Int. J. Heat Mass Transfer. Vol. 36, No. 14, pp. 3421–3427, 1993
Printed in Great Britain

0017–9310/93 $6.00+0.00

Forced convection and flow boiling heat transfer for liquid flowing through microchannels

X. F. PENG and B.-X. WANG
Thermal Engineering Department, Tsinghua University, Beijing 100084, China

(*Received* 21 *October* 1992 *and in final form* 12 *January* 1993)

Abstract—Experiments were conducted to investigate the single-phase forced-flow convection and boiling characteristics of subcooled liquid flowing through microchannels with a cross-section of 0.6×0.7 mm, machined on the stainless steel plate 2 mm thick. The influences of liquid velocity and subcooling on the boiling curve were experimentally inspected. It was observed that a steep increase of q'' emerged on the single-phase convection $q''-T_w$ curve. The experiments indicated that the single-phase convection and flow boiling characteristics are quite different from those in normally sized tubes, and their heat transfer was intensified. No apparent partial nucleate boiling exists for subcooled flow boiling, i.e. fully-developed boiling was induced much earlier in microchannels.

1. INTRODUCTION

MICROFABRICATION or nanotechnology emerged from the technology developed for integrated circuits and expanded rapidly into such fields as bioengineering and biotechnology, aerospace, mini heaters and mini heat exchangers, electronics and microelectronics, material processing and thin film deposition technologies, etc. But it may also provide new tools for examining physical phenomena, and supply new possibilities to experimentally study and measure the thermal phenomena that are difficult to measure in the usual manner.

The analyses of heat transfer phenomena in the aforementioned applications offer new and unique areas of research. For example, due to the urgent needs for cooling electronic components and devices, microscale heat transfer technologies such as mini heat exchangers with flow channels having dimensions ranging from the order of several hundred to 0.1 μm have been developed. These microchannels and mini heat exchangers have found their application in reactors for modification and separation of biological cells, selective membranes and liquid/gas chromatographies. As pointed out by Yang and Zhang [1], the last decade of the twentieth century may witness rapid progress in the research of micro- and nanoscale transport phenomena, which has important applications in technologies such as microelectronics and thermal control for spacecraft. Microscale heat transfer and transport phenomena are expected to be quite different from that in ordinary situations, and it is of absolute necessity to understand these phenomena so as to develop related new high technologies. These mean that exciting possibilities still exist for exploiting new active fields—microscale heat transfer and creating innovative new applications, and tremendous challenges and opportunities also exist to conduct theoretical and experimental research on the fundamentals of heat transfer at the microscale level.

Tuckermann and Pease [2, 3] demonstrated that the electronic chip can be effectively cooled by means of water flow in microchannels fabricated on the circuit board on which the chips are mounted. Their results also indicated that the heat transfer coefficient of laminar flow through microchannels might be higher than that of turbulent flow through normally sized channels. Wu and Little [4], Pfahler *et al.* [5], and Choi *et al.* [6] noted that the flow and heat transfer characteristics of fluid through microchannels or microtubes departs from the thermofluid experimental results for conventional sized channels. These investigations provided substantial experimental data and considerable insight into some behaviors of the heat transfer and fluid flow in microchannels or microtubes without phase change. The studies conducted for the microscale heat transfer with phase changes are limited only for micro heat pipes, micro convection and evaporation in sessile drops and augmentation of boiling heat transfer on composite surfaces. To the present authors' knowledge, there are few investigations of the flow boiling of liquid through microchannels or microtubes reported in the literatures until now.

It is certain that some new behaviors will emerge from flow boiling in microchannels. This is of critical importance in several practical cases where boiling or phase change occurs, such as the design of high-capacity micro heat pipes for spacecraft thermal control, the cooling of electronic chips and devices, and the development of mini heat exchangers. In this paper, attempts were made to investigate experimentally the boiling heat transfer characteristics for subcooled liquid flow through microchannels.

NOMENCLATURE

A_1	area of the sides of the microchannel
A_2	area of the bottom of the microchannel
d_e	equivalent diameter
I	current
L	length of microchannel
Nu	Nusselt number
Pr	Prandtl number
Q	total applied power
q''	heat flux
Re	Reynolds number
T	temperature
U	voltage
u	flow velocity.

Greek symbol

μ	dynamic viscosity.

Subscripts

f	liquid
w	wall.

