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Convective heat transfer and flow friction for water flow in microchannel structures

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Abstract—The single-phase forced convective heat transfer and flow characteristics of water in microchannel structures/plates with small rectangular channels having hydraulic diameters of 0.133–0.367 mm and distinct geometric configurations were investigated experimentally. The results indicated that the geometric configuration had a significant effect on the single-phase convective heat transfer and flow characteristics. The laminar heat transfer was found to be dependent upon the aspect ratio and the ratio of the hydraulic diameter to the center-to-center distance of the microchannels. The turbulent heat transfer was found to be a further function of a new dimensionless variable, Z, such that $Z = 0.5$ will be the optimum configuration for turbulent heat transfer regardless of the groove aspect ratio. The friction factor or flow resistance reached a minimum value as Z approaches 0.5. The turbulent flow resistance was usually smaller than that predicted by classical relationships, and the Reynolds number for flow transition to fully developed turbulent flow became much smaller than the ordinary channel flow. Empirical correlations were suggested for calculating both the heat transfer and pressure drop. Copyright © 1996 Elsevier Science Ltd.

INTRODUCTION

The importance of convective heat transfer with and without phase change in microchannels and/or microchannel structures has increased dramatically due to practical applications involving the thermal control of electronic devices. In the earliest investigation of microscale flow and heat transfer, Tuckermann and Pease [1] studied the fluid flow and heat transfer characteristics in microchannels, and demonstrated that electronic chips could be effectively cooled by means of the forced convective flow of water through microchannels fabricated either directly in the silicon wafer or in the circuit board on which the chips were mounted. The dissipated heat flux was of the order of 1.3×10^7 W m^{-2} while the surface temperatures were maintained at a level of less than 130°C. These results confirmed the potential of this technology. Since this initial study, other investigations have supported the earliest findings and have served to illustrate the unusually high levels of heat removal that can be accomplished using microchannel structures [2]. Several of these investigations have suggested that the heat transfer coefficient for laminar flow through microchannels might be higher than for turbulent flow through larger more conventionally sized channels.

Shortly after the initial work of Tuckermann *et al.* [1, 2], Wu and Little [3, 4] measured the flow friction and heat transfer characteristics of gases flowing through microchannels and observed that the convective heat transfer characteristics departed from the typical experimental results for conventionally sized channels. In addition, the frictional pressure drop for laminar flow was found to be higher than the classical prediction. These measurements indicated a transition from laminar to turbulent flow at Reynolds numbers of 400–900 for a number of different test configurations. Pfahler *et al.* [5, 6] later found that the frictional flow constant, $C = f^* Re$, was generally lower than what was normally obtained from theoretical predictions, and that this value increased with increasing Reynolds number. Other investigations by Choi *et al.* [7], Weisberg *et al.* [8] and Bowers and Mudawar [9] all provided additional information and considerable evidence that the behavior of fluid flow and heat transfer in microchannels or microtubes without phase change is substantially different from that which typically occurs in larger channels and/or tubes.

In an attempt to clarify some of the questions surrounding this issue, Peng and Wang [10, 11] and Peng *et al.* [12] investigated microchannels and microchannel structures both with and without phase change. In these investigations, the heat transfer and flow mode conversions for single-phase forced convection in microchannels, and the transitions induced by or associated with variations in the liquid ther-

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NOMENCLATURE

C_f	empirical coefficient for flow friction	W_c	center-to-center distance of microchannel
D_h	hydraulic diameter	W_t	width of microchannel structure/plate
f	friction factor	Z	dimensionless variable defined by equation (9)
f^*	dimensionless variable defined by equation (13)	μ	dynamic viscosity
H	height of microchannel	ν	viscosity
h	heat transfer coefficient	ρ	density
k	conductivity	τ	shear stress.
L	length of microchannel structure/plate		
Nu	Nusselt number		
Pr	Prandtl number		
p	pressure drop		
Re	Reynolds number		
q''	heat flux		
T	temperature		
T	temperature difference		
T_m	log mean temperature difference		
U	velocity		
W	width of microchannel		

Subscripts

ex	at exit
f	liquid
H	heat transfer
in	at inlet
l	laminar flow
t	turbulent flow
w	wall.

mophysical properties due to the increases in the liquid temperature through the heated microchannels, were studied. Wang and Peng [13] also experimentally studied the forced flow convection of liquid in microchannels and found that the fully developed turbulent convection was initiated at Reynolds numbers in the range of 1000–1500, and that the conversion from the laminar to transition region occurred in the range of 300–800. The heat transfer behavior in the laminar and transition regions was found to be quite unusual and complicated. Peng and Peterson [14] later confirmed these experimental observations using methanol flowing through similar microchannel structures and analyzed experimentally the effects of the thermofluid and the geometric variables on the heat transfer. Most recently, Peng *et al.* [15, 16] measured both the flow friction and the heat transfer for single phase convection in an array of parallel microchannels. These measurements validated previous observations and conclusions and were used to develop a set of heat transfer and friction factor correlations for both laminar and turbulent flow.

