An Experiment on Thermosyphon Boiling in Uniformly Heated Vertical Tube and Asymmetrically Heated Vertical Channel

Jin-Seok Jeon, Jung Hee Na, Hong Chul Park, Ho-Young Kwak*
Department of Mechanical Engineering, Chung-Ang University

Continuing efforts to achieve increased circuit performance in electronic package have resulted in higher power density at chip and module level. As a result, the thermal management of electronic package has been important in maintaining or improving the reliability of the component. An experimental investigation of thermosyphonic boiling in vertical tube and channel made by two parallel rectangular plates was carried out in this study for possible application of the direct immersion cooling. Fluorinert FC-72 as a working fluid was used in this experiment. Asymmetric heated channel of open periphery with gap size of 1, 2, 4 and 26 mm and uniformly heated vertical tubes with diameter of 9, 15 and 20 mm were boiled at saturated condition. The boiling curves from tested surfaces exhibited the boiling hysteresis. It was also found that the gap size is not a significant parameter for the thermosyphonic boiling heat transfer with this Fluorinert. Rather pool boiling characteristics appeared for larger gap size and tube diameter. The heat transfer coefficients measured were also compared with the calculation results by Chen's correlation.

Key Words : Boiling Hysteresis, Immersion Cooling, Thermosyphon Boiling

Nomenclature

C_f	: Friction factor
C_p	: Specific heat at constant pressure (kJ/kg · K)
d	: Hydraulic diameter based on wetted perimeter
d_h	: Hydraulic diameter based on heated perimeter
F	: Macroscopic heat transfer coefficient multiplier
g	: Gravitational constant (m/s²)
G	: Mass flux (kg/m² · s)
h	: Heat transfer coefficient(W/m² · K)
h_{fg}	: Latent heat of vaporization(kJ/kg)
k_l	: Fluid thermal conductivity(W/m · K).
L	: Effective length of the cold plate (m).
P	: Pressure(N/m²)
Pr_l	: Liquid Prandtl number
ΔP_{SAT}	: Saturation pressure difference corresponding to ΔT_{SAT}
q	: Surface heat flux to base area of cold plate(W/cm²).
Re	: Reynolds number
S	: Nucleate boiling suppression factor
T_W	: Wall temperature of prime surface (base wall) of channel
T_{SAT}	: Saturation temperature of coolant
ΔT_{SAT}	: Wall superheat, $T_W - T_{SAT}$
x	: Mass quality
X_{tt}	: Martinelli parameter
z	: Downstream (vertical) coordinate measured from inlet edge of active cold plate surface

Greek Letters

α	: Vapor void fraction
μ	: Dynamic viscosity (N · s/m²)
ρ	: Density (kg/m³)
Φ	: Two-phase multiplier used in Eq. (2)

* Corresponding Author,
E-mail : kwakhy@cau.ac.kr
TEL : +82-2-820-5278 ; **FAX** : +82-2-823-8616
Department of Mechanical Engineering Chung-Aug University, 221, Huksuk-dong, Dongjak-gu, Seoul 156-756, Korea. (Manuscript **Received** April 17, 2000; **Revised** October 13, 2000)

σ : Surface tension (N/m)
τ : Shear stress (N/m)

Subscripts

CB : Convective boiling
FZ : Forster-Zuber
g : Vapor properties or corresponding to vapor flow alone in the channel
l : Liquid properties or corresponding to liquid flow alone in the channel
l_o : Liquid properties evaluated at the total mass flux

1. Introduction

As a result of continued growth in functional density, advanced electronic devices have been a trend towards ever higher heat flux at chip, module, and system level. The heat flux for a single chip is already up to 0.4 MW/m^2 and is expected to exceed 1 MW/m^2 in the near future. Therefore, the thermal management of electronic package has played an important role in maintaining or improving reliable operation since the temperature of all components must keep within their specified functional and maximum limit(Lyman, 1982; Chu and Simon, 1984; Bar-Cohen, 1983).

The direct liquid immersion cooling is being considered for these applications because it effectively cools the heated surface and offers a benign local environment for microeletronic device (Lyman, 1982). The cooling method has also been considered as one of the promising methods to cool the high density chips in supercomputer, high heat flux modular electronics and high-powered X-ray and other diagnostic devices. Some selected dielectric fluid such as FC-72 is very attractive coolant during nucleate boiling since it is chemically inert one of high dielectric strength and low boiling point (Marto and Lepere, 1982; 3M, 1980). However, its high wettability induces the phenomenon called boiling hysteresis (You, et al., 1990) due to temperature overshoot(or thermal excursion). As a result, it is important to accurately establish the incipient boiling characteristics (Baker, 1973). Also, as the spacing for cooling is getting smaller and smaller to minimize packaging volume correspondingly propagation delay, the boiling characteristics in constrained space should be considered importantly.

