Enhancement in heat transfer coefficient of water by using nano-fluids for corrugated plate heat exchanger

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ABSTRACT— Plate heat exchangers (PHEs) are extensively used for heating, cooling and heat–regeneration applications in the chemical, Food and pharmaceutical industries. In a conventional plate heat exchanger, increase in heat transfer area directly affects the size of heat exchanger by making it bulky. To overcome this limitation and to test the capability of compact heat exchanger namely corrugated plate heat exchanger, there is a need to increase the heat transfer coefficient of the base fluid. Preferably, addition of Nano particles along with the provision of corrugations on the plates will lead to an increase in effectiveness of the heat exchanger. This study is carried to enhance heat transfer of water by addition of Nano fluids and test the test rig for parallel flow arrangement and counter flow arrangement for different mass flow rates of hot fluid. design of heat exchanger is done based on sizing, that is determination of heat transfer parameters, number of thermal plates required and physical dimensions. Testing of counter flow and parallel flow arrangement is carried out by varying mass flow rate considering water as working fluid. It is observed that there is a slight drop in effectiveness with increase in heat capacity ratio. For water as a working fluid the value of effectiveness varies from 0.66 to 0.80 while that of for Nano fluids the value of effectiveness varies from 0.70 to 0.82.

***Keywords*—** Corrugated PHE, Counterflow, Effectiveness, Nano Fluids, Sizing etc.

I. INTRODUCTION

Heat exchangers are devices that facilitate the exchange of heat between two fluids that are at different temperature while keeping them without mixing with each other. These exchangers are classified according to construction, flow arrangement; number of fluids, compactness, etc. The use of heat exchanger gives higher thermal efficiency to the system. In many applications like power plants, petrochemical industries, air conditioning etc. heat exchangers are used. Plate heat exchanger is generally used in dairy industry due to its ease of cleaning and thermal control. The plate heat exchangers are built of thin metal heat transfer plates and pipe work is used to carry streams of fluid. Plate heat exchangers are widely used in liquid to liquid heat transfer and not suitable for gas to gas heat transfer due to high pressure drop.

According to (Shah and Sekulic ,2003), Heat exchanger is a test rig used for transfer of heat energy between two or more fluids, between a solid surface and a fluid or solid particulates and a fluid, at different temperature and in thermal contact, usually without external heat and work interactions. They are widely used in refrigeration, air conditioning, power plants, chemical plant, petrochemical plant and automobile.

(Shah and Kandilkar ,1989) had presented classification of PHE on the basis of number of passes, flow arrangement and by considering end plate effect. Configurations used for this study were 1-1 (1 Pass-1 Pass), 2-1, 2-2, 3-3, 4-1, 4-2, and 4-4 arrangements, and six configurations for the 3-1 arrangement. Results were presented for temperature effectiveness, log-mean temperature difference correction factor as a function of number of transfer units, heat capacity ratio and total number of thermal plates. Also this study provides guidelines for selection of appropriate PHE configuration.

(Murugesan and Balasubramani ,2013) had investigated experimentally heat transfer enhancement in PHE with regard to effects of various operating and design parameters. In this study tests are conducted on plate pack of length 31 mm with 100°C work temperature and design pressure 6 kg/cm2. The parameters such as flow rates, temperature, pressure and properties of test fluid were varied. The outcomes of this study were overall heat transfer coefficient and individual heat transfer coefficient increases with mass flow rate.

(Chouda, 2014) In this paper the thermal performance analysis of a counter flow heat exchanger is performed by varying the composition of Nano fluids used, which is a mixture of coolant and iron particles. An experimental analysis has been performed on counter flow heat exchanger. The volume fraction of coolant varies from 0-2.0% by mass. Experimental results such as heat transfer rates, overall heat transfer coefficient, and heat exchanger effectiveness have been calculated for assessing the performance of heat exchanger. The objective of this project is to determine whether the use of Nano fluids improves the heat exchanger performances and at what percentage of Nanoparticles-coolant mixture the performance of counter flow heat exchanger obtain maximum heat exchange rate and at what percentage deteriorates.

(N. Targui and H. Kahalerras ,2014) In this work a numerical simulation of nanofluids flow in a double pipe heat exchanger provided with porous baffles. The hot nanofluid circulates in the annular gap. The Darcy Brinkman-Forchheimer model is adopted to describe the flow in the porous regions, and the governing equations with appropriate boundary conditions are solved

by finite volume method. The highest heat exchanger performances are obtained when nanoparticles are added only to the cold fluid.

