Influence of an Aspect Ratio of Rectangular Channel on the Cooling Performance of a Multichip Module

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Experiments were performed by using PF-5060 and water to investigate the influence of an aspect ratio of a horizontal rectangular channel on the cooling characteristics from an in-line 6×1 array of discrete heat sources which were flush mounted on the top wall of the channel. The experimental parameters were aspect ratio of rectangular channel, heat flux of simulated VLSI chip, and channel Reynolds number. The chip surface temperatures decreased with the aspect ratio at the first and sixth rows, and decreased more rapidly at a high heat flux than at a low heat flux. The measured friction factors at each aspect ratio for both water and PF-5060 gave a good agreement with the values predicted by the modified Blasius equation within $\pm7\%$. The Nusselt number increased as the aspect ratio decreased, but the increasing rate of Nusselt number reduced as the aspect ratio decreased. A 5:1 rectangular channel yields the most efficient cooling performance when the heat transfer and pressure drop in the test section were considered simultaneously.

Key Words : Aspect Ratio, Rectangular Channel, Discrete Heat Source, Friction Factor, Nusselt Number

Nomenclature

A : Surface area (m^2)
D_h : Hydraulic diameter (m)
f : Fanning friction factor
h : Heat transfer coefficient ($W/m^2 \cdot K$)
H : Channel height (cm)
k : Thermal conductivity ($W/m \cdot K$)
L_h : Chip length (m)
Nu_{Lh} : Nusselt number ($=h \cdot L_h/k_f$)
q'' : Heat flux (W/cm^2)
R : Resistance (Ohm)
Re_{Dh} : Reynolds number ($=\rho_f \cdot U \cdot D_h/\mu_f$)
T : Temperature (℃)
U : Average velocity (m/s)
V : Voltage (V)
W : Channel width (cm)

Greek letters

α^* : Aspect ratio of rectangular channel ($=H/W$)
δ : Uncertainty
μ : Viscosity ($N \cdot s/m^2$)
ρ : Density (kg/m^3)

Subscripts

f : Working fluid
s : Chip surface

1. Introduction

Thermal characteristics of turbulent flow through a noncircular duct is of special interest because of the application for the thermal management of computer and compact heat exchangers. The rectangular channel, because of its technical importance, has received more attention than the other noncircular geometries. For electronics, the power dissipation per single chip has increased dramatically in recent years (Ramaswamy et al., 1998; Pautsch and Bar-

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Cohen, 1999). It has provided thermal engineers with challenge of managing the increased thermal budget in order to maintain the chip at a safe operating temperature. For high-heat-flux application, forced convection cooling using liquid as a coolant must be used. In this case, discrete heat sources are embedded at one wall of a rectangular channel and the heat fluxes generated by the heat sources are dissipated by coolants. Numerous theoretical and experimental results have been obtained for laminar and turbulent flow in a continuously heated channel (Sparrow et al., 1966; Tan and Charters, 1970; Kostic and Hartnett, 1986), and there have been a few data on the forced-convection heat transfer from discrete heat sources attached to the walls of a channel (Incropera et al., 1986; Garimella and Eibeck, 1990; Heindel et al., 1992; Choi and Cho, 1999).