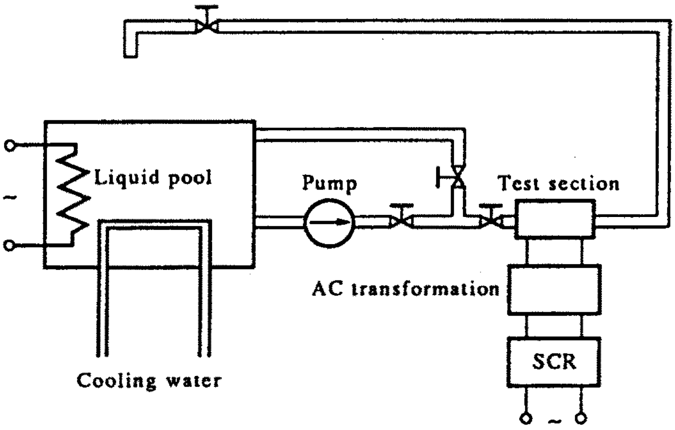

FIG. 1. Test facility.

2. TEST FACILITY AND EXPERIMENTAL DESCRIPTION

The test facility was set up as shown schematically in Fig. 1. It consists of a liquid pool, liquid pump, test section and the control valves for adjusting flow rate. The liquid temperature in the pool was kept constant by different valves for heating or cooling. Liquid was pumped into the circuit line. It flowed partly through the test section and returned back to the pool, and partly through the by-pass tube line back to the pool. An open-loop system was chosen for this investigation and the flow rate was determined by the weighing method. The measurement indicated that the flow rate might be determined within $\pm 0.1\%$.

The microchannels to be tested were machined on stainless steel plate 15 mm wide, 60 mm long and 2 mm thick. There are three identically sized microchannels uniformly distributed on the plate, as shown in Fig. 2. The cross-section of the channels was rectangular, 0.6 mm wide and 0.7 mm high. The test section was mounted on a frame structure. Microchannels were machined into the top surface of the stainless steel heating plate, with thermal and electrical insulating materials between the heating plate and frame.

Thermocouples for measuring liquid temperatures were located at the inlet and outlet of the test section. Six thermocouples were mounted on the back of the microchannel plate, two for each microchannel, one located at the upstream end and another one at the downstream end. The stainless steel plate on which the three microchannels were machined, was electrically heated by use of an electrical current transformer matched with an SCR voltage regulator to provide low voltage and high electric current, hence, higher heat flux along the test section. The input voltage and current were adjusted to control the applied heat flux.

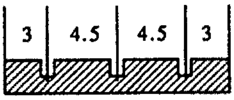

FIG. 2. Microchannel distribution.

Deionized water was employed as the working fluid. The water temperature was varied from 30 to 60°C, i.e. the liquid subcooling varied from 40 to 70°C at ambient pressure, and liquid velocity varied from 1.5 to 4.0 m s^{-1}. The test procedure was first to adjust the heat flux applied to the test section, and then to obtain the specified liquid velocity and subcooling condition. For each test, the system was allowed to reach its steady state, and the corresponding liquid flow rate, the temperature of the liquid and the plate wall, and the voltage and current were measured and recorded, respectively.

3. EXPERIMENTAL OBSERVATIONS AND DATA PROCESSING

Since the cover, as shown in Fig. 3, was made of Pyrex, the flow could be observed while the experiment was ongoing. The liquid flowed steadily through the microchannels and it seemed to be laminar flow, even for highest flow rate in the tested range. This situation did not change as the applied heat flux increased. When the plate was heated with its temperature rising up to higher than the saturation temperature at ambient conditions, approximately 104°C, vapor bubbles were observed on the end of the plate adjacent to the liquid mixing chambers near both the inlet and outlet of the test section, A and B. Yet no bubbles existed in all three microchannels. Three strings of small bubbles were observed in the mixing chamber at the outlet as the heated liquid flowed out of the microchannels. This phenomenon was greatly intensified by increasing the applied heat flux or plate surface temperature. However, there were still no bubbles observed in the microchannels throughout the experiment, even at the highest surface temperature of this investigation. There did not exist any observable difference between the three strings of small bubbles, which implied that the heat transfer and flow characteristics of three microchannels were about the same. Analyses of the measured data in detail and experimental boiling curves indicated that the experiment was actually carried out in the fully developed nucleate boiling regime when the plate surface temperature was a little bit over 100°C. As a result, we may conclude that no bubbles grow for flow boiling of liquid flowing through microchannels.

