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Research Paper

Analysis of corrugation angle on thermal performance of plate heat exchanger

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ABSTRACT

In thermal engineering application, Plate Heat Exchangers (PHEs) are the most common devices. Enhancement in the efficiency of heat exchanger is a challenge for the R&D. In this study, three structures with corrugation angle, 45°, 33.8° and 0° (flat plate) were numerically investigated with water as working fluid using FLUENT. Effectiveness, heat transfer rate and thermal hydraulics performance (in term of pressure drop and heat transfer coefficient) were calculated. Heat transfer rate and coefficient increased with corrugation angle because of heat transfer area and turbulence, respectively which eventually enhanced the effectiveness. Effectiveness and heat transfer rate with 45° and 33.8° corrugation angles were 26 and 17%, respectively more than that of flat plate. As corrugation becomes sharp, there is an enhancement in thermal performance and pressure drop, which can be neglected as compared with increment in thermal performance. The velocity of working fluid decreased up to 30%, when increased from 33.8° to 45°. Therefore, corrugated PHE with 45° can be used as efficient heat extracting device in auxiliary loop of nuclear power reactor.

Key words: Plate heat exchanger, corrugated plate, pressure drop, heat transfer rate, efficiency, heat transfer coefficient.

Nomenclature: $\mathbf{D_e}$**,** Equivalence diameter m, N**u,** Nusselt Number; **ΔT,** temperature difference °C; Q, hate transfer rate W; **Re,** Reynolds number; $\mathbf{C_p}$**,** specific heat KJ/kg K; **Pr,** Prandlt number; **V,** volume flow rate m^3/s, **h,** heat transfer coefficient W/m^2K; **P,** pressure Pascal; **m,·** mass flow rate kg/s; **K,** thermal conductivity W/m K; **A,** area m^2.

Greek alphabet: μ, Dynamic viscosity N s/m^2; **υ,** kinetic energy dissipation rate; **k,** turbulence kinetic energy; **ε,** turbulence, kinematic viscosity m^2/s; **ρ,** density kg/m^3.

Subscripts: T, turbulent; **CFD,** computational fluid dynamics; **PHE,** plate heat exchanger; **L,** length; **W,** wall.

INTRODUCTION

The enhancement and development of heat transfer devices are mainly purposed for saving capital cost and energy, through controlling the cost (energy or material) (Elmaaty et al., 2017). Heat transfer devices are important part of nuclear reactors and used to exchange the heat from primary to secondary coolant for power generation system. It can be used to extract the heat from secondary coolant after expansion in the turbine, so that secondary working fluid can extract the maximum heat from primary coolant inside the core. Exchange of heat between two fluid media separated by the solid wall is due to temperature gradient (Kumar et al., 2016; Galeazzo et al., 2006). The heat transfer device used to exchange the thermal energy between two or more medium is known as heat exchanger (Lin et al., 2007; Elmaaty et al., 2016). Therefore, there is the need for efficient heat exchanger which can extract maximum heat from the hot fluid and also plays important role in the operation of many system such as power plants, process industries, nuclear power plants and heat transfer units (Elmaaty et al., 2017; Huminic and Huminic, 2012; Gaddamwar and Bhoyar, 2016).

During the last few decades, R&D has been push forward and focused on effectiveness (Dvořák and Vít, 2015). Double pipe or concentric pipe heat exchanger are conventionally used and replaced by plate heat exchangers (PHEs) (Achaichia and Cowell, 1988). To meet the hygienic problems of dairy industry, PHE in dairy industry was introduced in 1930 and still, most famous heat transfer devices are used in different appliances because of various merits such as lightweight, small in size, ease of handling and cleaning, high thermal performance as compared with other type of compact heat exchanger (Huminic and Huminic, 2012; Tiwari et al., 2014; Khoshvaght-Aliabadi and Hormozi, 2015; Khoshvaght-Aliabadi et al., 2016). They are safe to use, the safety of the environment and the worker around the setup is the important factor which cannot be neglected (Wu et al., 2016). PHE provided more heat transfer area as compared with the conventional type of exchangers and corrugation pattern increased heat transfer area (Gut et al., 2004). There are two methods to improve the heat transfer, active and passive methods. In active method, extra energy is provided to system in order to enhance the heat transfer rate. In passive method, no extra heat is provided and the only shape is changed (Gaddamwar and Bhoyar, 2016; Khoshvaght-Aliabadi and Hormozi, 2015). Corrugated/ Vortex-Generator Plate-Fin (CVGPF), wavy plate-fin HE using passive methods: (winglets, nanofluid and perforation) and analysis of corrugated/perforated fin with nano fluids were analyzed experimentally on the bases of hydrothermal performance (Khoshvaght-Aliabadi et al., 2016; Khoshvaght-Aliabadi et al., 2016; Khoshvaght-Aliabadi et al., 2018). A PHE consists of corrugated metallic plates with specific chevron angle. The purpose of corrugated plates is to create turbulence in working fluids during their flow, which in turn enhanced the strength of plate pack. Proper selection of corrugation angle with depth and pitch of corrugation is very important.

Durmus et al. (2009) experimentally investigated exergy loss using flat, corrugated and asterisk structure for PHE. Also, Dovic et al. (2007) experimental analyzed the effect of geometrical parameters spacing between plates and pitch of corrugation for corrugation angle 28° and 61° with air as working fluid. While Gherasim et al. (2011a, b) experimentally investigated frictional factor and Nusselt number in a heat exchanger and then extended it numerically for assessment of laminar and two equation turbulent model by comparing the simulation result with the experimental.

