# Testing and Design of a Microchannel Heat Exchanger with Multiple Plates

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The microheating system is one of the hard cores of a microchemical system. In this paper, the performance of microchannel heat exchangers (MCHEs) with two plates made of stainless steel was investigated experimentally. The maximum volumetric heat transfer coefficient was up to 5.2 $MW/m^3 \cdot K$ with a corresponding pressure drop of less than 20 kPa under a Reynolds number of around 65. The correlations of average Nusselt number and pressure drop to Reynolds number in microchannels were presented for designing MCHEs with multiple plates with the same geometric structure being researched, and the validity of correlations was verified through MCHEs with two plates and ten plates. Moreover, experimental results verified that MCHEs can be applied to recover energy in integrated microstructure systems of thermal and chemical processes via a system of ethanol dehydration to ethylene.



## 1. Introduction

Since the emergence of silicon integrated circuit technology, the integrated circuit density has increased by several orders of magnitude. Accordingly, the heat dissipation problem has become a serious limitation on development in the semiconductor industry. On the basis of this background, the microchannel heat exchanger (MCHE), as a novel cooling technique, has developed gradually in order to maintain the electronic components at acceptable temperature levels.[1] Recently, MCHEs have been applied widely in other fields such as microchemical engineering,[2−8] cooling of powerful laser mirrors,[9] air conditioning,[10] etc.

Compared with a conventional heat exchanger, MCHEs have the main advantages of a high heat transfer coefficient, large surface area to volume ratio, and low thermal resistance due to their small characteristic dimensions. An MCHE is composed of thin plates of metal or nonmetal with microchannels etched on one side or two sides. There are two main types of MCHEs, i.e., a microchannel heat sink for heat exchange between a wall surface and fluid and a microchannel heat exchanger for heat exchange between hot and cold fluids. The concepts of the microchannel heat sink and microchannel heat exchanger were first presented by Tukerman and Pease[11] in 1981 and Swift et al.[12] in 1985, respectively.

So far, most research has been mainly focused on fully developed flow and heat transfer characteristics in microchannels.[13−17] According to a chronological analysis of the experimental results, researchers prefer to support the viewpoint that the conventional theory is still to be valid for microchannels with characteristic dimensions of tens to hundreds of micrometers. However, there is little research related to the developing flow and the overall performance of MCHEs. Owing to the special structure characteristics of microdevices, the fully developed region is relatively small and the effect of the inlet and outlet is prominent. Therefore, the correlations of the Fanning friction factor and the Nusselt number to the Reynolds number (*Re*) for fully developed flow can not be effectively applied to the design of MCHEs, especially counterflow MCHEs.

Among earlier research work related to the overall performance of two-fluid MCHEs, Alm et al.[4] experimentally investigated and simulated the performance of counterflow ceramic micro-heat exchangers using water as a test fluid with a channel width of 250 $\mu$m, fin width of 520 $\mu$m, and channel depth ranging from 320 to 420 $\mu$m. The range of the heat exchanger effectiveness was from 0.10 to 0.22, and the maximum heat transfer coefficient was 19.5 $kW/m^2 \cdot K$ with a corresponding pressure drop of more than 7 bar under the flow rate of around 2000 mL/min. Ceramic micro-heat exchanger can be applied in new fields of micro-process engineering for its advantages, such as fast heat transfer rate, good thermal stability, high corrosion resistance, etc. However, the application of ceramic micro-heat exchanger is limited due to difficult fabrication and easy breakage.

In this paper, the heat transfer performance of counterflow MCHEs made of stainless steel with deionized water as the working fluid was studied experimentally. The aim of this experiment was to define the correlations of average Nusselt number and pressure drop to Reynolds number in microchannels for the design of MCHEs with multiple plates with the same geometric structure being researched. The correlations were obtained through MCHEs with two plates and verified that the relations obtained can be used to design MCHEs with multiple plates through MCHEs with two and ten plates.