The present work was undertaken to investigate the liquid flow and heat transfer characteristics in microchannel structures in an attempt to experimentally determine the effects of the geometric configuration on the flow and heat transfer and to further develop heat transfer and flow friction correlations, which would be readily applicable to engineering design and practical applications.

EXPERIMENTAL DESCRIPTION

The experimental test apparatus employed in the current investigation was similar to that described previously [15, 16]. However, for the current investigation, twelve different microstructure configurations, each with several rectangular microchannels were designed and evaluated. The geometric and characteristic parameters are summarized in Table 1. The test module shown in Fig. 1 consisted of a stainless steel plate substrate into which the microchannels were machined, and a cover which served as both an insulator and a sealant. On the microchannel plate or test section, two sumps were machined and connected by microchannels. A pressure tap was located at both the inlet and outlet sumps in order to measure the inlet and outlet pressures and the pressure drop through the microchannels. Thermocouples were installed in the sumps at the entrance and exit to measure the inlet and outlet liquid temperature. In addition, six thermocouples were mounted on the back of each microchannel plate to measure the axial wall temperature distribution and ensure that the flow rate was equally distributed in the channels. These thermocouples were distributed longitudinally at three different locations. During the experiment, the test module was wrapped in an insulating shroud to minimize the heat loss to the surroundings through convection and radiation.

Water was selected as the working fluid. To account for fluid property variations, the experiments were conducted over an inlet liquid temperature range of 20–45°C. The liquid velocity was relatively high, ranging from 0.20 to 12 m s^{-1}. Accordingly, the Reynolds number spanned a range of pure laminar to highly turbulent, 50–4000. The flow rate of water entering the test module was measured using a rotameter, which had been calibrated by the weight method. The pressure drop of the liquid flowing through the microchannel was measured to determine the flow friction.

Table 1. Geometric parameters of the test sections

Plate	W [mm]	W_c [mm]	W_t [mm]	L [mm]	H [mm]	D_h [mm]	D_h/W_c	H/W	Z	$C_{f,l}$	$C_{f,t}$
1	0.4	4.5	18	45	0.2	0.267	0.0593	0.5	0.5	28 600	40 400
2	0.4	2.8	18	45	0.3	0.343	0.1225	0.75	0.75	44 800	34 200
3	0.4	2.0	18	45	0.3	0.343	0.1715	0.75	0.75	44 800	34 200
4	0.3	4.6	18	45	0.2	0.24	0.0533	0.667	0.667	42 600	18 200
5	0.3	2.8	18	45	0.3	0.30	0.107	1.00	1.00	109 000	38 600
6	0.3	2.0	18	45	0.3	0.30	0.15	1.00	1.00	109 000	38 600
7	0.2	4.5	18	45	0.2	0.20	0.0444	1.00	1.00	32 400	20 100
8	0.2	2.8	18	45	0.3	0.24	0.0857	1.50	0.667	42 600	18 200
9	0.2	2.0	18	45	0.3	0.24	0.12	1.50	0.667	42 600	18 200
10	0.1	4.5	18	45	0.2	0.133	0.0296	2.0	0.50	5200	1820
11	0.1	2.8	18	45	0.3	0.15	0.0536	3.0	0.333	24 200	6920
12	0.1	2.0	18	45	0.3	0.15	0.075	3.0	0.333	24 200	6920

The microchannel stainless steel plate was electrically heated by directly connecting the test sections to a d.c. transformer that provided low voltage and high electric current. In this way, heat generated in the thin section of the plate was transferred directly to the liquid from the two sides and the bottom of the microchannel, as illustrated in Fig. 1(c). Because the center section of the test plate was uniform in thickness and width, i.e. cross-sectional area, the heat flux was considered to be uniform along both the longitudinal length and around the wetted periphery, except for the top surface, which was insulated. The input voltage and current were used to measure and control the heat flux. These parameters were varied from 0.15 to 0.92 V and 50 to 400 A, respectively.

The experiments were carried out under steady-state conditions. For each different set of conditions, the experimental data, including flow rate, liquid temperatures, inlet and outlet pressures (or pressure drop between the entrance and exit), the test section wall surface temperatures, and the input voltage and current, were measured and recorded. The measurement uncertainties were analyzed and are given in Table 2.

