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BOILING FLOW HEAT TRANSFER IN MICROCHANNEL: EXPERIMENTAL AND NUMERICAL INVESTIGATION

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ABSTRACT

Rapid miniaturization in electronic industry has resulted in continuous increase of heat dissipation requirements thereby making a call for highly efficient cooling technology. One potential solution for this is the use of two phase flow boiling heat transfer. Also microelectronic device performance and reliability is known to increase when effective temperatures are kept below some specific temperature say 40 °C. Hence there is need for a correlation to calculate amount of coolant required to keep temperature below 40 °C as a function of power to be dissipated. This paper reports the results of computational and experimental studies on two phase forced convection in microchannels using water as the coolant. The study mainly concentrates on flow modeling using homogeneous equilibrium model (HEM), fabrication of microchannel of plexiglas using laser beam machining (LBM) and finally experimental analysis to validate results of numerical simulation and establish empirical correlation for calculating coolant flow rate requirement for specified wattage of electronic devices. Correlation is found to be within ±12.6 % of error limits for all the test cases studied to keep the temperature of the device below 40 °C.

Keywords: Electronic Cooling, Forced Convection, Two Phase Flow Heat Transfer

NOMENCLATURE

A	area (m^2)
C_p	specific heat (J/kg K)
d	depth (m)
dP	pressure drop (bar)
D_h	hydraulic diameter (m)
f	friction factor
G	mass flux (kg/m^2)
h	enthalpy (J/kg)
hconv	heat transfer coefficient (W/m^2K)
h_{fg}	latent heat (J/kg)
k	thermal conductivity (W/m K)
L	total length (m)
m	mass flow rate (kg/s)
Nu	Nusselt number
P	pressure (Bar)
P_c	perimeter (m)
Q	heat flow (W)

Re	Reynolds number
t	temperature (^oC)
v	velocity (m/s)
w	width (m)
X	Quality
Z	position across length (m)
z_c	boiling front position (m)
Greek Letters	
ρ	density (Kg/m^3)
Subscripts	
f	fluid
l	liquid
m	mean
sp	single phase
sat	saturation
tp	two phase
v	vapour
w	wall

INTRODUCTION

The rapid growth of high density power electronics with the increased miniaturization of microelectronic devices and processing speed, thermal issues are affecting overall electronic packaging and system capabilities. In the past few years, heat flux dissipation requirement has increased from 100 W/cm^2 to 1000 W/cm^2 which exceeds the capability of present cooling technology "[1]". Tuckerman and Pease stated that single phase forced convective cooling in microchannels should be feasible for circuit power densities of more than 1000 W/cm, and demonstrated a microchannel heat sink that removes 790 W/cm with 71 ^oC temperature increase at 600 mL/min flow rate "[2]".

With the increase in heat flux dissipation, device temperature also increases, limiting the heat dissipation capability "[2]". But flow boiling due to extraction of latent heat component can dissipate heat without increase in temperature. Also, single phase heat transfer coefficient for laminar flow of water in a 200µm square channel is around 0.1 MW/m^2-^oC, whereas flow boiling heat transfer coefficient can exceed 1 MW/m^2-^oC making two phase flow more attractive for cooling of electronic devices "[3-5]". In the present study, HEM model has been adopted for two phase flow through microchannel. Numerical analysis has been conducted on microchannels of 250 µm x 500 µm cross section. Results of HEM studies on two phase flow through microchannels using water as the fluid medium were analyzed to predict the location of boiling front, pressure drop, variation of fluid coolant temperature and wall temperature along the length of microchannel.

Conventionally, any channel or tube with hydraulic diameter less than 0.5 mm is called a microchannel "[5-6]". The microchannels are fabricated on the surface of electronic devices by either precision micromachining or micro-fabrication technology. The coolant is forced to pass through microchannels to carry away heat from the hot surface of electronic device. There are various fabrication techniques available for manufacturing of micro devices, namely laser cutting, photolithography, chemical vapour deposition, physical vapour deposition, electric discharge machining, to name a few "[7-9]". In the present study laser beam machining (LBM) with very high energy density laser was used for fabricating microchannel.