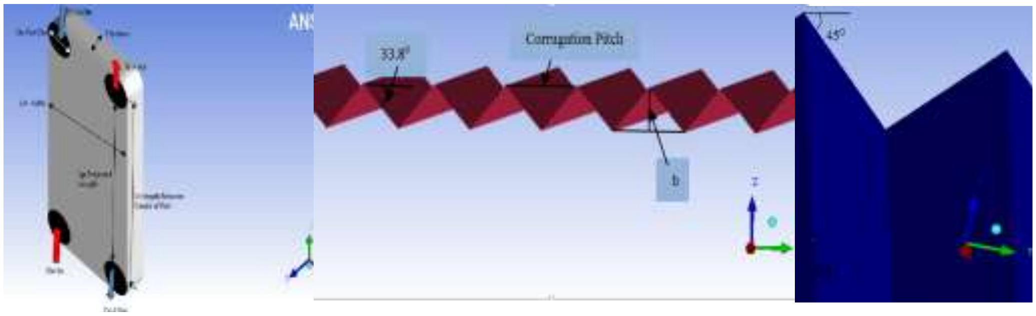
Furthermore, Tisekar et al. (2016) experimentally investigated the heat transfer rate for parallel and counter flow heat exchanger with corrugated plate structure and Reynolds number within the range of 500 to 1200. According to Rao et al. (2014), pressure drop increased with the corrugation angle. Plate with corrugation angle 40° transferred the maximum heat as compared with 30°. Pandey and Nema (2011) exergy loss was reduced due to change of the structure from rectangular to corrugation with angle of 30° because it decreased friction. The structure of PHE is very complex, therefore, it is difficult to measure the accurate value of heat transfer coefficient (Gaddamwar and Bhoyar, 2016; Faizal and Ahmed, 2012). In corrugated PHE, turbulence is created at low velocity because of corrugation pattern, therefore it is also suitable for high viscosity fluid (Mujumdar, 2000; Picon-Nunez et al., 1999). The NTU (Number of Transfer Units) and LMTD (Log Mean Temperature Difference) methods are the classical methods for designing heat exchanger, which are based on prototype and assumptions iterations through design. Due to all this reasons, CFD techniques are adopted in the design of heat exchangers (Bhutta et al., 2012).

However, there is a dearth of study regarding hydrothermal analysis and flow insight of PHE with 0°, 33.8° and 45° corrugation angle using FLUENT. From the literature and recent studies, it was reported that run simulation with phase change of fluid results evaluated are not so good (Bhutta et al., 2012), therefore water as working fluid was chosen at 373K using FLUENT. To achieve better performance, two factor are important and need to be considered: to increase the heat transfer rate, and to reduce pressure drop. The potential of proposed angle 45° is investigated for enhancement of overall performance against 33.8° and 0°. There is also dearth of study regarding the quantitative and specifically analysis of impact of corrugation angle on performance. Therefore, this study will enhance the literature and understanding for the numerical analysis of 3D model which consist of more realistic geometry, so that complex structures can be simulated with CFD.

Physical model

The size and shape of PHE depend upon its use and application. PHEs are also known as compact heat exchanger because of their size. In this study, three model were selected. One was flat plate (with zero corrugation angle) second and third were corrugated PHE with 33.8° and 45° corrugation angles, respectively as shown in Figure 1. The sharpness of corrugated structure depend upon the depth and angle of corrugation.

Some design parameters of PHEs are given in Table 1. Vertical distance between the ports is L_v and actual width of the plate is L_w. All parameters were designed within the limitation of design (Kakac and Liu, 2012).



a) Flat PHE b) Corrugation 33.8° c) Corrugation 45°

Figure 1: Plate heat exchanger.

Table 1: Dimension of PHEs.

Dimension	PHE with 45° corrugation	PHE 33.8° corrugation	Flat PHE
Length (L_v) (m)	1.1	1.1	1.1
Width (L_w) (m)	0.68	0.68	0.68
Thickness (m)	0.0007	0.0007	0.0007
Port diameter D_p(m)	0.2	0.2	0.2
Corrugation pitch p (m)	0.004	0.004	
Corrugation angle	45^0	33.8^0	
Depth of corrugation b(m)	0.002	0.003	

Table 2: Design and working fluid used in PHE.

Design of heat exchanger	Hot side	Cold side	No of cases
Flat PHE	Hot water	Cold water	Case I
Corrugated PHE with 33.8^0	Hot water	Cold water	Case II
Corrugated PHEwith 45^0	Hot water	Cold water	Case III

THEORY AND CALCULATIONS

Instead of simulating the whole system, only (2 fluid bodies and one plate) were consider as whole heat exchanger because of computational limitations. A fluid body existed between every two plates, and number of cases and working fluid used in this study are given in Table 2. Launder and Spalding (1972), in their study, presented a standard k-ε mode. Fast convergence, simplicity, up to mark accuracy and robustness are the reasons for its popularity (Launder and Spalding, 1972; FLUENT 6.3 User's Guide, n.d.). In this study, k-ε model was used with suitable coefficient.