The parameters used in the data reduction and analysis are summarized below:

the hydraulic diameter, D_h

$$D_h = \frac{4(\text{cross-section area})}{\text{wetted perimeter}} = \frac{2WH}{W+H} \quad (1)$$

the Reynolds number, Re

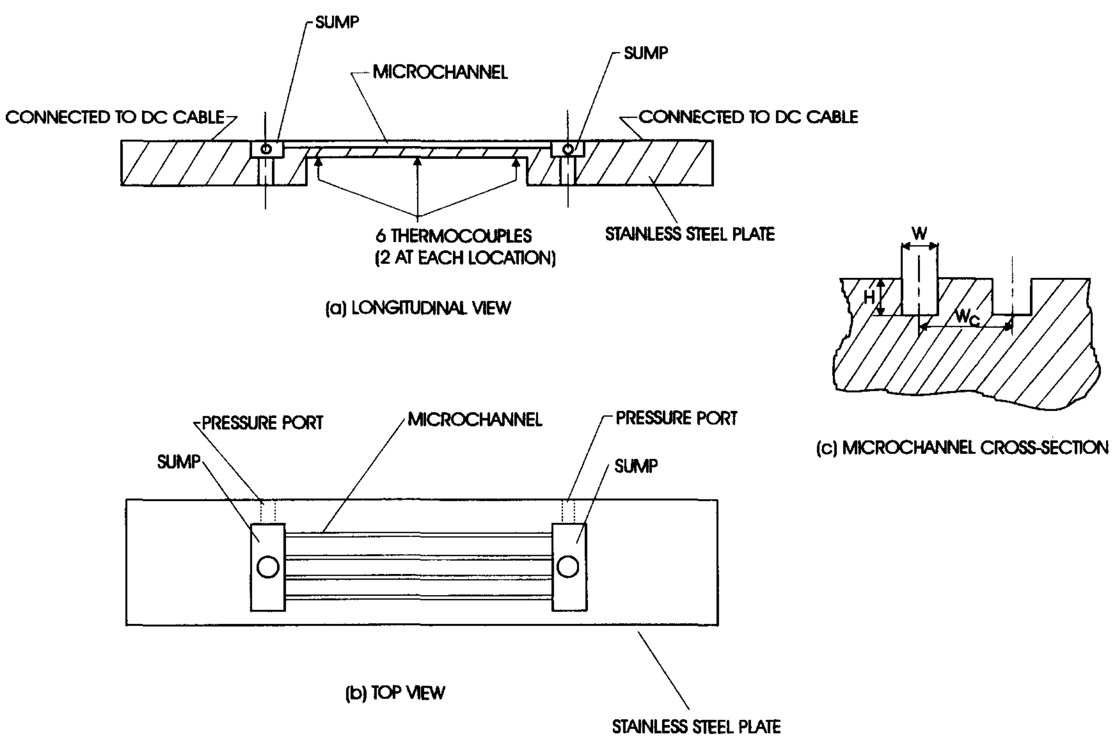

Fig. 1. Experimental section.

Table 2. Experimental uncertainties

Variables	Uncertainty [%]
Flow rate	1.5
Velocity	3.5
Voltage	0.5
Current	0.5
Heat flux	8
Length scale	12
Pressure drop	1.5
Liquid temperature	1.5
Wall temperature	1
Heat transfer coefficient	10
Flow friction factor	10
Nusselt number	16
Reynolds number	8

$$Re = \frac{UD_h}{\nu} \tag{2}$$

where U represents the mean channel velocity, the flow friction factor, f,

$$f = \frac{8\tau_w}{\rho U^2} = \frac{2D_h\,\Delta p}{\rho U^2 L} \tag{3}$$

and the Nusselt number, Nu,

$$Nu = \frac{hD_h}{k}. \tag{4}$$

In the calculation of the friction factor, f, and the Reynolds number, Re, the thermophysical properties of water were assumed to be independent of liquid pressure. Because of variations in the temperature of the liquid, the friction factor and the Reynolds number varied along the length of the plate. For simplicity, the inlet properties and conditions of the liquid were used to calculate these values. Since there existed a sudden flow contraction and expansion at the entrance and exit of the microchannels, the actual measured pressure drop included these contraction and expansion losses. The pressure drop, Δp, in equation (3), however, represents the pressure drop only along the microchannels, hence, the calculated pressure drop caused by the contraction and expansion were subtracted from the measured values. These were 5–7% of the total.

In the determination of the total heat generation or heat flux, the heat losses induced by conduction, convection, and radiation were modified using an energy balance. The heat flux of the microchannel structure was defined as

$$q'' = \frac{Q}{A} \tag{5}$$

where Q is total heat input, and A denotes the microstructure plate area (i.e. $A = W_t L$). The heat transfer coefficient was evaluated as,

$$h(x) = \frac{q''}{\Delta T_m} \tag{6}$$

where T_m is the log-mean temperature difference, determined from

$$\Delta T_m = \frac{\Delta T_{in} - \Delta T_{ex}}{\ln(\Delta T_{in}/\Delta T_{ex})}. \tag{7}$$

For all cases, the relative size of the heat losses was small.

The final results are presented as a function of the Nusselt number, Reynolds number and friction factor. Using conventional techniques, the uncertainties for all of the experimental values measured were estimated and are given in Table 2.

