VOL. 11, NO. 6, MARCH 2016 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

A THERMOSYPHON STUDY: CORRELATION OBTAINED AS FUNCTION OF HEAT FLUX AND TEMPERATURE DIFFERENCE BETWEEN CHIP AND COOLANT INLET TEMPERATURE

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ABSTRACT

The development of electronics industry in keeping the chip size to be smaller has challenged its thermal management to maintain the maximum chip temperature below 80 °C. The correlation as a method to estimate chip temperature, when only the chip's heat rate or power input and inlet coolant temperature are known, is obtained. The equations involving pressure drop, pool boiling and film condensation characteristics that explain the heat transfer process and flow phenomenon inside the thermosyphon flow are derived to obtain the correlation. A new design of a two-phase thermosyphon water-cooled condenser system as the electronic cooling system has been built to acquire the experimental data. This thermosyphon design uses R-113 as working fluid and water as external coolant condenser (flows through the cooling coil). The condensation occurs as film condensation at the outer of radial water coolant coil. The R-113 liquid condensate will then return back to the evaporator section through the downcomer section by gravity rather than by capillary forces. The thermosyphon system (excluding the water coolant coil section) is thus a passive system, and its evaporator section must be located below the condenser. In the present work, the heat supply component is simulated by a cartridge heater. This cartridge heater is in direct contact with the working fluid. The water coolant condenser is supplied by solution bath. Furthermore, the evaporator, riser, condenser, and downcomer are well insulated and all applied heat is assumed to be transferred through the thermosyphon. In the experiment, the varying mass flow rate and voltage at a constant inlet water temperature ( ) of 10ºC, 15ºC and 20ºC were measured. The correlation obtained as function of heat flux and temperature difference between chip and coolant inlet temperature is valid as they are in good agreement with the experimental results.

Keywords: thermosyphon, chip temperature, electronic cooling system, correlation obtained.

INTRODUCTION

Recent development of semiconductor design has resulted in the need for better thermal management in the electronics industry. Moreover, there is a maximum allowable chip temperature, i.e. 80 °C [1]. Therefore, power dissipation has indeed become a major challenge, and its thermal management, i.e. cooling system, a critical technology in the electronics industry. The development of cooling technology involving thermosyphon flow is amongst other possibilities to solve the problem.

In the present study, an empirical equation is obtained. The objective of finding this empirical equation is to obtain or estimate chip temperature when only the heat rate or power input of chip and inlet coolant temperature are known. This empirical equation is obtained by deriving the equations of nucleate boiling heat transfer from chip surface to refrigerant, film condensation heat transfer from refrigerant vapor to coolant coil surface and heat transfer process from coolant coil surface to water coolant.

METHODOLOGY

A new design of a two-phase thermosyphon water-cooled condenser system has been built. Its schematic drawing is shown in Figure-1. In the experiment, the varying mass flow rate and voltage at a constant inlet water temperature of 10ºC, 15ºC and 20ºC were measured. Meanwhile, the equations that explain heat transfer and flow phenomenon inside the thermosyphon flow are derived.

There are three basic characteristics of heat transfer which can explain the process which occurs in thermosyphon system. There are pressure drop, pool boiling and film condensation characteristics.

Pressure drop characteristics

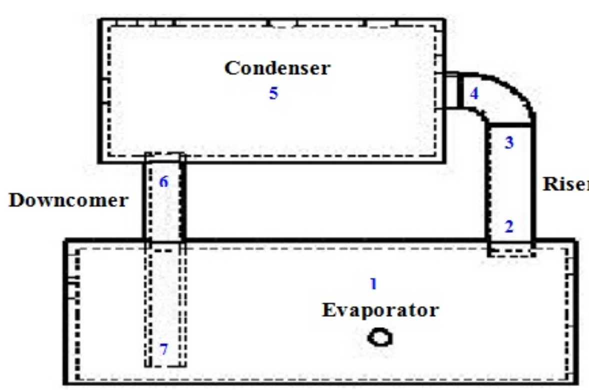

Figure-1. Schematic drawing of thermosyphon system

Assumptions:

1. Steady state.
2. Closed system.

3. Only R-113 vapor enters riser tube (single phase).
4. R-113 vapor is fully condensed; thus only R-113 liquid enters downcomer tube (single phase).

