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Kimtiow Ooi
Nanyang Technological University

Yongliang Teh
Nanyang Technological University

Florence Pei Fan Ng
AEM Singapore Pte. Ltd.

Tiewtoon Phay
AEM Singapore Pte. Ltd.

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Kim Tiow OOI[1]*, Yong Liang TEH[2], Florence Pei Fan NG[3], Tiew Toon PHAY[4]

[1,2] Nanyang Technological University,
School of Mechanical & Aerospace Engineering,
Thermal & Fluids Engineering Division,
Singapore,
[1] (65) 6790 5511, mktooi@ntu.edu.sg
[2] (65) 6790 6174, tehy0002@ntu.edu.sg

[3,4] AEM Singapore Pte Ltd,
Precision & System Integration,
Singapore,
[3] (65) 6551 0318, florenceng@aem.com.sg
[4] (65) 6551 0318, ttphay@aem.com.sg

ABSTRACT

Flow boiling in mini and micro-channels offers high heat removal rates and is widely employed in many developing technologies such as electronics cooling. However, the minuteness of such channels imposes precision demands on their fabrication, leading to increased manufacturing costs. In this paper, a new miniature heat exchanger design named Thermal Control Unit (TCU) which employs wire inserts disposed within the mini-channels is presented. Through theoretical and experimental studies, the new design has been shown to achieve high heat transfer capabilities comparable to that of micro-channels, while alleviating precision demands through the reduction of the hydraulic diameters by partial blockage of the flow area using the wire inserts. Furthermore, the components of the TCU are easily fabricated by conventional machining which makes it appealing for industrial applications. The TCU when use in conjunction with the multi-evaporator system presented in Part II of this paper series (Ooi *et al*., 2008) is anticipated to provide an effective and cost-efficient thermal solution for the electronics cooling industry.

1. INTRODUCTION

Typically, all high-performance electronic devices are subjected to a 100% functional test prior to being shipped by the manufacturer (Pfahnl *et al.*, 1999; Sweetland and Leinhard, 2003). For example, high power microprocessor devices are in general subjected to a classification test to determine the effective operating speed of the device. During this classification test, it is important to keep a temperature of a die in the microprocessor device at a single prescribed temperature while the power of the device is varied from 0% to 100% of the power rating in a predetermined test sequence (Sweetland and Leinhard, 2003). In order to ramp and maintain the die at the prescribed temperature for testing purposes, equipments known as thermal control unit (TCU) have been designed.

A TCU is basically used to provide heating and cooling to maintain the set temperature of a device under test (DUT) such as microprocessor devices. The heating process is simply achieved by installing a heater within the TCU. As for the cooling process, it is accomplished by passing a cooling medium through the TCU, which can either exist in a single phase flow of either gas or liquid, or a two-phase liquid-vapour flow.

In the microprocessor testing, chilled water TCU technology is commonly used which employs the use of single phase flow to remove heat by forced convection. Recently, it has been found that the power densities in microprocessor devices have approached high levels of about 50 W/cm^2 to about 100 W/cm^2 (Kromann, 1996; Malinoski *et al.*, 1998; Tadayon, 2000). As the power densities of the microprocessor device are expected to get more intense, it is possible that the single phase cooling technology would reach its limits in terms of testing microprocessors at lower temperatures as well as removing the larger amount of heat being generated.

In order to overcome such restrictions, the use of two-phase liquid-vapour flow which capitalizes on the high adsorption ability of latent heat of vaporization has been considered. Apart from utilizing two-phase heat transfer, mini-channels which are known to provide high heat transfer capability is employed in the new TCU design. The innovation in the design lies with the insertion of wires within the mini-channels to improve the heat transfer rate that is comparable to that of micro-channels. The theoretical and experimental studies of the new TCU design will be discussed shortly in this paper.

2. DESIGN OF THE THERMAL COOLING UNIT

The key focus in the TCU design is on the cooling aspects where the specifications are presented in Table 1. It can be seen that the value of the maximum cooling capacity is often governed by the requirement to ramp down the temperature of DUT within a specified time period. For example, although the cooling requirement during steady state based on the maximum heat load is about 18 W/cm^2, the maximum cooling capacity requirement is increased to 110 W/cm^2 when the maximum temperature drop is taken into account (assuming a DUT mass of 20g made of copper). Therefore, in order to achieve such high heat removal rates, it is anticipated that two-phase flow must be used.