HEAT TRANSFER PERFORMANCE

Peng *et al.* [15] demonstrated that the critical Reynolds number for laminar transition in microchannel structures occurs at approximately 200–700 and for fully developed turbulent flow at Reynolds numbers of 400–1500. In addition, the transition Reynolds numbers have been shown to decrease as the hydraulic diameter decreased. The experiments conducted in the current investigation demonstrated that these conclusions are true only for liquid flow in microchannel structures with identical microchannel dimensions. The current data indicates that the geometric parameters such as hydraulic diameter, ratio of the microchannel height and width, H/W, and ratio of hydraulic diameter and microchannel center-to-center distance, D_h/W_c, all have a significant influence on both the flow regime and the heat transfer.

Laminar flow

The laminar convective heat transfer data for water flowing through the twelve different microchannel structures was shown to be correlated by the following relationship

$$Nu = 0.1165\left(\frac{D_h}{W_c}\right)^{0.81}\left(\frac{H}{W}\right)^{-0.79} Re^{0.62} Pr^{1/3}. \tag{8}$$

In Fig. 2, this correlation is compared with the experimental results. Although the Reynolds number range for laminar convective heat transfer is not identical, due to the difference in the transition Reynolds number for different microchannel structures or plates, this comparison indicates reasonably good agreement between the experimental data obtained from the twelve microchannel plates and the correlation given by equation (8). The range of deviation for this correlation is approximately $\pm 30\%$.

Figures 3 and 4 depict the average experimental results for the individual test plates as a function of D_h/W_c and H/W, respectively, and illustrate the effects of the geometric parameters on the laminar convective heat transfer. Careful thought was given to the selection of these dimensionless parameters in order to explore and generalize the effect of the various channel dimensions. Figure 3 indicates that the laminar heat transfer will be augmented by enlarging the hydraulic

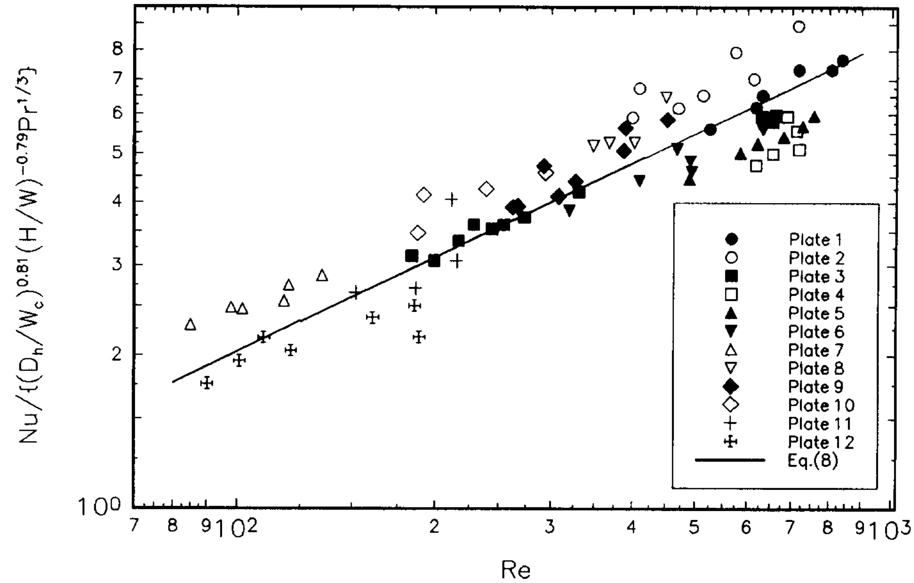

Fig. 2. Experimental results of laminar heat transfer.

diameter or decreasing the center-to-center distance. Figure 4 indicates that increasing the width or decreasing the height of the microchannel enhances the laminar heat transfer.

Turbulent flow

For turbulent convection, the ratio D_h/W_c, was found to be a very important parameter, particularly when compared to the case of laminar convective heat transfer. Figure 5 illustrates the change of the turbulent heat transfer for each test plate with the dimensionless value D_h/W_c. This relationship is similar to that shown in Fig. 3, however, a more detailed analysis indicates that the turbulent convection is somewhat more heavily dependent on this ratio than the laminar case. The significance of H and W is nearly the same, however, the effect of these two parameters on the turbulent heat transfer cannot be simply described by the dimensionless parameter, H/W. For this reason, a new dimensionless variable, Z, defined as

$$Z = \frac{\min(H, W)}{\max(H, W)} \tag{9}$$

has been introduced, which helps to define the effect of the aspect ratio. The significance of this relationship was determined empirically.