(Arun Kumar Tiwari, 2015) In this paper, an attempt is made to experimentally investigate the thermal performance of shell and tube heat exchanger using nanofluids. The cold water based nanofluids flow in tube side and water as hot fluid flows on shell side. Use of nanoparticles in water based nanofluid as coolant in shell and tube heat exchanger improves the effectiveness by a considerable amount, while convective and overall heat transfer coefficient increases even further with the addition of 3% Al2O3 nanoparticles in water based fluid

(Arun Kumar Tiwari ,2015) In the present work theoretically investigated the performance characteristics of hot water-based nanofluid containing Al2O3 nanofluid and water as a cold fluid which cross a rectangular arrangement of tubes in a shell and tubes heat exchanger. The hot nanofluid flow in tube side and water as cold fluid flows on shell tube. The results show the effectiveness and cooling capacity increases by a considerable amount, while the overall heat transfer coefficient increases even further at higher nanoparticle concentrations. It was also found that the performance of heat exchanger increases appreciably due to use if Al_2O_3 nanofluid.

II. METHODOLOGY

2.1 Heat duty (Q):

Heat rejected by hot water,

$Q = m_h Cp_h \Delta T_h$ …… (2. 1)

$Q = m_h Cp_h (T_{h1} - T_{h2})$

$= 0.25 \times 4194 \times (85-65)$

Q $= 20970 W$

Where, m_h =mass flow rate of hot water in Kg/s,

Cp_h=specific heat capacity of hot water J/Kg K

ΔT_h = temperature difference between inlet and outlet of hot water in K

Heat absorbed by cold water,

$Q = m_c Cp_c \Delta T_c$ …… (2. 2)

$= m_c Cp_c (T_{c2} - T_{c1})$ 20970

$= 0.25 \times 4178 \times (T_{c2} - 25)\ T_{c2}$

$= 45.07^0 C$

Where, m_c =mass flow rate of cold water in Kg/s,

Cp_c= specific heat capacity of hot water in J/Kg K

ΔT_c=temperature difference between inlet and outlet of cold water in K

2.2 Thermo physical properties at fluid mean temperature:

1. *Hot water mean temp* = (85+65)/2 = 75 ^0C
2. *Cold water mean temp* = (25+45.07)/2 = 35.03 ^0C

Table 1: Thermo Physical properties of fluid

Property	Unit (metric)	Hot water (mean temp)	Cold water (mean temp)
Heat capacity(Cp)	J/kgK	4197	4178
Thermal conductivity (k)	W/mK	0. 668	0. 6316
Dynamic viscosity (μ)	Ns/m^2	0. 000378	0. 000370
Density (ρ)	Kg/m^3	974.851	994.034

2.3 The hydraulic diameter:

$D_e = 2b/\psi$

$= 2 \times 0.00224 / 1.16$

D_e = 0.00386 m

The flow area for water:

$A = NWb$…… ……. (2.3)

Where,

A= flow area for water in m^2, N=number of water chambers

W =width of plate in m b =distance between two plates in m

2.3.1 For hot water

$A_h = N_h W b$ …… ……. (2.4)

Here,

N_h= 6, W=0.090 m, b =0.00224 m

$A_h = 6 \times 0.0.090 \times 0.00224$

A_h=0.0012 m^2

2.3.2 For cold water

$A_c = N_c W b$…… …… (2.5)

Here,

N_h= 7, W=0.090 m, b =0.00224 m

A_c=7×0.090×0.00224

A_c=0.001411 m^2

2.4 Velocity of water:

$V = m \div (A\ \rho)$ ……. (2.6)

Where

V= velocity of hot water in m/s, m=mass flow rate in Kg/s

A= flow area for water in m^2, ρ = density of water in Kg/m^3

2.4.1 For hot water

$V_h = m_h A_h \rho_h$……. …..……. (2.7)

Here,

m_h=0.25 Kg/s, A_h =0.0012 m^2, ρ_h =974.851 Kg/m^3

=0.25/ (974.851×0.0012)

V_h =0.02137 m/s

2.4.2 For cold water

$V_c = m_c A_c \rho_c$……. (2.8)

Here,

m_c=0.25 Kg/s, A_c =0.001411 m^2, ρ_c = 994.034 Kg/m^3

=0.250.001411×994.11

=0.1783 m/s

2.5 Reynolds number

2.5.1 For hot water:

$Re_h = \rho_h V_h De/\mu_h$ …… …… (2.9)

Here,

ρ_h = 973.851 Kg/ m^3, Vth = 0.2137 m/s, De= 0.00386 m, μ_h = 0.000378 Ns/m^2

= (973.851× 02137× 0.00386)/0.000378

Re_h=2127.34

2.5.2 For cold water:

$Re_c = \rho_c V_c D_c/\mu c$ ……...……. (2.10)

