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Numerical Investigation and Analysis on Thermal-Hydraulic Characteristics of an Elliptical Embossment Plate Heat Exchanger

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ABSTRACT

Dimpled and embossed surfaces are widely used to enhance the performance of plate heat exchangers (PHEs). In this work, the thermal–hydraulic behavior of a PHE with a newly proposed elliptical embossment pattern was examined through numerical simulation. The study focused on two key geometric parameters—embossment depth and pitch—and their influence on heat transfer and flow resistance. Three-dimensional simulations were carried out in ANSYS Fluent using the realizable k–ε turbulence model with enhanced wall functions, covering Reynolds numbers from approximately 100 to 2000. The results indicate that both the Nusselt number (Nu) and the thermal–hydraulic performance factor (F) rise with increasing embossment depth and pitch, while the friction factor (f) shows a decreasing trend under the same conditions. These findings suggest that careful tuning of the elliptical profile can achieve a favorable balance between heat transfer enhancement and pressure drop, offering practical guidance for the design of next-generation high-efficiency PHEs.

Keywords: Elliptical Embossment, Plate Heat Exchanger, Computational Fluid Dynamics (CFD), Heat Transfer Enhancement.

1. Introduction

Plate heat exchangers (PHEs) are widely used across multiple sectors, such as chemical manufacturing, energy production, heating, ventilation, and air conditioning (HVAC). Their characteristics include increased thermal efficiency, less space requirements, and straightforward maintenance[1], [2], [3], [4]. The efficacy of a PHE is mainly influenced by the configuration of its corrugated plates. Although the chevron pattern has been the preferred choice for an extended period [5], [6], [7], [8] recent research has explored various alternative designs, including dimpled or embossed, in an effort to enhance performance. Ducts of circular-dimple arrays deployed at uniform spacings demonstrate high heat transfer efficiency and lower losses of pressure than conventional designs[9], [10]. Multiple numerical analyses, sometimes backed by empirical validation, have indicated that the k-ε class of turbulence models is capable of capturing the details of the fluid dynamics, such as vortex generation and secondary flows, that govern such heat transfer by such channels[11], [12], [13]. The majority of the studies have been conducted on circular dimples. Non-circular embossments like ellipses, unlock geometric options that could potentially manage secondary distributions of flows such that there is an optimal balance between enhanced heat transfer and friction losses. Fewer studies have been done on primary parameters of elliptical embossments, such as depth and pitch, and their resultant influences on PHE overall performance. This paper aims at bridging that gap by comprehensively exploring the effects of varying embossment depth and pitch on heat transfer as well as pressure loss. In order to explain the recorded performance tendencies, the internal flow processes are examined thoroughly. The results can be used for the optimization of PHE's design to use in industrial practice.

2. Methodology

2.1 Problem Description

This analysis considers a plate heat exchangers channel that is arranged with elliptical forms. Fig. 1 is an image of the shape.

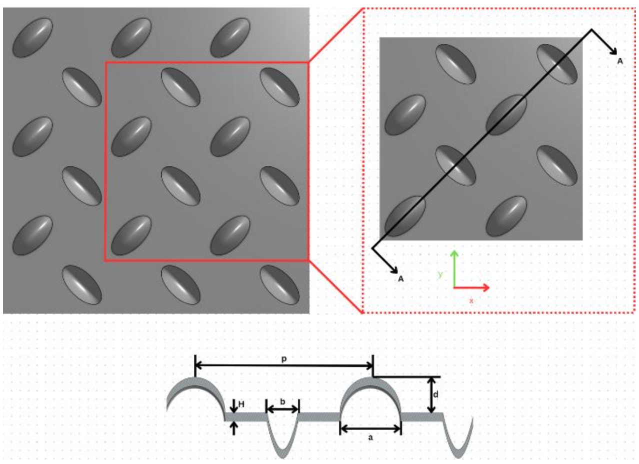

Fig. 1 Schematic Diagram of the elliptical embossment.

Table 1 outlines key measurements for the system, such as thickness, embossment depth, and pitch. These are selected from actual use considerations and have a range that is common for use in industry. This way permits an intensive examination of how they will affect performance.

Table 1 Structural parameters of CFD simulations.

