Pergamon

Int. J. Heat Mass Transfer. Vol. 38, No. 1, pp. 127–137, 1995

0017–9310/95 $9.50+0.00

0017–9310(94)00136–7

Experimental investigation of heat transfer in flat plates with rectangular microchannels

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(*Received* 12 *October* 1993 *and in final form* 31 *March* 1994)

Abstract—An experimental investigation was conducted to determine the heat transfer characteristics and cooling performance of rectangular-shaped microgrooves machined into stainless steel plates. Using methanol as the cooling fluid, grooves with different aspect ratios and a variety of center-to-center spacings were evaluated. The influence of liquid velocity, subcooling, property variations and microchannel geometric configuration on the heat transfer behavior, cooling performance, and heat transfer and liquid flow mode transition were analyzed experimentally. Measurements made to clarify the flow nucleate boiling attributes indicated an increased heat transfer rate and a behavior that was quite different from what typically occurs in larger tubes or channels, due to the relatively large portion of the surface area associated with the thin-film region.

INTRODUCTION

Microstructure technology has practical applications in many fields, including bioengineering and biotechnology, aerospace, mini-heaters and mini-heat exchangers, materials processing and manufacturing, etc. For example, microchannels and mini-heat exchangers with flow channels having dimensions ranging from several hundred microns to 0.1 μm have found application in bioreactors for the modification and separation of biological cells and selective membranes. In addition, microstructure technology may provide new tools for examining more closely physical phenomena and may result in the development of a method or methods by which thermal phenomena, often considered very difficult or impossible to investigate due to the small length scale, can now be studied. These suggest the types of creative and innovative applications that may result from a better understanding of flow characteristics and heat transfer behavior, and give some insight into the tremendous challenges that exist.

As pointed out by Yang and Zhang [1], the last decade of the twentieth century may witness rapid progress in research of micro- and nano-scale transport phenomena, with important applications in technologies such as microelectronics and spacecraft thermal control. Perhaps the most promising of these is the area of microelectronics, where the efficient thermal control of high-density, high-power electronic components and devices is of critical importance in the development of ultra-large-scale integrated (ULSI) circuits. With existing heat flux levels in ULSI circuits already in excess of 5×10^5 W m^{-2} and new circuit and device designs requiring dissipation requirements in excess of 10^6 W m^{-2} with low operating temperatures [2], new thermal control methods are necessary. Peterson and Ortega [3] and Yang and Zhang [1] have reviewed the recent research and resulting developments in the thermal control of high-density electronic components and devices, and concluded that the best near-term solution is to locate the heat sink as close to the device junction as possible. One way this can be accomplished is through the use of microchannels, such as those recently investigated by 3M [4].

Tuckermann and Pease [5, 6] demonstrated that electronic chips can be effectively cooled by means of water flow in microchannels fabricated within the silicon chips or the circuit board on which the chips were mounted, and the heat transfer coefficient for laminar flow through microchannels might be higher than that of turbulent flow through larger channels. Wu and Little [7], Pfahler *et al.* [8] and Choi *et al.* [9] noted that the flow and heat transfer characteristics of fluid flowing through microchannels and microtubes depart from the characteristics observed in previous experimental results in more conventionally sized channels. These investigations provided substantial experimental data and considerable insight into some

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NOMENCLATURE

A	area
h	heat transfer coefficient
H_c	depth of microchannel
I	current
L	length of microchannel
N	number of microchannels
Q	total applied power
q''	heat flux
r	modification coefficient accounting for heat loss
T	temperature
T_i	temperature sample, $i = 1, 2, 3 \ldots$
U	voltage
u	flow velocity
W	width of test plate
W_c	width of microchannel
W_t	distance between two microchannels.

Subscripts

f	liquid
w	wall.

characteristics of heat transfer and fluid flow in microchannels or microtubes without phase change.

A critical review of the literature indicates that only a limited number of investigations of forced-flow convection in microchannels or microtubes without phase change have been conducted and even fewer with phase change. These studies of microscale heat transfer with phase change are limited to micro-heat pipes, micro-convection, evaporation in sessile drops, and augmentation of boiling heat transfer on composite surfaces [4]. As evidenced by the aforementioned studies, the behavior and characteristics of microscale heat transfer and transport phenomena may be quite different from what typically occurs in larger scale situations. In the current investigation, the heat transfer characteristics of a subcooled liquid flowing through microchannels both with and without phase change was investigated and the cooling performance was measured in an attempt to determine if the flow characteristics and heat transfer behavior actually do differ from what has been observed in larger-scale structures.

TEST FACILITY AND EXPERIMENTAL DESCRIPTION

The test facility illustrated in Fig. 1 consists of a liquid pool, liquid pump, test section and control valves for adjusting the liquid flow rate. Throughout the tests, the liquid pool or reservoir was maintained at a constant temperature through the use of heated and chilled water and an external heat exchanger. The working fluid was pumped from the liquid pool through the test section and the flow rate was controlled by means of a bypass loop, which allowed some of the liquid to return back to the pool. This bypass loop also insured good mixing and helped maintain a more uniform temperature in the liquid pool. Use of this particular test configuration provided a carefully controlled fluid temperature with a known and constant flow rate.

As shown in Fig. 2, the test sections utilized in the current investigation were fabricated from stainless steel plates, 18 mm wide and 125 mm long, into which a series of parallel microchannels were machined. By varying the microchannel groove width, center-to-center distance and number of microgrooves (either four or six), a total of six different microchannel groove configurations were formulated, manufactured and evaluated. For each configuration, the region of the plate where the microchannels were machined was 2 mm thick and the length of the microchannels was 45 mm. In addition, the microchannels all had a rectangular cross-section, 0.7 mm deep. The geometrical parameters for the six different test sections are summarized in Table 1.

