THERMAL ANALYSIS OF CORRUGATED PLATE HEAT EXCHANGER BY USING ANSYS SOFTWARE THROUGH FEA METHOD

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Abstract – *The effects of nanofluid that is Al2O3 in water in a counter flow corrugated plate heat exchanger were investigated through an experiment by Shive Dayal Pandey and V. K. Nema. It had been discovered that the heat transfer characteristics improve with the decrease in nanofluid concentration. Power consumption and heat transfer rates were lower for water compared to the nanofluid. For a given heat load the nanofluid needed the lower rate of flow, however, suffered a higher pressure drop than that for water. For a given pumping power additional heat might be removed by the nanofluids relative to water, though the most heat transfer rate was found with very cheap concentration of nanofluids. Correlation equations were obtained for the Nusselt number and the Friction factor for both water and the nanofluid. N. Putra, W. Roetzel and S.K. Das showed that natural convection heat transfer by applying nanofluid of Al2O3/water and CuO/water was not up to the mark as compared to the base fluid. Also, they found that this might result in too many factors like the sinking of nanoparticles and speed distinction between nanoparticles and base fluid. Considering these facts, I concluded that because of additional drop-in pressure just in case of nanofluid, there is additional pumping power needed thus in place of nanofluid solely water is being thought-about for experiment. There is a passive methodology to extend the heat transfer by increasing the amount of heat transfer that is tested here through ANSYS FEA methodology. Corrugated plate heat exchanger with the counterflow has been studied along with the arrangement of the hot fluid (water) on either side of the cold fluid channel for various volume flow rates of the cold fluid (water) along with 30-degree of corrugation angle and 20-degree of corrugation angle. The setup was run through ANSYS Fluent and the reports were observed and compared for the optimum results.*

Key Words: Heat Exchanger, Nanofluid, Heat Transfer, Thermal Energy, Effectiveness, Passive Methods, Corrugated Plate.

1. INTRODUCTION

Heat exchanger is a device which is used to transfer heat between two fluids at different temperature either by direct contact or indirect contact with the help of a separating wall made up of a highly conductive material. These are widely used in various industries, vehicles, engines etc and domestic applications as well. Global warming is a phenomenon which has made us realize the need for optimum usage of everything that we have. Every researcher is putting their efforts in the field of energy-saving and finding some ways that we can follow to make it reality. Utilizing thermal energy in an efficient manner is our prime focus here. Heat exchangers have already been one of the most talked topics among the researchers so a lot of work has been done in this field already and this research work is also aimed to acknowledge their efforts in this field.

There are two ways to improve the rate of heat transfer by active methods and passive methods. Active methods need some energy input to achieve higher efficiency and it has limitations to do it whereas passive methods don't need any energy input so researchers are more concerned about these methods. It has also many constraints but still lots of scope to achieve higher efficiency for heat exchanging process. Technical constraints which are responsible for the improvement in the rate of heat transfer are surface area and overall heat transfer coefficient. Our focus is to study the relations and key features which can improve the rate of heat transfer by improving the heat transfer coefficient.

2. Literature Survey

The performance of heat exchangers may be improved by heat transfer improvement techniques for playing a particular heat-transfer duty. Improvement of the heat transfer allows the scale of the heat exchanger to be considerably shrunken. Many researchers have done their experiments and analysis over the factors which are responsible for improvement of heat transfer rate and the same has been discussed and reviewed here in this section.

Fluids, like water and engine oil, have poor heat transfer performance and so, high compactness and effectiveness of heat transfer systems are necessary to realize the desired heat transfer. Among the efforts for improvement of heat transfer, the applying of additives to liquids is critical **[1,2]**. The word nanofluid can be defined as a suspension of nano-sized solid particles in typical fluids; such fluids embrace increased heat transfer characteristics, like convective heat transfer. This hefty increase in heat transfer could result in remittent energy expenditure and raw material-input, also reduces the size of the apparatus and consequently reduces expenses and exaggerates system potency.

Primarily **Choi [3]** used the nanometer-sized particles in the typical fluids and showed improvement in heat transfer characteristics.

The majority of experimental studies on the applying of nanofluids at single-phase forced convective heat transfer are involved with streamline flow principally in outwardly heated circular tubes or micro-channels. Among the varied sorts of nanofluids used, metal oxides with partial volume concentrations are the foremost common, most likely because of their cheaper price **[4]**.

Studies on heat transfer of suspension of metal oxides in fluids were restricted to suspensions with millimeter or micron-sized particles. Such giant particles could cause severe issues in heat transfer instrumentation. specifically, giant particles tend to quickly settle out of suspension and thereby passing through micro channels causes severe wearing and increase the pressure drop significantly **[5]**.

Experimental investigation performed on the suspension of 4 percentage volume & 35 nm sized particles of CuO in ethylene glycol resulted in high improvement in the thermal physical phenomenon **[6]**.

The investigation of 35 nm-sized Cu/deionized water nanofluid flowing during a tube with constant wall heat flux showed the improvement in Nusselt number compared to pure water for identical rate of flow by increasing the volume fraction of nanoparticles from 0.5% to 1.2% **[7]**.

In another study, Al2O3/water nanofluid heat transfer in streamline flow underneath constant wall heat flux and reported a rise in the nanofluid heat transfer constant with Reynold's number and nanoparticle concentration significantly within the entrance region in which thermal developing length for nanofluid was larger than in case of pure water. **Wen and dingdong [8]**.

The numerical investigation for heat transfer of Al2O3/ethylene glycol and Al2O3/water nanofluids during a radial flow system showed improvement in heat transfer rate. Conjointly, researchers showed that wall shear stress improved with concentration and Reynold's number **[9]**.

It had been reported by **Putra et al. [10]**, that natural convection heat transfer by applying nanofluid of Al2O3/water and CuO/water was not up to the mark as compared to the base fluid and all over that this might result in too many factors like the sinking of nanoparticles and speed distinction between nanoparticles and base fluid.

Thermo-physical properties of CuO nanofluid were measured and investigated the performance of nanofluid and compared it to the base fluid (i.e., water) by **Pantzali et al. [11]**.

The performance of a nanofluid containing carbon nanotubes resulted in improvement of the heat transfer which was observed as 3.5 times higher than the base fluid, which was studied by **Ding et al. [12]**.