## 2. Analysis

For analysis and design of the MCHE, the heat transfer coefficients on both hot and cold sides are very important. In order to determine the heat transfer coefficients on the two sides, the Wilson plot method is applied.[18] The method is based on separation of the total thermal resistance into the thermal resistances on both fluid sides and wall thermal resistance, in which the hot side area was taken as the standard area, as shown in eq 1.

$$R_t = R_h + R_w + R_c \tag{1}$$

$$R_t = \frac{1}{KA_h} \tag{2}$$

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$$R_h = \frac{1}{k_h A_h} \tag{3}$$

$$R_w = \frac{\delta_w}{\lambda_w A_w} \tag{4}$$

$$R_c = \frac{1}{k_c A_c} \tag{5}$$

Combining eqs 1−5, the overall heat transfer coefficient can be expressed as

$$\frac{1}{KA_h} = \frac{1}{k_h A_h} + \frac{\delta_w}{\lambda_w A_w} + \frac{1}{k_c A_c} \tag{6}$$

Due to the relatively high conductivity of stainless steel and the small wall thickness, the thermal resistance of the wall $R_w$ is about 1.5–3.5% of the total thermal resistance and can be negligible in this work. If the hot fluid side and the cool fluid side have almost the same dimensions of microchannels and chambers of exchanger plates and mass flow rates, it is reasonable to assume the convection heat transfer coefficients of both the hot fluid side and the cool fluid side are the same, that is

$$k_h = k_c = 2K \tag{7}$$

The overall heat transfer coefficient of the MCHE is defined by

$$K = \frac{Q_m}{A_h \cdot \Delta t_m} \tag{8}$$

Where, $A_h$ is equal to the total area of the bottom wall and side walls of the microchannel and partial chamber (overlapped vertical projection region of any two adjacent plates' chambers). $Q_m$ is the average heat transfer rate between the hot and cold fluids.

$$Q_m = 0.5(Q_h + Q_c) \tag{9}$$

$Q_h$ and $Q_c$ are the heat transfer rates in the hot and cold fluids, respectively.

$$Q_h = m_h c_p (T_{h,i} - T_{h,o}) \tag{10}$$

$$Q_c = m_c c_p (T_{c,o} - T_{c,i}) \tag{11}$$

The logarithmic mean temperature difference $\Delta T_m$ is defined by

$$\Delta T_m = ((T_{h,i} - T_{c,o}) - (T_{h,o} - T_{c,i}))/\ln \frac{T_{h,i} - T_{c,o}}{T_{h,o} - T_{c,i}} \tag{12}$$

Therefore, the functional relations of the convection heat transfer coefficients to flow rate of the plate of MCHE can be obtained.

Correspondingly, the overall heat transfer coefficient of MCHE with multiple plates can be estimated through the heat transfer coefficients on both sides which can be calculated by simplified form of eq 6 when the hot fluid side and the cool fluid side have almost the same heat transfer area and the thermal resistance of the wall $R_w$ is negligible.

$$\frac{1}{K} = \frac{1}{k_h} + \frac{1}{k_c} \tag{13}$$

The hot fluid outlet temperature $T_{h,o}$ and cool fluid outlet temperature $T_{c,o}$ are calculated by

$$T_{h,o} = T_{h,i} - \varepsilon (mc_p)_{min} (T_{h,i} - T_{c,i})/KA \tag{14}$$

$$T_{c,o} = T_{c,i} + \varepsilon (mc_p)_{min} (T_{h,i} - T_{c,i})/KA \tag{15}$$

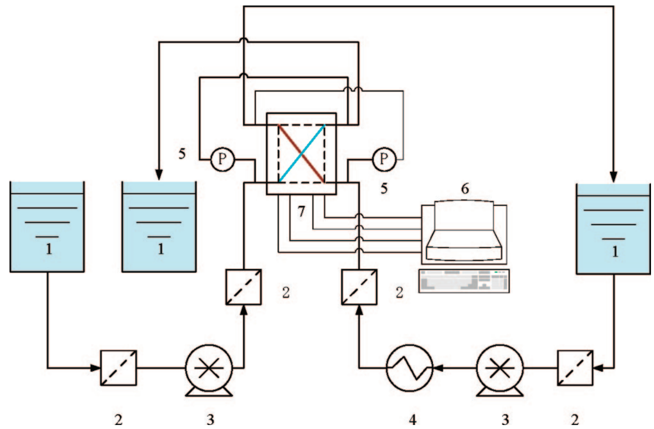


**Figure 1.** Experimental apparatus: (1) water tank; (2) filter; (3) water pump; (4) electric heater; (5) differential pressure transducer; (6) multiplexer and computer; (7) MCHE.

