An Experimental Study on Heat Transfer and Pressure Drop of MTHE

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Abstract A micro-sized shell and tube heat exchanger (MTHE) was fabricated, and its performance in heat transfer and pressure drop was experimentally studied. The single-phase forced convection heat transfer correlation in the tube side of the MTHE was proposed and compared with previous experimental data in the Reynolds number range of 500-1800. The averaged deviation of the correlation in calculating the Nusselt numbers is about 6.59%. The entrance effect in the thermal developing region was discussed in detail. In the same range of Reynolds number the pressure drop and friction coefficients were found to be considerably higher than those predicted by the conventional correlations. The product of friction factor and Reynolds number is also a constant, but about one fold higher than the conventional. The reasons resulting in these physical phenomena have been preliminary discussed.

Keywords: Micro-tube heat exchanger, Heat transfer, Pressure drop, Entrance effect

1. Introduction

During the last two decades more and more compact systems in MEMS have resulted in a great demand for developing efficient heat transfer equipment to accommodate these high heat fluxes. One of the simplest arrangements that can be used for the heat removal involves using single-phase forced convection or flow boiling in small straight circular tubes or rectangular channels.

A number of theoretical and experimental investigations devoted to this problem were performed in these years. One of the early studies concerning circular microtube configurations was reported by Choi et al (1991). The conclusion was that the Reynolds analogy does not hold for microchannels. Bruno's (2004) experiments indicated that the derived formula of friction factor and heat transfer coefficient were consistent with those of large diameter tubes. Also experiments of Wahib and Biorn (2004) showed good agreement between the experimentally measured data and classical correlations .Primal et al (2008) found that the Nusselt numbers agreed with those predicted by the Gnielinski correlation within about 5% accuracy. It was found that the measured Nusselt numbers and friction factors were in good agreement with the theoretical values expected from Poiseuille flow in Boris (2010) studied. Krzysztof et al (2008) found that the critical Reynolds number is Re=2000, which is concurrent with macrotubes.

However, the results obtained by Mala and Li's (1999) , Hsieh et al. (2004), Jiang et al. (2006) and Liu et al (2006) showed that the pressure drops in microchannels were higher than the predictions by classical correlations.

In a summary that the heat transfer and fluid flow in microtubes or microchannels is a very complex issue. Many international scholars have devoted to this issue by both theoretical and experimental investigations. However, there is a little common accepted conclusion so far.

The aim of the present study is for further understanding the single phase convection heat transfer and pressure drop performance in MTHE.

2. Experimental setup and procedure

A micro sized shell and tube heat exchanger (MTHE) was made using 61 copper tubes, with outside diameter of 1.0 mm and wall thickness of 0.16 mm. The tubes, each with the total length of 154 mm (active heat transfer length of 140 mm), were with equilateral triangles arrangement having a tube pitch of 2.0 mm. The tube bundle was placed inside a copper shell and the ends of the tubes were welded to a tube plate. Connections were added at the ends of the MTHE to serve as inlet and outlet ports for the tube-side fluid. The total length of the MTHE was 178 mm and the outside diameter of the shell was 28 mm only.

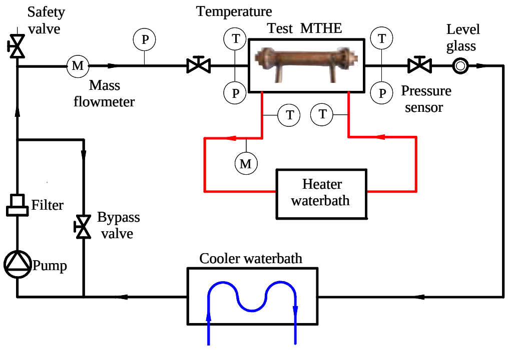

Fig. 1. Schematic of the experimental setup

The schematic diagram of the experimental test loop is shown in Figure 1. The circulation medium, R142b pumped by a metering pump, firstly entered into the mass flow meter and then passed into the tube side of the MTHE where heated by the hot water circulated through a temperature controlled water bath from the shell side. The heated working fluid, R142b, came from the MTHE were cooled to an appropriate temperature in the cool thermostatic water bath, then pumped by metering pump for constituting a closed cycle.

The MTHE was mounted horizontally. Two pressure sensors were mounted on the inlet and outlet of the MTHE, respectively, to measure the pressure drop of the working fluid. The four temperatures at both the inlet and outlet of working fluid and warm water were measured by T-type thermocouples. The mass flux rate of the working fluid in the MTHE was measured by mass flow meter.

After confirming the experimental system reached a hydrodynamic and thermal steady state, temperatures, pressures, and flow rates were recorded every 3s for approximately 5 min. The working fluid rate was then changed incrementally while the warm water flow rate was fixed. At each increment, measurements were recorded at stable conditions. In seven different shell side conditions, more than 100 different working fluid flows have been measured.