Experimentation with Al2O3 and TiO2 as nanofluids along with a shell and tube exchanging device **[13]** has shown that there's an optimum volume concentration for the maximum value of overall heat transfer coefficient. This optimum value of concentration differs fluid to fluid.

Thermo-physical properties are altered by the addition of nanoparticles, and fluid viscosity and the nature of flow, are the crucial parameters for determining the effectualness of nanofluid **[14]**.

Further, the heat transfer rate and heat transfer constant during a PHE are above those use base fluid **[15]**.

The work on two nanofluids specifically, Al2O3/EG and CuO/EG having concentrations 0.1, 0.5, and 1 percent respectively **[16]** have shown that the heat transfer constant or coefficient will increase with temperature and concentration each.

Al2O3/water of 6 volume percentage concentration in a fluid with Reynold's number greater than 100 but lesser than 500, temperature greater than 20˚C but lesser than 40˚C resulted that the performance of the PHE at a given rate of flow didn't improve with the fluid **[17]**.

Shive Dayal Pandey, V.K. Nema [18]: The effects of nanofluid that is Al2O3 in water two, three and four vol.% and water as coolants on heat transfer, resistance losses, and loss of available energy during a counter flow corrugated plate device were by experimentation investigated. The desired properties of the nanofluid were measured. It had been determined that the heat transfer characteristics improve with the increase in Reynolds- and Peclet-number and with the decrease in nanofluid concentration. For a given heat load, the desired pumping power exaggerated with the increase in nanofluid concentration. Each power consumption and heat transfer rates were lower for water as compared to the nanofluid for flow rates of 2–5 LPM for hot and cold fluids. Further, for a given heat load the nanofluid needed a lower rate of flow however suffered a higher pressure drop than that for water.

3. Observations

Shive Dayal Pandey and V. K. Nema improved rate of heat transfer by using Nano Fluid (Al2O3 in Water) but having more pressure drop consequently more power consumption and more wear friction factor.

N. Putra, W. and Roetzel, S.K. Das showed that natural convection heat transfer by applying nanofluid of Al2O3/water and CuO/water was not up to the mark as compared to the base fluid.

Also, this might result in too many factors like the sinking of nanoparticles and speed distinction between nanoparticles and base fluid.

Mainly researches on heat transfer coefficients are found for an unvarying wall temperature or an unvarying heat flux. The condition of constant wall temperature is idealized in heat exchangers with natural action like condensers. The boundary condition of constant heat flux finds application in electrically heated tubes and nuclear fuel components. However, the case of liquid-liquid heat exchange has not been studied well. In case the use of nanofluid for increasing the heat transfer rate by enhancement of the thermal conductivity of fluid has given better result, but when observed closely then found that there is more drop in fluid pressure from the inlet to outlet that means there is greater resistance to flow due to increase in friction which will damage the walls of the heat exchanger, so in the long term it will not be good enough where we need to keep our system wear less also. In current work fluid to fluid heat exchange is taken into thought and analysed. Overall heat transfer coefficient is taken into consideration based on the ANSYS result and our calculation. ANSYS results were used to determine effectiveness, overall heat transfer coefficient, and also LMTD for counter flow Heat Exchanger. Results are compared and validated by the CFD analysis of the same with the help of ANSYS Fluent software between nanofluid and water with the changing corrugation angle as 30 degrees and 20 degrees.

3. Highlights

- Collected the CFD data pertaining to heat transfer and fluid flow in a corrugated plate heat exchanger with different corrugation angle.
- Flow rate of Cold Fluid varies from 2 to 5 LPM
- Constant flow rate of Hot Fluid 2 LPM
- Counter Flow arrangement of fluid flow
- Cold Fluid flowing through the inner channel and
- Hot Fluid flowing through the annulus
- Once Corrugation Angle kept 30 Degree and then 20 Degree

4. Methodology

This lesson deals with the process methodology for calculating effectiveness, LMTD and overall heat transfer coefficient formulas used in calculation given in this part. Mathematical calculation and value gained for Corrugated Steel Plate counterflow, with corrugation angle of 30 degree and 20 degree have been discussed here in this lesson. The present work is to identify the effect of corrugated plate in place of simple plane steel plate for heat transfer with simple water, not with nanofluid which was used for heat transfer by Shive Dayal Pandey, V.K. Nema [18] to increase thermal conductivity. And then we will compare the results of nanofluid and water in counter-flow heat exchanger.

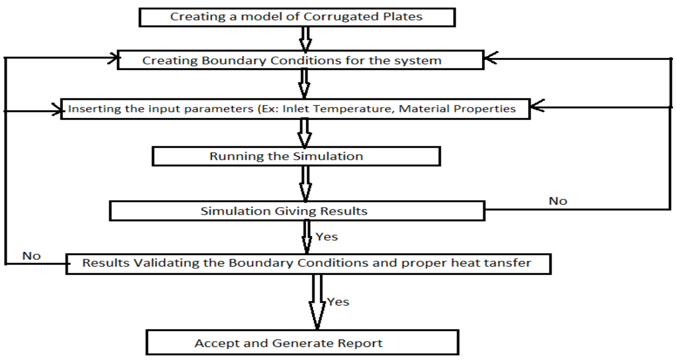

Fig 4.1 Methodology Flow Chart

The complete methodology includes some steps which are as follows.

a. Designing 4 corrugated plates of 30-degree corrugation angle and 20-degree corrugation angle having plate thickness 1mm and 20 mm apart from each other, with the help of Solidworks Designing Software.

