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Thermal Performance Comparison of Various Corrugated Channels on a Corrugated Heat Exchanger using CFD

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Abstract- **In recent years, research on the methods for heat transfer enhancement in heat exchangers has received great attention in order to cater for the growing needs of higher efficiencies in these devices. Corrugated surface geometry is one of the many suitable techniques to enhance the heat transfer in heat exchangers. When fluid flows in a corrugated channel, the flow becomes disturbed due to growing recirculation regions near the corrugated wall, which enhances the mixing of fluid as well as heat transfer. On the coolant side, the use of nanofluids, a liquid in which nanoparticles are added to a base fluid, can also enhance the heat transfer due to the improved thermal conductivity of the fluid.**

The basic geometry of the present problem consists of corrugated channel which included velocity inlet condition and temperature of 300K, pressure outlet condition while slip velocity was ignored. Top and bottom walls for the test section are subjected to a constant heat flux of 10kW/m2 while adiabatic boundary condition is applied to the remaining walls. A numerical simulation is performed on thermal performance comparison of a corrugated channel with three corrugation profiles. Semicircle, triangular, and mixed are considered as corrugation profiles for corrugated walls of channel using nanoparticles volume fractions of Al2O3 and Reynolds number ranging from 0 to 0.06 and 10,000–30,000, respectively. Moreover, the results of triangular and mixed corrugated channel are compared with Semi-corrugated channel. Governing equations are discretized by using a finite volume method (FVM) and SIMPLE algorithm. The second order upwind scheme is applied and standard k-ε turbulent model with standard wall function is selected. The results showed that the Triangular channel has the average Nusselt number greater than 1.3 times of mixed shape (Semi-circular + Triangular) and 1.7 times of Semi-circular corrugated channel. Finally, it was found that among corrugated channels, the triangular channel provides the highest thermal performance followed by the mixed shape (Semi-circular + Triangular) and Semi-circular corrugated channels.

Index terms- **Nanofluids, Corrugated channels, Reynold's number, Heat transfer, Nusselt Number, CFD**

I.INTRODUCTION

Heat transfer is a science that explores energy exchange between two entities because of the difference in temperature. In principle, the kinetic energy of microscopic molecules is related to thermal energy. The thermal agitation of its constituent molecules is enhanced if the material's temperature increases [1]. The areas with higher molecular kinetic energy will then shift this energy to areas with lower kinematic energy. Therefore, if an individual or material is at temperatures separate from their surroundings, heat transfer happens to maintain a thermal equilibrium between the body and the surrounding environment [2].

Generally there are three types of heat transfer which are conduction, convection and radiation. Conduction is the transfer of heat within an object or between two objects in contact. Convection heat transfer occurs when a fluid (liquid or gas) comes in contact with a material of a different temperature. For natural convection, it occurs when the flow of a fluid is primarily due to density differences within the fluid due to cooling or heating of that fluid [3,4]. Meanwhile, forced convection occurs when the flow of fluid is primarily due to pressure differences. Radiation is the transfer of heat from one object to another by means of electro-magnetic waves. Radiation does not require objects to be in contact or fluid flow between those objects, it occurs in the void of space (that's how the sun warms us). Transfer of thermal energy can also occur with any combination of the three. In our study which relates to plate heat exchanger, it consists of heat transfer by conduction and convection.

Enhancement of heat transfer surfaces has developed over the years, and is the main focus in the heat exchanger industry. Enhanced surfaces yields higher heat transfer coefficient when compared to unenhanced surfaces. Surface can be enhanced basically by two types; either active enhancement which requires deployment of external power which is obviously higher in operational and capital cost thus commercially unviable, and passive enhancement which adding extended surfaces(e.g. fins),or employing interrupted surfaces (e.g. corrugations) [5,6,7].

1.1 Heat Transfer Enhancement

Heat transfer enhancement is a process of increasing the heat transfer rate and thermo hydraulic performance of a system using various methods. The methods of heat transfer enhancement are employed for developing the heat transfer without affecting the overall realization of the systems significantly, and it covers a wide range of areas where heat exchangers are used for such functions as air-conditioning, refrigeration, central heating systems, cooling automotive components, and many uses in the chemical industry.

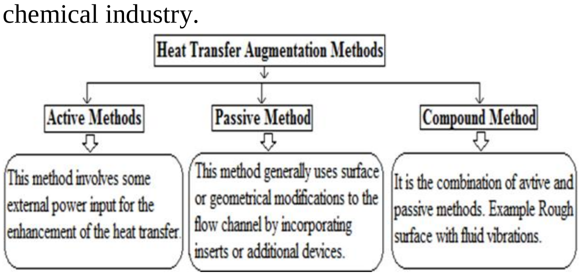

Figure 1. Augmentation methods

Recently corrugated types of heat exchangers have come into picture. Their advantages include increased heat transfer, reduced servicing costs, compact design and minimal fouling. Because of these advantages, corrugated type heat exchangers are gaining popularity. A lot of research is being done in their design field.

