal conductivity

q_E effective heat flux

h heat-transfer coefficient

L test-piece lengths

p pressure

W test-piece widths

Z distance from test-piece inlet

Greek characters

Δp pressure drop

Subscripts

sat saturated value

sub sub-cooled value

tc at thermocouple location

w wall value

References

[1] D. Schrage, J. T. Hsu, and M. Jensen, "Two-phase pressure drop in vertical crossflow across a horizontal tube bundle," AIChE journal, vol. 34, pp. 107-115, 1988.

[2] R. Dowlati, M. Kawaji, and A. Chan, "Pitch-to-diameter effect on two-phase flow across an in-line tube bundle," AIChE journal, vol. 36, pp. 765-772, 1990.

[3] A. Chan and M. Kawaji, "Hydrodynamics of two-phase flow across horizontal in-line and staggered rod bundles," 1992.

[4] P. Feenstra, D. Weaver, and R. Judd, "An improved void fraction model for two-phase cross-flow in horizontal tube bundles," International Journal of Multiphase Flow, vol. 26, pp. 1851-1873, 2000.

[5] K. Ishihara, J. Palen, and J. Taborek, "Critical review of correlations for predicting two-phase flow pressure drop across tube banks," Heat Transfer Engineering, vol. 1, pp. 23-32, 1980.

[6] H. Honda and J. Wei, "Enhanced boiling heat transfer from electronic components by use of surface microstructures," Experimental Thermal and Fluid Science, vol. 28, pp. 159-169, 2004.

[7] D. McNeil, A. Raeisi, P. Kew, and P. Bobbili, "A comparison of flow boiling heat-transfer in in-line mini pin fin and plane channel flows," Applied thermal engineering, vol. 30, pp. 2412-2425, 2010.