Parameter	Value
Thickness, H [mm]	0.25
Depth, d [mm]	0.5 - 1.3 with intervals of 0.2 @fixed pitch of 5.2 mm
Pitch, p [mm]	2.8 - 6 with intervals of 0.8 @fixed pitch of 1.1 mm
Major axis, a [mm]	1.83
Minor axis, b [mm]	0.92
Chevron angle, β [°]	45

2.2 Governing Equations

The geometric model is made simpler by retaining only the major flow and heat transfer characteristics. Solid parts are also approximated as homogeneous, while the roughness of surfaces is also overlooked. Water is used as the working fluid. Numerical analysis assumes a steady-state, stable, single-phase incompressible turbulent regime. The principles of calculation of incompressible fluids are given by the laws of conservation of mass, momentum, and energy. They are expressed by Eqs. (1) - (3):

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \bar{v}) = S_m \tag{1}$$

$$\frac{\partial}{\partial t}(\rho \bar{v}) + \nabla \cdot (\rho \bar{v} \bar{v}) = -\nabla p + \nabla \cdot (\tau) + \rho \bar{g} + \bar{F} \tag{2}$$

$$\frac{\partial}{\partial t}(\rho E) + \nabla \cdot \left(\bar{v}(\rho E + p)\right)$$

$$= \nabla \cdot (k_{eff} \nabla T - \textstyle\sum_j h_j J_j + (\vec{\tau}_{eff} \cdot \vec{v})) + S_h \tag{3}$$

2.3 Numerical Method

A procedure for plates with patterns of dimples was first developed. Next, numerical calculation targets were selected. Following that, mesh generation and independence verification were conducted. The study examined the differences in thermal performance by examining the paths of the flows. This served to demonstrate the potential causes. The test employed an assessment index of quantitative analysis of the performance of varying plate structures. The expression of the index is given by Eq. (4).

$$F = Nu / f^{\frac{1}{3}} \tag{4}$$

Where, Nu is the Nusselt number represented by the relation, $N_u = h_w D / \lambda_f$. Here, h_w is the water heat transfer coefficient over the dimple plate. It is obtained by substituting $q/(Tw - T_f)$. In the equation, q represents the heat flow at the solid-liquid heat transfer interface. Tw and T_f represent the fluid's plate wall temperature and the fluid's temperature, respectively. λ_f is the fluid's thermal conductivity, while D is the hydraulic diameter of the key region[12].

The friction factor was obtained from Eq.(5). In this equation, ΔP represents the pressure drop across the channel; ρ is the fluid density. The channel length of the characteristic zone is given by L; v_m is the mean fluid viscosity of the zone.

$$f = \frac{\Delta P D}{2 \rho L v_m^2} \tag{5}$$

Fig. 2(a) shows the plates used for the calculations. The model had a three-channel heat transfer setup with four stacked plates. A study by Zhang et al. [14] found that fluid flow in the plate stabilizes after going through 3 to 5 cells. So, a 3 × 6 cell layout was chosen for this study. Fig. 2(b) shows the fluid domains for the calculations. The central fluid domain was the main focus in this study. Hence, the top and bottom plates were left out in the simulation. The fluid inlets, named inlets 1, 2 and 3, were set in opposite directions to match the symmetrical counterflow conditions. A 30 mm stabilization zone was made at both the inlet and outlet to remove the entrance effect and make sure the flow is fully developed in the characteristics zone.

For numerical simulation, a second-order upwind technique was applied, which was developed from the finite volume approach. It was considered to be stable if the outlet temperature, the pressure drop across cells, and the heat flux across the surface experienced minute changes of less than 10^{-6}. Močnik et al. [11] presented a report stating that the realizable k-ε model of turbulence with enhanced wall functions performed superior for dimple structures over other models. Thus, a realizable k-ε model with enhanced wall functions was used to capture similar behavior.

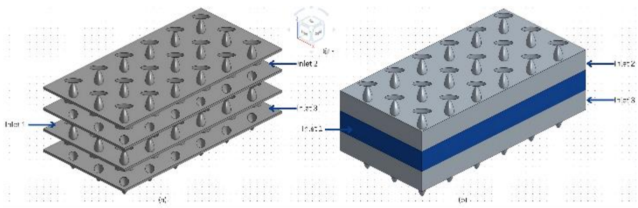

Fig. 2 Schematic diagram of the elliptical plate assembly used for numerical simulation.

The mass fluxes at inlets 2 and 3 were maintained at a constant value of 58.3 kg/m²s while keeping the temperature at 50 °C. The mass flux at inlet 1 was kept constant at 116.5 kg/m^2s while keeping the temperature at 65 °C. The fluid velocity at the characteristic region varied from around 0.0 to 0.12 m/s. The complete boundary conditions are given in Table 2.

These boundary conditions from Table 2 were selected depending on the needs for operations as well as study objectives. Inlet mass flux value of 116.5 kg/m²s was picked to cover the usual working conditions of plate heat exchangers for industry and commercial applications. The inlet temperature was selected based on realistic conditions of hot water for heat transfer applications, assuming that it is connected to practical systems. This temperature was kept constant to ensure consideration of the influence of the mass flux on heat transfer as well as flow characteristics. The thermal conductivity of the plate material was given as 16.8 W/m·K, which is stainless steel material that is commonly employed in plate heat exchangers due to its high mechanical as well as thermal properties.