Here,

ρ_c = 993.034 Kg/ m^3, V_c = 0.1783 m/s, D_e = 0.00386 m

μ_h = 0.000720 Ns/m^2

= (993.034× 0.1783× 0.00386)/0.000720

Re_c =950.18

2.6 Prandtl number

2.6.1 For hot water,

$Pr_h = \mu_h Cp_h/k_h$ …… (2.11)

Here,

μ_h= 0.000378 Ns/m^2, Cp_h= 4194 J/Kg K, k_h =0.668 W/mK

= (4194×0.000378)/0.668

Pr_h=2.3732

2.6.2 For cold water,

$Pr_c = \mu_c Cp_c/k$ ……. (2.12)

Here,

μ_c= 0.000720 Ns/m^2, Cp_c= 4178 J/Kg K

k_c =0.6316 W/mK

=4178×0.0007200.6316

Pr_h=4.7627

2.7 Nusselt Number

Here Re < 2000 so taking relation for laminar flow,

$N_u = 0.662\ Re^{0.5} Pr^{0.33}$… ……. (2.13)

2.7.1 For hot water

$Nu_h = 0.662\ Re_h^{0.5}$……. (2.14)

Convective heat transfer coefficient for hot water (h_h):

$h_h = (0.662)\ (k_h/D_e)\ Reh^{0.5} Pr_h^{0.33}$

Here,
k_h=0.668W/mK, Re_h=2127.32, Pr_h=2.3732
=0.662× (0.668/0.00386) × $(2127.32)^{0.5}$× $(2.3732)^{0.33}$
h_h=7027.90$W/m^2.K$
Where,
h_h is the hot fluid heat transfer coefficient

2.7.2 For cold water
$Nuc = 0.662\ Re_c^{0.5} Pr_c^{0.33}$ …… ……. (2.15)
Convective heat transfer coefficient (h_c):
$h_c = (0.662)\ k_c/D_e)\ Re_c^{0.5} Pr_c^{0.33}$… …….. (2.16)
= (0.662) × (0.6316/0.00386) × $(950.18)^{0.5}$× $(4.7627)^{0.33}$
h_c=5588.65W/m^2K
Where, h_c is the cold fluid heat transfer coefficient

2.8 Overall heat transfer co efficient:
The overall heat transfer coefficient for a clean surface is given by
$1/U=1/h_h+t/k_p+1/h_c$…… (2.17)
Here,
h_h =7027.90 W/m^2. K, h_c = 5588.90 W/m^2. K
K_p = 17.5 W/m^2K t = 0.0005m
= (1/7027.90) +(0.0005/17.5) +(1/5588.90)
$1/U$= 0.0003497
U = 2858.81$W/m^2.K$

2.9 Logarithmic Mean Temperature Difference (LMTD):
Θ_m =[[$(T_{h1}-T_{c2})$ −$(T_{h2}-T_{c2})$] / ln[$(T_{h1}-T_{c2})/(T_{h2}-T_{c2})$]] ……. (2.18)
= [[(85−45.07) −(65−45.07)]/ ln [85−45.0765−45.07]]

Θ_m= 28.78 ^{0}C
2.10 Area required
$Q=UA\Theta_m$… ……..…….… (2.19)
20970=2858.81×A×28.78.78
A= (20970 / 2858.81×28.78)
A=0.2548 m^2
Area for single plate = 0.231×0.090=0.020m^2
Total no of plates = $Total\ area$ / $Area\ of\ single\ plate$
= 0.2548 / 0.020
= 13
In this design total area of heat transfer obtained is 0.2548m^2 and total numbers of plates required are 13.

EXPERIMENTAL SETUP

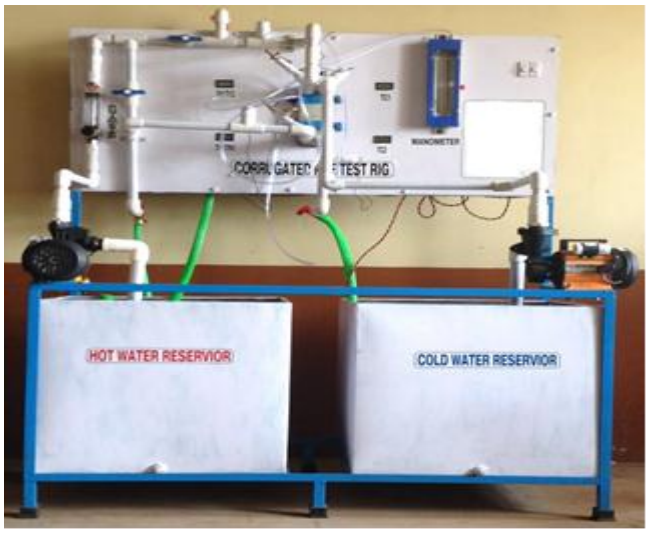


Fig.1 Experimental Setup

Experimental setup consists of tanks, pumps, Rotameter, U tube manometer, temperature indicators, corrugated plate heat exchanger. test the performance of heat exchanger for parallel and counter flow arrangement with water as a working fluid. The heat exchanger has total 13 plates and it is constructed using Stainless Steel AISI 316. Each plate is flat and has thickness of 0.5 mm. The total heat transfer area is 0.2548 m^2. Thermometers are placed to measure inlet and outlet temperatures of hot and cold water. following trials are taken