Oktay (1982) tested the heat sink with vertical holes in copper block and obtained enhanced nucleate boiling and a reduced temperature overshoot in FC-86, which is attributed to the pumping effect of liquid through vertical holes. Bar-Cohen and Schweitzer (1985) conducted experiments to investigate boiling heat transfer from a pair of flat, closely spaced, isoflux plates immersed in saturated water. In their result, the wall superheat at constant imposed heat flux was found to decrease as the channel space was narrowed. Fujita et al. (1988) also carried out experiments to study on the heat transfer and critical heat flux of boiling in a narrow space. They found that at a moderate gap size, marked heat transfer enhancement was observed, but the effect of boiling surface orientation which was changed from upward facing to downward facing was very small in the case of open periphery.

Xia et al (1992) conducted similar experiments with subcooled and saturated conditions in vertical channel of different gap sizes. They found that there existed an optimal gap size for the heat transfer enhancement. Recent experiments by Zhao et al (1998) with working fluid of water revealed that the heat transfer was enhanced by decreasing the gap size from 5 mm to 1 mm, however, a further reduction in the gap size caused a degradation of heat transfer. They also found that the gap size had relatively smaller influence on the heat transfer in subcooled condition rather than in saturated condition.

In the present study, an experimental investigation of thermosyphonic boiling heat transfer in vertical tube and channel made by two parallel rectangular plates with open periphery to understand the effect of gap size and pumping action during nucleate boiling in Fluorinert FC-72. Asymmetrically heated isoflux plates and uniformly heated tube were employed in this study. Chen's correlation (1980) was employed to estimate the heat transfer coefficient for the

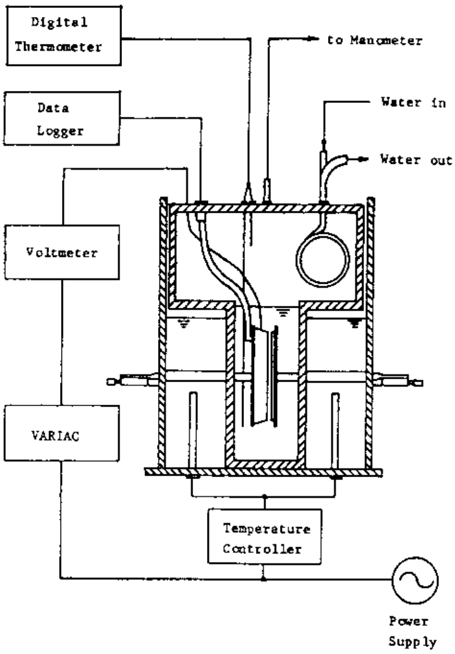


Fig. 1 Schematic of experimental apparatus

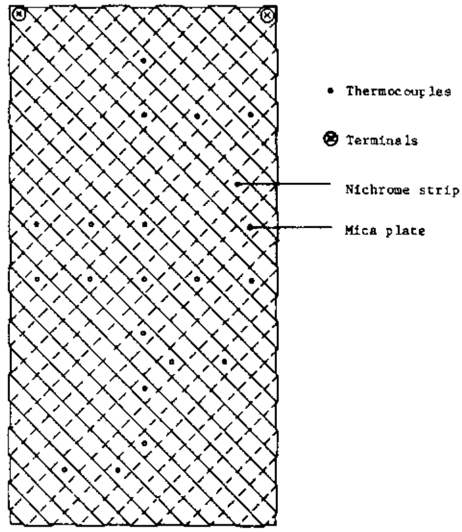


Fig. 2 Heater design with a nichrome strip

convective boiling in the constrained space.

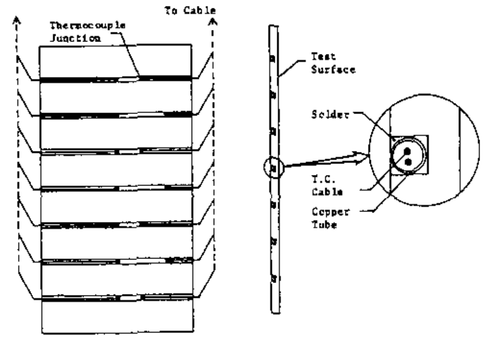
2. Experimental Apparatus

The experimental apparatus is shown schematically in Fig. 1. An acryl chamber to contain working fluid and test heater was placed in aluminum vessel (400 mm×400 mm×500 mm). Fluorinert FC-72, a working fluid was heated indirectly by controlling the temperature of distilled water which was filled in aluminum vessel. The aluminum vessel included glass windows, on the front and rear walls, to facilitate visual observation. The top cover of acryl chamber had taps and thermocouple connector, manometer, power cable and condenser.

