PERGAMON

International Journal of Heat and Mass Transfer 44 (2001) 55–61

International Journal of HEAT and MASS TRANSFER

www.elsevier.com/locate/ijhmt

Effect of the aspect ratio of rectangular channels on the heat transfer and hydrodynamics of paraffin slurry flow

Mingoo Choi, Keumnam Cho*

School of Mechanical Engineering, Sungkyunkwan University, 300 Chunchun-dong, Changan-ku, Suwon 440-746, South Korea

Received 28 September 1999; received in revised form 28 February 2000

Abstract

The objective of the present study was to investigate the effect of the aspect ratio of rectangular channels on the cooling characteristics of paraffin slurry flow with a linear array of discrete heat sources. Four key experimental parameters were the aspect ratio of the rectangular channel, the mass fraction of paraffin slurry, the heat flux of simulated VLSI chips, and the channel Reynolds number. The influence of the aspect ratio of the rectangular channel on the local heat transfer coefficients for both water and paraffin slurry at a heat flux of 40 W cm^{-2} was greater than at a low heat flux. The paraffin slurry with a mass fraction of 5% showed the most efficient cooling performance in a rectangular channel with an aspect ratio of 0.20 regardless of the heat flux and the channel Reynolds number.

Keywords: Paraffin slurry; Aspect ratio; Rectangular channel; Discrete heat source; Mass fraction

1. Introduction

Convective heat transfer in ducts is encountered in a wide variety of engineering situations. In particular, the heat transfer behavior of turbulent flow through rectangular ducts is of special interest because of the application for the thermal management of computer and compact heat exchangers. For the thermal management of electronics, the power dissipation per chip has increased dramatically in recent years [1,2]. It has provided thermal engineers with the challenges of managing the increased thermal budget in order to maintain the chip at a safe operating temperature. For high-heat-flux applications, the effective liquid cooling method must be used that can keep the surface temperature of the chips with a high heat flux within a certain temperature limit. The application of a phase change material (PCM) slurry as an active liquid cooling method is attractive for the thermal management of high power electronics. The potential benefits of using PCM slurry in the thermal control of electronic systems are the enhanced heat absorption due to the phase change process and the conductivity enhancement induced by the motion of the particles [3–7].

In a PCM slurry flow applied to electronics, discrete heat sources are embedded at one wall of a rectangular channel, whereas coolants in the channel remove the heat generated by the heat sources. For a single phase flow, numerous theoretical and experimental results have been reported for laminar and turbulent flows in a continuously heated channel [8–14], and there have been a few data on the forced-convection heat transfer

* Corresponding author. Tel.: +82-331-290-7445; fax: +82-331-290-5849.

E-mail addresses: mingoo@nature.skku.ac.kr (M. Choi), keumnam@yurim.skku.ac.kr (K. Cho).

0017-9310/01/$ - see front matter
PII: S0017-9310(00)00095-8

Nomenclature

A	surface area (m^2)
D_h	hydraulic diameter (m)
EF	enhancement factor, Eq. (6)
f	Fanning friction factor, $(\Delta P \cdot D_h)/(2 \cdot \rho \cdot L \cdot U^2)$
H	channel height (mm)
h	heat transfer coefficient (W m^{-2} K^{-1})
L	length in the axial direction (m)
L_h	chip length (m)
ΔP	pressure drop (Pa)
q''	heat flux, $V^2/(R \cdot A_s)$ (W cm^{-2})
R	resistance (Ω)
Re_{D_h}	Reynolds number, $(\rho_f \cdot U \cdot D_h)/\mu_f$
T	temperature (°C)
U	average velocity (m s^{-1})
V	voltage (V)
W	channel width (mm)
x	mass fraction of paraffin slurry (%)

Greek symbols

α^*	aspect ratio of rectangular channel, H/W
μ	viscosity (N s m^{-2})
ρ	density (kg m^{-3})