General transport and governing equations

Governing equations continuity, momentum and energy are derived on the basic law of conservation of mass, momentum and energy (Kundu and Hu, 2004). The energy, mass and momentum equation for steady state fluid flow are given as (FLUENT 6.3 User's Guide-12.4.1,Standard -Model, n.d.):

$$\frac{\partial(\rho.\bar{u}_i)}{\partial x_i} = 0 \tag{1}$$

$$u_j \frac{\partial(\rho \bar{u}_i)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[(\mu + \mu_t)\frac{\partial \bar{u}_i}{\partial x_j}\right] \tag{2}$$

$$c_p.u_j.\frac{\partial(\rho.T)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[(k + k_t)\frac{\partial T}{\partial x_j}\right] \tag{3}$$

Reynolds number (Re) Prandtl number (Pr) and Nusselt number (Nu) are defined as:

$$Re = \frac{\rho u d_e}{\mu} \tag{4}$$

$$Pr = C_p\, \mu / K_f \tag{5}$$

$$Nu = \frac{h D_e}{k} \tag{6}$$

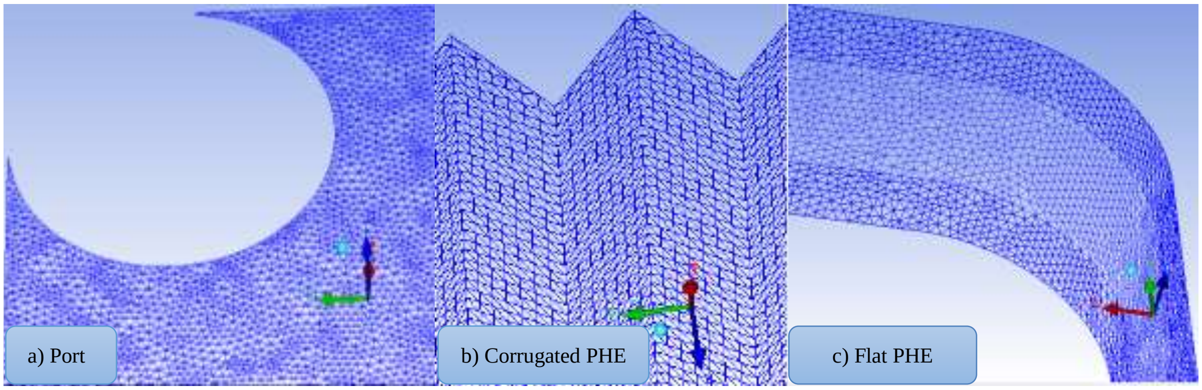


Figure 2: Mesh view of different part of heat exchanger.) Mesh across the port, b) Mesh across corrugation, c) Mesh of outside wall.

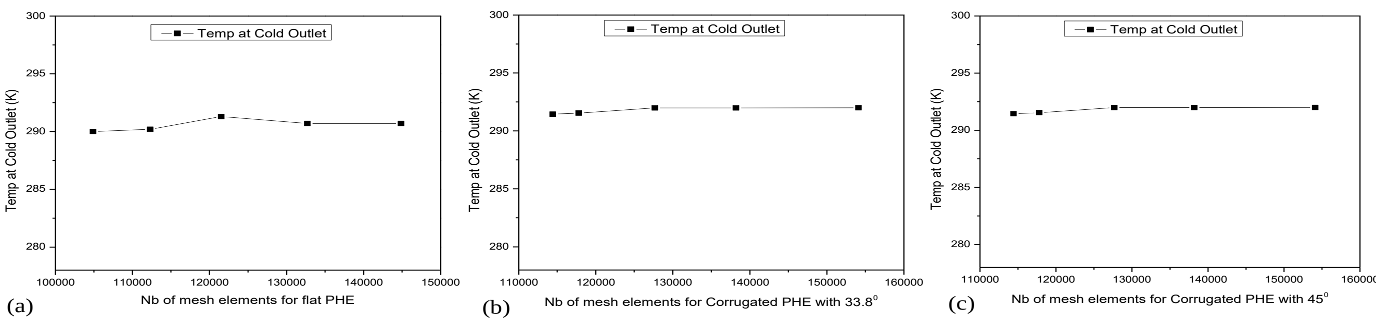


Figure 3: Grid dependency test.) Flat PHE; b) Corrugation with 33.8°; c) Corrugation with 45°.

Where C_p is the specific heat at constant pressure, μ is viscosity of the working fluid, d_e equivalence diameter, K_f is thermal conductivity of the fluid, L_h is characteristic length of the plate and h is heat transfer coefficient:

$$h = \frac{Q}{A.\Delta t_m} \quad (7)$$

Where Q is the heat transfer rate, A is the surface area of wall and Δt_m is the logarithmic mean temperature difference. Relation for the Δt_m is:

$$Q = m_h C_{ph} (T_{hi} - T_{ho}) = m_c C_{pc} (T_{co} - T_{ci}) \quad (8)$$

$$\Delta t_m = \frac{[(t_{hi} - t_{ho}) - (t_{hi} - t_{ci})]}{ln[(t_{hi} - t_{ho})/(t_{hi} - t_{ci})]} \quad (9)$$

Pressure drop was calculated by:

$$\Delta P = P_{in} - P_{out} \quad (10)$$

Mesh

Zones were created firstly in the geometry involving two fluids which one is the solid zone. Untrusted meshes were generated in ICEM, and boundary layer grids were meshed in fluid domain near to adjacent structural wall. The mesh skewness was adjusted within the range of 0.90 and mesh smoothening was high.

The number of elements and nodes for the flat plate (121528, 19940), the corrugated plate with 33.8° (127691, 19788) and the corrugated plate with 45^0 (96402, 14791) are shown in Figure 2. Also, Mesh across the port of heat exchangers, across corrugation and outside wall mesh of flat PHE are shown in Figure 2.

