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Enhancement of Heat Transfer in Plate Heat Exchangers Using Corrugated Geometries: An ANSYS-Based Study

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***Abstract*— Plate heat exchangers (PHEs) are widely used in industrial applications due to their compact size and high thermal efficiency. Enhancing heat transfer within these exchangers is critical for improving overall performance. This study investigates the impact of corrugated geometries—specifically triangular, sinusoidal, and rectangular patterns—on heat transfer enhancement in PHEs using computational simulations in ANSYS Fluent. The effects of corrugation shape on flow behavior, pressure drop, Nusselt number, and overall heat transfer coefficient were analyzed across a range of Reynolds numbers. Results indicate that corrugated channels significantly increase turbulence intensity, disrupt boundary layers, and improve fluid mixing, leading to higher heat transfer rates compared to smooth channels. Among the geometries studied, rectangular corrugations exhibited the highest Nusselt number but also the largest pressure drop, highlighting a trade-off between thermal performance and hydraulic resistance. This study provides valuable insights for optimizing PHE designs by selecting appropriate corrugation geometries to balance heat transfer efficiency and pressure drop constraints.**

Keywords*— *Heat Transfer Enhancement, Plate Heat Exchanger (PHE), Triangular, Sinusoidal, Rectangular Channels.

I. INTRODUCTION

It has been recognized that plate heat exchangers possess many unique characteristics, which are capable of recovering heat efficiently at low temperature differentials (as low as 1^0C) because of the high turbulence even at low velocities. Enhanced surfaces yield higher heat transfer coefficient when compared to unenhanced surfaces. A surface can basically be enhanced in two ways, either active enhancement which requires deployment of external power which is obviously high in operational and capital cost thus commercially unviable, and passive enhancement which involves adding extended surfaces (e.g. fins), or employing interrupted surfaces (e.g. corrugations). The plate surface geometry is characterized by corrugation profile, corrugation inclination angle corrugation pitch (P) and corrugation height (h).The enhancement of heat transfer between corrugated plates is directly related to these features, which provide increased effective heat transfer area, disruption and reattachment of boundary layers, swirl or vortex flow generation, and small hydraulic diameter flow channels.

The plate heat exchangers are most economical and efficient type of heat exchanger in the market. Due to its low cost, easy maintenance, flexibility, and high thermal efficiency.

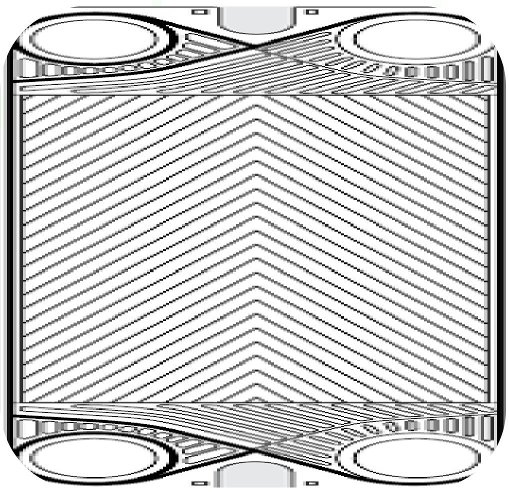

Fig. 1: Heat Exchanger Plates

Those mechanisms are associated with friction expenditure which increases the pressure loss, flow separation, and reattachment processes. As the channel

with complicated passage is formed as a result of the arrangement of the two plates, the breakup and reattachment of boundary layers together with vortices flows occurring in the passage are the most affective contributes to the high heat transfer efficiency. The corrugation patterns induces turbulent flows at very low velocity of flow, it not only achieves unmatched efficiency, but also creates a self-cleaning effect that reducing fouling on surface of a heat exchanger plate [1]. The most common surface pattern used is the corrugated design.Plate heat exchangers are usually designed to achieve turbulence across the total flow passage and increase heat transfer area in order to get the maximum possible heat transfer coefficient with the minimum pressure drop, and allow for reach close temperature difference. It means smaller heat transfer area, smaller size of heat exchangers, and sometimes less number of heat exchangers [1]:
The corrugated plate heat exchangers have gasketed plates which are fixed between a upper carrying rod and a bottom guide rod. The plates are tightened and compressed to closed packing by means of tie bolts between a fixed frame part and a moveable frame part [1].