Where, $\varepsilon$ is the heat exchanger effectiveness, defined by

$$\varepsilon = \frac{Q_m}{Q_{max}} \tag{16}$$

Where

$$Q_{max} = \min((mc_p)_c, (mc_p)_h)(T_{h,i} - T_{c,i}) \tag{17}$$

In the $\varepsilon$−NTU method, $\varepsilon$ can be calculated through NTU (number of heat transfer units) and $C_R$ (the ratio of heat capacity rate).[19]

$$\varepsilon = \frac{1 - \exp[(1 - C_R) \cdot \mathrm{NTU}]}{C_R - \exp[(1 - C_R) \cdot \mathrm{NTU}]} \tag{18}$$

Where, NTU and $C_R$ are defined by

$$\mathrm{NTU} = \frac{KA}{(mc_p)_{min}} \tag{19}$$

$$C_R = \frac{(mc_p)_{min}}{(mc_p)_{max}} \tag{20}$$

## 3. Experimentation

**3.1. Experimental Apparatus.** The experimental apparatus, shown schematically in Figure 1, consisted of an electrical heater, an instrument to measure temperatures, a PC data acquisition system, two pumps, two differential pressure transmitters, three water tanks, filters, and MCHEs. After filtration, the hot water was driven through an electrical heater to the inlet of the MCHE and, then, went back to the hot water tank for recycling. The cold water was supplied from a cold water tank and, after filtration, flowed through MCHE to another cold water tank in order to recycle when the water recovered to room temperature.

The MCHEs made up of different numbers of plates were bonded via vacuum diffusion bonding under the operating conditions of temperature (1273−1373 K), vacuum degree ($10^{-3}$ Pa), and pressure on MCHEs (6−20 MPa). Figure 2 represents an exploded view of a counterflow MCHE with four plates. For MCHE with multiple plates, a similar arrangement pattern is adopted. The channels were fabricated on one side of the plate by a chemical etching method from 400 $\mu$m thick stainless steel plates. To begin with, the plates used were cleaned and dried. Next, the etching film was stuck to the both sides of the plates. Then, the plates with etching film were exposed and developed via the solution of sodium carbonate. After cleaning

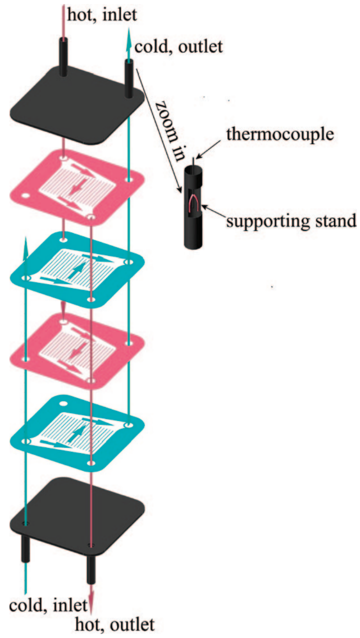



**Figure 2.** Exploded view of a counterflow MCHE with four plates.

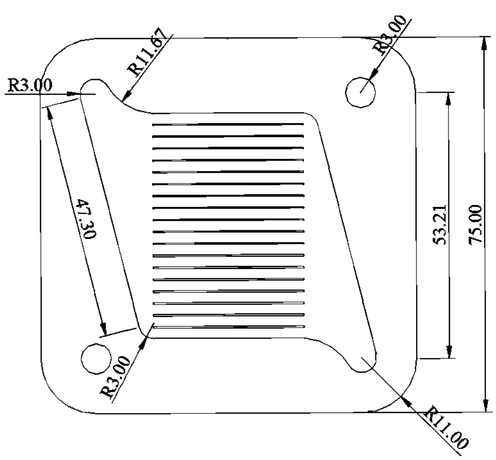



**Figure 3.** Geometry dimension of a plate of the MCHE.

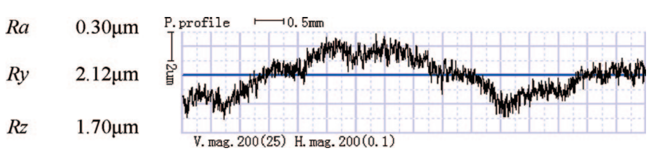


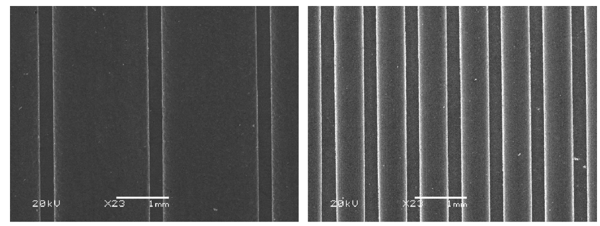

**Figure 4.** Surface roughness of etched area.

and drying, the exposed plates were etched in the solution of ferric chloride. Finally, the etching film on the plates was removed in the solution of sodium hydroxide.

