

The effect of hemispherical chevrons angle, depth, and pitch on the convective heat transfer coefficient and pressure drop in compact plate heat exchangers

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ABSTRACT

Plate heat exchangers are widely used in industries due to their special characteristics, such as high thermal efficiency, small size, light weight, easy installation, maintenance, and cleaning. The purpose of this study is to consider the effect of depth, angle, and pitch of hemispheric Chevrons on the convective heat transfer coefficient and pressure drop. In the simulation of the heat exchanger, water and stainless steel are chosen for fluid and plate materials, respectively. The process is considered to be steady state, single-phase, and turbulent. In brief results show that the convective heat transfer coefficient and pressure drop decrease where the Chevrons depth and pitch increase. Moreover, these parameters enhance increment of the Chevrons angle up to 90°, after which they decrease with the Chevron angle. lastly, results are compared with Kumar equation which has been presented for corrugated plates. Maximum relative difference in this comparison is approximately 30%. As a result, a new correlation is proposed for the convective heat transfer coefficient in terms of the Reynolds number and the plate geometry.

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1. Introduction

Plate heat exchangers (PHE) are among the most employed thermal facilities in thermal facilities, e.g. in oil and gas plants, refrigeration, etc., due to their high thermal efficiencies. PHEs are mainly made of thin plates that are pressed into a whole frame and shaped in a package. According to the application, PHEs are designed in various types based on the plates and frame specifications. Gasket PHEs are much more sensitive to temperature and pressure than

other types, mainly because of their structures which are basically specified by plate perforations and gasket designs.

A single plate comprises four corner ports and a corrugated area. The corrugation pattern has a Chevron angle evaluated as the angle between the corrugated passes. In a PHE unit, the plates are installed with the Chevron apexes pointing in opposite directions. The Chevron design increases the flow turbulence level, the effective heat transfer area, and the stiffness of the plate pack; it may also decrease the fouling risk mainly due to high level of turbulence and wall shear force. The Chevron angle of commercially available plates is perhaps the most important geometric parameter of a

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PHE relating to its thermal and hydraulic performance. Conventional plates have approximately sinusoidal profiles.

The complex geometry of PHEs makes it difficult to develop a correlation for its convective heat transfer coefficient. As a result, numerical simulation is a powerful tool to determine the flow thermal behavior in these heat exchangers. Focke et al. [1] verified that at the Chevron angles up to about 80°, the fluid flows mostly along the corrugations. In addition, a secondary swirling flow may be imposed on the fluid flow as it is crossed by the opposite plate stream.

Gaiseret et al. [2] studied the effect of the combination of wavelength and Chevron angle of the corrugated plate on the local heat transfer coefficient, overall heat transfer, and flow characteristics in PHEs. Their results indicated that both the wavelength and the angle inclination influence the average heat transfer and homogeneity of the local heat transfer coefficient. Moreover, the results showed that large wavelengths and medium angles may cause more uniform local heat transfer.

Muley et al. [3] considered three different corrugated plate arrangements in a single-pass U-type counter flow plate heat exchanger in order to examine its heat transfer rate and pressure drop. The results indicated that as the Chevron angle increases, Nusselt number enhances dramatically. The increase of surface area enlargement factor also has similar, though smaller, influence on Nusselt number.

Dovic et al. [4] investigated the effect of different geometric parameters including the Chevron angle, the depth, and the corrugation wavelength on the PHEs performance. This study showed that the effect of flow components on the heat transfer rate depends not only on the Chevron angle and Reynolds number, but also relies on the aspect ratio. In addition, their results indicated that at fixed Reynolds number and Chevron angle, the pressure drop increases more than the heat transfer for plates with higher aspect ratio.

Durmus et al. [5] experimentally investigated the effect of surface geometry of three various types of heat exchangers (flat, corrugated, and asterisk) on the heat transfer rate, the friction factor, and the exergy loss. Andersson et al. [6] examined how well a fouling model of a shell-and-tube heat exchanger fits data obtained from monitoring of a Compabloc unit. Besides, they obtained

an optimized operating condition for these types of heat exchanger in preheat trains.

Han et al. [7] simulated a corrugated plate heat exchanger and obtained three dimensional temperature, pressure, and velocity fields and compared the results with the experimental data. Within the temperature field, it is noticeable that the temperature gradient gradually increases in the entrance zone and becomes smaller again in the central zone. In addition, it is remarkable that the pressure gradually reduces along the flow direction.