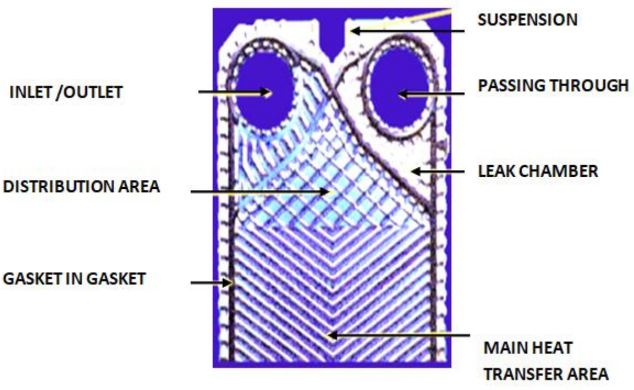

Fig 2: Main Component of Heat Exchanger Plate

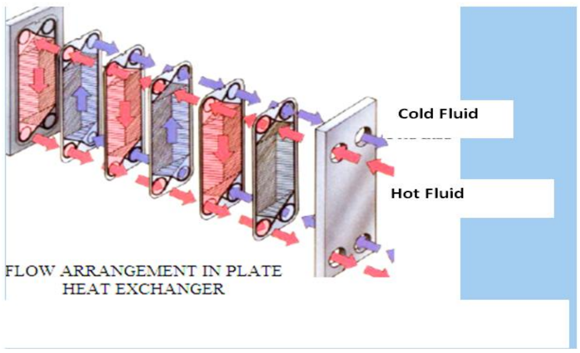

Fig 3: Flow Arrangement in Plate Heat Exchanger

The plate heat exchanger is designed for single-pass or multi-pass flow, according to requirement. For most applications, the single-pass heat exchanger is suitable and preferred because it makes all connections of corrugated plates on the stationary frame part and it makes the overall disassembly easier. However, for applications with low flow rates or close approach temperatures, the multi-pass solution is required.

A. Turbulent Flow

- **Irregularity**. Turbulent flow is irregular, random and chaotic. The flow consists of different eddy sizes where largest eddies are due to flow geometry, and the smallest eddies which are due to viscous forces dissipated in the form of internal energy.
- **Diffusivity**. In turbulent flow the due to random motion of particle diffusivity increases. The turbulence increases due to the exchange of momentum in the boundary layers and decreases or delays by separation at bluff-bodies such as sphere airfoils geometry. The increased diffusivity also increases the surface friction in internal flows such as in pipes.
- **Dissipation**. Turbulent flow is dissipative in nature, which means that kinetic energy in the smaller size eddies are transformed into internal energy. Smaller size eddies form due to turbulent floe get the kinetic energy from slightly larger sizes of eddies, and the larger size eddies get their kinetic energy from even largest eddies. The largest eddies receive their energy from the mean flow.

II. Literature Review

the study carried out by Jain, S. Joshi, A. Bansal, and P. K. [2] titled ‘ Approach to Numerical Simulation of Small Sized Corrugated Plate Heat Exchangers ’ have experimental study of total heat transfer and fluid flow in a single pass counter flow plate heat exchanger with corrugated plates presented. Computational fluid dynamic analysis of small sized corrugated plate heat exchanger was conducted by taking the full geometry of the total heat transfer surface and realistic thermal and hydrodynamic boundary conditions. A cold water flow channel with two corrugated plates and two parts of hot channel on both side having flat boundaries was selected as the computational fluid domain. For Reynolds number range 400-1300 and Prandtl number range 4.4-6.3 heat transfer data and pressure drop data were collected experimentally for water. The work considered a cold channel (with two corrugated plates on either side) and two half hot channels on either side as the computational domain. The virtual flat boundaries of the outer half hot channel were treated as periodic surfaces to represent the complete heat exchanger. In applying the periodic boundary conditions, the pack was assumed to contain infinite plates (neglecting the end effects), although the end effects may be significant if the number of platesis less than 50.

Shah, R. K. and Focke, W. W. [3]. has found for the selected geometry, 3-D, steady state numerical simulations were carried out using ANSYS WORKBENCH as the preprocessor for geometry creation and mesh generation and FLUENT 14 as the solver and postprocessor. The inputs to the simulation included the inlet velocities and temperatures of hot and cold fluids. The simulations predicted the heat transfer coefficient and pressure drop along with the velocity, pressure, and temperature distribution in the channels. The realizable k −εturbulence model with non-equilibrium wall function was use.

