A study on investigating the thermal and energy saving performances of flat plate heat exchanges (FPHE) with different corrugated channels configurations

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ABSTRACT

Due to scarce conventional energy resources, energy consumptions in the buildings occupied a large proportion of the overall energy consumptions. Previous studies investigated the application of the plate heat exchangers in the heat recovery of air ventilation systems. The objective of this paper is to offer new methods to improve the thermal and energy saving performances of flat plate heat exchanges (FPHE) with different corrugated channel configurations and provide a guidance for the optimization design of a novel plate heat exchanger.

This paper firstly compared the predicted simulation results with previous experimental results and verified the validation of the numerical methods. Then, this paper used the numerical method to investigate the heat transfer characteristics and the pressure drop as well as the flow distributions in plate heat exchangers with three different corrugated configurations(triangular, semicircle and trapezoidal) and different the channel spacing. The numerical results showed that, corrugated configurations not only had a significant positive impact on heat transfer enhancement, but also led to an increase in the pressure drop through the corrugated channel. It can be seen that the effects of the corrugated channel configurations on the heat transfer and pressure drop were various for different corrugated channels at different *Re* numbers. In addition, the comparisons of performance factors identified that when the channel spacing was larger, it was more appropriate to choose a triangular corrugated channel at a lower *Re* number. However, at a higher *Re* number, selecting the semicircle corrugated channel may be more suitable.

1. INTRODUCTION

A plate heat exchanger (PHE) is a type of heat exchanger that uses metal plates to transfer heat between two fluids. This facilitates the transfer of heat, and greatly increases the speed of the temperature change. The corrugated plate heat exchanger has many advantages compared to plate heat exchangers as it gives maximum heat transfer; there is an increase in area due to the presence of corrugations which gives rise to turbulence and this helps in its usage in high viscosity applications as turbulence is also induced at low flow rates. However, the heat transfer increase is typically accompanied by an increase in the fluid pressure drop and pumping power. Therefore, the identification of the corrugated geometry and arrangement giving the best heat transfer performance should take into account both heat and momentum transfer characteristics. Many parameters affect the performance of rib-roughened channels: the shape and dimensions of the channel, the rib characteristics (including shape, size, arrangement, etc.), and the flow characteristics (through the Reynolds and Prandtl numbers) [Ekadewi *et al*., 2016, Naznil *et al*., 2016 and Giovanni, 2011]. Among these parameters, the role played by the rib installation has often been recognized as the most important. Consequently, a considerable amount of literature work has been devoted to evaluating heat transfer and friction characteristics of channels with various forms of configurations.

Sparrow and Comb[1983] studied the effect of channel spacing on heat transfer performance of a triangular parallel corrugated plate. When the channel spacing was increased from 5.08 mm to 7.37 mm, its heat transfer performance increased by 30% and pressure drop also increased by nearly one time. Therefore, the overall heat transfer performance decreased. Xu and Min [2004] performed steady-state and non-steady-state simulations on a triangular corrugated plate. The simulation results obtained by the two methods were nearly identical, so the steady state method can be used for investigation method. It is also pointed out that when the Reynolds (*Re*) number was less than 400, the laminar flow model can be used. When the *Re* number was greater than 400, the k-turbulence model will be used. Sui et al. [2010] performed a three-dimensional numerical simulation of the laminar fluid flowing into the corrugated plate channel. The results showed that the secondary swirling flow was generated in the corrugated channel, and chaos convection also generated along the direction of the flow, which promoted the mixing of the convection fluid and enhanced the heat transfer performance. Naphon[2007 and 2009] performed numerical studies on the heat transfer performance of wave plates with different geometries, under constant wall heat flow conditions. The results showed that a sharp edged corrugated surface, such as a triangle, had a more pronounced effect on flow distribution and heat transfer enhancement. Yin et al. [2011] studied the fluid flow and heat transfer performance in a sinusoidal parallel channel at a uniform wall surface temperature. The results found out that the frictional resistance and the average Nusselt (*Nu*) number increased with the degree of sinusoidal fluctuation and the channel spacing of the corrugated plates. Moreover, the experimental results show that when the corrugation angle is about 80°, the heat transfer performance was best and the pressure drop also higher. Sunden and Karlsson [1991] carried out experiments to investigate heat transfer and pressure drop performance in trapezoidal channels. It was shown that the corrugated channel with a larger angle led to better heat transfer enhancement, scarifying with a higher pressure drop.