Experimental observations and analysis implied that bubble growth in liquid might be concerned with the scale of liquid bulk. If the scale of liquid bulk is large enough, bubbles could grow, otherwise, no bubbles grow and exist in liquid. Apparently, there is a critical scale for liquid bulk to determine whether bubbles can grow in liquid. This critical scale, we refer to as 'evaporating scale or space', is the minimum liquid bulk size at which bubbles can grow in liquid. If liquid bulk size is smaller than this evaporating scale, bubbles cannot exist in liquid. This concept may explain why no bubbles were observed in the microchannels, i.e. the scale of microchannels was smaller than the necessary evaporating space. However, more experiments are still needed to approve this new idea.

For the data processing, the plate surface temperature was corrected from the measured temperature on the back of the plate to the inner surface temperature of the microchannel. The flow rate and velocity through the microchannels were the average as calculated from the measured mass flow of all three microchannels. It was much more complicated to determine the actual heat flux from the side surfaces of the microchannel to the liquid flow, because all these three microchannels were machined on the heating plate, so that the thickness of the heating plate was not uniform. This resulted in heterogeneous heat generation of the heating plate. Consequently, the applied heat fluxes from the three sides of the channel, as shown in Fig. 4, were not uniform. Considering this heterogeneity, great care was taken to design the distribution of the three microchannels on the plate as shown in Fig. 2 and to manufacture them accordingly. The test inspection and corresponding analysis indicated that this distribution ensured that the applied heat flux from the microchannel sides was reasonably uniform. Hence, the applied surface heat flux can be calculated from total input power as

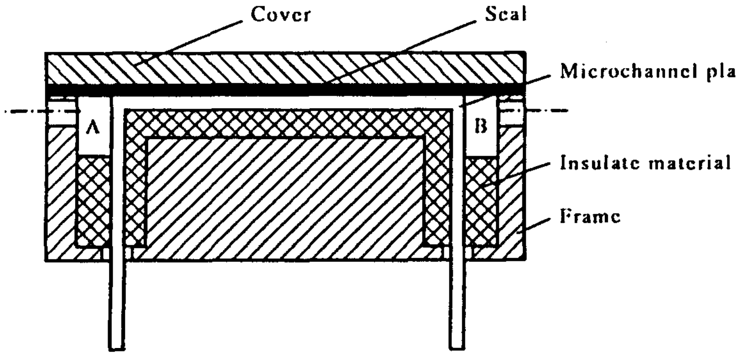

FIG. 3. Test section.

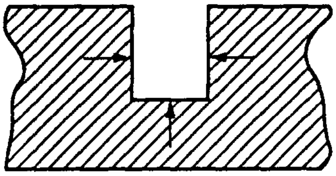

FIG. 4. Analytical model for heat flux.

$$q'' = \frac{Q}{3(2A_1 + A_2)} \tag{1}$$

where A_1 and A_2 are the area of the side and the bottom of the channel, respectively. The total applied power, Q, was measured or determined from the measured current, I, and the voltage, U, i.e.

$$Q = IU. \tag{2}$$

The measurement of the plate surface temperature and the experimental observations showed that there exists no observable difference between the fluid flow and heat transfer characteristics in the three microchannels. This implies that the method of data processing employed is reasonably acceptable. We will, therefore, focus our discussions on the measured results of the intermediate microchannel.

4. EXPERIMENTAL RESULTS AND DISCUSSION

Figure 5 illustrates the typical flow boiling curves measured experimentally. Very clearly, when the temperature of the plate was a little higher than the saturation temperature of water at ambient conditions, the flow boiling had reached fully developed nucleate boiling, and no apparent partial nucleate regime existed, although the liquid subcooling was sufficiently high. This is quite different from that of the well-known subcooled flow boiling in conventional sized tubes. These results also show that, for all test runs, these nucleate boiling data were distributed very close to an average boiling curve, i.e. the liquid velocity and subcooling do not influence the fully developed nucleate boiling, which is expected with the normal flow boiling.