Table 1: TCU Design Specifications

Maximum Heat Load	250 W
Temperature Range	15 °C to 100 °C
Ramp Rate	12 °C/s
DUT Surface Dimensions	37.5 x 37.5 mm^2
Overall TCU Dimensions	85 x 70 x 50 mm^3

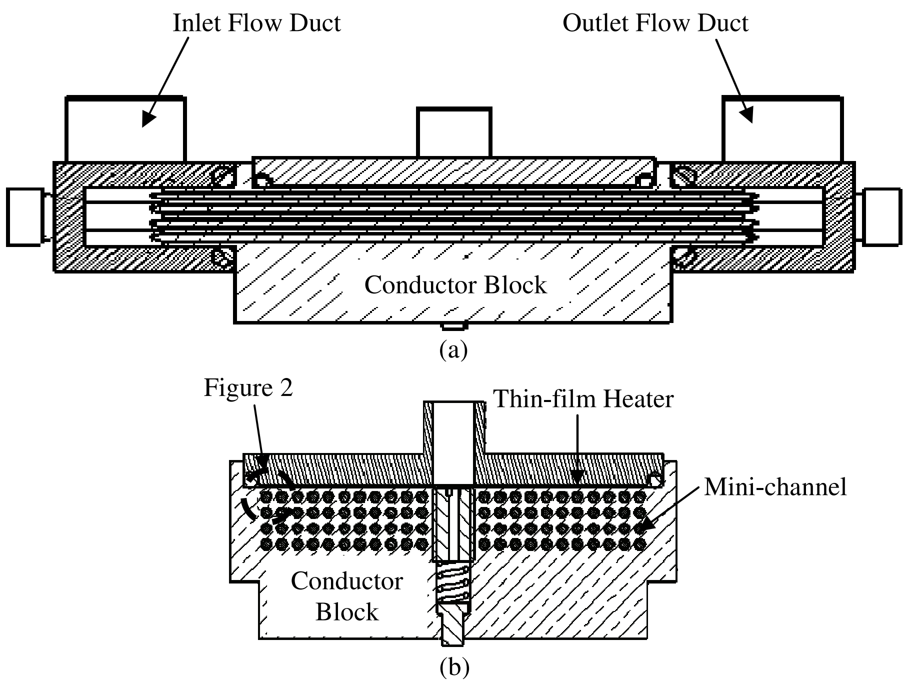

Figure 1: (a) Front View of TCU Design; (b) Side View of TCU Design

Figure 1 presents the basic design of the TCU which consists of few components, namely, a) a conductor block made of copper having multiple through mini-channels, b) a wire insert in each mini-channel, and c) inlet and outlet flow ducts. In each mini-channel, the wire insert creates a partial blockage of the flow which reduces the flow area, hence achieving a reduced hydraulic diameter which dimensions are comparable to that of a micro-channel. It is illustrated in Figure 2. By doing so, heat transfer coefficients are dramatically increased which enhances the heat transfer performance. An important point to note is that the design bears the characteristic of the micro-channels in terms of heat transfer ability but yet allows low cost fabrication to be carried out using conventional machining.

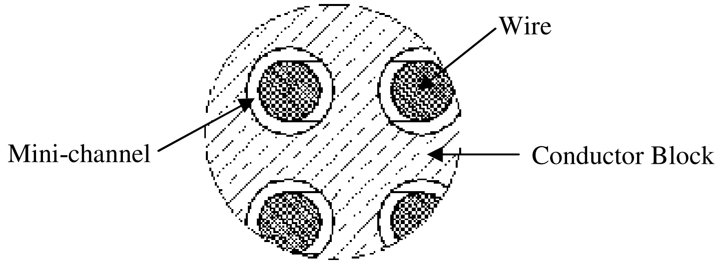

Figure 2: Wire Inserted into Each Mini-channel

The operation mode of the TCU is as follows. When the TCU comes into contact with the DUT, a cooling medium in liquid form (eg. liquid refrigerant) is received by the inlet flow duct and channeled through the mini-channels containing the inserts. Flow boiling takes place in the flow channels when heat is transferred from the DUT to the cooling medium. The vaporized fluid then exits the flow channels through the outlet flow duct. In the process of heat removal, the temperature of the TCU may be ramped down or maintained at a prescribed temperature of the DUT.