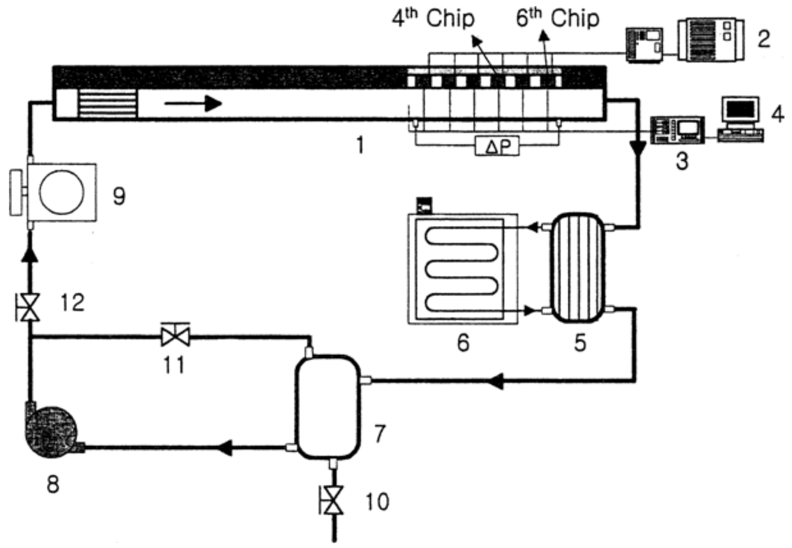
In many practical applications involving a rectangular channel, the aspect ratio is one of the most important parameter because the heat transfer characteristics in a rectangular channel depend on the aspect ratio. However, no data on the effect of this parameter have been reported under the fully developed turbulent flow condition. This problem is pertinent to the design of cooling channels of electronic systems. Consequently, the practicing engineer has limited information on the design criteria.

Therefore, the objective of present study is to investigate the influence of the aspect ratio of rectangular channel on the cooling performance of a linear array of square heat sources with high heat flux which plays the role of a multichip module, cooled by indirect liquid cooling method.

2. Experimental Apparatus

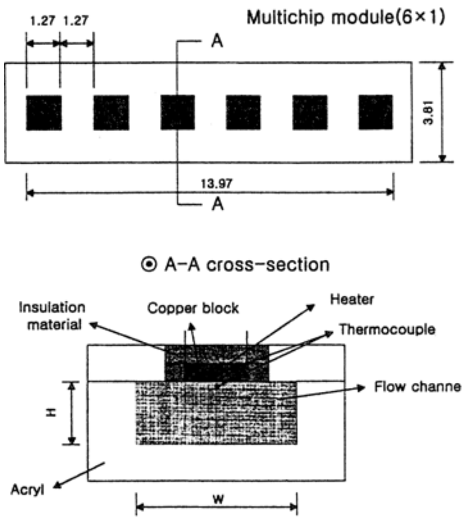
The schematic diagram of the experimental apparatus is shown in Fig. 1. The apparatus consisted of a main test section, a constant temperature bath, a power supply, a mass flow meter, a pump, a data acquisition system etc.

The test section consisted of a rectangular channel and a multichip module as shown in Fig.



1. Rectangular channel
2. DC power supply
3. Data acquisition system
4. Personal computer
5. Plate-type heat exchanger
6. Constant temp. bath
7. Receiver tank
8. Pump
9. Mass flow meter
10 ~ 12. Valves

Fig. 1 Schematic diagram of the present experimental apparatus



$\alpha^*=H/W$ (Unit: cm)

α^*	0.1	0.2	0.5
H	0.38	0.76	1.90
W	3.81	3.81	3.81

Fig. 2 Details of the present test section

2. The multichip module had an in-line 6×1 array of discrete heat sources simulating VLSI chips, and it was flush-mounted on the top wall of a horizontal rectangular channel. The multichip module was located at a downstream location 50 times of the hydraulic diameter from the inlet of the test section, where flow was hydrodynamically fully developed. A honeycomb section was located at the inlet of the channel to make flow uniform. Two static pressure taps were located at the bottom wall just before and after the test section and connected to an U-tube manometer to measure the pressure drop across the test section. A heating wire with a resistance of $18.6\pm0.2\Omega$ as a heat source was attached to each chip by silicon glue. Six heating wires on the chips were connected in parallel to a power supply within accuracy of ±0.03V to supply uniform heat flux to each heater. Each heat source had a square shape with a size of 1.27cm, a typical size for a VLSI chip. The distance between two adjacent chips was 1.27cm. A copper heat sink with a thickness of 6mm was attached at the bottom of each chip, and the top side was thermally insulated by fiberglass to force the heat flow only to the channel. The copper heat sink was a square shape of the same size as the heater. The maximum temperature difference at three positions on a copper heat sink was approximately 0.5℃. The temperatures of the copper heat sink and the fluid temperatures at 0.5mm below the copper heat sink surface were measured by type T thermocouples inserted through machined holes. The thermocouples were calibrated within accuracy of ±0.15℃ by a standard RTD.

