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The effect of thermofluid and geometrical parameters on convection of liquids through rectangular microchannels

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INTRODUCTION

The research into microscale flow and heat transfer phenomena conducted by Tuckermann and Pease [1, 2], Wu and Little [3, 4], Pfahler *et al.* [5, 6], Choi *et al.* [7] and Weisberg *et al.* [8] provided substantial experimental data and considerable evidence that the behavior of fluid flow and heat transfer in microchannels or microtubes without phase change may be substantially different from that which typically occurs in larger more conventionally sized channels and/or tubes. In an attempt to clarify some of the questions surrounding this issue, Peng and Wang [9, 10] and Peng *et al.* [11] recently investigated the heat transfer characteristics of liquid flowing through microchannel structures. In that work, the heat transfer and flow mode conversions for single-phase convection in microchannels, and the transitions induced by or associated with variations in the liquid thermophysical properties due to the increases in the liquid temperature through the heated microchannels, were studied. Wang and Peng [12] also studied the forced flow convection of liquid in microchannels both with and without phase change experimentally. It was found that fully-developed turbulent convection was initiated at Re = 1000–1500, and the heat transfer behavior in the laminar and transition regions was quite unusual and complicated.

In the present work, a series of experiments with several different microchannels were conducted to examine the single-phase convective heat transfer and better understand the fundamental physical phenomena associated with this type of flow situation. The physical fundamentals and nature of the flow and heat transfer phenomena in microchannels were examined to determine experimentally the influence of the liquid flow, thermal conditions and microchannel size on the flow and forced convective heat transfer characteristics for single-phase water flowing through microchannels.

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EXPERIMENTS

The test facility utilized in the current investigation and the experimental procedure have been described in detail by Wang and Peng [12]. The geometric parameters for the four different microchannels utilized are summarized in Table 1. Both water and methanol were employed as the working fluid and, for various tests, the temperature was varied from 11 to 28°C and 12–20°C (i.e. the liquid subcooling varied from 72 to 89°C and 45–53°C at ambient pressure) for water and methanol, respectively. The liquid velocities evaluated ranged from 0.2 to 2.1 m s^{-1} for water and 0.2–1.5 m s^{-1} for methanol.

As discussed previously, the applied surface heat flux of microchannel was calculated from the total input power as

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

Table 1. Geometric parameters of the test sections

Test	L [mm]	W [mm]	W_c [mm]	H_c [mm]
Section 1	45	18	0.8	0.7
Section 2	45	18	0.6	0.7
Section 3	45	18	0.4	0.7
Section 4	45	18	0.2	0.7

NOMENCLATURE

A	area
D_h	hydraulic diameter
h	heat transfer coefficient
k	conductivity
N	number of microchannel
Nu	Nusselt number
Q	total applied power
q''	heat flux
Re	Reynolds number
T	temperature
U	flow velocity.

Greek symbol

ν	viscosity.

Subscripts

f	liquid
w	wall.

where $Q = rIU$. The local heat transfer coefficients were evaluated as

$$h(x) = q''/[T_w(x) - T_f] \tag{2}$$

where T_f denotes the liquid temperature measured at the inlet. The local temperature, $T_w(x)$, was determined from the three wall temperatures measured at the same x position along the flow direction. Two dimensionless parameters were used in the analysis of the experimental results, the Nusselt number,

$$Nu = \frac{h(x)D_h}{k_f} \tag{3}$$

and the Reynolds number,

$$Re = \frac{UD_h}{\nu_f}. \tag{4}$$

The experimental uncertainties was estimated exactly as same as that given by Wang and Peng [12].

GENERAL PERFORMANCE

The experimental results for methanol are presented in Fig. 1. As illustrated, the laminar region is in the region for Re less than about 400. This number is a little larger than that of 300 given by Wang and Peng for water [12]. However, the results clearly indicate that Nu may decrease with an increase in Re and Nu is apparently associated with other factors. For $Re > 400$, the results are similar to those obtained by Wang and Peng [12]. When Re is greater than 400 but less than 1000, i.e. $1000 > Re > 400$, the data diverge from a single straight line configuration and form two fairly distinct regions, one which represents the turbulent regime and another which represents the laminar flow regime. In this range of Re, the relation of Nu to Re is quite complicated. For the situation where Re is greater than about 1000, all of the experimental data fall along a fairly narrow region that traces a straight line in which Nu increases with increasing Re (the right-hand region of Fig. 1). The results shown in Fig. 1 show the existence of three clearly distinct flow regimes and again indicate that laminar flow exists for Re less than 400, a transition regime in the region $400 < Re < 1000$, and a fully developed turbulent regime for the region where $Re > 1000$.