Table 2 Boundary conditions.

Parameter	Value
Working Fluid	Water
Inlet mass flux, [$kgm^{-2}s^{-1}$]	116.5
Temperature, [°C]	65
Plate thermal conductivity [$Wm^{-1}K^{-1}$]	16.8

2.4 Grid Independence and CFD Model Validation

A polyhedral mesh was used to create the grid because it is better than traditional tetrahedral meshes. Its benefits include, needing less memory, quicker computation times, and having fewer elements for similar accuracy. The results of the grid independence study are shown in Table 3. When the mesh size went up from 0.82 million to 0.97 million cells, the differences in the Nusselt number (Nu) and friction factor (f) dropped a lot, from 2.84% and 1.48% to 0.28% and 0.29%, respectively. Making the mesh finer to 1.02 million cells led to very small changes in Nu (0.09%) and f (0.12%). Increasing the cell count to 1.15 million showed little change, showing that it was stable. Based on these results, the mesh with 1.02 million cells is enough for getting grid-independent results, and it will be used for future simulations with other shapes.

Table 3 Grid independence study at G=58.3 kg/m^{2}s for *p*=5.2 mm

No of cells [10^5]	Difference in ΔP (%)	Difference in Nu (%)
8.3		
9.7	2.84	1.48
10.2	0.28	0.29
11.5	0.09	0.12

3. Result and Discussion

3.1 The Effect of Elliptical-Plate Parameters on Heat Transfer

Fig. 3 indicates the variation of the Nusselt number (Nu) with varying embossing depth and pitch at various mass flux levels. The data evidently reveal that, enhancing the embossing depth from 0.5 mm to 1.3 mm at a fixed mass flux of 116.5 kg/m²s results in an enhanced value of Nu. This trend again depicts the way deeper elliptical grooves enhance local recirculation of the flows and convective heat transfer. It is also observed that, an increase in the pitch from 2.8 mm to 6.0 mm tends to give an enhancement of the Nusselt number. This implies that an increased embedding pitch of the test surfaces does not necessarily degrade thermal performance but rather sometimes facilitates beneficial recirculation zones that enhance the removal of heat.

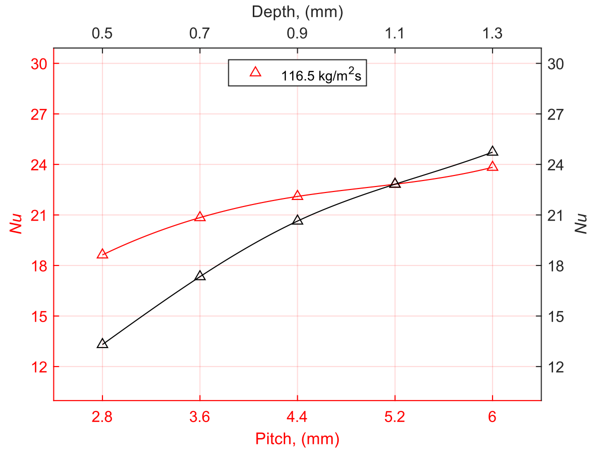

Fig. 3 Variation in *Nu* with depth and pitch

3.2 The Effect of Elliptical-Plate Parameters on Pressure Loss

Fig. 4 indicates that the friction factor (f) decreases with the embossing depth and pitch. For the measured mass flux value, the friction factor substantially decreases with an increased embossment pitch from 2.8 mm up to 6.0 mm. That implies that an increased gap between the embossments creates lower flow interference and lower hydraulic resistance, which contributes towards lower pressure drop. Notably, the friction factor decreases as the embossment depth rises from 0.5 mm to 1.3 mm. This result is unexpected but significant. Intuitively, deeper bumps would tend to rise form drag and friction losses. However, the uncommon geometry of the elliptical embossment appears to direct the flow better at increased depths. This could decrease flow separation or generate a more stable vortex structure that decreases overall pressure loss.

This two-part advantage that raises both depth and pitch tend to reduce friction, indicating that the elliptical embossment profile is extremely effective for hydraulics. It provides a great deal of design freedom for optimizing the transfer of heat while never causing a large decrease in pressure.