A conduction analysis was performed on the heat source and the substrate using the values of thermal conductivities for copper block (386W/m · K), acryl (0.2W/m · K), thermal silicon (1.6W/m · K), and insulation (0.02W/m · K). The heat loss from the heater to a surrounding ranged from 2.3 to 3.8% within the whole experimental range and was compensated in the calculation of heat flux.

Three key experimental parameters were the aspect ratio of rectangular channel of 0.1, 0.2, 0.5, the heat flux of 10, 20, 30, 40 W/cm^2 and the channel Reynolds number ranged from 3,000 to 15,000. The hydraulic diameters of each rectangular channel were 2.54, 1.27, 0.69cm. The heat flux was obtained by dividing the heat supplied to each chip by the chip area. The Reynolds number was calculated by using the hydraulic diameter of the rectangular channel. The inlet temperature of the test section was set to 15 ± 0.15℃ for all runs. The heat transfer fluids used in the present study were water and PF-5060 (a dielectric liquid manufactured by 3M Co.). PF-5060 was an alternative coolant of FC-72, and its typical properties were shown and compared with FC-72 and water in Table 1.

The local heat transfer coefficient was calculated by dividing the heat flux supplied to each chip by the temperature difference between the surface temperature of a chip and the fluid temperature measured at a location 0.5mm below the chip as shown in Eq. (1). The heat flux was obtained using the supplied voltage and the resistance of heater as shown in Eq. (1).

$$h=\frac{q''}{(T_s-T_f)}=\frac{V^2}{RA_s}/(T_s-T_f) \qquad (1)$$

The determination of a bulk temperature in the

Table 1 Properties of the Fluorinert liquids and water (at 25℃ and 1atm)

Property	PF-5060	FC-72	water
Boiling point(℃)	56	56	100
Density(Liquid) (g/cm³)	1.68	1.68	1.0
Viscosity(Liquid) (cs)	0.4	0.4	0.9
Surface tension (dyne/cm)	12.0	12.0	72
Vapor pressure (kPa)	30	30	3.3
Heat of vaporization (cal/g)	21	21	540
Specific heat (cal/(g・℃))	0.23	0.25	1.0

rectangular channel required 2-dimensional temperature and velocity profiles. This made it difficult to determine the bulk temperature. Thus, the convective heat transfer coefficients were obtained by using the fluid temperature just below a heat sink instead of the bulk temperature.

The equations for the uncertainty analysis of the local heat transfer coefficient and the heat flux are shown in Eqs. (2)～(3) by following procedure in the literature (Moffat, 1985).

$$\frac{\delta h}{h}=\sqrt{\left(\frac{\delta q''}{q''}\right)^2+\left(\frac{\delta T_s}{T_s-T_f}\right)^2+\left(\frac{\delta T_f}{T_s-T_f}\right)^2} \quad (2)$$

$$\frac{\delta q''}{q''}=\sqrt{\left(\frac{\delta R}{R}\right)^2+\left(2\frac{\delta V}{V}\right)^2+\left(2\frac{\delta L_h}{L_h}\right)^2} \quad (3)$$

The uncertainties of T_s, T_f, resistance, voltage, and the heater length (L_h) were 0.65℃, 0.15℃, 0.2Ω, 0.03V and 0.02mm, respectively for the whole experimental range. The uncertainty of the heat flux was ±1.19% for both PF-5060 and water, and the uncertainties of the local heat transfer coefficient were ±4.1% for PF-5060 and ±3.7% for water.