The measured results of the turbulent heat transfer

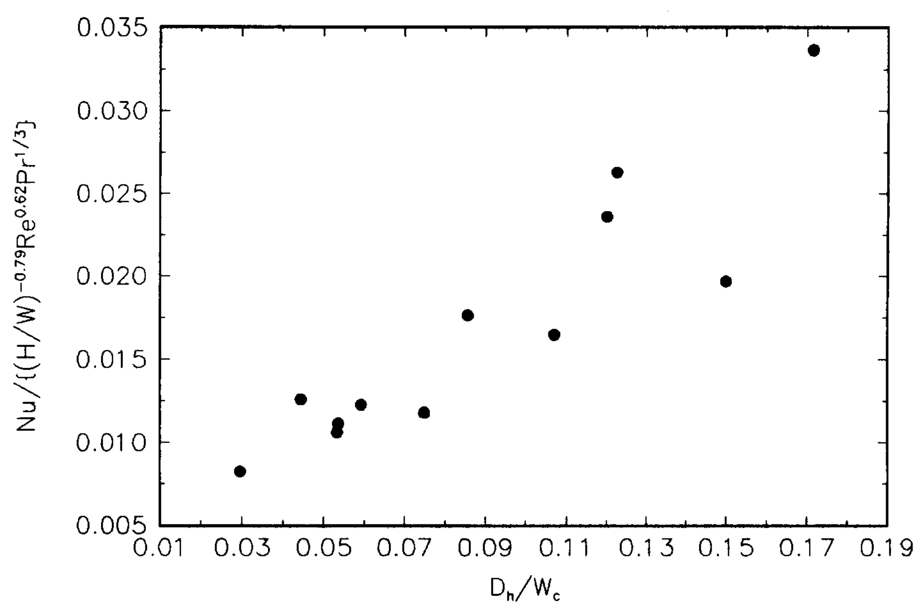

Fig. 3. Effect of D_h/W_c on laminar heat transfer.

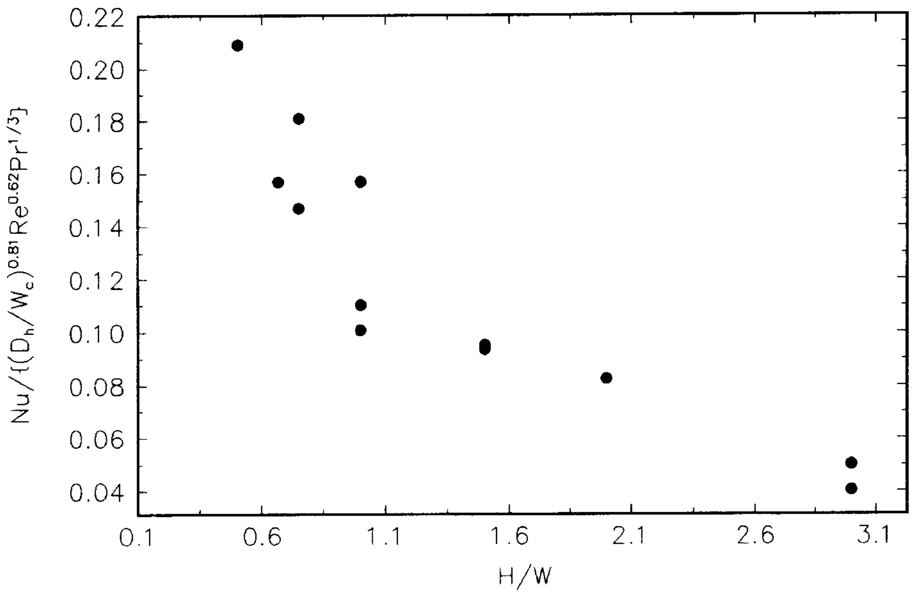

Fig. 4. Effect of H/W on laminar heat transfer.

for different test plates are depicted as a function of Z, in Fig. 6. This figure indicates that there exists an optimum value of Z which maximizes the turbulent convective heat transfer, and that this value, approximately $Z = 0.5$, is constant, regardless of the ratio of H/W or W/H.

For fully developed turbulent flow in microchannel structures, the forced convective heat transfer correlation can be expressed as

$$Nu = 0.072\left(\frac{D_h}{W_c}\right)^{1.15}[1-2.421(Z-0.5)^2]Re^{0.8}Pr^{1/3}. \tag{10}$$

Figure 7 compares the correlation given in equation (10) with the turbulent heat transfer data obtained for the twelve test plates. As shown, the experimental data for all twelve test plates fall along this line with a maximum deviation of $\pm 25\%$.

While providing a means for predicting the heat transfer characteristics in microchannel structures, these measurements indicate that the transition heat transfer for a liquid flowing through microchannel plates is highly complicated. As noted above, the geometric configuration is of significant importance for the single phase forced-convective heat transfer, and has a somewhat different effect for laminar and turbulent convection.