Subscripts

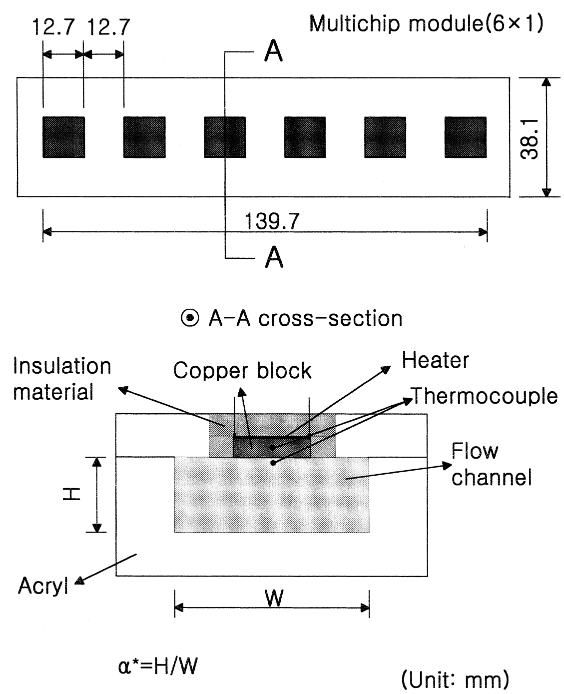
f	working fluid
p	paraffin
s	chip
w	water
+	dimensionless.

from discrete heat sources attached to one wall of a channel [15–17]. However, very few data were reported on the heat transfer characteristics of the PCM slurry flow in the rectangular duct with discrete heat sources. In addition, the influence of the geometric parameters of a rectangular channel on the cooling performance of PCM slurry flow must be studied. In particular, the aspect ratio of a rectangular duct is one of the most important parameters because the heat transfer characteristics in the rectangular channel critically depend on the aspect ratio. However, to the authors' knowledge, no data have been reported on the effect of the aspect ratio on the cooling performance 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 of such systems.

Therefore, the objective of the present study was to investigate the effect of the aspect ratio of rectangular channels on the cooling characteristics of a PCM slurry flow with a linear array of discrete square heat sources with high heat flux, an important design issue which affects the overall performance of a multichip module, cooled by the PCM slurry flow.

2. Experimental apparatus and procedure

The present experimental apparatus consisted of a main test section, a power supply, a plate-type heat exchanger, a constant temperature bath, a pump, a mass flow meter, and a data acquisition system. The test section consisted of a rectangular channel and a multichip module (MCM) as shown in Fig. 1. The multichip module had an in-line 6 × 1 array of discrete heat sources simulating VLSI chips. The module was flush-mounted on the top wall of a horizontal acrylic rectangular channel and located at a downstream location 50 times of the channel hydraulic diameter from



α*	0.1	0.2	0.33	0.5
H	3.8	7.6	12.7	19.0
W	38.1	38.1	38.1	38.1

Fig. 1. Details of the present test section.

the inlet of the test section so that the flow in the main test section was hydrodynamically fully developed.

More detailed descriptions on the present test section were given at the previous study [7]. The rectangular channels that have four different aspect ratios were used.

Present experiments were performed with water and paraffin slurry. A commercial grade paraffin ($C_{22}H_{46}$) was used. In order to make fine paraffin particles, an emulsifier was used. The process of making fine paraffin slurry was presented elsewhere [7].

Four key experimental parameters were the aspect ratio of the rectangular channel, the mass fraction of the paraffin slurry, heat flux, and the channel Reynolds number. The aspect ratios of the channel were 0.10, 0.20, 0.33, and 0.50; the mass fraction of the paraffin slurry varied from 0 (water) to 10%; the heat flux varied from 10 to 40 W cm^{-2}; and the channel Reynolds number ranged from 3000 to 15,000. The hydraulic diameters of four rectangular channels were 0.69, 1.27, 1.90, and 2.54 cm. The aspect ratio of the rectangular channel varied by changing the channel height, and the heat flux was obtained by dividing the heat supplied to each chip by the chip area. The channel Reynolds number was calculated by using the hydraulic diameter of the rectangular channel. The inlet temperature of the test section was set to 15°C for all runs.