Figure 2 illustrates the typical experimental results for single-phase convective heat transfer for the microchannel plates evaluated (see Table 1) in terms of the heat transfer coefficient as a function of the wall surface temperature. It is not difficult to see that there are some differences in the change of heat transfer coefficients, h, with wall surface temperature, T_w, for the different experiments. The variation of heat transfer coefficient can be divided into three distinct regions, the region on the left side of the graph, in which the coefficient decreases slightly, the middle region where a great variation emerges, and a region of generally small monatomic increases in the heat transfer coefficient. The variations in the functional relationship between the heat transfer coefficient and the wall temperature, h and T_w, respectively, implies a change in the heat transfer mechanism which corresponds to the different heat transfer regimes identified in Fig. 1. Peng *et al.* [9–12] discussed similar phenomena observed in previous experiments. Prior to the steep increase in the heat transfer coefficient laminar flow might be expected. The conversion to the transition regime occurs where the heat transfer coefficient dramatically increases with the wall temperature.

The variation of the heat transfer coefficient with the wall temperature is also a consequence of the small size of the channel. Because of the extremely small size, the heating through the channel creates a large change in the liquid temperature. As a result, the thermophysical properties of the flowing liquid change dramatically. For example, Re at the inlet was typically about 500–1000, while, at the outlet, Re ranged from 1800 to 2500 or larger, depending on the range of water flow velocity and other experimental con-

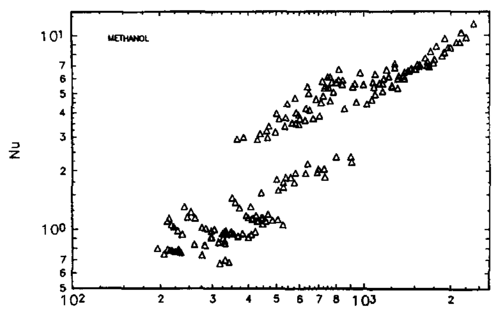


Fig. 1. Nusselt number vs Reynolds number for methanol.

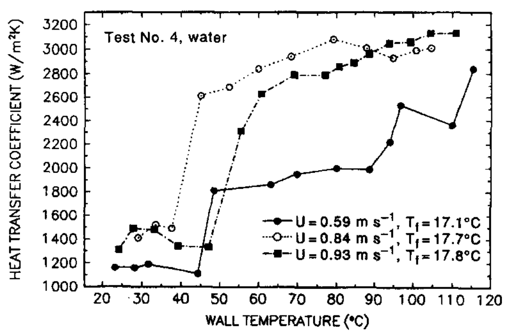


Fig. 2. Variation of heat transfer coefficient with wall temperature (test section 4).

ditions used. This implies that, for water flow, Re could possibly double over the length of the microchannel for an inlet Re of around 1000. Figure 3 more clearly illustrates the effect of wall temperature on Re. As shown, variations in the heating rate or wall temperature induce relatively large changes in Re.

THE EFFECT OF THERMOFLUID AND GEOMETRICAL PARAMETERS

From Fig. 2, it can also be seen that the liquid inlet temperature and velocity have a significant influence on the relationship between the heat transfer and the wall temperature, and hence the change from one type of flow regime to another. Generally speaking, the heat transfer coefficient becomes large as the liquid temperature decreases or the liquid velocity increases. All of the experimental results made in this investigation support this conclusion. Increasing velocity promotes the large variations in the heat transfer coefficient and the transitions are shifted to the left of the h–T_w graph, and reducing the liquid temperature causes an identical situation. For the generally monatomic or hardly changing region, increasing the liquid velocity and/or reducing the liquid temperature causes the slope of the h–T_w curve to diminish, as shown in Fig. 2.

To explore the significance of the microchannel size, the experimental results of different test sections with same liquid velocity are illustrated in Fig. 4(a) for water. Clearly, for a specified condition, the microchannel size significantly alters the heat transfer performance. The heat transfer coefficient of test section 3 is much higher than those of the other three test sections, which demonstrate no significant difference. The results for methanol, illustrated in Fig. 4(b), show the same behavior.

Figure 5(a) and (b) compares the measured heat transfer coefficient for mass flow rates of 0.32 and 0.13 g s^{-1}, respectively. For both cases, i.e. regardless of the mass flow rate, the performance of test section 3 is superior to the other test sections. As illustrated the smaller microchannel, test section 2, results in a slightly greater heat transfer coefficient than the larger one, test section 1. Unfortunately, there is no way to properly compare the results for test section 1 with test section 4. However, it is apparent that the heat transfer performance improves as microchannel size decreases from that of test section 1 to that of test section 3.