During the experiment, in order to avoid direct contact between the apex of thermocouple and the pipe wall of the inlet or outlet, a special small supporting stand was designed to measure fluid temperature accurately (shown in Figure 2). Figure 3 shows the geometric dimension of the plate. The value of surface roughness (Ra) of the plate measured by step profiler (ET4000M) ranges from 0.24 to 0.30 $\mu$m, and one of the measure maps is shown in Figure 4. The channel size, channel number, and plate number of different types of MCHEs are listed in Table 1, where the channel size was measured through a scanning electron microscope (SEM), and the SEM photographic images are shown in Figure 5.

**Table 1. Geometric Dimensions of Counterflow MCHEs**

| MCHE | 1 | 1* | 2 | 2* | 3 | 3* |
|---|---|---|---|---|---|---|
| number of plates | 2 | 10 | 2 | 10 | 2 | 10 |
| number of channels | 1 | | 19 | | 49 | |
| channel length/mm | 30 | | 30 | | 30 | |
| channel width/$\mu$m | 45000 | | 2122 | | 639 | |
| channel height/$\mu$m | 197 | | 197 | | 197 | |
| fin width/$\mu$m | 0 | | 260 | | 285 | |




a. plate of 2# and 2*# MCHEs b. plate of 3# and 3*# MCHEs

**Figure 5.** Scanning electron microscope photographic images of microchannels. (a) Plate of 2 and 2* MCHEs. (b) Plate of 3 and 3* MCHEs.

**3.2. Experimental Procedure.** For each experiment, hot and cold water flow rates, inlet and outlet water temperatures, and pressure drop between the inlet and outlet were measured at steady state. Temperatures were measured using four thermocouples of type T, connected through a multiplexer to a personal computer. The pressure drops between the two ends of the hot fluid and cold fluid were measured by differential pressure transmitters connected to digital display meters. The flow rates were measured by a weighing method. The measurement uncertainty of thermocouple of type T and differential pressure transmitters was within $\pm$1 K and $\pm$0.25 kPa, respectively. The maximum error in the flow rate was less than 1.5%, and the experimental errors in the heat balance were less than 5%.

For measuring the convection heat transfer coefficients of the MCHE plate conveniently under a given flow rate, the nearly equal volume flow rates in both passages were increased from 20 to 80 mL/min in steps of 10 mL/min in MCHEs with two plates, corresponding *Re*s in microchannels ranged from around 15 to 65. Subsequently, in order to verify that the correlations of the average Nusselt number and pressure drop obtained can be used to design MCHEs with multiple plates, the performance of corresponding MCHEs with two plates and ten plates was measured experimentally under the condition that hot fluid flow rate was fixed and the cool fluid flow rate was varied in a certain range.

## 4. Results and Discussion

**4.1. Thermal Performance of MCHEs.** The volumetric heat transfer coefficient at the cold and hot side tested in the MCHEs with two plates is shown in Figure 6. The effect of fins is not obvious, and volumetric heat transfer coefficients of plates of different types are nearly the same. This is due to the fact that the effect of the inlet and outlet is stronger compared with the influence of fins. The maximum volumetric heat transfer coefficient is up to 5.2 $MW/m^3 \cdot K$, and the density of heat transfer area of the plates is about 885 $m^2/m^3$. The maximum heat transfer coefficient is around 5.8 $kW/m^2 \cdot K$ with a corresponding pressure drop of less than 20 kPa under a Reynolds number of around 65. The ratio of heat transfer coefficient and pressure drop is about 0.29, much larger than the value of approximately 0.03 introduced by Alm et al.[4]

Figure 7 shows the average Nusselt number as a function of *Re* in microchannels which ranges from 1.2 to 3.7 under the