The upstream and downstream liquid temperatures were measured using two T-type thermocouples (AWG 36), one at the inlet and one at the outlet of the test section. The axial wall surface temperature distribution was measured by six thermocouples mounted on the back of each test plate, three at the upstream end and three at the downstream end. The stainless steel plate into which the microchannels were machined was electrically heated by directly connecting the test sections to an a.c. electrical current transformer matched with an SCR voltage regulator that provided low voltage and high electric current. In this way, a highly uniform heat flux was applied along the entire length of the test section and the input voltage and current could be adjusted to control the applied heat flux. This type of heat generation closely approximates the type of heat generation occurring in electronic components and devices.

Methanol was used as the working fluid. The average liquid temperature was varied from 14 to 19°C (i.e. the liquid subcooling varied from 45 to 50°C at ambient pressure), and the liquid velocity ranged from 0.2 to 1.5 m s^{-1}. The test procedure utilized was to first adjust the heat flux applied to the test section, and then adjust the flow rate and liquid pool temperature to obtain the specified liquid velocity and subcooling conditions. For each test, the system was allowed to reach steady state prior to measuring and recording the liquid flow rate, liquid and plate wall temperatures and voltage and current.

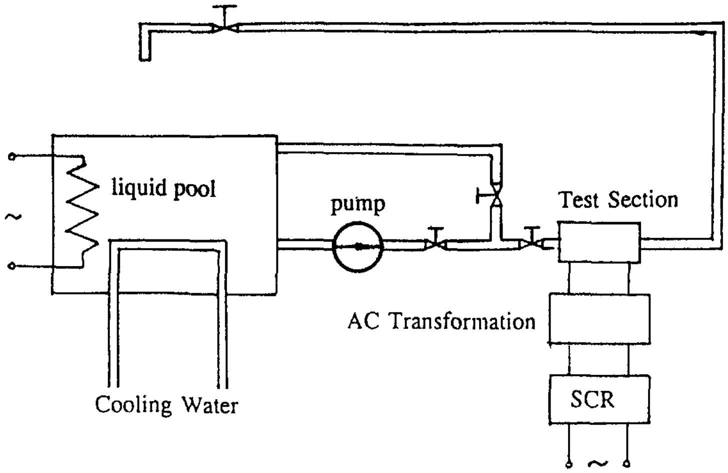


Fig. 1. Test facility.

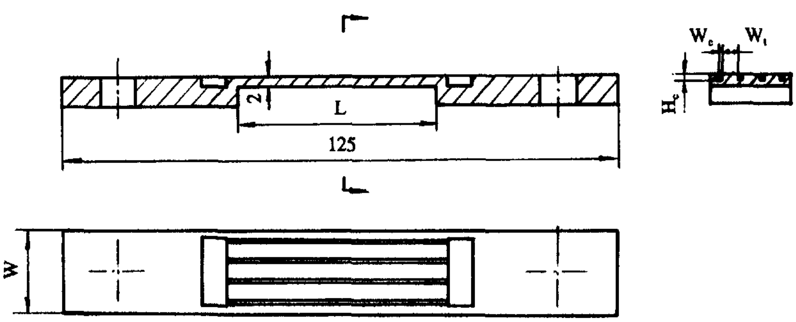


Fig. 2. Test section (dimensions in millimeters).

DATA ANALYSIS

The heat flux generated within the test section was determined from the measured power input, i.e.

$$q'' = \frac{Q}{A}. \tag{1}$$

The microchannel structure area, A, and the applied heat generation rate, Q, were calculated as follows:

$$A = LW \quad \text{and} \quad Q = rIU \tag{2}$$

where L and W are the length and width, respectively, of the region in which the microchannels were fabricated. The wall heat flux shown in equation (1) was calculated by measuring the voltage and current, U and I, to determine the internal heat generation and then subtracting conduction and other heat losses. These losses were estimated assuming natural convection on the outside surface of the insulation which surrounded the test section under steady-state flow conditions without phase change. The modification coefficient, r, in equation (2) can be used to account for all these heat losses and was determined from an experimental energy balance. Prior to conducting the tests, the measurement technique was verified by comparing the calculated value with the sensible heat gained by the liquid between the inlet and the outlet of the microchannels. These values consistently had a variation of less than $\pm 5\%$.

The local heat transfer coefficients were evaluated as

$$h(x) = q''/[T(x) - T_f], \tag{3}$$

where T_f denotes the liquid temperature measured at

Table 1. Geometric parameters of test section

Test section	L [mm]	W [mm]	W_c [mm]	W_t [mm]	H_c [mm]	N
No. 1	45	18	0.6	3.6	0.7	4
No. 2	45	18	0.4	3.8	0.7	4
No. 3	45	18	0.4	2.4	0.7	6
No. 4	45	18	0.2	4.0	0.7	4
No. 5	45	18	0.2	2.6	0.7	6

the inlet and $T(x)$ was determined from the three wall temperatures measured at identical axial positions along the flow direction. To compare the cooling characteristics of the various test configurations, the heat transfer coefficient was calculated at the downstream end of the microchannels.