As for the TCU to provide heating for the DUT, a thin-film heater is used and is positioned in the TCU as shown in Figure 1(b).

3. TWO-PHASE HEAT TRANSFER IN MINI AND MICROCHANNELS

3.1 Sub-Cool Liquid Model

This section presents the mathematical model used during the design phase of the TCU. Considering a flow section along a channel in the TCU, by classification of the flow phases, it may be divided into three regions, namely, liquid, liquid-vapour and gas phases. The pressure drop along the sub-cool liquid region is obtained using the following equation:

$$\left(\frac{dP}{dz}\right)_F = \frac{fG^2}{2d}\left[v_f\right] \tag{1}$$

where f is given by the Colebrook Friction Factor given in equation (2).

$$\frac{1}{\sqrt{f}} = -2\log\left[\frac{(\varepsilon/D)}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right] \tag{2}$$

3.2 Homogeneous Two-Phase Flow Model

The following section describes the homogenous two-phase flow model. Defining the dryness fraction as the ratio of the mass fraction of gas to total mass:

$$\text{x} = \text{D}\textit{ryness}\ \text{Fraction} = \frac{\text{M}_\text{g}}{\text{M}} \tag{3}$$

the specific volume of the mixture is given by the following expression:

$$v_m = xv_g + (1-x)v_f \tag{4}$$

The two-phase mixture velocity is thus given by:

$$U_m = \frac{\dot{m}}{A}\left[xv_g + (1-x)v_f\right] = U_f = U_g \tag{5}$$

From the conservation of momentum, the total pressure drop along the channel may be expressed as (Wong and Ooi, 1995 and 1996):

$$-\left(\frac{dP}{dz}\right)_T=\left(\frac{dP}{dz}\right)_G+\left(\frac{dP}{dz}\right)_F+\left(\frac{dP}{dz}\right)_M \tag{6}$$

The homogenous two-phase friction factor can be determined using the Colebrook friction factor given in equation (2) together with the Reynolds number, which the latter is defined as:

$$\mathrm{Re}=\frac{U_m d}{\mu_{tp} v_m} \tag{7}$$

The two-phase viscosity is obtained using the Dukler *et al.*(1964) relationship:

$$\mu_{tp}=\mu_g\beta+\mu_f\left(1-\beta\right) \tag{8}$$

where $\beta=\dfrac{(1-y)v_g}{(1-y)v_g+yv_f}$

From equation (6) it can be observed that the pressure drop along the channel is due to gravity, friction and momentum. However, in a horizontally placed channel, there is no pressure drop due to gravity and hence the following expression can be used (Wong and Ooi, 1995; Wong and Ooi, 1996):

$$\left(\frac{dP}{dz}\right)_T=\frac{-\left[\left(\frac{dP}{dz}\right)_F+C\left(\frac{dx}{dz}\right)\right]}{D} \quad \text{and} \quad \left(\frac{dP}{dz}\right)_F=-\left(\frac{fG^2}{2D}\right)\left(v_m\right) \tag{9}$$

where the changes in the dryness fraction, taking the heat transfer into consideration is given by:

$$\frac{dx}{dz}=\frac{-A\left(\frac{dP}{dz}\right)_F+\frac{D\,\hbar\pi\, d\left(T_w-T\right)}{M}}{BD-AC} \tag{10}$$

in which

$$A=x\frac{dh_g}{dP}+\left(1-x\right)\frac{dh_f}{dP}+G^2v_m\left[x\frac{dv_g}{dP}+\left(1-x\right)\frac{dv_f}{dP}\right] \qquad B=h_{gf}+G^2v_mv_{gf}$$

$$C=G^2v_{gf} \qquad D=1+G^2\left[x\frac{dv_g}{dP}+\left(1-x\right)\frac{dv_f}{dP}\right]$$

$$h_{gf}=h_f-h_g \qquad v_{gf}=v_f-v_g$$

The differential equations (9) and (10) are integrated numerically using the 4th order Runge-Kutta numerical integration technique. As the refrigerant flows, the pressure reduces and the velocity increases. The rise in the velocity is accelerated by both the liquid pressure drop due to flow and also due to evaporation of the liquid refrigerant. The latter effect accelerates the flow velocity significantly as the heat transfer process occurs, and this further improves the heat transfer in the channel as the Reynolds number increases. If the channel is long enough, at a certain point when the refrigerant reaches the local sonic velocity, the flow chokes. This condition occurs when the denominator of equation (9) approaches 0, i.e. when $(dP/dz)_T \rightarrow \infty$. When this happens, further drops in the pressure downstream will not accelerate the flow further.

