Thermal performance of a print circuit heat exchanger in a steam-air test facility

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Abstract. Printed circuit plate heat exchanger (PCHE) is a new type of microchannel heat exchanger with high compactness and excellent heat transfer performance. In this study, the performance of a 100×100×200 mm PCHE was evaluated. The thermal hydraulic performance of the PCHE was experimentally investigated in a steam-air heat exchange test setup. Experiments were carried out in the region of 1000<Re<3500. Pressure and temperature at the inlet and outlet of the PCHE were measured during the experiment for both the hot steam and the cold air side. Global Nusselt number correlations were then calculated and regressed from the experimental data. In addition, a full-scaled three-dimensional numerical simulation was performed using CFD method to make more discussion on the friction factors. With these correlations, PCHEs can be further optimized and designed.

1. Introduction

With the continuous development and the diversification of industry, the demand for energy utilization is increasing. Heat exchanger, as the basic equipment to realize heat transfer, has been attracted with more attention for the improvement of its efficiency [1]. A kind of plate heat exchanger named printed circuit plate heat exchanger (PCHE) can be expected to be used in those special industrial fields which needs higher heat transfer rate. PCHE was manufactured by the processes of chemical etching and diffusion bonding. Therefore, they are considered as a new type of plate heat exchangers with high compactness and high effectiveness. They are extremely suitable for gas-gas heat transfer with high pressure and high temperature.

The heat transfer experiment investigation offers an exact way to understand the characteristic of hydraulic performance and the pressure drop in order to design a PCHE. Kim et al. [2] carried out a lot of high pressure heat transfer experiments for different working fluids. Chen et al. [3] fabricated different PCHEs made of Alloy 617 and performed a test in the High Temperature Helium test Facility at the Ohio State University. He suggested that PCHEs with zigzag channels have better overall heat transfer performance. Meanwhile, for heat exchanger optimization, the CFD study is the easiest way. Some details that cannot be observed in the experiment will present a new perspective to understand the performance of the PCHE only by CFD. Aneesh et al. [4] proposed a new design of straight channel PCHE by 3D numerical simulation. CFD also plays an important role in the development of PCHE channel structure. For straight, zigzag, S-shaped and airfoil type channels, researchers have expended great efforts in attempting to optimize a PCHE [5~8].

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The present work is a continuous study in order to enrich the data evaluating the thermal hydraulic performance of PCHEs. The empirical correlations of the global Nusselt number and friction factor calculated from the experiment and simulation will guide our design of this kind of heat exchanger in the future.

2 Experiment setup and data

2.1 PCHE product and experiment setup

The PCHE used in this study was fabricated by chemical etching and diffusion bonding, made of stainless steel 316L for both the heat exchanger plates and the headers. Figure 1 shows the appearance of the PCHE used in the experiment. The dimensions of the PCHE core were 200x100x40 mm in size. In addition, the core was covered by a plate of 30 mm in thick.

![Figure 1. The product of the PCHE.](image)

Figure 1(b) shows the stacked PCHE core and one hot-side plate and one cold-side plate. A total of 10 hot plates and 10 cold plates were diffusion bonded together to form the core. Each plate was 200 mm * 100mm * 2 mm in volume. A total of 13 straight channels were photo-chemically etched in the plate. Table 1 lists the detailed geometry parameters of the PCHE.

| Parameter                          | Cold side | Hot side |
|-----------------------------------|-----------|----------|
| Number of plates                  | 10        | 10       |
| Number of channels in each plate  | 13        | 13       |
| Plate thickness, mm               | 2         | 2        |
| Channel diameter, mm              | 2.5       | 2.5      |
| Hydraulic diameter, mm            | 1.22      | 1.22     |
| Channel pitch, mm                 | 2.5       | 2.5      |

Experiments were carried out to test the thermal-hydraulic performance of the straight-channel PCHE under various steady-state conditions. Figure 2 is a simplified diagram of the experimental setup.