3. Results and discussion

It should be noted that both the internal and external heat transfer performances of the MTHE are unknown, Wilson Plot Method (WPM) were used here. In the method, the following assumptions have been made: firstly, the micro copper tubes have the uniform surface roughness and the same circular cross section; secondly, the fluid flow in the tube side has no maldistribution problem, and thirdly, the axial thermal conduction of the tubes is negligible.

3.1 Heat transfer

More than 116 measurements of the overall heat transfer coefficients at different flow velocities on tube side have been done in seven different shell side conditions. In each stabilized shell side heating and flowing condition, the overall heat transfer coefficients vary mainly with the flow of the tube side.

The inverse of the overall heat transfer coefficient ($1/K$) was plotted versus the inverse of the tube side fluid flow rate with a power index of n ($1/U^n$). Assuming the exponent n initially to be 0.8 in each measurement, the best linear fit to the obtained data for every stabilized shell side condition can be determined by using the least squares method.

It is found that the fitting linear curve having the exponent of n = 1.08 has the minimum deviation for all experimental data. But the intercepts are different, which are as shown in Fig.2. The intercept A decreases gradually with increasing the external heat transfer coefficient, h_0. The experimental *Re* number ranges from 500 to 1800.

Presenter	*Time*	*Correlations*	*Applicable scope*	*Scale*
Sieder-Tate	1936	$\overline{Nu} = 1.86\left(\frac{RePr}{l/d}\right)^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14}$	$0.48 < Pr < 16700,$ $0.0044 < \frac{\mu}{\mu_s} < 9.75$	conventional
Dittus-Boelter	1930	$Nu = 0.023Re_f^{0.8}Pr_f^{n}$	$Re_f = 10^4 \sim 1.2\times10^5$ $Pr_f = 0.7 \sim 120, \frac{l}{d} \geq 60$	conventional
Primal	2008	$Nu = \left(4.526\times10^{-4}\right)Re^{1.25}Pr^{0.4}\left(\mu/\mu_s\right)^{0.14}$	$2300 < Re < 6000$	microchannel
Debray	2001	$Nu = 0.0593Re^{3/4}Pr^{1/3}$	$3000 < Re < 20000$	microchannel
Yu et al.	1995	$Nu = 0.007Re^{1.2}Pr^{0.2}$	$250 < Re < 20000$	microchannel
Wu and Little	1984	$Nu = 0.00222Re^{1.09}Pr^{0.4}$	$3000 < Re < 20000$	microchannel
Choi et al.	1991	$Nu = 0.000972Re^{1.17}Pr^{1/3}$	$Re < 2000$	microchannel
Choi et al	1991	$Nu = \left(3.82\times10^{-6}\right)Re^{1.96}Pr^{1/3}$	$2500 < Re < 20000$	microchannel

Table 1: Correlations from the literature for both conventional and microchannels

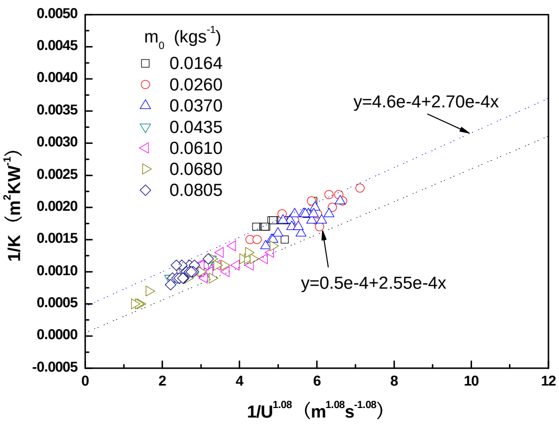

Fig. 2. The 1/K versus the inverse of the tube side flow rate with an exponent of 1.08 ($1/U^{1.08}$)

Therefore, the correlation for the single phase forced convection heat transfer was proposed as equation 1.

$$Nu = 0.0043Re^{1.08}Pr^{0.4}\left(\frac{\mu}{\mu_s}\right)^{0.14} \quad 500 < Re < 1800 \qquad (1)$$

The comparison of the experimental Nusselt number, Nu_e, with the calculated, Nu_c, is shown in Figure.3. It shows that the experimental *Nu* number, Nu_e, agree well with the predicted values by the correlation. The averaged negative and positive errors are 15% and 15% respectively. The mean absolute error (MAE) defined as below is about 6.59%, where *M* is the number of experimental condition.

$$MAE = \frac{1}{M}\sum\frac{\left|Nu_e - Nu_c\right|}{Nu_e}\times100\% \qquad (2)$$

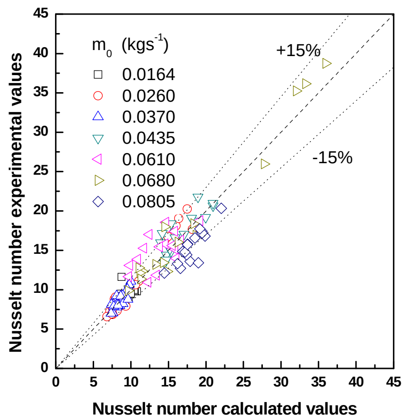

Fig. 3. Comparison of the experimental Nu_e with the calculated Nu_c

3.2 Experimental result comparison

In order to compare the present experimental results with those in the previous literature, some correlations for heat transfer in convectional tubes, microtubes, and microchannels are summarized and listed in Table 1.

