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Investigation of Heat Transfer Rate of water by addition of Al_2O_3 Nanofluid

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ABSTRACT

This article aims to report an investigation of an experimental study of forced convective heat transfer and flow characteristics of a nanofluid consisting of different volume concentrations of Al2O3 nanofluid (0.2–2)% and water flowing in a parallel and counter flow arrangement. The size of of about 10 nm Al2O3 nanoparticles is used in this research. It has been observed in the resultthat the convective coefficient of heat transfer is slightly higher than the base liquid. The rate of heat transfer coefficient of mass flow when increased with the volume concentration of the Al2O3 nanofluidthe coefficient of heat transfer increases. However increase in the nanofluids viscosity leads to increase in friction factor due to increase in volume concentration. The improvement of rate of Heat transfer of a nanofluid is done by improving the thermo-physical properties and by increasing the mass fraction of nanoparticles. This results in increase in the viscosity of the nanofluid due to increase in the number of nanoparticles in the base fluid.This research is to investigate the forced convective rate of heat transfer for an heat exhanger with various Al2O3-water nanofluids. The effects of various mass fractions of the nanofluid (0.2–2)% onthe heating efficiency of the heat exchanger were analyzed over time, and results indicate that the heat exchanger with the 2.0% Al2O3-waternanofluid has the highest heating efficiency.

Index Terms- : Al2O3, Al2O3-waternanofluid,Parallel Flow, Counter Flow.

1. Introduction

A few years ago, instead of particles of micrometre size, particles of nanometre size (normally between 1 to 100 nm) are used to disperse in base liquids, and they are called nanofluids. Undetectable by the human eye, nanoparticles can exhibit significantly different physical and chemical properties to their larger material counterpart. Nano fluid is popularsince it was first introduced by Choi in 1995. Recently nanofluids have attracted great interest beacause of enhanced heat transfer rate and thermal conductivity.

To enhance the heat transfer characteristics and thermal conductivity, nanoparticles used to mix should not react with base fluids and remain stable fo a very long time.

Recently, various active and passive heat transfer technologies have been developed due to the increasing demand of heat transfer enhancement in many industrial fields. Nanofluids are heat transfer working media with uniform, stable, and high heat conduction. Nanofluids are prepared by mixing non-metallic or metallic into traditional heat transfer fluids such as glycol, water, etc. nanofluids are extensively used in heat transfer systems such as heat exchangers. For forced convective heat transfer, nanoparticles can significantly change the thermal-physical properties (thermal conductivity, viscosity, density, and so on) of the base fluid. To investigate various rates of heat transfer of nanofluids, the concentration of nanoparticles and effects have been investigated in various thermal systems, such as cooling system pool boiling system , heat exchanger , and transformer system . Selvam et al.experimentally studied heating efficiency in an automobile cooling radiator, and they found that 0.5 vol %. Madhesh et al.experimentally investigated heat transfer characteristics and pressure drop of Ag-water and CuOwaternanofluids in a heat exchanger. They found that compared with the base fluid, heat transfer coefficients of 1.0 vol.% Ag-water and CuO-water nanofluids increased by 52.0% and 27.6%, respectively. Karthikeyan et al.analyzed the nanofluid stability on boiling heat transfer system. It has been observedthat the stabilityof ethanol-based nanofluid of 80 ppm multi-wall carbon nanotubes has a 7% rate of cooling higher than the pure ethanol (base fluid). Sozen € et al.numerically and experimentally studied heat transfer enhancement of aqueous clinoptilolitenanofluid in a heat pipe. Their results indicated that the heat pipe efficiency under 400 W heating load increased from 86.2% to 94.5%. In Miroshnichenkoet a’s

Nomenclature

Cp	specific heat, J/kg K
d	nanoparticle diameter, m
D	tube diameter, m
f	friction factor
U	overall heat transfer coefficient, W/m^2 K
K	thermal conductivity, W/mK
$\dot{m}$	mass flow rate, L/s
Nu	Nusselt number
Re	Reynolds number
Pe	Peclet number
Pr	Prandtl number Q heat transfer, W
T	temperature, °C
V	mean velocity, m/s
Greek symbols	
υ	kinematic viscosity, m^2/s
$\varnothing$	volume concentration, %
ρ	density, kg/m^3
α	thermal diffusivity, m^2/s
μ	viscosity, kg/ms
ΔT	°C
Subscripts	
wi	water inlet
wo	water outlet
ni	nanofluid inlet
no	nanofluid outlet
in	inlet
out	outlet
n	nanofluid
f	base fluid
ρ	nanoparticles