1. Test on corrugated plate heat exchanger with parallel flow arrangement (with water)
2. Test on corrugated plate heat exchanger with counter flow arrangement (with water)
3. Test on corrugated plate heat exchanger with parallel flow arrangement (with water and Nano-fluids)
4. Test on corrugated plate heat exchanger with counter flow arrangement (with water and Nano-fluids)

RESULT AND DISCUSSION

4.1 Results for parallel flow arrangement

fig 2 shows variation of convective heat transfer coefficient with Reynolds number. It is observed that heat transfer coefficient increases with increase in Reynolds number. Increase in Reynolds number is attributed to more turbulent flow and it leads to higher heat transfer rate. Value of heat transfer coefficient varies from 4000-6000 W/m^2K. Maximum value of heat transfer coefficient is 5632.48 for water with Reynolds number 1040 whereas, value of heat transfer coefficient is 5807 W/m^2K for Nano fluids with Reynolds number 1048.

Table 2 Results for parallel flow arrangement(water as working fluid)

Parameter	Units	1	2	3	4	5
m_h	Kg/s	0.11	0.12	0.15	0.19	0.20
m_c	Kg/s	0.25	0.25	0.25	0.25	0.25
T_{h1}	^0C	47.8	56	49	49.2	52.7
T_{h2}	^0C	37.7	39.2	38.2	39.7	41.8
T_{c1}	^0C	34.9	30.7	32.8	33.1	33.1
T_{c2}	^0C	37	36.9	36.8	37.8	39.3
Q	W	4642.86	8778	6891.83	7701.89	9490.56
Re_h	-	562.18	594.30	794.18	1027.46	990.88
h_h	W/m^2K	4114.22	4362.03	4861.07	5497.06	5632.48
A_m	^0C	4.1868	9.5917	6.04	6.6449	8.30
U	W/m^2K	4352.15	3591.7	4478.14	4548.93	4487.60

Fig.2 Heat transfer coefficient Vs. Reynolds number for parallel flow

Table 3 Results for counter flow arrangement(water as working fluid)

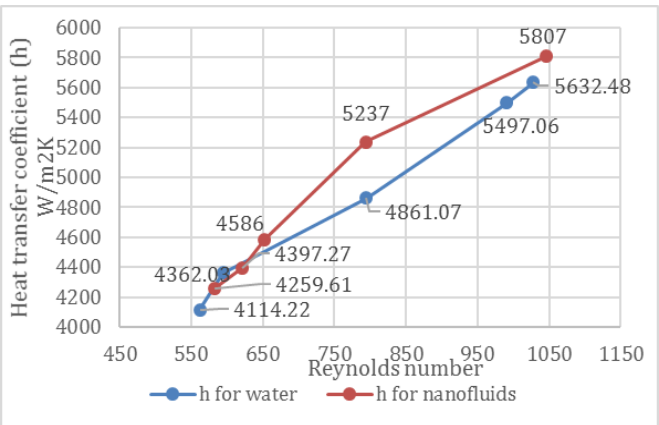
Parameter	Units	1	2	3	4	5
m_h	Kg/s	0.11	0.12	0.15	0.19	0.20
m_c	Kg/s	0.25	0.25	0.25	0.25	0.25
T_{h1}	^0C	48.6	53.7	49.5	49.7	46.5
T_{h2}	^0C	35.4	37.7	25.5	39.7	38.8
T_{c1}	^0C	32.8	34.4	17.7	35.2	35.8
T_{c2}	^0C	37.2	40.1	26.7	40.2	40.3
Q	kW	6.07	8.39	13.9	8.11	6702.73
Re_h	-	550.09	577.7	633.085	1027.4	1064.75
h_h	kW/m^2K	4.01	4.11	4.50	5.52	5662.05
A_m	^0C	5.95	7.27	13.98	6.69	4.4080
U	kW/m^2K	4.02	4.52	3.91	4.76	5.91
R	-	0.44	0.50	0.5547	0.77	0.83
ε	-	0.80	0.79	0.72	0.70	0.66

Table: 4 Results for parallel flow arrangement (Nano fluids as working fluid)