SN	Parameter	Value
1	Length of the test section	L 350 mm
2	Width of the test section	W 80 mm
3	Gap between two corrugated plates	H 20 mm
4	Total height of the test section	120 mm
5	Developed length of the corrugated plate	410 mm
6	Corrugation angle	30

Table 4.1 Design Parameters of Corrugated Plate Heat Exchanger

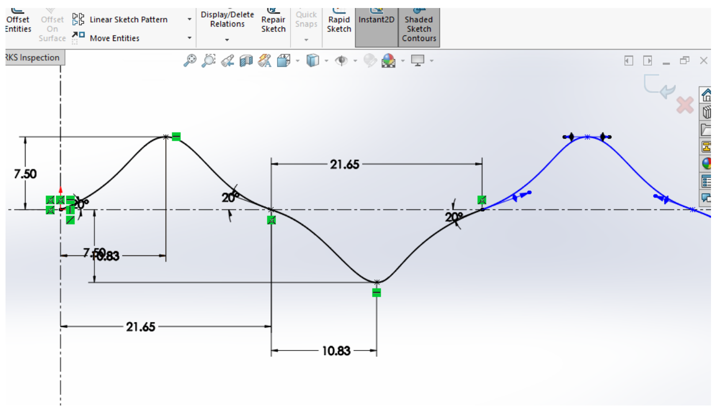

Fig 4.2 20-Degree Corrugation Angle

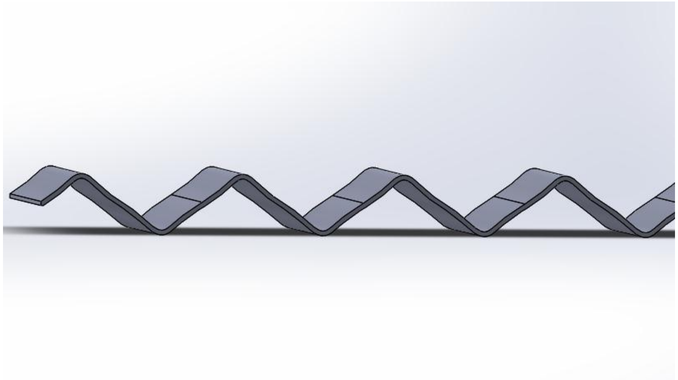

Fig 4.3 Base Plate with 30 Degree Corrugation Angle

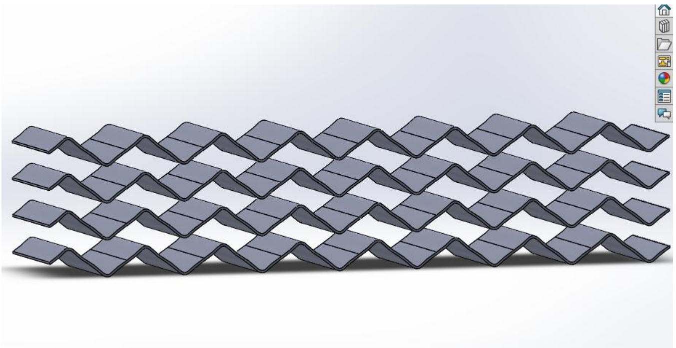

Fig 4.4 Set of Corrugated Plates with 30 Degree Corrugation Angle

b. Imported the Solidworks model in ANSYS and then designed the outside box and enclosed all plates inside that box.

c. Named all the parts of that model and then given boundary conditions with the help of ANSYS Fluent software.

d. Then initiated calculation for 200 iterations and then with the help of Reports>Surface Integrals commands, found all results for 30-degree corrugation angle and then same values for 20-degree corrugation angle.

e. Used mathematical relations to calculate the performance parameters of all the heat exchangers and then compared all.

5. Experimental Procedure

1) A CFD analysis on ANSYS-Fluent (2018.2) software is performed with different volume flow rates of the cold fluid in counterflow arrangement to the hot fluid. The volume flow rate of the hot fluid maintained at 2LPM and the volume flow rate of cold water varied with values of 2, 2.5, 3, 3.5, 4, 4.5 and 5LPM (Litre Per Minute) respectively.

2) The same temperature of hot and the cold fluid inlet was provided for every simulation and results are checked for cold and the hot fluid outlet temperatures. This same simulation was run for 30-degree corrugation angle and 20-degree corrugation angle. As it was found that outlet temperatures are coming out to be different for all the simulations so the research moved forward.

3) After giving cold domain for inlet conditions as cold water at atmospheric temperature 303 K every time and atmospheric pressure.

4) After that, the hot fluid in the upper hot domain and lower hot domain given same 350 k temperature as inlet condition in opposite direction to the cold fluid that is counter-flow arrangement kept all time and then the setup was run for taking the readings.

5) Firstly, corrugation angle of steel plates was taken as 30 degrees and flow rate of the hot fluid is maintained at 2LPM and flow rate of the cold fluid varied and for every flow rate of the cold fluid started with 2LPM, the setup was run and readings were taken for every simulation and results.

6) After taking the first set readings, the flow rate of the cold fluid was changed to 2.5LPM and when the system reaches the next steady-state another set of readings were taken. Similar procedure was used for taking the readings with flow rates of 3, 3.5, 4, 4.5 and 5 LPM of the cold fluid.

7) For every set of readings, the system is allowed the time to reach steady-state.

8) Similarly, readings are taken and noted for plates with corrugation angle of 20 degrees with flow rate of the hot fluid keeping at 2LPM and varying flow rate of the cold fluid as 2, 2.5, 3, 3.5, 4, 4.5, 5 LPM respectively.

6. Mathematical Calculations

The readings obtained by the ANSYS setup of 30-degree corrugation angle of steel plates, were noted and tabulated and by utilizing those details, calculations were done like the following steps:

For the cold fluid flow rate of 2LPM.

Surface Area of Corrugated Plate, Assumed Rectangular,

$$A\ (m^2) = 410\ mm * 80\ mm = 0.0328\ m^2$$

Total Surface Area of Heat Transfer for the cold fluid,

$$A_S = 2 * 0.0328\ m^2,$$

$$A_s = 0.0656\ m^2$$

Total Heat Transfer Rate, Q (W) from Wall Cold Domain Plate 2 Shadow = Heat Flux * Area

= 8007.904 * 0.0328

= 262.6592512 W

Total Heat Transfer Rate, Q (W) from Wall Cold Domain Plate 3 Shadow = Heat Flux * Area

= 8044.5772* 0.0328

= 263.86213216 W

Total rate of heat transfer to the cold fluid,

Q = 262.6592 + 263.8621 = 526.52138336 W

Wall Temperature for Wall Cold Domain Plate 2 Shadow,

T_s = 326.31457 k

Wall Temperature for Wall Cold Domain Plate 3 Shadow,

T_s = 326.40516 k

Bulk Mean Temperature,

T_b = Average of (Temperature of the cold fluid Inlet & Temperature of the cold fluid Outlet)

T_b = Avg (T_{ci}, T_{ce}) = (303 + 307.1526) ÷ 2 = 305.07633 k

Temperature Difference ($dT = T_s - T_b$) for Wall Cold Domain Plate 2 Shadow = 326.31457 – 305.07633