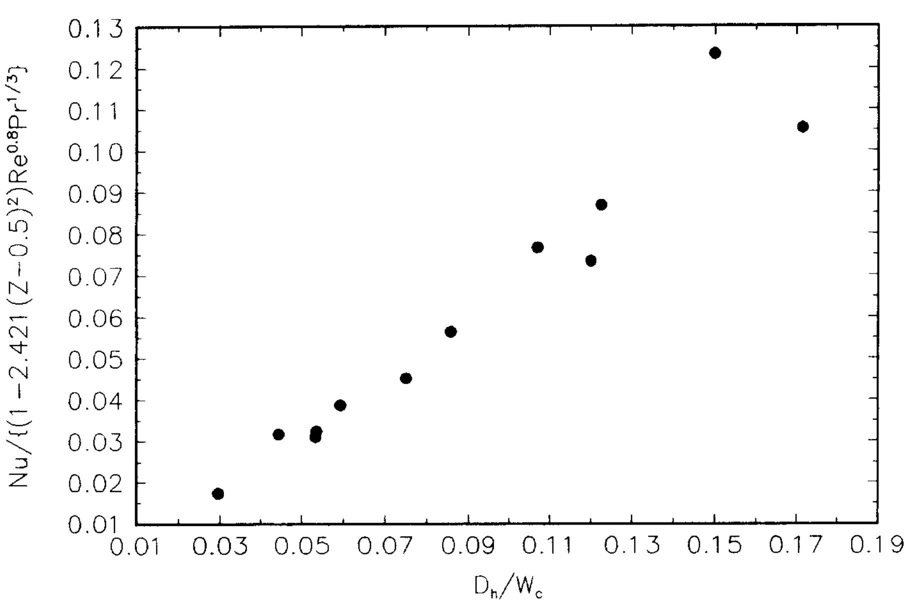

Fig. 5. Effect of D_h/W_c on turbulent heat transfer.

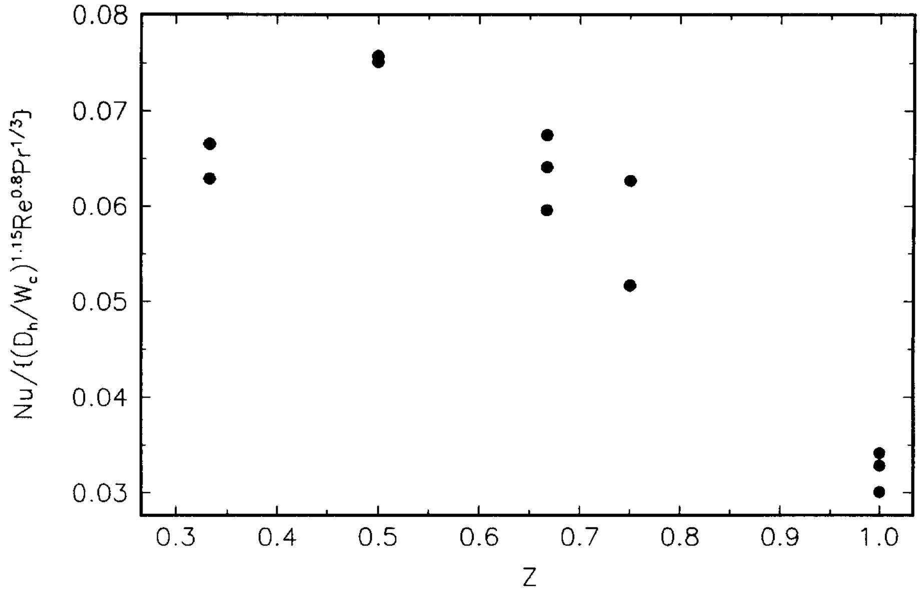

Fig. 6. Effect of Z on turbulent heat transfer.

FLOW RESISTANCE

As a result of the experimental flow measurements of Peng *et al.* [15] empirical correlations for the calculation of the friction factor for single-phase liquid flow in microchannels have been proposed. These relations are given as:

$$f = \frac{C_{f,l}}{Re^{1.98}} \tag{11}$$

for laminar flow and

$$f = \frac{C_{f,t}}{Re^{1.72}} \tag{12}$$

for fully developed turbulent flow, where $C_{f,l}$ and $C_{f,t}$ are empirical coefficients for laminar and turbulent flow, respectively. These values are also dependent upon the microchannel configuration and geometric parameters.

The experimental investigation of microchannel plates or structures indicates that the analyses of Peng *et al.* [15] are applicable to the current case, and that the flow resistance has the same form given by equations (11) and (12) for laminar and turbulent flow, respectively. Comparison of experimental data also indicates that the hydraulic diameter, D_h, and the dimensionless variable, Z, are critical parameters in