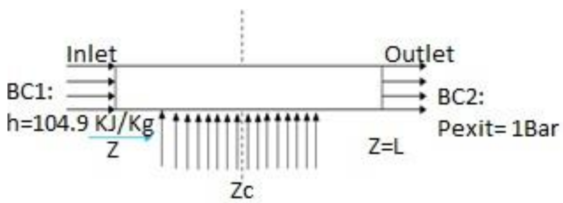
Large number of experimental studies has been done on boiling heat transfer in microchannel encompassing various flow regimes, phase transition and heat transfer correlations "[10-12]". But theory of flow boiling in microscale devices is still not well established. In this study experimental investigation on cooling effect at varied coolant flow rate was carried out. For the same, it was difficult to provide heat flux to the walls of fabricated plexiglas channel. So, copper microchannel of the same dimension was then used for the experiments. Two phase flow experiments were conducted mainly to validate the results obtained from numerical simulation of HEM model. Microelectronic device performance and reliability are known to increase when effective temperatures are kept below some specific temperature say 40°C "[13]". Experimental results were used to establish empirical relation between mass flow rate and heat flux supplied. The aim was to compute amount of mass flow rate of the coolant to maintain the temperature of the device below 40 ºC for increased performance of electronic devices.

NUMERICAL MODELLING USING HEM

Numerical simulations were carried out on a microchannel with cross section dimensions 250 µm x 500 µm and length of 2 cm. Water was taken as coolant that enters the channel at ambient temperature of 25 ºC. Constant heat flux was applied to the bottom surface. Figure 1(a) shows schematic diagram of microchannel with boundary conditions using which simulations were performed. Water extracts heat continuously, resulting in phase transition from single phase to two phase flow.

In two phase flow studies, it is very important to obtain the position of boiling front beyond which vapour quality is greater than zero. Hence two phase flow exists beyond this point. Fluid starts boiling whenever bulk enthalpy of fluid matches saturated vapour enthalpy corresponding to given pressure as shown in Fig 1(b). Point of intersection of two enthalpy curves indicates position of boiling front. In the two phase region properties like density, viscosity, enthalpy, etc. behave dynamically. Hence role of quality becomes important, adding extra complexity to the problem. Inlet enthalpy of water at 25 ^{o}C and outlet pressure have been taken as boundary conditions which look more intuitive than what have been used in previous works "[14]". As a result one needs to march backward for solving governing differential equations.

1(a)



1(b)

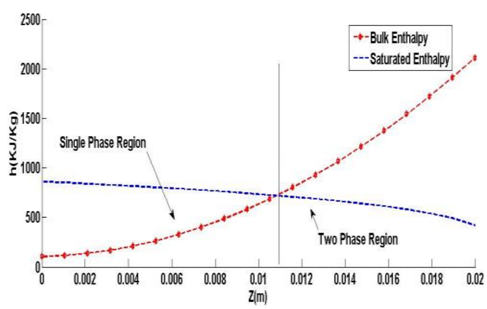


FIGURE 1: (A) SCHEMATIC OF BOILING FLOW HEAT TRANSFER THROUGH MICROCHANNEL (B) LOCATION OF BOILING FRONT

Pressure distribution in single phase region is given by Eqn. (1), where f_{sp} is single phase friction factor. Effect of pressure drop due to contraction can be ignored because of constant area of cross section. Pressure distribution in two phase region is given by Eqn. (2) in which first term signifies pressure drop due to friction while second term signifies pressure drop due to acceleration. Capillary effect i.e. wall effect was ignored to simplify the analysis. Here f_{tp} is two phase friction factor, which is given by Cichitti relation "[15]".

$$\Delta P_{sp} = \frac{f_{sp}\, z\, G^2}{2\, \rho\, D_h} \tag{1}$$

$$\frac{-dp}{dz} = \frac{f_{tp}\, G^2}{2\, \rho\, D_h} + \frac{d}{dz}\left(\frac{G^2}{2\, \rho}\right) \tag{2}$$