= 21.23824 k

Similarly,

dT for Wall Cold Domain Plate 3 Shadow = 21.32883 k

ΔT_i (Counter Flow) $= T_{hi} - T_{ce}$

= 350 – 307.15266

= 42.84734 k

ΔT_e (Counter Flow) $= T_{he} - T_{ci}$

= 347.64229 – 303

= 44.64229 k

Logarithmic Mean Temperature Difference (LMTD),

$$\theta_m = \frac{\Delta Ti - \Delta Te}{LN\left(\frac{\Delta Ti}{\Delta Te}\right)} = 43.7386 \text{ k}$$

Heat Capacity, for the cold fluid,

mC = 0.033333333 kg/s * 4187 j/kg.k

= 139.5666667

Heat Capacity, for the cold fluid,

mC = 0.033333333 kg/s * 4187 j/kg.k

= 139.5666667

Heat Transfer Rate (Q),

Q $= mC * (T_{ce} - T_{ci})$

= 139.5666667 * (307.15266 – 303)

= 579.572914 W

Overall Heat Transfer Coefficient (U),

U $= Q/(A_s * \theta_m)$ Watts/m^2k

ΔT_{max} = Thi – Tci = 47 k

Maximum Heat Transfer Rate (Q_{max}),

Q_{max} $= mC_{min} * \Delta T_{max}$ W

= 6559.633333 W

Effectiveness (∈),

∈ $= Q/Q_{max}$

= 0.088354468

Flow Velocity (V),

V = (Discharge in m^3/s)/(Domain Flow Area in m^2)

= 0.0000333333/(0.08*0.02)

= 0.020833333 m/s

Peclet Number (Pe),

Pe = Re * Pr

= [$\rho V L_c/\mu$] * [$\mu C/k$]

= $\rho V C . L_c/k$

Where,

Re = $\rho V Lc/\mu$ (Reynold's Number)

Pr = $\mu C/k$ (Prandtl Number)

ρ = density of water,

= 998.2 kg/m^3 (from Ansys)

V = Flow velocity, in m/s

C = Specific Heat of Water, 4182 j/kg

L_c = Characteristic Length of the Flow Channel,

L_c = (4*Flow Area of Channel)/(Wetted Perimeter)

L_c = (4*A)/P

L_c = (4*(0.08*0.02))/(2(0.08+0.02))

= 0.0064/0.2

= 0.032 m

Thermal Conductivity of Steel (k),

k = 16.27 w/mk

Peclet Number (Pe),

Pe = (998.2*0.02083333*4182*0.032)/16.27

= 171.04985

Friction Factor (f),

f = 4 * Skin Friction Coefficient

= 0.051307176

Similarly, all the results have been arranged in table form here, for the fixed hot fluid flow rate of 2 LPM and varying the rate of flow of cold fluid as 2, 2.5, 3, 3.5, 4, 4.5 and 5 LPM respectively.

Volume Flow rate of the cold fluid (LPM)	Surface Area of Corrugated Plate, Assumed Rectangular, A (m^2) (410 mm * 80 mm)	Total Surface Area of Heat Transfer for the cold fluid, A_s	Heat Flux (w/m²)	
			Wall Cold Domain Plate 2 Shadow	Wall Cold Domain Plate 3 Shadow
2	0.0328	0.0656	8007.904	8044.5772
2.5	0.0328	0.0656	9222.3905	8631.0543
3	0.0328	0.0656	10223.734	9049.7345
3.5	0.0328	0.0656	11140.362	9400.4383
4	0.0328	0.0656	11979.312	9700.2765
4.5	0.0328	0.0656	9874.5406	9993.6528
5	0.0328	0.0656	10136.705	10236.88

Table 6.1 Calculation of Heat Transfer Rate

Volume Flow rate of the cold fluid (LPM)	Total Heat Transfer Rate, Q (W)		Total Rate of Heat Transfer, Q (W)	Overall Heat Transfer Coefficient, U (Watts/m²k)
	Wall Cold Domain Plate 2 Shadow	Wall Cold Domain Plate 3 Shadow	The cold fluid	The cold fluid
2	262.6592512	263.8621322	526.5213834	183.5044222
2.5	302.4944084	283.098581	585.5929894	203.181263
3	335.3384752	296.8312916	632.1697668	218.6136113
3.5	365.4038736	308.3343762	673.7382498	232.5419938
4	392.9214336	318.1690692	711.0905028	245.0458304
4.5	323.8849317	327.7918118	651.6767435	224.572165
5	332.483924	335.769664	668.253588	229.735524

Table 6.2 Calculation of Heat Transfer Rate

Volume Flow rate of the cold fluid (LPM)	Temperature Difference (dT= Ts – Tb)		ΔTi (Counter Flow)	ΔTe (Counter Flow)	Logarithmic Mean Temperature Difference (LMTD)
	Wall Cold Domain Plate 2 Shadow	Wall Cold Domain Plate 3 Shadow	$Th_i - Tc_e$	$Th_e - Tc_i$	$\theta m = \frac{\Delta Ti - \Delta Te}{LN(\frac{\Delta Ti}{\Delta Te})}$
2	21.23824	21.32883	42.84734	44.64229	43.73867673
2.5	21.211905	19.886355	43.19377	44.6842	43.93477167
3	21.244625	18.750185	43.46649	44.70153	44.08112648
3.5	21.18937	17.77576	43.64126	44.6945	44.16578693
4	21.18748	16.9299	43.78646	44.68817	44.23578329
4.5	16.375525	16.482445	44.43059	44.04129	44.23565449
5	15.762785	15.902375	44.71971	43.96521	44.34139014

Table 6.3 Calculation of Various Temperatures

Volume Flow rate of the cold fluid (LPM)	Heat Capacity, mC		Heat Transfer Rate (Q)
	The cold fluid	The hot fluid	The cold fluid
2	139.5666667	139.5666667	579.572914
2.5	174.4583333	139.5666667	664.0285421
3	209.35	139.5666667	739.7403185
3.5	244.2416667	139.5666667	820.3442555
4	279.1333333	139.5666667	897.006132
4.5	314.025	139.5666667	806.8589753
5	348.9166667	139.5666667	795.6311858