3.3 Mini and Micro-channel heat transfer

In equation (10), the two-phase heat transfer coefficient $\hbar$ is computed based on a method recommended by Kandlikar and Balasubramanian (2004). Based on the mean free path of molecules in the single phase flow, surface tension effects, and flow patterns in the two-phase flow, Kandlikar and Grande (2003) recommends that for the purpose of the heat transfer applications, the following classifications can be made:

Conventional channels :	$D_h \geq 3$mm
Mini-channels:	$200\ \mu m \leq D_h < 3$mm
Micro-channels:	$10\ \mu m \leq D_h < 200\ \mu m$
Transition micro-channels:	$1\ \mu m \leq D_h < 10\ \mu m$
Transition nano-channels:	$0.1\ \mu m \leq D_h < 1\ \mu m$
Molecular nano-channels:	$D_h < 0.1\ \mu m$

where D_h is the hydraulic diameter of the flow channel. Kandlikar also modified the heat transfer correlations for large diameter tubes that he formulated previously to cater for heat transfer in mini and micro-channels (Kandlikar, 1990[a] and 1990[b]). The correlations for mini and micro-channel (Kandlikar and Balasubramanian, 2004) are shown below:

For laminar and transition flow regions in mini-channels for $1600 < Re_{LO} < 3000$, the following can be used:

$$\hbar = 0.6683\,Co^{-0.2}\left(1-x\right)^{0.8}\hbar_{LO} + 1058.0\,Bo^{0.7}\left(1-x\right)^{0.8}F_{FL}\hbar_{LO} \tag{11}$$

For Re_{LO} <1600, this laminar flow heat transfer coeficient for single phase is obtained from $\hbar_{LO}$. For transition region, the linear interpolation between the turbulent and laminar single phase correlation is recommended. In equation (11), F_{FL} is the fluid surface parameter value which depends on the nucleation site density, fluid properties etc. In flow boiling correlations recommended by Kandlikar (2004), he recommends that F_{FL} =1 for stainless steel tubes for all fluids. The value of $\hbar_{LO}$ is given by Pethukhov and Popov (1963) and Gnielinski (1976), respectively:

$$\hbar_{LO} = \frac{\mathrm{Re}_{LO}\,\mathrm{Pr}_L\left(f/2\right)\left(K_L/D_h\right)}{1+12.7\left(\mathrm{Pr}_L^{\,2/3}-1\right)\left(f/2\right)^{0.5}} \qquad \text{for} \qquad 10^4 \leq \mathrm{Re}_{LO} \leq 5\times10^6 \tag{12}$$

$$\hbar_{LO} = \frac{\left(\mathrm{Re}_{LO}-1000\right)\mathrm{Pr}_L\left(f/2\right)\left(K_L/D_h\right)}{1+12.7\left(\mathrm{Pr}_L^{\,2/3}-1\right)\left(f/2\right)^{0.5}} \qquad \text{for} \qquad 3000 \leq \mathrm{Re}_{LO} \leq 10^4 \tag{13}$$

where f is the friction factor given by:

$$f = \left[1.58\ln\left(\mathrm{Re}_{LO}\right) - 3.28\right]^{-2} \tag{14}$$

For flows at very low Reynolds number in micro-channels, equation (12) is also recommended. The full details on the heat transfer coefficients for the full range of the tube size are given in Kandlikar and Balasubramanian (2004).

4. Experimental Setup & Procedures

Firstly, the TCU is coupled to a vapour compression system to utilize the two-phase heat transfer, in which refrigerant R404A is used as the cooling medium. Next, a heater block (here after refer to as simulator) consisting of cartridge heaters is used to simulate the DUT. The simulator is placed in contact with the TCU as shown in Figure 3, where the cartridge heaters are connected to a control system for regulating the heat supplied to the TCU. The aim is to remove heat effectively from the simulator such that its temperature can be ramped down or maintained at a set-point. In addition, data acquisition equipment has been set up to acquire the necessary data, namely, TC1, T_{set} and Q_{in}. TC1 refers to the temperature at the center of the simulator surface in contact with the TCU; T_{set} refers to the temperature set point; and Q_{in} refers to the heat supplied to the TCU.