Quality of numerical simulation is dependent on grid independency test, therefore validation of mesh model is done from outlet temperature of cold water. It was calculated across different number of mesh elements for three models, as shown in Figure 3. For flat PHE, grid independency shows after 121528, while 33.8° corrugated PHE at 127691 mesh elements. Similarly, 45^0 shows 964021 number of mesh elements.

Thermal and boundary conditions

CFD techniques were used for the solution of governing equations with following assumptions:

1. Heat cannot enter or leave the system, therefore all outer surfaces or boundaries are assumed to be adiabatic.

Table 3: Inlet boundary condition of fluid.

S/N	Velocity inlet (m/sec)	Volume flow rate at inlet (Liter/min)	Hot fluid inlet temperature (K)	Cold fluid inlet temperature (K)
1	0.00201	3.71934084		
2	0.00402	7.4386861		
3	0.00603	11.1580224	373	278
4	0.00804	14.8774476		
5	0.009999	18.5023326		

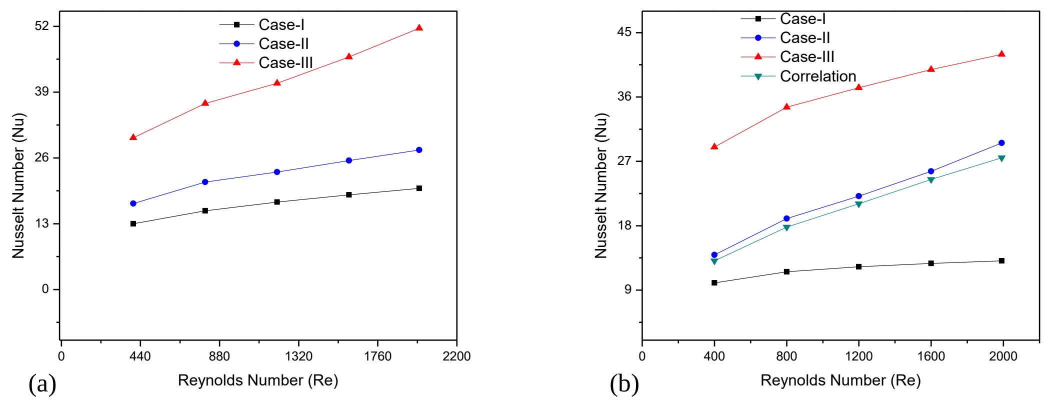


Figure 4: Verification of model. a) Nusselt and Reynolds number; b) Comparison of Nusselt number with correlation.

2. On surfaces, there was no effect of fouling.
3. System was steady state.
4. Volume flow rate was same for at inlet of both hot and cold fluids.
5. k-ε and energy model were used for numerical simulation of PHE with CFD, as these two model have ability to resolve the problem of simulation (Tiwari et al., 2014; Gherasim et al., 2011a).

Inlet: Velocity inlet boundary condition was defined at inlet with temperature in the thermal tab. Inlet temperatures and velocities of hot and cold fluids are given in Table 3. System was operated on atmospheric pressure, and every simulation turbulence intensity (Mehendale et al., 2000) was calculated using Equation 11. Hydraulic diameter for flat plate 0.002, 33.8° is 0.006 and for 45° is 0.004 m:

$I = 0.16.Re^{(-1/8)}$ (Khoshvaght-Aliabadi and Hormozi, 2015)

Outlet: Pressure outlet boundary condition was applied and back flow outlet temperature was average of both hot and cold fluid inlets temperature.

Wall: All outlet boundaries are defined as wall and coupled thermal condition for inner shadow wall. Motion of wall was stationary, no slip in shear condition with default setting.

Conjugate condition was applied at the interface of fluid and solid plate conjugated heat transfer condition was coupled by formula $q_s = q_f$, $T_s = T_f$, while, T and q represented temperature and heat flow, subscripts f and s referred to fluid and solid, respectively. Computations were performed on Fluent 16.0 solver. On the pressure based solver, steady state and velocity formulation method was chosen. Standard k-ε model with standard wall function with default setting was chosen. In solution, initialization standard method was chosen than run simulation.

Numerical model was verified by calculating Nusselt number (Nu) with correlation and simulation. It increased with the Reynolds number (Re), but its value was high for Case-III and II as compared with Case-I (Figure 4a). Nu decreased, if corrugation angle was reduced. Corrugation angle increased the turbulence, which increased the heat transfer coefficient. Therefore, the value of Nu is maximum with corrugation angle 45°. Nu is given by the relation $\mathrm{Nu} = \mathrm{C} * \mathrm{Re}^{\mathrm{m}} \mathrm{Pr}^{1/3}$ (Kakac and Liu, 2012)C, where m is the constant and m=0.668, C=0.3for 33.8°. The relationship between Re and Nu is shown in Figure 4b. The average uncertainty between simulation and correlation results is about 4.2%, which is in good agreement with them. It can be neglected because of limitation of software and correlation. There is a good agreement between experimental studies and simulations, yielding results within the range of 2 to 10%. In some exceptional cases, it may vary up to 36% (Bhutta et al., 2012).