Muthuraman [8] studied the characteristics of a brazed plate heat exchanger at different Chevron angles. Condensation heat transfer coefficient and pressure drop was experimentally measured for R22 and R410a refrigerants in three types of heat exchangers.

Tamakloe et al. [9] studied thermo-hydraulic performance of two welded PHEs. The first case was a heat exchanger in which heat from the residues leaving a vacuum distillation unit was used to heat crude oil. The second one was a heat recovery used between a column pump around stream and crude oil. They showed that integration of the fouling behavior in this type of heat exchanger is necessary for future designs as the fluid shear stress produced in the PHEs can be high enough to minimize the fouling but at a cost of pressure drop increase.

Faizal et al. [10] performed an experimental study on a corrugated plate heat exchanger for small temperature difference applications. The spacing between the plates was varied in order to determine the optimized configuration. The results showed that at a fixed space, the average heat transfer rate between two fluid flows, and the pressure drop increase with the water flow rates.

Fahmy [11] investigated the performance of a PHE used in a research reactor to update and raise its power to 10 MW. In their study, the effect of the Chevron angle and the corrugation pitch on the surface area density, the total number of corrugated plates, and the pressure drop are studied. In addition, the effect of Chevron angle on the heat transfer coefficient was investigated.

In this study, a numerical simulation is performed to investigate the flow thermal behavior in a new type of compact plate heat exchanger with hemispherical Chevrons. The focus is on the effect of the Chevron angle, depth, and pitch on the convective heat transfer coefficient and pressure drop.

2. Materials and Methods

Figure 1 shows the geometry of the heat exchanger model whose specifications are summarized in Table 1. Water and stainless steel are chosen for fluid and plate materials, respectively.

As shown in Fig.2, the geometry has been meshed using unstructured tet/hybrid and tgrid. To increase the accuracy of the results, the grid has been refined in regions with high velocity gradient. Figure 3 shows the variation of the convection heat transfer coefficient with the number of grids. Based on the

results, the reduction of the convection heat transfer coefficient is less than 1% as the number of grids is more than 320,000. As a result, 320,000 meshes guarantee the independency of the simulation results from the number of grids.

For an incompressible flow, the governing equations are given as

$$\nabla(\rho V\phi - \Gamma_{\phi}\nabla\phi) = S_{\phi}, \quad (1)$$

where V is the velocity vector and the effective diffusion coefficient, Γ_{ϕ} , and the source term, S_{ϕ} , for different parameters, ϕ , are listed in Table 2.

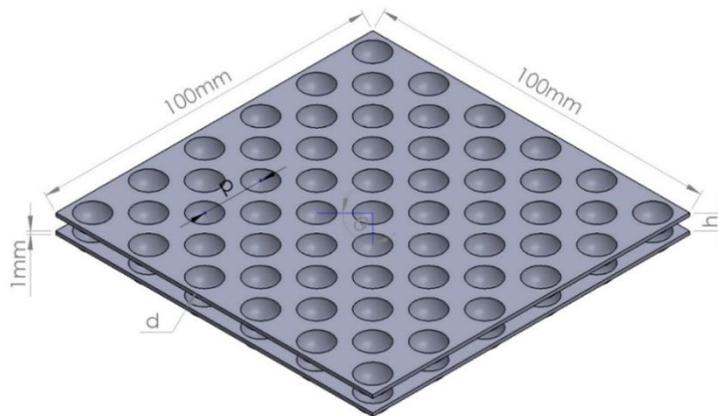


Fig. 1. Schematic of the model geometry

Table 1. Geometric characteristics of the models

$\Theta(^{\circ})$	P/D	H (MM)	D (MM)
70	1.25	2.5	2.5
80	1.5	5	5
90	2	7.5	7.5
110	2.5	-	-
120	-	-	-