The parallel plate channel was formed by a heated surface and an adiabatic surface. Each plate was supported by horizontal stainless steel bars. The gap size of channel can be varied continuously by a micrometer assembly during operation of the apparatus.

The heating surface which is 70 mm wide, 120 mm long and 4 mm thick is a rectangular copper plate. The heating element was made by nichrome strip which was wound with inclination angle of 45 degree on the thin mica plate as shown in Fig. 2. The distance between adjacent strip is 5 mm. The strip dimension is 1800 mm in length, 5 mm in width and 0.2 mm in thickness. The electric resistance of the strip heater is about 2.4 Ω. This heating element was attached to the heating surface using RTV, a silicon elastomer for electrical insulation and heat conduction. On the rear surface of heating element, a 10 mm rock wool was attached by ceramic adhesive. To connect the positioning mechanism to the test heater assembly, 6 mm expoxy glass was also attached to the surface of rock wool. After assembly, the edges of heater were covered with a layer of RTV, to provide sealing. The temperaturs of the heating surface were measured by T-type thermocouples with a potentiometer. The thermocouples were positioned inside copper tube soldered at seven rectangular grooves (2 mm×2 mm) as shown in Fig. 3.

In the case of tube heater, the heating surface was formed by two concentric copper tubes, whose inner diameter was varied as 9 mm, 15 mm and 20 mm. In the space between the inner and the outer copper tube, four T-type thermocouples covered by glass wools were inserted being spaced 90 deg apart in the middle of the tube and the remainder part of space was packed with solder. Nichrome wire of 0.5 mm were wound around



(a) Back view of copper plate (b) Side view

Fig. 3 Thermocouples imbeded in the copper plate attached to the heater shown in Fig. 2

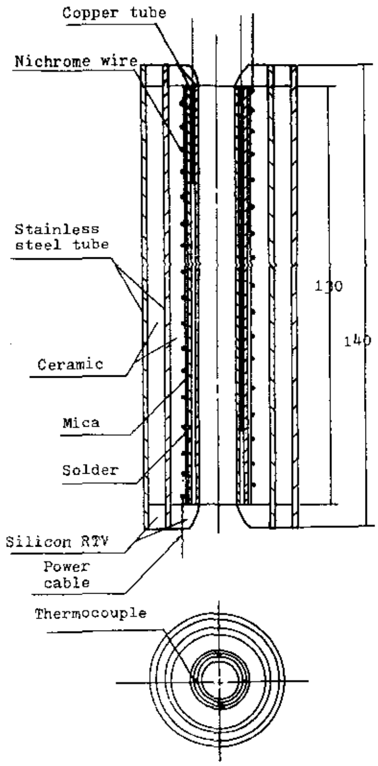


Fig. 4 Heater design for the convective boiling experiments in vertical tube

outer tube for electrical heating. The electric resistance of the heaters employed is 26.1 Ω for 9 mm D test tube, 36.4 Ω for 15 mm D test tube and 40.0 Ω for 20 mm D test tube. A sheet of mica was sandwiched between nichrome wire and the outer tube for electrical insulation. To prevent heat flow outwards, two stainless steel tubes with height of 140 mm were placed concentrically to the heater surface. The spaces between heater surface and inner stainless tube and the space between inner and outer stainless tubes were filled with ceramic adhesive to prevent the heat flow to outside. The top and bottom part of heater assembly were sealed with RTV. Detailed design of the tube heater is shown in Fig. 4.

The immersion heater was controlled by a temperature controller. The power applied to the test heater was measured by voltmeter and ampere meter. The surface heat flux was determined by calculating the heat power using measured voltage and current to the test heater. The working fluid temperature, and its vapor temperature were measured by a set of K-type thermocouples which were connected to digital thermometer.

3. Experimental Procedure

Test heater surface were firstly polished with #600, #1200 and finally with #1500 sand papers in order to insure uniform, smooth and clean heating surface. And it was finally cleaned with acetone. The acryl chamber was also cleaned with acetone before each run.

At the start of each run, the pool was boiled to saturated condition and the test heater was turned on about 4×10^4 W/m^2 in order to degas the test liquid. The pool was again maintained at saturated condition by controlling immersion heater. Heat was removed from the acryl chamber by a water cooled condenser coiled at the upper section of the chamber so that saturated condition was maintained during experiment.

When the system was stabilized to saturated condition, the power to the test heater was increased by 1 to 5 V increments for the vertical channel case and was increased by 2 to 10 V increments for the vertical tube case. The power to the test heater was then decreased in the same fashion. At each power setting, the system was permitted to achieve the steady state condition for 10 minutes and the following data were taken, heater voltage, current to the heater, vapor and liquid temperature of the working fluid and the temperature of the heating surface.