For the single-phase liquid convection regime on the boiling curves, there exist two different cases in Fig. 5: (i) run 1 and run 4; (ii) run 2, run 3, run 5 and run 6. During the single-phase liquid convection region for run 2, run 3, run 5 and run 6, there existed a steep increase in the plate temperature, and then, similar to the normal flow boiling curve, the heat flux increased slowly with the temperature. However, the single-phase convection region for run 1 and run 4 shows the same tendency as the normal flow boiling curve. Figure 5 shows the distinct effects of liquid velocity and subcooling on the single-phase liquid convection region of $q'' - T_w$ curves, especially on the steep increase of q''. This shows that the decrease in liquid subcooling and/or increase in flow velocity may induce the steep increase of q'' on the $q'' - T_w$ curve. Comparing the experimental results of run 1 and run 6 in Fig. 5, a steep increase of q'' emerges on the single-phase convection $q'' - T_w$ curve of run 6, only because of a somewhat lower liquid subcooling and a little higher flow velocity. Run 1 and run 2 have only 0.1°C difference in liquid subcooling, so the steep increase of q'' is obviously induced by the increased liquid velocity of run 2. The steep increase of q'' still exists for run 5, but its flow velocity is not higher than that

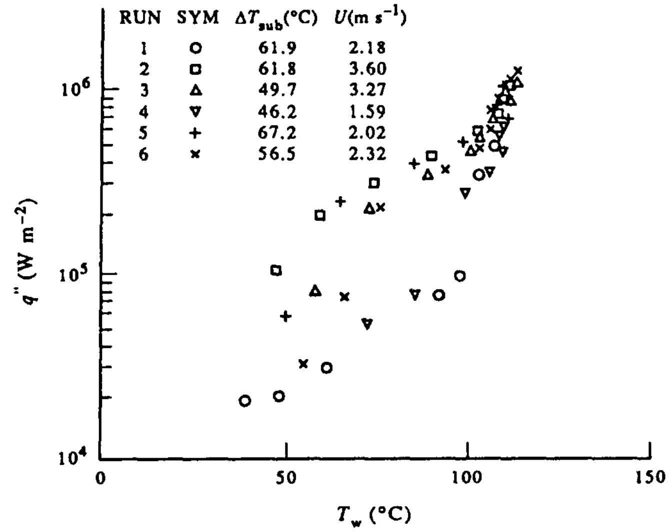

FIG. 5. Typical experimental results.

of run 1 and its liquid subcooling is not smaller than that of run 1. As a consequence, the steep increase of q'' may be affected by other factors besides flow velocity and liquid subcooling, and therefore further study is needed to clarify which parameters cause such variation.

Wu and Little [4] suggested a heat transfer correlation for single-phase liquid flow through microchannels with the range of $Re > 3000$ as

$$Nu = 0.00222 Re^{1.09} Pr^{0.4}. \tag{3}$$

The well-known heat transfer correlations of normally sized channels are given for laminar flow as [7]

$$Nu_f = 1.86\, Re_f^{1/3}\, Pr_f^{1/3} \left(\frac{d_e}{l}\right)^{1/3} \left(\frac{\mu_f}{\mu_w}\right)^{0.14} \tag{4}$$

and for transition flow

$$Nu_f = 0.116[Re_f^{2/3} - 125]\, Pr_f^{1/3} \left[1 + \left(\frac{d_e}{l}\right)^{2/3}\right] \left(\frac{\mu_f}{\mu_w}\right)^{0.14}. \tag{5}$$

The experimental results obtained here are compared with equations (4) and (5) plotted in Fig. 6, with the symbols '♂' and 'T' noting the data prior to and after the steep increase of q'', respectively. The heat transfer for microchannels is obviously higher than that for a normally sized channel when the steep increase of q'' occurs, and becomes lower prior to the steep increase in q''. This result may imply that there exists a transition of fluid flow and heat transfer regime from prior to after the steep increase in q''. From the comparisons shown in Fig. 6, the fluid flow and heat transfer performance for the cases that the steep increase in q'' has or has not occurred are different from those of conventional sized channels or tubes, perhaps due to the specific behavior of microchannel flow. The reasons promoting the transitions and the behaviors are not recognized or understood. The experiments of Wu and Little [4] indicated that the heat transfer was lower than that of a normally sized tube for the flow with $Re < 1000$. And generally speaking, the heat transfer performance in microchannels was better than in smooth tube for $Re > 1000$. The phenomenon for steep increase in q'' was not found in their investigation.