Investigation heat transfer performance of a Al2O3–water nanofluid in an multiple porous layersand open cavity. They figured that comparison of base fluid, a 2% amplification of heat efficiency was gained when 0.2% Al2O3–water nanofluid was used. results showed that for the same Reynolds number, nanofluid showed higher pressure drops than the basic fluid. Results showed that the electrical conductivity of this nanofluid was enhanced by increasing volume fraction. In addition, compared to pure oil, the electrical conductivity of 4 vol.% iron oxide oil nanofluid reached to 27.4 nS/m. Shen et al. reported an effect of ultrasonic waves on heat transfer of various Al2O3 nanofluids in a pool boiling system. They indicated that the highest heat transfer efficiency reached to 128%, when ultrasonic waves were used. Based on hemispherical surface Ciloglu investigated rate of heat transfer of the Sio2-water naofluid. Results indicated that compared2550 to the base fluid, the critical heat flux increased by 45% for 0.1 vol.% SiO2-water nanofluid. Etedali et al. investigated the effect of different surfactants on pool boiling heat transfer coefficient of a SiO2-water nanofluid on a copper surface. In their report nonionic and anionic surfactants resulted in the smallest and the largest rises in the boiling heat transfer of the nanofluids. Similarly Moldoveanu et al. analyzed the thermal conductivity of an Al2O3-SiO2 hybrid nanofluid. Results showed that higher volume fractions of SiO2 nanoparticles in the hybrid nanofluids resulted in higher thermal conductivity. However, many studies focused on experimental and numerical simulations in a partially heated rectangular/annular enclosure of natural convection heat transfer. Few researchers have studied the heat transfer characteristics of nanofluids in an electric heater. Electric heaters are used to supply proper indoor heating instead of a central heating system, which can reduce carbon emissions. In addition, a heater with water as the base fluid is easy to recycle without pollution. This experiment is conducted to investigate the performance of heatingof a mixture of Al2O3 nanoparticles and water in the test rig. Related results will be discussed at various nanofluid concentrations. This paper provides an effective method to enhance the heat transfer of electric heater.

2. Literature Survey

In many heat transfer applications Al2O3 water based nanofluids are used.Nanofluids that are prepared by the ultrasonic vibrators remain unstable for a very long time. Therefore researchers have to carefully prepare nanofluid which will give consistent result and will be stable for long time. Heat transfer characteristics and thermal conductivity of Al2O3 nanofluid are not consistent in many studies conducted. Following might be the limitations.

1. Surfactant.
2. Optimizing PH
3. Temperature for different nanofluid.
4. Surface modification.

Not many studies are available to understand the enhancement of thermal conductivity of the nanofluid by some modifications in shape and size. Due to this the rate of heat transfer of Al2O3 has adverse effects. It is observed in many literature surveys for same volume fraction different degree of improvement.The techniques used for measuring thermal conductivity canalso alter the values. Thermal conductivity of nanofluid at concentration upto 400K was recorded but has not yet been reported.

There are not many reports available that provide the information regarding the effect of temperature on viscosity at higher concentrations. The temperature of nanofluids viscosityhas to be carefully maintained at high concentrations to study the behaviour of nanofluids. The factors affecting the thermal conductivity improvement of Al2O3 nanofluid and its preparation methods is summarised. From the studies it has been realised that the nanofluids are must to improve the thermal efficiency of different systems.

3. Experimental setup

In this project a system is designed that is capable to transfer heat to nanofluid through heat exchanger from hot fluid stored in tank and record temperature differences with the help of four thermocouples. To check the flow rate of the fluid flow meters are installed in the pipes carrying nanofluids.The system used in this experiment is not very complex and easy to operate on.The actual experimental setup is shown is Fig. 1. which consists of a heating unit to heat the Al2O3 water nanofluid and temperature measurement system. The nanofluid and the cooling water flows through the arrangement of pipes. Arrangement of pipes is connected are connected toa pump with a flow meter, a bypass valveand a reservoir to maintain the required flow rate. The plate heat exchanger is of stainless steel type 170 L, 550 x 550 x 550 mm and one 2000-watt capacity heater is fitted inside the hot water tank. The diameter of tube is 2.4 mm with a wall thickness of 0.25 mm, having a heat transfer designed area of 0.06 m2. To measure the inlet and outlet temperatures four J-type thermocouples are attached to the heat exchange. Two single phase centrifugal pumps of 0.5 HP and 2800 rpm are used.