From the above previous experimental and simulation studies, using corrugated surfaces with various configurations improves energy performance of the plate heat exchanger, but also lead to a significant increase in pressure drop and hence the pumping power. However, selecting the optimal corrugated configuration for highest performances is complicated because it depends upon the height, the length and pitch of the ribs. The objective of this study is to investigate the thermal and energy saving performances of flat plate heat exchanges with different corrugated channel configurations (triangular, trapezoid and semicircle). Firstly, the model development is reported, including corrugation geometry, mesh sensitivity and numerical methods. And then, the numerical model will be validated, by compared with the numerical and the previous experimental results. Lastly, the results analysis is given in detail.

2. MODEL DEVELOPMENT

2.1 Corrugated channel configurations

Five geometric models will be developed in this paper, including flat plate channel and four kinds of corrugated plate channels with different shapes: triangular (20^o), triangular (45^o), trapezoid and semicircle. The purposes of the establishment of the five geometric models are as follows:

- Developing the flat plate channel model is to provide a baseline to compare its performance factor, PE, with those of four kinds of corrugated plate channels;
- Developing the triangular (20^o) plate channel model is to validate the numerical model with the previous experimental results;
- Developing the other three kinds of corrugated plate channel models is to investigate the effects of different corrugated channel con figurations on the heat transfer performance and pressure drop.

The five geometric models are shown in Figs 1.1 to 1.5 and their specific parameters listed in Table 1.