To estimate the uncertainties in the calculation of the wall heat flux and the heat transfer coefficients, the expressions given in equation (2), can be combined, to express the wall heat flux as

$$q'' = rIU/(WL). \tag{4}$$

Combining equations (3) and (4) yields

$$h(x) = \frac{rIU}{WL\,[(T_1+T_2+T_3)/3-T_f]} \tag{5}$$

where T_1, T_2 and T_3 are the temperatures from the three thermocouples mounted on the back of the plate near the downstream end. The uncertainties of both the current and voltage measurements were $\pm 0.2\%$. The uncertainties in the axial location was less than $\pm 1\%$. Type-T (copper–constantan) thermocouples with an uncertainty of $\pm 0.4\%$ were utilized to measure both the fluid and wall temperatures and the uncertainty of the thermocouple readout was $\pm 0.3\%$. Other uncertainties were estimated as follows: 4% for the average wall temperature, which indicates the variation in the readout from the three thermocouples, 2% for the a.c. current and voltage due to small fluctuations, 4% for the flow rate, and 6% for the liquid velocity. Combining these values, the maximum uncertainty for the wall heat flux was estimated to be approximately $\pm 8\%$, and the maximum uncertainty for the heat transfer coefficient, $h(x)$, was $\pm 10\%$.

EXPERIMENTAL RESULTS AND DISCUSSION

Tests were conducted at various power levels and resulted in data for both single-phase and flow boiling conditions. Figure 3(a–e) illustrates the typical experimental results for the various test sections and compares the relationship between the wall heat flux and the wall temperature for the six different microchannel configurations. The data shown in the figures include both single-phase flow convection and fully developed flow nucleate boiling. Because of the difficulty in determining the boundaries between single-phase flow convection and the onset of nucleate boiling (i.e. partial boiling), the only two-phase regime evaluated was fully-developed flow, nucleate boiling.

Single-phase forced convection

The experimental data for single-phase flow convection, shown in Fig. 3(a–e), all indicate a characteristic shape that is convex upward, and there exists a rapid increase in the wall heat flux value less than approx. 10 000 W m^{-2} when plotted as a function of the wall temperature. While, in general, the characteristic shape of these curves is not significantly different from what typically occurs for forced convection in earlier studies on larger channels and tubes, this rapid increase in the heat transfer coefficient at low wall temperatures indicates a transition of the liquid flow regime or heat transfer mechanism. One possible explanation may be the transition between laminar and turbulent liquid flow. By comparing the data in each test section, particularly the results shown in Fig. 3(a–d), it is easy to show that the liquid velocity had a significant influence on the flow mode or heat transfer mechanism. The sharp rise in heat flux with respect to wall temperature occurred at lower wall temperatures for higher velocities. It is unclear, however, how liquid temperature or subcooling affects the transition from these results, but it is anticipated that this parameter is still quite important and must be examined in future investigations. The size and possibly the number of microchannels also has a significant effect on the transition. Comparing Fig. 3(a–e) it is also apparent that the abrupt increases in heat flux were shifted to the right for the plates with smaller microchannels or those with a higher number of channels, in spite of variations in the liquid velocity and/or temperature.

The values of the heat transfer coefficient determined from equation (5) are illustrated in Fig. 4. The variation of the heat transfer coefficient, h, with respect to wall temperature, T_w, initially appears to be very complicated, particularly in Fig. 4(a–d). However, these data can be divided into three fairly distinct regions, which are more clearly apparent in Fig. 4(e). In the first of these, the forced liquid convection region, the behavior of the heat transfer coefficient vs temperature curve is apparently related to the steep increase in the heat flux data. However, at lower temperatures, especially those shown in Fig. 4(b–e), the heat transfer coefficient, h, increases sharply for some test runs, and decreases for others. This variation in the heat transfer coefficient indicates that the transitive behavior is quite elaborate in nature. In the middle region, the heat transfer coefficient, h, is nearly independent of the wall temperature. The third region occurs in the fully developed nucleate flow boiling regime. In this region, which occurs at significantly higher wall temperatures, the convection coefficient, h, increases very rapidly to large values with only small increases, and in some cases even a slight decrease, in the wall temperature. This behavior is quite different from the behavior observed in the single-phase forced-flow convection regime.

Because of the limited range of test parameters, the data obtained in this investigation cannot completely explain the fundamental physical phenomena but can provide a significant insight into the behavior and trends. Most importantly, it appears that, because of the extremely small channel dimensions, the heat addition process creates a large change in the liquid temperature not typically observed in larger channels. As a consequence, the thermophysical properties of the flowing liquid change dramatically over the length of the microchannel prior to the occurrence of any phase change. For example, the Reynolds number, *Re*, for the experiments described here was usually about 700–1000 at the inlet, and about 1800–2000 or larger at the outlet. This implies a Reynolds number that more than doubles over the length of the microchannel. Clearly, variations in the heating rate or wall temperature are the cause of these relatively large changes in the Reynolds number. For fluids flowing through microchannels, Wu and Little [7] found tran-

