International J. of Pure & Engg. Mathematics (IJPEM)
ISSN 2348-3881, VOL. 4 NO. II (AUGUST, 2016), PP. 145-153

SINGLE-PHASE HEAT-TRANSFER AND PRESSURE DROP IN PARALLEL MICRO- CHANNEL HEAT SINK ; EXPERIMENTS AND PREDICTIONS

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Abstract

Heat-transfer coefficients and pressure drops are reported for a test section containing fifty , one mm by one mm, parallel micro-channels. The channels were fifty mm long and had a glass top plate to allow visual observations. The data were produced while boiling R113 at atmospheric pressure. The mass flux range was 200,300,400 and 600 kg/m^2.s and the heat flux range was 10–50.40 kW/m2. The test section was heated from below by an electrical heating method that is normally associated with a constant heat flux boundary condition. However, because of the significant variation in the single- phase, heat-transfer coefficient in the entrance zone, the interceding aluminium and copper material is shown to produce a near isothermal wall boundary condition. The heat conduction effect is taken into account in the data analysis. Heat-transfer coefficients and pressure drops are reported for single-phase flow, with single-phase data obtained. The single-phase results are shown to be position dependent, consistent with a developing laminar flow

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.

Work on multiple parallel micro-channels has revealed the importance of instabilities that result in flows with different flow patterns appearing in different tubes simultaneously, Hestroni et al [1] Qu and Mudawar [2] concluded that existing correlations for boiling in single micro-channels were not applicable to multi-channel arrangements. Several instabilities have been identified. Xu et al [3] , working with water and methanol in 26 parallel rectangular channels 0.3 mm x 0.8 mm x 50 mm long, classified the instabilities in terms of duration. Large amplitude/long period oscillations, occurring at 117 s intervals, resulted in large pressure pulses, with smaller oscillations occurring at periods of the order of a 10 s. Wang et al [4] boiled water in 8 parallel channels, 30 mm long with a hydraulic diameter of 0.186 mm. Two instabilities were observed, one with periods of 3-7 s, increasing with decreasing mass flux, and one with a period of 0.1 s. Bogojevic et al [5] used 40 parallel channels 15 mm long with a hydraulic diameter of 0.194 mm. Again two frequency ranges were observed, one with periods greater than 0.3 s and one with a period of 0.04 s, with the dominant instability being dependent upon the heat- to mass-flux ratio.

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.

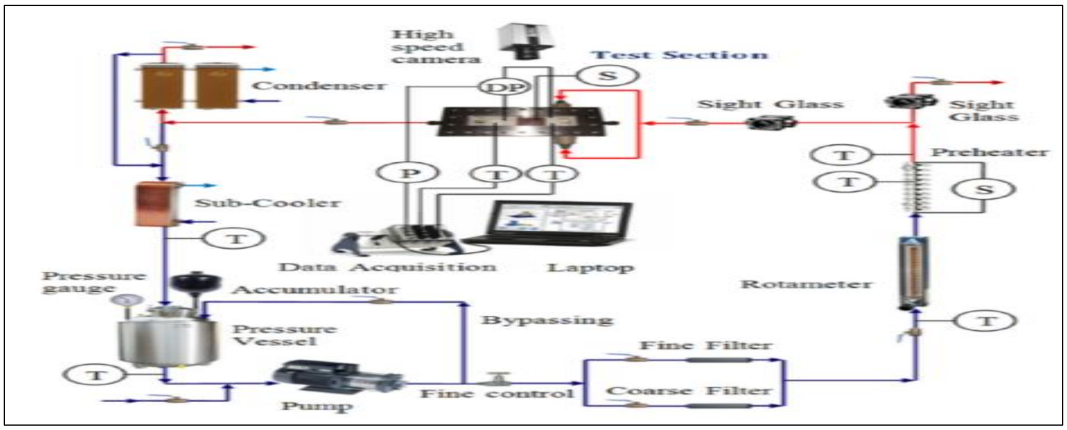

Fig. 1: Schematic of flow loop[6]

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.