II. LITERATURE REVIEW

Among all, plate type heat exchangers are widely used in industries such as automobile industries, dairy, chemical industries, power processing industries, pharmaceutical industries. The plate may be plane or may have some distortions in it. Creating uniform undulations is very typical and it is called corrugated plate heat exchanger. Corrugated plate heat exchanger is of sinusoidal shape which contains crest and trough due to which rate of heat exchanger increases. The performance, effectiveness and the rate of heat transfer of corrugated plate heat exchanger is much higher as compared to the plane plate heat exchanger.

Numerous works has been done on Plate Heat Exchangers (PHEs) and their data related to thermal and hydraulic characterization are available in open literature. But there is a widespread discrepancy in these reported correlations and before commencing the present study, it was necessary to analyses the experimental facilities and procedure, data reduction methods, results and conclusions of some of the important past works.

2.1. Previous work

Mohammed and Abed (2008) numerically studied laminar forced convection heat transfer and fluid flow characteristic in a corrugated channel. Temperature of the channel walls was maintained constant which was higher than fluid temperature. Effect of wavy angle and Reynolds number were studied on fluid flow and heat transfer. The range of the Reynold`s number was carried out for the solution was found out to be 500 to 2500, wavy angles range was from 0° to 60° and Prandtl number was 0.7. It was found that the optimum values of the heat transfer enhancement and pressure drop were 3.6 and 1.11 times higher than those from the plane channel at wavy angle $\lambda = 40°$, respectively [1].

Marjan et.al (2008) has made an experimental study over corrugated plate heat exchanger by using multi-walled carbon nanotubes (MWCNT). To investigate friction loss, heat transfer coefficient by convection, Nusselt number, pumping power and pressure drop in a counter flow corrugated plate heat exchanger different water-based nano-fluids such as Gum Arabic-treated multi-walled carbon nanotubes (MWCNT-GA), functionalized MWCNT with cysteine (FMWCNT-Cys) and silver (FMWCNT-Ag) were employed as coolants. From the experimentation it was found that by increasing Pecelet number, Reynolds number, or fraction of nonmaterial improve marks in the characteristics heat transfer of the nanofluid increased. In all investigated cases, it has been found that power consumption and heat transfer rate is less for water compared to nanofluids. Besides that it has been also found that for a specific pumping power, heat removal in nanofluids is higher as compared to water. Therefore, performance of the heat exchanger can be enhanced by choosing MWCNT water as the working fluid [2].

Khan and Kumar (2009) described performance and exergy of corrugated plate heat exchanger in parallel or in counter flow. Plate had sinusoidal wavy surface with corrugation angle of 45°C. Heat exchanger contained 3 Channels. Hot fluid flow at the middle channel which was cooled by water through outer channels. Hot water temperature was in the range of 40°C to 60°C. Reynolds number was in the range of 900< Re>1300 for hot and cold fluid. After performing experiment performance or effectiveness of corrugated plate heat exchanger for counter flow was found out to be 44.5% more as compared to parallel flow arrangement. As well as exergy loss in counter flow is 7.2% less as compared to parallel flow [3].

Rao et.al (2009) in their investigation used corrugated plate heat exchanger with corrugation angle 30°, 40°, 50°. Water was taken as heating medium while Glycerol was taken as test fluid. The inlet and outlet temperature of hot fluid and test fluid was measured by means of four thermocouples. From the experimental investigation it was found that 50° corrugation angle heat transfer increased. It is also found that 60% Glycerol had high rate of heat transfer as compared to the 50%, 60% and water. Hence in investigation it has been found that with the increase of corrugation angle as well as with the increase of viscosity of fluid heat transfer rate increases [4].

Kumar et.al (2010) has made an attempt to investigate the performance and effectiveness of corrugated plate heat exchanger. Experiment was conducted on three channels 1-1 passes of corrugated plate heat exchanger. Hot fluid was made to flow at the middle channel while the cold fluid flow at top and bottom channel in counter and in parallel flow. Plate had a sinusoidal shape at an angle of 30° corrugation angle. Temperature of hot fluid was in the range of 50°C to 70°C whereas temperature of the cold fluid was in the range if 30°C to 40°C inlet. It was found that the effectiveness of counter flow heat exchanger is 48% higher than the parallel flow. As well as exergy loss was also calculated and found 33% less in counter flow arrangement as compared to the parallel flow arrangement [5].

Rao et.al (2010) made experimental studies on a sinusoidal corrugated plate heat exchanger where water was taken as test fluid. Two stainless steel sheets of thickness of 1 mm were used to fabricate plate heat exchanger for performing test channels. It had clearance of 5 mm and length 30 cm. Total 3 heat exchangers were fabricated by using these plates with corrugation angles of 30°, 40° and 50°. From the experiment it has been found that corrugation angle affected the heat transfer rate and pressure drop. During experiment it was found that pressure drop of fluid increases as the corrugation angle of plate increases due to which friction factor decreases. Turbulence is generated in the channel due to increase in pressure drop. From the result it has been found that as the angle of corrugation increases, pressure drop of fluid increases due to which rate of turbulence increased which lead to increase the rate of heat transfer [6].