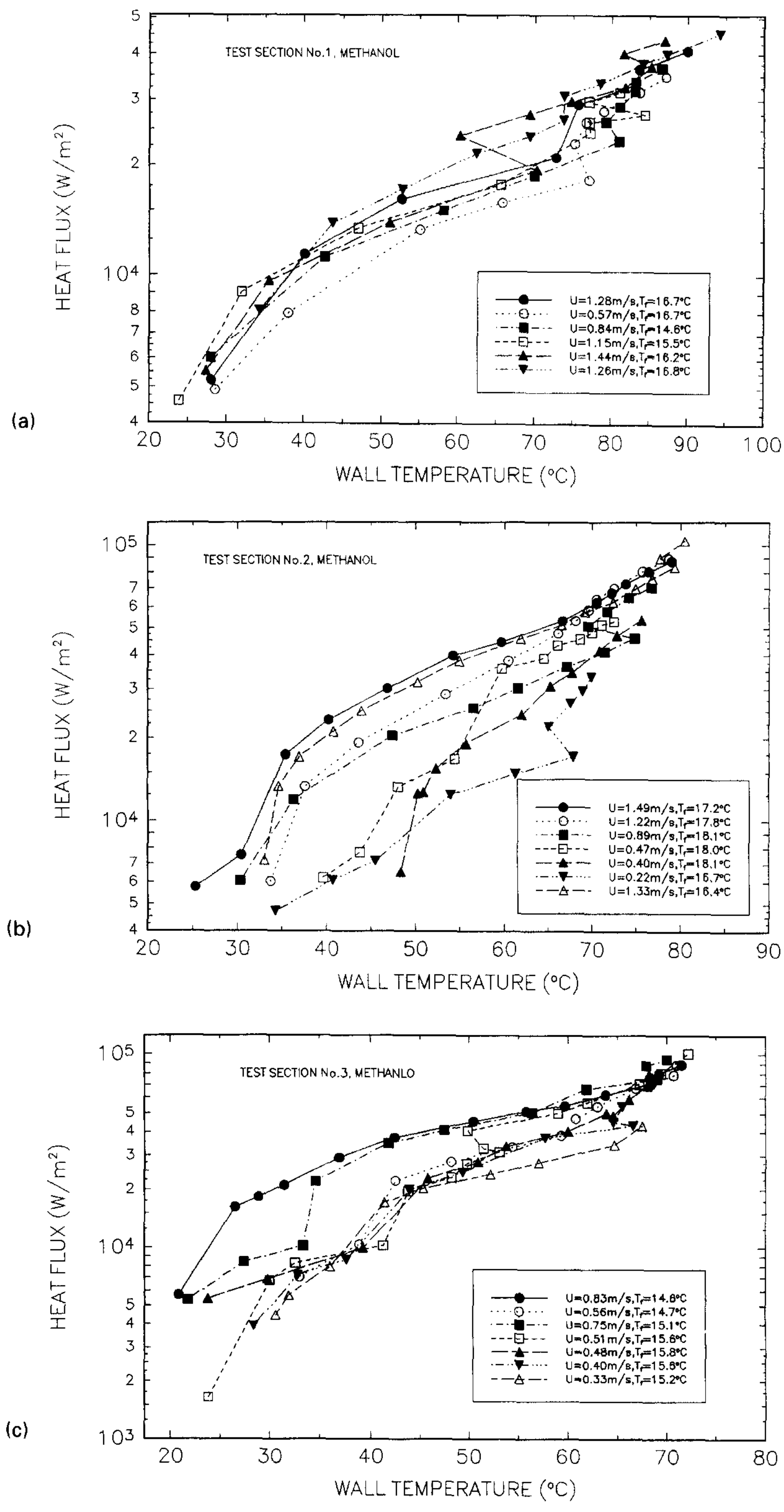


Fig. 3. Heat flux variation with wall temperature.

(d)

(e)

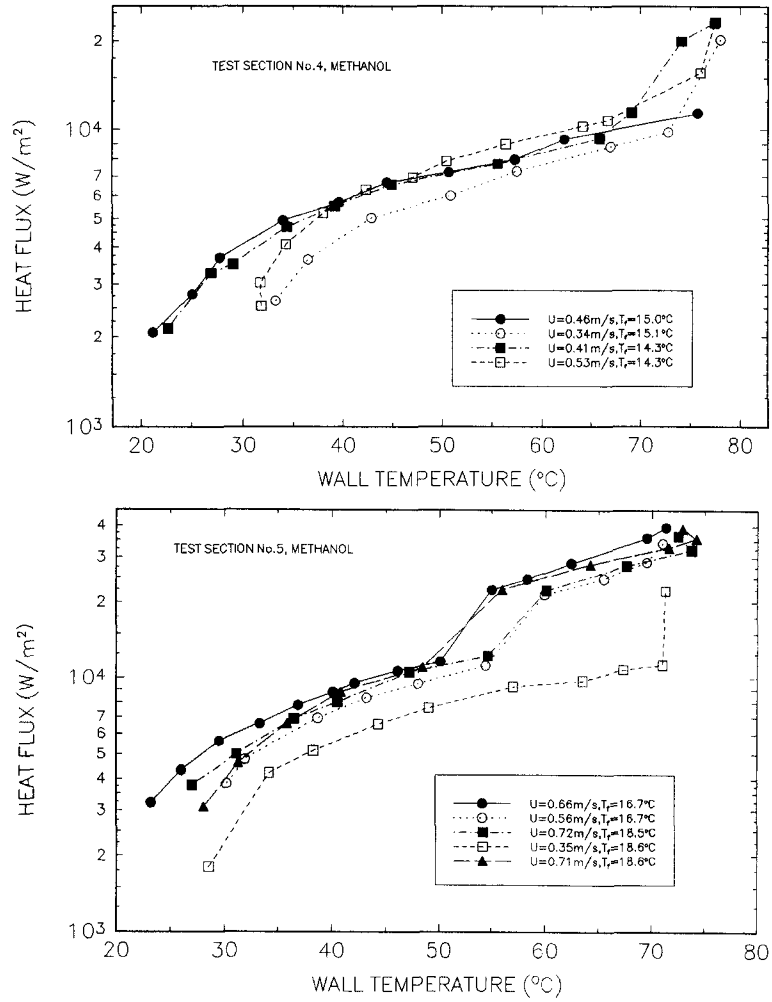


Fig. 3—*continued.*

sition to turbulence at much lower Reynolds numbers than for larger, more conventionally sized channels, which supports the hypothesis presented here that the rapid change in the heat flux and heat transfer coefficient are the result of changes in the flow regime. Hopefully, additional experimental investigations can provide further explanation of the observed phenomena.