A conduction analysis was performed on both the heat source and the acrylic substrate using the thermal conductivity values for copper block (386 W m^{-1} K^{-1}), acrylic (0.2 W m^{-1} K^{-1}), thermal silicon (1.6 W m^{-1} K^{-1}), and insulation material (0.02 W m^{-1} K^{-1}). The heat loss from the heater to the surroundings ranged from 2.3 to 3.8% in the whole experimental range and was compensated in the calculation of heat flux on the heater.

Local heat transfer coefficients were calculated by dividing the heat flux supplied to each chip by the temperature difference between the surface temperature of the chip and the fluid temperature just below the chip as shown in the following equation:

$$h = \frac{q''}{(T_s - T_f)} \tag{1}$$

The determination of a bulk temperature in the rectangular channel required temperature and velocity profiles in 2-dimensional directions. This made it difficult to determine the bulk temperature. Thus, the convective heat transfer coefficients were obtained by using fluid temperature just below a heat sink instead of the bulk temperature. The heat flux was obtained by using the supplied voltage and the resistance of the heater. The error analysis showed that the error range of the heat flux was ±1.93%, whereas that in the local heat transfer coefficient calculation was ±3.7%.

The size of a paraffin slurry was controlled by the amount of an emulsifier. When the concentration of the emulsifier relative to each mass fraction of the paraffin was 3.3%, the size of the paraffin slurry particles ranged from 10 to 40 μm [7].

The density of the paraffin slurry was obtained by the following equation [18]:

$$\frac{1}{\rho_f} = \frac{x/100}{\rho_p} + \frac{1 - x/100}{\rho_w} \tag{2}$$

which can be used for the slurries of very fine, mono-size, spherical particles.

The relationship between the viscosity and concentration of the suspension of paraffin slurry with the error range of ±5% was proposed by Shook [18]:

$$\frac{\mu_f}{\mu_w} = 1 + 2.5\frac{\rho_f}{\rho_p}\frac{x}{100} + 10.05\left(\frac{\rho_f}{\rho_p}\frac{x}{100}\right)^2 + 0.00273\exp\left(16.6\frac{\rho_f}{\rho_p}\frac{x}{100}\right) \tag{3}$$

3. Results and discussion

Fig. 2 shows the apparent Fanning friction factor obtained from the present test section as a function of the channel Reynolds number. The modified Blasius equation introduced by Jones [19] for rectangular ducts can be described as:

$$f = 0.079\left(Re^*_{D_h}\right)^{-0.25} \tag{4}$$

where the Kozicki Reynolds number, $Re^*_{D_h}$, is defined as:

$$Re^*_{D_h} = a(\rho U D_h/\mu) \tag{5}$$

Note that the value of a in Eq. (5) is constant, which depends on the aspect ratio of a rectangular channel, as given in Table 1.

Measured friction factors for water and the 5% paraffin slurry increased as the aspect ratio of a rectangular channel decreased due to the difference in the mean velocity. The friction factors for water gave good agreement with the values predicted by the modified

Table 1
Values of a in Eq. (5)

