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INVESTIGATION OF HEAT TRANSFER RATE OF WATER BY ADDITION OF SIO_2 NANOFLUID

Sakru Rathod [1], Avinash Salve [1], Sachin Gore [1], Arun Gore [1], Prof. Mahesh Shinde [2]

[1]*Mechanical Engineering, Jayawantrao Sawant College of Engineering, Hadapsar, Pune*
[2]*Mechanical Engineering, Jayawantrao Sawant College of Engineering, Hadapsar, Pune*

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.

1. INTRODUCTION

Plate heat exchanger (PHE) originally received particular attention from the dairy industry due to their suitability in hygienic application. Soon their use became wide spread in the food, juice, beverage and pharmaceutical industries due to the ease of cleaning and the thermal control required for sterilization and pasteurization. They are currently diverse industries such as the synthetic rubber industry, paper mills, petrochemicals plants, and in process heaters, coolers and closed-circuit cooling system. Plate heat exchangers are used in liquid-to-liquid heat exchange application. Here, the PHE consist of series of parallel plates that are corrugated both to increase turbulence and to give rigidity. The basic objective of providing corrugation to the plates is to impart high turbulence to the fluids, which results in a very high heat transfer rate compared to those obtainable in the shell and tube heat exchangers for similar duties.

In addition to this, many methods are available to improve heat transfer in processes. The flow of heat in a process can be calculated as, $Q = hA\Delta T$ where, Q is to heat flow, h is heat transfer coefficient, A is the heat transfer area, ΔT is the temperature difference that results in heat flow. It can be stated from this equation that increased heat transfer can be achieved by: increasingΔT, increasing$A$, increasing h. A greater temperature difference ΔT can lead to increase the heat flow but ΔT is often limited by process or material constraints. Maximizing the heat transfer area (A) leading to increase in the size of heat exchanger which ultimately leads to unwanted increase in weight and cost. And Heat transfer enhancement can also be achieved by increasing the heat transfer coefficient h either by using more efficient heat transfer methods, or by improving the transport properties of the heat transfer material. So in order to overcome above problems, addition of Nano-particles in the base fluid is recommended for the purpose of improving heat transfer. Nano fluids are potential heat transfer fluids with enhanced thermodynamic physical properties and heat transfer performance which can be applied in many devices for better performances i.e. energy, heat transfer and other performances.

In order to take a combined effect of both addition of Nano particle and corrugated plates on heat exchanger performance, this project is focused on “Enhancement in heat transfer coefficient of water using Nano fluids for corrugated plate heat exchanger.”

2. LITERATURE SURVEY

R. K. Shah and D. P. 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. Maintaining the Integrity of the Specifications.

R. K. Shah and S. G. 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.

R. Saidur, S.N. Kazi, M.S. Hossain (2011) : the paper presents thermalphysical properties of nanoparticles suspended in refrigerant and lubricating oil of refrigerating systems were reviewed. Heat transfer performance of different Nano refrigerants with varying concentrations was reviewed. Pressure drop and pumping power of a refrigeration system with Nano refrigerants were obtained from different sources.

R.Saidur, K.Y.Leong, H.A. Mohammad (2011) : Nano Fluids are potential heat transfer fluids with enhanced thermo physical properties and heat transfer performance can be applied in many devices for better performances (i.e. energy, heat transfer and other performances). Recent researches have indicated that substitution of conventional coolants by Nanofluids appears promising. Specific application of Nanofluids in engine cooling, solar water heating, cooling of electronics, cooling of transformer oil, improving diesel generator efficiency, cooling of heat exchanging devices, improving heat transfer efficiency of chillers, domestic refrigerator-freezers, cooling in machining, in nuclear reactor and defense and space have been reviewed. Nano fluids have a much higher and strongly temperature-dependent thermal conductivity at very low particle concentrations than conventional fluids.

Murugesan M.P. and Balasubramani R. (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.

Vishal R. Naik, V.K. Matawala (2013) : had studied experimentally the effect of chevron angle with wide range of Reynolds Number on heat transfer characteristics of Gasketed oil-water PHE. Results showed that increase in mass flow rate of both the fluids increases overall heat transfer coefficient. In addition to this 60° chevron angles gives better heat transfer coefficient for wide range of Reynolds number.

T K S Sai Krishna, S G Rajasekhar, C Pravarakhya (2013) : For many industrial applications plate heat exchangers are demonstrating a large superiority over the other types of heat exchangers. In this they performed analysis using two different working fluids which are i.e. CO2 and R134a used only for refrigeration system. They got the results regarding heat transfer and also observed a very low pressure drop.