Parameter	Units	1	2	3
m_h	Kg/sec	0.11	0.15	0.20
m_c	Kg/sec	0.25	0.25	0.25
T_{h1}	^{0}C	57.4	47.1	45.7
T_{h2}	^{0}C	32.2	34.2	35.8
T_{c1}	^{0}C	21.5	27.1	28
T_{c2}	^{0}C	28.6	31.8	33.3
Q	W	11584.18	7008	8617
Re_h	-	582.71	652.23	1045.24
h_h	W/m^{2}K	4259.61	4586	5807
A_m	^{0}C	13.62	8.30	8.24
U	W/m^{2}K	2834	2221.71	2750.91
R	-	0.439	0.52	0.83
ε	-	0.64	0.59	0.48

Table 5 Results for counter flow arrangement (Nano fluids as working fluid)

Parameter	Unit	1	2	3
m_h	Kg/sec	0.11	0.15	0.20
m_c	Kg/sec	0.25	0.25	0.25
T_{h1}	^{0}C	56.9	55.6	44.5
T_{h2}	^{0}C	25.6	31.6	32.9
T_{c1}	^{0}C	18.8	24.2	29.8
T_{c2}	^{0}C	27.9	33.3	34
Q	W	14388	13041.6	10099
Re_h	-	562.19	701.60	965.94
h_h	W/m^{2}K	4233.57	4645	5653.30
A_m	^{0}C	15.34	13.50	5.88
U	W/m^{2}K	2473.68	2540.84	4515.67
R	-	0.44	0.52	0.80
ε	-	0.82	0.74	0.70



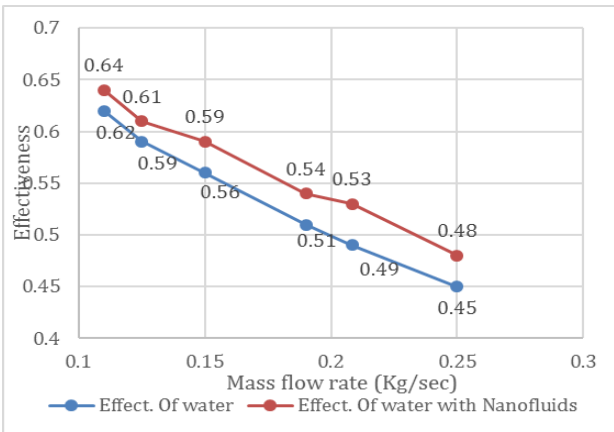


Fig.3 Effectiveness Vs. Mass flow rate of hot fluid

Fig.3 shows variation of effectiveness with mass flow rate of hot fluid. It is observed that effectiveness increases as mass flow rate of hot fluid decreases. Due to increase in mass flow rate time elapsed for heat transfer between two fluids goes on decreasing resulting into a lesser temperature drop. Effectiveness is found to be maximum for minimum mass flow rate of hot fluid. Maximum Effectiveness is found to be 0.62 at 0.11 kg/sec for whereas, that for Nano fluids it is 0.64 at 0.11 kg/sec.

Fig 4 shows effect of heat capacity ratio on effectiveness of corrugated PHE. It is observed that there is a slight drop in effectiveness with increase in heat capacity ratio. For water as a working fluid the value of effectiveness varies from 0.45 to 0.62 while that of for Nano fluids the value of effectiveness varies from 0.48 to 0.64.