α^*	1	0.75	0.50	0.23	0
a	1.125	1.106	1.029	0.878	0.667

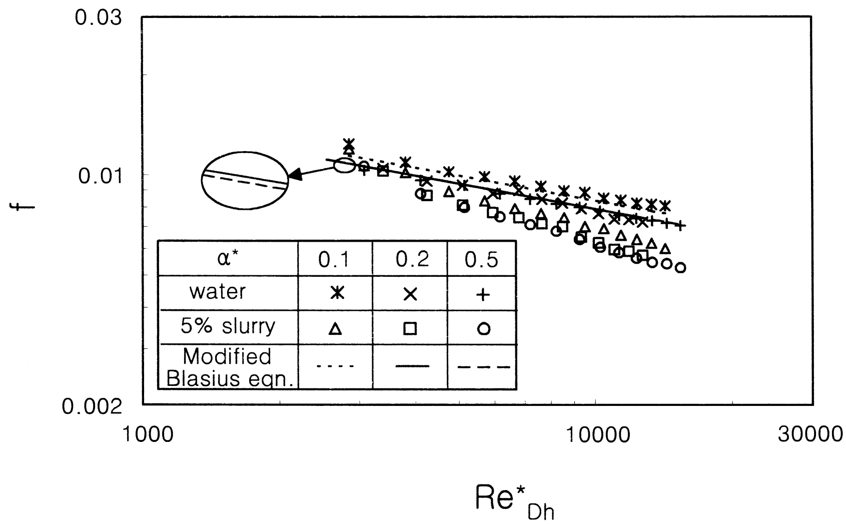


Fig. 2. Fanning friction factors for water and 5% paraffin slurry.

Blasius equation within ±7%. Pressure drop data for the 5% paraffin slurry were larger than those for water, whereas the apparent Fanning friction factors for the 5% paraffin slurry were smaller than those for water for all the three cases of aspect ratio. The reason is that the average velocity for the paraffin slurry increased as the mass fraction of the paraffin slurry increased at the same Reynolds number since the kinematic viscosity of the paraffin slurry was greater than that of water.

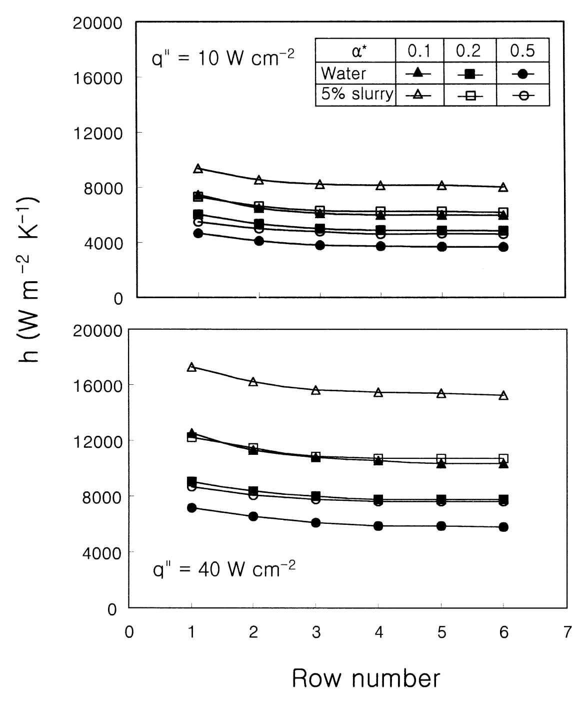


Fig. 3. Local heat transfer coefficients with respect to row number ($Re_{D_h} = 15{,}000$).

Local heat transfer coefficients were obtained by using the chip surface temperature and fluid temperature. Fig. 3 shows local heat transfer coefficients with respect to the row number for a channel Reynolds number of 15,000 and two different heat fluxes of 10 and 40 W cm^{-2}. The local heat transfer coefficients reached a uniform value approximatsely after the fourth row (located at a distance seven times of the chip length) regardless of coolants and aspect ratios. This means that the thermally fully developed condition was reached after the fourth row. Garimella and Eibeck [20] also observed the attainment of fully developed conditions after the fourth row. The local heat transfer coefficients for the paraffin slurry with a mass fraction of 5% were larger than those for water. The reason for the heat transfer enhancements was due to both the particle migration and subsequent collision against the wall in turbulent flows and the latent heat of the PCM. Transverse migration of particles adjacent to a surface can aid in both disrupting the laminar sublayer and increasing the heat transfer coefficient. The latent heat of the PCM, which can be viewed as a form of specific heat, increases the heat transfer coefficient because the heat transfer coefficient increases as the one-third power of the specific heat for the turbulent flows. The local heat transfer coefficients for a rectangular channel with an aspect ratio of 0.10 were larger than those for the rectangular channels with aspect ratios of 0.20 and 0.50. The reason is that the average flow velocity for a rectangular channel with an aspect ratio of 0.10 was the largest among the channels of three different aspect ratios at the same Reynolds number due to the smallest hydraulic diameter. The differences of the local heat transfer coefficients between the paraffin slurry with a mass fraction of 5% and water increased as the aspect ratio of the rectangular channel decreased. Especially, at a high heat flux of 40 W cm^{-2}, the influence of the aspect ratio of the rectangular channel on the local heat transfer coefficients for both water and the paraffin slurry with a mass fraction of 5% was greater than those at a low heat flux.