Average density is given by Eqn. (3) which has been derived from assumptions of HEM "[15]". Here subscript *m, v* and *l* are used for mean, vapour and liquid phase properties.

$$\frac{1}{\rho_m} = \frac{X}{\rho_v} + \frac{1-X}{\rho_l} \tag{3}$$

Effectiveness of cooling can be obtained in terms of temperatures of channel wall and fluid. For the same, in single phase region fluid temperature is given by Eqn. (4).

$$T_f(z) = T_1 + \frac{Q\, z}{m\, Cp_l L} \tag{4}$$

Wall temperature and enthalpy of fluid are given by Eqn. (5) and Eqn. (6) respectively. Here ɳ is the fin efficiency which was considered 90% during simulation "[16]". Heat transfer coefficient correlation by Kandliker was used for two phase region "[17]".

$$T_w = T_f + \frac{Q}{h_{conv}(w\, L + 2\, ɳ\, d\, L)} \tag{5}$$

$$m \frac{dh_f}{dz} - ɳ\, h_{conv} P_c (T_w - T_f) = 0 \tag{6}$$

Numerical Procedure

Solution to the problem involves solving all the equations discussed in previous section. Unfortunately all these equations are coupled hence these are solved using numerical methods. Wherever required, diffrential equations are descritized using finite difference method (FDM). Numerical procedure for the same is described by following steps.

- At first boiling front is located as discussed in previous section and Fig. 1(b).
- Pressure drop in single phase is found using Eqn. (1).
- For two phase region Eqn. (2) is discretized using FDM and pressure is calculated by marching in backward direction.
- Fluid temperature in single phase region is calculated using Eqn. (4) while in two phase region fluid temperature is same as saturation temperature corresponding to fluid pressure.
- Wall temperature in single phase region was found using Eqn. (5).
- Fluid enthalpy is again extracted using Eqn. 6 by FDM and this process is repeated until convergence is achieved. Relative error of 10^{-9} is considered as convergence criterion.

FABRICATION OF FLOW DEVICE

Epilog Legend 36 EXT Laser Machine with very high energy density laser was used for fabricating microchannels "[18]". A key advantage of LBM over other traditional patterning techniques is its ability to remove materials ranging from ceramics and metals to semiconductors and polymers without the need to change tooling or chemical processing "[19]". Other notable qualities include lesser thermal distortion i.e. smaller heat affected zone (HAZ), better cutting edge, better surface morphology, etc. Amongst available lasers, pulsed laser is being used more extensively for micromachining because it enables large degree of flexibility in terms of process parameters. Laser pulse may be varied by variation in traverse velocity, laser power, pulse frequency, spot size, to name a few "[18-19]". Detailed parametric study was carried out through experimental investigation and analysis of SEM images. Finally, microchannel of requisite dimension was cut after selecting appropriate process parameters.

Finally a microchannel of dimensions as shown in Fig. 2(a) was cut. This microchannel was further used as test section of microchannel flow device. Microchannel flow device mainly consists of a microchannel of requisite dimension, two reservoirs and capillaries at the end. Reservoirs were used to avoid overflow of coolant. Purpose of using capillaries was multifold. Besides acting as inlet and outlet to the device, they helped in maintaining the driving pressure. Reservoirs were fabricated using combined mode of cutting with very small power and increased velocity. Owing to limitation of spot size diameter, to cut reservoir of 1 mm diameter with 1 mm, number of passes were used with laser beam shifting by raster cut "[18]". Special care was taken while fabricating device to maintain smoothness at the junction of reservoirs and microchannel to avoid vortices formation which may result in larger pumping requirement. Figure 2(a) shows schematic view of fabricated microchannel with reservoirs.

EXPERIMENTAL ANALYSIS

Experiments were performed in microchannel flow device to analyze cooling characteristics. To carry out the flow experiments in microchannel, first and foremost, a fluid flowing and measurement system, together with microchannel flow device was properly designed and built up.