The digital thermometer of K-type employed to measure the saturation temperature has an accuracy of ± 0.2℃. The temperatures of the heating surface, which were measured by a potentiometer were accurate within ± 0.05℃ so that an estimated uncertainty in the wall superheat of ± 0.25℃. The test heater voltage was measured by a digital voltmeter accurate to ± 0.2 V up to 35 V and ± 1 V up to 150 V. Heater current was determined by measuring the voltage drop across a precision resister connected in series with the test heater. The uncertainty in the measurement of the current to the heater is less than ± 0.1 A. The heat flow rate from the heater was determined by the measured voltage and current to the heater. These calculations including heat loss to the environment yields an estimated uncertainty in the heat flux of about ± 3.0 %, and an estimated uncertainty in the heat transfer coefficient of ± 5.5 %.

4. Heat and Mass Transfer Analysis

An analysis was done for the heat transfer performance of a cold plate cooled by convective boiling of FC-72 in flow passage having smooth surface. The pressure drop in the flow passage was calculated using the separated flow model (Chisholm, 1983). The pressure gradient to account for friction, acceleration and gravitation components for vertical channel with uniform cross-section area is given by

$$-\frac{dP}{dz}=\frac{4\tau}{d}+G^2\frac{d}{dz}\left[\frac{x^2}{\alpha\rho_g}+\frac{(1-x)^2}{(1-\alpha)\rho_l}\right]+[\alpha\rho_g+(1-\alpha)\rho_l]g \quad (1)$$

The pressure gradient due to friction may be obtained using the two phase multiplier, Φ_l.

$$\frac{4\tau}{d}=\left(-\frac{dP}{dz}\right)_l\cdot\Phi_l^2 \quad (2)$$

where $(-dP/dz)_l$ is a single phase frictional pressure gradient calculated at a liquid mass flux only, which is given by

$$\left(-\frac{dP}{dz}\right)_l=\frac{2C_f}{d}\frac{G^2(1-x)^2}{\rho_l} \quad (3)$$

For the friction coefficient, the following correlations (White, 1994) were used,

$$C_f=0.079\ Re_{,lo}^{-\frac{1}{4}} \text{ if } Re_{,lo}<2\times10^4 \quad (4)$$
$$=0.046\ Re_{,lo}^{-\frac{1}{5}} \text{ if } Re_{,lo}>2\times10^4$$

where, $Re_{,lo}=G\dfrac{d}{\mu_l}$

For the two phase multiplier, Fridel correlation (Freidel, 1979) which is applicable to any fluid, except when $\mu_l>\mu_g>1000$, was utilized.

With the assumption of uniform heat flow to the boiling surface, the quality at an arbitrary point in the flow passage can be obtained by the energy balance equation. That is

$$\frac{dx}{dz}=\frac{4q}{Gh_{fg}d_h} \quad (5)$$

The void fraction may be obtained by the following equation

$$\alpha=\frac{x}{S_L+x\left(1-S_L\frac{\rho_g}{\rho_l}\right)} \quad (6)$$

The slip ratio S_L in Eq. (6) is just the velocity ratio of vapor to the liquid, which was obtained by CISE correlation (Whalley, 1987). The Martinelli parameter needed to calculate S_L and the two phase multiplier by Fridel correlation is given for turbulent flow in a smooth tube;

$$X_{tt}=\left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_g}\right)^{0.1} \quad (7)$$

A correct mass flow rate may be obtained when the pressure drop in the flow passage is equal to the hydrostatic head. For the assumed mass flow, G, one can calculate the pressure drop in the flow passage for a given heat flux. By iteration procedure, one can find G with given hydrostatic heads. For the flow passage with smooth surface, the Chen's correlation may be utilized. This is

$$h_{CB}=h_lF+h_{FZ}S \quad (8)$$

The single phase liquid convective heat transfer coefficient h_l based on the total mass flow rate was calculated using the following McAdams' correlation (Rohsenow and Choi, 1961) rather than the Dittus and Boelter's one (Incropera and Dewitt, 1985). The term $(\rho_l-\rho_g)/\rho_g$ in the heat transfer correlation turned out to be very important in the forced convection induced by thermosyphonic action in vertical channel (Forster and Zuber, 1955).

$$h_l = 0.023 \frac{k_l}{d} \mathrm{Pr}_l^{0.4} Re_{lo}^{0.8} \left(x \frac{\rho_l - \rho_g}{\rho_g} + 1 \right)^{0.8} \tag{10}$$