Jixiang et.al (2011) has investigated about heat transfer and flow characteristics on corrugated plate heat exchanger by shifting upper and lower plate and varying Reynold's number $2000 \leq Re \leq 10000$. Hydraulic performance of corrugated plate heat exchanger changed due to the effect of phase shift and Reynold's number which was numerically investigated. Based on the numerical results relation between heat transfer coefficient, Nusselt Number Nu and flow friction factor f are established. By using streamlines, flow characteristics were visualized. By increasing the phase shift, Nusselt Number and friction factor decreases. By increasing Reynold's number and friction factor f goodness factor G decreases. While Nusselt is opposite change trend. The distribution of streamlines is closely related to the performance of thermal hydraulic. When the streamline distorts is more the resistance loss is greater and the heat transfer rate is high. These channels of phase shift from 0° to 90° had better overall performance, and the 0° channel had the optimal performance in lower Reynolds number region [7].

Faizal and Ahmed (2012) performed an experiment on corrugated plate heat exchanger for small temperature difference applications. It had 20 corrugated plates placed parallel to each other and its total heat transfer area was 1.16298 m2. The spacing between the plates varied 6 mm, 9 mm, and 12 mm. Minimum rate of heat transfer was obtain when distance between plates was 6 mm. Water had been used as a hot and cold fluid which flow in alternate channels. In this experiment effect on rate of heat transfer was determined while varying flow of hot water whereas the cold side flow rate and the hot and cold water inlet temperatures were kept constant. In the result it was found that when the mass flow rate of hot fluid increased corrugated plate enhance turbulent at higher velocity which increases heat transfer rate. The overall heat transfer coefficient U, the pressure losses and the average thermal length are found to increase with increasing hot fluid flow rate and heat transfer rate of heat exchanger with 6 mm heat exchanger compared to other values. The plate heat exchanger with 6 mm is found to be appropriate due to effective high thermal length and heat transfer when the pressure loss is higher [8].

Pandey and Nema (2012) made an attempt to determine the characteristics of heat transfer of corrugated plate heat exchanger. The test section contained three identical channels having corrugation angle 30° with cold air flow in the middle section while hot water was made to flow at adjacent channels. Sinusoidal wave arcs connected with tangential flat portion made corrugation angle in transverse direction. Reynold's number of water was varied from 750 to 3200 and 566 to 2265 for air which depends on the hydraulic diameter. In this experiment Prandtl number were approximate constant 2.55 for water and 0.7 for air. Experimental investigation was done by changing mass flow rate of air and water. It has been found that as pressure decreases heat transfer rate increases. Nusselt number correlation is developed for both air and water while friction factor was developed for air only. Nusselt number depends on the Reynold's number. Heat transfer coefficient of air varied from 162.73 W/m2K to 204.18 W/m2K and Nusselt Number varied from 15 to 33.Whereas rate of heat transfer of water varied from 633.42w/m2K to 2029.17w/m2K and its Nusselt number varies from 65 to 210 [9].

Dnyaneshwar et.al (2013) focused on the modeling a copper plate heat exchanger for milk pasteurization in a food industry using high temperature for a short time. This paper presents analytical and CFD analysis of pressure drop of counter flow for milk and water over copper and steel plate type heat exchanger for determining the energy required for circulating fluid. Knowing all operations parameters problem was first solved theoretically by LMTD. After that comparison between CFD and analytical result it was done. It was found that energy required for the circulation of water & milk is very low in copper plate type of heat exchanger as compared to the steel plate type of heat exchanger [10].

Giurgiua et.al (2014) numerically studied two different models on plate heat exchanger. Geometry of plate influence rate of heat transfer. One plate heat exchanger contains mini channels at 30° while other plate heat exchanger contains mini channels at 60°. In result from CFD and numerical analysis it has been found that plate heat exchanger with 60° mini channels give high rate of heat transfer as compared to 30° mini channel heat exchanger [11].

Kumar et.al (2014) investigated the performance of baffle shell and tube heat exchanger by using CFD tool ANSYS. The work was carried out to determine the performance of heat exchanger by changing the inclination of baffles in shell and tube heat exchanger. Three different baffles inclination angles namely 0°, 45° and -45° were used in CFD modeling to find the impact of baffle inclination angle on the characteristics of heat transfer and also on fluid flow. As the result of CFD comes out it had been observed that the steady state heat flux comes out to be more in the case of +45 degree baffles case than -45 degrees baffles case. The heat flux comes of 0 degree baffles come out intermediate between 45 and -45 degrees cases [12].

Melvinraj et.al (2014) has investigated on a parallel flow heat exchanger corresponding ribbed tube heat exchanger has also been modeled and numerically analysed. For designing and analysis purpose Pro-e and ANSYS 14.5 has been used respectively. The effectiveness of two heat exchangers has been compared using CFD. The ribbed heat exchanger effectiveness is more than that of simple heat exchanger. Due to the shape of ribbed helical tube fluid flow is not parallel but in swirls, which increases turbulence and thereby increasing the effectiveness [13].