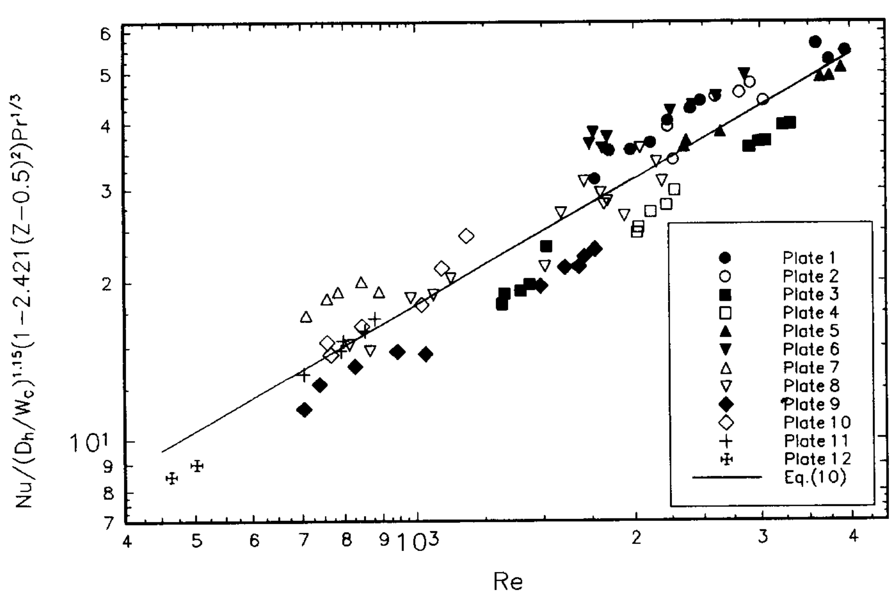

Fig. 7. Experimental results of turbulent heat transfer.

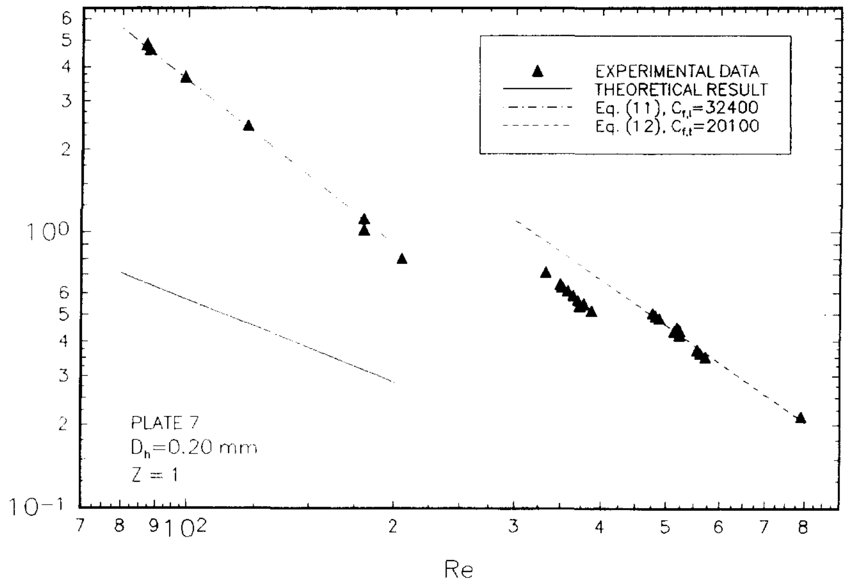

Fig. 8. Friction factor of plate 7 [15].

the determination of the empirical coefficients in equations (11) and (12). Figures 8 and 9 illustrate some of experimental results, and the values for $C_{f,l}$ and $C_{f,t}$ for the plates tested are listed in Table 1.

Comparing Figs. 8 and 9, it is clear that by selecting the appropriate geometric configuration, the friction factor for the laminar conditions can be reduced and made smaller than the value predicted by the classical relation. A new parameter, f^*, can be introduced to relate the experimental and theoretical friction factor for the laminar conversion Reynolds number, Re_{cri},

$$f^* = \left(\frac{f_{l,exp}}{f_{l,theo}}\right)_{Re_{cri}}. \tag{13}$$

This value, f^*, is depicted as a function of Z in Fig. 10. Clearly f^* has a minimum value, which might be used as an optimum parameter in the design of microchannel plates or structures and other similar devices. For turbulent flow, the measured flow resistance in microchannels is lower than the theoretical results predicted by conventional relationships, and the Reynolds number becomes smaller than the Reyn-