The nucleate boiling heat transfer coefficient h_{FZ} may be calculated from the Forster-Zuber equation (1955) as follows

$$h_{FZ} = 0.00122 \left[\frac{k_l^{0.79} C_{pl}^{0.45} \rho_l^{0.49}}{\sigma^{0.5} \mu_l^{0.29} h_{fg}^{0.24} \rho_g^{0.24}} \right] \Delta T_{SAT}^{0.24} \Delta P_{SAT}^{0.75} \tag{11}$$

In the calculation of the nucleate boiling heat transfer coefficient h_{FZ}, the heat transfer area increase due to fins was also taken into account by the same way to obtain h_l. Since h_{FZ} depends on ΔT_{SAT}, the local wall superheat and heat transfer coefficient can be obtained by iteratively solving the following energy balance equation with Eq. (8).

$$q = h_{CB} \Delta T_{SAT} \tag{12}$$

For the calculations of the two-phase heat transfer coefficient multiplier F and the nucleate boiling suppression factor S, the curve-fit equations recommended by Collier (1981) were used. These are

$$F = 1 \text{ for } 1/X_{tt} < 0.1 \tag{13}$$

$$= 2.35(1/X_{tt} + 0.213)^{0.736} \text{ for } 1/X_{tt} > 0.1$$

$$S = 1/(1 + 2.53 \times 10^{-6} Re_{TP}^{1.17}) \text{ and}$$

$$Re_l = \frac{(1-x)Gd}{\mu_l} \tag{14}$$

where $Re_{TP} = Re_l F^{1.25}$ and the calculation of the pressure drop and the heat transfer coefficient were accomplished by dividing the length of the flow channel in cold plate into about 500 elements of equal size. The local pressure drop and the local heat transfer coefficient were calculated for each element. In turn, the local heat transfer coefficient was used to determine the local wall superheat.

5. Results and Discussions

Each experiment was performed at least twice to check the reproducibility of the results. Experimental data were presented in the form of heat flux versus excess temperature, $T_w - T_s$ where T_w is wall temperature and T_s is saturated temperature of liquid. The average of wall temperature measured at all points were taken as the value of T_w.

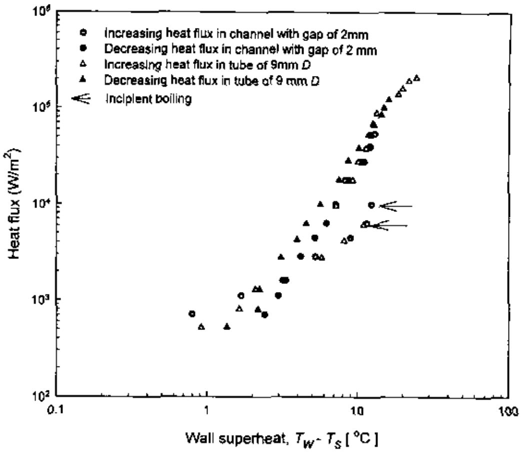


Fig. 5 Boiling curves from the vertical channel with gap of 2 mm and the vertical tube of 9 mm D

Figure 5 shows the boiling curves obtained from asymmetrically heated vertical channel with gap size of 2 mm and from the vertical tube with diameter of 9 mm. Boiling hysteresis due to the highly wettable characteristics of FC-72 is evident. The excess temperature at the incipient boiling is shown to be about 10.5 ℃, which is comparable to the Marto and Lepere's result obtained from horizontal plain tube (Marto and Lepere, 1982) as shown in Fig. 9. The incipient boiling point noted as ← in the boiling curves can be found by the sudden temperature drop of the boiling surface. Very similiar boiling curves are obtained from the both cases because the hydraulic diameters of heated area are almost same for both cases. In Fig. 6, the boiling curves obtained from the vertical channel with the gap size of 2 mm and 4 mm are shown; even though the temperature overshoot is reduced slightly for 2 mm case, the boiling curves of the two cases are essentially same. This is somewhat different result from the outcome of Bar-Cohen and Schweitzer (1985). Their results revealed that at constant imposed heat flux, the wall temperature decreases as the spacing narrows below 10 mm in water.