Fig. 4 shows dimensionless local heat transfer coefficients (h^+) at the sixth row with respect to the mass fraction of the paraffin slurry for the case of a Reynolds number of 15,000. The dimensionless local heat transfer coefficient is defined as the ratio of the local heat transfer coefficient for the paraffin slurry to that for water. As the mass fraction increased from 0 to 10%, the dimensionless local heat transfer coefficients at heat fluxes of 10 and 40 W cm^{-2} slowly increased when the mass fraction increased over 5%. This indicates that the local heat transfer coefficients reached an asymptotic value beyond a mass fraction of 10% when the Reynolds number was 15,000 and the heat flux varied from 10 to 40 W cm^{-2}. The dimensionless local heat transfer coefficients increased as the aspect ratio of the rectangular channel decreased, but the increasing rate was not linear. This means that there may exist an optimal aspect ratio of the rectangular channel based on the heat transfer aspect for the paraffin slurry flow.

The dimensionless local heat transfer coefficients at heat fluxes of 10 and 40 W cm^{-2} were larger at the sixth row than those at the first row. This means that the effect of the latent heat of the paraffin slurry on the local heat transfer coefficient was larger at the downstream than that at the upstream. At the sixth row, the cooling effect by the sensible heat of the paraffin slurry decreased since the temperature of the working fluid increased. However, since the fluid temperature reached the melting temperature of the paraffin, the cooling effect by the latent heat increased downstream of the test section. When the mass fraction of the paraffin slurry was 10%, the dimensionless heat transfer coefficients at a heat flux of 40 W cm^{-2} for the rectangular channels with the aspect ratios of 0.50, 0.20, and 0.10 were 1.42, 1.54, and 1.59, respectively.

Fig. 5 shows enhancement factors (EFs) at the sixth row for three aspect ratios of rectangular channels obtained by using the local heat transfer coefficient and the pressure drop at a Reynolds number of 15,000. Note that the EF is defined as:

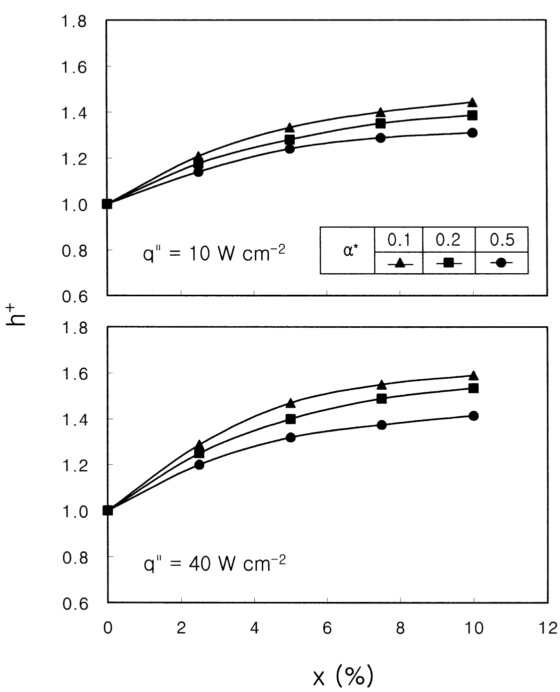


Fig. 4. Dimensionless local heat transfer coefficients at the sixth row ($Re_{D_h} = 15{,}000$).