- Pressure drop between evaporator section (1) and riser inlet (2):

Pressure drop between 1 and 2 is a pressure drop due to sudden contraction. According to the revised Bernoulli equation [2]:

$$\frac{P_1}{\rho}+\frac{V_1^2}{2}=\frac{P_2}{\rho}+\frac{V_2^2}{2}+\frac{P_{loss}}{\rho} \tag{1}$$

with $P_{loss}=\frac{V_2^2\rho}{2}\left(\frac{1}{C_c}-1\right)^2$ (2)

This yields the pressure drop between 1 and 2:

$$P_2-P_1=\frac{\rho_l}{2}\left\{V_1^2-\left(V_2^2\left(1+\left[\frac{1}{Cc}-1\right]^2\right)\right)\right\} \tag{3}$$

- Pressure drop between riser inlet (2) and elbow inlet (3):

Pressure drop between 2 and 3 is a pressure drop of R-113 vapor flowing through a straight duct of circular cross section. The fundamental equation is

$$\Delta P=f\,\frac{L}{D}\frac{V^2}{2}\rho \tag{4}$$

Thus the pressure drop between 2 and 3 becomes

$$P_2-(P_3+H\rho_v g)=f_R\,\frac{L_R}{D_{i,R}}\frac{V_2^2}{2}\rho_v$$

$$P_2-P_3=\rho_v\left(\left[f_R\,\frac{L_R}{D_{i,R}}\frac{V_2^2}{2}\right]+Hg\right) \tag{5}$$

With $f_R=\frac{64}{\mathrm{Re}}$ if Re < 2300 and $f_R=\frac{0.316}{\mathrm{Re}^{0.25}}$ when Re > 2300.

- Pressure drop between elbow inlet (3) and elbow outlet/condenser inlet (4):

The pressure drop between these two inlets,

$$P_{loss}=\frac{V^2\rho}{2}(geometry\ factor).$$

Using the revised Bernoulli equation, the pressure drop between 3 and 4 may be determined:

$$P_4-P_3=\left(\frac{\rho_v(V_3^2-V_4^2)}{2}\right)-\left(\frac{V_3^2\rho_v}{2}(geometry\ factor)\right) \tag{6}$$

And since the ratio of radius of curvature to diameter of riser is very small (mitered), thus the geometry factor is 1.3 [2]. Therefore, the pressure drop between 3 and 4 is:

$$P_4-P_3=-\frac{\rho_v}{2}\left(0.3V_3^2+V_4^2\right) \tag{7}$$

- Pressure drop between condenser inlet (4) and condenser section (5):

Pressure drop between 4 and 5 is a pressure drop due to sudden enlargement with pressure loss, $P_{loss}=\frac{V_4^2\rho_v}{2}\left(1-\frac{A_4}{A_5}\right)^2$. Therefore, by implementing the revised Bernoulli equation and pressure loss, the pressure drop between 4 and 5 becomes:

$$P_5-P_4=\frac{\rho_v}{2}\left\{V_4^2\left(1-\left[1-\frac{A_4}{A_5}\right]^2\right)-V_5^2\right\} \tag{8}$$

- Pressure drop between condenser section (5) and downcomer inlet (6):

Pressure drop between 5 and 6 is same as pressure drop between 1 and 2, which is a pressure drop due to sudden contraction. Therefore, pressure drop between 5 and 6 is:

$$P_6-P_5=\frac{\rho_l}{2}\left\{V_5^2-\left(V_6^2\left(1+\left[\frac{1}{Cc}-1\right]^2\right)\right)\right\} \tag{9}$$

- Pressure drop between downcomer inlet (6) and downcomer outlet (7):

Pressure drop between 6 and 7 is same as pressure drop between 2 and 3 which is a pressure drop of R-113 liquid flowing through a straight duct of circular cross section Therefore, the pressure drop between 6 and 7 is:

$$P_6-P_7+H\rho_l g=f_D\,\frac{L_D}{D_{i,D}}\frac{V_D^2}{2}\rho_l$$

$$P_6-P_7=\rho_l\left(\left[f_D\,\frac{L_D}{D_{i,D}}\frac{V_6^2}{2}\right]-Hg\right) \tag{10}$$