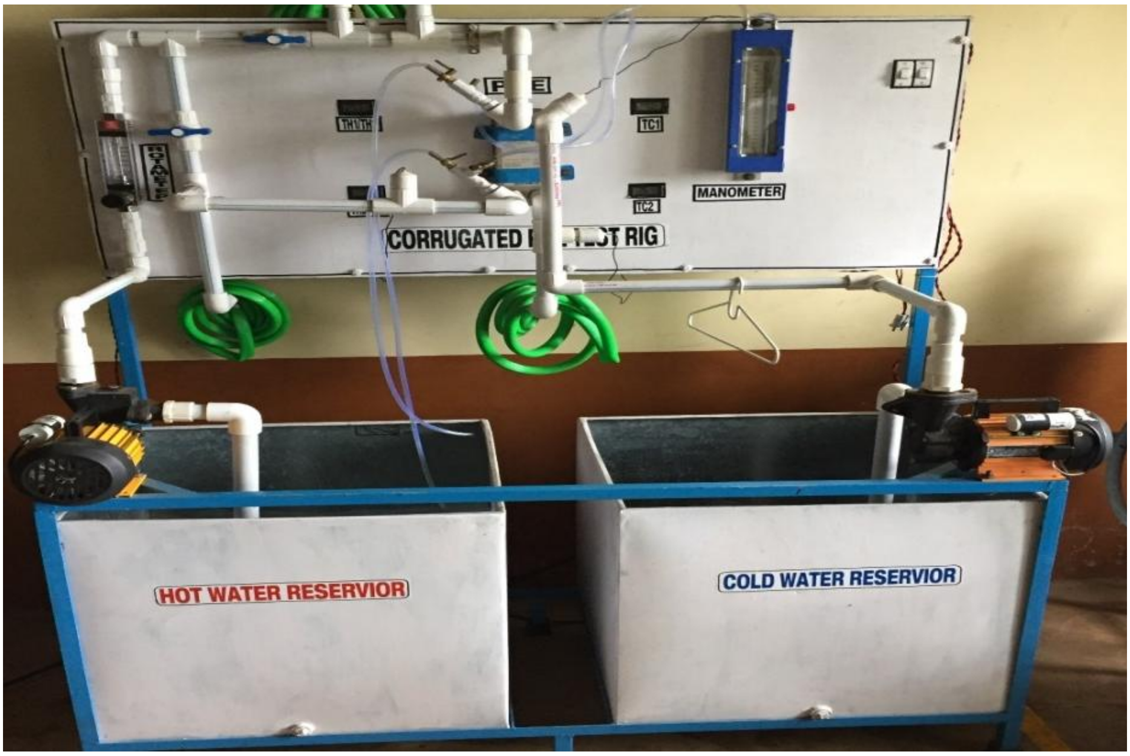

Fig.1. Actual Experimental Setup

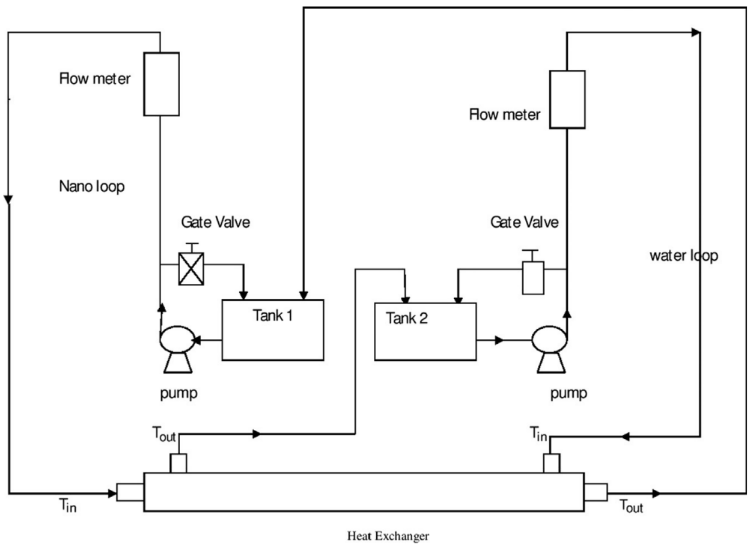

Fig.2. Test Rig

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:

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 (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:

a) For hot water:

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

b) 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):

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

c) For cold water:

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

Convective heat transfer coefficient (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):

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

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

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

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

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

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 ºC		Cold water temperature ºC		Friction Factor
	m_h	m_c	T_{h1}	T_{h2}	T_{c1}	T_{c2}	f
1	0.11	0.25	47.8	37.7	34.9	37	0.025
2	0.12	0.25	56	39.2	30.7	36.9	0.0328
3	0.15	0.25	49	38.2	32.8	36.8	0.042
4	0.19	0.25	49.2	39.7	33.1	37.8	0.052
5	0.20	0.25	52.7	41.8	33.1	39.3	0.0625
6	0.25	0.25	57.6	43.9	31.3	40.6	0.074

Observation table for counter flow with water

Sr. No	Mass flow rate (Kg/sec)		Hot water temperature ºC		Cold water temperature ºC		Friction Factor
	m_h	m_c	T_{h1}	f	T_{c1}	T_{c2}	f
1	0.11	0.25	48.6	35.4	32.8	37.2	0.028
2	0.12	0.25	53.7	37.7	34.4	40.1	0.035
3	0.15	0.25	49.5	25.5	17.7	26.7	0.048
4	0.19	0.25	49.7	39.7	35.2	40.2	0.060
5	0.20	0.25	46.5	38.8	35.8	40.3	0.070
6	0.25	0.25	50.7	33	21.7	35.2	0.081

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 ºC		Cold water temperature ºC		Friction Factor
	m_h	m_c	T_{h1}	Δh	T_{c1}	T_{c2}	F
1	0.11	0.25	59.3	34.3	23.4	29.7	0.032
2	0.12	0.25	54.2	35.3	26.5	32.4	0.035
3	0.15	0.25	49.3	36.7	29.2	33.7	0.041
4	0.19	0.25	48.5	37	29.55	34.6	0.047
5	0.20	0.25	47.6	37.9	30	35.2	0.062

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		Friction Factor
	m_h	m_c	T_{h1}	Δh	T_{c1}	T_{c2}	f
1	0.11	0.25	58.7	37.8	20	29.8	0.04
2	0.12	0.25	58.4	30.4	23.4	32.8	0.046
3	0.15	0.25	57.7	33.7	26.5	35.3	0.055
4	0.19	0.25	52.08	34.5	29	35.66	0.071
5	0.20	0.25	46.9	34.8	31.7	36.1	0.092

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}	^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	6205
Θ_m	^0C	4.1868	9.5917	6.04	6.6449	8.30
U	W/m^2K	2000	2527.85	3200	3400	3900
R	-	0.4401	0.50	0.6109	0.7761	0.8335
ε	-	0.62	0.61	0.56	0.51	0.4440

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	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	W	6067.9	8388	13917.7	8107.26	6702.73
Re_h	-	550.09	577.7	633.085	1027.4	1064.75
h_h	W/m^2K	4094.87	4111.05	4499.39	5515.08	5662.05
Θ_m	^0C	5.95	7.27	13.98	6.69	4.4080
U	W/m^2K	2473.68	2600	2852	3528.26	4515.67
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	0.20
m_c	Kg/sec	0.25	0.25	0.25	0.25	0.25
T_{h1}	^0C	59.3	54.2	49.3	48.5	47.6
T_{h2}	^0C	34.3	35.3	36.7	37	37.9
T_{c1}	^0C	23.4	26.5	29.2	29.55	30
T_{c2}	^0C	29.7	32.4	33.7	34.6	35.2
Q	W	11533.5	9511.922	7926.66	7815.5	8817
Re_h	-	582.71	617.74	652.23	848.735	1045.24
h_h	W/m^2K	4328.76	4422.9	4588	5196.7	6429
Θ_m	^0C	13.63	10.97	8.32	8.29	8.25
U	W/m^2K	2300	3591.9	4478.16	4548.95	4487.8
R	-	0.439	0.4795	0.52	0.675	0.83
ε	-	0.65	0.64	0.60	0.534	0.49