Liquid flow velocity, temperature and degree of subcooling are all important parameters for heat transfer and cooling performance of both large-scale and microchannel structures. Generally speaking, the heat transfer or cooling performance is improved by increasing the liquid velocity. This can easily be seen by comparing the experimental data presented here, for instance the results shown in Figs. 3a and 4b. A more detailed comparison implies that changes in the inlet velocity significantly alter the profiles of the wall heat flux or heat transfer coefficient vs the wall temperature curves, and results in the transition from laminar flow to transition flow or from transition flow to turbulent flow. The characteristics of the wall heat flux vs the wall temperature curves seemed to vary from those normally observed for conventionally sized channels, to a convex upward curve with increases in the liquid velocity, as illustrated in Fig. 3. The flow or heat transfer transition occurred at lower wall temperatures when the velocity was higher, and heat transfer exhibited a behavior in the middle region in which heat transfer coefficient, h, was almost independent of wall temperature, as discussed above. It is also clear that, for a given wall temperature, microchannels can significantly enhance heat transfer performance. The results in Figs. 3 and 4 clearly support these conclusions.

The effects of liquid subcooling on the heat transfer or cooling performance of microchannels are similar to those of liquid flow velocity. For increasing liquid subcooling, the characteristic wall heat flux vs the wall temperature curve tends to assume a convex upward shape. The flow or heat transfer transition occurs more sharply at lower wall temperatures, i.e. heat transfer performance or cooling is more highly augmented at lower temperatures, as shown in Fig. 3. Accordingly, the heat transfer coefficient increases with decreasing liquid temperature (see Fig. 4). It is apparent that the liquid temperature or degree of subcooling also plays an important role in determining the heat transfer performance or cooling augmentation. Actually, the experimental data analyses conducted in the current investigation indicate that the effect of liquid temperature/subcooling on heat transfer is greater than that of liquid velocity.