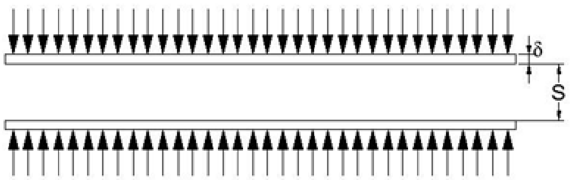

Figure 1.1: Flat plate channel

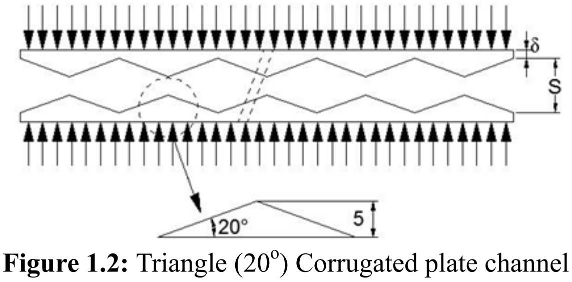

Figure 1.2: Triangle (20^o) Corrugated plate channel

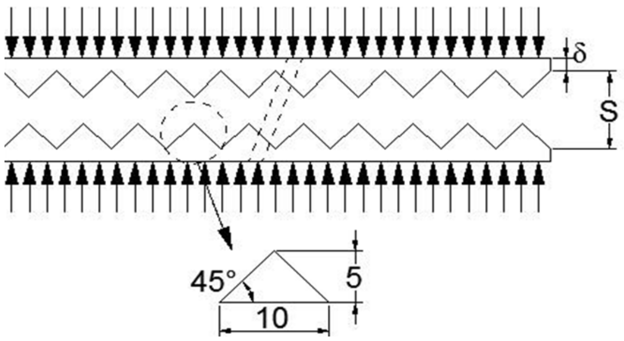

Figure 1.3: Triangle (45^o) Corrugated plate channel

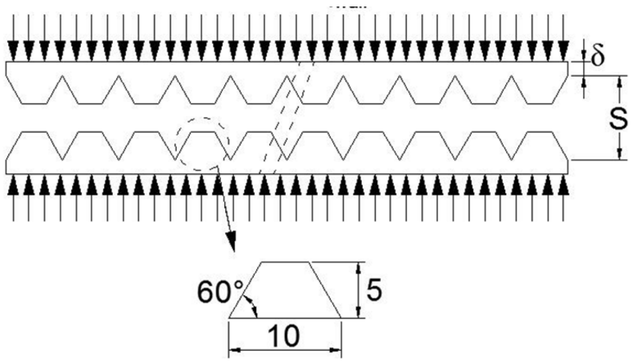

Figure 1.4: Trapezoid corrugated plate channel

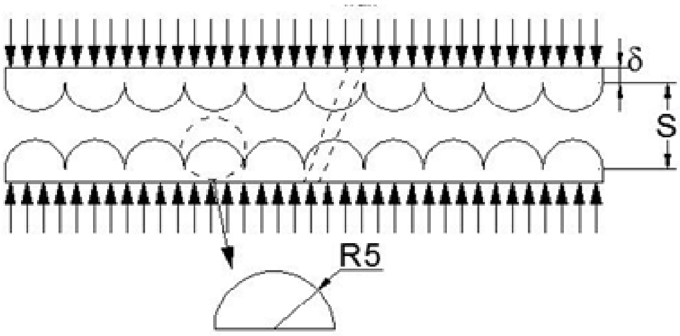

Figure 1.5: Semicircle corrugated plate channel

Figure 1: The five different corrugated plate channel

Table 1: Dimensions of different corrugated plate channels

Rib-grooved wall types	**Depth (Y) (mm)**	**Height of the base (δ) (mm)**	**Channel spacing (S) (mm)**	**Corrugated length (mm)**	**Number of the ribs and grooves**	**Channel length (L) (mm)**
Flat	—	2.5	12.5 15 17.5	300	—	300
Triangular (20^o)	5	2.5	12.5 15.0 17.5	319.8	11	300.3
Triangular (45^o)	5	2.5	12.5 15.0 17.5	424.3	30	300
Trapezoidal	5	2.5	12.5 15.0 17.5	473.2	30	300
Semi-circular	5	2.5	12.5 15.0 17.5	471.2	30	300

2.2 Grid generation

In this section, the model for triangular (20^o) corrugated plate channel was taken as a example for grid independence testing. To evaluate the number of elements required, four different meshes were tested. The grid independence test is carried out in the analysis by adopting four different grid distributions of 80,642、104,148、130,864 and 153,884. The grid independence test indicated that the grid systems of 48,000 ensure a satisfactory solution. It is found that after 104148 cells, further increase in cells gives less than 3% variation in average *Nu* umber value which is taken as criterion for grid independence.

2.3 Boundary conditions

The numerical solution domain is considered 2D, enclosed by the outlet, inlet and wall boundaries. For all walls, no slip condition is assumed for momentum equations on walls. The inlet velocity values have been derived from the given Re numbers. The outlet boundary condition is called 'pressure outlet', which implies a static (gauge) pressure at the outlet boundary, which means the pressure will be extrapolated from the flow in the interior.

2.3.1 Wall boundary conditions

$$q_w = 290\frac{W}{m^2} \tag{1}$$

2.3.2 Inlet boundary condition

The uniform profiles for all properties are as follows: a fully developed steady flow condition. The inlet velocity values have been derived from the given Reynolds numbers. The specification method for inlet boundary is intensity and hydraulic diameter. The intensity is at 5% for all cases and the hydraulic diameter is calculated according to the channel spacing.

2.4 Boundary conditions

The Realizable k-epsilon turbulent model was selected with the enhanced Wall Treatment. The second-order upwind was used for convections and diffusions, respectively. The discretisation was achieved by using the PRESTO method for pressure and the second-order upwind method for momentum, energy, turbulent kinetic energy and the turbulent dissipation rate equation. The SIMPLE algorithm was used for flow computations.

3. MODEL VALIDATION

Due to the limitations of experimental equipment and conditions, the numerical results will be compared with the experimental results in a previous paper [Naphon, 2007]. The experimental details can be referred to the paper [Naphon, 2007]. In this paper, the experimental results of the triangular (20^o) corrugated plate channel (S=12.5mm, q_w=580W/m^2) were compared with the corresponding simulation results to verify the numerical method.

Fig. 2 shows the comparison between the predicted averaged *Nu* number in the corrugated channel and the experimental results. It can be seen that the predicted values of averaged *Nu* number in the corrugated channel, obtained by the numerical model, were similar with those by the experiments and the errors were less than 5%. Therefore, the numerical method, developed in Section 2, is reliable and can be used to study the thermal and energy saving performance of the plate heat exchanges (FPHE) with different corrugated channels configurations in the next Section.