Kansal and Sahabat (2015) deals with the study of shell and tube heat exchanger by using KERN method and CFD simulation. Main aim of this work is to determine effectiveness of shell and tube heat exchanger. Methanol has been used as a hot fluid in shell side its inlet temperature is 368 K whereas water was used as fluid flow in tube at the inlet temperature of 298 K. From simulation it has been found that outlet temperature of methanol is 313 K and 315.53 K from KERN method and CFD respectively. Whereas outlet temperature of water is 313 K and 308.43 K from KERN and CFD respectively. From KERN method the effectiveness of heat exchanger was found out to be 0.79 and from CFD it has been found to be 0.76. Both the results are in close agreement with each other [14].

Hasanpour et al. (2016) have experimentally studied a double pipe heat exchanger with inner tube corrugated filled with various categories of twisted tapes from conventional to modified types (perforated, V-cut and U-cut). The twist ratio, the hole diameter, the width and depth ratio of the cuts have been varied and the Reynolds number has been changed from 5000 to 15000. Overall more than 350 experiments were carried out. Nusselt number and friction factor for corrugated tube equipped with modified twist tapes are found out to be higher than typical tapes [15].

Johnson et.al (2017) studied the analytical design of the heat exchanger which has been also numerically analyzed. On the basis of standard k- ε modelling CFD analysis have been done. The solution of the problem yields when the optimum values of flow rate, outer diameter of pipe and inner diameter of pipe to be used at an effective length for a double pipe heat exchanger. When the stream processes for specified flow rates then it was treated for a given inlet to outlet temperature. From the result it has been found that the design and analysis of the double pipe heat exchanger would be a great success [16].

R K Ajeel et.al (2017) studied CFD study on turbulent forced convection flow of Al_2O_3-water nanofluid in semi-circular corrugated channel. Computational Fluid Dynamics (CFD) simulations of heat transfer and friction factor analysis in a turbulent flow regime in semi-circle corrugated channels with Al_2O_3-water nanofluid is presented. Simulations are carried out at Reynolds number range of 10000-30000, with nanoparticle volume fractions 0-6% and constant heat flux condition. The results for corrugated channels are examined and compared to those for straight channels. Results show that the Nusselt number increased with the increase of nanoparticle volume fraction and Reynolds number. The Nusselt number was found to increase as the nanoparticle diameter decreased. Maximum Nusselt number enhancement ratio 2.07 at Reynolds number 30,000 and volume fraction 6% [17].

Junqi et al. (2018) has experimentally investigated the thermal hydraulic characteristics for three types of fluids (R245fa, glycol & water) on plate heat exchanger surface. To overall evaluate the enhanced heat transfer, concept of pump power is provided. Using multiple regression method, dimensionless correlation equation of Nusselt number & friction factor are given. It is concluded that the plate chevron angle affect thermal hydraulic performance. Heat transfer increases with increase in chevron angle & vice versa [18].

Sharif Asal et al. (2018) used Computational Fluid Dynamics approach with the Reynolds stress model to investigate the influence of the apex angle on the thermal and hydraulic features of triangular cross-corrugated heat exchangers. The Reynolds number was varied from 310 to 2064. The numerical results varied by 5% than experimental results. On increasing the apex angle, pressure forces increase which lead to pressure drop along with heat exchanger coefficient. It is concluded that on increasing apex angle from 45^0 to 150^0, vorticity magnitude & pressure forces along the direction of flow increase which lead to higher heat transfer [19].

Khavin G. (2018) studied about the different height of corrugation for heat exchangers with a circular plate. For designing of such heat exchanger, use of plates with different corrugation heights along hot and cold side can prove to be very helpful. Due to this design, resistance to contamination increases [20].

From the study on various literature reviews it can be observed that the heat transfer is the most important parameter to be measured as the thermal performance and heat transfer efficiency of the plate heat exchanger. Among all types of heat exchangers corrugated heat exchangers are found to have highest rate of heat transfer. In recent years, corrugated channels are a popular heat transfer augmentation device used in different heat- exchanging channels such as the internal cooling channels in gas turbine blades. Using this technique a significant enhancement in the flow mixing between hotter fluid layers near channel wall and cooler fluid layers in core region is demonstrated. In other words, the flow disturbance caused by the corrugations greatly increases the production of turbulent kinetic energy, which enhances turbulent heat transfer in the channel. Corrugated plate heat exchangers are used to transfer heat but its manufacturing is very typical as compared to the plate heat exchangers. Besides that, many studies have been done using both experimental and numerical analysis based on working fluid. As summarized above, many researchers have investigated various aspects of corrugated channel geometry to enhance heat transfer. However, the heat transfer performances of many other corrugated shapes have not yet been reported. In addition, very little studies have been done to investigate the impact of geometrical parameters through the corrugated channels.