Masoud Asadi, Ramin Haghighi Khoshkhoo (2014) : had demonstrated effect of chevron angle on heat transfer coefficient, friction factor and pressure drop of corrugated PHE. From this study it was concluded that, optimal value of chevron angle is 60° and friction factor varies inversely with mass flow rate of both fluids in laminar and turbulent zone. This paper focuses on thermal design of corrugated PHE for one pass one arrangement and water-water heat transfer.

Raman Chouda (2014) : In this paper the thermal performance analysis of a counter flow heat exchanger is performed by varying the composition of Nanofluids 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.

N. Targui, H. Kahalerras (2014) : In this work a numerical simulation of Nanofluids flow in a double pipe heat exchanger provided with porous baffles. The hot Nanofluids 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.

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 Nanofluids 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.

Tambe Shahanwaj K, Pandhare Nitin T, Bardeskar Santosh J, Khandekar S.B (2015) : Compact heat exchangers are most widely used for heat transfer applications in industries. Plate heat exchanger is one such compact heat exchanger, provides more area for heat transfer between two fluids in comparison with shell and tube heat exchanger. Plate type heat exchangers are widely used for liquid-toliquid heat transfer applications with high density working fluids. This study is focused on use of plate type heat exchanger for water as a working fluid. This research work deals with experimental investigation of plate.

3. EXPERIMENTAL SETUP

To design a project that could be used to transfer heat from hot water in a heat exchanger to nanofluid stored in a separate tank and make temperature calibrations for the same by employing two thermocouples. Also, flow meters will be installed in the pipes carrying nanofluid to check its flowing rate. The complete system will be very dynamic and easy to use. Mechanical structure design is shown is Fig. 1. It consists of two flow loops, a heating unit to heat the nanofluid or the distilled water, and temperature measurement system. The two flow loops carries heated nanofluid or distilled water and the other cooling water. Each flow loop includes a pump with a flow meter, a reservoir and a bypass valve to maintain the required flow rate. The shell and tube heat exchanger is of stainless steel type 170 L, 550 mm x 550 mm x 550 mm and one 2000-watt capacity heater is fitted inside the hot water tank. The tube diameter is 2.4 mm with a tube wall thickness of 0.25 mm, having a designed heat transfer area of 0.05 m2. Four J-type thermocouples with removable bulbs are inserted on the heat exchanger to measure the bulk temperatures of inlet and outlet fluid streams. Two single phase centrifugal pumps of 0.5 HP and 2800 rpm are used.