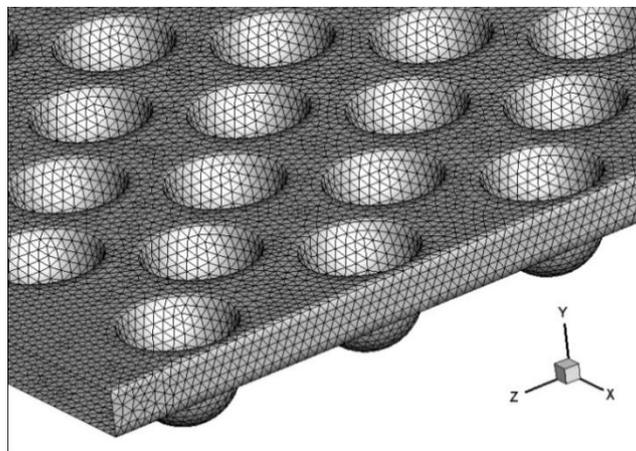


Fig. 2. Schematic of the unstructured mesh

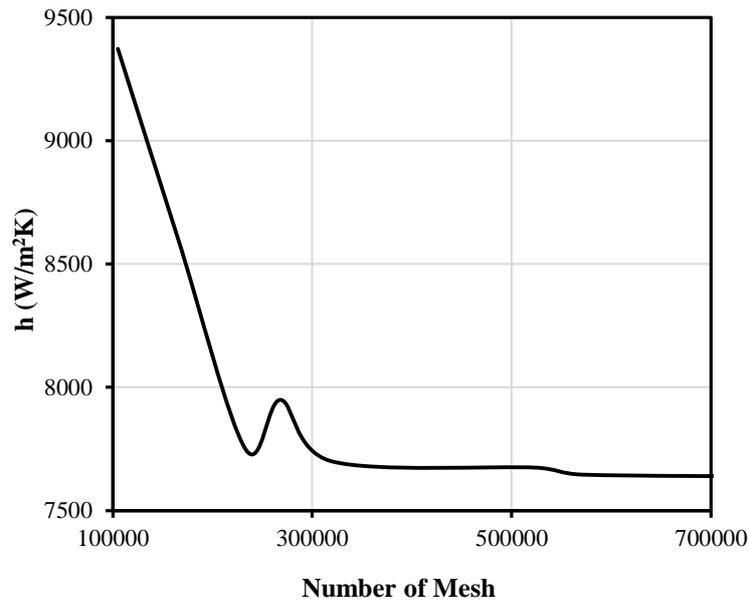


Fig. 3. The effect of the grids number on the convective heat transfer coefficient

Table 2. Coefficients and source terms of the flow governing equations

Equation	ϕ	Γ_{ϕ}	S_{ϕ}
Continuity	1	-	-
Momentum	V	μ_{eff}	$-\nabla p$
Turbulent kinetic energy	k	$\mu_{\text{eff}} / \sigma_k$	$P_k - \rho \varepsilon$
Turbulent kinetic energy dissipation rate	ε	$\mu_{\text{eff}} / \sigma_{\varepsilon}$	$\varepsilon(C_1 P_k - C_2 \varepsilon) / k$

Note: $\mu_{\text{eff}} = \mu + \mu_t$ $\mu_t = \rho C_{\mu} k^2 / \varepsilon$ $C_{\mu} = 0.0845, C_1 = 1.42, C_2 = 1.6$

To resolve the turbulence closure problem, the RNG k- ε model [12], accompanied by the standard wall function [13], is implemented due to its simplicity, robustness, and relatively accurate results. The governing equations are solved on a collocated grid using ANSYS FLUENT commercial code. Pressure-velocity coupling is established through the SIMPLE algorithm [14] and all terms are discretized using the first order schemes. The minimum residuals for stopping the solution procedure are equal to 10^{-3} for continuity equation and 10^{-6} for the others.

At the inlet, uniform constant velocity of 0.5, 0.75, and 1.00 m/s with 5% turbulence intensity is used. At the outlet, pressure outlet boundary condition is introduced. The water inlet and wall temperatures are 300 and 353, respectively. As the flow is turbulent, it is expected that the heat transfer coefficient does not depend on the wall thermal boundary condition.

3. Results and Discussion

In this section, the results of simulation and numerical analysis are presented. The model with the Chevrons depth of 5 mm and pitch of 1.25D is assumed as the base one. In Fig.4, the velocity contours at a longitudinal section are shown for the base model as the water inlet velocity is 0.75 m/s. In addition, Fig.5 depicts the temperature contour in the same section. As shown in these figures, the fluid velocity reduces to zero as it impacts the walls, while the velocity in the middle of the heat exchanger reaches its maximum value. In addition, the flow velocity decreases in the hemispheric Chevrons due to the creation of vortexes. The fluid velocity is high in the Chevrons entrance and exit due to decrease in the cross section area. Vortexes are generated within the hemispheric Chevrons which results lower velocity in these areas. In addition, the fluid flow is uniform at the inlet