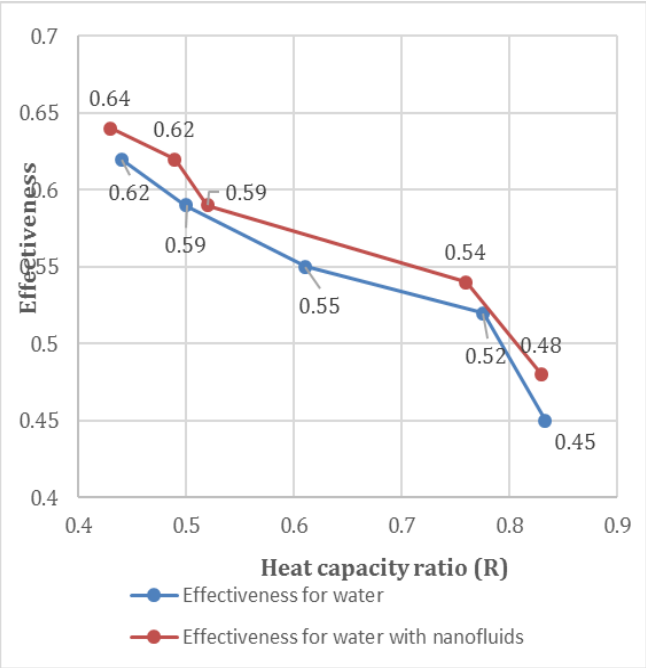


Fig.4 Effectiveness Vs. Heat capacity ratio

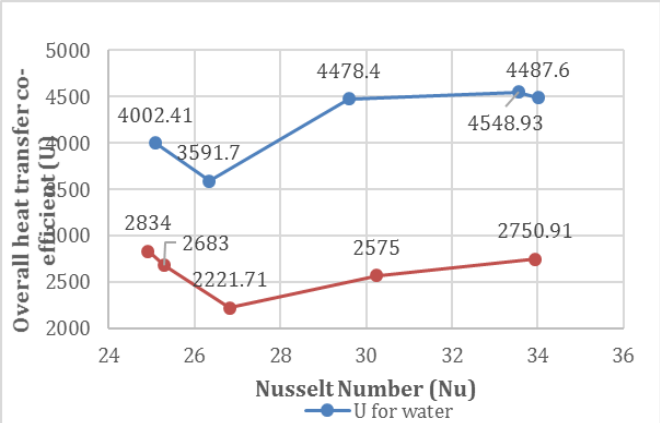


Fig.5 Overall heat transfer co-efficient Vs. Nusselt Number

Fig.5 shows the effect of Nusselt number on overall heat transfer co-efficient. Nusselt number is a function of Reynolds number. So as Nusselt number increases, overall heat transfer coefficient also increases. Overall heat transfers coefficient value changes from 2000 to 5000.

4.2 Results for counter flow arrangement:

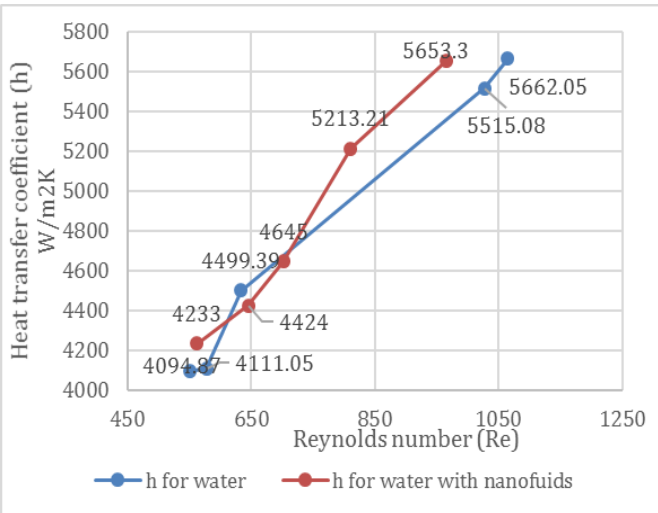


Fig.6 Heat transfer coefficient Vs. Reynolds Number

Fig.6 shows variation of convective heat transfer coefficient with Reynolds number. It is observed that heat transfer coefficient increases with increase in Reynolds number. Increase in Reynolds number is attributed to more turbulent flow and it leads to higher heat transfer rate. Value of heat transfer coefficient varies from 4000-5800 W/m^{2}K. Maximum value of heat transfer coefficient is 5662.05 for water with Reynolds number 1060 whereas, value of heat transfer coefficient is 5653 W/m^{2}K for Nano fluids with Reynolds number 965.

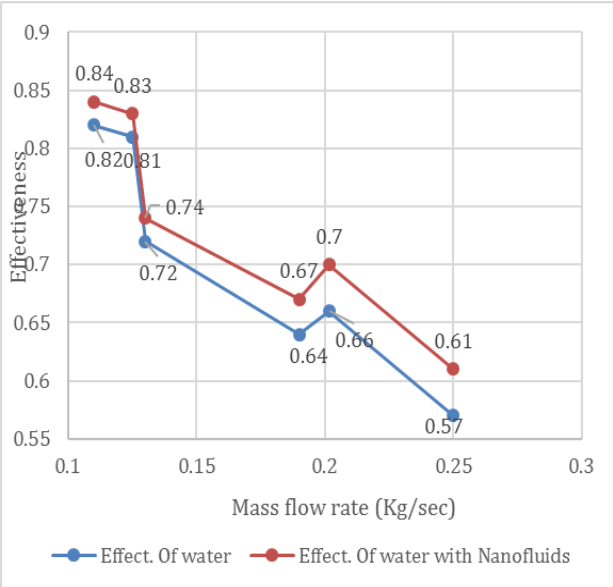


Fig.7 Effectiveness Vs. Mass flow rate of hot fluid

Fig.7 shows variation of effectiveness with mass flow rate of hot fluid. It is observed that effectiveness increases as mass flow rate of hot fluid decreases. Due to increase in mass flow rate time elapsed for heat transfer between two fluids goes on decreasing resulting into a lesser temperature drop. Effectiveness is found to be maximum for minimum mass flow rate of hot fluid. Maximum Effectiveness is found to be 0.82 at 0.11 kg/sec for whereas, that for Nano fluids it is 0.84 at 0.11 kg/sec.