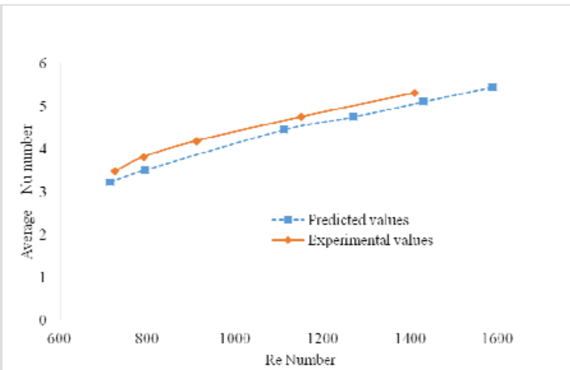

Figure 2: The comparisons between the predicted averaged *Nu* number in the corrugated channel and the experimental results

4. RESULTS ANALYSIS AND DISCUSSION

4.1 Performance index

The current numerical simulations are in order to study the influence of using different corrugated channel rib shapes on heat transfer characteristics and pressure drop for a fully developed flow in corrugated channels. Four performance indexes will be introduced firstly in this section, such as average Nu number, *Nu*, pressure drop per unit, △P/L and performance factor, PE.

The average reference velocity in the channel, was calculated by the Reynolds number based on the channel hydraulic diameter which is defined as:

$$u_{av} = \frac{R_{ev}}{D_h} \tag{2}$$

Where D_h is expressed as[Elshafei et al. 2008]

$$D_h = 4SW / (2S + 2W) \tag{3}$$

The average heat transfer coefficient along the corrugated channel was defined by:

$$\mathrm{h}_m = \frac{Q_{in}}{A_c\left(\Delta T_m\right)} \tag{5}$$

The logarithmic temperature difference between the wall and the fluid was calculated as

$$\Delta T_m = \frac{\left(T_w - T_{a,i}\right) - \left(T_w - T_{a,o}\right)}{\ln\left[\left(T_w - T_{a,i}\right) / \left(T_w - T_{a,o}\right)\right]} \tag{6}$$

The average heat transfer coefficient is given in terms of the mean Nusselt Number (Fernández-Seara et al. 2011) as follows:

$$Nu = \frac{h_c H \overline{\sigma}}{\lambda L_t} \tag{7}$$

The pressure drop per unit was as follows:

$$\frac{\Delta p}{L} = \frac{(p_{av,o} - p_{av,i})}{L_t} \tag{8}$$

To achieve the performance assessment of such narrow channels based on their heat transfer characteristics and pressure drop, Performance factor is almost considered, defined as (Elshafei et al. 2008)

$$\mathrm{PE} = \frac{\left(\mathrm{Nu_c} / \mathrm{Nu_s}\right)}{\left(\mathrm{f_c} / \mathrm{f_s}\right)^{1/3}} \tag{9}$$

Where f is the friction factor which is calculated by

$$f = \frac{\Delta \mathrm{p}\left(\mathrm{D_h} / \mathrm{L_t}\right)}{\left(1/2\right)\rho \mathrm{u}_{\mathrm{av}}^2} \tag{10}$$

4.2 Thermal and energy saving performance of the plate heat exchanges (FPHE) with different corrugated configurations

In this Section, air is used as the working fluid. The effects of different channel spacing (S=12.5mm, 15.0mm and 17.5mm) and different corrugated configurations (triangular (45o), semicircle and trapezoid) on the heat transfer and pressure drop characteristics in the corrugated channel will be investigated.

4.2.1 Comparisons of average *Nu* number

The values of average Nu number in three different corrugated configurations are shown in Figs 3 to 5, respectively. It can seen that, the average Nu numbers increased with the increase in *Re* number under all cases. It because that the *Re* number increased, the turbulences in the corrugated channel increased, destroying the thermal boundary layer near the wall surface and enhancing the heat transfer.

However, for different corrugated configurations, the effects of the channel spacing on average Nu number was slightly different. According to Equation (7), for the a certain corrugated configuration, the values of $\lambda, \overline{\sigma}$ and L_t are constants. The *Nu* number are determined by h_c and H. For the triangle corrugated channel, when the *Re* number is small, the value of H will decrease slightly and h_c increase significantly with the decrease in the channel spacing. However, when the *Re* number increased up to nearly 2000, the effect of the decrease in H on the *Re* number is similar with that of the increase in h_c. From Fig.3, it shows that for the triangular (45^{o}) corrugated channel, when the *Re* number and the spacing channel were small, the corresponding average *Nu* number will be smaller. However, when the *Re* number increased up to around 2000, the channel spacing have no effect on the average *Nu* number. From Figs 4 and5, for the semicircle and trapezoid corrugated channels, the effects of channel spacing on the *Nu* number was the same under different *Re* number.