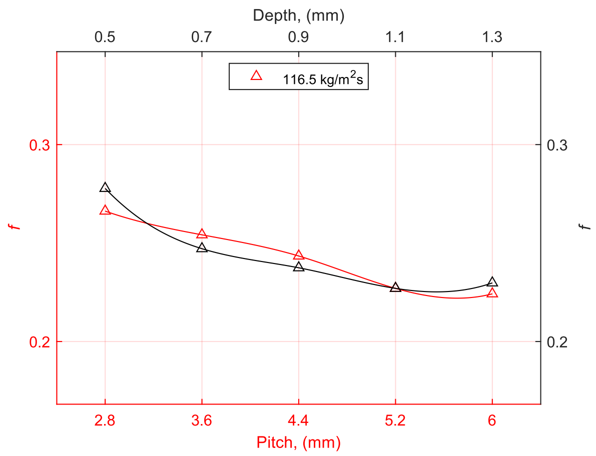

Fig.4 Variation in *f* with depth and pitch

3.3 The Effect of Elliptical-Plate Parameters on Overall Performance

Fig. 5 demonstrates the variation of the performance factor (F) with embossing depth and pitch for elliptical features.

When the average pitch is increased from 2.8 mm to 6.0 mm, this results in a moderate rise of F, by about 35% across the measured mass flux. And when, the embossing depth is increased from 0.5 mm to 1.3 mm, this results in a massive rise of F, by about 98% across the measured mass flux. These observations imply that deeper elliptical embossing features significantly enhance thermal-hydraulic performance. As noted earlier, the Nusselt number (Nu) and performance factor (F) increased with the increase of both depth and pitch under the studied structural parameters. Industrially, there needs to be a compromise between the maximization of heat transfer efficiency to enhance energy performance on one hand and the minimization of pressure drop to reduce costs on the other hand. Hence, the optimal plate heat exchanger design should ideally be one that strikes a balance between enhancing heat transfer (by increasing the depth) against balancing the pressure loss that accompanies its increment. This work reveals that for typical commercial plate heat exchangers, the dimple depth is more significantly influential on design than the pitch when overall performance is considered, particularly for energy use and costs of operation.

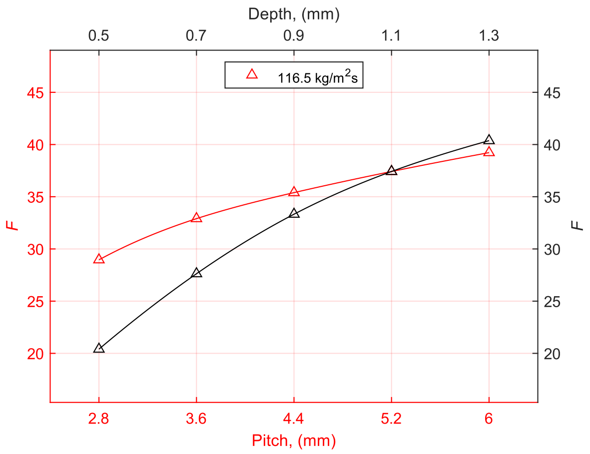

Fig.5 Variation in *F* with depth and pitch

4. Conclusion and Prospects

In this study, numerical computations were carried out to analyze the thermal-hydraulic performance of a plate heat exchanger with elliptical embossment pattern. The numerical model was validated and the effect of embossment depth and pitch was analyzed. The following are the main conclusions:

- The heat transfer (*Nu*) is higher with greater embossment depth. It also rises considerably with a rise in pitch. It expresses that, deeper and further spaced elliptical embossments provide better thermal performance.
- The friction factor (*f*) reduces with a rise in embossment depth and pitch. This combined effect, lowers hydraulic resistance, which is significant in design optimization.
- The performance factor (*F*) rises considerably with pitch and depth. The rise in pitch is especially notable since it is caused by the combined effect of enhanced heat transfer and lower pressure drop.
- Compared to traditional embossments, elliptical embossments provide a significant advantage, mostly in configurations with increased pitch. This places them as an option to consider in optimizing heat exchanger performance in practical applications.

Further work could be done on parametric optimization, using genetic algorithms, to find the optimal depth, pitch, ellipse aspect ratio, and chevron angle. These numerical findings will be key to next-generation embossed plate heat exchanger designs.

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6. Nomenclature

Symbol	Meaning	Unit
a	Major axis	(m)
b	Minor axis	(m)
D	Hydraulic diameter	(m)
d	Corrugation depth	(m)
f	Friction factor	Dimensionless
F	Performance factor	Dimensionless
G	Mass flux	($kgm^{-2}s^{-1}$)
h	Heat transfer coefficient	($Wm^{-2}K^{-1}$)
H	Thickness	(m)
L	Length	(m)
Nu	Nusselt number	Dimensionless
p	Pitch	(m)
q	Heat flux	(Wm^{-2})
Re	Reynolds number	Dimensionless
T	Temperature	(K)
V	Velocity	(ms^{-1})
β	Chevron angle	[°]
ΔP	Pressure drop	(Pa)
ρ	Density	(kgm^{-3})