Experimental Setup

The experimental system was divided into three parts namely a channel section, a water driving system and a temperature measurement system as shown in Fig. 2(b). Owing to heating limitations, experiments were performed on copper microchannel having diameter of 500 µm and length of 2 cm instead of plexiglas microchannel. Copper microchannel with sand base was used as test section. Water was injected drop by drop and mass flow rate was controlled by adjusting size and number of water drops using dripper. Required heat flux was provided by heating nichrome wire electrically with specific input values of current and voltage. Thermocouple was used for temperature measurement at different nodes along length. Due to facility constraints only inlet and outlet temperature of fluid was measured, not the fluid temperature at other locations. The whole device was kept in the casing to avoid extra cooling due to convection from atmosphere. For each data point being measured, the flow was considered to be at steady state condition when final observation was made.

Experimental Procedure

At first, microchannel was heated using the heating system as discussed above and then left free for an hour to allow it to reach steady state. Once steady state was reached, temperature was measured at prescribed locations along the length. To cool the wall of the microchannel, coolant water was pumped through a dripper and syringe needle arrangement. Dripper was adjusted to obtain desired coolant flow rate. Water temperature at the inlet was kept 25 °C. Flow rate was measured manually using stopwatch and number of drops. Temperature of coolant varied along the flow direction. The above process was repeated for various flow rates for the same heat input. This completed one experiment. Similar experiments were performed with various other heat inputs. It was ensured that all the working fluid had passed through microchannel and no left over bubbles or impurities remained to block the flow passage. The process was repeated after evacuating channel for three times to ensure reliability and repeatability of the results.

RESULTS

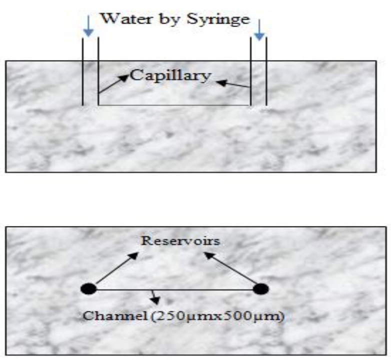
Results Based On Simulation

The numerical model developed was exercised for the microchannel with cross section of 250 µm x 500 µm and length of 2 cm. Exit pressure of 1 bar and inlet enthalpy of 104.9 KJ/Kg (Enthalpy of water at 1 bar and 25 ^{o}C) are taken as bounadry conditions for the problem. Results for the flow rate of 10 µl/s are discussed here.

Pressure Drop Along Microchannel Length: Figure 3(a) shows variation of pressure along the microchannel with 0.5 W heat supply. Observed pressure drop in single phase region was very mild while that in two phase region was very high. The reason being that in the single phase region pressure drop is only due to friction while in latter acceleration terms also contribute to pressure drop as given in Eqn. 3. Also friction loss is accompanied with a multiplier whose value was greater hence pressure drop due to acceleration term dominated in the two phase flow region.

Pressure Drop vs Heat Supplied: Figure 3(b) shows variation of total pressure drop in microchannel with heat supply. Pressure drop increased with the increase in heat input owing to the increase in convection current with increase in heat input. Moreover as heat input increases, two phase region would be larger resulting in larger pressure drop.

2(a)



2(b)

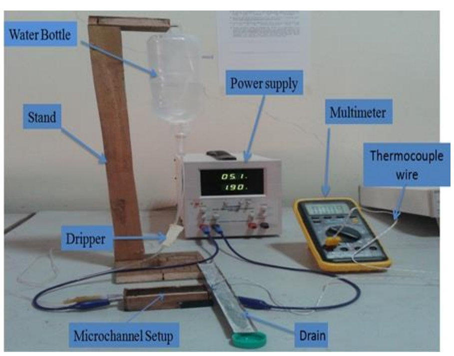


FIGURE 2: (a) SCHEMATIC OF MICROCHANNEL FLOW DEVICE. (b) EXPERIMENTAL SETUP

Fluid And Wall Temperature: Figure 3(c) shows variation of T_f and T_w along microchannel length. Up to Z_c, T_f and T_w increased approximately linearly while beyond Z_c (i.e. in two phase region) T_f and T_w both decreased. The reason being that in single phase region fluid enthalpy increases due to heat input while in the two phase region there is existence of saturation conditions and since pressure decreases so temperature also decreases. Up to Z_c difference between T_f and T_w was almost constant while after Z_c they were same and equal to the T_{sat}. However, the vapour quality kept on increasing as the coolant kept flowing down the microchannel.