At lower heat flux below about 1.3×10^3 W/m²

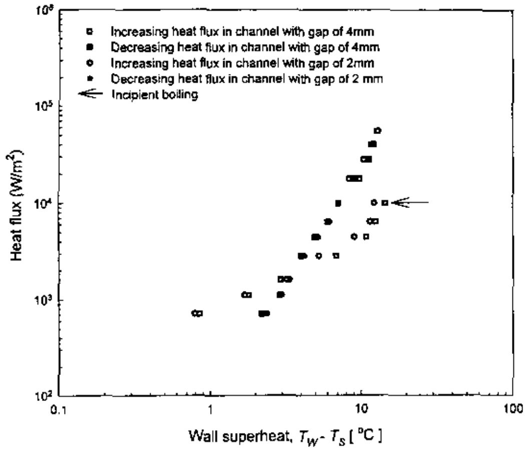


Fig. 6 Boiling curves from the vertical channels with gap of 2 mm and 4 mm

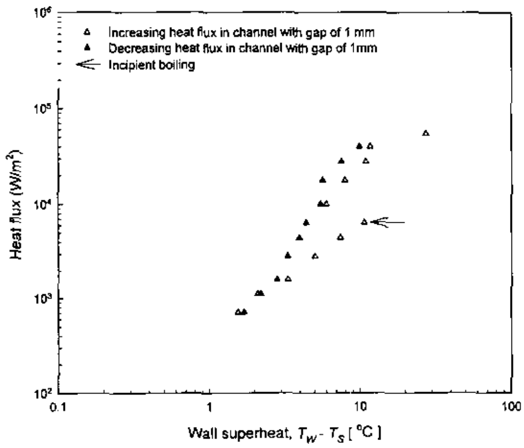


Fig. 7 Boiling curves from the vertical channel with gap of 1 mm

the heat transfer mode inside vertical channel is natural convection apparently as can be seen in Figs. 5 and 6. As the heat flux was increased to approximately 6.4×10^3 W/m^2, nucleation started at the top of test section which was also confirmed visually. The boiling curves in the case of decreasing heat flux took different path from that of the increasing case as shown in Figs. 5 and 6. The nucleation sites are still active at the heat flux of 1.5×10^3 W/m^2, which is well below the heat flux needed for incipient boiling. However, nucleation activity is entirely died out at about 1.2×10^3 W/m^2. For the gap size of 1 mm as shown in Fig. 7, it was found that the wall temperature is abruptly increased at the heat flux of 5.5×10^4

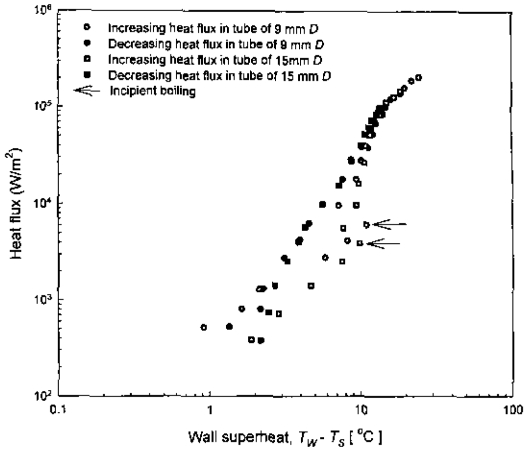


Fig. 8 Boiling curves from the vertical tubes of 9 mm D and 15 mm D

W/m^2, which is apparently CHF in this case so that some degradation in the heat transfer coefficient occurs in the decreasing case of heat flux for 1 mm gap. This value is certainly less than the CHF of 10^5 W/m^2 for the plain tube which has been confirmed by previous experiment (Marto and Lepere, 1982).

Figure 8 shows the boiling curves obtained from uniformly heated vertical tube with diameter of 9 mm and 15 mm. Similar to the vertical channel case, the boiling curves are almost identical especially at high heat flux region. Boiling hysteresis was also observed in this case. For each case, the excess temperature at incipient boiling point was not exceeded over 10℃ as can be seen from the boiling curves. However, the temperature overshoot was reduced (The temperature drop after boiling start is about 2~4℃) and nucleate boiling was enhanced comparing to the vertical channel case, which might be due to vigorous pumping because of the closed side periphery. For open periphery, the thermosyphon action due to the generation of bubble inside channel induces entrainment of fluid from the outside, which causes some degradation in the heat transfer coefficients. Actually, in the case of 9 mm diameter, the vapor was pumped about 25 cm from the top of tube at the heat flux of 1.0×10^5 W/m^2. Up to the heat flux of 1.7×10^5 W/m^2, the wall temperature increased slowly. However, further increase in power resulted in the dryout

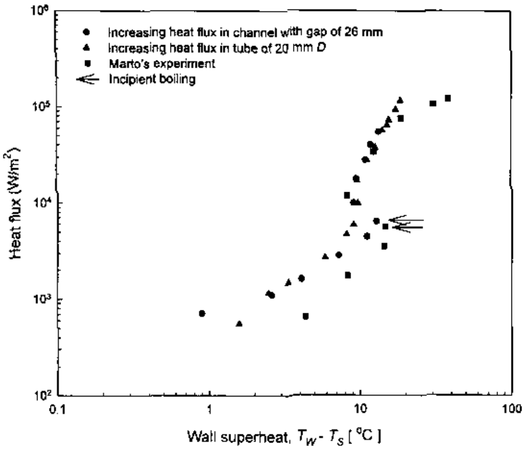


Fig. 9 Pool boiling curves from the vertical channel, the vertical tube and horizontal cylindrical surface (Reference, 4)