The experimental data obtained here are also compared with equation (3), as shown in Fig. 7. The experimental results here are smaller than the result predicted by equation (3), however, they give a good agreement in tendency when $Re > 3000$. The difference between the experimental and predicted results may result from the size of microchannels and heating way. The microchannels used here are bigger than those tested in Wu and Little's experiments from which equation (3) was correlated. The empirical correlation of equation (3) was really suggested for a channel heated from one side; however, the experimental data obtained here are those of microchannels heated from three sides, as shown in Fig. 4. The results for $Re < 3000$ give a deviation from the trend predicted by equation (3) and show the Reynolds number as not very important. It seems reasonable to expect that the thermocapillary flow or so-called Marangoni effect will play a significant role in this range, owing to the very small size of microchannels and increase the importance in capillary force with the decreasing importance of Reynolds number.

The test results for water flow through horizontal microchannels are compared with that of flow boiling through horizontal circular tubes with 9 mm inside diameter [7] in Figs. 8 and 9. For the microchannel results quoted in Fig. 8, prior to the steep increase of q'' in the single-phase liquid convection region, the

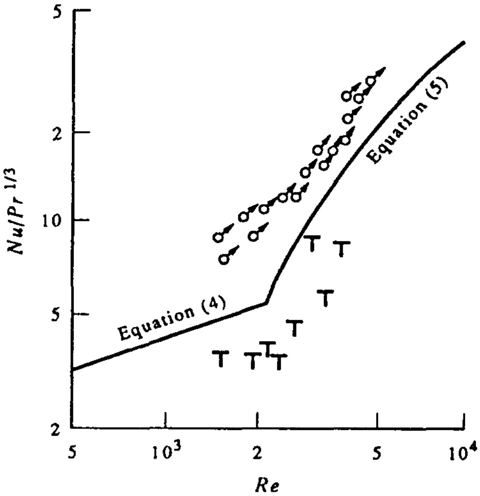

FIG. 6. Comparison of experimental data with the ordinary correlation—equations (4) and (5).

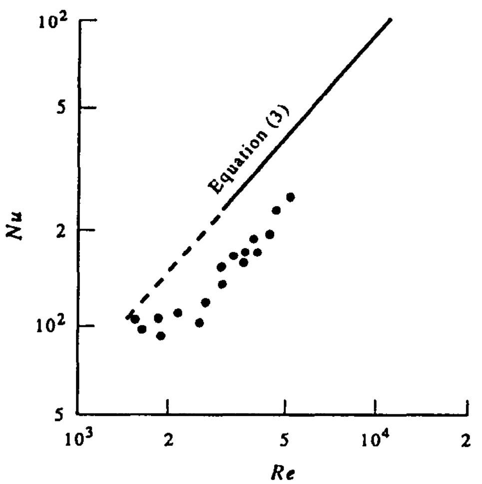

FIG. 7. Comparison of experimental data with the microchannels correlation—equation (3).

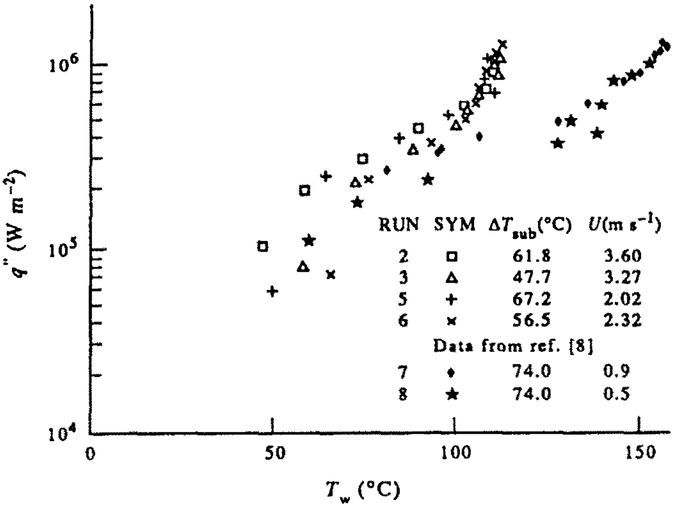

FIG. 8. Comparison of results from different experiments.

heat flux, q'', along the microchannel is smaller than that of the ordinary tube; and after the steep increase in q'', the heat flux for the microchannel is higher than that for the normally sized tube. In Fig. 9, all boiling curves are similar to each other, even with no steep increase in q'' even for the microchannel. The convection heat flux, q'', of the microchannel is smaller than that of the normally sized tube. However, the Reynolds number for all microchannel flow ranged from 1600–6000, while those for the tube flow [8] were reported as 5090 and 9160, respectively. These indicate that the laminar or transition liquid convection heat transfer in microchannels may reach or exceed the level of turbulent heat transfer in normally sized tubes.