Lee Y. S. Sun Y.M [4] has investigated the heat transfer characteristics of the corrugated channels in plate heat exchangers. A flow channel with wavy surfaces was constructed within which air was passed thru. The temperature of the wall was maintained constant. Starting from the entrance, the air temperatures within two wavelengths were measured at about 300 locations. The Reynolds number of air was varied from 300-9000. Results show that both the average and local Nusselt number decreases with the distance from the entrance and is greater at sections near the crest.

Davidov, B. I. [5] has numerically investigated the two-dimensional time dependent fluid flow and heat transfer in two geometrical configurations for the unsteady regime with Reynolds number (175 to 200) for sinusoidal channel, and Reynolds number varying from 60 to 80 for the arc-shape channel. The results showed that as long as the heat transfer enhancement was high, the pressure loss was high too.

Harlow, F. H[7] studied numerically on the fully unsteady fluid flow and heat transfer in sine shaped wavy channels. Jones, W. P [8]considered the influence of the sinusoidal wavy-surface plate- fin geometry on hydrodynamic and thermal behaviors of laminar air flow under a constant wall temperature and through the range of Reynolds number from 10 to 1000. The predicted results were identical to the results of the above mentioned researches. The vortices which were responsible for enhancing of heat transfer increased as the Reynolds number increased and with sharp and sudden sinusoidal corrugation. These results were approved under the same conditions by Launder, B. E. and Sharma, B. I[9] but with viscous liquids (Prandtlnumber5, 35, and150).

Mansour, N. N. and Kim, J. Moin, P. [10] has investigated the same model but changing flow type to force flow, and using both the constant wall temperature and constant wall heat flux as boundary conditions. However, the predicted flow phenomena were comparable, but the thermal performance with the constant heat flux was higher than that with constant temperature boundary condition. Under the same condition as above i.e. constant wall temperature and constant wall heat flux Morris,

It is observed by F. H. Whitman, [11] undertook a fully developed laminar water flow for sinusoidal channels of circular and semi-circular cross-section, and investigated the influence of Reynolds number $(5<Re<200)$ and amplitude to half wavelength ratio $(0.222<A/L<0.667)$ on heat transfer enhancement and pressure drop. With varying the Reynolds number and amplitude to half wavelength ratio, the enhancement efficiency obtained with as large as 1.8, and 1.5 for the circular and semi-circular section respectively. The author found that the numerical results obtained show that the heat transfer performance can suitably be described by the field synergy principle.

The investigation of developing laminar forced convection and entropy generation in both double- and half-sine ducts subject to the variation of Reynolds number (86 to 2000) was numerically carried out by Incropera, F. P. DeWitt, D. P. [13]. investigated that the Nusselt number in double-sine ducts was higher than that in half-sine, whereas the entropy generation in half-sine duct was much higher in double-sine ducts.

Pham, M. V. [14] used the Large-eddy simulation together with dynamic modeling toexamine 3D turbulent flow in wavy channels exposed to the variation of Reynolds number (750 to 4500), spacing ratio, and waviness aspect ratio. It was found that the spacing ratio played a key role to produce vortices, control turbulent kinetic energy. And also, the friction factor was highly sensitive to the variation of spacing ratio.

Ismail, S. L[15] has studied the design data and flow patterns for three types of offset fins and sixteen types of wavy fins for a compact plate heat exchanger. This study addressed the quantification of geometry generating mal-distribution. The type of flow inside the corrugated channel in the compact heat exchanger as function of Reynolds number is a significant issue.

Heggs, J and Walton, C. [17] has utilized the electrochemical mass transfer technique to calculate values of the local transfer coefficient for a corrugated plate heat exchanger channel, confirmed that the pure laminar flow did not exist for the tested Reynolds number range from 150 to 11,500. When the compact heat exchangers were used as reflux condenser, Drosos, E. I. P and Karabelas, A. [18] has suggested that the limit imposed by the onset of flooding, reduces the Reynolds number for less than 2000.

Vlasogiannis, P [19] has experimentally tested a plate heat exchanger under two-phase flow conditions and visualized the flow regime, validated that the flow was turbulent for Re 650.