Experimental Results

Experiments were performed for various values of heat supplied using water with inlet temperature 25 °C as coolant. Also, required coolant flow rate was obtained to keep temperature of microchannel below 40 °C.

Single Phase Flow Heat Transfer: Temperature profile obtained was found to increase monotonically along the length showing coolant to be in single phase as shown in Fig. 4(a), Fig. 4(b) and Fig. 4(c). With the increase in flow rate, wall temperature was found to decrease. For 4.41 heat input, with flow rate of 10.4 µl/s device peak temperature was found to be below 40 °C and hence was considered appropriate under this heat input. Experiment was repeated for heat input of 5.04 W, 5.6 W and 6.1 W as well. Coolant flow rate required for cooling below 40 °C corresponding to various heat inputs is summarized in Tab. (1).

Two Phase Flow Heat Transfer: Figure 4(d) and Fig. 4(e) show temperature profiles obtained for heat inputs of 7 W and 9 W respectively. With no flow condition, temperature of the wall everywhere was well above 100°C. Similar to numerical simulation results temperature profile was found to first increase then decrease in two phase region as shown in Fig. 3(d). Reason for difference in profile is discussed in section 6. Also required flow rate is much more to keep the temperature below 40 °C in comparison to single phase cooling. With flow rates of 40.7 µl/s and 71 µl/s device temperature was found to be below 40 °C for heat inputs of 7 W and 9 W respectively.

Establishment of Correlation

Overheating should be avoided for proper functioning of the electronic devices. For the same, coolant has to flow continuously. It was observed that the amount of coolant required was a function of the heat input. If one knows the amount of coolant flow rate required for keeping the peak temperature below 40 °C, then the problem of overheating can be handled effectively and hence functioning of device can be improved. Coolant flow rates required corresponding to various heat inputs were extracted from Tab. (1).

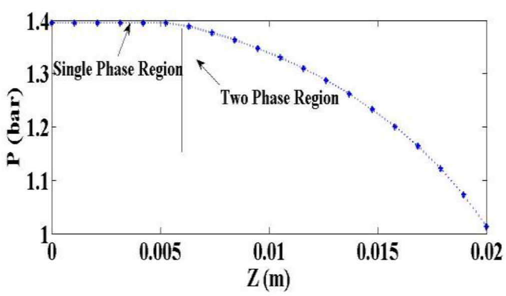
Figure 4(f) shows the plot of coolant flow rate vs heat input obtained using curve fitting tool in MATLAB®. Coolant flow rate required for cooling below fixed temperature for different heat input is given as input to MATLAB® curve fitting toolbox. Curve fitting was done using various degree curves viz. linear, quadratic and cubic. The norm of residuals was observed for every case of the curve fitting. Its minimum value gave the best curve fitted on the data points. With minimum value of norm of residual, cubic curve was chosen as final correlation between coolant flow rate and heat input as given in Eqn. (8).

$$Y = 0.62x^3 - 5.1x^2 + 13x + 0.016 \quad (8)$$

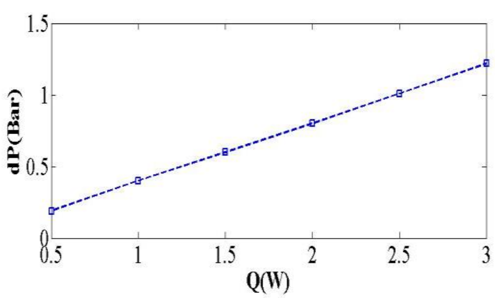
Where Y= Coolant flow rate in µl/s x=Heat input in W

Simulated Experiments: To test the accuracy of the correlation obtained, two more experiments with heat input 4.7 W and 6 W were performed to obtain respective required coolant flow rates. The same were then compared with corresponding coolant flow rates obtained from correlation shown as points A and B in Fig. 4(f).The simulated experiment with 6 W heat supplied gave a required flow rate of 32.253 µl/s experimentally while the correlation provided value of 28.36 µl/s (12.6% deviation). Similarly when experiment was repeated with 4.7 W heat supplied, the required flow rate was found to have deviation of 12.03%.