The friction factor in the test section was obtained by using the measured pressure drop, and was defined as

$$f_{Fanning}=\frac{\Delta P D_h}{2L_{test\ section}\rho U_i^2} \quad (4)$$

and the range of uncertainty of the friction factor was within 2% for Reynolds number above 3,000.

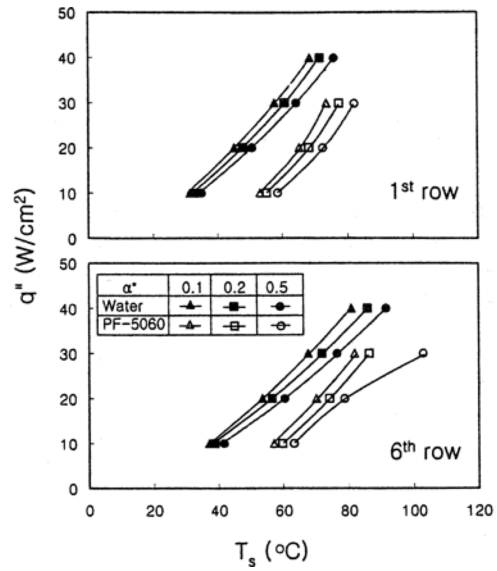


Fig. 3 Removable heat flux with respect to chip surface temperatures (Re_{Dh}=15,000)

3. Results and Discussion

Figure 3 shows the heat flux ranges that can be removed from the chips with respect to the chip surface temperatures at the first and sixth rows of the multichip module at a channel Reynolds number of 15,000. The chip surface temperatures for water were lower than those for PF-5060 at the same heat flux since the specific heat and thermal conductivity for PF-5060 were approximately 25% and 10.6% of those for water. The chip surface temperatures at the same heat flux decreased as the aspect ratio decreased at the first and sixth rows, and the surface temperature decreased more rapidly at a high heat flux than at a low heat flux. For water, the chip surface temperatures for 10:1 rectangular channel were lower than those for 2:1 rectangular channel by 4.6～7.3℃ at the first row, and by 4.8～10.6℃ at the sixth row. For PF-5060, the chip surface temperatures for 10:1 rectangular channel were lower than those for 2:1 rectangular channel by 5.5～8.6℃ at the first row, and by 6.1～21.1℃ at the sixth row. For 2:1 rectangular channel, the surface temperature at the sixth row abruptly increased over 100℃ when the heat flux was over 30W/cm². This meant that the local dryout condition occurred at the sixth row. The dryout condition puts a limit on the allowable amount of evaporation in a rectangular channel. It is extremely important in the design of cooling channels for electronics cooled by forced convection boiling.

Figure 4 shows the local heat transfer coefficients with respect to the row number at a channel Reynolds number of 15,000 and the heat fluxes of 10 and 30W/cm². The local heat transfer coefficients for water reached an uniform value approximately after the fourth row (seven times of the chip length). This means that the thermally fully developed condition was reached after the fourth row. The local heat transfer coefficients for 10:1 rectangular channel were larger than those for 2:1 and 5:1 rectangular channels. The reason is that the average flow velocity for 10:1 rectangular channel showed the largest among three different channels at the same Reynolds number due to the smallest hydraulic diameter. For water, the influence of the aspect ratio on the local heat transfer was more severe than for PF-5060 at a heat flux of 10W/cm². Therefore, the difference of the local heat transfer coefficients between water and PF-5060 at each aspect ratio increased as the aspect ratio decreased. But at a heat flux of 30W/cm², the local heat transfer coefficient for PF-5060 was affected by aspect ratio as much as water. Especially, the local heat transfer coefficient for 2:1 rectangular channel abruptly decreased after the third row due to the local dryout at the heating surface. At a heat flux of 30W/cm², the difference of the local heat transfer coefficients between water and PF-5060 at each aspect ratio decreased as the aspect ratio increased except the 2:1 rectangular channel. The reason is that a subcooled nucleate boiling near a heating surface made the local heat transfer coefficient for PF-5060 increase for 5:1 and 10:1 rectangular channels. However, for a 2:1 rectangular channel case, after the third row, the liquid deficient region occurred at the heating surface and the cooling effect by evaporation could not be expected anymore.