III. GEOMETRY SETUP AND MODELLING

Figure 2. Shows the geometry of the corrugated channel which will be denoted here as the test section while Figure 3, 4 and 5 shows the computational model of different corrugated channel. The total length of the channel is Ltotal=700mm. The length of the test section is L2=200mm, with an upstream rectangular section of L1=400mm upstream to ensure a fully developed flow at the leading edge. The downstream section has a length of L3=100mm which is used to prevent the occurrence of adverse pressure effects caused by reversed flow which might at the trailing section. The channel height (H) is 10mm while the channel width (W) is 50mm. The geometry configuration was achieved by using Ansys design modeler.

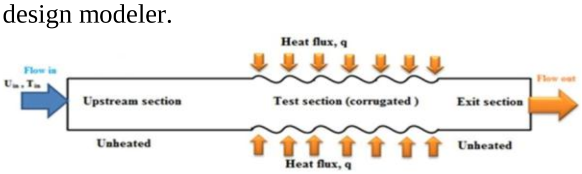

Figure 2. Physical model of the present study

The semi-circular corrugated channel width (w) =5 mm with fixed pitch (p= 15mm).

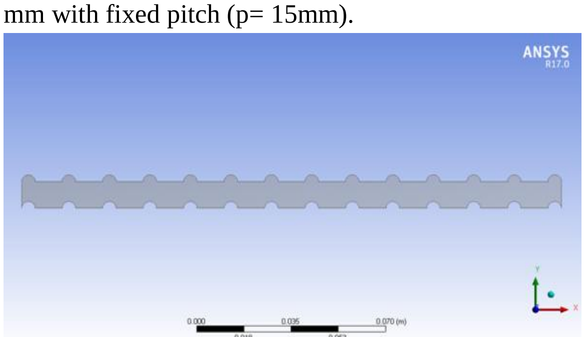

Figure 3. Semi-circular corrugated channel (test section).

The equilateral triangular corrugated channel width (w) =4.75 mm with fixed pitch (p= 15mm).

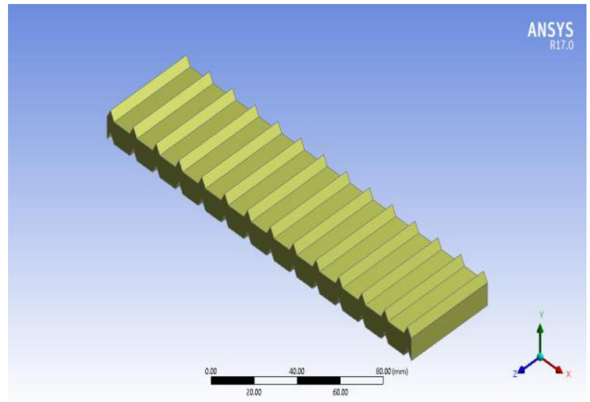

Figure 4. Triangular corrugated channel (test section)

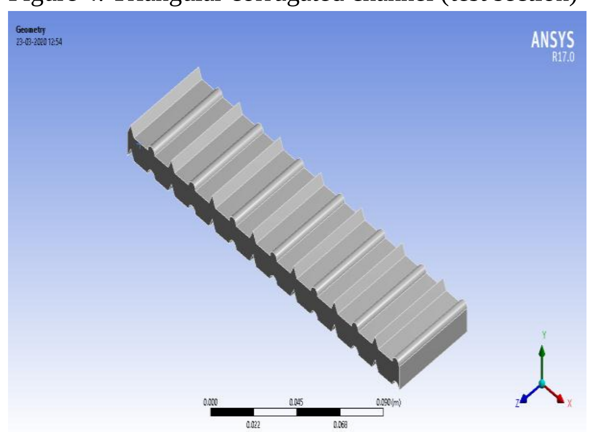

Figure5. Mixed (Semi-circular + Triangular) corrugated channel (test section)

Mesh generation was performed by using Ansys Meshing modular.

Table 1. Meshing Details

MODEL TYPE	NODES	ELEMENTS	MESH TYPE
Semi circular	132804	118400	Tetrahedral & quad core
Equilateral triangle	118250	102459	Tetrahedral & quardcore
Mixed (Semi-circular+Equilateral triangle)	135256	118716	Tetrahedral & quad core

In this study, the finite volume method (FVM) is used for discretization of governing equations. The upwind scheme is used to discretize the convection terms of governing equations, while diffusion terms were discretized using the central differencing scheme. The SIMPLE algorithm was used, for coupling of the velocity and pressure equations, to determine pressure field. The second order upwind scheme is applied and standard k-ε turbulent model with standard wall function is selected.