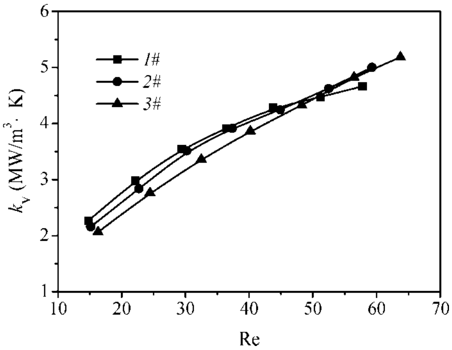



**Figure 6.** Volumetric heat transfer coefficient as a function of *Re* in microchannels.

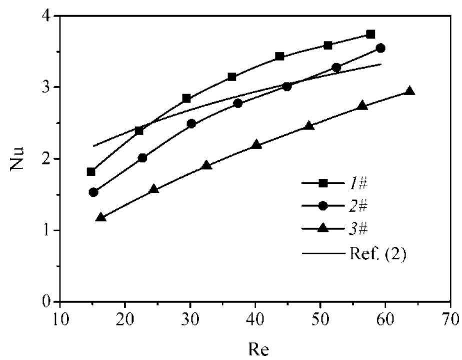



**Figure 7.** Average Nusselt number as a function of *Re* in microchannels.

corresponding *Re* in microchannels from around 15 to 65. Compared with results in ref 2, the average Nusselt number in MCHEs 2 and 3 is relatively smaller when *Re* is less than 45. However, the average Nusselt number in MCHE 1 is larger than the literature value when *Re* is larger than 25. And, the correlation of the average Nusselt number and *Re* in microchannels gained by the least-squares fitting is expressed as follows

$$Nu^{(1)} = 0.46Re^{0.527} \tag{21}$$

$$Nu^{(2)} = 0.30Re^{0.606} \tag{22}$$

$$Nu^{(3)} = 0.18Re^{0.675} \tag{23}$$

In Figure 8, experimentally measured data of outlet temperatures of MCHEs with two plates are compared with estimated results calculated through eqs 14 and 15 by using the correlations of eqs 21−23 under the condition that the hot fluid flow rate was fixed at 40 mL/min and the cool fluid flow rate varied from 20 to 80 mL/min. From Figure 8, it can be seen that the heat transfer performance of the MCHE with two plates can be well predicted and the maximum temperature difference between the value of the experiment and estimation is only 1.6 K.

Figure 9 shows the comparison of heat exchanger effectiveness of MCHEs with two plates gained through experiment and estimation, and the difference is small. Besides, the difference among plates with different types is not obvious. This is because the strong effect of the inlet and outlet overwhelms the influence of channels of different types. The heat exchanger effectiveness ranging from roughly 0.56 to 0.80 decreases at first with the increasing volumetric flow rate and reaches the bottom when the cool fluid volume flow rate is almost equal to the fixed hot fluid volume flow rate and then increases with the increasing volumetric flow rate.