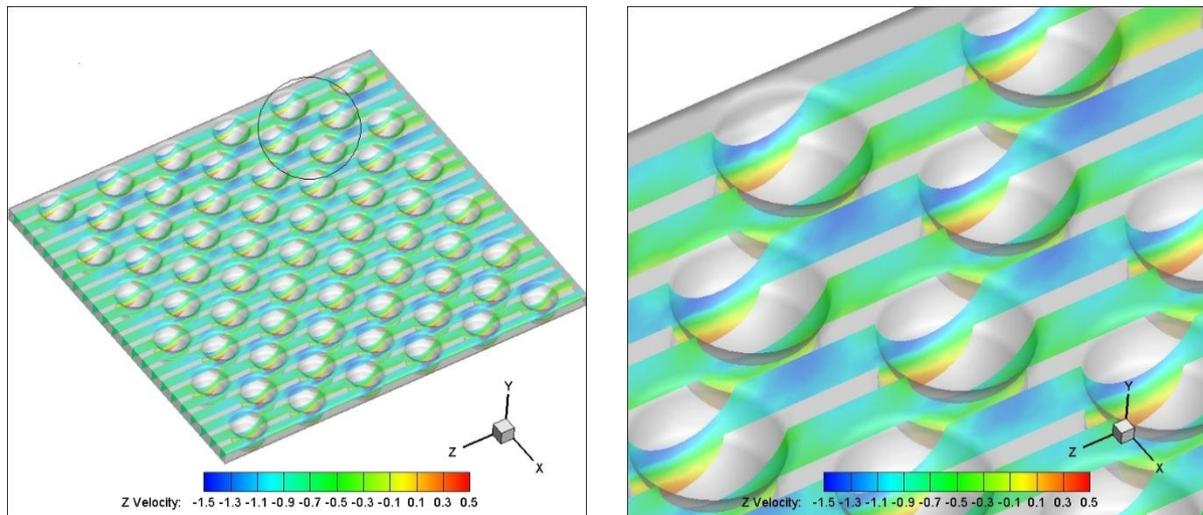


Fig. 4. Contour of velocity distribution at the inlet velocity of 0.75 m/s

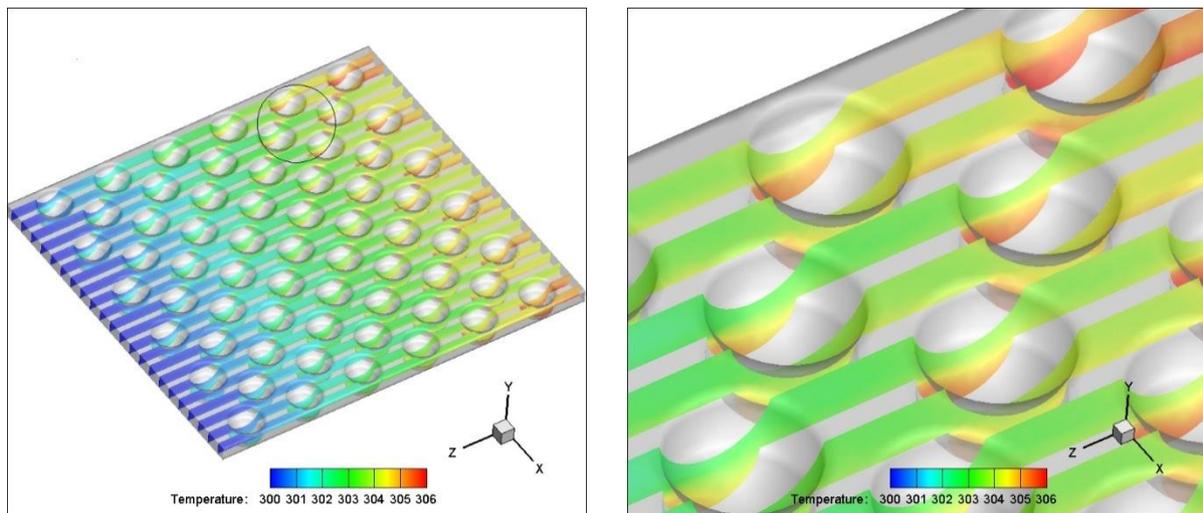


Fig. 5. Contour of temperature distribution at the inlet velocity of 0.75 m/s

where the turbulent intensity is minimum. On the other hand, in the middle parts of the heat exchanger- especially in the hemispheric Chevrons - the turbulent intensity reaches its maximum value. As the fluid flows through the heat exchanger, its temperature increases monotonically.