It is observed by Lioumbas, I.S [20] that the flow in narrow passages during counter-current gas–liquid flow, recommended that for gas Reynolds numbers in the range of 500–1300, the flow exhibited the basic features of turbulent flow.

III. Methodology

The two plates are superimposed in the such a way that the opposite corrugations formed a cross-type pattern where the peaks and valleys in the top and bottom-plates lie on the same longitudinal plane the geometry is consisted of fourteen equal sized and uniformly spaced corrugations and two side-channels as illustrated and detailed in Fig. 3.1, and Table 3.1. However, this thesis is carried out for a single phase flow of water.

Case 1- A triangular vertical channel of a corrugated plate heat exchanger, is formed by two plates, 110 mm width and 200 mm length. On each plate, the inclination angles are suggested to be a 45°. The two plates are superimposed in the such a way that the opposite corrugations formed a cross-type pattern where the peaks and valleys in the top and bottom-plates lie on the same longitudinal plane the geometry is consisted of fourteen equal sized and uniformly spaced corrugations.

Case 2- A Sinusoidal vertical channel of a corrugated plate heat exchanger is also formed by two plates, 110 mm width and 200 mm length. On each plate, the inclination angles are suggested to be a 45°. The two plates are superimposed in the such a way that the opposite corrugations formed a cross-type pattern where the peaks and valleys in the top and bottom-plates lie on the same longitudinal plane the geometry is consisted of fourteen equal sized and uniformly spaced corrugations, but the configuration of the valleys and peak is sinusoidal as shown in Fig. 5

Case 3- A Rectangular vertical channel of a corrugated plate heat is formed by two plates, 110 mm width and 200 mm length. On each plate, the inclination angles are suggested to be a 45°. The two plates are superimposed in the such a way that the opposite corrugations formed a cross-type pattern where the peaks and valleys in the top and bottom-plates lie on the same longitudinal plane the geometry is consisted of fourteen equal sized and uniformly spaced corrugations, but the configuration of the valleys and peak is Rectangular as shown in Fig.6

Fig 4 Triangular corrugated channel

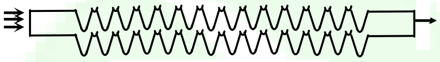

Fig 5: Sinusoidal corrugated channel

Fig 4: Rectangular corrugated channel

IV. RESULTS AND DISCUSSION

Heat transfer characteristics for constant wall temperature, were analysed locally and globally with CFD code based modelling, giving important information about the effect of geometries of corrugated channels on hydrodynamic and thermal behaviours of the flow crossing the channels. Firstly, the velocity magnitude, static pressure, turbulence intensity, and temperature distribution within Reynolds number varying from 400 to 1400, were all presented and analyzed for the triangular corrugated channels. Moreover, gradient of pressure, shear stress, friction factor, total heat flux, Nusselt number, and the surface heat transfer coefficient for the lower plate of triangular corrugated channel were graphed and locally analysed.

A. *Hydrodynamic Prediction*

Fig.7 and Fig.8 show the flow motion represented in velocity magnitude for both Reynolds numbers 400, and 1400. It can be seen that the main flow deflects into the corrugated confined area where the vortices are created. The vortex grows larger and its center moves to the downstream as Reynolds number increases, and occupies the confined area. The vortices in the confined corrugated area repeat the propagation that they travel to the downstream simultaneously in the lower and upper plates.