- Pressure drop between downcomer outlet (7) and evaporator section (1):

Pressure drop between 7 and 1 is same as pressure drop between 4 and 5 which is a pressure drop due to sudden enlargement. Therefore, the pressure drop between 7 and 1 becomes:

$$P_1-P_7=\frac{\rho_l}{2}\left\{V_7^2\left(1-\left[1-\frac{A_7}{A_1}\right]^2\right)-V_1^2\right\} \tag{11}$$

- The net driving head caused by the different in density between the liquid in downcomer and the vapor/liquid mixture in riser:

$$\Delta P_{riser}=\Delta P_{downcomer}$$

$$(P_2-P_3)=-(P_6-P_7)$$

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$$\dot{m}_r^2 \frac{8}{\pi^2}\left\{\frac{f_r L_r}{D_{i,r}^5 \rho_v} + \frac{f_d L_d}{D_{i,d}^5 \rho_l}\right\} = H\, g(\rho_l - \rho_v)$$

Thus, mass of refrigerant flowing in thermosyphon system can be determined as:

$$\dot{m}_r = \sqrt{\frac{\pi^2 H\, g(\rho_l - \rho_v)}{8\left\{\frac{f_r L_r}{D_{i,r}^5 \rho_v} + \frac{f_d L_d}{D_{i,d}^5 \rho_l}\right\}}} \qquad (12)$$

Pool boiling characteristics

Pool boiling is defined as the process in which the formation of vapor is due to heat added to the liquid by a surface in contact with or submerged within the liquid [3]. Pool nucleate boiling is assumed as boiling phenomenon in the present study, since in nucleate boiling, the bubbles of vapor are formed within the liquid which this vapor bubble plays a key role in this cooling system. Furthermore, saturated boiling is also assumed in the present study rather than subcooled boiling. This assumption is made as in saturated boiling, the bubbles can detach to liquid surface, which then flow to the riser and then condenser. However, some of experimental data are from subcooled boiling. Therefore, the cooling characteristics are not as dominant in this particular case.

Heat transfer correlation of nucleate boiling is defined by Rohsenow's correlation:

a) Nusselt number

$$Nu_L = \frac{h_{nb} L_c}{k_l} = \frac{Ja^2}{C_{NB}^{\ 3}\, \Pr^m}, \qquad (13)$$

Where Jakob number (Ja) is $Ja = \frac{C_{p,l}\left(T_{chip} - T_{sat}\right)}{h_{fg}}$

Prandtl number (Pr) is $\Pr = \frac{C_{p,l}\ \mu_l}{k_l}$

The values of the constants C and m depend on the boiling liquid and the nature of the heating surface. In the present study, the boiling liquid is R-113 and the heating surface is made of copper, thus C = 0.0022 and m = 2.25 [4].

b) Heat transfer rate

$$\dot{Q}_{nb} = A_{chip}\, h_{nb}\left(T_{chip} - T_{sat}\right) \qquad (14)$$

Equation (13) and (14) follows that:

$$\dot{Q}_{nb} = \left[\frac{A_{chip}\, k_l\, C_{p,l}^{\ 2}}{L_c\, C^3\, \Pr_l^m\, h_{fg}^{\ 2}}\right]\left(T_{chip} - T_{sat}\right)^3 \qquad (15)$$

c) Characteristic length

Characteristic length is considered to be proportional with the radius of the bubbles at the time it breaks away from the heating surface. The estimation as follows

$$L_c = R_b = \left[\frac{\sigma_t}{\left(\rho_l - \rho_v\right)\, g}\right]^{1/2} \qquad (16)$$

Film condensation characteristics

Steady-state laminar film condensation on radial system is assumed as a condensation phenomenon between vapor and outer surface of coolant coil (Figure-2).