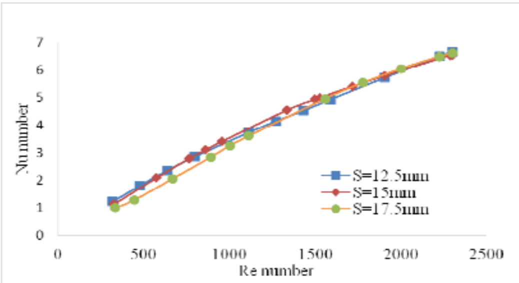

Figure 3: The relationships between the average *Nu* number and the *Re* number for the triangular (45^{o}) corrugated channel (S=12.5mm, 15.0mm and 17.5mm)

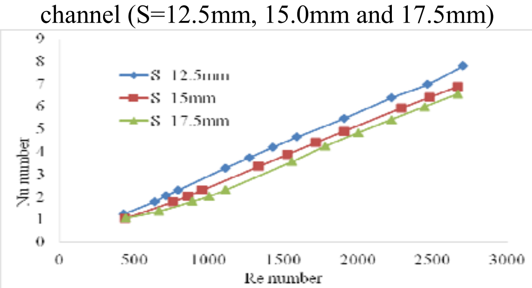

Figure 4: The relationships between the average *Nu* number and the *Re* number for the semicircle corrugated channel (S=12.5mm, 15.0mm and 17.5mm)

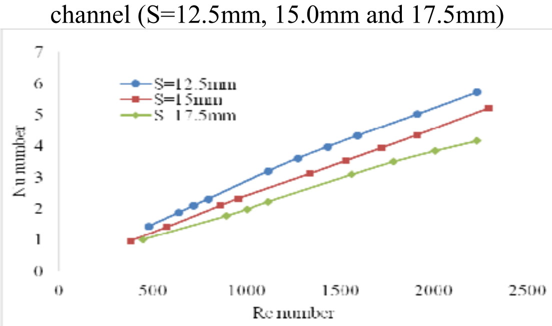

Figure 5: The relationships between the average *Nu* number and the *Re* number for the trapezoid corrugated channel (S=12.5mm, 15.0mm and 17.5mm)

4.2.2 Comparisons of pressure drop per length

Figs.6 to 8 show the relationships between the pressure drop per length and the *Re* number for the three different corrugated channels, respectively. Under all the cases, the pressure drop per length increased with the increase in *Re* number. Especially at the channel spacing, 12.5mm, the pressure drop per length were significantly larger than that for the other channel spacing, 15mm and 17.5 mm. With respect to the flat plate channel, the flow through the corrugated channels were complicated. Swirling and turbulent flows were significantly increased due to the corrugated channel, resulting a larger flow resistance. The pressure drop per length for the triangle corrugated channel was obviously larger than that for the other corrugated channels. The pressure drop per length for the trapezoid corrugated channel was slightly larger than that for the semicircle corrugated channel. As show in Figs. 9 to 11 , the vortex flow area on the triangle corrugated channel is larger and the direction of the vortex flow is not the same as the direction of the mainstream fluid, so the generated flow resistance will be larger. Compared with triangle corrugated channel, there is buffer zone which is parallel with the main fluid flows for the trapezoidal channel, so the flow resistance becomes smaller. In the semi-circular corrugated channel, there is a smooth transition for the corrugated shape, so their pressure drop is the relatively smallest.