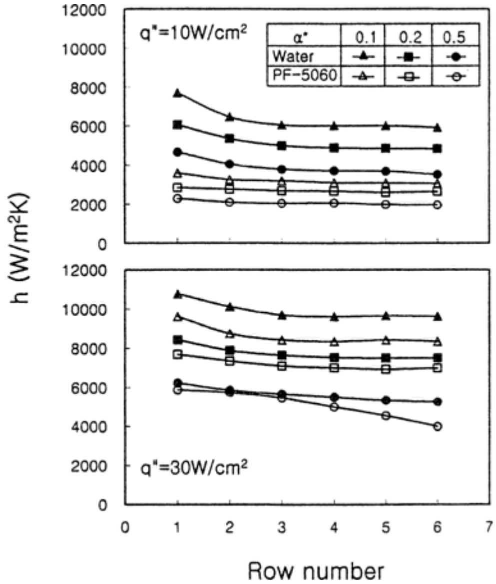


Fig. 4 Local heat transfer coefficient with respect to row number ($Re_{Dh}=15{,}000$)

Figure 5 shows the apparent Fanning friction factor in the test section as a function of Reynolds number.

The Blasius equation which can be used to predict the fully developed turbulent friction factor in smooth circular ducts was introduced by Jones (1976) for rectangular ducts. The modified Blasius equation is defined as

$$f=0.079\,(Re^{*}_{Dh})^{-0.25} \tag{5}$$

The Kozicki Reynolds number, Re^{*}_{Dh}, in Eq. (5) is given by the following expression

$$Re^{*}_{Dh}=a\,(\rho U D_h/\mu) \tag{6}$$

where the value of a in Eq. (6) depends on the aspect ratio, as shown in Table 2.

The measured friction factors for water and PF-5060 increased as the aspect ratio of rectangular channel decreased due to the difference of the mean flow velocity at each aspect ratio, and gave

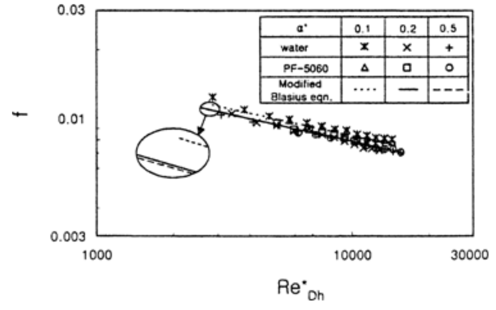


Fig. 5 Friction factor for water and PF-5060 in the present test section

Table 2 Values of a in Eq. (6)