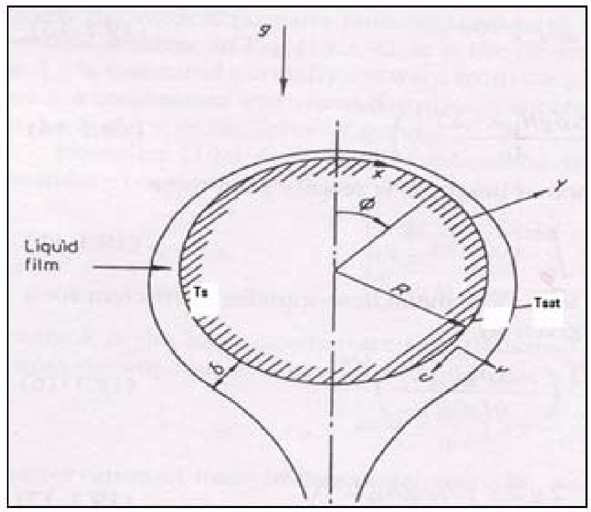

Figure-2. Schematic of film condensation in radial system [4].

Assumptions
1. The coolant coil surface is maintained at a constant temperature (Ts).
2. The saturation temperature of the quiescent vapor in which the surface is placed is Tsat.

Fluid mechanics consideration

Consider the schematic and fluid element shown.

Assumption:
- Neglect inertia force
- Steady flow
- Viscous force and buoyancy force is balance.

Hence, the equilibrium equation of the fluid element:

$$L_{cc}\delta x \delta y \rho_l g + PL_{cc} - (P + \delta P)L_{cc}\delta y - \tau_{xy}L_{cc}\delta x + \left(\tau_{xy} + \left[\frac{\partial \tau_{xy}}{\partial y}\right]\delta y\right)L_{cc}\delta x = 0 \qquad (17)$$

For the quiescent vapor: $\delta P = \delta x \rho_v g$

For a Newtonian fluid: $\tau_{xy} = \mu_l \frac{\partial u}{\partial y}$

Thus, by integrating for the velocity, the solution gives the velocity distribution as:

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$$u(y) = \frac{g(\rho_l - \rho_v)\sin\theta}{\mu_l}\left[\delta y - \frac{y^2}{2}\right] + C \tag{18}$$

a) Boundary condition

At the coolant coil surface: u(0) = 0, thus C = 0. Hence, the velocity distribution becomes:

$$u(y) = \frac{g(\rho_l - \rho_v)\sin\theta}{\mu_l}\left[\delta y - \frac{y^2}{2}\right] \tag{19}$$

b) Mass flow rate of condensate liquid

The liquid flow rate at a section of the film at a half side is given by:

$$\Gamma\Big|_{\frac{1}{2}side} = \int_0^{\delta} L_{cc}\rho_l u dy \tag{20}$$

Substituting u(y) and then integrate, thus:

$$\Gamma\Big|_{\frac{1}{2}side} = \frac{L_{cc} g(\rho_l - \rho_v)\rho_l \sin\theta}{\mu_l}\left[\frac{\delta^3}{3}\right] \tag{21}$$

And thus, $\delta = \dfrac{L_{cc}^{1/3}\Gamma_{1/2}^{1/3}\ \mu_l^{1/3}}{\left[\rho_l(\rho_l - \rho_v)g\sin\theta\right]^{1/3}}$ (22)

c) Rate of condensation

Rate of condensation for half side is defined as:

$$\dot{m}_c = \frac{\partial\Gamma_{\frac{1}{2}}}{\partial x}, \text{ where } x = R\theta \tag{23}$$

d) Film condensation characteristic flow

The characteristic velocity to denote the flow of condensate film is:

$u_m = \dfrac{\Gamma}{(\rho_l \delta L_{cc})}$ and $D_h = \dfrac{4A}{p} = \dfrac{4(\delta \times L_{cc})}{1}$, thus the Reynolds number is given by $\mathrm{Re}_\delta = \dfrac{u_m D_h \rho_l}{\mu_l} = \dfrac{4\Gamma}{\mu_l}$ (24)

Therefore, the flow characteristic of film condensation is as follow:

- $\mathrm{Re}_\delta < 30$ is laminar flow
- $30 < \mathrm{Re}_\delta < 180$ is wavy flow
- $\mathrm{Re}_\delta > 1800$ is turbulent flow