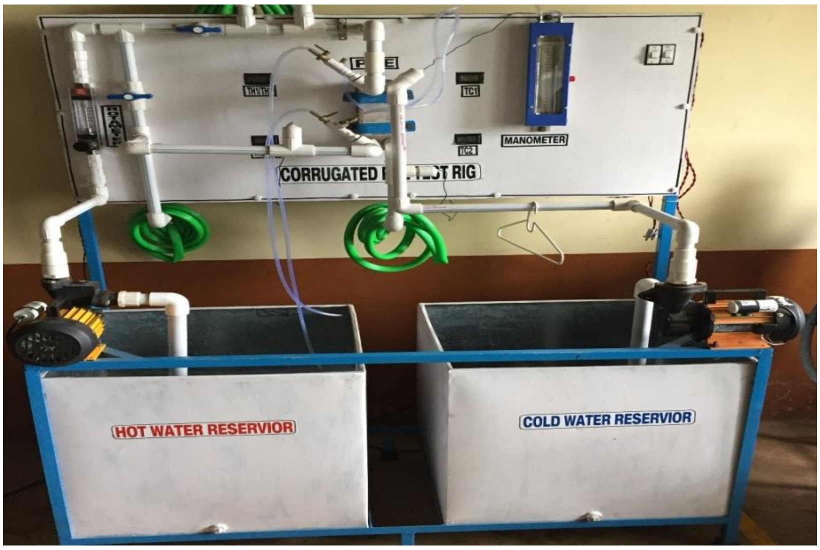

Fig.1. Actual Experimental Setup

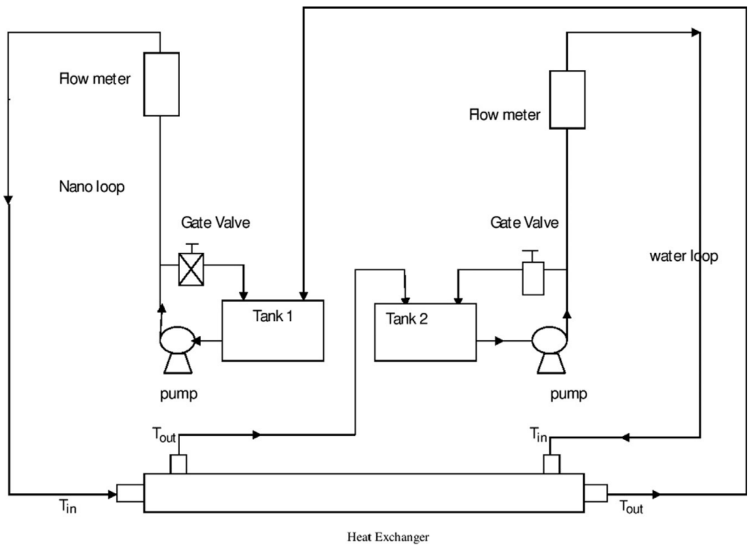

Fig.2. Test Rig

4. STEPS OF CALCULATION FOR PARALLEL FLOW WITH WATER:

4.1. Heat Duty:

$$Q = m_h C_{ph} \Delta T_h$$

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

4.2. Velocity of water:

a) For hot water

$$V_h = \frac{m_h}{A_h \rho_h}$$

b) For cold water

$$V_c = \frac{m_c}{A_c \rho_c}$$

4.3. Reynold Number:

a) For hot water:

$$Re_h = \frac{\rho_h V_h D_e}{\mu_h}$$

b) For cold water:

$$Re_c = \frac{\rho_c V_c D_e}{\mu_c}$$

4.4. Prandtl Number:

a) For hot water:

$$Pr_h = \frac{\mu_h c_{Ph}}{K_h}$$

b) For cold water:

$$Pr_c = \frac{\mu_c Cpc}{K_c}$$

4.5. Nusselt Number:

For hot water:

$$Nu_h = 0.662\ Re_h{}^{0.5} Pr_h{}^{0.33}$$

Convective heat transfer coefficient for hot water (h_h):

$$h_h = (0.662)\left(\frac{K_h}{D_e}\right) Re_h{}^{0.5} Pr_h{}^{0.33}$$

For cold water:

$$Nu_c = 0.662\ Re_c{}^{0.5} Pr_c{}^{0.33}$$

Convective heat transfer coefficient for cold water (h_c):

$$h_c = (0.662)\left(\frac{K_c}{D_e}\right) Re_c{}^{0.5} Pr_c{}^{0.33}$$

4.6. Logarithmic Mean Temperature Difference (LMTD):

$$\Theta_m = \left[\frac{[(T_{h1}-T_{c2})-(T_{h2}-T_{c2})]}{ln\left[\frac{(T_{h1}-T_{c2})}{(T_{h2}-T_{c2})}\right]}\right]$$

4.7. Overall heat transfer coefficient:

$$Q = UA\,\Theta m$$

4.8. Effectiveness

(ε- NTU):

$$\mathbf{C_h = m_h \times C_{ph}}$$

$$C_c = m_c \times C_{pc}$$

$$NTU = \frac{UA}{Cmin}$$

$$R = \frac{C_{min}}{C_{max}}$$

$$(\varepsilon)_{parallel\ flow} = \frac{1-exp[-NTU(1+R)]}{1+R}$$

5. STEPS FOR CALCULATION FOR COUNTER FLOW WITH WATER:

5.1. Heat Duty:

$$Q = m_h C_{ph} \Delta T_h$$

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

5.2. Velocity of water:

a) For hot water

$$V_h = \frac{m_h}{A_h \rho_h}$$

b) For cold water

$$V_c = \frac{m_c}{A_c \rho_c}$$

5.3. Reynold Number:

a) For hot water:

$$Re_h = \frac{\rho_h V_h D_e}{\mu_h}$$

b) For cold water:

$$Re_c = \frac{\rho_c V_c D_e}{\mu_c}$$

5.4. Prandtl Number:

For hot water:

$$Pr_h = \frac{\mu_{h}\, c_{Ph}}{K_h}$$

For cold water:

$$Pr_c = \frac{\mu_c C_{p_c}}{K_c}$$

5.5. Nusselt Number:

a) For hot water:

$$Nu_h = 0.662\, {Re_h}^{0.5} {Pr_h}^{0.33}$$

Convective heat transfer coefficient for hot water (h_h):

$$h_h = (0.662)\left(\frac{K_h}{D_e}\right) {Re_h}^{0.5} {Pr_h}^{0.33}$$

b) For cold water:

$$Nu_c = 0.662\, {Re_c}^{0.5} {Pr_c}^{0.33}$$

Convective heat transfer coefficient for cold water (h_c):

$$h_c = (0.662)\left(\frac{K_c}{D_e}\right) {Re_c}^{0.5} {Pr_c}^{0.33}$$

5.6. Logarithmic Mean Temperature Difference (LMTD):

$$\Theta_m = \left[\frac{[(T_{h1}-T_{c2})-(T_{h2}-T_{c1})]}{ln\left[\frac{(T_{h1}-T_{c2})}{(T_{h2}-T_{c1})}\right]}\right]$$