![Figure 2. Flow diagram of the experiment setup.](image)
It was made up of two flow path, one flowing the hot steam and the other flowing the cold air. The steam of 0.4 MPa was generated by a steam generator. The mass flow rate of the steam was measured by a vortex flowmeter. The air was transferred from the compressor, with the pressure setting to 0.8 MPa. Thermocouples, absolute and differential pressure gauges were installed along the flow loop and at the inlet and outlet of the hot and cold sides of the PCHE to record the data. In addition, insulation was provided along the steam flow path and on the PCHE in order to minimize the heat loss.

2.2 Experimental Condition and Data Reduction
As mentioned before, the steam was used as the hot side fluid of which the inlet temperature varied from 129 to 145 °C. In addition, the outlet state of the steam was carefully monitored to ensure that there was no condensate. The inlet temperature of the air varied from 6 to 15 °C. The total flow rate of the steam was measured in a range of 12.5 to 34.5 kg/h at the local pressure condition while the air flow rate changing from 8.5 to 22.5 kg/h at atmosphere condition.

To calculate the global Nusselt number, the heat transfer rate and coefficient of both the hot and cold sides should be first confirmed. The heat transfer rate was calculated mainly based on cold side. Using enthalpy in Eq. (1), the total heat transferred can be calculated.

\[ Q = m_c (H_{oc} - H_{ic}) \]  

(1)

Using the logarithmic mean temperature difference (LMTD), the effective heat transfer area and heat transfer coefficient, the heat transfer rate can be also obtained by Eq. (2):

\[ Q = KA \Delta T_{LMTD} \]  

(2)

where,

\[ \frac{1}{KA} = \frac{1}{h_c A_c} + \frac{1}{h_i A_i} + R_w \]  

(3)

Because the temperature of the inner wall was not measured, some assumptions need to be made to obtain the heat transfer coefficient directly from the experiment data. Suppose the temperature distributions along both the hot and cold sides were linear and the wall temperature on the plate was half of local temperatures of the hot and the cold fluid. Assume further that there was no heat loss across the solid plate and the effective heat transfer areas on both the hot and cold sides were the same. Moreover, the thermal resistance of the plate was neglected since the estimated value of \( R_w \) was 1% of the hot or the cold side thermal resistance in this experiment. Finally, the convective heat transfer coefficient of one side can be obtained from the experimental data directly. Further, the global Nusselt number can be obtained as,

\[ \frac{h D}{\lambda} \]  

(4)

The global friction factor was used to describe the pressure changed in the PCHE, shown as Eq.(5),

\[ \Delta p = 4 f_{PCHE} \frac{L}{D} \frac{1}{2} \rho v^2 \]  

(5)

Therefore,

\[ f_{PCHE} = \frac{\Delta p D}{2L \rho v^2} \]  

(6)

2.3 The experiment results and analysis
Figure 3 shows the global Nusselt number of both the hot and cold sides using the aforementioned method. The regression correlation is also plotted in Figure 3(a). The Nusselt numbers increase as the growth of Reynolds number. The best correction of the global Nusselt number was proposed in form of the Reynolds number and the Prandtl number, as follows,

\[ Nu = 0.016 Re^{0.81} Pr^{0.3} \]  

(7)
The coefficient of determination of the regression correlation was 96.1%. As seen in Figure 3, most of the experimental calculated Nusselt numbers were predicted within 5% by the global Nu correlation. The maximum deviation was no more than 20%.

![Figure 3. The global Nusselt numbers and regression correlation.](image)

3. Three dimension CFD simulation

3.1. A simple two channels PCHE model for validating the numerical method

![Figure 4. The physical model and three dimension grid for numerical certification.](image)

The two channels PCHE model is shown in Figure 4(a). The PCHE was simplified as two periodic channels with hot fluid flowing above while cold fluid flowing underneath as the corresponding solid plate. The total number of the generated meshes was approximately 2,656,486, as seen in Figure 4(b). Applying the periodic arrangement on the boundary, the model was used for the simulation. The geometry and the hydraulic parameter of the model can be found in the literature [2].