**4.2. Hydraulic Performance of MCHEs.** During the test of pressure drops, a discrepancy usually exists between the hot

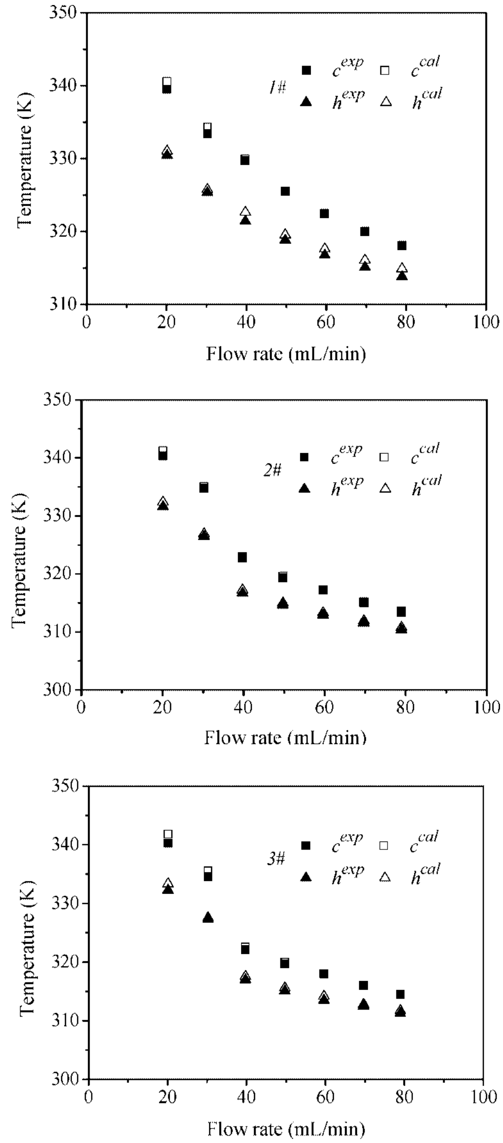



**Figure 8.** Calculated and measured outlet temperatures in the MCHEs with two plates.

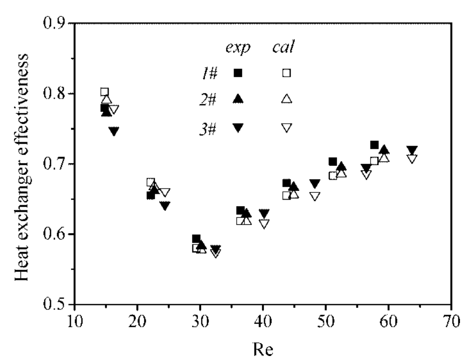



**Figure 9.** Calculated and measured heat exchanger effectiveness in the MCHEs with two plates.

side and cold side which are made up of the same type of plates. It is most likely that the difference is caused by diffusion bonding and the inconsistency of plates induced in the process of chemical etching far more than the viscosity change induced by temperature, because the result of the experimental study shows that the difference of the pressure drops of fluid with different temperature in the same plate is relatively small. In this study, the smaller pressure drop between the both sides

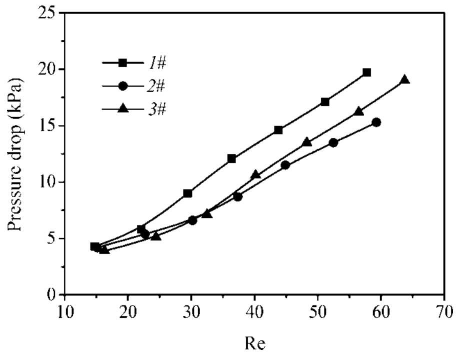



**Figure 10.** Measured pressure drop as a function of *Re* in microchannels.

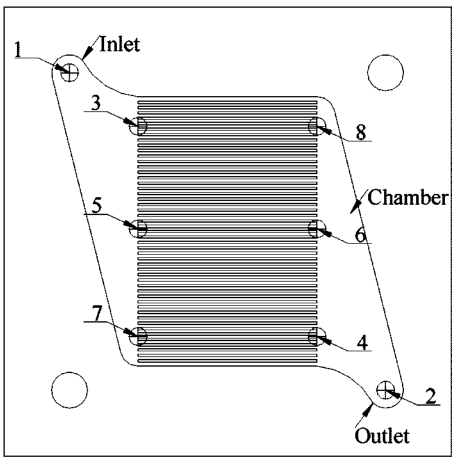



**Figure 11.** Schematic diagram of the microstructure for pressure measurement.

**Table 2. Pressure Drop of Different Pressure Measurement Points (kPa)**

| $P_{1,7}$ | $P_{7,4}$ | $P_{4,2}$ | $P_{1,5}$ | $P_{5,6}$ | $P_{6,2}$ | $P_{1,3}$ | $P_{3,8}$ | $P_{8,2}$ |
|---|---|---|---|---|---|---|---|---|
| 2.0 | 1.9 | 1.4 | 1.7 | 1.6 | 2.0 | 0.9 | 1.5 | 2.9 |

made up of the same type plates is chosen to analyze the difference of pressure drops among plates of different types. As shown in Figure 10, the difference of pressure drops among plates with different types is small, similar to heat transfer performance. The main pressure drop is induced by the inlet, outlet, and chamber rather than microchannels. And one of the correlations of pressure drop to flow rate gained by the least-squares fitting is expressed as follows,

$$\Delta p^{(2)} = 0.0022Re^2 + 0.1035Re + 1.9558 \tag{24}$$

For the purpose of confirming the effect of the inlet, outlet and chamber, a similar microstructure to the plate of an MCHE is fabricated on PMMA via micromachining technology with a channel width of 0.5 mm, depth of 0.2 mm, length of 30 mm, and fin width of 0.4 mm. The detailed distribution of pressure measurement points is shown in Figure 11. The pressure drop between the pressure measurement points 1 and 2, $P_{1,2}$, is 5.3 kPa under the condition of a volumetric flow rate of around 80 mL/min, which is far lower than the corresponding pressure drop in the MCHE. This further illustrates the point that the process of diffusion bonding and the inconsistency of plates induced in the process of chemical etching may cause a pressure drop increase. The pressure drop between other pressure measurement points is shown in Table 2, where we can conclude that the main pressure drop is induced by the inlet, outlet, and chamber, which accounts for 64−72%.