5.7. Overall heat transfer coefficient:

$$Q = UA\, \Theta m$$

5.8. Effectiveness (ε- NTU):

$$C_h = m_h \times C_{ph}$$

$$C_c = m_c \times C_{pc}$$

$$NTU = \frac{UA}{Cmin}$$

$$R = \frac{C_{min}}{C_{max}}$$

$$(\varepsilon)_{counter\ flow} = \frac{1 - exp[-NTU(1-R)]}{1 - R\, exp[-NTU(1-R)]}$$

6. PRESSURE DROP & FRICTION FACTOR DETERMINATION:

Area $=(\pi/ 4 \times d^2)$

Velocity, $v = (Q/A)$

Pressure drop, $\Delta p = (\rho_{hg} - \rho_{water}) \times g \times \Delta h$

Friction factor f= $(\Delta p \times d)/(2 \times L \times \rho \times v^2)$

7. OBSERVATION TABLES

Part A - It consists of observation table for parallel and counter flow arrangement with water.

Observation table for parallel flow with water

Sr.no	Mass flow rate (Kg/sec)		Hot water temperature °C		Cold water temperature °C		Pressure Drop
	m_h	m_c	T_{h1}	T_{h2}	T_{c1}	T_{c2}	Δp
1	0.11	0.25	46.8	36.7	33.9	36	0.025
2	0.12	0.25	55	38.2	29.7	35.9	0.0328
3	0.15	0.25	48	37.2	31.8	35.8	0.042
4	0.19	0.25	48.2	38.7	32.1	36.8	0.052
5	0.20	0.25	51.7	40.8	32.1	38.3	0.0625

Observation table for counter flow with water

Sr.no	Mass flow rate (Kg/sec)		Hot water temperature °C		Cold water temperature °C		Pressure Drop
	m_h	m_c	T_{h1}	T_{h2}	T_{c1}	T_{c2}	Δp
1	0.11	0.25	47.6	34.4	31.8	36.7	0.028
2	0.12	0.25	52.7	36.7	33.4	39.1	0.035
3	0.15	0.25	48.5	24.5	16.7	25.7	0.048
4	0.19	0.25	48.7	38.7	34.2	39.2	0.060
5	0.20	0.25	45.5	37.8	34.8	39.3	0.070

Part B - It consists of observation table for parallel and counter flow arrangement with water and Nano-fluids

Observation table for parallel flow with water and Nano-fluids

Sr.no	Mass flow rate (Kg/sec)		Hot water temperature °C		Cold water temperature °C		Pressure Drop
	m_h	m_c	T_{h1}	T_{h2}	T_{c1}	T_{c2}	Δp
1	0.11	0.25	58.8	34.8	18.1	26.8	0.031
2	0.12	0.25	55.42	27.21	20.9	29.2	0.042
3	0.15	0.25	54.8	31.8	24	33.1	0.053

4	0.19	0.25	49.1	31.24	26.2	32.9	0.061
5	0.25	0.25	32.1	31.28	28.9	33.8	0.089

Observation table for counter flow with water and Nano-fluids

Sr.no	Mass flow rate (Kg/sec)		Hot water temperature ºC		Cold water temperature ºC		Pressure Drop
	mh	mc	Th1	Th2	Tc1	Tc2	∆p
1	0.11	0.25	57.7	36.8	19	28.8	0.03
2	0.12	0.25	57.4	29.8	22.4	31.8	0.036
3	0.15	0.25	56.8	32.8	25.5	34.3	0.045
4	0.19	0.25	51.8	33.5	28	34.66	0.061
5	0.25	0.25	45.1	39.8	30.7	35.8	0.088

8. RESULT TABLES

Results for parallel flow arrangement (water as working fluid)

Sr. No	Units	1	2	3	4	5
m_h	Kg/sec	0.11	0.12	0.15	0.19	0.20
m_c	Kg/sec	0.25	0.25	0.25	0.25	0.25
T_{h1}	^{0}C	46.8	55	48	48.2	51.7
T_{h2}	^{0}C	36.7	38.2	37.2	38.7	40.8
T_{c1}	^{0}C	33.9	29.7	31.8	32.1	32.1
T_{c2}	^{0}C	36	35.9	35.8	36.8	38.3
Q	W	4641.86	8777	6890.83	7700.89	9489.56

Re_h	-	561.18	593.30	793.18	1026.46	990.88
h_h	W/m^2K	4113.22	4361.03	4860.07	5496.06	6204
Θ_m	^0C	4.1868	9.5917	6.04	6.6449	8.30
U	W/m^2K	1998	2526.85	3199	3399	3900
R	-	0.4501	0.51	0.6209	0.7661	0.8235
ε	-	0.62	0.61	0.57	0.52	0.4540

Results for counter flow arrangement (water as working fluid)