Figure 4 shows the present experimental data and the correlations proposed by Sider and Tate (1936), and Dittus and Boelter (1930) for convectional tubes. It is shown that both the correlations could underestimate the Nusselt number in the *Re* number range of 500-1800. Further more studies are needed in order to determine whether the conventional correlations can be used for the microtube heat exchanger, for example, the outer diameter is 1mm or even less.

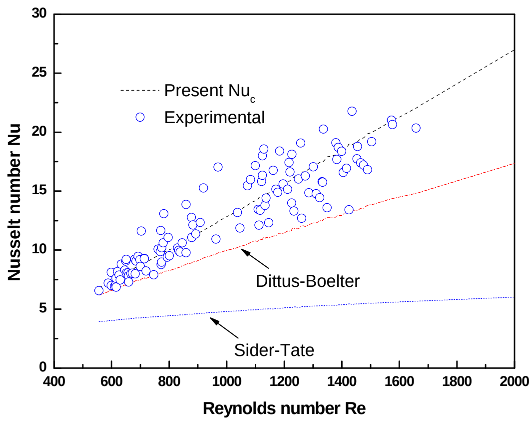

Fig. 4. Comparison of the experimental correlations proposed by various authors for conventional tubes

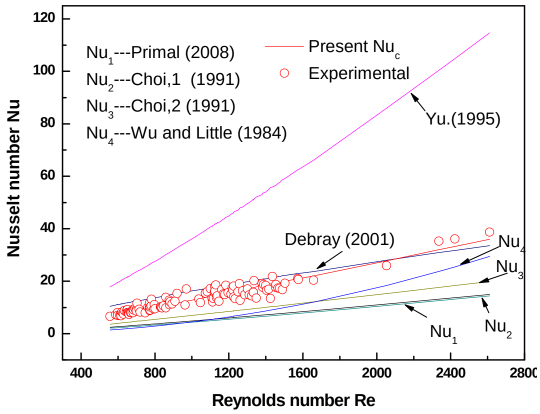

Fig. 5. Comparison of the experimental correlations proposed by various authors for conventional tubes

The comparison of the present experimental Nusselt number with those from the others correlations can be shown in Figure.5.It is that the experimentally measured Nusselt numbers are in good agreement with the result of Debray et al (2001) .The correlation proposed by Wu and Little (1984) 、Choi et al (1991) and Primal's (2008) will give a lower Nu number. The correlation suggested by Yu et al. (1995) give the highest Nusselt number among all of them .

The points are much scattered beyond our expectation as shown above. This could be probably caused by the relative surface roughness. In a fact, there has been no a convincing explanation so far about impact of the relative surface roughness on micro scale heat transfer.

3.3 The entrance effect

When fluid comes from a large space into a single pipe, the local surface heat transfer coefficient at the entrance region should be higher than that in the fully development section.

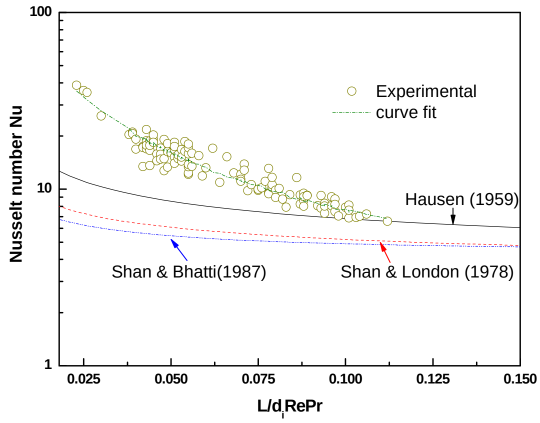

Fig. 6. The averaged Nu as a function of the inverse of Gz (axis title is Gz^{-1})

The experimental averaged Nusselt numbers, *Nu*, versus the inverse of the Graetz number, Gz^{-1} from the present experiments are also plotted in Figure 5. The results were also compared with those approximately averaged Nusselt numbers by integration of the Hausen formula, the Shah and London formula, and the Shah and Bhatti formula, respectively.