Figure 5 compares the results obtained from the

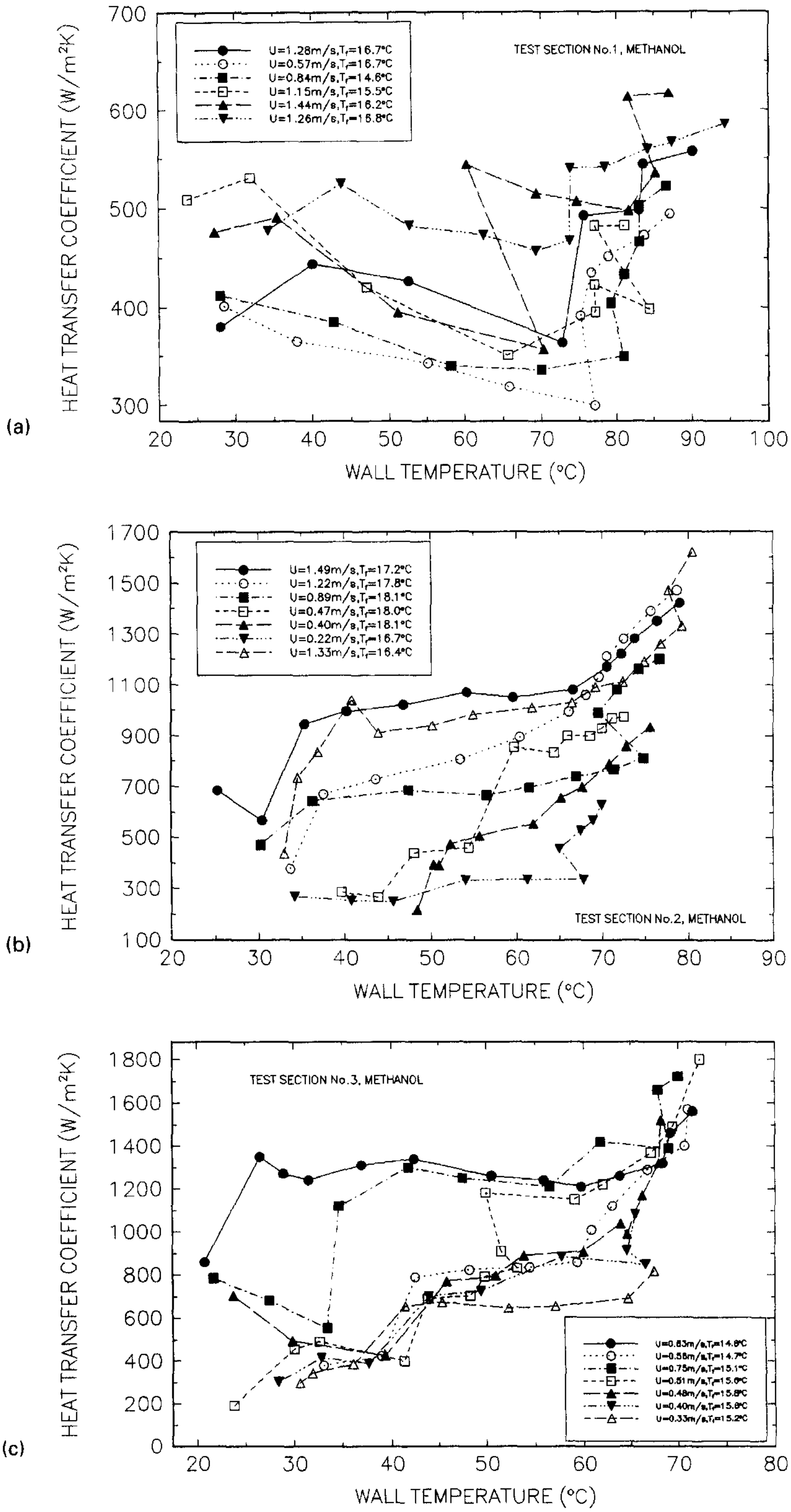
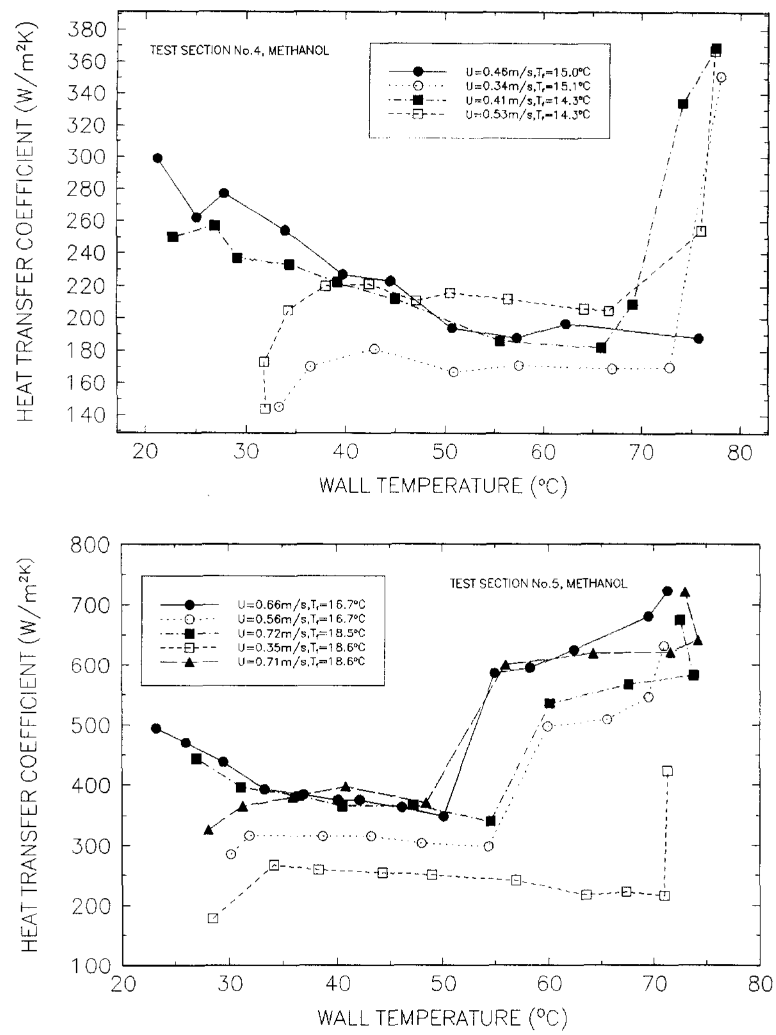


Fig. 4. Variation of heat transfer coefficient with wall temperature.

(d)



(e)

Fig. 4—*continued.*

test articles having the same number, but different sizes of microchannels under the condition of about the same flow velocity. In Fig. 5a, the heat transfer coefficient of the section with the larger channels, No. 1, is less than that of section with the smaller channels, No. 2. Conversely, in Fig. 5(b and c) the heat transfer performance for the section with large channels is better than that obtained for the section with the smaller channels. Because the channels on sections Nos 2 and 4 are same as those on sections Nos 3 and 5, respectively, the 0.4×0.7 mm channels appear to be best suited for all the cases tested. Actually, the heat transfer coefficient of both sections Nos 2 and 4 was decidedly higher than that of other tested sections, though variations in the liquid temperature diminished this enhancement in Fig. 5(b and c). The foregoing discussion indicates that there may exist an optimum channel size and spacing, which is near that utilized in sections Nos 2 and 3.

Figure 6(a and b) illustrates the results of microchannels of the same cross-section but different spacing (i.e. four as opposed to six). As would be expected, regardless of the heat transfer mode, forced-flow convection with or without phase change, the heat transfer performance or cooling capacity was noticeably augmented by increasing the number of channels. In Fig. 7, the difference in performance resulting from variations in channel size and number is illustrated for liquid flowing with nearly identical velocities. Clearly the heat transfer of section No. 2 is much better than that of section No. 1, while the mass flowrate through section No. 1 is nearly 50% greater. In addition, the heat transfer coefficient of section No. 3 is approximately twice than that of section No. 2, and three times greater than that of section No. 1. Clearly, enlarging the number of channels has a greater effect on the overall heat transfer performance and is the most important parameter, followed to a somewhat lesser degree by optimization of the channel size and geometrical configuration.