Heat transfer consideration

Assumptions:

- Velocity is small, so that convective energy transfer can be neglected
- Temperature gradient in the θ direction is also small

Energy balance for the small fluid element:

$$qdx = \left[q + \frac{\partial q}{\partial y}dy\right]dx \tag{25}$$

Hence, heat flux at the coolant coil surface is:

$$\dot{Q}_s = -\left(\frac{k_l}{\delta}\right)(T_s - T_{sat})p_2 \tag{26}$$

Heat transfer coefficient is defined as:

$$\dot{Q}_s = \bar{h}_c p_2 (T_s - T_{sat}) \tag{27}$$

From (26) and (27):

$$h_c = \left(\frac{k_l}{\delta}\right) \tag{28}$$

is the expression for the laminar condensation heat transfer coefficient.

The energy balance:

$$\Gamma h_f + h_g \dot{m}_c dx = (\Gamma + d\Gamma)h_f + \dot{Q}_s dx \tag{29}$$

Using relation (23) and from (26):

$$\Gamma_{\frac{1}{2}} = 1.1438\left[\frac{g\rho_l(\rho_l - \rho_v)}{\mu_l^{\ 3}}\frac{k_l^{\ 3}(T_{sat} - T_s)^3 R^3}{h_{fg}^{\ 3}}\right]^{\frac{1}{4}} \tag{30}$$

a) Average heat transfer coefficient and Nusselt number

Hence, for a tube where the condensate flows on both sides, and representing the heat transfer rate by the heat transfer coefficient:

$\dot{Q}_s = 2\Gamma_{\frac{1}{2}} h_{fg} = 2\pi R\bar{h}_c(T_{sat} - T_s)$, thus by substituting (30), the average film condensation heat transfer coefficient is:

$$\bar{h}_c = 0.728\left[\frac{g\rho_l(\rho_l - \rho_v)}{\mu_l(T_{sat} - T_s)D}\frac{k_l^{\ 3}h_{fg}}{}\right]^{\frac{1}{4}} \tag{31}$$

Therefore, the average Nusselt number for a tube of diameter 2R becomes:

$$\overline{Nu} = \frac{\bar{h}_c(2R)}{k_l} = 0.728\left[\frac{g\rho_l(\rho_l - \rho_v)}{\mu_l k_l(T_{sat} - T_s)}\frac{h_{fg}D^3}{}\right]^{\frac{1}{4}} \tag{32}$$

And heat transfer rate:

$$\dot{Q}_s = 0.728\left[\frac{g\rho_l(\rho_l - \rho_v)}{\mu_l(T_{sat} - T_s)D}\frac{k_l^{\ 3}h_{fg}}{}\right]^{\frac{1}{4}} p_2(T_s - T_{sat}) \tag{33}$$

b) Modified latent heat

The work by Rohsenow showed from experiments that the modified latent heat for condensation is given by:

$$h''_{fg} = h_{fg}\left[1 + 0.68Ja\right] \tag{34}$$

Thus, the latent heat which is used in equation (31), (32) and (33) should be as modified latent heat, hence average heat transfer coefficient, Nusselt number and heat transfer rate becomes:

$$\bar{h}_c = 0.728\left[\frac{g\rho_l(\rho_l - \rho_v)}{\mu_l(T_{sat} - T_s)D}\frac{k_l{}^3 h''_{fg}}{}\right]^{1/4} \tag{35}$$

$$\overline{Nu} = \frac{\bar{h}_c(2R)}{k_l} = 0.728\left[\frac{g\rho_l(\rho_l - \rho_v)}{\mu_l k_l(T_{sat} - T_s)}\frac{h''_{fg}D^3}{}\right]^{1/4} \tag{36}$$

$$\dot{Q}_s = 0.728\left[\frac{g\rho_l(\rho_l - \rho_v)}{\mu_l(T_{sat} - T_s)D}\frac{k_l{}^3 h'_{fg}}{}\right]^{1/4} p_2(T_s - T_{sat}) \tag{37}$$