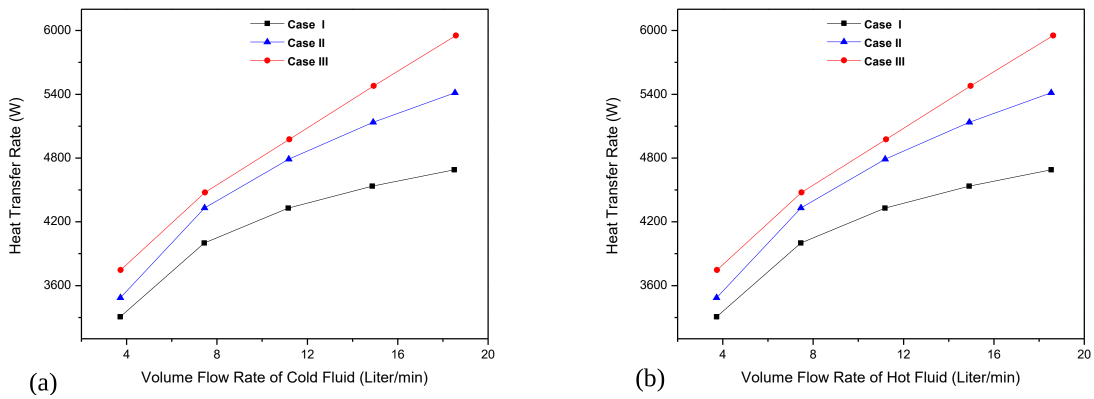


Figure 5: Effect of structure on heat transfer rate . a) Heat transfer rate verse volume flow rate of cold fluid. b) Heat transfer rate verse volume flow rate of hot fluid.

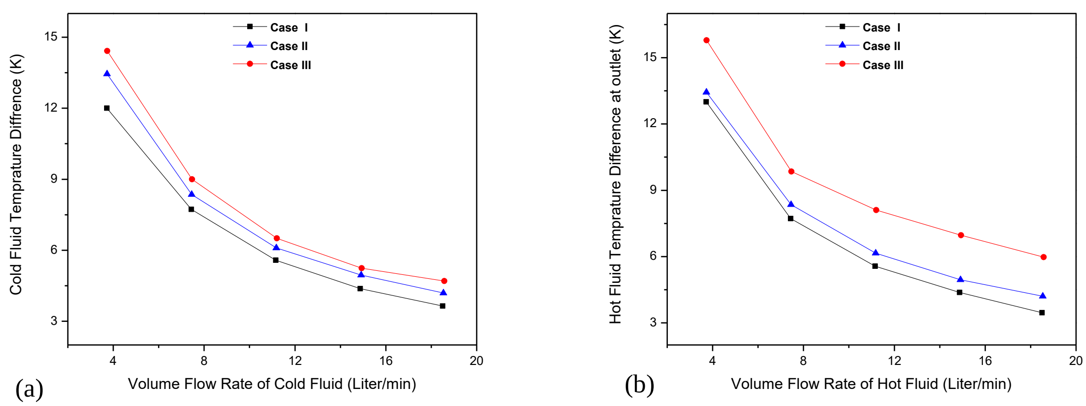


Figure 6: Effect of structure on temperature difference at outlet. a) Cold fluid temperature difference verse its volume flow rate. b) Hot fluid temperature difference verse its volume flow rate.

RESULTS AND DISCUSSION

The detail of design and number of cases with working fluids are given in Table 2. The working parameters and inlet temperature for both fluids are listed in Table 3. CFD results are evaluated and discussed hereafter.

Corrugation angle of the plate has effect on heat transfer rate between fluids through the plate. Therefore, heat transfer rate as a function of volume flow rate of cold and hot fluids is shown in Figure 5a and b. The curves for heat transfer rate moved high as volume flow rate increased. For Case II and III, curves were higher as compared with Case I because of different corrugation angle of plates. In Case I, the structure of the plate was flat and showed less heat transfer area as compared with the other corrugation angle. Plate structure with corrugation45^0 provides more heat transfer rate, as the corrugation becomes sharp heat transfer rate and contact area of fluid are increased. The increment in the heat transfer rate for the corrugated plate with 45^0corrugation was 17% as compared with flat plate. Similarly, the increment in heat transfer rate between corrugation angle 33.8° and 45° was 7%.

The important parameter in heat exchanger is the outlet temperature because heat transfer rate is difference from inlet and outlet temperature for hot side, outlet minus inlet temperature for cold side. So the effect of volume flow rate on difference of temperatures at outlets is shown in Figure 6a and b. If the volume flow rate of the hot and cold fluids is increased, then outlet temperature difference is decreased. In Case III, there is further temperature difference at low flow rate. This is because at low flow rate of hot fluid outlet, temperature decreased, which increased the cold fluid outlet temperature, respectively.

Tiwari et al. (2014) reported that temperature difference decreased by increasing the volume flow rate of nanofluid, and Gherasim et al. (2011b) examined the same trend of curves during assessment of two equation turbulence model. The temperature of cold fluid at outlet of corrugated

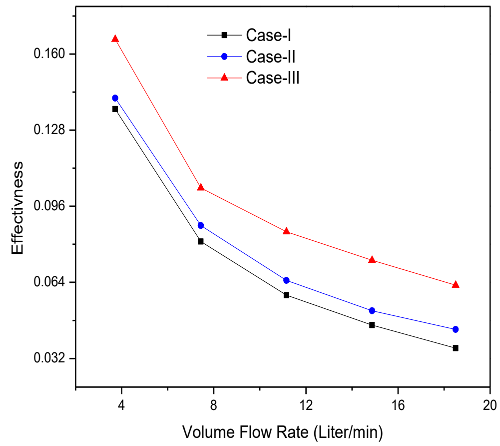


Figure 7: Effect of corrugation angle on performance of PHE.