Flow boiling

As discussed previously, Figs. 3 and 4 actually include some data within the flow nucleate boiling

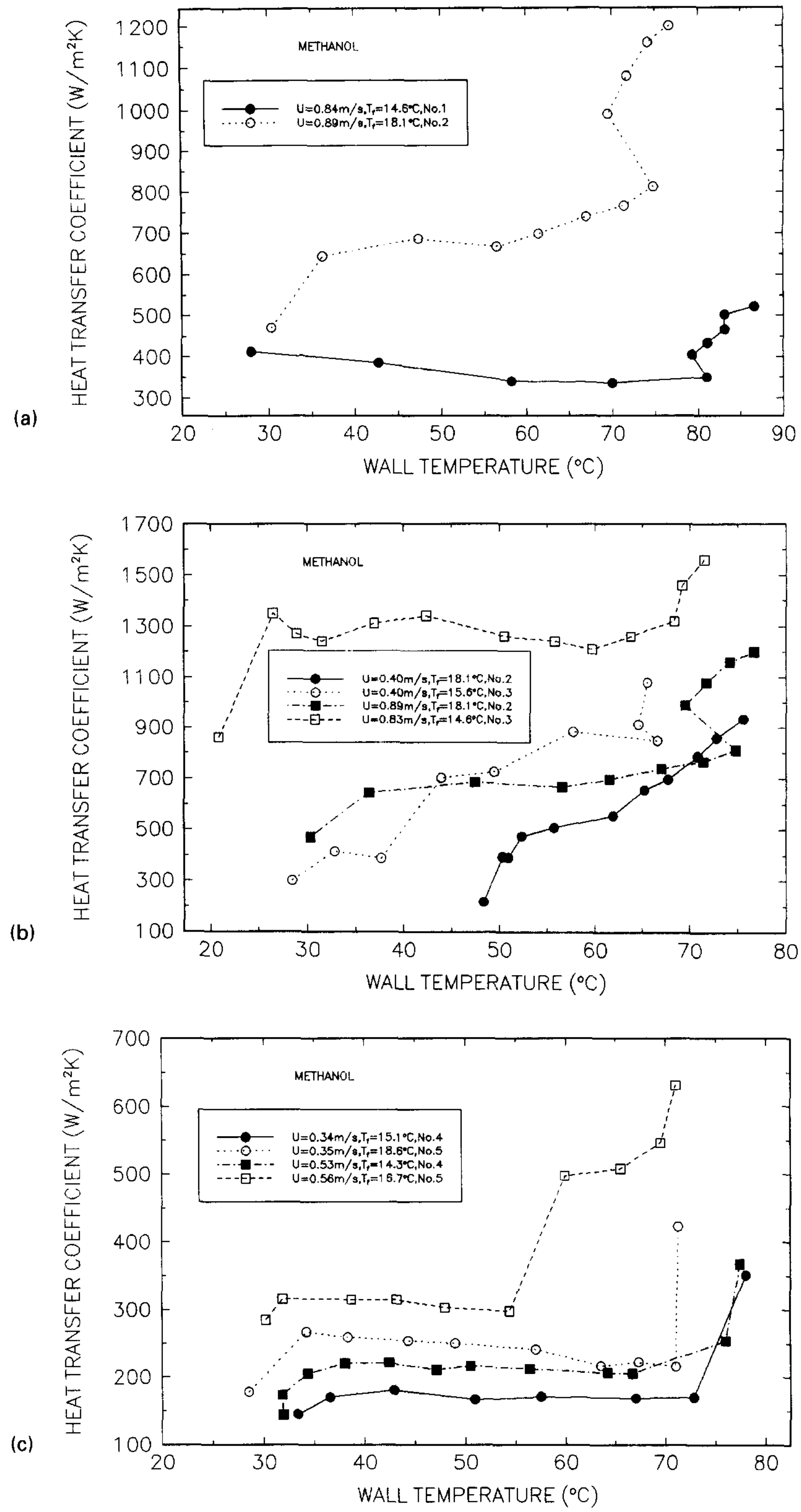


Fig. 5. Effect of microchannel size on heat transfer.

regime. With a sufficiently high superheat, flow boiling was initiated at once and immediately developed into fully developed nucleate boiling. Hence, although the liquid subcooling was sufficiently great, no apparent partial nucleate regime exists. Because of the rapid change to fully developed nucleate flow boiling, the heat removal rate and performance was greatly enhanced. The behavior, however, is again sig-