When using the above expression, all liquid properties are evaluated at the mean temperature between coolant coil surface and the saturated vapor (T_f). Meanwhile, the vapor properties are evaluated at saturation vapor temperature (Tsat).

c) Heat transfer at coolant coil

Heat rate of water flowing in coolant coil is also meant heat dissipation.

$$d\dot{Q}_d = d\dot{Q}_w = \dot{m}_w C_{p,w} dT_w$$

$$\dot{Q}_d = \dot{Q}_w = \dot{m}_w C_{p,w}\int_{T_{wi}}^{T_{wo}} dT$$

$$\dot{Q}_d = \dot{Q}_w = \dot{m}_w C_{p,w}(T_{wo} - T_{wi}) \tag{38}$$

Nusselt number for water coolant inside the tube is given by:

$$Nu_D = 0.023\mathrm{Re}_D^{0.8}\,\mathrm{Pr}^{0.4},$$

Furthermore, $Nu = \dfrac{h_w D_i}{k_w}$, thus heat transfer coefficient for water coolant flow inside the tube is:

$$h_w = \frac{k_w(0.023\mathrm{Re}_D^{0.8}\,\mathrm{Pr}^{0.4})}{D_i} \tag{39}$$

d) Overall heat transfer coefficient

Heat transfer coefficient from R-113 vapor to water coolant is determined by overall heat transfer coefficient through the following equation:

$$U = \frac{Di \times h_w \times h_c}{(Do \times \bar{h}_c) + (Di \times h_w)} \tag{40}$$

Furthermore, heat transfer from coolant coil surface to water coolant is determined by:

$$\dot{Q}_w = p_1 h_w(T_s - T_w)\text{, hence} \tag{41}$$

It is necessary to define Tw which is mean temperature of water coolant (Tm), thus the heat transfer becomes:

$$\dot{Q}_w = p_1 h_w(T_s - T_m) \tag{42}$$

By deriving Equations of Rohsenow's nucleate boiling, heat transfer at coolant coil and heat transfer coefficient from R-113 vapor to water coolant, it was found that different temperatures between chip and inlet coolant water is a function of its heat transfer rate as following: assuming $\alpha = \left[\dfrac{A_{chip}\ k_l\ C_{p,l}{}^2}{L_c\ C^3\ \mathrm{Pr}_l{}^m\ h_{fg}{}^2}\right]$, Rohsenow's nucleate boiling becomes:

$$\dot{Q}_{nb} = \alpha\left(T_{chip} - T_{sat}\right)^3 \tag{43}$$

assuming $\beta = 0.728\left[\dfrac{g\rho_l(\rho_l - \rho_v)}{\mu_l D}\dfrac{k_l{}^3 h'_{fg}}{}\right]^{1/4} P_2$,

heat transfer at coolant coil equation becomes:

$$\dot{Q}_s = \beta(T_s - T_{sat})^{3/4} \tag{44}$$

since $T_m = \dfrac{T_{wi} + T_{wo}}{2}$, and by substituting heat transfer at coolant coil equation to T_m, equation of heat transfer coefficient from R-113 vapor to water coolant can be written as:

$$(T_s - T_{wi}) = \dot{Q}_w\left(\frac{1}{P_1 h_w} + \frac{1}{2\dot{m}_w C_{p,w}}\right) \tag{45}$$

At steady state, $\dot{Q}_{nb} = \dot{Q}_s = \dot{Q}_w$; thus by adding Equations (43), (44) and (45), we obtain:

$$(T_{chip} - T_{wi}) = \dot{Q}\left(\frac{1}{P_1 h_w} + \frac{1}{2\dot{m}_w C_{p,w}}\right) + \left[\frac{\dot{Q}}{\beta}\right]^{4/3} + \left[\frac{\dot{Q}}{\alpha}\right]^{1/3} \tag{46}$$

Hence, the empirical equation between chip and coolant inlet temperature difference and heat flux becomes:

$$(T_{chip} - T_{wi}) = a\dot{Q} + b\dot{Q}^{4/3} + c\dot{Q}^{1/3} \tag{47}$$

RESULTS AND DISCUSSION

The correlation obtained as functions of chip and coolant inlet temperature difference and heat flux in the current study is presented as in Equation.47:

$$(T_{chip} - T_{wi}) = a\dot{Q} + b\dot{Q}^{4/3} + c\dot{Q}^{1/3}$$

VOL. 11, NO. 6, MARCH 2016 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

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where:

T_{wi}	a	b	c
10 °C	-20.6	7.5	53.8
15 °C	-38.43	16.52	60.6
20 °C	-56.33	27.79	64.31

Figures-3,-4, and -5 show that the empirical equations are valid as they are in good agreement with the experimental results.