3.1. The Effect of the Chevrons Depth

As shown in Fig.6, the convective heat transfer coefficient decreases with increase of the inlet velocity or as the Chevrons depth increases from 2.5 to 5 mm. The convective heat transfer coefficient reduces due to

decrease in the number of Chevrons and enhancement of the heat transfer area with increasing the Chevrons depth. As the depth increases from 5 to 7.5 mm, the convective heat transfer coefficient is expected to decrease; however, it increases because the distance between the plates enhances, which itself increases the rate of fluid flow. In addition, increase of the fluid flow velocity enhances the mass flow rate and momentum, which raises the convective heat transfer coefficient in its turn. Obviously, the values of the convective heat transfer coefficient are very high compared to shell-and-tube heat exchanger.

Figure 7 shows the effect of the fluid flow inlet velocity and the Chevrons depth on the pressure drop. As depicted in this figure, with increase of the Chevrons depth at a constant inlet velocity, the pressure drop decreases asymptotically. This feature is mainly due to reduction in the number of Chevrons and increase in the distance of the plates. Moreover, the results show that the pressure drop increases with the fluid flow velocity at a constant depth, which is in accord with the previous discussions.

3.2. The Effect of the Chevrons Pitch

The effect of the Chevrons pitch on the convective heat transfer at a constant depth of 5 mm and the velocity of 0.5 m/s is shown in Fig.8. The convective heat transfer coefficient increases with an increase in the Chevrons pitch; however, it is anticipated that the convective heat transfer coefficient declines due to decrease in the number of Chevrons. Considering a small temperature difference observed in the outlet flow among different

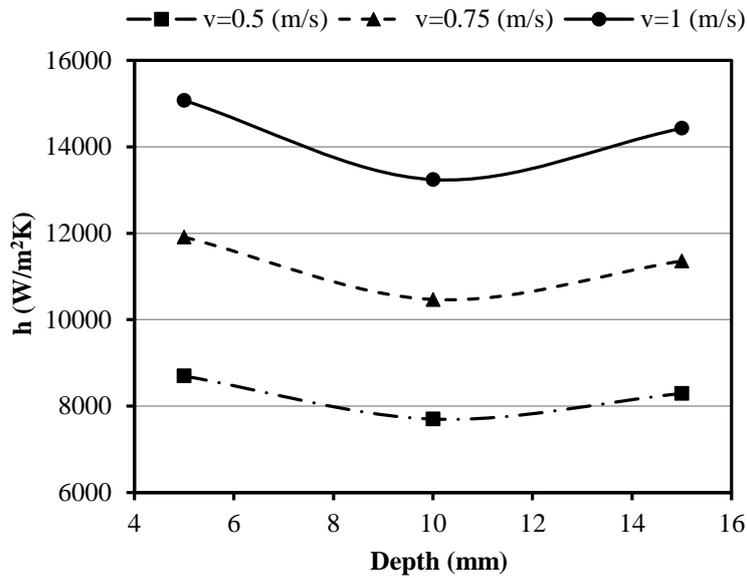


Fig. 6. The effect of Chevrons depth on the convective heat transfer coefficient

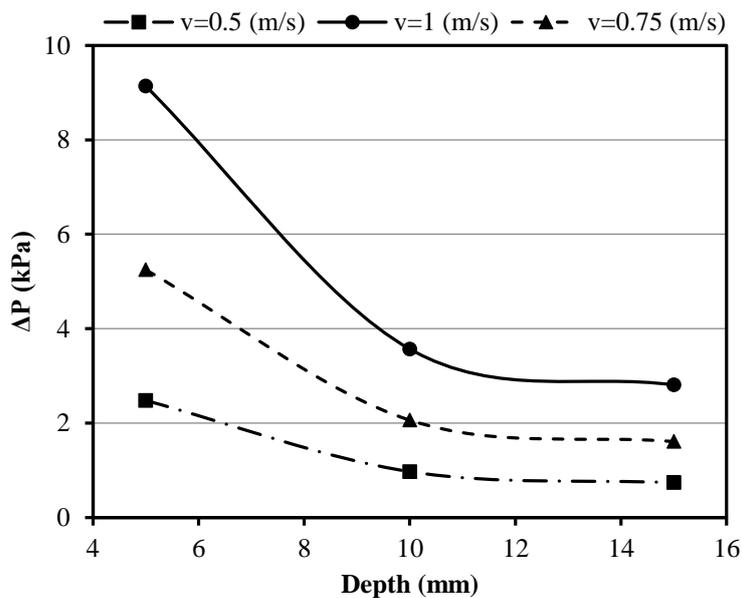


Fig. 7. The effect of Chevrons depth on the pressure drop

models with the increase of the Chevrons pitch and a decrease in the heat transfer area, a constant mass flow rate analysis may provide logical results.