Aspect ratio	1	0.75	0.50	0.23	0
a	1.125	1.106	1.029	0.878	0.667

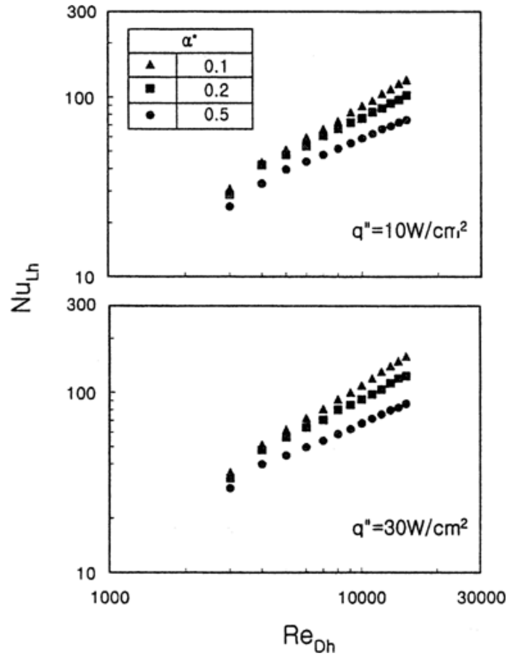


Fig. 6 Nusselt number with respect to Reynolds number for water

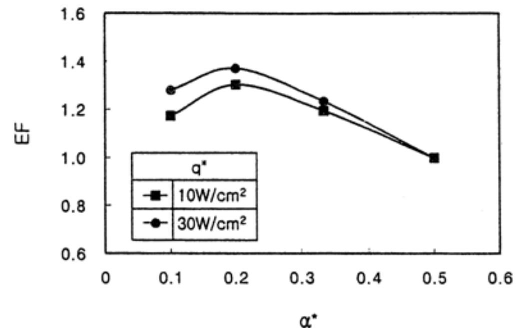


Fig. 7 Enhancement factor at the 6th row with respect to the aspect ratio of rectangular channel ($Re_{Dh}=15{,}000$)

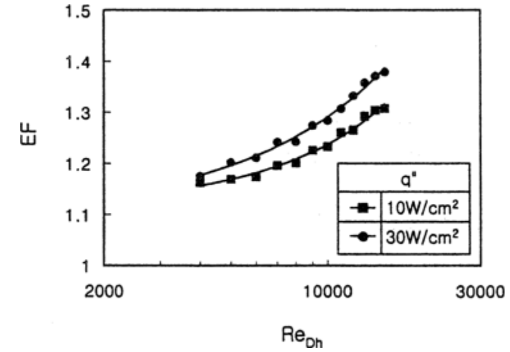


Fig. 8 Enhancement factor of a 5:1 rectangular channel with respect to Reynolds number at the 6th row

good agreement with the values predicted by the modified Blasius equation within $\pm 7\%$.

To investigate the influence of the aspect ratio of a rectangular channel on the Nusselt number for water, the Nusselt number data at the sixth row for the heat fluxes of 10 and 30W/cm^2 are shown in Fig. 6. Nusselt number using the chip length is defined as shown in Eq. (7)

$$Nu_{L_h}=\frac{hL_h}{k_f} \tag{7}$$

Nusselt number increased as the aspect ratio decreased, but the increasing rate of Nusselt number diminished as the aspect ratio decreased. At a heat flux of 30W/cm^2, the local Nusselt numbers for 5:1 rectangular channel were larger by 13~45% than those for 2:1, and the local Nusselt numbers for 10:1 rectangular channel were larger by 7~28% than those for 5:1.

Figure 7 shows the enhancement factors (EFs) obtained by using the Nusselt number and the friction factor at a Reynolds number of 15,000. Note that the EF is defined as

$$EF=\frac{Nu_{L_h}/Nu_{L_h,2:1}}{f/f_{2:1}} \tag{8}$$

where $Nu_{2:1}$ and $f_{2:1}$ are the Nusselt number and the friction factor for 2:1 rectangular channel. In order to obtain the reasonable trend for the relation between the EFs and the aspect ratio, the data of 3:1 rectangular channel were included. For heat fluxes of 10 and 30W/cm^2, the EFs for 5:1 rectangular channel showed the largest value, and the values were 1.30 at a heat flux of 10W/cm^2 and 1.37 at 30W/cm^2. This means that the 5:1 rectangular channel yields the most efficient cooling performance when the heat transfer and pressure drop in the test section were considered simultaneously.

Figure 8 shows the EFs for a 5:1 rectangular channel which showed the most efficient cooling performance as a function of Reynolds number at the sixth row. The EFs for each heat flux increased with the channel Reynolds number. The influence of the channel Reynolds number on the EFs at a heat flux of $30W/cm^2$ was greater than that at a heat flux of $10W/cm^2$. The reason is that at high heat flux, the increasing rate of the heat transfer coefficient is comparable to that of the pressure drop as the channel Reynolds number increases.