In this study, the single-phase approach has been used in the modeling of nanofluid. Therefore, the two-dimensional governing for steady, incompressible flow in terms of Cartesian coordinates are [19]:

Continuity equation:

$$\frac{\partial}{\partial x}(\rho u)+\frac{\partial}{\partial y}(\rho v)=0$$

U-momentum equation:

$$\frac{\partial}{\partial x}(\rho uu)+\frac{\partial}{\partial y}(\rho uv) = -\frac{\partial P}{\partial x}+\frac{\partial}{\partial x}\left[(\mu+\mu_t)\frac{\partial u}{\partial x}\right] +\frac{\partial}{\partial y}\left[(\mu+\mu_t)\frac{\partial u}{\partial y}\right] +\frac{\partial}{\partial x}\left[(\mu+\mu_t)\frac{\partial u}{\partial x}-\frac{2}{3}\rho k\right] +\frac{\partial}{\partial y}\left[(\mu+\mu_t)\frac{\partial v}{\partial x}\right]$$

V-momentum equation:

$$\frac{\partial}{\partial x}(\rho uv)+\frac{\partial}{\partial y}(\rho vv) = -\frac{\partial P}{\partial y}+\frac{\partial}{\partial x}\left[(\mu+\mu_t)\frac{\partial v}{\partial x}\right] +\frac{\partial}{\partial y}\left[(\mu+\mu_t)\frac{\partial v}{\partial y}\right] +\frac{\partial}{\partial y}\left[(\mu+\mu_t)\frac{\partial v}{\partial y}-\frac{2}{3}\rho k\right] +\frac{\partial}{\partial x}\left[(\mu+\mu_t)\frac{\partial u}{\partial y}\right]$$

Energy equation:

$$\frac{\partial}{\partial x}(\rho uT)+\frac{\partial}{\partial y}(\rho vT) = \frac{\partial}{\partial x}\left[\left(\frac{K}{c_p}+\frac{\mu_t}{Pr_t}\right)\frac{\partial T}{\partial x}\right] +\frac{\partial}{\partial y}\left[\left(\frac{K}{c_p}+\frac{\mu_t}{Pr_t}\right)\frac{\partial T}{\partial y}\right]$$

Table 2 Thermophysical properties of nanoparticle and base fluid at T=300 K

Material	Density(Kg /m^3)	Specific heat(J/Kg-K)	Thermal conductivity (W/m-K)	Dynamic viscosity(N-s/m^2)
Water	998.2	4182	0.6	0.001
Al_2O_3	3600	765	36	-------

Here the effective properties of the Al_2O_3/water nanofluid are defined as follows:

The below equations for determining density, thermal conductivity, specific heat and viscosity of nanofluids.

The density and heat capacity of nanofluid are given by:

$$\rho_{nf}=\emptyset_p\rho_p+\left(1-\emptyset_p\right)\rho_{bf}$$

$$\left(\rho C_p\right)_{nf}=(1-\emptyset_p)(\rho C_p)_{bf}+\emptyset_p(\rho C_p)_p$$

In order to compute the effective thermal conductivity by using nanoparticles in corrugated channel, the effect of Brownian motion will be taken into consideration by utilizing the empirical correlation below:

$$K_{effective}=K_{static}+K_{Brownian}$$

$$K_{static}=K_{bf}\left\{\frac{K_p+2K_{bf}-2\emptyset_p(K_{bf}-K_p)}{K_p+2K_{bf}+\emptyset_p(K_{bf}-K_p)}\right\}$$

$$K_{Brownian}=5\times10^4\beta\,\emptyset\,(\rho C_p)_{bf}\sqrt{\frac{kT}{\rho_p d_p}}\,f(T,\emptyset)$$

Where, Boltzmann constant, $k=1.381\times10^{-23}$ *J/K*

$$f(T,\emptyset)=(2.8217\times10^{-2}\,\emptyset+3.917\times10^{-3})\frac{T}{T_0} +(-3.0669\times10^{-2}\,\emptyset-3.9112\times10^{-3})$$

The effective dynamic viscosity of nanofluid is:

$$\mu_{nf}=\mu_{bf}\,A_1e^{A_2\,\emptyset}$$

Where A_1=0.983 and A_2= 12.959

Table 3. Thermophysical properties of distilled water and Al_2O_3–water nanofluid with different volume fractions at 300 K.

Ø(%)	ρ($\frac{Kg}{m^3}$)	$c_p(\frac{J}{Kg-K})$	K($\frac{W}{m-K}$)	μ($\frac{N-s}{m^2}$)
0	998.2	4182	0.6	0.001
2	1050.236	3947.74	0.6355	0.001273
4	1102.272	3735.60	0.6718	0.001650
6	1154.308	3542.59	0.7083	0.002139

Computational domains and boundary conditions were applied at the semicircle corrugated channel which included velocity inlet condition and temperature of 300K, pressure outlet condition while slip velocity was ignored. Additionally, there was uniform heat flux (q=10 KW/m^2) on the corrugated walls whereas adiabatic condition applied for the remaining walls which are straight. The specific thermal conditions for the complex flow field as well as the boundary conditions can be illustrated as below.

The boundary conditions at the inlet:

$$u = u_{in}, v = w = 0, T_{in} = 300K$$

The boundary conditions at the wall:

$$u = v = w = 0, q = q_{wall}$$

The boundary conditions at the outlet:

$$\frac{\partial T_f}{\partial x} = 0, \frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial w}{\partial x} = 0$$

In current study, fully developed for the properties are assumed at the outlet.