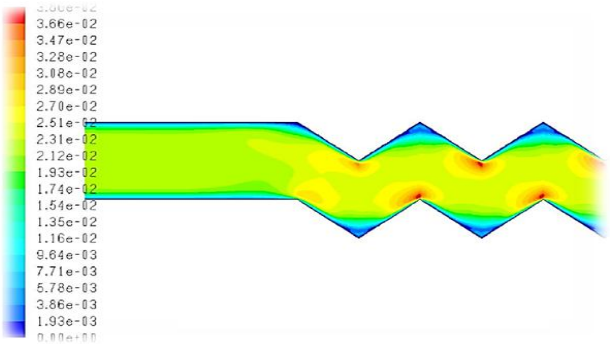

Fig7 Velocity magnitude at Re400

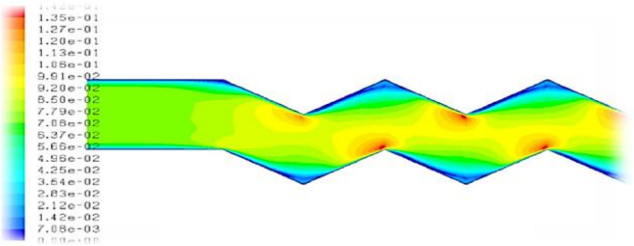

Fig 5: Velocity magnitude at Re1400

The phenomena of the creation of vortex in the channel can be qualitatively interpreted to the pressure gradients within the periodic corrugation. Fig. 5 represents the pressure gradients through two periodic corrugations where the first peak is located at 0.4 cm from the total length of the channel and second and third peaks are settled at 0.64, a0.78 cm respectively. Similarly, the relevant valleys are positioned at 0.47, and 0.71 cm. As the flow progress along the corrugation, the pressure decreases through the entrance flat channel, and then increases gradually when the flow passes from the peak to the valley. And, correspondingly, the pressure decreases again when the flow elapses from valley to peak. This behavior of pressure repeats itself for the second corrugation. This fluctuation of the pressure is a function of wall shear stress variation as it can be seen later and to the variation of cross sectional area. Seeing that the increase of pressure and reduction in velocity are related through the Bernoulli equation, the increasing of pressure causes a reduction in velocity, and when the momentum of the fluid layers near the surface is not sufficiently high to overcome the increase in pressure, the velocity gradient at the surface comes to zero, where the boundary layer of the flow reaches the separation point

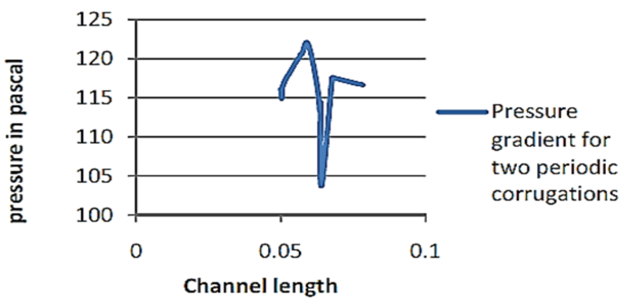

Fig 6: Pressure Gradient For Two Periodic Corrugations

B. Thermal Prediction

Fig. 7 and Fig.8 show the static temperature along the triangular corrugated channel for a constant wall temperature boundary condition at Reynolds numbers 400, and 1400. The figures illustrate that the water cools gradually as it crosses the channel. Also, it is clear that the temperature difference is quite high in the corrugated area compared to the flat entrance and exit.

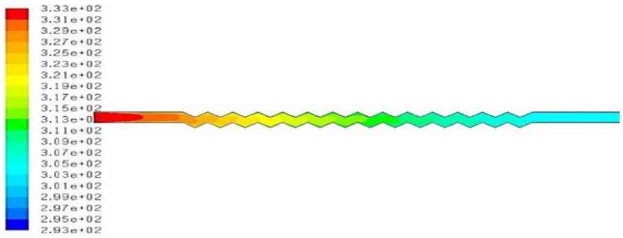

Fig 7: Static Temperatures at Re400

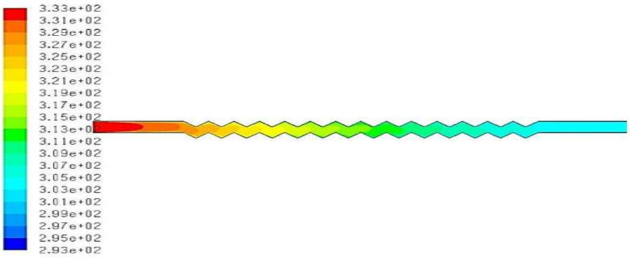

Fig 8: Static Temperatures at Re400

Fig. 9 exhibits the local heat flux in W/m^2 for the triangular corrugated channel for the lower plate at Reynolds number changing from 400 t0 1400. The graph shows that with progression of Reynolds number, the rate of total heat flux increases. The noticeable feature is that the heat flux slumps cross the flat plate before it fluctuates through the corrugated channel. In addition, it is also obvious that the heat flux is quite high at the peak of corrugation.