Very clearly, the microchannels can effectively intensify the nucleate boiling heat transfer as shown in Figs. 8 and 9. No apparent partial nucleate boiling may exist in microchannels as mentioned above, and the plate surface superheat is much smaller than that of nucleate boiling curves measured for the ordinary tube [8].

5. CONCLUSION

The flow boiling characteristics of subcooled water flowing through microchannels with rectangular cross-section 0.6 × 0.7 mm were experimentally investigated. The results provide significant data and considerable insight into the behavior of the flow boiling and single-phase liquid convection heat transfer in

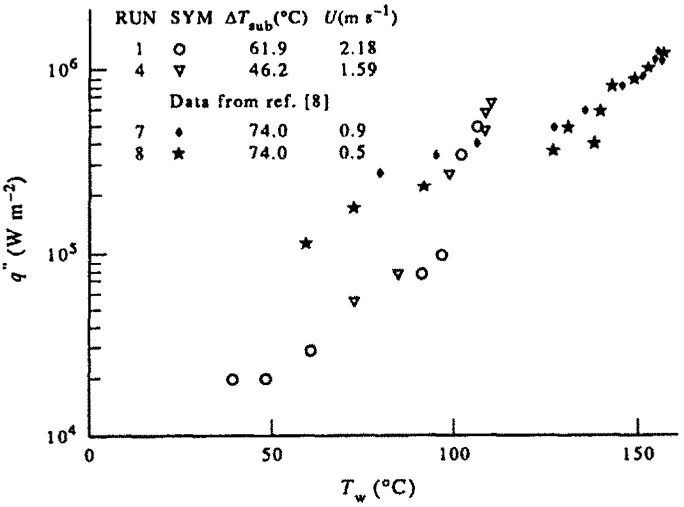

FIG. 9. Comparison of results from different experiments.

microchannels. It shows that the flow boiling and single-phase liquid convection characteristics are quite different from that of the conventional cases and can be summarized as the following:

(1) For single-phase liquid convection, a steep increase of q'' was observed. The liquid subcooling and flow velocity may affect the single-phase convection and the steep increase of q''. The single-phase convective heat transfer for microchannels would be smaller than that for normally sized channels prior to the steep increase of q'', and becomes higher than that of ordinary cases after the steep increase of q''.

(2) The nucleate boiling is greatly intensified, and the wall surface superheat for flow boiling may be much smaller than that of the normal case for the same wall heat flux. The velocity and liquid subcooling appear to have no obvious effect on the flow nucleate boiling.

(3) No partial nucleate boiling of subcooled water through microchannels has been observed or inspected for the transition from single-phase liquid convection to nucleate boiling. For this 'nucleate boiling' region, no bubbles were observed throughout the investigation. This is a very strange phenomenon.

REFERENCES

1. W. J. Yang and N. L. Zhang, Micro and nano-scale heat transfer phenomena research trends. In *Transport Phenomena Science and Technology 1992* (Edited by B. X. Wang), pp. 1–15. Higher Education Press, Beijing (1992).
2. D. B. Tuckermann and R. F. Pease, High-performance heat sinking for VLSI, *IEEE Elect. Dev. Lett.* **EDL-2**(5), 126–129 (1991).
3. D. B. Tuckermann and R. F. Pease, Optimized convective cooling using micromachined structure, *J. Electrochem. Soc.* **129**(3), C98 (1982).
4. P. Y. Wu and W. A. Little, Measurement of friction factor for the flow of gases in very fine channels used for microminiature Joule–Thompson refrigerators, *Cryogenics* **23**, 273–277 (1983).
5. J. Pfahler, J. Harley, H. H. Bau and J. Zemel, Liquid transport in micron and submicron channels, *J. Sensors Actors* **A21–23**, 431–434 (1990).
6. S. B. Choi, R. F. Barron and R. O. Warrington, Liquid flow and heat transfer in microtubes. In *Micromechanical Sensors, Actuators, and Systems*, ASME DSC-Vol. 32 (Edited by D. Choi *et al.*) pp. 123. ASME (1991).
7. S. M. Yang, *Heat Transfer*. People's Education Press, Beijing (1980).
8. J. G. He, Experimental investigation for the enhancing effect of polymeric additives on subcooled nucleate flow boiling with higher velocity, M.S. Thesis, Tsinghua University, Beijing (1988).