The hydraulic diameter is computed based on cross section area (A_{cross}) and the perimeter of wetted (P)

$$D_h = 4\,\frac{A_{cross}}{P} = H_{max} + H_{min}$$

At the inlet, the water velocity can be set for different values according to the Reynolds number. The flow is simulated at five Reynolds number values which are 10000, 15000, 20000, 25000 and 30000. Top and bottom walls for the test section are subjected to a constant heat flux of $10kW/m^2$ while adiabatic boundary condition is applied to the remaining walls. The inlet temperature of the working fluid is T_{in}=300K. Al_2O_3-water nanofluid is set to flow steadily over the test section at Reynolds numbers stated above.

IV. RESULTS AND DISCUSSIONS

4.1 Validation of semi-circular corrugated channel simulation results using water as a working fluid

For validation, In order to numerically validate the CFD model of heat exchanger for initial case we considered water as a working fluid. Here water flows as different Reynolds number inside the semi-circular corrugated heat exchanger and the results obtained by simulation for semi-corrugated channel which included Nusselt number (Nu) are compared with R K Ajeel et.al (2017) results [17]:

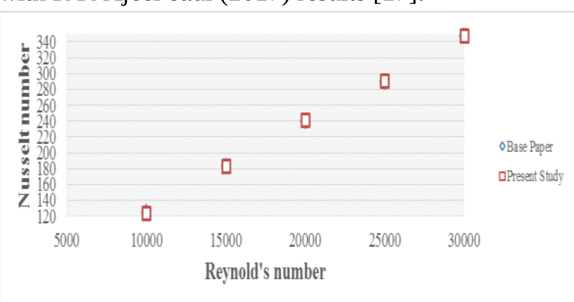

Figure 6. shows the values of Nusselt number calculated from the CFD modeling compared with the values obtained from the analysis performed by R K Ajeel et.al (2017) for semicircle corrugated channel using water as a working fluid.

From the above analysis it is found that the value of Nusselt number calculated from CFD analysis is close to the value of Nusselt number obtained from the base paper. So here we can say that the CFD model of corrugated channel is correct.

4.2 The effect of different types of corrugated channels on the flow and thermal characteristics in the corrugated heat exchanger

The turbulent forced convection of Al_2O_3-water nanofluid in different corrugated channel were investigated. The CFD results obtained due to the impact of the different models, ϕ and Reynold's number have been displayed in terms of Nusselt number.

4.2.1 Effect of Triangular corrugated channel on the flow and thermal characteristics in the corrugated heat exchanger

In this the 3-D turbulent forced convective flow of Al_2O_3 –water nanofluid for triangular corrugated channels over *Re* ranging from $10{,}000 \leq Re \leq 30{,}000$ were examined under constant heat flux while ϕ ranged from 0 to 0.06

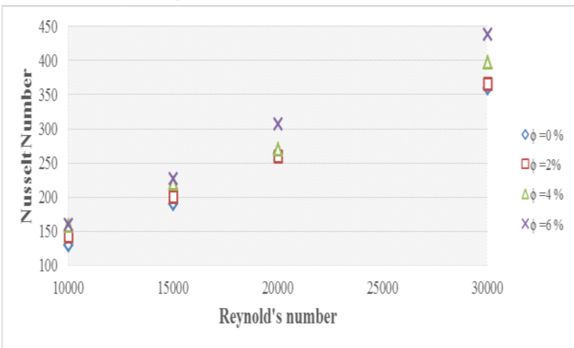

Figure 7. Shows the value of Nusselt number calculated through CFD analysis at different Reynold's number for Triangular corrugated channel under constant heat flux while ϕ ranged from 0 to 0.06

4.2.2 Effect of Mixed corrugated channel on the flow and thermal characteristics in the corrugated heat exchanger

In this the 3-D turbulent forced convective flow of Al_2O_3 –water nanofluid for mixed corrugated channels over *Re* ranging from $10{,}000 \leq Re \leq 30{,}000$ were examined under constant heat flux while ϕ ranged from 0 to 0.06.

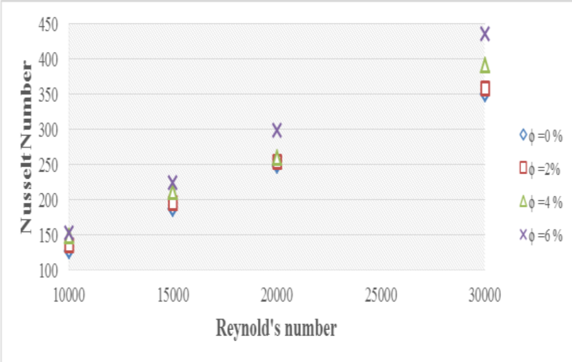

Figure 8. Shows the value of Nusselt number calculated through CFD analysis at different Reynold's number for mixed corrugated channel under constant heat flux while ϕ ranged from 0 to 0.06.