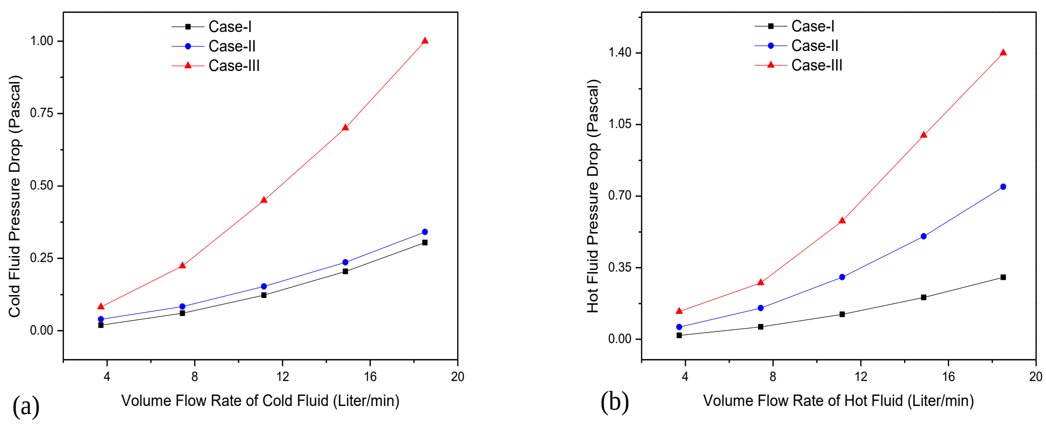


Figure 8: Effect of corrugation angle on pressure drop. a) Cold fluid pressure drop verse its volume flow rate. b) Hot fluid pressure drop verse its volume flow rate.

plate with corrugation 45° was 16%, which is more as compared with flat PHE. This temperature increased to about 10%, when corrugation angle increased from 0° to 33.8°. As corrugation increased from 33.8° to 45°, then the temperature at the outlet of cold fluid increased to about 7%. Thermodynamic laws revealed that heat transfer rate and outlet temperature depend on the heat transfer area, these results are also in accordance with these laws.

The effectiveness of heat exchanger is depended on flow rate and heat capacity of the fluid. It is the ratio of actual heat transferred from hot to cold fluid with maximum heat transfer rate. Figure 7 shows the trend of curves for effectiveness with volume flow rate. During the numerical analysis, it was observed in the results that effectiveness decreased with the flow rate because actual heat transfer reduced as compared to maximum heat, but it also changed with the corrugation angle. The area of contact of fluids and turbulence across the flow path depends on corrugation angle of the plate, secondly heat transfer rate increased with the corrugation angle. While in Case III, effectiveness was high as compared with other two cases.

Plate with 45° corrugation was suitable for recovery of the energy as compared to other structures. For the same amount of energy, PHE with corrugation pattern of 45° required less number of plates. Effectiveness with corrugation angle 45° was 26.91% more than that of flat one. If corrugation increased from 33.8° to 45° then effectiveness increased 20%.

Pressure drop for hot and cold fluids with volume flow rate is shown in Figure 8a and b. Pressure drop increased with the flow rate. Highest pressure drop rate is found in Case III. Meanwhile, outlet temperature difference is also high in Case III. Pressure drop depends on geometrical dimension and density. Difference between Case III and II is on bases of corrugation angle of plate. Secondly, density variation for the case III is more because of the variation in temperature of the working fluid. In Case III, more turbulence is provided to fluid and friction as compared

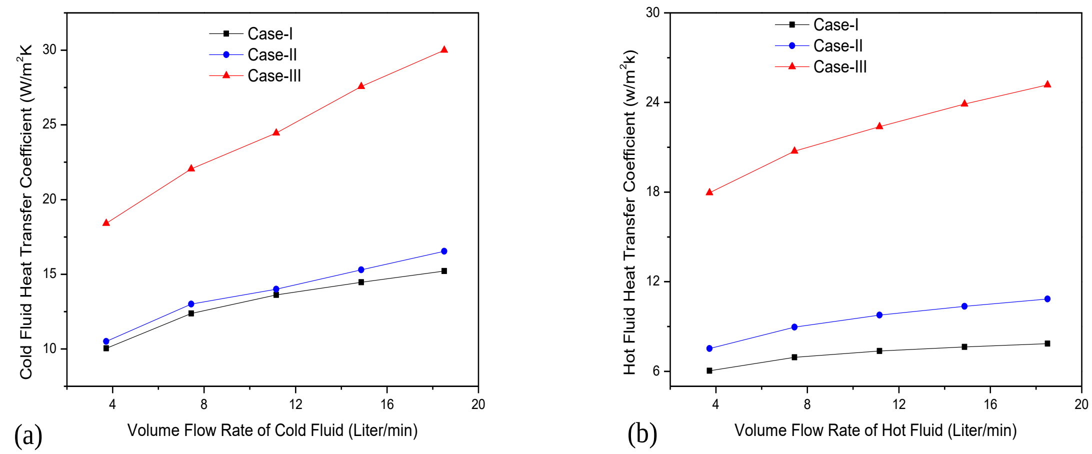


Figure 9: Volume flow rate effect on heat transfer coefficient.