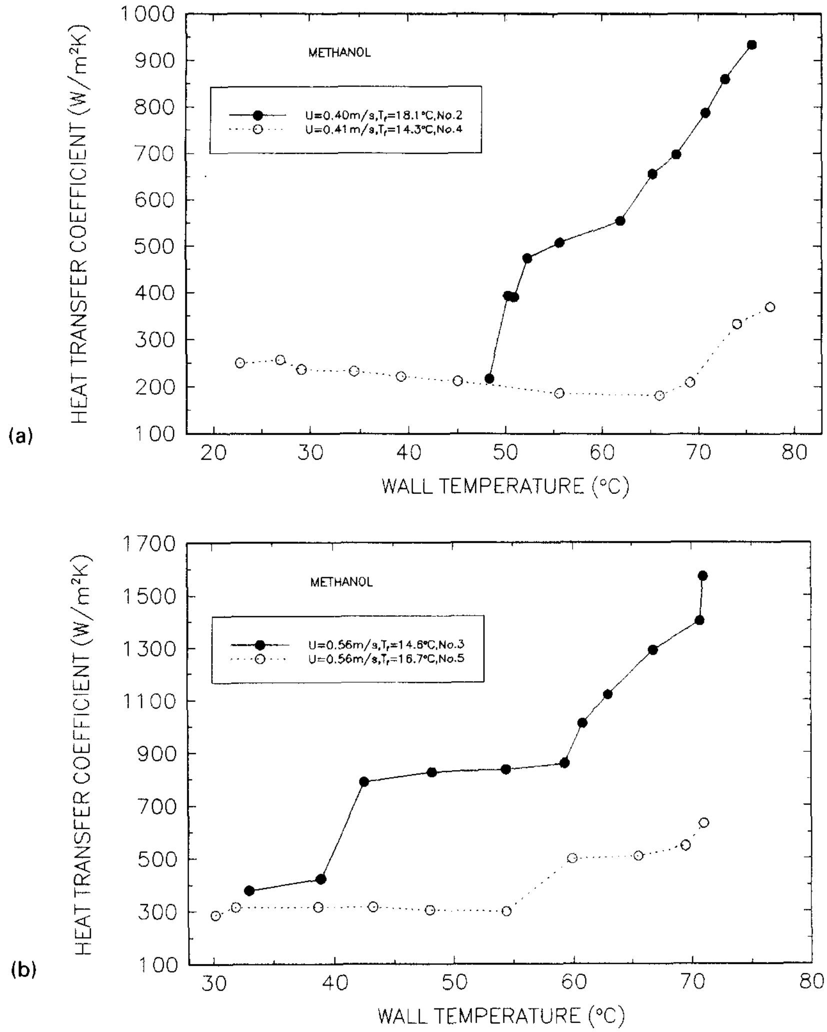


Fig. 6. Effect of microchannel number on heat transfer.

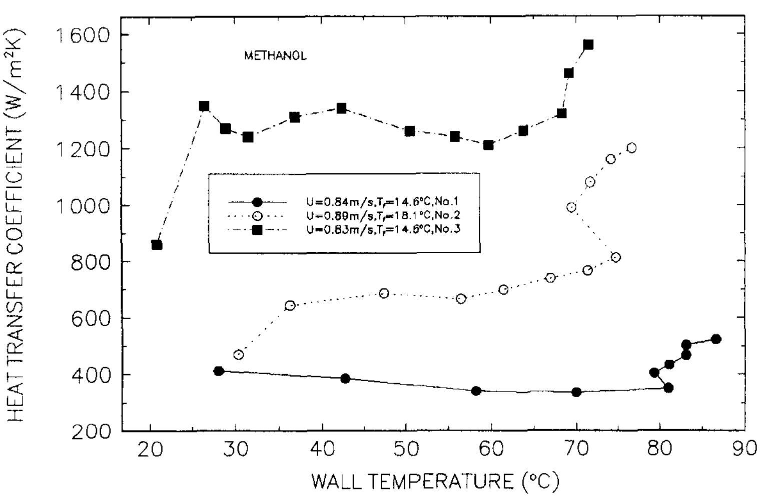


Fig. 7. Comparison for effect of channel number and size on heat transfer.

nificantly different from that observed in subcooled flow boiling in larger channels and tubes. The characteristic boiling curve does exhibit a slight hysteresis phenomenon, i.e. the plate temperature decreases slightly while the heat flux rapidly increases to the initial boiling point, as illustrated in Figs. 3 and 4. In addition, the experimental data for fully developed flow nucleate boiling illustrated in Figs. 3 and 4 fall very close to an average boiling curve, i.e. the liquid velocity and subcooling seem not to influence the fully developed nucleate boiling, which is the case for "normal" flow boiling. The experiments described here indicate that the liquid velocity and temperature (or subcooling) have little effect on the boiling initiation for flow in the microchannels evaluated here, but that higher liquid temperatures (or greater liquid subcooling) promoted and increased velocity and suppressed the initiation of flow boiling.

CONCLUSIONS

The heat transfer performance and cooling characteristics of subcooled liquid flowing through microchannels with rectangular cross-sections were investigated experimentally. The results provide significant data and considerable insight into the behavior of the heat transfer and cooling performance with microchannels machined onto a plate. The results show that the liquid velocity, liquid subcooling, liquid properties and geometry of the microchannels all have significant influence on the heat transfer performance, cooling characteristics and liquid flow mode transition.

The results and conclusions can be summarized as follows: first, there exists a heat transfer or liquid flow mode transition region for single-phase flow through microchannels, beyond which the heat transfer coefficient is nearly independent of the wall temperature. These transitions are a function of the heating rate or wall temperature conditions within the microchannel itself. This transition is also a direct result of the large liquid temperature rise in the microchannels which causes significant liquid thermophysical property variations and, hence, significant increases in the relevant flow parameters, such as the Reynolds number. As a result, the liquid velocity and subcooling were found to be very important parameters in determining the point or region where this transition occurs. Second, nucleate boiling is intensified and the wall surface superheat for flow boiling may be significantly smaller in microchannel flow than in the "normal" case for a given wall heat flux. The velocity and liquid subcooling appear to have no obvious effect on the flow nucleate boiling: however, they may significantly alter the inception or onset of flow nucleate boiling. Finally, the heat transfer performance and cooling characteristics of the microchannels improved as the liquid flow velocity and/or subcooling increased. The heat transfer characteristics performance and cooling characteristics were also enhanced as the channel number increased and size of the microchannels approached an optimum geometry.

It is clear from the results of the current investigation that, if selected properly, the correct combination of these parameters can provide significant improvements in the thermal performance and heat removal rate for a wide variety of applications, particularly those where space is limited such as electronic circuits and devices.

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