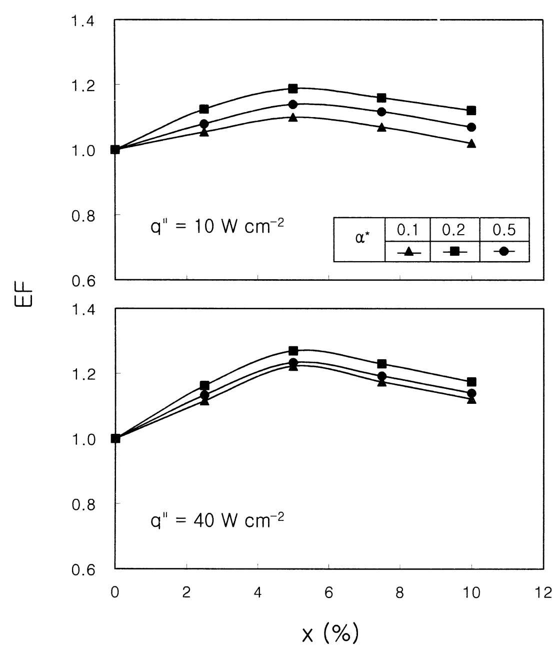


Fig. 5. Enhancement factors at the 6th row ($Re_{D_h} = 15{,}000$).

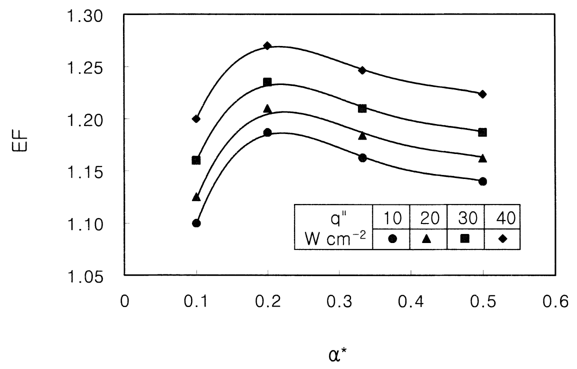


Fig. 6. Enhancement factors for 5% paraffin slurry with respect to aspect ratio at four different heat fluxes ($Re_{D_h} = 15{,}000$).

$$\mathrm{EF} = \frac{h^+}{\Delta P^+} \tag{6}$$

where ΔP^+ is the ratio of the pressure drop for the paraffin slurry to that for water. For heat fluxes of 10 and 40 W cm^{-2}, the EFs for the paraffin slurry with a mass fraction of 5% showed the largest value regardless of an aspect ratio of rectangular channel. Moreover, the EF for a rectangular channel with an aspect ratio of 0.20 showed the largest value among three aspect ratios. The EFs for the rectangular channels with aspect ratios of 0.50, 0.20, and 0.10 were 1.14, 1.19, and 1.10 at a heat flux of 10 W cm^{-2}, and were 1.23, 1.27, and 1.22 at a heat flux of 40 W cm^{-2}, respectively, for a mass fraction of the paraffin slurry of 5%. The values of EF for the rectangular channel with an aspect ratio of 0.10 were the smallest among three aspect ratios due to the largest pressure drop.