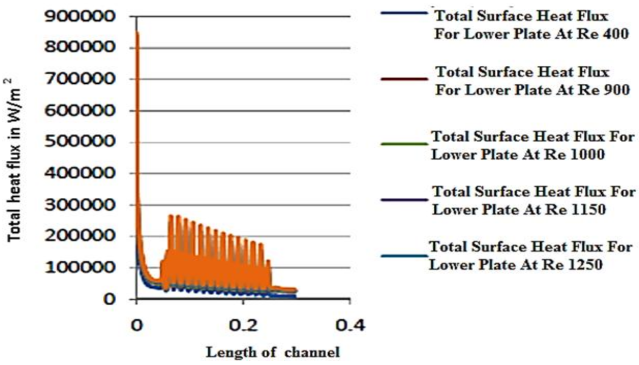

Fig 9: Total Heat Flux for Triangular Corrugated Channel

C. Results Predicated for Sinusoidal Corrugated Channel

Fig. 10 and Fig. 11 show the flow motion represented in velocity magnitude for both Reynolds numbers 400, 1400. Likewise to the triangular corrugated channel, the figures show that the vortices are created too, and the vortices grow larger and their centre move to the downstream as Reynolds number increases. But the vortices seem fractionally less than that created in triangular corrugated channel

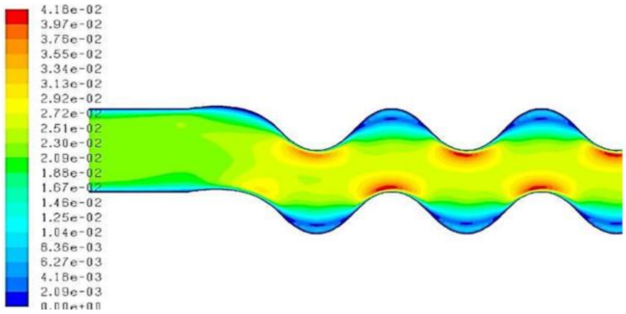

Fig 10: Velocity magnitudes at Re 400

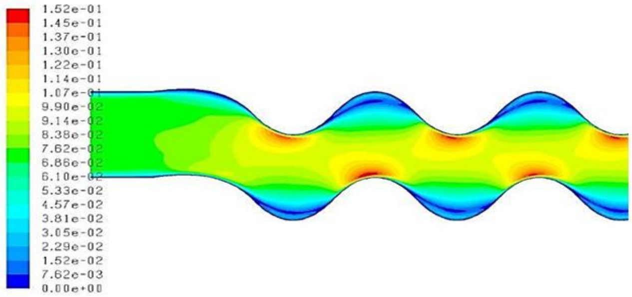

Fig 11: Velocity Magnitude at Re1400

Fig.12 and Fig. 13 show static temperature along the sinusoidal corrugated channel for a constant wall temperature boundary condition at Reynolds numbers 400, and 1400. The figures demonstrate that while the water passes through the channel, the changing in temperature becomes visible

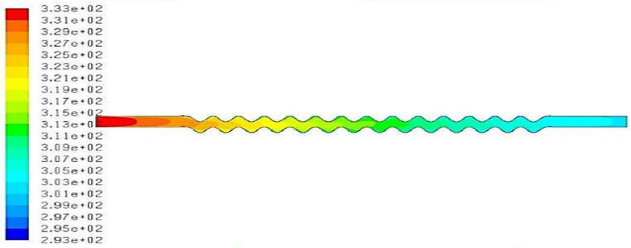

Fig 12: Static Temperature at Re400

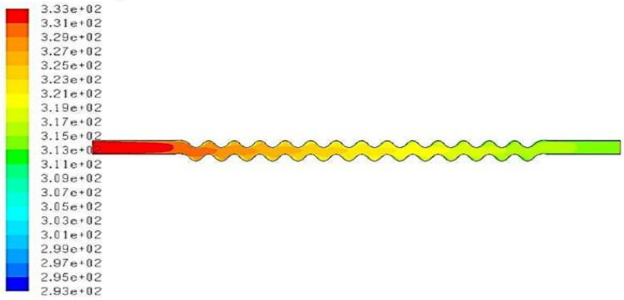

Fig 13: Static Temperature at Re1400

D. Comparison between the Triangular, Sinusoidal, and Rectangular Corrugated Channels

Fig 14 illustrates the variation of the friction factor along the lower plate of triangular, sinusoidal, and rectangular corrugated channels for Reynolds numbers ranging from 400 to 1400. It is evident that the friction factor decreases with an increase in Reynolds number for all three channel geometries. The triangular and sinusoidal corrugated channels exhibit nearly identical decreasing trends, with friction factor values varying within the range of 0.39 to 0.147. In contrast, the rectangular corrugated channel shows a much steeper decline in friction factor, with values decreasing from 5.848 to 1.7. This behavior can be attributed to the higher sharpness of the rectangular corrugations, which induces greater flow resistance compared to the triangular and sinusoidal configurations.