In addition, the effect of the Chevrons pitch on the pressure drop at a constant depth and velocity is depicted in Fig.9. As it can be seen, by increasing the Chevrons pitch, which leads to decrement of the number of Chevrons, disruption of the flow is reduced and, as a result, the pressure drop decreases.

3.3. The Effect of the Chevrons Angle

The effect of the Chevrons angle on the convective heat transfer coefficient in different velocities is shown in Fig.10. Based on the results, the convective heat transfer coefficient increases with the Chevrons angle until it reaches the maximum value at 90° . Afterwards, the convective heat transfer coefficient decreases. In addition, the value of the convective heat transfer coefficient

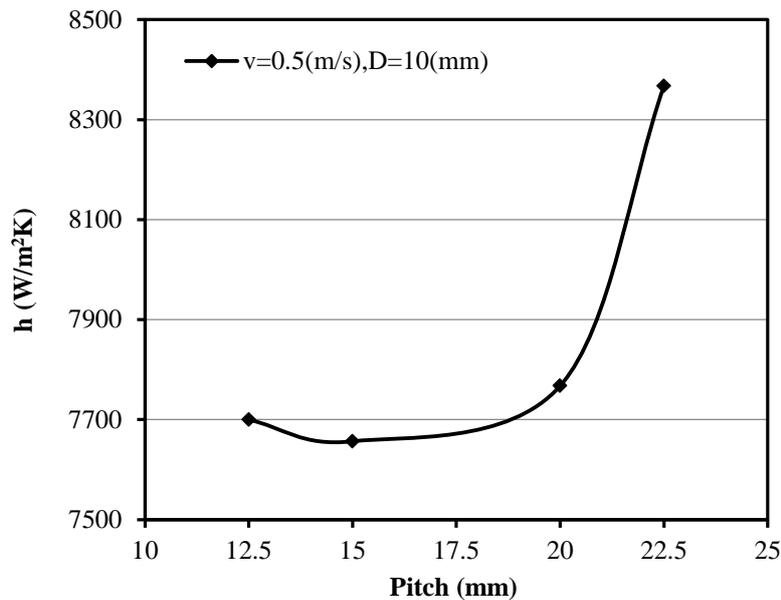


Fig. 8. The effect of Chevrons pitch on the convective heat transfer coefficient

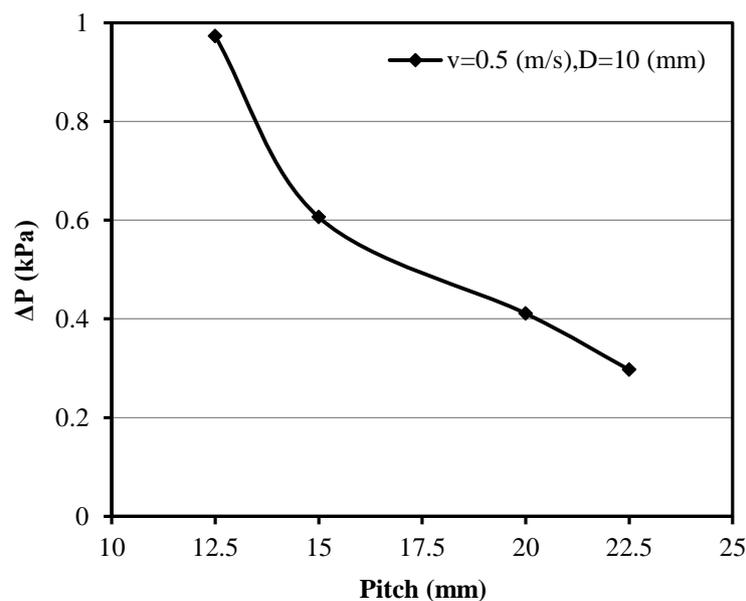


Fig. 9. The effect of Chevrons pitch on the convective heat transfer coefficient

decreases. In addition, the value of the convective heat transfer coefficient increases with the enhancement of the fluid flow inlet velocity, which is mainly due to increase of Reynolds number.

Figure 11 depicts the effect of the inlet velocity and the Chevrons angle at a constant depth and pitch on the pressure drop. As shown in this figure, the pressure drop increases with enhancement in the Chevrons angle and reaches its maximum value at the angle of 90°; after that, the pressure drop decreases.