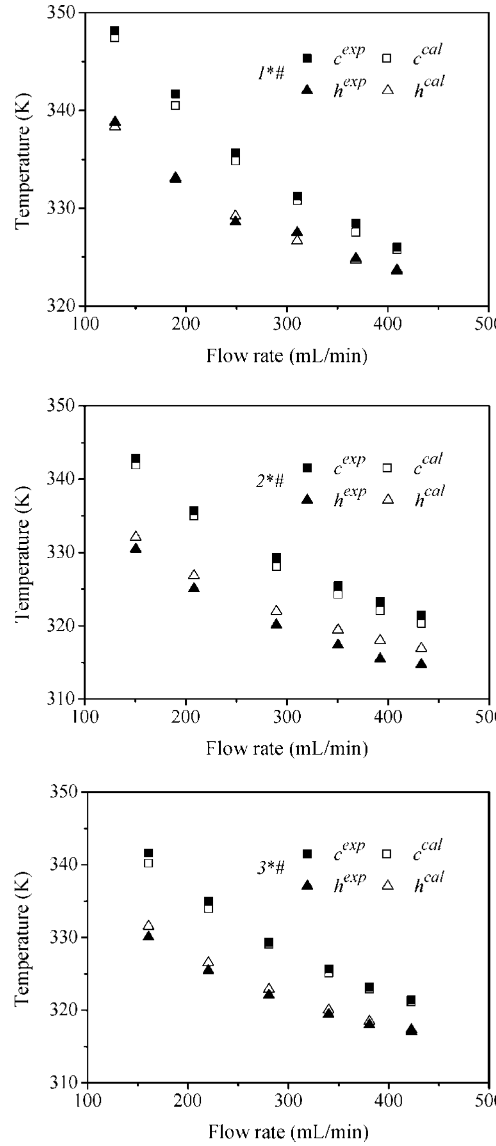



**Figure 12.** Calculated and measured outlet temperatures in the MCHEs with ten plates.

**4.3. Estimation in Scale-up Performance of MCHEs.** In order to verify that the correlations of the average Nusselt number obtained via MCHEs with two plates can be used to design MCHEs with multiple plates, the performance of corresponding MCHEs with ten plates was measured experimentally under the condition that the hot fluid flow rate was fixed and the cool fluid flow rate varied in a certain range. Figure 12 describes that the experimental measurements of the outlet temperatures of MCHEs with ten plates are in good agreement with the predictive values through eqs 14 and 15 by using the correlations of eqs 21−23. What needs to be noticed is that most outlet temperatures of the experimental results of the cool fluid and predictive values of the hot fluid are both larger than the predictive values and the experimental results of their counterparts, indicating that the experimental heat transfer performance of MCHEs with multiple plates is slightly better than that of evaluated value. This implies that the performance of MCHEs with multiple plates is conservatively estimated through the correlations obtained from MCHEs with two plates. In addition, the range of heat exchanger effectiveness of MCHEs with ten plates is similar to that of MCHEs with two plates.

When it comes to the pressure drop of MCHEs with ten plates, it is assumed that the flow rate and pressure drop of each

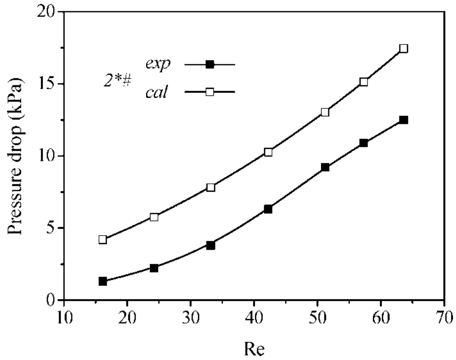



**Figure 13.** Calculated and measured pressure drop in one MCHE with ten plates.

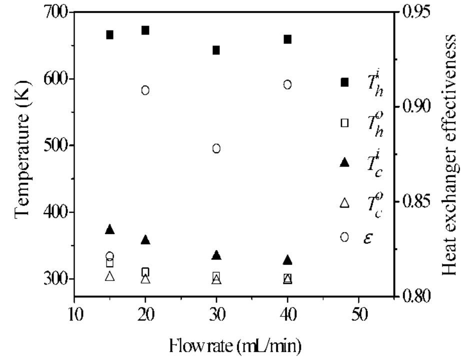



**Figure 14.** Outlet temperatures and heat exchanger effectiveness of MCHE 2.



plate is the same. Figure 13 shows that the experimental measurements of MCHE 2* are far less than the values estimated through the eq 24. This is mainly because of the handicap caused by diffusion bonding and the fact that the interface between the inlet, outlet, and main body of MCHEs is decreasing compared with the MCHE with two plates.