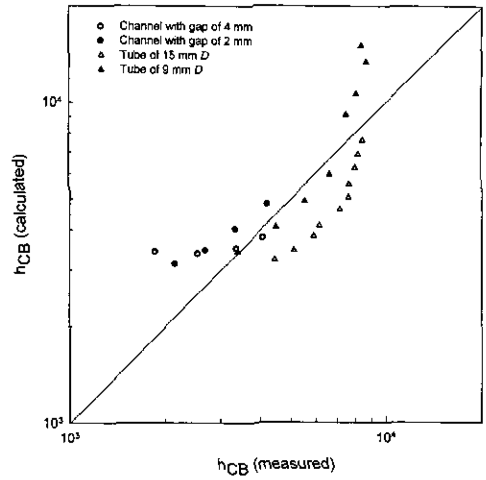


Fig. 10 Comparison of calculated and measured heat transfer coefficients

inside tube and caused sharp increase of wall temperature. Indeed, in the case of 9 mm diameter tube, the wall temperature was sharply increased at 2.15×10^5 W/m^2, and the heater stopped its operation at this point.

Figure 9 provides boiling curves obtained from the vertical channel with gap of 26 mm, vertical tube of 20 mm diameter along with the data obtained form the plain tube by Marto and Lepere (1982) for comparison purpose. As can be seen in this figure, all the boiling curves show similiar behavior, which indicate the pool boiling characteristics as expected for larger hydraulic diameter of heated region. However for the case of smaller gap size, the heat transfer with vertical tube case is superior in natural convection as well as in nucleation boiling region. Also, the critical heat flux obtained for the vertical tube is two times higher than that for plain tube as can be seen from Figs. 5 and 9. This good performance may be attributed to the pumping effect in the vertical tube or channel with closed periphery, which has also been verified by Oktay (1982).

The measured heat transfer rates from the vertical channel with open periphery and from vertical tubes along with the calculated values by the Chen's correlation are shown in Fig. 10. Reasonable agreement between the measured and calculated values especially for the cases where two-phase forced convection is dominant over the

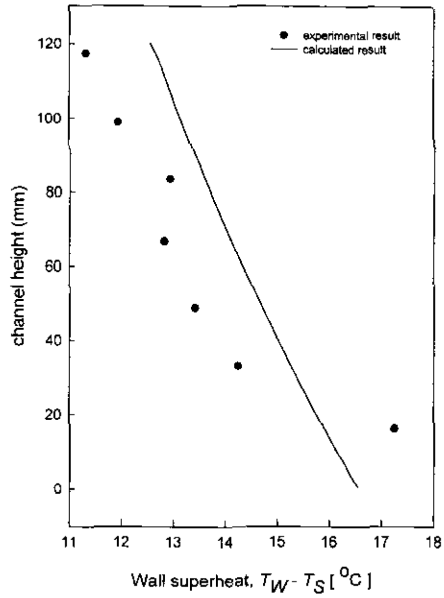


Fig. 11 Wall temperature along the channel height in the vertical tube with gap of 4 mm at the heat flux of 5.44×10^4 W/m^2

nucleate boiling (Hong et al., 1998) can be seen. However near the region where the onset of nucleate boiling(ONB) occurs and the range of the critical heat flux(CHF). the deviation between experimental results and calculated ones becomes greater. The temperature profile along the channel from the bottom end for the case of the vertical channel at the heat flux of 5.44×10^4

W/m^2 is shown in Fig. 11. Fair agreement between the observed data and the calculated values obtained by the Chen's correlation can also be seen. The calculated slip ratios which have the value between unity at lower heat flux and 5.0 around the critical heat flux are reasonable.

Finally, it is noted that the optimal spacing for the maximum heat transport in thermosyphonic boiling is closely related to the departure diameter of bubble. For ebullition in the dielectric fluids such as freon or fluorinert, the departure diameter of bubbles becomes larger because of large contact angle so that the optimal spacing should be larger than the case of ebullition in water. The squeezed and flattened bubbles to form thin liquid film in narrow channel (Zhao et al., 1998) may help improving the heat transfer coefficient. However, a large number of squeezed bubble which move slowly in the channel less than the optimal spacing may yield lower heat transfer coefficient. In the channel with optimal spacing, the increase in the bubble departure frequency due to the agitation induced by thermosyphonic action results in the higher heat transport than the case in unconfined channel

5. Conclusions

From the experimental results observed in this study, the following conclusions were obtained.