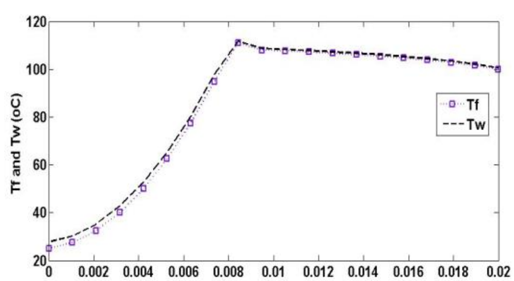
3(a)



3(b)



3(c)



3(d)

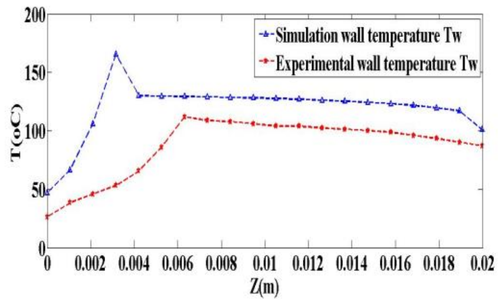


FIGURE 3: SIMULATION RESULTS WITH MICROCHANNEL DIMENSIONS 250 µm X 500 µm AND LENGTH OF 2 cm (A) VARIATION OF PRESSURE DISTRIBUTION ALONG CHANNEL LENGTHFOR 0.5 W HEAT SUPPLY AND 10 µl/s COOLANT FLOW RATE (B) EFFECT OF HEAT SUPPLIED ON PRESSURE DROP ALONG MICROCHANNEL (C) SIMULATED FLUID AND WALL TEMPERATURE DISTRIBUTION ALONG LENGTH OF MICROCHANNEL FOR 0.5 W HEAT SUPPLY AND 10 µl/s COOLANT FLOW RATE (D) COMPARISON OF SIMULATION WALL TEMPERATURE DISTRIBUTION AND EXPERIMENTAL WALL TEMPERATURE DISTRIBUTION FOR HEAT INPUT OF 7 W AND 10 µl/s COOLANT FLOW RATE

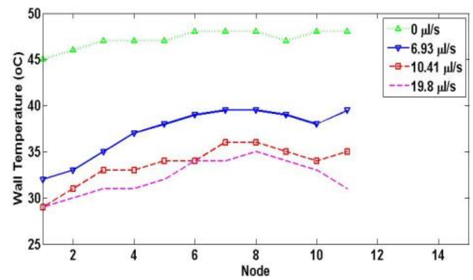
VALIDATION

Under varied flow rate conditions, cooling behaviour of microchannel wall can be seen clearly and was found matching with simulation results of flow boiling heat transfer as shown in Fig. 3(d) for 7 W heat supplied. The reason for difference in temperature magnitudes may be attributed to difference in convection heat transfer as well as in dimensions of microchannel. During experiments, in addition to coolant heat transfer, convection loss from outside wall surface was also responsible to reduce the temperature of the wall significantly. In two phase region, again both the plots are matching. In two phase region, heat transfer coefficient is already enhanced and temperature difference between the fluid and the wall is not significant.