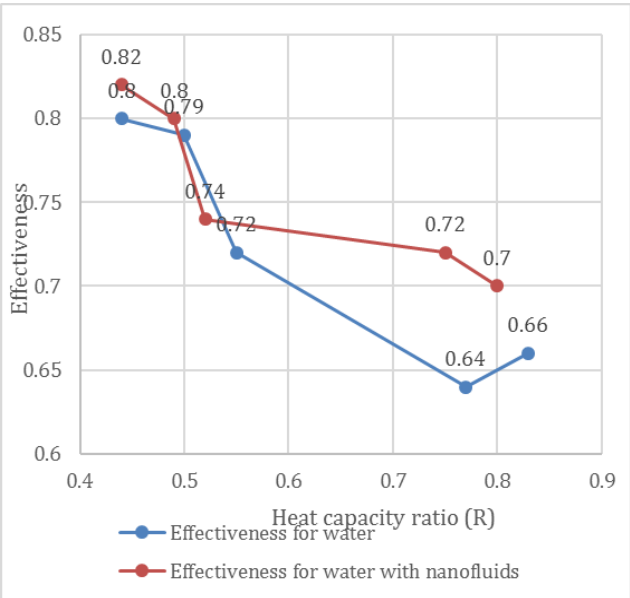


Fig.8 Effectiveness Vs. Heat capacity ratio

Fig.8 shows effect of heat capacity ratio on effectiveness of corrugated PHE. It is observed that there is a slight drop in effectiveness with increase in heat capacity ratio. For water as a working fluid the value of effectiveness varies from 0.66 to 0.80 while that of for Nano fluids the value of effectiveness varies from 0.70 to 0.82.

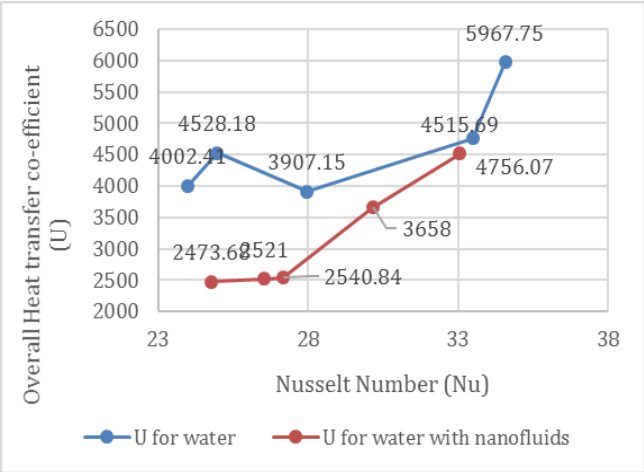


Fig.9 Overall heat transfer coefficient Vs. Nusselt number

Fig.9 shows the effect of Nusselt number on overall heat transfer co-efficient. Nusselt number is a function of Reynolds number. So as Nusselt number increases, overall heat transfer coefficient also increases. Overall heat transfers coefficient value changes from 2000 to 6500.

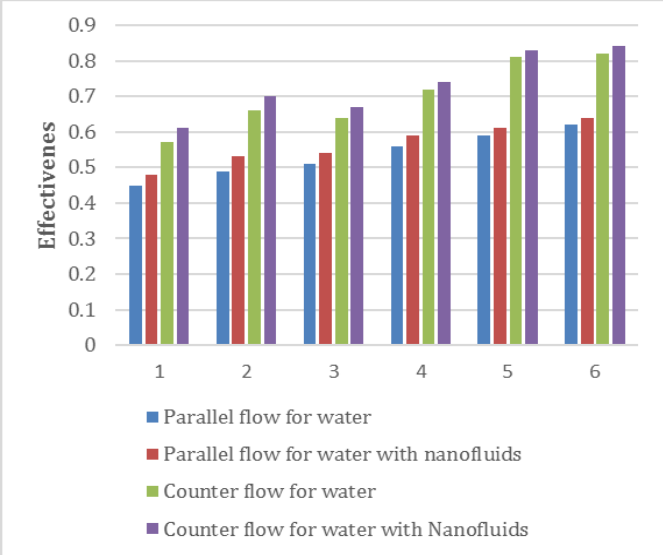


Fig.10 Comparison of effectiveness for water and Nano fluids

Fig.10 represents the comparison between the effectiveness for parallel flow and counter flow respectively associated to water and Nano fluids. Above graph promotes counter flow with Nano fluids for the highest exchanger effectiveness. The highest value of effectiveness obtained is 0.84 which is for counter flow arrangement with Nano fluids as working medium.