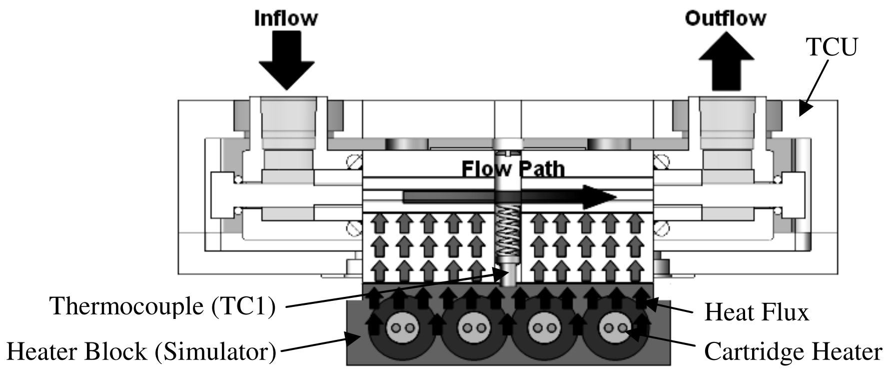

Figure 3: TCU Experimental Setup with External Housing

The experiment is being conducted in two portions. Firstly, under steady state conditions, Q_{in} is being varied from 600 W to 350W at intervals of 50W, with T_{set} being maintained at a steady temperature of 15 °C. Each heat load is maintained for 15 minutes. In the second portion, a Q_{in} of 250W is set and a ramp down of TC1 is effectuated by increasing the refrigerant flow instantaneously within a short period of time. During the entire duration of the experiment, the room temperature is consistently maintained at 22 °C. As the walls of the simulator block are exposed to the surrounding, the heat loss resulted from natural convection is estimated to be about 10 W. It is therefore insignificant as compared to the amount of heat removed by the TCU.

5. RESULTS AND DISCUSSIONS

Figure 4(a) shows the variation of Q_{in}, TC1 and T_{set} with time. The effectiveness of the TCU in removing the various heat loads is clearly shown as the contact surface of the simulator is well maintained at the T_{set} of 15 °C with small fluctuations of within ±1 °C. It is also noticed that during each step-down in Q_{in}, the range of TC1 did not show signs of decrement. This indicates that the cooling capacity has been regulated effectively during the transition to maintain the constant T_{set}. The reason behind this efficient control is the novel vapour compression system discussed in Part II of this paper series (Ooi *et al*., 2008).

Figure 4(b) shows the comparison between theoretical and experimental cooling capacities, in which the former is calculated based on measured volumetric flow rates. During the course of the experiment, as it is observed that the cooling medium exists in two-phase flow at the inlet of the TCU, the inlet dryness fraction is being varied in the possible range of 0.1 to 0.3 during theoretical calculations. From the figure, it is seen that experimental results and theoretical predictions are in reasonably good agreement, where actual cooling capacities fall within the range of the calculated results (shaded).

Figure 5(a) demonstrates the high heat transfer capability of the TCU by showing the temperature ramp-down measured at TC1. A temperature ramp rate of 3.93 °C/s is obtained when the cooling capacity is increased instantaneously from 250 W to 600 W. Although this value is much smaller than the required ramp rate of 12 °C/s listed in Table 1, this is due to the large thermal mass of the simulator used in the test. Figure 5(b) explains this statement by showing the calculated temperature ramp rate using the lumped capacitance method. At a heat load of 250 W, the 600W cooling capacity achieves a theoretical ramp-rate of 4.63 °C/s at the simulator, differing from the experimental value by 0.7 °C/s. The discrepancy is suspected to be due to errors in estimating the thermal mass of the simulator, as well as heat gain from the environment. Nevertheless, the important point is that the experimental ramp rates are in reasonably good agreement with the theoretical values. This indicates that when the actual device such as a lidded microprocessor having a mass of about 22.4 g is used, the desired ramp rate can be achieved. In fact, as shown in the same figure, the required cooling capacity to attain a ramp rate of 12 °C/s is only 485 W. Therefore, a 600 W cooling capacity achieved in the TCU exceeds the given requirements. In general, the TCU has been validated to achieve effective heat transfer from the results shown.