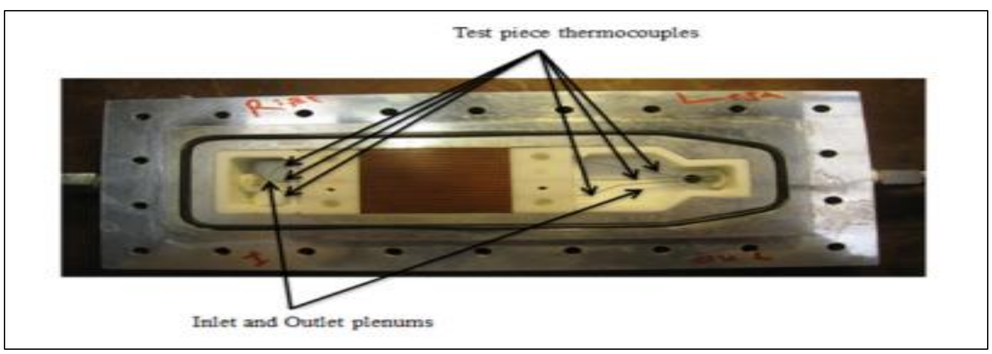

Fig.2. Test section[6]

The parallel channel test piece is shown in Fig.3.The parallel 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..

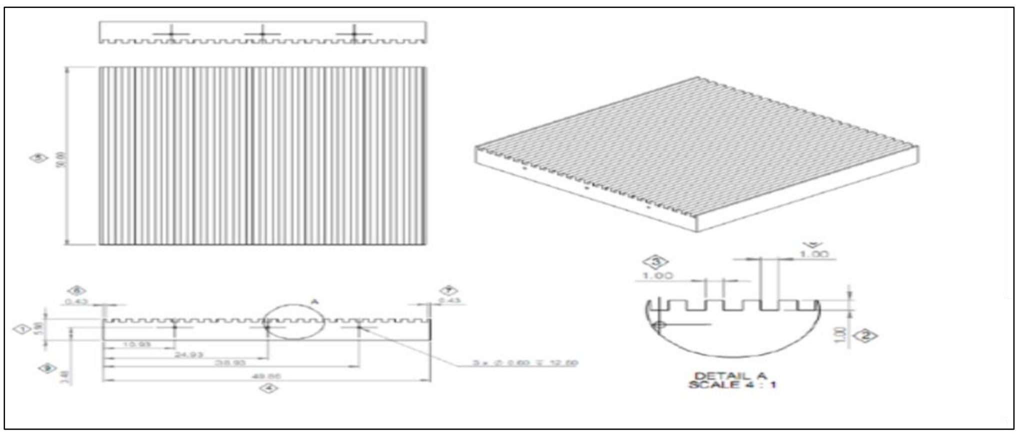

Fig.3. parallel micro- channel test piece[6]

3. Processing method

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

$$q_E = 2.3\,(T_h - T_w) - 0.9 \qquad (1)$$

A specific heat capacity for aR113 working fluid with of ($C_{p,f}$), the liquid temperature was found from

$$T_f = T_{in} + \frac{W}{MC_{p,f}} \int_0^z q_b d_z \qquad (2)$$

In which (M) is the mass flow rate of fluid supplied , (T_{in}) is the liquid inlet temperature and (W) is the width of the test piece. [7] reported that the local-channel, laminar-flow Nussle number, (Nu_z),can be calculated from

$$Nu_z = \left[{Nu_d}^4 + {Nu_f}^4\right]^{0.25} \qquad (3)$$

Where (Nu_f) is the Nussle number in the fully-developed flow regime and (Nu_d) is the value in the developing flow regime. In the fully developed was calculated the Nussle number by.

$$Nu_f = 8.235(1 - 1.833A_L + 3.767{A_L}^2 - 5.814{A_L}^3 + 5.361{A_L}^4 - 2{A_L}^5) \qquad (4)$$