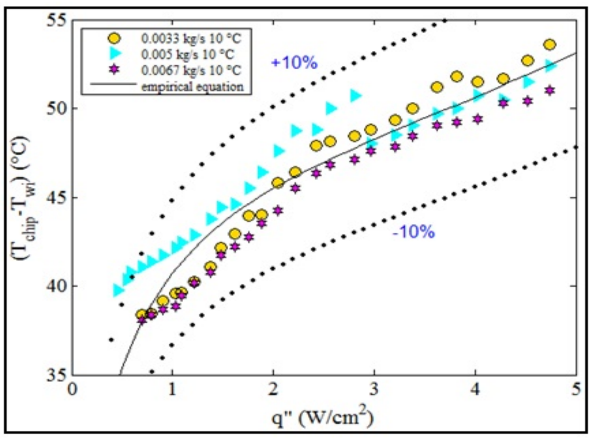

Figure-3. Correlation between chip and inlet water temperature difference and heat flux at constant of 10 °C.

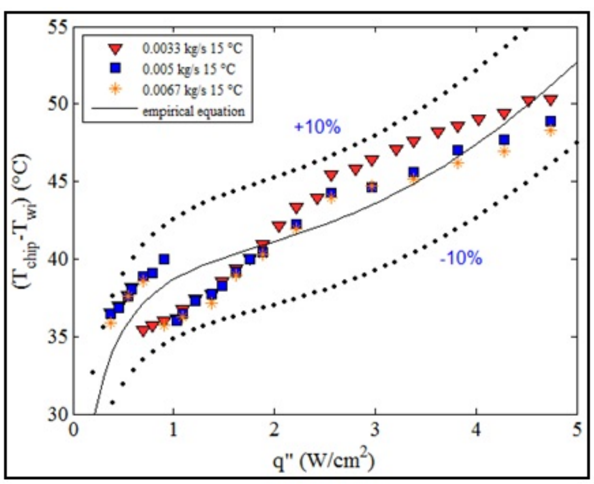

Figure-4. Correlation between chip and inlet water temperature difference and heat flux at constant T_{wi} of 15 °C.

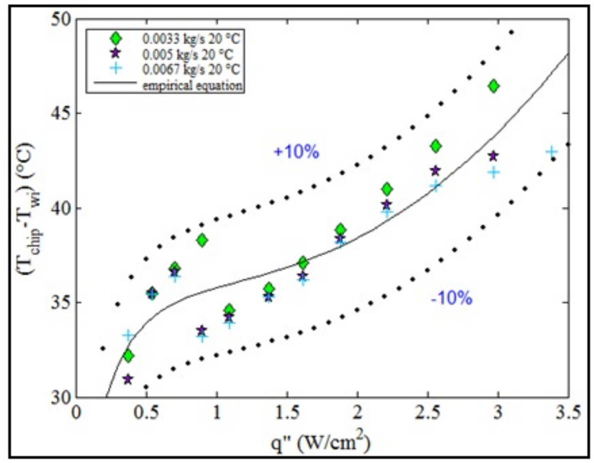

Figure-5. Correlation between chip and inlet water temperature difference and heat flux at constant of 20 °C.

CONCLUSIONS

These empirical equations are the first known method to estimate chip temperature when only the heat rate or power input of chip and inlet coolant temperature are known. This preliminary study shows a valid result as they are in good agreement with the experimental results.

However, these equations are only applicable to the T_{wi} range of 10 °C to 20 °C. Future research with various T_{wi} range of 25 °C to 40 °C is required to capture the general trend of a broader range of T_{wi}.

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