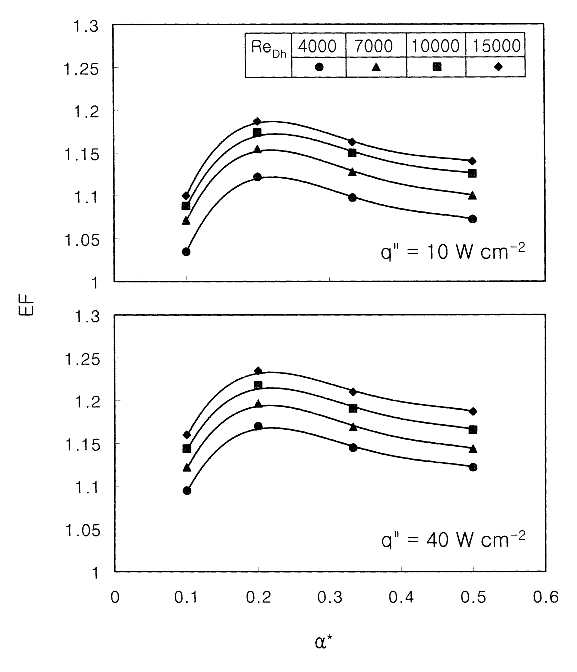


Fig. 7. Enhancement factors for 5% paraffin slurry with respect to aspect ratio at four different Reynolds numbers.

Fig. 6 shows EFs for the 5% paraffin slurry as a function of the aspect ratio at four different heat fluxes. In order to obtain the reasonable data trend, the data for a rectangular channel with an aspect ratio of 0.33 were added. The EFs increased as the heat flux increased. The effect of latent heat of the paraffin slurry became great as the heat flux increased because the melting rate of the paraffin particles on the heating surfaces increased, while the pressure drop at each aspect ratio was slightly affected by the heat flux. The EFs for the rectangular channel with an aspect ratio of 0.20 showed the largest values among four aspect ratios, regardless of the heat flux.

Fig. 7 shows EFs for a 5% paraffin slurry with respect to aspect ratio at four different Reynolds numbers. The EFs increased as the channel Reynolds number increased, but the increasing rate became small as the channel Reynolds number increased, especially at a heat flux of 10 W cm^{-2}. The reason is that at low heat flux, the increasing rate of the pressure drop was larger than that of the heat transfer coefficient as the channel Reynolds number increased. At a high heat flux such as 30 W cm^{-2}, the increasing rate of the heat transfer coefficient was comparable to that of the pressure drop as the channel Reynolds number increased.

In summary, the paraffin slurry with a mass fraction of 5% for the rectangular channel with an aspect ratio of 0.20 showed the most efficient cooling performance when the heat transfer and pressure drop in the test section were considered simultaneously, regardless of the heat flux and the channel Reynolds number.

4. Conclusions

A brief summary of the present study is given below.

1. Fanning friction factors for a paraffin slurry in rectangular channels were smaller than those for water.
2. Local heat transfer coefficients for the paraffin slurry were larger than those for water due to the disruption of laminar sublayer by paraffin particles, the direct heat transfer due to particle–wall impact, and the role of the latent heat of the paraffin.
3. The influence of the aspect ratio of the rectangular channel on the local heat transfer coefficients for both water and the paraffin slurry at a high heat flux of 40 W cm^{-2} was greater than that at a low heat flux.
4. The paraffin slurry with a mass fraction of 5% for

the rectangular channel with an aspect ratio of 0.20 showed the most efficient cooling performance regardless of the heat flux and the channel Reynolds number.

Acknowledgements

The authors wish to acknowledge the financial support of the Korea Research Foundation (1998-018-E00018), the Brain Korea 21 Project and useful discussion with Dr. Young I. Cho at Drexel University.

References

[1] C. Ramaswamy, Y. Joshi, W. Nakayama, W.I. Johnson, Performance of a compact two-chamber two-phase thermosyphon, in: Proceedings of 11th International Heat Transfer Conference Kyongju, KSME, 1998, pp. 127–132.

[2] G. Pautsch, A. Bar-Cohen, Thermal management of multichip modules with evaporative spray cooling, in: Proceedings of the Pacific Rim/ASME International and Intersociety Electronic and Photonic Packaging Conference, Hawaii, ASME, 1999, pp. 1453–1461.

[3] C.W. Sohn, M.M. Chen, Microconvective thermal conductivity in disperse two-phase mixture as observed in a laminar flow, ASME J. Heat Transfer 103 (1981) 47–51.