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	.19	0.20
m_c	Kg/sec	0.25	0.25	0.25	.25	0.25
T_{h1}	^0C	58.7	58.4	57.7	52.08	46.9
T_{h2}	^0C	37.8	30.4	33.7	34.5	34.8
T_{c1}	^0C	20	23.4	26.5	29	31.7
T_{c2}	^0C	29.8	32.8	35.3	35.66	36.11
Q	W	14388	13715.8	13043.6	11573.3	10099
Re_h	-	562.19	632.895	702.60	835.77	967.94

h_h	W/m^2K	4235.59	4440.30	4647	5623	6526
Θ_m	^0C	15.35	14.43	13.51	9.70	5.89
U	W/m^2K	4002.43	4528.20	4724	5300	5967.77
R	-	0.44	0.48	0.52	0.66	0.80
ε	-	0.85	0.78	0.75	0.73	0.71

8.1. Results

8.1.1. Results for Parallel Flow Arrangement

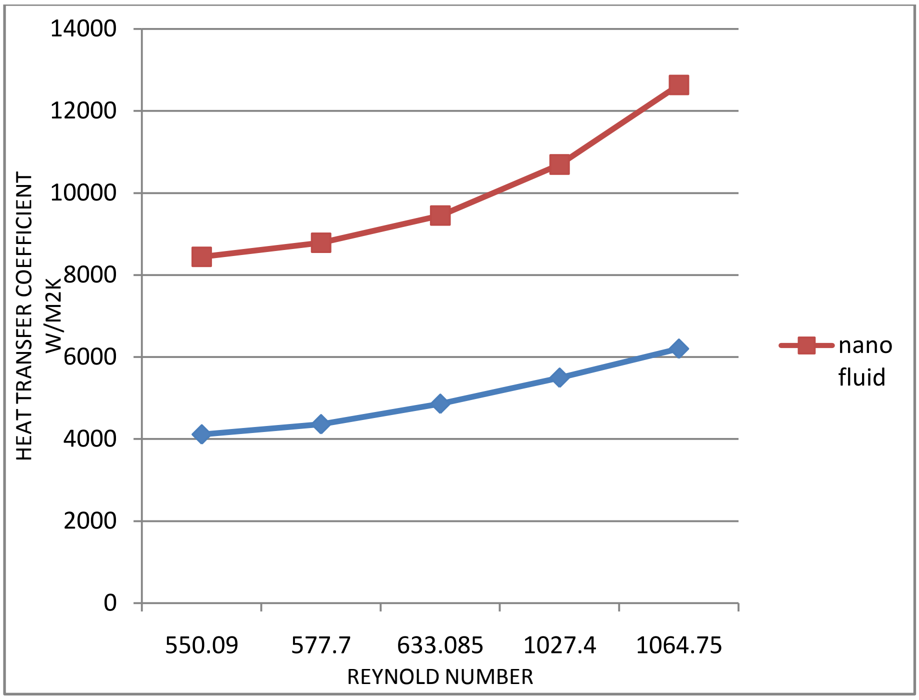

Fig.3.Variation of Heat transfer coefficient with Reynolds number

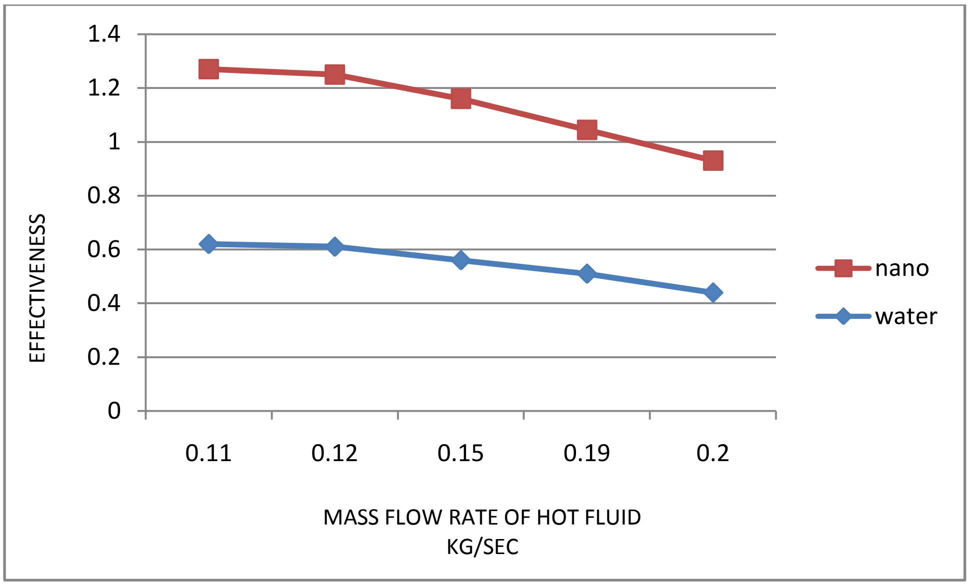