In Which the ratio of the channel height to the channel width is (A_L). The developing flow Nussle number was calculated by [8]

$$Nu_d = 1.54 \left(\frac{Re_f\, Pr_f D_{th}}{z}\right)^{0.33} \qquad (5)$$

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.

$$T_w = T_{tc} - \frac{q_E H_{tc}}{K_c} \qquad (6)$$

the heat-transfer coefficient was found from

$$q_b\,(W_{ch} + W_w) = \alpha\,(T_w - T_f)\,[W_{ch} + 2\eta H_{ch}] \qquad (7)$$

The fin efficiency was determined by considering that the fins could be processed as rectangular fins with adiabatic tips, i.e.

$$\eta = \frac{tanh(\lambda\, H_{ch})}{\lambda\, H_{ch}} \qquad (8)$$

In which λ was the fin parameter, given by

$$\lambda = \sqrt{\frac{2\alpha}{k_c W_w}} \qquad (9)$$

Where k_c , is the thermal conductivity of copper and α single-phase heat transfer coefficient . An initial analysis revealed that the fin efficiency was always near to unity. It was subsequently got as one. The copper test piece has three thermocouples 12.5 mm from the inlet was set inside and also three thermocouples 12.5 mm from the outlet was set inside the copper test piece. A typical two-dimensional unit cell at the location of a thermocouple situated inside the copper test piece for the parallel test pieces is seen in the table below

Table 1: Dimension for channel heat transfer analysis

Parallel channel test piece									
W_w	W_{ch}	W_{cell}	L_{cell}	H_{w1}	H_{w2}	H_{w3}	H_{ch}	H_{tc}	H_{cell}
mm	mm	mm	mm	mm	mm	mm	mm	mm	mm
1.0	1.0	2.0	2.0	10.0	5.0	5.0	1.0	1.5	21.0

$$Pr_f = \frac{C_{p,f}\, \mu_f}{k_f} \qquad (10)$$

$$Re_f = \frac{G\, D_{th}}{\mu_f} \qquad (11)$$

In which (μ_f) is the liquid dynamic viscosity, $(C_{p,f})$the liquid specific heat capacity at constant pressure is and (G) is the mass flux, k_f thermal conductivity (W/m.K). The equation of the hydraulic diameter to the copper channel in the existence of the heat-transfer procedures, calculated by

$$D_h = \frac{4\, H_c W_c}{2(W_c + H_c)} \qquad (12)$$

4. Experimental results

Single-phase analysis, heat transfer coefficients measurements and predictions which were presented are analyzed in this paper. For the parallel channel surfaces, several popular macro- and micro-channel correlations for the empirical data were compared with the heat-transfer data. The variation of single phase heat transfer with mass flux is shown for the parallel channel surface in Figure 4. The heat transfer coefficients are shown to be a power law function of the mass flux.