Table 6.4 Heat Capacity and Rate of Heat Transfer

Volume Flow rate of the cold fluid (LPM)	Heat Transfer Rate (Q)	ΔT_{max}	Maximum Heat Transfer Rate, Q_{max} (W)	Effectiveness
	The cold fluid	$Th_i - Tc_i$		$\epsilon = \frac{Q}{Qmax}$
2	579.572914	47	6559.633333	0.0883545
2.5	664.0285421	47	6559.633333	0.1012295
3	739.7403185	47	6559.633333	0.1127716
3.5	820.3442555	47	6559.633333	0.1250595
4	897.006132	47	6559.633333	0.1367464
4.5	806.8589753	47	6559.633333	0.1230037
5	795.6311858	47	6559.633333	0.121292

Table 6.5 Calculation of Effectiveness of Heat Exchanger

Volume Flow rate of the cold fluid (LPM)	Flow Velocity, V (m/s)	Peclet Number, Pe	Friction Factor, f = 4*Skin Friction Coefficient
	(Discharge in m³/s)/(Domain Flow Area in m²)	$\frac{\rho VC.Lc}{k}$	
2	0.020833333	171.0498832	0.051307176
2.5	0.026041667	213.812354	0.07377554
3	0.03125	256.5748248	0.099016276
3.5	0.036458333	299.3372956	0.126969712
4	0.041666667	342.0997664	0.1576103
4.5	0.046875	384.8622372	0.192070332
5	0.052083333	427.6247081	0.228850672

Table 6.6 Calculation of 'Pe' and 'f'

Similarly, all the results have been arranged in table form here, for the plates of 20-degree corrugation angle with the fixed hot fluid flow rate of 2 LPM and varying the rate of flow of cold fluid as 2, 2.5, 3, 3.5, 4, 4.5 and 5 LPM respectively.

Volume Flow rate of the cold fluid (LPM)	Surface Area of Corrugated Plate, Assumed Rectangular, A (m^2) (410 mm * 80 mm)	Total Surface Area of Heat Transfer for the cold fluid, As	Heat Flux (w/m^2)	
			Wall Cold Domain Plate 2 Shadow	Wall Cold Domain Plate 3 Shadow
2	0.0328	0.0656	8154.8537	8078.1639
2.5	0.0328	0.0656	8732.4448	8642.2739
3	0.0328	0.0656	9144.0175	9042.8587
3.5	0.0328	0.0656	9511.3509	9400.438
4	0.0328	0.0656	9818.658	9699.6827
4.5	0.0328	0.0656	10085.207	9958.3969
5	0.0328	0.0656	10319.149	10184.262

Table 6.7 Calculation of Heat Transfer Rate

Volume Flow rate of the cold fluid (LPM)	Total Heat Transfer Rate, Q (W)		Total Rate of Heat Transfer, Q (W)	Overall Heat Transfer Coefficient, U (Watts/m^2k)
	Wall Cold Domain Plate 2 Shadow	Wall Cold Domain Plate 3 Shadow	The cold fluid	The cold fluid
2	267.4792014	264.9637759	532.4429773	184.5418811
2.5	286.4241894	283.4665839	569.8907734	196.5977601
3	299.923774	296.6057654	596.5295394	207.0418566
3.5	311.9723095	308.3343664	620.3066759	214.4987685
4	322.0519824	318.1495926	640.201575	221.0101503
4.5	330.7947896	326.6354183	657.4302079	226.6694652
5	338.4680872	334.0437936	672.5118808	230.8425856

Table 6.8 Calculation of Heat Transfer Rate

Volume Flow rate of the cold fluid (LPM)	Temperature Difference (dT= Ts – Tb)		ΔTi (Counter Flow)	ΔTe (Counter Flow)	Logarithmic Mean Temperature Difference (LMTD)
	Wall Cold Domain Plate 2 Shadow	Wall Cold Domain Plate 3 Shadow	$Th_i - Tc_e$	$Th_e - Tc_i$	$\theta m = \frac{\Delta Ti - \Delta Te}{LN\left(\frac{\Delta Ti}{\Delta Te}\right)}$
2	21.333225	21.466385	42.84119	45.14275	43.9819338
2.5	19.97272	20.13398	43.37422	45.0129	44.18849607
3	18.879235	19.066925	43.76341	44.07851	43.92077162
3.5	18.03686	18.24117	44.17124	43.99623	44.0836771
4	17.26584	17.48339	44.38806	43.92697	44.15711377
4.5	16.58339	16.81705	44.56138	43.86701	44.21328625
5	16.126935	16.376755	45.01235	43.81293	44.40994055

Table 6.9 Calculation of Various Temperatures

Volume Flow rate of the cold fluid (LPM)	Heat Capacity, mC		Heat Transfer Rate (Q)
	The cold fluid	The hot fluid	The cold fluid
2	139.5666667	139.5666667	580.431249
2.5	174.4583333	139.5666667	632.5475358
3	209.35	139.5666667	677.5801165
3.5	244.2416667	139.5666667	690.901057
4	279.1333333	139.5666667	729.0795187
4.5	314.025	139.5666667	765.7876455
5	348.9166667	139.5666667	693.5242125

Table 6.10 Heat Capacity and Rate of Heat Transfer

Volume Flow rate of the cold fluid (LPM)	Heat Transfer Rate (Q)	ΔT_{max}	Maximum Heat Transfer Rate, Q_{max} (W)	Effectiveness
	The cold fluid	$Th_i - Tc_i$		$\epsilon = \frac{Q}{Qmax}$
2	580.431249	47	6559.63333	0.08848532
2.5	632.5475358	47	6559.63333	0.09643032
3	677.5801165	47	6559.63333	0.10329543
3.5	690.901057	47	6559.63333	0.10532617
4	729.0795187	47	6559.63333	0.11114638
4.5	765.7876455	47	6559.63333	0.11674245
5	693.5242125	47	6559.63333	0.10572606

Table 6.11 Calculation of Effectiveness of Heat Exchanger

Volume Flow rate of the cold fluid (LPM)	Flow Velocity, V (m/s)	Peclet Number, Pe	Friction Factor, f = 4*Skin Friction Coefficient
	(Discharge in m^3/s)/(Domain Flow Area in m^2)	$\frac{\rho V L.Lc}{k}$	
2	0.020833333	171.0498832	0.050458148
2.5	0.026041667	213.812354	0.072233812
3	0.03125	256.5748248	0.096598248
3.5	0.036458333	299.3372956	0.123913084
4	0.041666667	342.0997664	0.15343964
4.5	0.046875	384.8622372	0.18576196
5	0.052083333	427.6247081	0.220710644

Table 6.12 Calculation of 'Pe' and 'f'

7. Results Comparison and Discussion

CFD Results of Corrugated Plate Heat Exchanger with different volume flow rates of the cold fluid & constant volume flow rate of the hot fluid as 2 LPM.