Sr. No	Units	1	2	3	4	5
m_h	Kg/sec	0.11	0.12	0.15	0.19	0.20
m_c	Kg/sec	0.25	0.25	0.25	0.25	0.25
T_{h1}	^0C	47.6	52.7	48.5	48.7	45.5
T_{h2}	^0C	34.4	36.7	24.5	38.7	37.8
T_{c1}	^0C	31.8	33.4	16.7	34.2	34.8
T_{c2}	^0C	36.2	39.1	25.7	39.2	39.3
Q	W	6066.9	8389	13916.7	8106.26	6701.73
Re_h	-	550.09	577.7	633.085	1027.4	1064.75
h_h	W/m^2K	4093.87	4110.5	4498.39	5514.08	5661.05

Θ_m	^0C	5.94	7.26	13.97	6.68	4.408
U	W/m^2K	2472.68	2600	2851	3527.26	4514.67
R	-	0.44	0.50	0.5547	0.77	0.83
ε	-	0.8	0.7	0.65	0.54	0.5

Results for parallel flow arrangement (Nanofluids as working fluid)

Sr. No	Units	1	2	3	4	5
m_h	Kg/sec	0.11	0.12	0.15	0.19	0.25
m_c	Kg/sec	0.25	0.25	0.25	0.25	0.25
T_{h1}	^0C	58.8	53.9	48.8	48.1	46.6
T_{h2}	^0C	33.3	34.3	36.8	36	36.9
T_{c1}	^0C	22.5	25.9	28.9	29.11	29
T_{c2}	^0C	29.1	31.4	32.7	33.6	34.2
Q	W	11534.5	9611.99	7936.66	7715.5	8717
Re_h	-	581.71	616.74	650.23	846.735	1044.24
h_h	W/m^2K	4258.61	4420.8	4585	5194.5	6325
Θ_m	^0C	13.61	10.94	8.29	8.26	8.22

U	W/m^2K	2300	3591.8	4477.14	4547.92	4486.6
R	-	0.439	0.4795	0.52	0.674	0.83
ε	-	0.63	0.62	0.58	0.536	0.47

Results for counter flow arrangement (Nanofluids as working fluid)

Sr. No	Units	1	2	3	4	5
m_h	Kg/sec	0.11	0.12	0.15	0.19	0.25
m_c	Kg/sec	0.25	0.25	0.25	0.25	0.25
T_{h1}	^0C	57.7	57.4	56.8	51.8	45.1
T_{h2}	^0C	36.6	29.8	32.8	33.5	39.8
T_{c1}	^0C	19	22.4	25.5	28	30.7
T_{c2}	^0C	28.8	31.8	34.3	34.66	35.8
Q	W	14389	13715.8	13042.6	11571.3	10099
Re_h	-	561.19	630.89	700.60	832.77	964.73
h_h	W/m^2K	4234.5	4438.28	4643	5618	6522
Θ_m	^0C	15.33	14.41	13.49	9.68	5.86
U	W/m^2K	4002.4	4528.19	4721	5299	5967.76