Fig.4. Mass flow rate and effectiveness of hot fluid variation

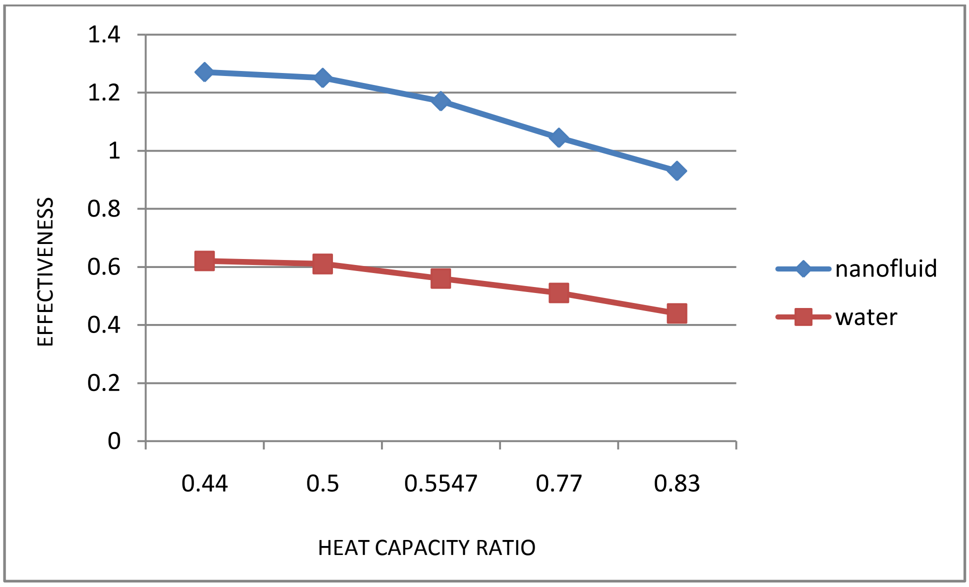

Fig.5. Variation of Effectiveness with Heat capacity ratio

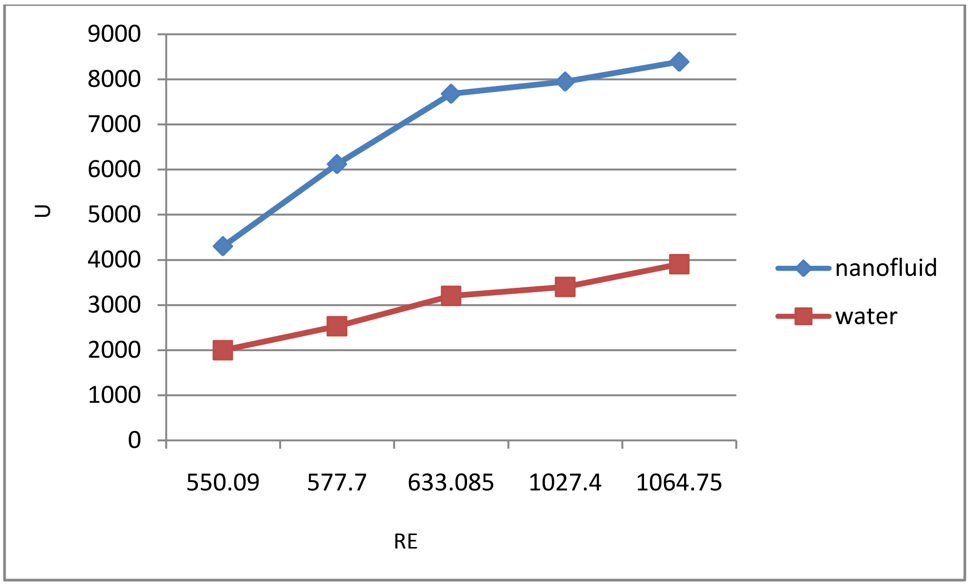

Fig.6.Reynold Number and overall heat transfer coefficient variation

8.1.2. Results for counter flow arrangement

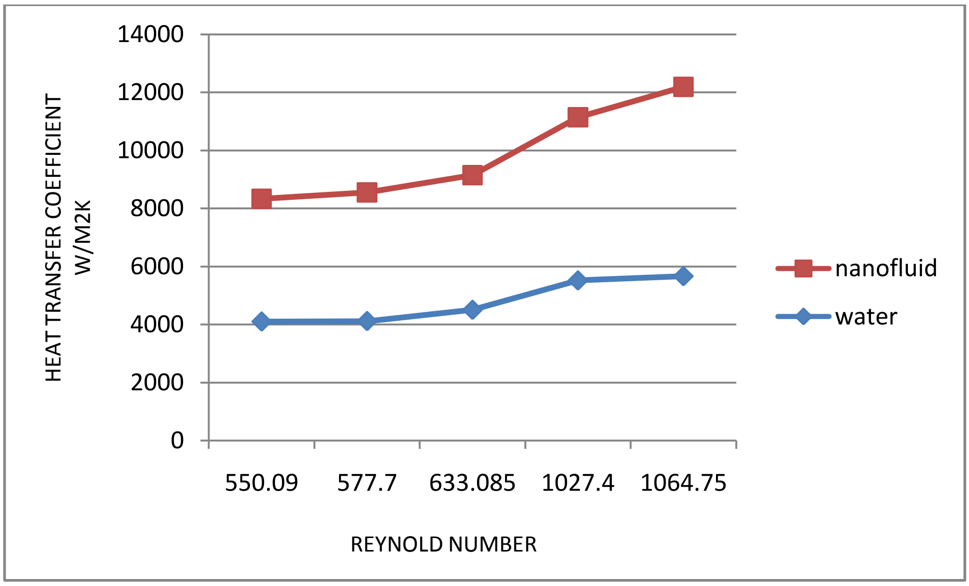

Fig.7.Variation of Heat transfer coefficient with Reynold Number

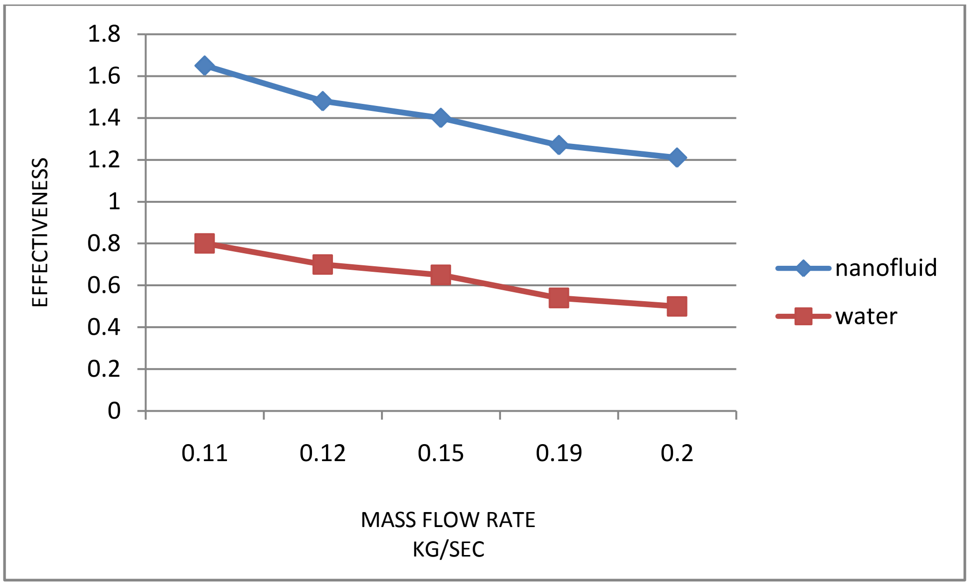

Fig.8. Mass flow rate and Effectiveness of hot fluid variation

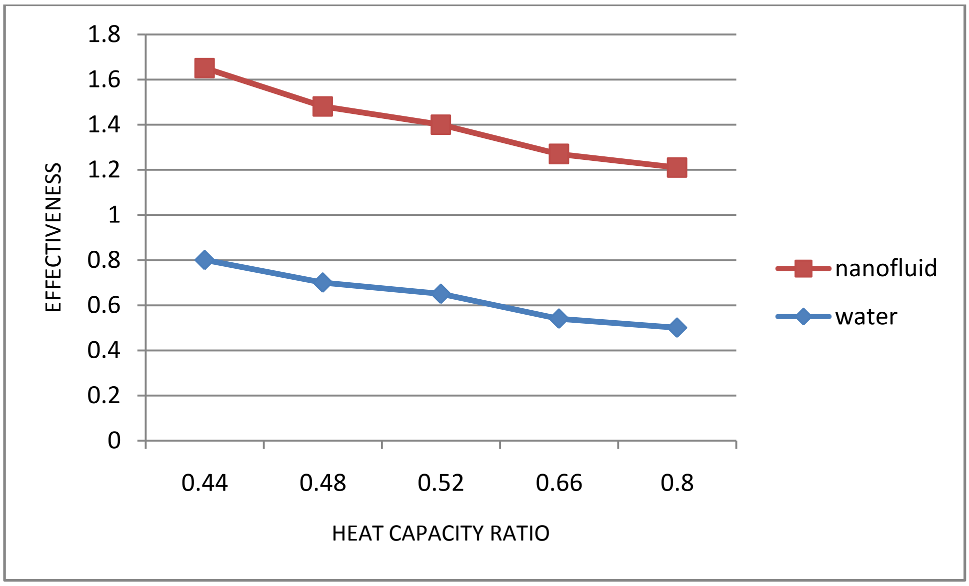

Fig.9. Variation of Effectiveness with Heat capacity ratio

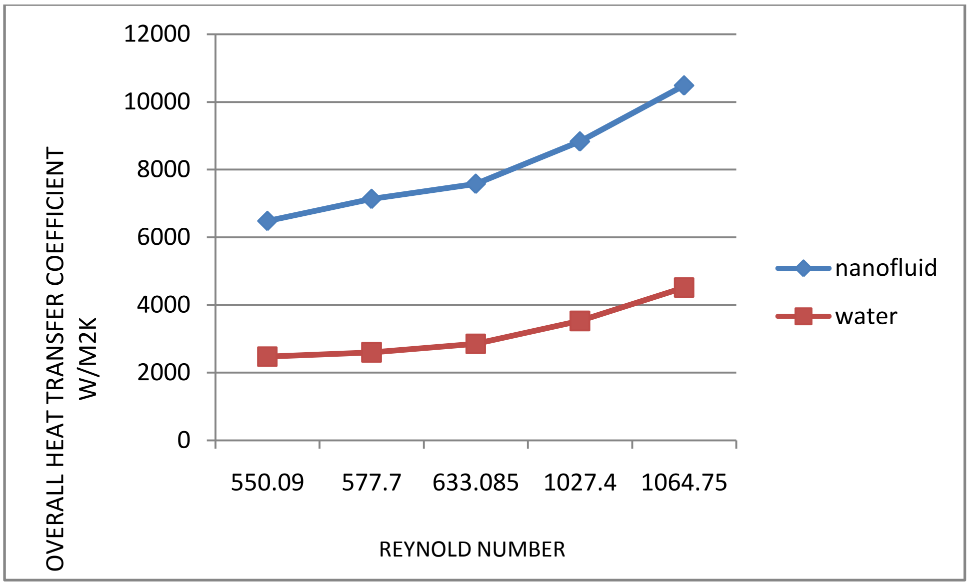

Fig.10.Variation of Overall heat transfer coefficient Vs. Reynold number

9. Conclusion

In this study the experiments are carried out in turbulent conditions to understand the flow characteristics and convective rate ofheat transfer of Al2O3 nanofluid flowing in parallel and counter flow arrangement of the test rig has been experimentally investigated. Reynolds number and hysteresis phenomenonis determined with the help of the effect of particle concentration. Important conclusions are obtained are summarized as follows:

- 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.
- With increase in mass flow rate of hot fluid effectiveness of heat exchanger decreases. 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).
- Exchanger effectiveness considerably increases when nanoparticles are added into the base fluid. Maximum effectiveness obtained is 0.64 and 0.84 for parallel and counter flow arrangements respectively (for Nanofluids as working medium).
- Increase in effectiveness of PHE by addition of 0.4% Nano fluids by volume into base fluid is obtained as 4% for this study.
- Increase in overall heat transfer coefficient Nano fluid is obtained as 32% with respect to water as working fluid.
- In order to increase exchanger effectiveness, it is required to reduce the mass flow rate of cold fluid for Nanofluids as working medium.
- 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 thermo physical properties as well as plate geometry.
- Pressure drop is lower due to addition of nano particles to water.

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