Table 4. Comparison of Nusselt number calculated through CFD analysis at different corrugated channel under constant heat flux at different Reynold's number while ϕ ranged from 0 to 0.06

Reynold's number	Corrugated Shape	Nusselt number			
		ϕ =0 %	ϕ =2%	ϕ =4 %	ϕ =6 %
Reynold's number=10,000	Semi-Circular	125	130	140	150
	Triangular	129.16	142.97	158.88	159.35
	Mixed	128.22	136.67	149	153.85
Reynold's number=15,000	Semi-Circular	183	190	200	218
	Triangular	190.88	200.27	220.96	227.04
	Mixed	187.24	196.61	212.2	224.82
Reynold's number=20,000	Semi-Circular	240	248	252	290
	Triangular	263.24	259.53	270.7	307.63
	Mixed	249.62	254.67	259.91	298.92
Reynold's number=30,000	Semi-Circular	348	355	380	425
	Triangular	360.03	366.86	398.34	438.3
	Mixed	350.73	359.62	391.63	435.68

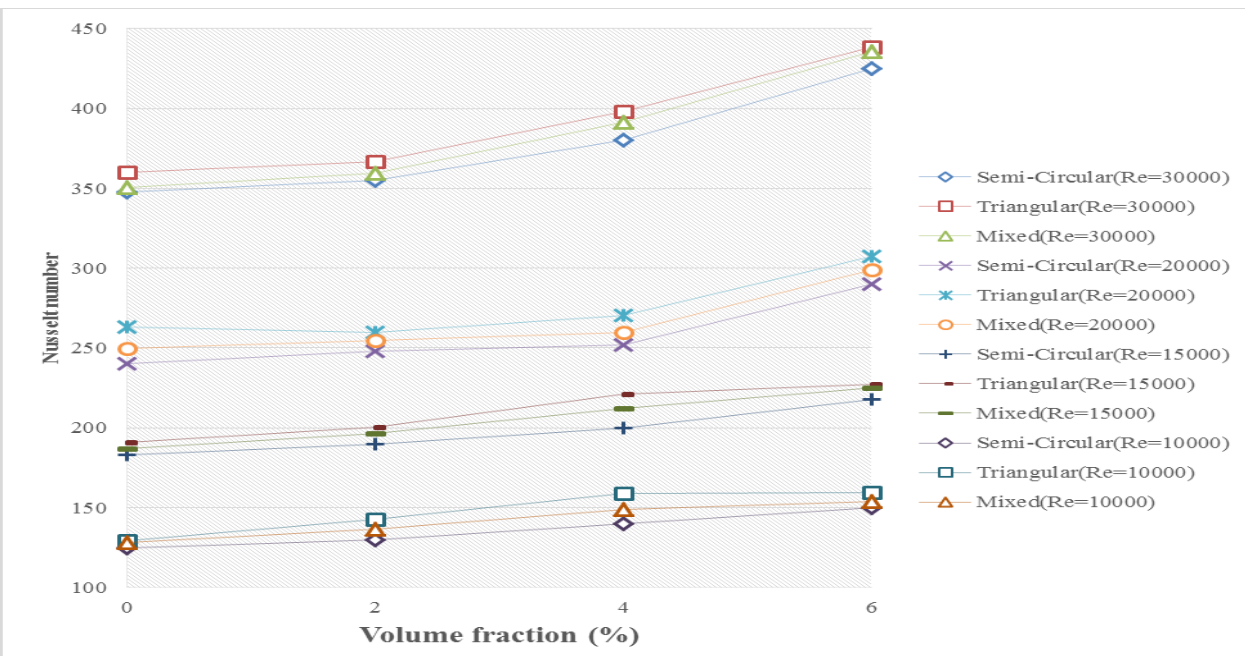

Figure 9. Comparison of Nusselt number calculated through CFD analysis at different corrugated channel under constant heat flux at different Reynold's number while ϕ ranged from 0 to 0.06.

V. CONCLUSIONS

In this paper, the 3-D turbulent forced convective flow of Al_2O_3 –water nanofluid for different types of corrugated channels over Re ranging from $10{,}000 \le Re \le 30{,}000$ was examined. Three shapes of corrugated channel namely Semi-circular, Triangular and Mixed shape (Semi-circular + Triangular) were tested under constant heat flux while ϕ ranged from 0 to 0.06 and the diameter of the nanoparticles was 20 nm. The conclusions of this CFD study are as follows:

- For all channels average Nusselt number increased with increasing ϕ *and Re.*
- It is observed that Triangular channel has the highest average Nusselt number followed by the mixed shape (Semi-circular + Triangular) and Semi-circular corrugated channel due to better mixing of working fluids caused by the corrugation that acted like a swirls promoter for the fluid layers.
- As Re increased, heat transfer also increased. The explanation for this is associated with increased velocity due to the increasing Reynolds number, which improves and promotes thermal exchange between the layers of fluid. Of all corrugated channels, the Triangular channel was the best at processing Nusselt numbers, followed by the mixed shape (Semi-circular + Triangular) and Semi-circular corrugated channel with the lowest results.
- Nusselt number which indicates the optimum configuration is Triangular channel at Reynolds number 30000 and volume fraction 6%.
- Triangular channel has the average Nusselt number greater than 1.3 times of mixed shape (Semi-circular + Triangular) and 1.7 times of Semi-circular corrugated channel.

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