R	-	0.44	0.48	0.52	0.66	0.80
ε	-	0.83	0.76	0.73	0.71	0.69

8.1. Results

8.1.1. Results for Parallel Flow Arrangement

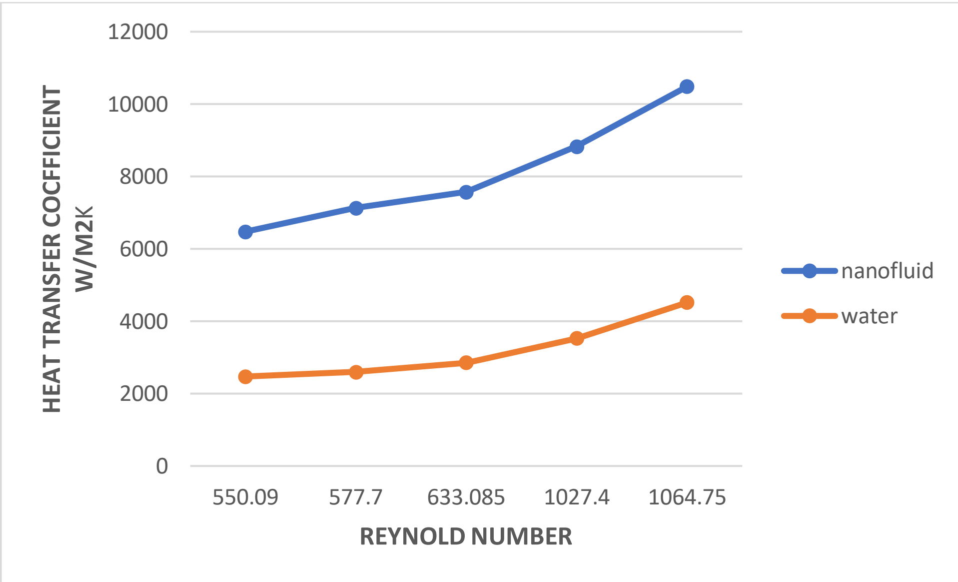

Fig.3. Variation of Heat transfer coefficient with Reynolds number

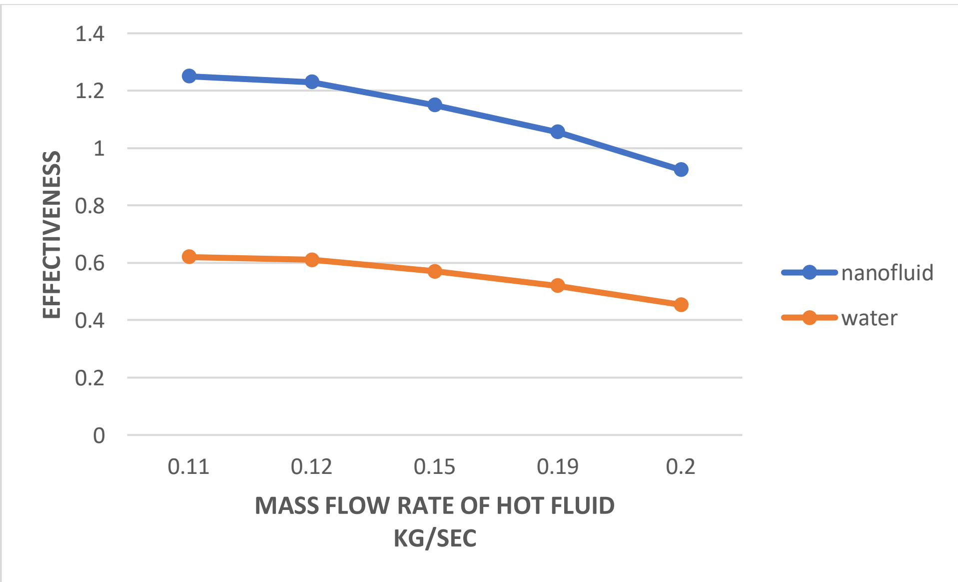

Fig.4. Variation of Effectiveness with Mass flow rate of hot fluid

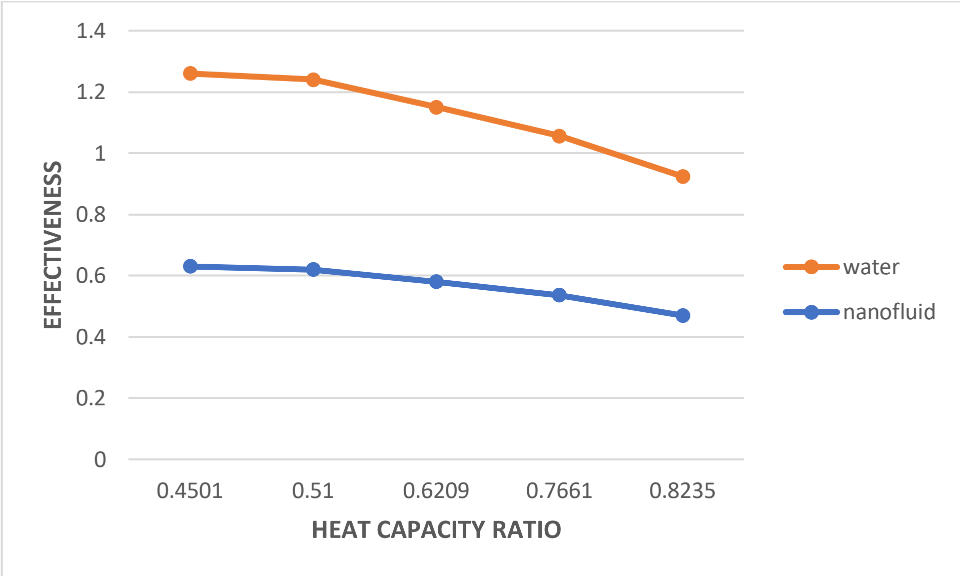

Fig.5. Variation of Effectiveness with Heat capacity ratio

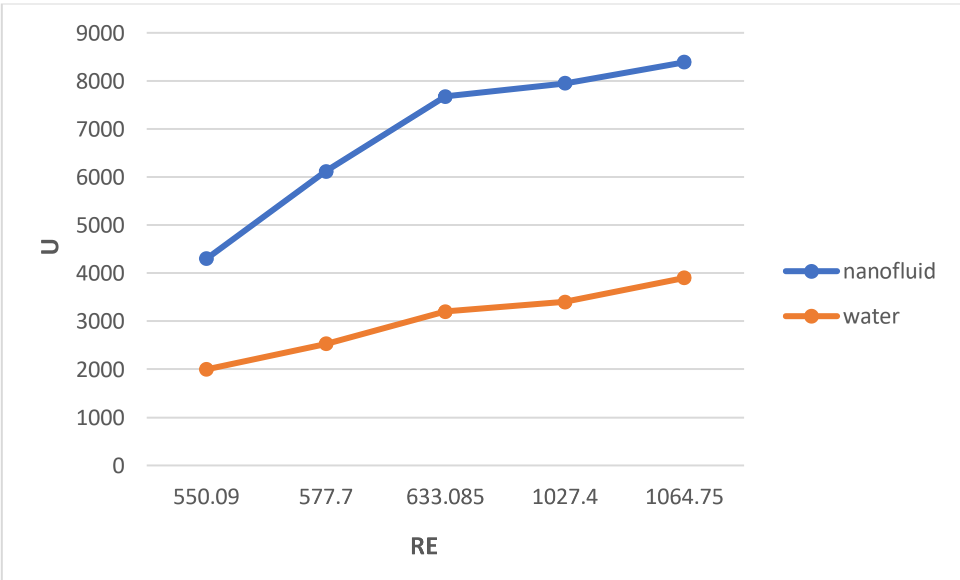

Fig.6. Variation of Overall heat transfer co-efficient with Reynold Number

8.1.2. Results for counter flow arrangement

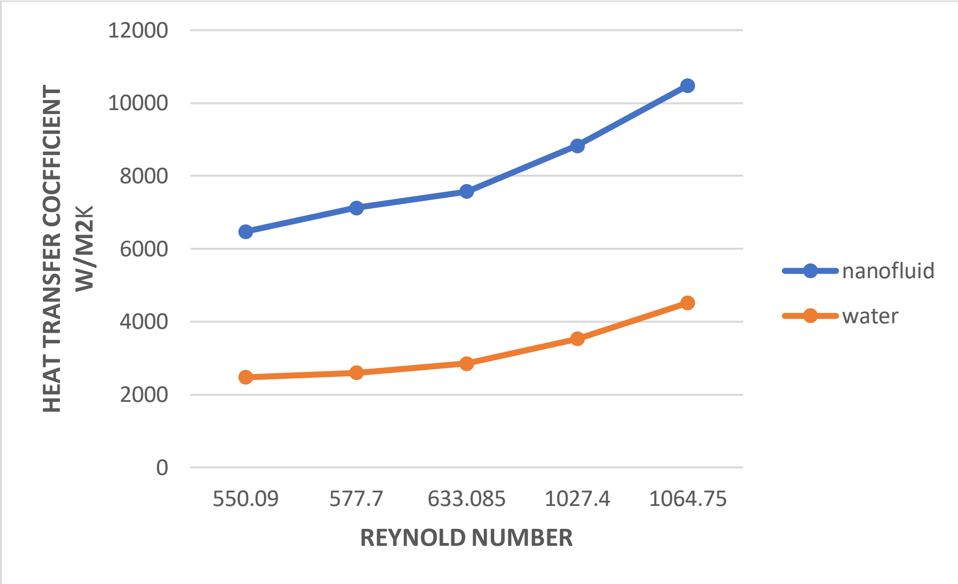