SN	Volume Flow Rate of The Cold fluid (LPM)	Heat Transfer Rate for 30 Degree Corrugation Angle	Heat Transfer Rate for 20 Degree Corrugation Angle
1	2	579.572914	580.431249
2	2.5	664.0285421	632.5475358
3	3	739.7403185	677.5801165
4	3.5	820.3442555	690.901057
5	4	897.006132	729.0795187
6	4.5	806.8589753	765.7876455
7	5	795.6311858	693.5242125

Table 7.1 Variation of Heat Transfer Rate with Coolant Flow Rate

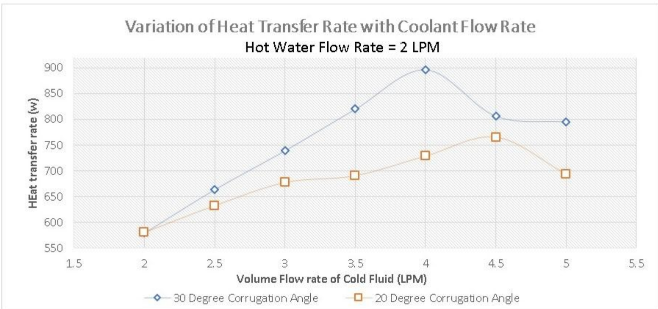

Fig 7.1 Variation of Heat Transfer Rate with Coolant Flow Rate

Based on the data 30-degree corrugation angle is giving the best result for the heat transfer. 30-degree corrugation angle favors the heat transfer till coolant flow rate of 4 LPM and then shows the drop whereas 20-degree corrugation angle shows the steady growth with the coolant flow rate till 4.5 LPM then drops but for optimum result 30-degree corrugation angle with 4 LPM of coolant flow rate is suggested here which gives highest rate of heat transfer of 897.0061 Watt.

SN	Volume Flow Rate of the cold fluid (LPM)	Surface Heat Transfer Coefficient (w/m²k) for 30 Degree Corrugation Angle	Surface Heat Transfer Coefficient (w/m²k) for 20 Degree Corrugation Angle
1	2	212.1259	214.49905
2	2.5	245.85618	240.11064
3	3	273.31394	260.78038
4	3.5	299.53821	279.728
5	4	323.19576	296.55612
6	4.5	307.92197	311.96456
7	5	325.09376	326.1789

Table 7.2 Variation of Heat Transfer Coefficient with Coolant Flow Rate

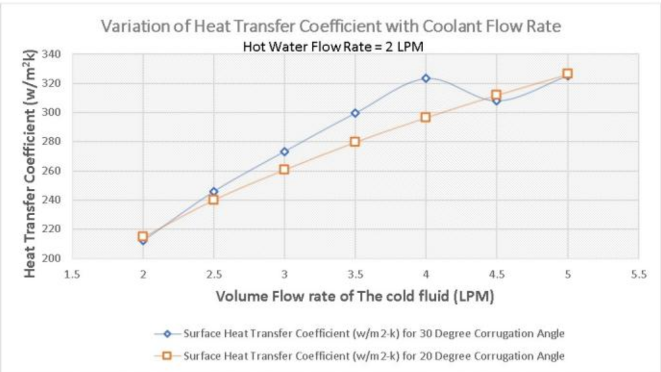

Fig 7.2 Variation of Heat Transfer Coefficient with Coolant Flow Rate

30-degree corrugation angle favors the heat transfer coefficient till coolant flow rate of 4 LPM and then shows the drop whereas 20-degree corrugation angle shows the steady growth with the coolant flow rate and highest heat transfer coefficient of 326.1789 W/m2K for 5 LPM, for optimum result 20-degree corrugation angle with higher coolant flow rate is suggested here.

Volume Flow Rate of the cold fluid (LPM)	Peclet Number (Pe) for 30 & 20 Degree Corrugation Angle	Heat Transfer Coefficient (W/m²k) for 30 Degree Corrugation Angle	Heat Transfer Coefficient (W/m²k) for 20 Degree Corrugation Angle
2	171.0498832	212.1259	214.49905
2.5	213.812354	245.85618	240.11064
3	256.5748248	273.31394	260.78038
3.5	299.3372956	299.53821	279.728
4	342.0997664	323.19576	296.55612
4.5	384.8622372	307.92197	311.96456
5	427.6247081	325.09376	326.1789

Table 7.3 Variation of Heat Transfer Coefficient with Peclet Number (Pe)

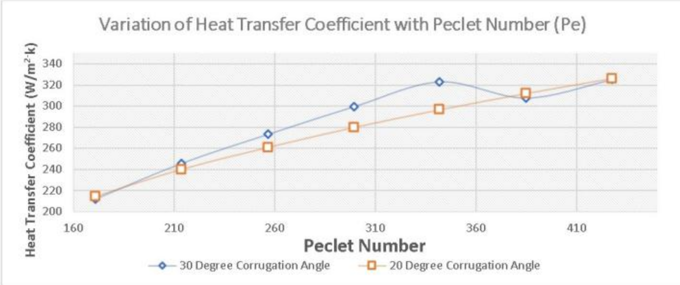

Fig 7.3 Variation of Heat Transfer Coefficient with Peclet Number (Pe)

30-degree corrugation angle shows the higher heat transfer coefficient till coolant flow rate of 4 LPM for Peclet Number of around 342.0997664 and then shows the drop whereas 20-degree corrugation angle shows the steady growth with the coolant flow rate and for optimum result 20-degree corrugation angle with higher or 5 LPM of coolant flow rate is suggested here which gives highest heat transfer coefficient of 326.1789 W/m²K.