It is found to be that the average experimental Nusselt numbers are much higher than the others. With increasing the inverse of *Gz*, the difference between the experimental averaged *Nu* and the predicted by Hausen's formula decreases. The experimental averaged *Nu* gradually decreases to a stable value, which is more close to the curve of Hausen rather than those of the others. The experimental demarcation point separating from the entrance and developed regions also seems larger than that of the convectional tube of 0.05, which is about 0.1 or even larger according to our experimental data. This means that the entrance effect can have greater influence on micro scale than the conventional scale.

3.4 Pressure drop

The variation of the experimental pressure drop with the mass flux of R142b is shown in Figure 6. The pressure drop increases monotonically for the micro-tubes with the increase of the mass flux. Compared with the results for the conventional channels or tubes, the pressure drop through the micro tubes is relatively large. This is probably because that the inner surface roughness might have a larger effect on the microtube than the conventional tube.

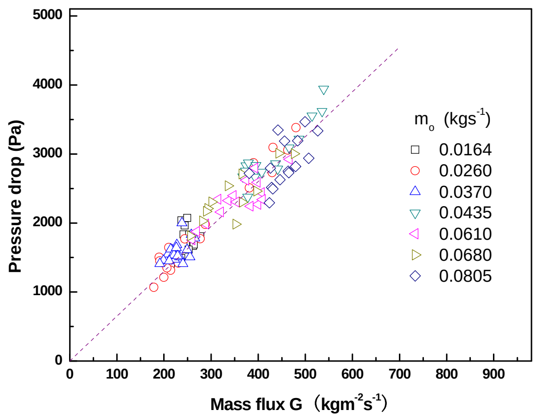

Fig. 7. Experimental pressure drop at different mass flowrate

The real MTHE may be sensitive to the external force considering the tiny geometric dimension of the heat exchange tube. The vibration caused by the cross flow is much lager than by the vertical mobility. Therefore the main reason for scattered experimental data may be that the vibration caused by the cross flow in the shell side of the MTHE.

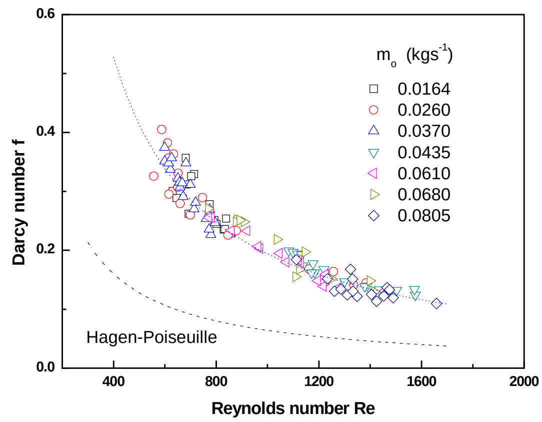

Fig. 8. Darcy friction factor f as a function of the Reynolds number

The correlation of Darcy friction factor *f* as a function of the Reynolds number is shown in Figure 7. With increasing the Reynolds number, the Darcy friction factor *f* correspondingly decreases, but at a higher level than the predicted by Hangen-Poiseuille correlation in all experiments. The correlation of Darcy friction factor *f* and the Reynolds number can be expressed as below

$$f = 368.6Re^{-1.094} \qquad 500 < Re < 1800 \qquad (4)$$

3.5 Uncertainty analysis

The detailed experimental uncertainties were listed in table 2.

Parameter	*Uncertainties (%)*
Temperature	0.67
Pressure	0.31
Mass flow rate (working fluid)	0.17
Mass flow rate (heat source)	0.1
Overall heat transfer coefficient	8.19
Convective heat transfer coefficient(tube - side)	8.40
Friction factor	1.32

Table 2: The variable uncertainties of parameters

The surface roughness on the inner side of the microtube and the wall shape specific distribution are not still clear. This experimental investigation can only give a preliminary understanding on the fluid flow and heat transfer characteristics in the microtube.

4. Conclusions

According to our experimental results, the following conclusions of this research may be summarized:

1. A single-phase forced convection heat transfer correlation in the tube side of the MTHE was proposed in a Reynolds number range from 500 to1800. and with a standard deviation of 6.59%. The experimental results were compared with correlations from the literature.

2. It is shown that the averaged Nusselt number is about 1.89-5.82 times larger than that of the conventional scale heat exchanger, and in the fully developed region, the averaged

Nusselt number is about 1.44 times larger. According to our results, the thermal entry length in a microtube is longer than that of the conventional tubes for laminar flow.

3. In the Reynolds number range of *Re* = 500-1800, the pressure drop and friction coefficients were found also considerably higher than those predicted by the conventional correlations.

The reasons resulting in these physical phenomena have been preliminary analyzed. The experimental uncertainties and their evaluation were given.

Acknowledgements

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