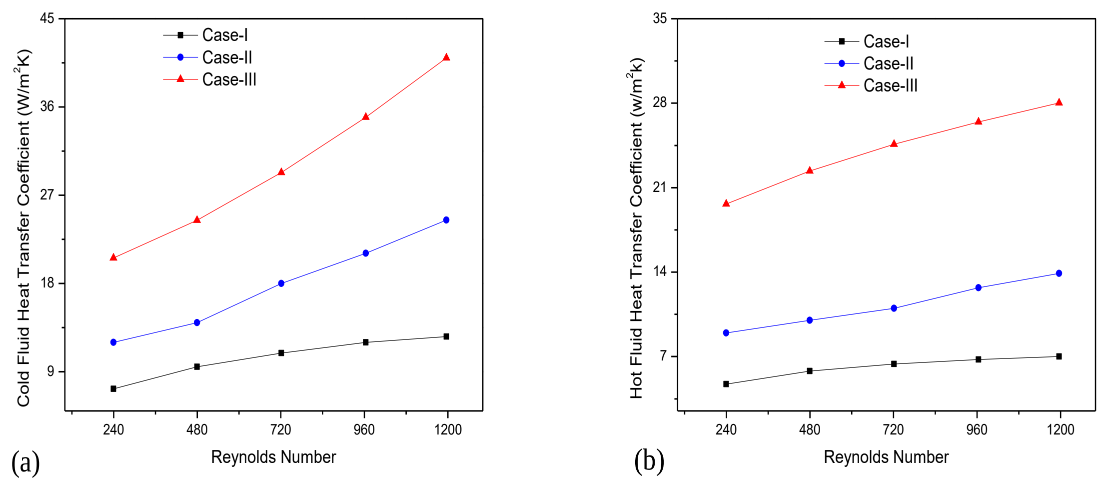
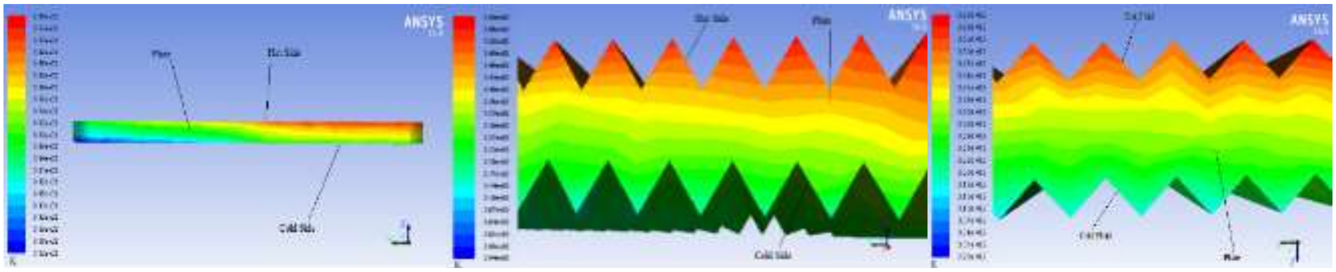


Figure 10: Effect of Reynolds number on heat transfer coefficient.

with other structures. The increment rate in pressure drop is less for Case III, but it can be neglected as compared to increment in heat transfer rate. Rao et al. (2014) also explained experimentally in the evaluation of pressure drop of viscous fluid. Increment in heat transfer rate is more important because the purpose of heat exchanger is to extract the maximum heat from the hot working fluid.

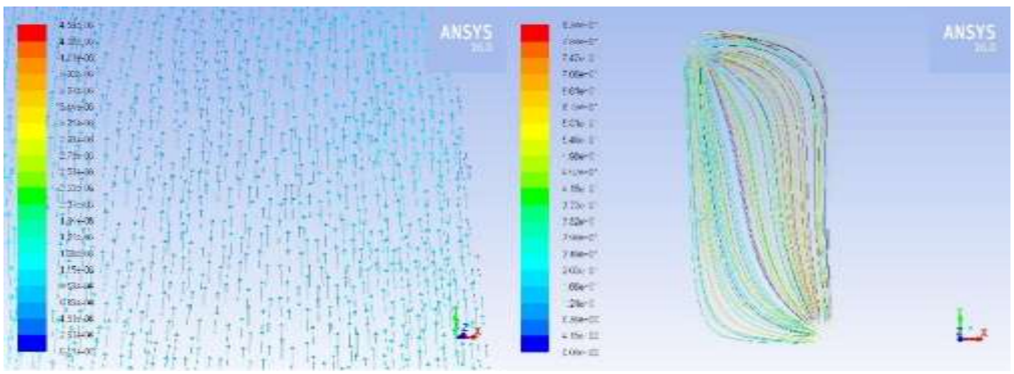
The relationship between volume flow rate and heat transfer coefficient is shown in Figure 9a and b. It can be concluded that if flow rate is properly controlled, then heat transfer coefficient can be enhanced. Heat transfer coefficient depends on the turbulence of fluid, which further depends on the design of plate and velocity of working fluid. It can be increased by changing the volume flow rate of fluids. When flow rate is increased, then turbulence increased, and in case of corrugation structure at low value of velocity, turbulence can be achieved. While in Case III, heat transfer coefficient is high as compared with the other two cases because velocity decreased about 30%, when corrugation angle increased from 33.8° to 45°. Murugesan and Balasubramani (2012) analyzed during experimental study of two fluids that heat transfer coefficient increased by increasing the volume flow rate of fluids. Average increase in heat transfer coefficient of both sides for flat and corrugated plate with 33.8° corrugation is about 14%, but it increased interestingly up 29% for corrugation 45° due to sharpness of edges. Its mean heat transfer coefficient also increased with the corrugation angle and small corrugation angle have no specific effect on it. This study also show the similar behavior with Case III, coefficient value is high because of turbulence phenomena, which enhance the importance of corrugation with 45°.

Reynolds number (Re) has effect on the heat transfer coefficient because it enhanced turbulence of the fluid between the plates which further increased heat transfer coefficient. Figure 10a and b shows the increment of heat transfer coefficient with Re. Therefore, in Case III, heat transfer coefficient is more as compared with other cases.