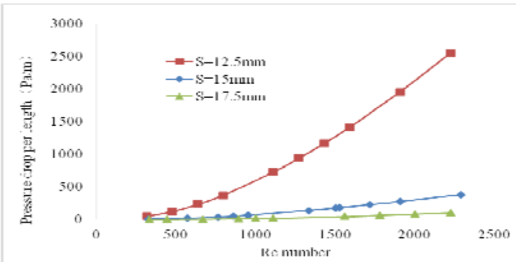

Figure 6: The relationships between the pressure drop per length and the *Re* number for the triangular (45^o) corrugated channel (S=12.5mm, 15.0mm and 17.5mm)

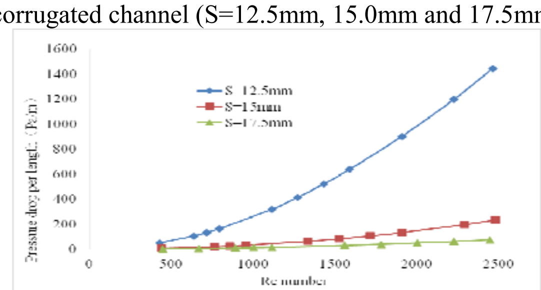

Figure 7: The relationships between the pressure drop per length and the *Re* number for the semicircle corrugated channel (S=12.5mm, 15.0mm and 17.5mm)

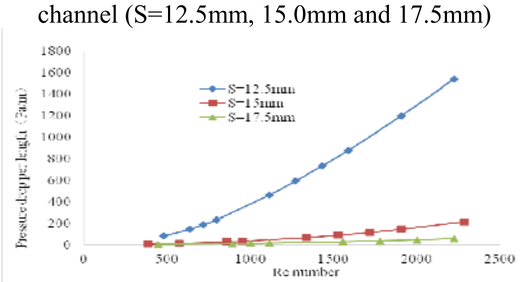

Figure 8: The relationships between the pressure drop per length and the *Re* number for the trapezoid corrugated channel (S=12.5mm, 15.0mm and 17.5mm)

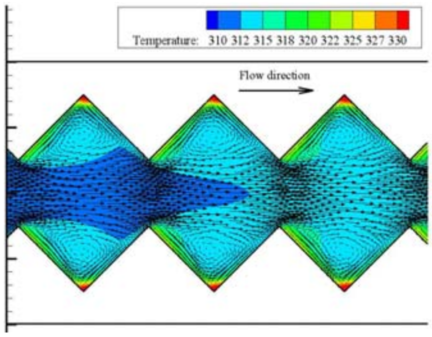

Figure 9: The air temperature distribution and flow velocity vectors in the triangle corrugated channel (S=15.0mm, Re=1500)

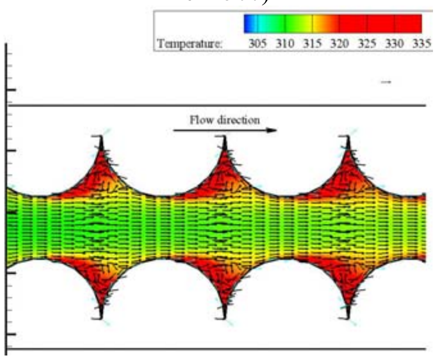

Figure 10: The air temperature distribution and flow velocity vectors in these semicircle corrugated channel (S=15.0mm, Re=1500)

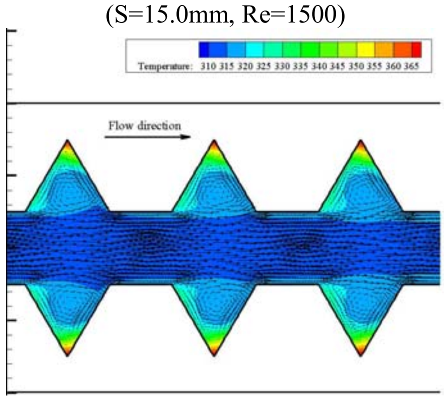

Figure 11: The air temperature distribution and flow velocity vectors in the trapezoid corrugated channel (S=15.0mm, Re=1500)

4.2.3 Comparisons of performance factors, PE

Figs.12 to 14 show the comparisons among the performance factors for the three different corrugated channels, respectively, at three different channel spacing. It can be seen that the values of the performance factor increased with the increase in *Re* number and were smaller than 1 for all the cases. This meant that compared with the flat plate, the increase in pressure drop, resulted by the corrugated configuration, were larger than that in heat transfer performance.