III CONCLUSION

The main focus of this project is to understand the effect of nanoparticles when added with working medium and to investigate experimentally its effect on various performance parameters of PHE. The following are the findings of this experimental investigation:

1. Convective heat transfer coefficient increases with Reynolds number and mass flow rate for both parallel and counter flow arrangement. This is due to the fact that flow becomes more turbulent and cause for turbulence can be attributed to plate geometry i.e., corrugations as well as high flow velocity.
2. Effectiveness of heat exchanger decreases with increase in mass flow rate of hot fluid. Maximum effectiveness for parallel flow arrangement is 0.62 and that of for counter flow arrangement is 0.80 (for water as a working fluid).
3. Exchanger effectiveness considerably increases when nanoparticles are added into the base fluid. Maximum effectiveness obtained is 0.64 and 0.82 for parallel and counter flow arrangements respectively (for Nano fluid as working medium).
4. Increase in effectiveness of PHE by addition of 0.4% Nano fluids by volume into base fluid is obtained as 4% for this study.
5. In order to increase exchanger effectiveness, it is required to reduce the mass flow rate of cold fluid for Nano fluid as working medium.
6. Maximum temperature drop achieved for this heat exchanger is in the range of 24^0C – 27^0C, which is comparatively higher. This can be attributed to enhanced thermos physical properties as well as plate geometry.

IV REFERENCES

[1] R. K. Shah and D. P. Sekulic, (2003). "A Textbook of Fundamentals of Heat exchanger design"

[2] R. K. Shah and S. G. Kandilkar MAY (1989) "Multipass Plate Heat Exchangers- Effectiveness-NTU Results and Guidelines for Selecting Pass Arrangements", *Transactions of ASME,* Vol.111,

[3] R. Saidur, S.N. Kazi, M.S. Hossain (2011) "A review on the performance of nanoparticles suspended with refrigerants and lubricating oils in refrigeration system"

[4] R. Saidur, K.Y. Leong, H.A. Mohammad (2011) "A review on applications and challenges of nanofluids", *Renewable and Sustainable Energy Reviews,* 1646–1668

[5] Jogi Nikhil G., Lawankar Shailendra M. Oct (2012) "Heat Transfer Analysis of Corrugated Plate Heat Exchanger of Different Plate Geometry", *International Journal of Emerging Technology and Advanced Engineering*, ISSN 2250-2459, Volume 2,

[6] Murugesan M.P. and Balasubramani R. February (2013), "The Experimental Study on Enhanced Heat Transfer Performance in Plate Type Heat Exchanger" *Research Journal of Engineering Sciences,* Vol.2(2),16-22, ISSN 2278-9472.

[7] Vishal R. Naik, V.K. Matawala (April 2013) "Experimental Investigation of single phase Chevron Type Gasket Plate Heat Exchanger", *International Journal of Engineering and Advanced Technology(IJEAT),* ISSN: 2249-8958, Volume-2, Issue-4,

[8] T K S Sai Krishna, S G Rajasekhar, C Pravarakhya July-August (2013) "Design and Analysis of Plate Heat Exchanger with CO2 and R134a as working fluids", *International Journal of Mechanical Engineering and Technology (IJMET),* ISSN 0976-6359 Volume 4. Issue 4

[9] Masoud Asadi, Ramin Haghighi Khoshkhoo (February 2014) "Effects of Chevron Angle on Thermal Performance of Corrugated Plate Heat Exchanger", *International Journal of Engineering Practical Research (IJEPR)* Volume 3 Issue 1,

[10] Raman Chouda (September 2014) "Analysis of Heat Exchanger with Nano fluids", *International Journal of Core Engineering & Management (IJCEM)* Volume 1, Issue 6,

[11] N. Targui, H. Kahalerras (2014) "Analysis of a Double Pipe Heat Exchanger Performance by Use of Porous Baffles and Nano fluids", *International Journal of Mechanical, Aerospace, Industrial, Mechatronic and Manufacturing Engineering* Vol: 8, No"9,

[12] Arun Kumar Tiwari (2015) "Enhancement of Heat Transfer by Using Nano fluids in a Heat Exchanger", *International Journal of Advances in Production and Mechanical Engineering (IJAPME),* ISSN: 2394-6210, Volume -1, Issue-1,

[13] Arun Kumar Tiwari (2015) "Thermal Performance of Shell and Tube Heat Exchanger Using Nano fluids", *International Journal of Advances in Production and Mechanical Engineering (IJAPME),* ISSN: 2394 - 6210, Volume - 1, Issue-1,

[14] Tambe Shahanwaj K, Pandhare Nitin T, Bardeskar Santosh J, Khandekar S.B(2015) ("Experimental Investigation of Performance of Plate Heat Exchanger for Water as Working Fluid", IJRET: *International Journal of Research in Engineering and Technology, eISSN: 2319-1163* | pISSN: 2321-7308.