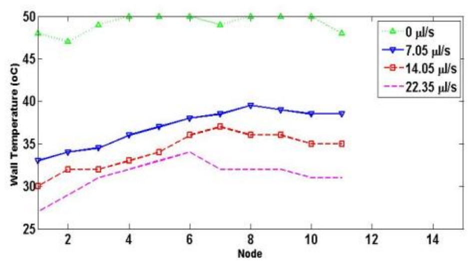
CONCLUSIONS

The paper reports the results of our studies on two phase forced convection in microchannel using water as the coolant medium. Numerical simulation was performed using homogeneous equilibrium model for which governing differential equations were solved to predict the boiling front, pressure drop and cooling effectiveness as functions of exit pressure and heat input. A microchannel flow device was fabricated using laser beam machining for which parametric analysis was performed. From parametric analysis it was concluded that both increased power and reduced velocity result in enhanced concentration of laser but thermal effects would be more pronounced in case of increase in power. While smoother surface finish may be obtained by increasing pulse frequency, larger width of cut channel may be obtained by increasing spot size. Preliminary experiments showed similar trend as found in simulation studies. Also a numerical correlation was established to find out coolant flow rate required for cooling device below fixed temperature as a function of heat supplied. Simulated experiments show accuracy of correlation to be within error limits of ±12.6%.

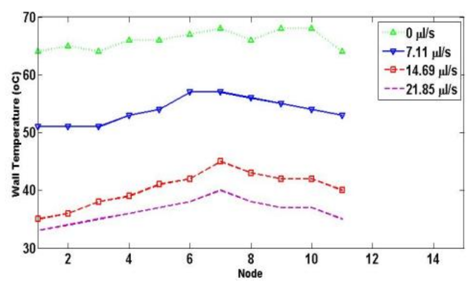
4(a)



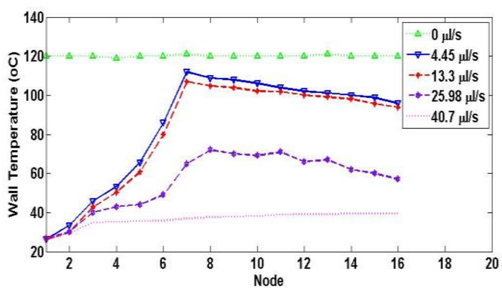
4(b)



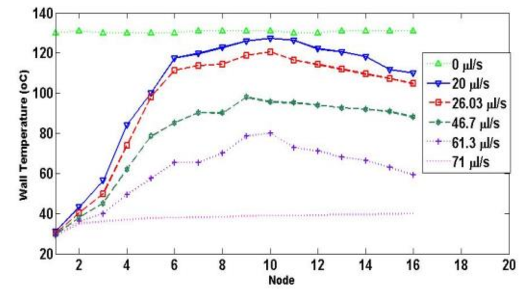
4(c)



4(d)



4(e)



4(f)

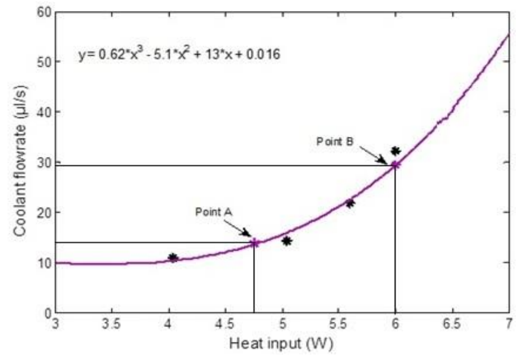


FIGURE 4: EXPERIMENTAL WALL TEMPERATURE DISTRIBUTION FOR (A) 4.41 W (B) 5.04 W (C) 5.6 W HEAT SUPPLIED (D)7W (E) 9 W HEAT SUPPLY (F) CORRELATION FOR CALCULATING COOLANT FLOW RATE REQUIRED AS A FUNCTION OF HEAT INPUT FOR KEEPING DEVICE BELOW 40^{O}C

TABLE1. SUMMARIZED RESULTS OF EXPERIMENTS

S. NO	Q (W)	Coolant Flow Rate (µl/s)
Single Phase Flow		
1	4.41	10.4
2	5.04	14.05
3	5.6	21.85
4	6.1	32.8
Two Phase Flow		
5	7	40.7
6	9	71
Simulated Experiments		
7	4.7	13.6
8	6	32.25

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