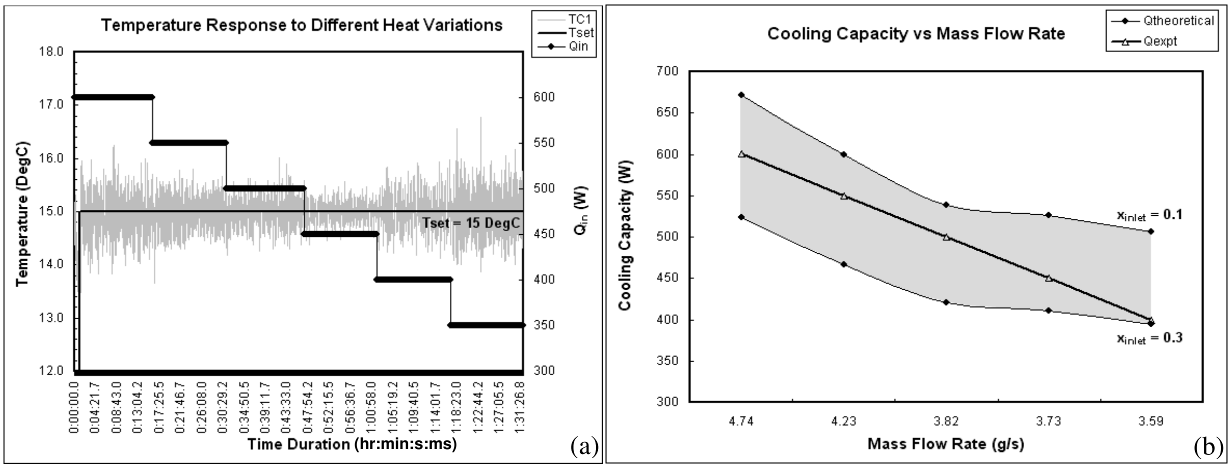

Figure 4: (a) Temperature Response to Different Heat Variations; (b) Theoretical and Experimental Cooling Capacities against Mass Flow Rate

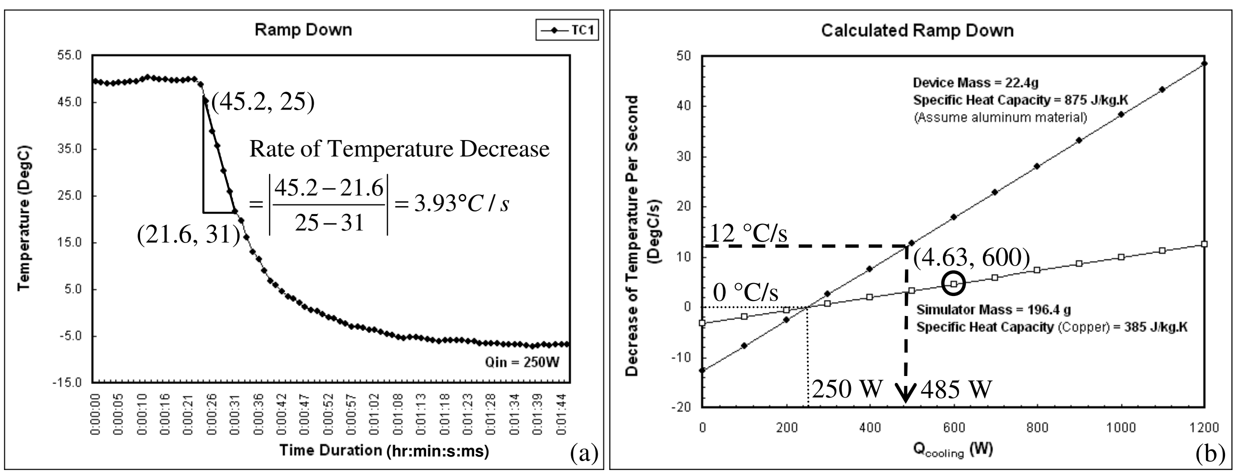

Figure 5: (a) Effects of Ramping Down the Temperature of the Simulator; (b) Calculated Temperature Drop per Second against the Cooling Capacity

6. CONCLUSION

Through theoretical and experimental studies, the new TCU design has been validated to achieve high heat transfer performance comparable to that of micro-channels, while alleviating precision demands through the reduction of the hydraulic diameters by partial blockage of the flow area using the wire inserts. As such, the components of the TCU are easily fabricated by conventional machining which makes it cost effective for industrial application. The TCU when use in conjunction with multi-evaporator system presented in Part II of this paper (Ooi *et al*., 2008) is anticipated to provide an effective and cost-efficient thermal solution for the electronics cooling industry. Last but not least, it is anticipated that the TCU being investigated in the experiments carries the potential to facilitate even higher heat removal rates.