3.4. Heat Transfer Coefficient Correlation

A correlation has been proposed by Kumar for corrugated Chevron plate heat exchangers as [15]

$$\frac{hD_h}{k} = C_h Re^n Pr^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0.17}, \quad (2)$$

in which values of C_h and n are constants [15] which depend on the flow characteristics and the Chevron angle. The Reynolds number, Re , based on the mass velocity and the hydraulic diameter, D_h , of the channels is defined as

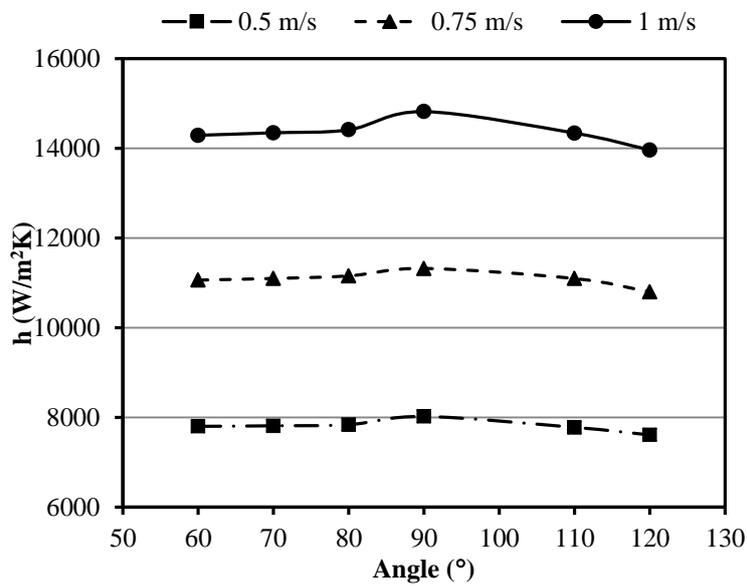


Fig. 10. The effect of Chevrons angle on the convective heat transfer coefficient

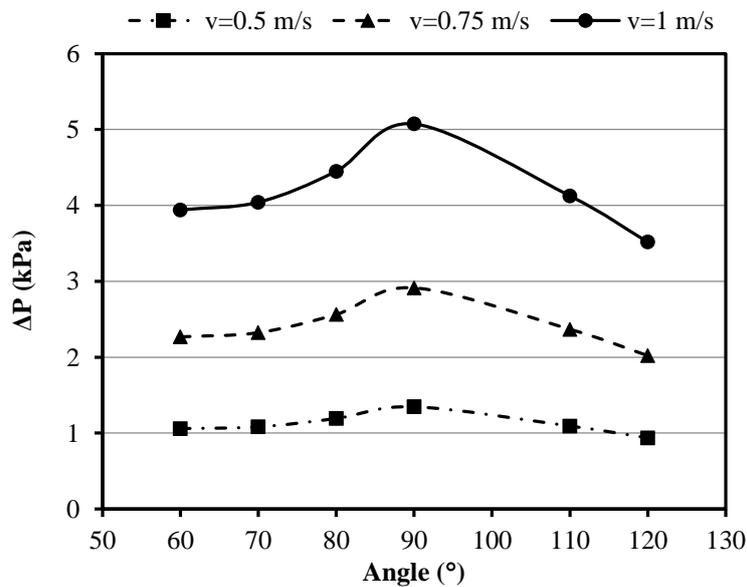


Fig. 11. The effect of Chevrons angle on the convective heat transfer coefficient

$$Re = \frac{G_c D_h}{\mu} \quad (3)$$

The mass velocity is given by

$$G_c = \frac{\dot{m}}{N_{cp} b L_w} \quad (4)$$

where N_{cp} is the number of channels per pass and is obtained from

$$N_{cp} = \frac{N_t - 1}{2N_p} \quad (5)$$

where N_t is the total number of plates and N_p is the number of passes.

In Fig.12, the convective heat transfer coefficient obtained from the numerical analysis is compared with the Kumar equation. Based on the results, there is a noticeable difference between the numerical

results and Kumar equation which is mainly because Kumar equation has been derived from the experimental studies on the plate heat exchangers with corrugated Chevrons. Thus, using this equation to determine the convective heat transfer coefficient for a plate with hemispheric Chevrons may cause significant errors in results.

Accordingly, a new correlation for the flow Nusselt number in hemispherical Chevron plate heat exchangers may be introduced as

$$Nu = C_h Re^n \quad (6)$$

where C_h and n are presented in Table 4 as $5000 \leq Re \leq 15000$.

The comparison between numerical results and the proposed correlation values is shown in Fig.13. As represented, the results are in good agreement with the proposed equation.