Fig.7. Variation of Heat transfer coefficient with Reynold Number

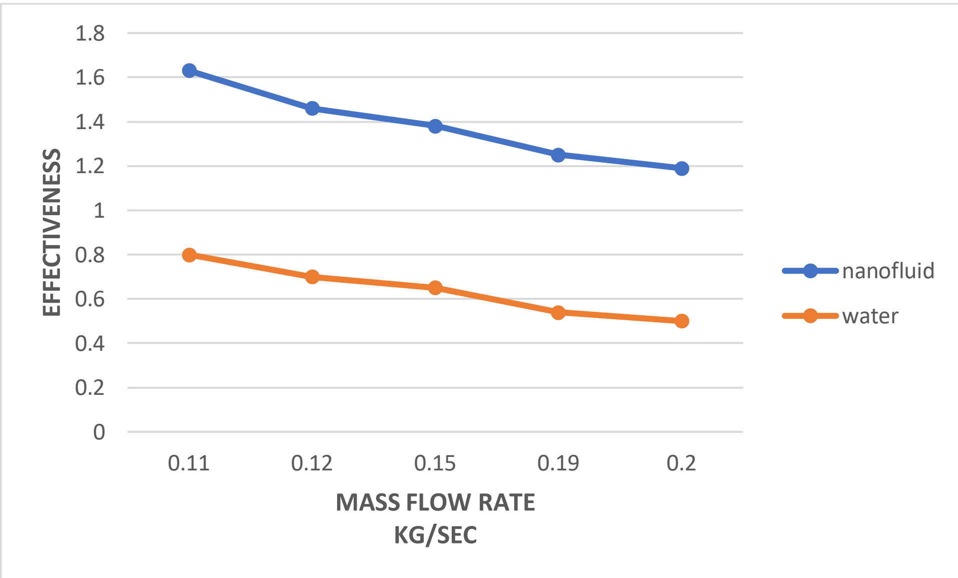

Fig.8. Variation of Effectiveness with Mass flow rate of hot fluid

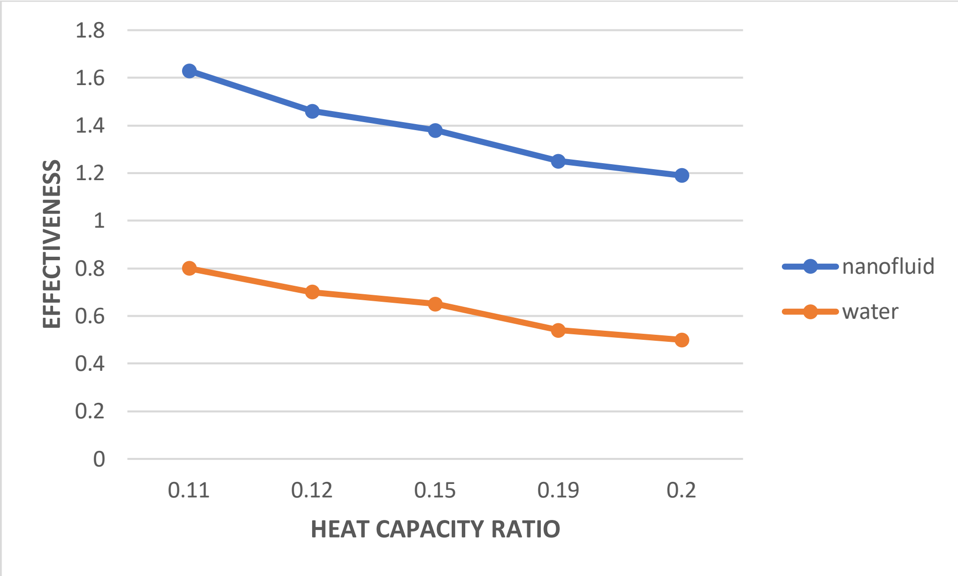

Fig.9. Variation of Effectiveness with Heat capacity ratio

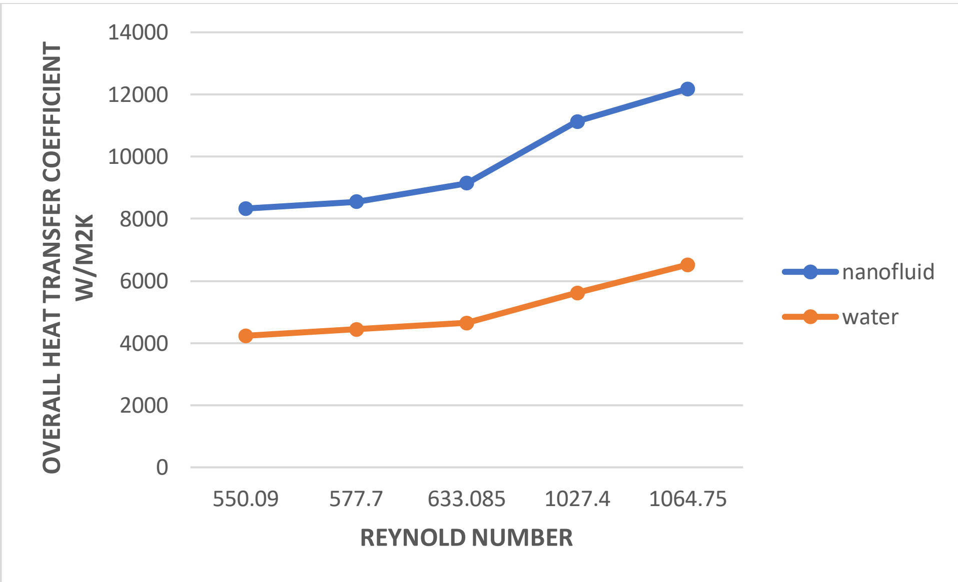

Fig.10. Variation of Overall heat transfer coefficient Vs. Reynold number

9. 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:

- 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.
- Effectiveness of heat exchanger decreases with increase in mass flow rate of hot fluid. Maximum effectiveness for parallel flow arrangement is 0.64 and that of for counter flow arrangement is 0.82 (for water as a working fluid).
- Exchanger effectiveness considerably increases when nanoparticles are added into the base fluid. Maximum effectiveness obtained is 0.66 and 0.86 for parallel and counter flow arrangements respectively (for Nanofluids as working medium).
- Increase in effectiveness of PHE by addition of 0.2% Nano fluids by volume into base fluid is obtained as 2% for this study.

- Increase in overall heat transfer coefficient Nano fluid is obtained as 30% with respect to water as working fluid.

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