NOMENCLATURE

Symbols

A	Area, m^2
d	Diameter, m
G	Mass Flux, kg/m^2s
h	Specific Enthalpy, J/kg
M	Mass, kg
P	Pressure, bar
T	Temperature, K
$\dot{m}$	Mass Flow Rate, kg/s
U	Velocity, m/s
$\hbar$	Two-phase heat transfer coefficient $W/m^2{\cdot}K$

Subscripts

F	Friction
G	Gravity
M	Momentum
T	Total
g	Gas
f	Liquid
m	Mixture
p	Pressure
w	Wall

Greek

υ	Specific Volume, kg/m^3
ρ	Density, m^3/kg
μ	Viscosity, Ns/m^2
ε	Internal Roughness, mm

REFERENCES

Pfahnl, A.C., Lienhard, J.H.V., Slocum, A.H., 1999, Thermal management and control in testing packaged integrated circuit devices, Proc. 34th Intersociety Energy Conversion Conf., Vancouver BC, Paper No. 1990-01-2723.

Sweetland, M., Leinhard, J.H.V., Active thermal control of distributed parameter systems with application to testing of packaged IC devices. ASME Journal of Heat Transfer, 125(1), 2003.

Tadayon, P., Thermal challenges during microprocessor testing. Intel Technology Journal, Q3, 2000.

Kromann, G.B., 1996, Thermal management of a C4/Ceramic-ball-grid-array: The Motorola PowerPC 603 and PowerPC 604 RISC microprocessors. Proceedings of the Twelfth IEEE SEMI-THERM Symposium:36-42

Malinoski,M., Maveety, J., Knostman, S., Jones, T., 1998, A test site thermal control system for at-speed manufacturing testing. In Proc. IEEE Intl. Test Conf.: 119-128

Wong, T.N., Ooi, K.T. 1995. Refrigerant flow in capillary tube: an assessment of the two-phase viscosity correlations on model prediction. Int'l Comm. in Heat and Mass Transfer, 22(4):595-604

Wong T.N., Ooi K.T., 1996, Adiabatic capillary tube expansion device: A comparison of the homogeneous flow and separated flow models, *Applied Thermal Engineering (United Kingdom)*, 16(17):625 - 634.

Dukler, A.E., Wicks III, M., Cleveland, R.G., 1964, Frictional pressure drop in two-phase flow, A.I.Ch.E. Journal, 10(1): p 38-51

Kandlikar, S.G., Balasubramanian, P., 2004. An extension of the flow boiling correlation to transition, laminar, and deep laminar flows in minichannels and microchannels. Heat Transfer Engineering 25(3):86 – 93.

Kandlikar S.G., Grande W.J. 2003. Evolution of microchannel flow passages-thermohydraulic performance and fabrication technology. Heat Transfer Engineering, 24(1): 3-17

Kandlikar, S. G.1990[a]. A general correlation for two-phase flow boiling heat transfer coefficient inside horizontal and vertical tubes, J. heat transfer, 112: 219-228

Kandlikar, S. G., 1990[b] Model for flow boiling heat transfer in augmented tubes and compact evaporators. *American Society of Mechanical Engineers (Paper)*, 7p

Petukhov, B.S.; Popov, V.N., 1963. Theoretical calculation of heat exchange and frictional resistance in turbulent flow in tubes of an incompressible fluid with variable physical properties**,** *Teplofizika Vysokikh Temperatur (high temperature heat Physics)*, 1(1)

Gnielinski, V., 1976. New equations for heat and mass transfer in turbulent pipe and channel flow, Int'l chemical Engineer, 16: 359-368

Ooi, K.T., Teh,Y.L., Ng, P.F., Phay, T.T., 2008, A Novel Thermal Solution for Electronics Cooling, Part I: Design of a Multi-Evaporator Refrigeration System, Int'l Refrigeration and Air Conditioning Conf at Purdue, No. 2381