Volume Flow Rate of the cold fluid (LPM)	Peclet Number (Pe) for 30 & 20 Degree Corrugation Angle	Overall Heat Transfer Coefficient (U), W/m²K for 30 Degree Corrugation Angle	Overall Heat Transfer Coefficient (U), W/m²K for 20 Degree Corrugation Angle
2	171.0498832	183.5044222	184.5418811
2.5	213.812354	203.181263	196.5977601
3	256.5748248	218.6136113	207.0418566
3.5	299.3372956	232.5419938	214.4987685
4	342.0997664	245.0458304	221.0101503
4.5	384.8622372	224.572165	226.6694652
5	427.6247081	229.735524	230.8425856

Table 7.4 Variation of Overall Heat Transfer Coefficient with Peclet Number (P_e)

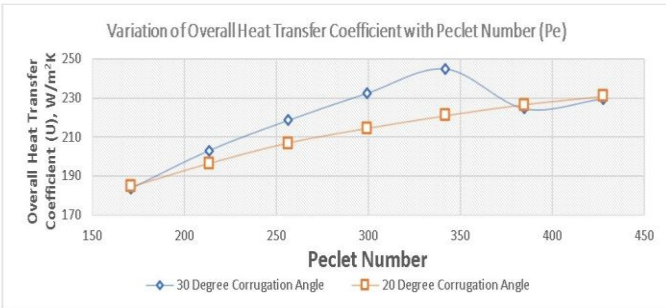

Fig 7.4 Variation of Overall Heat Transfer Coefficient with Peclet Number (Pe)

30-degree corrugation angle shows the higher overall heat transfer coefficient till coolant flow rate of 4 LPM for Peclet Number of around 342.0997664 and then shows the drop whereas 20-degree corrugation angle shows the steady growth with the coolant flow rate and for optimum result 30-degree corrugation angle with 4 LPM of coolant flow rate is suggested here which gives highest overall heat transfer coefficient of 245.0458304 W/m²K.

Volume Flow Rate of the cold fluid (LPM)	Peclet Number (Pe) for 30- & 20-Degree Corrugation Angle	Nusselt Number (Nu) for 30 Degree Corrugation Angle	Nusselt Number (Nu) for 20 Degree Corrugation Angle
2	171.0498832	353.28514	353.38593
2.5	213.812354	395.88882	394.98385
3	256.5748248	430.65335	428.41646
3.5	299.3372956	461.81447	459.07676
4	342.0997664	490.25598	486.31426
4.5	384.8622372	521.37007	511.10883
5	427.6247081	547.60664	533.81186

Table 7.5 Variation of Nusselt Number (Nu) with Peclet Number (Pe)

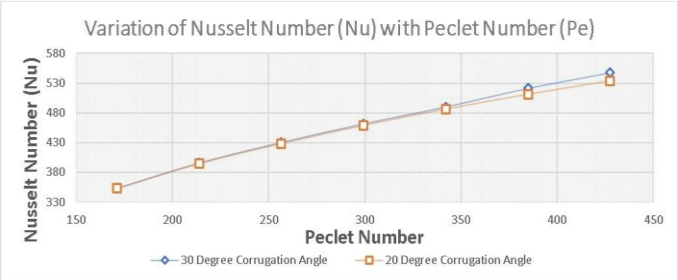

Fig 7.5 Variation of Nusselt Number (Nu) with Peclet Number (Pe)

30-degree corrugation angle & 20-degree corrugation angle both are showing the steady growth with the coolant flow rate and for optimum result 30-degree corrugation angle with higher coolant flow rate is suggested here which gives highest Nusselt Number (Nu) of 547.60664 and Peclet Number (Pe) of 427.6247081.

SN	Volume Flow Rate of the cold fluid (LPM)	Pressure Drop in the cold fluid for 30 Degree Corrugation Angle	Pressure Drop in the cold fluid for 20 Degree Corrugation Angle
1	**2**	3.5961163	3.9677907
2	**2.5**	5.2447381	5.7949247
3	**3**	7.1503347	7.9051784
4	**3.5**	9.3269869	10.309905
5	**4**	11.783925	13.005095
6	**4.5**	14.55938	15.996585
7	**5**	17.646032	19.27589

Table 7.6 Variation of Pressure Drop with Coolant Flow Rate

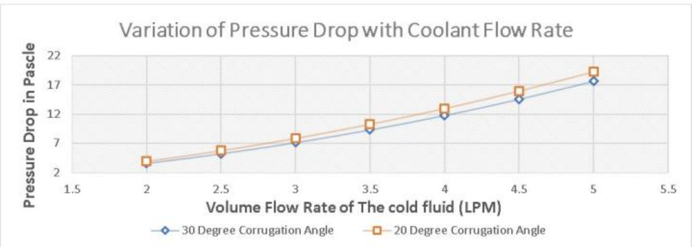

Fig 7.6 Variation of Pressure Drop with Coolant Flow Rate

30-degree corrugation angle & 20-degree corrugation angle both are showing the steady growth in Pressure Drop with the coolant flow rate and for optimum result 30-degree corrugation angle with lower coolant flow rate is suggested for lower pressure drop ie for requirement of lesser pumping power.

Volume Flow Rate of the cold fluid (LPM)	Peclet Number (Pe) for 30 & 20 Degree Corrugation Angle	Friction Factor (f) for 30 Degree Corrugation Angle	Friction Factor (f) for 20 Degree Corrugation Angle
2	171.0498832	0.051307176	0.050458148
2.5	213.812354	0.07377554	0.072233812
3	256.5748248	0.099016276	0.096598248
3.5	299.3372956	0.126969712	0.123913084
4	342.0997664	0.1576103	0.15343964
4.5	384.8622372	0.192070332	0.18576196
5	427.6247081	0.228850672	0.220710644

Table 7.7 Variation of Friction Factor (f) with Peclet Number (Pe)

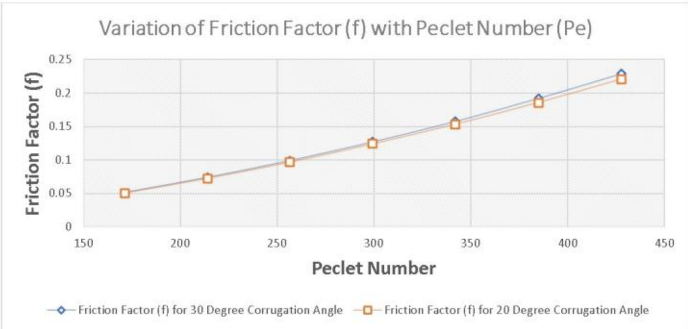

Fig 7.7 Variation of Friction Factor (f) with Peclet Number (Pe)

Here 20-degree corrugation angle should be preferred for lower friction factor ie minimum losses in pumping power.