In general, it appears that an optimum channel size exists in terms of the heat transfer performance in microchannels. From both Fig. 4 and Fig. 5, it is apparent that the microchannel size plays an important role in the determination of the flow mode and heat transfer mechanism, and hence the heat transfer performance. For similar velocities or mass flow rates the flow mode and heat transfer regime might be different for different sizes of microchannels, with

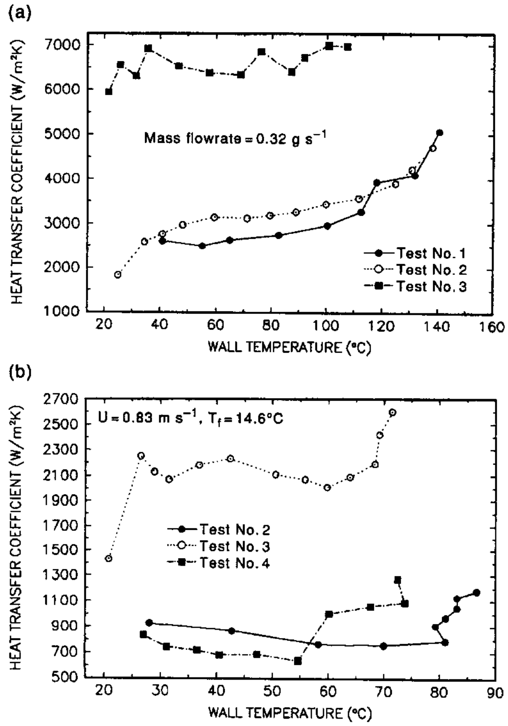
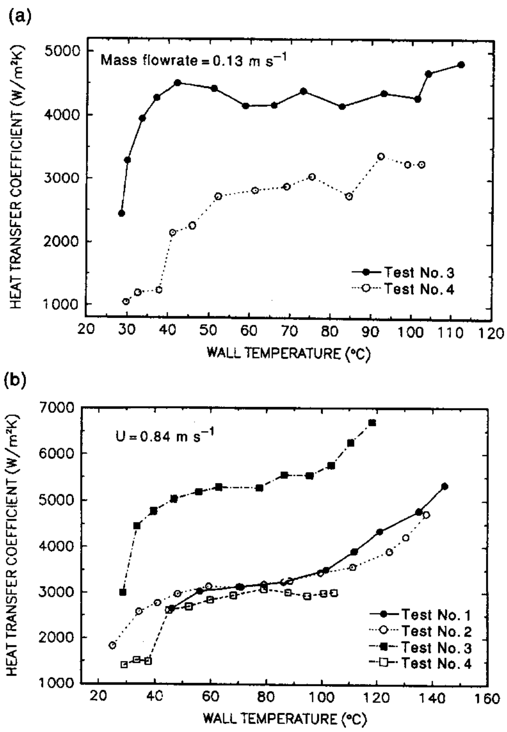


Fig. 4. Effect of microchannel size for a specified liquid velocity: (a) for water, (b) for methanol.



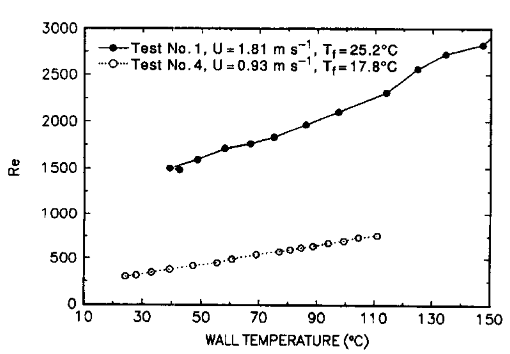


Fig. 3. Effect of wall temperature on the liquid Reynolds number.

Fig. 5. Effect of microchannel size for a specified liquid mass flow rate.

the size of the microchannel also affecting the transition from one flow regime to another.

CONCLUSION

The single-phase forced convective heat transfer characteristics of water flowing through microchannels with a rectangular cross-section were investigated experimentally. The experimental results indicate that the liquid convection characteristics are quite different from those observed in conventionally sized channels. The conversions of flow modes and heat transfer regimes are initiated at much lower *Re* than for the conventional situation. The transition from the laminar flow regime occurs at *Re* of approximately 300, and the transition to the fully turbulent flow regime at about $Re = 1000$. The transitions are influenced by liquid temperature, velocity and microchannel size.

Transition and laminar heat transfer in microchannels are significantly different from those of liquid flowing through conventionally sized channels, and are considerably more complicated. The range of the transition zone, and the heat transfer characteristics of both the transition and laminar flow regimes, are strongly affected by the liquid temperature, liquid velocity and microchannel size, and, hence, are not only determined by *Re*. Evidence was presented to support the existence of an optimum channel size in terms of the forced convective flow heat transfer of a single-phase liquid flowing in a rectangular microchannel.

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Boiling curve correlation for subcooled flow boiling

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1. INTRODUCTION

A precise knowledge of the subcooled flow boiling curve is essential in many engineering applications, which include research fission and fusion reactor components, high-power synchrotron and optical components, and advanced electronic components. Such examples are characterized as high heat flux (HHF) applications, which can be accommodated by few other means except subcooled flow boiling. Accurate subcooled flow boiling conditions are usually represented by the boiling curve, which describes the relationship between the applied heat flux and the wall temperature or wall superheat, and hence the heat transfer coefficient for the given flow conditions. Complete and accurate representation of this curve for HHF applications requires the identification and characterization of various flow regimes and transition boundaries. Although much work in characterizing the boiling curve at low heat flux levels has been completed, there are still many uncertainties and inaccuracies in HHF applications.

The objective of this work is to improve the present ability to predict local (axial) heat transfer in the subcooled flow