4. Conclusions

A summary of the present study is given below.

(1) The chip surface temperatures decreased with the aspect ratio at the first and sixth rows, and they decreased more rapidly at a high heat flux than at a low heat flux.

(2) Thermally fully developed conditions were observed after the fourth row at each aspect ratio. The difference of the local heat transfer coefficients between water and PF-5060 at each aspect ratio increased as the aspect ratio of rectangular channel decreased at a heat flux of $10W/cm^2$. But at a heat flux of $30W/cm^2$, the difference decreased as the aspect ratio increased.

(3) The measured friction factors at each aspect ratio for both water and PF-5060 gave good agreement with the values predicted by the modified Blasius equation within $\pm 7\%$.

(4) The Nusselt number data increased as the aspect ratio decreased, but the increasing rate of Nusselt number data reduced as the aspect ratio decreased.

(5) A 5:1 rectangular channel yields the most efficient cooling performance when the heat transfer and pressure drop in the test section were considered simultaneously.

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References

Choi, M. and Cho, K., 1999, "Thermal Characteristics of a Multichip Module Using PF-5060 and Water," *KSME Int. J.*, Vol. 13, No. 5, pp. 443~450.

Garimella, S. V. and Eibeck, P. A., 1990, "Heat Transfer Characteristics of an Array of Protruding Elements in Single Phase Forced Convection," *Int. J. Heat Mass Transfer*, Vol. 33, No. 12, pp. 2659~2669.

Heindel, T. J., Ramadhyani, S. and Incropera, F. P., 1992, "Liquid Immersion Cooling of a Longitudinal Array of Discrete Heat Sources in Protruding Substrates: Ⅱ-Single-phase Convection," *Trans. of the ASME, J. Electronic Packaging*, Vol. 114, pp. 55~62.

Incropera, F. P., Kerby, J. S., Moffatt, D. F. and Ramadhyani, S., 1986, "Convection Heat Transfer from Discrete Heat Sources in a Rectangular Channel," *Int. J. Heat Mass Transfer*, Vol. 29, pp. 1051~1058.

Jones, Jr, O. C., 1976, "An Improvement in the Calculation of Turbulent Friction in Rectangular Ducts," *J. Fluid Engineering*, Vol. 98, pp. 173~181.

Kostic, M. and Hartnett, J. P., 1986, "Heat Transfer to Water Flowing Turbulently through a Rectangular Duct with Asymmetric Heating," *Int. J. Heat Mass Transfer*, Vol. 29, No. 8, pp. 1283~1291.

Moffat, R. J., 1985, "Using Uncertainty Analysis in the Planning of and Experiment," *J. Fluid Engineering*, Vol. 107, pp. 173~182.

Pautsch, G. and Bar-Cohen, A., 1999, "Thermal Management of Multichip Modules with Evaporative Spray Cooling," *Proceedings of the Pacific Rim/ASME International and Intersociety Electronic and Photonic Packaging Conference*, Hawaii, pp. 1453~1461.

Ramaswamy, C., Joshi, Y., Nakayama, W. and Johnson, W. I., 1998, "Performance of a Compact Two-chamber Two-phase Thermosyphon: Effect of Evaporator Inclination, Liquid Fill Volume and Contact Resistance," *Proceedings of 11th IHTC*, Kyongju, Korea, pp. 127~132.

Sparrow, E. M., Lloyd, J. R. and Hixon, C. W., 1966, "Experiments on Turbulent Heat Transfer in an Asymmetrically Heated Rectangular Duct," *Trans. of the ASME, J. Heat Transfer*, Vol. 88, pp. 170~174.

Tan, H. M. and Charters, M. S., 1970, "An Experimental Investigation of Forced Convective Heat Transfer for Fully-Developed Turbulent Flow in a Rectangular Duct with Asymmetric Heating," *Sol. Energy*, Vol. 13, pp. 121~125.