SN	Volume Flow Rate of the cold fluid (LPM)	Effectiveness = (Q/Q_{max}) for 30 Degree Corrugation Angle	Effectiveness = (Q/Q_{max}) for 20 Degree Corrugation Angle
1	**2**	0.088354468	0.088485319
2	**2.5**	0.101229521	0.096430319
3	**3**	0.112771596	0.103295426
4	**3.5**	0.125059468	0.10532617
5	**4**	0.136746383	0.111146383
6	**4.5**	0.12300367	0.116742447
7	**5**	0.121292021	0.105726064

Table 7.8 Variation of Effectiveness with Coolant Flow Rate

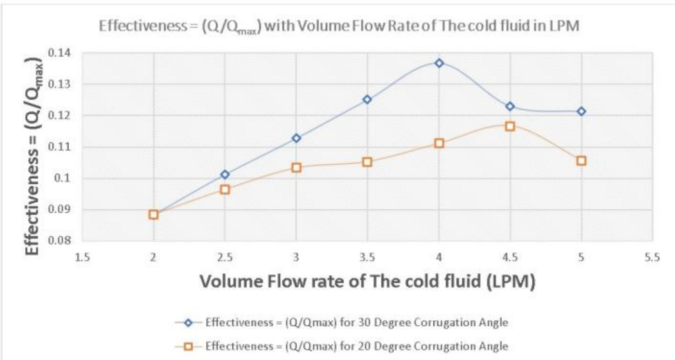

Fig 7.8 Variation of Effectiveness with Coolant Flow Rate

Effectiveness of heat exchanger is showing growth with 30-degree corrugation angle and highest effectiveness at coolant flow rate of 4 LPM.

SN	Volume Flow Rate of the cold fluid (LPM)	LMTD for 30 Degree Corrugation Angle	LMTD for 20 Degree Corrugation Angle
1	**2**	43.73867673	43.9819338
2	**2.5**	43.93477167	44.18849607
3	**3**	44.08112648	43.92077162
4	**3.5**	44.16578693	44.0836771
5	**4**	44.23578329	44.15711377
6	**4.5**	44.23565449	44.21328625
7	**5**	44.34139014	44.40994055

Table 7.9 Variation of LMTD with Coolant Flow Rate

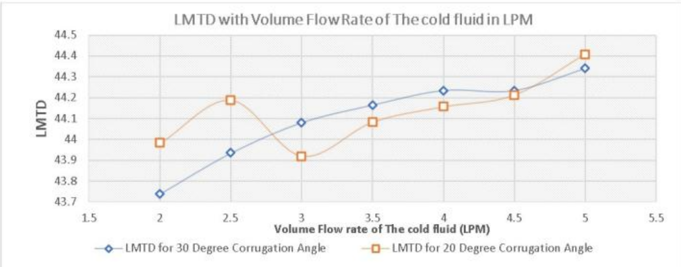

Fig 7.9 Variation of LMTD with Coolant Flow Rate

20-degree corrugation angle is showing zig zag trend in LMTD (Logarithmic Mean Temperature Difference) with coolant flow rate whereas 30-degree corrugation angle is showing steady growth rate with a little drop at 4.5 LPM of coolant flow rate. For the optimum result from 3 LPM to 4.5 LPM, 30-degree corrugation angle should be preferred whereas for highest value of LMTD, 20-degree corrugation angle with 5 LPM of coolant flow rate should be preferred.

8. CONCLUSIONS

It has been observed in above study that 30-degree corrugation angle with 4 LPM of coolant flow rate gives highest rate of heat transfer of 897.0061 Watt, 20-degree corrugation angle shows the steady growth with the coolant flow rate and gives highest heat transfer coefficient of 326.1789 W/m2K for 5 LPM, and for the optimum result 20-degree corrugation angle with higher coolant flow rate is suggested here. Where 20-degree corrugation angle with higher or 5 LPM of coolant flow rate is suggested here which gives highest heat transfer coefficient of 326.1789 W/m2K.

On in other hand 30-degree corrugation angle with 4 LPM of coolant flow rate is suggested here which gives highest overall heat transfer coefficient of 245.0458304 W/m2K.

Results show that 30-degree corrugation angle with higher coolant flow rate gives highest Nusselt Number (Nu) of 547.60664 and Peclet Number (Pe) of 427.6247081.

Whereas 30-degree corrugation angle with lower coolant flow rate gives lower pressure drop that is for requirement of lesser pumping power 30-degree of corrugation angle would be better to prefer. In another perspective 20-degree corrugation angle should be preferred for lower friction factor that is minimum losses in pumping power.

Effectiveness of heat exchanger is showing growth with 30-degree corrugation angle and we get highest effectiveness at coolant flow rate of 4 LPM. For the optimum result of LMTD, from 3 LPM to 4.5 LPM, 30-degree corrugation angle should be preferred whereas for the highest value of LMTD, 20-degree corrugation angle with 5 LPM of coolant flow rate should be preferred.

Results for pressure drop were compared with the experimental results obtained in the research work of S.D. Pandey, V.K. Nema [18]. We found that in our research work that pressure drop for 2 LPM volume flow rate of the cold fluid and hot fluid was obtained as 3.59 to 3.96 Pa with the total range of pressure drop as 3.59 Pa to 19.27 Pa whereas in case of experiment with nanofluid [18] its range was 12 Pa to 160 Pa. We know that the pressure drop is responsible for the consumption of power, so in place of using nanofluid, only water could be used for better results along with the passive way to improve the rate of heat transfer that is by improving the area of heat transfer and changing the corrugation angle of sheet.

8. Future Scope

It has been observed that still whole surface of corrugated plate is not being used for the heat transfer by observing the stream lines which are not completely in contact with corrugated steel plates and to avoid this, one can also change angle of corrugation as 25-degree, 40-degree, 45-degree or 50-degree for corrugated plates. One can also take the corrugation profile as pure Sine Curve and then compare with the results of different angles. In this research work as we have found all the results for various coolant flow rates by keeping the the hot fluid flow rate constant as 2 LPM, so there is scope for variation in the hot fluid flow rate for better results. If the corrugated plate material is changed as copper then due to more thermal conductivity than the steel, there would be greater rate of heat transfer and also the effectiveness will be higher.

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