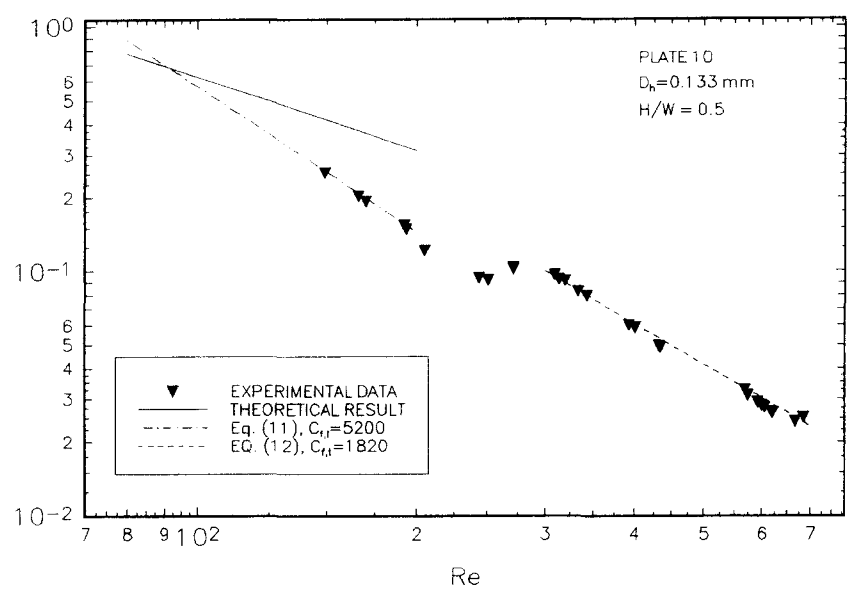

Fig. 9. Friction factor of plate 10 [15].

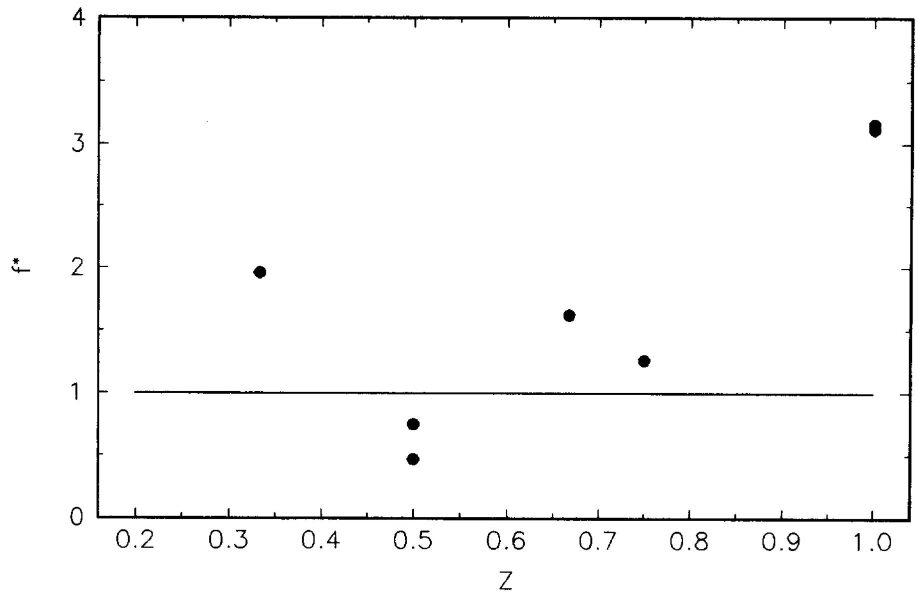

Fig. 10. Variation of f^* with Z [15].

olds number at which the flow transitions to fully developed turbulent flow occur. However, it is not clear whether W_c has any influence on the flow. How and how much these geometric factors affect the flow resistance or friction factor requires additional investigation.

CONCLUSIONS

The forced convective heat transfer and flow characteristics of water flowing through microchannel plates with extremely small rectangular channels having hydraulic diameters of 0.133–0.367 mm and different geometric configurations were investigated experimentally. Measurements indicated that the geometric configuration of the microchannel plate and individual microchannels has a critical effect on the single phase convective heat transfer, and that the effect on the laminar and turbulent convection is quite different. While the thermal conductivity of the material from which the plates were fabricated could be a factor, the microchannels are so small that the hydraulic radius is comparable to the sublayer thickness and therefore, the resistance in the sublayer for these cases becomes much more important than for larger more conventional channels. As a result, for channels as small as those evaluated here, the shape of the channels plays a negligible role for both the laminar and turbulent flow conditions. The laminar heat transfer, however, does depend on the parameters D_h/W_c and H/W, while the turbulent convection is related to D_h/W_c and Z rather than H/W. It was found that the dimensionless ratio, $Z = 0.5$, is the optimum configuration for turbulent heat transfer regardless of H/W. In addition, empirical correlations were suggested for predicting the heat transfer for both the laminar and turbulent case. Physically, for a given plate with the same size and number of channels, the convection heat transfer from the two sides of the channel in the center section of the plate decreases as the center-to-center distance is decreased. These conclusions are of fundamental importance in the practical applications and design of these types of structures.

The flow resistance of the liquid flow in the microstructures was also investigated experimentally and analytically, and correlations were proposed for the calculation of the flow resistance. The experiments demonstrated the importance of the geometric parameters, including the hydraulic diameter, H/W or Z, and D_h/W_c, on the friction factor. The laminar friction factor or flow resistance reaches a minimum value as Z approaches 0.5. The turbulent flow resistance is usually smaller than the value predicted by the classical relationships and the Reynolds number becomes smaller than the Reynolds number at which the flow transitions to fully developed turbulent flow, as the flow resistance decreases, i.e., the smaller the resistance, the larger the flow transition Reynolds number.

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