**4.4. Applications of MCHEs in Thermal and Chemical Processes.** Heat management is important and ubiquitous in thermal and chemical processes. MCHEs can be applied in these processes, especially with space limitations, such as hydrogen generation systems for fuel cells[20] and miniplants for bioethanol dehydration to ethylene.[21] To fully illustrate this problem, a system of ethanol catalytic dehydration to ethylene was developed in our laboratory, which consisted of a dehydration reactor and a catalytic burner that supplies energy for ethanol−water vaporization and endothermic ethanol dehydration.

$$C_2H_5OH = C_2H_4 + H_2O \quad +46.6\ \mathrm{kJ/mol}\ (673\ \mathrm{K}) \tag{25}$$

For example, for a 1000 g ethanol solution (93.8 wt %), the energy required to heat ethanol to reaction temperature (673 K) from room temperature (298 K) is 1779 kJ, while that for an endothermic reaction is only 948 kJ, which means around 65% of the energy must be recovered to save the process energy. In order to recover energy from products of dehydration reaction, MCHE 2 was integrated to the dehydration reactor of this system.

In this experiment, the flow rate of ethanol solution with 93.8 wt % is kept at 3.8 mL/min. The experimental results indicate that more than 90% of the energy required to heat the reactant can almost be recovered via MCHE 2 with deionized water as the working fluid. The relationship of inlet and outlet temperatures and water volumetric flow rate was presented in Figure 14, in which the temperature of the products of the dehydration reaction decreases with the increasing flow rate and the temperature decreases from 659 to 301 K when the flow rate of the cold fluid is around 40 mL/min. Moreover, the heat exchanger effectiveness of MCHE 2 is more than 88% except when the flow rate is about 15 mL/min, which is because some of the deionized water has evaporated in this case.

## 5. Conclusion

In order to satisfy the heat transfer quantity or fluid outlet temperature, corresponding operation conditions (flow rate of working fluid, inlet temperature of the working fluid, working pressure, etc.) should be determined for the design of microchannel heat exchangers with multiple plates. To begin with, the material and structure of MCHEs' plates and the flow pattern of the working fluid are chosen. Further, the performance of MCHEs with two plates is experimentally tested and correlations of average Nusselt number and pressure drop to *Re* in microchannels are presented for designing MCHEs with multiple plates. In this work, the validity of correlations is verified experimentally through MCHEs with two and ten plates. Lastly, the number of plates and the total flow rate are determined. If the working pressure exceeds the maximum working pressure, the structure of MCHEs' plates needs to be redesigned. To sum up, the heat transfer quantity and working pressure should meet the qualifications simultaneously.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## Nomenclature

$A$ = heat transfer area, $m^2$
$c$ = cold fluid
$c_p$ = specific heat, J/kg·K
$C_R$ = ratio of heat capacity rate
$h$ = hot fluid
$k$ = heat transfer coefficient, kW/$m^2$·K
$k_v$ = volumetric heat transfer coefficient, MW/$m^3$·K
$K$ = overall heat transfer coefficient, kW/$m^2$·K
$m$ = mass flow rate, kg/s
$Nu$ = Nusselt number
$p$ = pressure, kPa
$Q$ = heat transfer rate, W
$R$ = thermal resistance, $m^2$·K/W
$Ra$ = arithmetical mean deviation of the profile
$Re$ = Reynolds number
$Ry$ = maximum height of the profile
$Rz$ = the point height of irregularities
$T$ = temperature, K

*Greek Symbols*

$\delta$ = wall thickness, $\mu$m
$\lambda$ = thermal conductivity, W/m·K
$\varepsilon$ = heat exchanger effectiveness

*Subscripts*

c = cold fluid
h = hot fluid
i = inlet
m = mean
o = outlet

t = total
w = wall

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