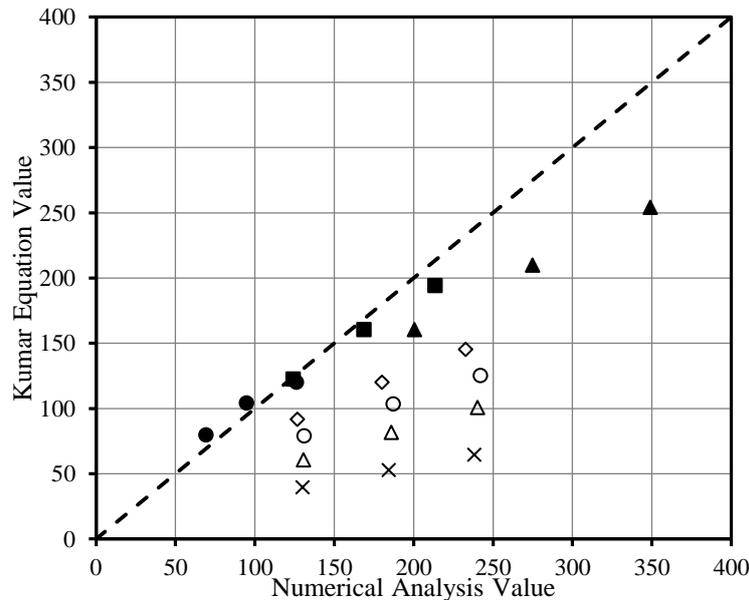


Fig. 12. The Comparison between numerical results and Kumar Equation

Table 4. The proposed formulas for C_h and n coefficients in Eq. (6)

Parameter	C_h	n
$5 \leq D \leq 15$	$-0.0014D^2 + 0.0351D - 0.0801$	$0.0019D^2 - 0.0445D + 1.0339$
$12.5 \leq P \leq 22.5$	$0.0047P^2 - 0.1597P + 1.4027$	$-0.005P^2 + 0.1734P - 0.6056$
$60^\circ \leq \Theta \leq 120^\circ$	$0.0083\Theta^4 - 0.0309\Theta^3 + 0.0347\Theta^2 - 0.0172\Theta + 0.0823$	$0.0349\Theta^4 - 0.2757\Theta^3 + 0.7364\Theta^2 - 0.7969\Theta + 1.175$

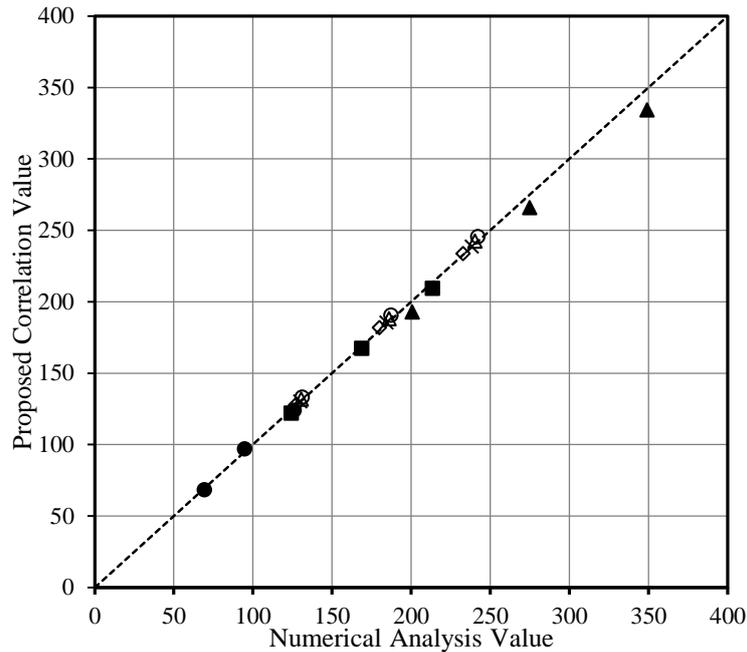


Fig. 13. The comparison between the numerical results and Eq. (6)

4. Conclusion

The results of this study may be summarized as below:

- The convective heat transfer coefficient and the pressure drop decreases with increase of the Chevrons depth and reduction of the number of Chevrons;
- The convective heat transfer coefficient and the pressure drop enhance as the Chevrons angle increases and reach the maximum at the angle of 90°; afterwards they decrease;
- The convective heat transfer coefficient and pressure drop increase with enhancement of the fluid flow inlet velocity or Reynolds number at a constant Chevrons geometric specifications;
- Based on the Kumar equation, a new correlation was proposed to predict the Nu number in the plate heat exchangers with hemispheric Chevrons.

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