(1) For Fluorinert FC-72, the gap size is not significant parameter for thermosyphonic boiling heat transfer within the experimental conditions tested.

(2) Comparing to the horizontal smooth surface and the vertical channel with open periphery, the uniformly heated vertical tube has higher CHF, and the heat transfer performance is better, which is due to pumping effect.

(3) For narrow spacing such as the case of vertical channel with gap size of 1 mm, CHF may be substantially reduced because the mass flux decreases as the gap spacing decreases.

References

3M Company, "Fluorinert Electronic Liquids," Commercial Chemicals Division, 1980 Bulletin.

Bar-Cohen, A., 1983, "Thermal Design of Immersion Cooling Module for a Electronic Components," *Heat Transfer Engineering*, Vol. 4, pp. 35~50.

Bar-Cohen, A. and Schweitzer, H., 1985, "Thermosyphon Boiling in Vertical Channels," *ASME J. Heat Transfer*, Vol. 107, pp. 772~778.

Baker, E., 1973. "Liquid Immersion Cooling of Small Electronic Devices," *Microelectronics and Reliability*, Vol. 12, pp. 163~173.

Bennett, D. L., and Chen, J. C., 1980, "Forced Convective Boiling in Vertical Tubes for Saturated Pure Components and Binary Mixtures," *AIChE J.*, Vol. 26, pp. 454~461.

Chisholm, D., 1983, *Two-phase Flow in Pipelines and Heat Exchangers*, George Godwin, London.

Chu, R. C. and Simon, R. E., 1984, "Thermal Management of Large Scale Digital Computers," *The International Society for Hybrid Micro Electronics*, Vol. 7, pp. 35~43.

Collier, J. G., 1981, Forced Convective Boiling, Chap. 8 in *Two-Phase Flow and Heat Transfer in the Power and Process Industries*, Hemisphere Publishing Co., New York.

Forster, H. K., Zuber, N., 1955, "Dynamics of Vapor Bubble and Boiling Heat Transfer," *AIChE Journal*, Vol. 1, pp. 531~535.

Freidel, L., 1979, "Improved Friction Pressure Drop Correlation for Horizontal and Vertical Two-phase Flow," *in Europian Two-Phase Flow Group Meeting*, Italy.

Fujita, Y., Ohta, H. and Uchida, S., 1988, "Effect of Surface Orientation on Nucleate Boiling Heat Transfer in a Confined Narrow Space," *The First KSME-JSME Thermal and Fluids Engineering Conf.*, pp. I-73~I-78.

Hong, J., Park, C and Kwak, H., 1998, "Forced Convective Boiling in Vertical Tube for Binary Refrigerant Mixtures of R11 and R113," *KSME International J.*, Vol. 13, pp. 493~503.

Incropera. F. P. and Dewitt, D. P., 1985, *Intro-*

duction to Heat Transfer, John Wiley & Sons, Yew York.

Lyman, J., 1982, "Supercomputers Demands Innovation in Packaging and Cooling," *Electronics*, September, pp. 136~143.

Marto, P. J. and Lepere, V. J., 1982, "Pool Boiling Heat Transfer from Enhanced Surfaces to Dielectric Fluids," *ASME J. Heat Transfer*, Vol. 104, pp. 292~299.

Oktay. S., 1982, "Departure from Natural Convection(DNC) in Low-temperature Boiling Heat Transfer Encountered in Cooling Microelectronic Devices," *Proceedings 7th International Heat Transfer Conf.*, Munich, Germany, Vol. 4, pp. 113~118.

Rohsenow, W. M., Choi, H., 1961, *Heat, Mass and Momentum Heat Transfer*, Prentice-Hill, Englewood Cliffs.

Whalley, P. B., 1987, *Boiling Condensation and Gas-Liquid Flow*, Clarendor Press, Oxford.

White, F. M., 1994, *Fluid Mechanics*, 3rd Edition, McGraw-Hill, New York.

Xia, C., Guo, Z., and Hu, W., 1992, "Mechanism of Boiling Heat Transfer in Narrow Channels," in HTD- Vol. 197, *Two Phase Flow and Heat Transfer*. pp. 111~119.

You, S. M., Simon, T. W., Bar-Cohen, A., and Tong, W., 1990, "Experimental Investigation of Nucleate Boiling Incipience with a Highly-wetting Dielectric Liquid(R-113)," *Int. J. Heat Mass Transfer*, Vol. 33, pp. 105~113.

Zhao, T. S. Guo, X. C. and Cheng, P., 1998, "The Influence of Subcooling on Nucleate Boiling Heat Transfer in an Immersed Confined Space," *in Proceedings of 11th IHTC*, Vol. 2, pp. 485~490.