[4] C.W. Sohn, M.M. Chen, Heat transfer enhancement in laminar slurry pipe flows with power law thermal conductivities, ASME J. Heat Transfer 106 (1984) 539–542.

[5] P. Charunyakorn, S. Sengupta, S.K. Roy, Forced convection heat transfer in microencapsulated phase change material slurries: flow in circular ducts, Int. J. Heat Mass Transfer 34 (3) (1991) 819–833.

[6] E. Choi, Y.I. Cho, H.G. Lorsch, Forced convection heat transfer with phase-change-material slurries: turbulent flow in a circular tube, Int. J. Heat Mass Transfer 37 (1994) 207–215.

[7] M. Choi, K. Cho, Liquid cooling for a multichip module using Fluorinert liquid and paraffin slurry, Int. J. Heat Mass Transfer 43 (2000) 209–218.

[8] R.K. Shah, A.L. London, Laminar flow forced convection in ducts, in: J.P. Hartnett, T.F Irvine Jr. (Eds.), Advances in Heat Transfer, Academic Press, New York, 1978, pp. 196–222.

[9] R.K. Shah, M.S. Bhatti, Laminar convective heat transfer in ducts, in: S. Kakac, R.K. Shah, W. Aung (Eds.), Handbook of Single Phase Convective Heat Transfer, Wiley/Interscience, New York, 1987, pp. 3.45–3.52.

[10] M.S. Bhatti, R.K. Shah, Turbulent and transition flow convective heat transfer in ducts, in: S. Kakac, R.K. Shah, W. Aung (Eds.), Handbook of Single Phase Convective Heat Transfer, Wiley/Interscience, New York, 1987, pp. 4.73–4.85.

[11] E.M. Sparrow, J.R. Lloyd, C.W. Hixon, Experiments on turbulent heat transfer in an asymmetrically heated rectangular duct, ASME J. Heat Transfer 88 (1966) 170–174.

[12] H.M. Tan, M.S. Charters, An experimental investigation of forced convective heat transfer for fully-developed turbulent flow in a rectangular duct with asymmetric heating, Sol. Energy 13 (1970) 121–125.

[13] M. Kostic, J.P. Hartnett, Heat transfer to water flowing turbulently through a rectangular duct with asymmetric heating, Int. J. Heat Mass Transfer 29 (1986) 1283–1291.

[14] J.P. Hartnett, M. Kostic, Heat transfer to Newtonian and non-Newtonian fluids in rectangular ducts, in: J.P. Hartnett, T.F. Irvine Jr. (Eds.), Advances in Heat Transfer, Academic Press, New York, 1989, pp. 247–356.

[15] F.P. Incropera, J.S. Kerby, D.F. Moffatt, S. Ramadhyani, Convection heat transfer from discrete heat sources in a rectangular channel, Int. J. Heat Mass Transfer 29 (1986) 1051–1058.

[16] T.J. Heindel, S. Ramadhyani, F.P. Incropera, Liquid immersion cooling of a longitudinal array of discrete heat sources in protruding substrates: II — forced convection boiling, ASME J. Electronic Packaging 114 (1992) 63–70.

[17] M. Choi, K. Cho, Thermal characteristics of a multichip module using PF-5060 and water, KSME Int. J. 13 (1999) 443–450.

[18] C.A. Shook, Slurry pipeline flow, in: P.A. Shamlou (Ed.), Processing of Solid–Liquid Suspensions, Butterworth–Heinemann, Oxford, 1993, pp. 287–309.

[19] O.C. Jones Jr., An improvement in the calculation of turbulent friction in rectangular ducts, ASME J. Fluid Engineering 98 (1976) 173–181.

[20] S.V. Garimella, P.A. Eibeck, Heat transfer characteristics of an array of protruding elements in single phase forced convection, Int. J. Heat Mass Transfer 33 (12) (1990) 2659–2669.