From Fig.12, when the *Re* number was small, the performance factors for the three different corrugated channels had no much difference. However, when *Re* number increased, the performance factors for the semicircle corrugated channel were larger than that for the other two corrugated channels. From Fig.13 and Fig.14 , when the *Re* number was small, the performance factors for the triangular corrugated channel were larger than that for the other two corrugated channels. However, when *Re* number increased, the performance factors for the semicircle corrugated channel increased largely and were larger than that for the other two corrugated channels at last.

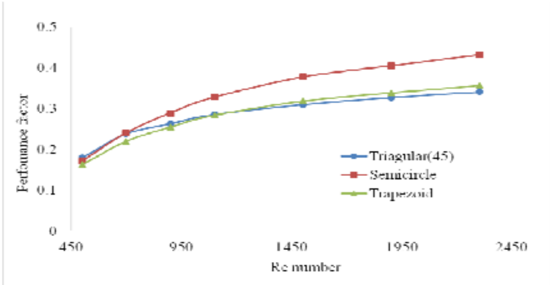

Figure 12: The comparisons among the performance factors for the three different corrugated channels (S= 12.5mm)

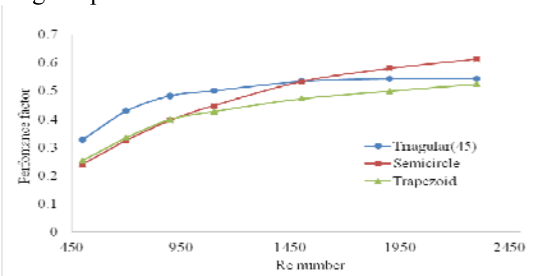

Figure 13: The comparisons among the performance factors for the three different corrugated channels (S= 15mm)

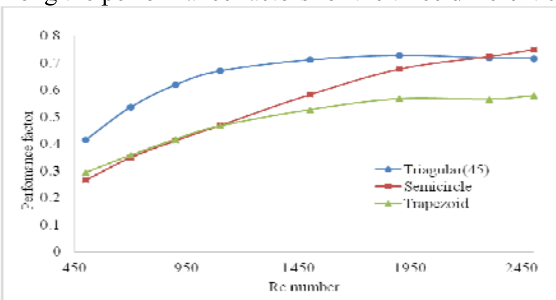

Figure 14: The comparisons among the performance factors for the three different corrugated channels (S= 17.5mm)

In conclusion, the performance factor is a parameter that comprehensively measures the overall heat transfer and pressure drop performance of different corrugated channels. It provides the designer the guideline to design the appropriate corrugated plate, according to different channel spacing and *Re* number. It has important significance for designing and manufacturing compact plate heat exchangers. Therefore, a specific corrugated channel plate should be selected in conjunction with different applications and operating conditions.

5. CONCLUSIONS

The following conclusions can obtained in this paper:

- Compared to the flat plate channel, the three corrugated channels had a significant effect on enhancing heat transfer, due to the increased heat transfer surface area and increased flow turbulence, thinning and destroying the thermal boundary layer, especially at a larger *Re* number;
- If only the heat transfer performance is considered, the heat transfer performance of the triangular (45^o) corrugated channel is significantly better than that of the semicircular and trapezoid corrugated channels.
- If only the pressure drop is taken into account, the pressure drop for the three corrugated channels are similar at a lower Re number. With the increase in *Re* number, the pressure drop of the triangular (45^o) corrugated channel is the largest and that of the trapezoid corrugated channel the smallest.
- If the heat transfer performance and pressure drop are considered together, at the channel spacing, 12.5 mm, the overall performance of the semicircular channel is the best. When S=15 or 17.5mm, the triangular (45^o) corrugated channel is the proper choice at the lower *Re* Number and the semicircular channel at a larger *Re* Number.

NOMENCLATURE

A	area	m^2
C_p	specific heat	kJ/(kg·°C)
D_h	hydraulic diameter	m
f	friction factor	-
h	heat transfer coefficient	kW/(m^2·°C)
H	the channel spacing	m
L	length	m
m	mass flow rate	kg/s
Nu	Nusselt number	-
ΔP	pressure drop	kPa
q_w	the heat flux	W/m^2
Q	heat transfer rate	kW
Re	Reynolds number	-
T	temperature	°C
u	velocity	m/s
W	input power	W
$\overline{\sigma}$	Distance the leading edge of the corrugated along the corrugated surface	m

Subscript

a	air
av	average
in	inlet
out	outlet
c	corrugated plate

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