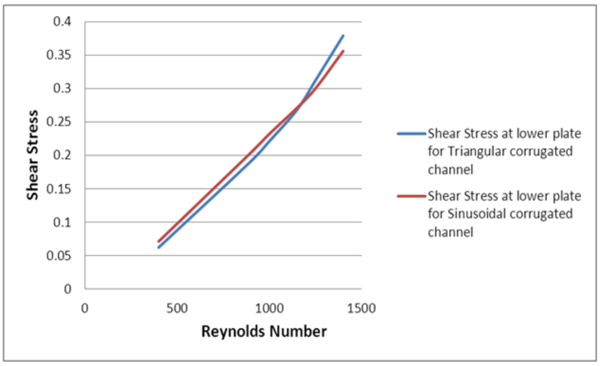

Fig 14: Comparison of shear stress between the triangular and sinusoidal channels (enlarged view).

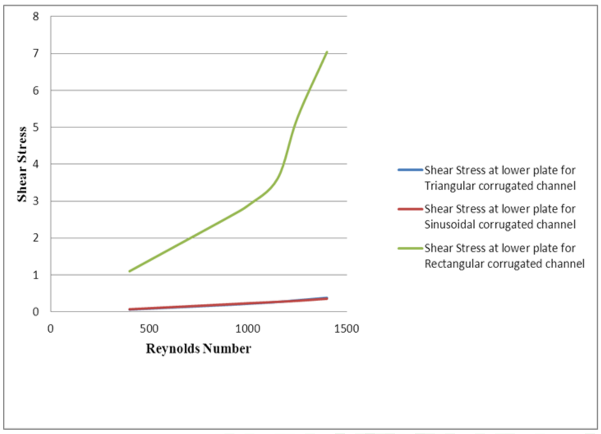

Fig 15: Comparison of Shear Stress between the Triangular, Sinusoidal, and Rectangular Channels

Table -1 Comparison of shear Stress between the Triangular, Sinusoidal, and Rectangular Channel

Reynolds number	400	900	1000	1150	1250	1400
Shear Stress at lower plate for Triangular corrugated channel	0.062	0.191	0.221	0.268	0.312	0.379
Shear Stress at lower plate for Sinusoidal corrugated channel	0.071	0.204	0.232	0.271	0.3	0.356
Shear Stress at lower plate for Rectangular corrugated channel	1.1	2.56	2.88	3.61	5.25	7.04

V. CONCLUSION

This research article presented the effect of geometry of the passage on heat transfer and pressure drop in the plate heat exchanger has been undertaken. The heat transfer enhancement effect in corrugated plates is primarily due to induced turbulence which gives higher heat transfer rate.

From earlier definitions of the Nusselt number and friction factor, it is shown that the performance of a plate heat exchanger is dependent on the plate geometry. Since this is the case, the question now lies in 'the choice of suitable plate geometry.

The pressure drop is dependent to the Reynolds number and to the pattern of corrugated surface. With increasing of Reynolds number, the pressure drop increases as well. The interesting observation is that while the pressure drop for triangular corrugation compared to sinusoidal corrugation, has increased with 10.7%, 27.94%, and 31.6% for Re 400, 1250, and 1400 respectively, the pressure drop for rectangular corrugation is equal to 160.1, 153.11, and 180.52 times of the pressure drop for the sinusoidal corrugation within the same Reynolds numbers.

The shear stress increases as a consequence of Reynolds number. The highest value of shear stress is in the peak of corrugation. From global point of view for Re 400, the shear stress for the rectangular model shows 1.1Pa and for Re 1400, the shear stress for the rectangular model shows 7.04 Pawhereasfor triangular models it is 0.062Pa and 0.379 Pa respectively

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