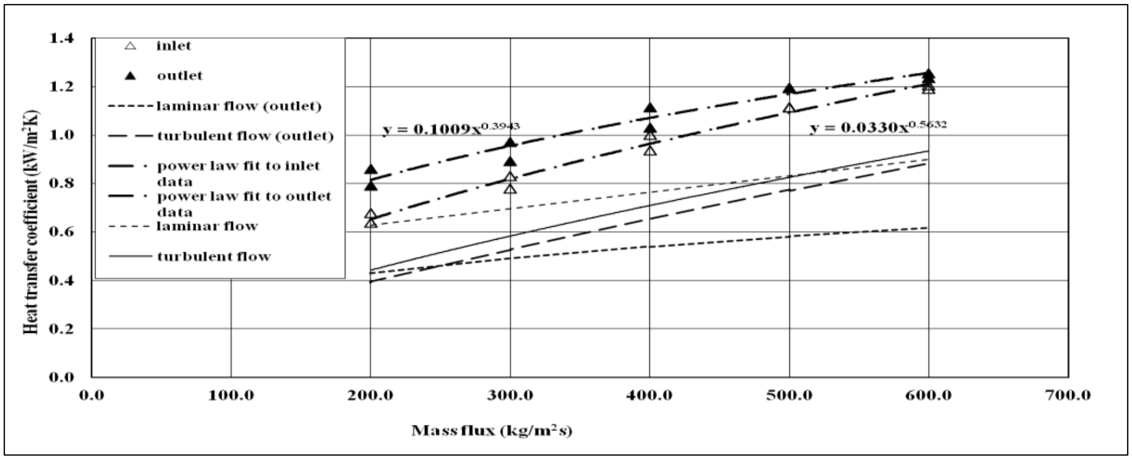

Fig.4 .Variation of heat transfer coefficient with mass flux

In Fig.5 shows the comparison of the prediction of the correlations with the empirical database. The pressure drop is plotted for measured and prediction against experiment mass fluxes for 200,300,400 and 600 kg/m^2.s .T he pressure drop is increased with increasing heat flux, In the laminar flow range the pressure drop is directly proportional to the velocity .The pressure drop is shown to be a power law function of the mass flux, as would be expected for laminar flow in channels.

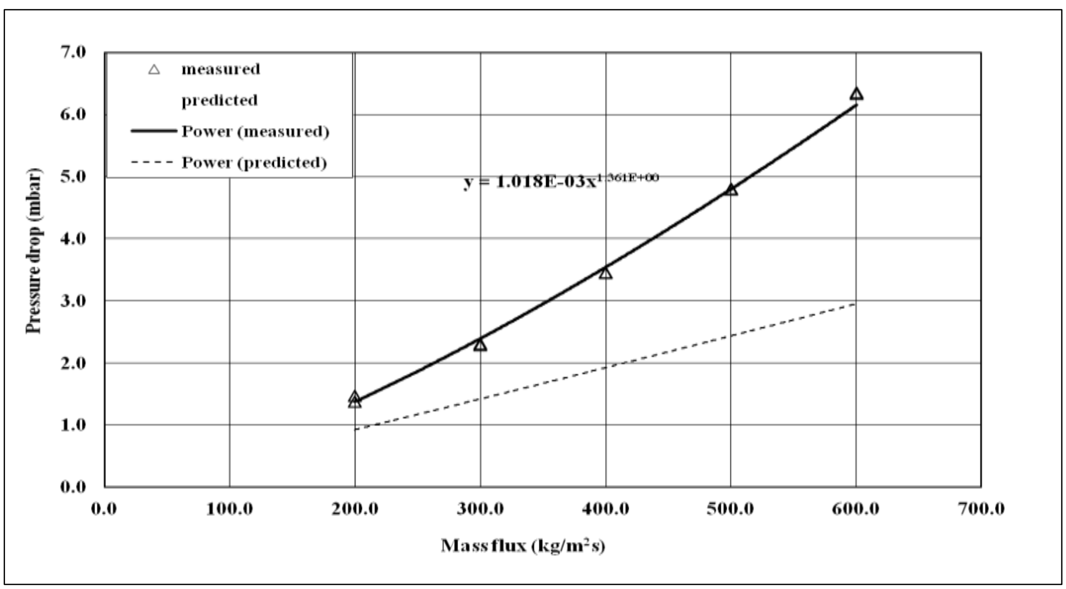

Fig.5 .Variation of pressure drop with mass flux

Conclusions

A twenty five parallel channel test piece boiling R113, Heat transfer and pressure drop have been obtained for them at atmospheric pressure. The channel's dimensions were fifty mm long by one mm square and by a constant heat-flux the channels were heated from below. The following conclusions can be drawn from this study:-

- The data for single- phase heat transfer coefficients and pressure drops for R113 working fluid flow were calculated experiments and predictions and the error were very small.
- All mass fluxes the flows were laminar at outlet and inlet values in the test section. Also the effective heat flux experiments data were very close to predictions data for a range of mass fluxes.
- Total prediction single-phase pressure drop evaluated from Ref [9] ;the measured single-phase pressure drop has taken from experiments data .

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