![Figure 5. The simulation results compare with the experimental data from literature.](image)

Figure 5 gives the simulation results of fanning friction factor and Nusselt number compared with the experimental data from the literature [1]. The friction factor is lower than that in the literature. This
was because that the experiment in the literature was carried out using the whole PCHE including the headers and some other connections. These would result in larger pressure drop. The most relative error of the friction factors simulation is under 5% and the Nusselt number performs excellent in prediction. These implied the simulation and the mesh generation method was reliable.

3.2. Full-scaled three dimension PCHE simulation

![Figure 6. Full-scaled 3D PCHE model and the meshes of the core.](image)

The full-scaled three-dimension PCHE was modelled following the geometry in Figure 2. The total number of generated meshes was approximately 82,309,296, as shown in Figure 6. Turbulence effects were considered by the RNG k-ε model and the enhanced wall treatment. The working conditions were the same as that in the experiment.

![Figure 7. Comparison of the global Nusselt number from simulations and experiments.](image)

The simulated global Nusselt numbers of full-scaled three dimension PCHE were compared with the experimental results, which is shown in Figure 7. The comparison showed that the simulated global Nusselt number agreed well with the experimental global Nusselt number. The maximum differences between the calculated values and experimental data were less than 10%.

![Figure 8. Simulated friction factors and regression deviation analysis.](image)
In the simulation, the total pressure drops were obtained and transferred to friction factors by Eq. (6), as shown in Figure 8. Because the structure of each channel was the same, the non-uniform flow maldistribution was avoided to the most extent. The regression equation of the friction factors can be expressed as Eq. (8). The regression deviation was less than about 5%.

\[ f_{PCHE} = 10.7 \text{Re}^{-0.8} \]  

(8)

4. Conclusion
Thermal hydraulic experiments and three dimension numerical simulations were performed on the PCHE. The experiment setup was built to test the performance of the designed PCHE. Using direct method with some idealized assumptions, the global Nusselt numbers were calculated and the empirical equation was proposed. As a supplement, the full-scaled three dimension PCHE simulation was performed to calculate the friction factors for reference. A simple two channels PCHE model for the validation of the simulation and the mesh generation method was firstly performed and the results showed that the 3D model was reliable. The full-scaled three dimension PCHE model was then applied using the same numerical method. The Nusselt number simulated agreed quite well with the experimental results and the deviation was less than 10%. The regression equation of the simulated friction factors was finally proposed and the regression deviation was less than about 5%.

Nomenclature

\begin{tabular}{ll}
\textbf{\(A\)} & heat transfer area, m² \\
\textbf{\(D\)} & hydraulic diameter, m \\
\textbf{\(f\)} & friction factor \\
\textbf{\(h\)} & convective heat transfer coefficient, W·m⁻²·K⁻¹ \\
\textbf{\(h\)} & overall heat transfer coefficient, W·m⁻²·K⁻¹ \\
\textbf{\(l\)} & channel length, m \\
\textbf{\(m\)} & mass flow rate, kg·s⁻¹ \\
\textbf{\(Nu\)} & Nusselt number \\
\textbf{\(p\)} & pressure, Pa \\
\textbf{\(Re\)} & Reynolds number \\
\textbf{\(Rw\)} & thermal resistance of the plates, m²·K·W⁻¹ \\
\textbf{\(T\)} & temperature, °C \\
\textbf{\(v\)} & channel velocity, m·s⁻¹ \\
\textbf{\(\Delta\)} & plate thickness, mm \\
\textbf{\(\rho\)} & density, kg·m⁻³ \\
\textbf{\(\lambda\)} & thermal conductivity, W·m⁻¹·K⁻¹ \\
\textbf{\(h\)} & hot side \\
\textbf{\(c\)} & cold side
\end{tabular}

Greek symbols

\begin{tabular}{ll}
\textbf{\(K\)} & overall heat transfer coefficient, W·m⁻²·K⁻¹ \\
\textbf{\(\delta\)} & plate thickness, mm \\
\textbf{\(\rho\)} & density, kg·m⁻³ \\
\textbf{\(\lambda\)} & thermal conductivity, W·m⁻¹·K⁻¹
\end{tabular}

Subscripts

\begin{tabular}{ll}
\textbf{\(h\)} & hot side \\
\textbf{\(c\)} & cold side
\end{tabular}

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