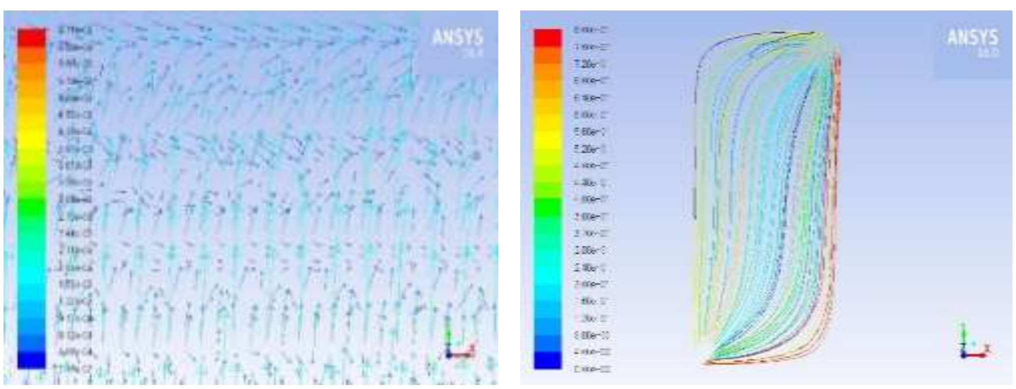
a) Flat plate b) Corrugated plate with 33.8° c) Corrugated plate with 45°

Figure 11: Temperature contour.



a) Velocity vector b) Path line

Figure 12: Flat PHE.



a) Velocity vector b) Path line

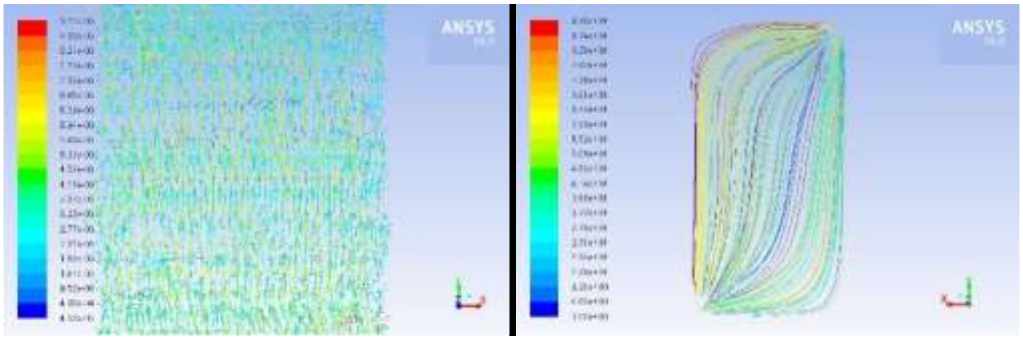
Figure 13: Corrugated 33.8° PHE.

Corrugation pattern increased turbulence which is more higher in Case III, therefore heat transfer coefficient increased with Re (Safri et al., 2017).

The temperature contours for three structures are shown in Figure 11. It can be seen from the figure that solid plate provides the heat transfer due to conduction, and fluid zone provides convection heat transfer.

There is heat transfer between two zones due to formation of temperature profile, as shown in Figure 11. The highest temperature appears in the upper zone due to hot fluid and the lowest across the lower zone which is cold fluid is due to its flow over the plate. The formation of these counters can be attributed to the transfer of heat from hot to cold fluid through the plate. Highest temperature counter is shown across the plate with corrugation angle 45°.

Velocity vectors and path lines are shown in Figures 12 to 14 in which: a) represented velocity vector for flat and corrugated PHE consecutively and, b) shows the path line of each fluid flow from cold inlet. It can be estimated from part (a) that velocity vectors are smooth for flat plate and



a) Velocity vector   b) Path line

Figure 14: Corrugated 45° PHE.

velocity variation for corrugated plate 45° and 33.8° are not smooth. Maximum variation in velocity vectors is only for corrugation 45° because it produces maximum turbulence.

Conclusion

This study provides complete understanding about the hydrothermal analysis and flow insight into the complex structure of PHE with water at high temperature using FLUENT, as three different structures were simulated with CFD. Based on numerical analysis and results, the followings points are concluded:

1. Heat transfer rate increased with corrugation angle or sharpness of the corrugation pattern. It increased up to 17% as corrugation angle increased from 0° to 45°.
2. The outlet temperature difference also depends on corrugation angle. It was16% higher for the corrugated plate with 45° as compared with flat plate. Therefore, effectiveness was increased 26% as the corrugation angle increased from 0° to 45° corrugation angle.
3. Heat transfer coefficient is depend on the angle of corrugation and flow rate. The velocity of the working fluid decreased about 30% when corrugation angle increased up to 45° and turbulence was increased. Pressure drop with sharp corrugation was more but it can be neglected as compared with increase in heat transfer coefficient.
4. Nusselt number value also increased for corrugation angle range of 33.8° to 45° and the presented analysis agreed well with the correlation. Therefore, the data will be helpful for designing the efficient heat exchanger with less number of plates for high energy generating plants.
5. Corrugation structure with 450 corrugation angle will be suitable for the heat exchanger used in auxiliary system of fission reactors.
6. It can be suggested that selection of angle is important, which reduced the size of heat exchanger by increasing effectiveness. Use of corrugated heat exchanger is better than the tube type heat exchanger with the same effectiveness as it covers less space.

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