

EXPERIMENTAL AND MATHEMATICAL PERFORMANCE ANALYSIS OF A CORRUGATED PLATE HEAT EXCHANGER USING CuO NANO FLUIDS

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ABSTRACT

In experimental analysis and investigated study to carried out to the heat transfer rate and their characteristic exergy loss effectiveness and friction factor of water based CuO nanofluid as a coolant in corrugated plate type heat exchanger. The analysis has been carried out for a 1-1 pass heat exchanger under parallel and counter flow situations, with different weight concentration CuO nanofluid. The effect of nanofluid (CuO in water .5, 1.in volume concentration and 70gm in weight %) and water as coolants as on heat transfer and those required properties of the nano fluids wear measured. The required pump power increased with increased in nanofluid weight concentration for better heat transfer rate in effectiveness in lower consumption of power or LMTD reduced in lower rate of 0.5 - 1LPM. The average heat transfer rate coefficient has been found to reduce by 45-60% when the angle of inclination of corrugation angle of inclination plate of heat exchanger is 45°.

Keywords: Mass Flow Rate, Enhanced Heat Flow, Reynolds Number, Nusselt Number, PHE

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NOMENCLATURE A= total heat transfer area, (m^2), C_p =specific heat, ($\text{J kg}^{-1} \text{K}^{-1}$)

h =convective heat transfer coefficient, ($\text{Wm}^{-2} \text{K}^{-1}$), L = plate length, (mm), N number of corrugated plates, Nu = Nusselt number, P =pressure, Pr = Prandtl number, Q = heat transfer rate, (W), Re =Reynolds number, t =corrugation pitch, mm, U =overall heat transfer coefficient, ($\text{Wm}^{-2} \text{K}^{-1}$), V =volume flow rate, (L min^{-1}) W =plant width, (mm), E = exergy loss (W), m = mass flow rate, A = area of cross section, i = inlet, P = pressure, F = friction factor, L =length (m), R = capacity ratio, Q = heat transfer rate(W).

1. INTRODUCTION

A heat exchanger is the device used to exchange the thermal heat energy between two or more different fluids, in this mechanism of a heat transfer are one of the most important engineering tools to save the energy in different applications, such as chemical and food industries mechanical automotive radiator etc. This design requirement is to saving energy and reduced to cost effectiveness, in globalised to enhance better performance rate. Nanofluids are mainly used as working fluids in heat transfer equipment, such as heat exchangers. As pointed out by Kumar et al. [1], any debate about heat transfer and heat exchangers may not be able to draw correct and logical understandings and conclusions without referring to nanofluids, and vice versa. Heat exchangers can be indexed as indirect contact type and direct contact type. Tubular heat exchangers are efficient process equipment that produce significantly improved heat transfer with high fluid flow rates. Plate heat exchangers are usually arranged of a stack of thin corrugated metal plates, with apertures at the corners to supply channels for two fluid systems, allowing heat transfer between the two fluid media. Nowadays, nanofluids have been widely used in solar energy systems, heat exchangers, automobile radiators, electronic chips, etc. However, especially in a solar energy system, nanofluids have a great potential [2], some practical limitations and enormous challenges [3]. Michael and Iniyan [4] improved the thermal performance of photovoltaic thermal collectors using CuO-water nanofluids. They found that the thermal efficiency increased up to 45.76% for a CuO-water nanofluid with 0.05% volume fraction, compared to water at a mass flow rate of 0.01 kg/s. Chen et al. [5] analyzed the entropy generation and exergy destruction of a graphene nanoplatelets nanofluid in a ribbed triple-tube heat exchanger (RTTHX). It was observed that the total exergy destruction of the whole RTTHX was reduced when the nanoparticle mass fraction increased. Kumar and Chandrasekar [6] analyzed heat transfer characteristics of double helically coiled tube heat exchangers with MWCNT- water nanofluids based on a comparison of the Dean Number. It was found that for a 0.6 vol.% nanofluid, the Nusselt number and pressure drop increased by 30% and 10% at a dean number of 2000, respectively. Radka et al. [7] investigated convective heat transfer characteristics of helical copper tube heat exchangers under a constant wall temperature condition. They found that the average Nusselt number increased by 18.6% for 0.25 vol.% ZnO nanofluid. Bianco et al. [8] numerically investigated the heat transfer performance of an asymmetric heated channel filled with Al₂O₃ -water nanofluid. The increase of the Nusselt number was 15% for the 6% Al₂O₃ water nanofluid at a Reynolds number of 1000. Huang et al. [9] investigated thermal performance of Al₂O₃ and Al₂O₃-MWCNT hybrid nanofluids in a chevron plate heat exchanger. They proposed a correlation to predict all the experimental data within an error band of $\pm 10\%$. Bhatt ad et al. [10] pointed out that increasing the volume ratio of MWCNT nanoparticles in an Al₂O₃ -MWCNT hybrid nanofluid was beneficial for the performance improvement of the plate heat exchangers. This Is very important role of industrial application and widely used to significant increase their thermal conductivity and eventually their heat transfer coefficient. their where efficient cooling is a strong need to reduced cost, and energy consumption and environmental impact of the system.

2. ABOUT COPPER OXIDE (CUO) NANO PARTICLE

COPPER OXIDE	Description
Purity	99.9%
Average Particle Size	30-50nm
SSA	60-80m ² /g
Molecular Weight	79.549 g/mol
Molecular Formula	CuO
Melting Point	1326°C
Bulk Density	2.7g/cm ³
CAS NO.	1317-38 ⁻¹
Physical Form	Powder
Morphology	Spherical
Colour	Black

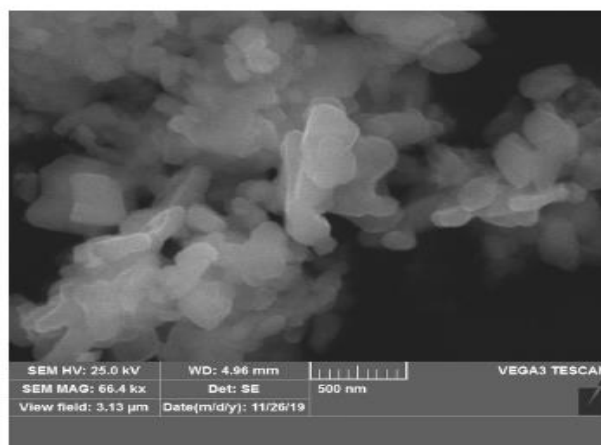
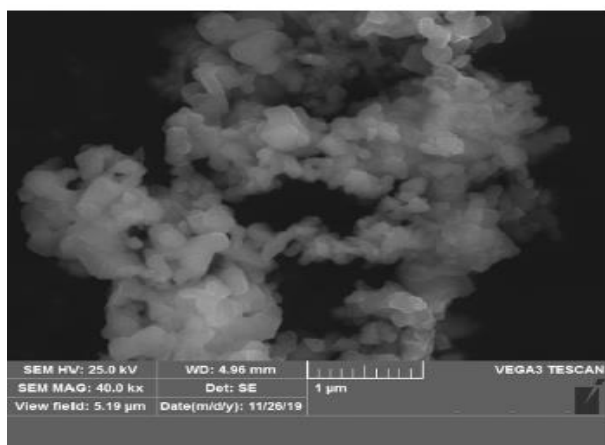
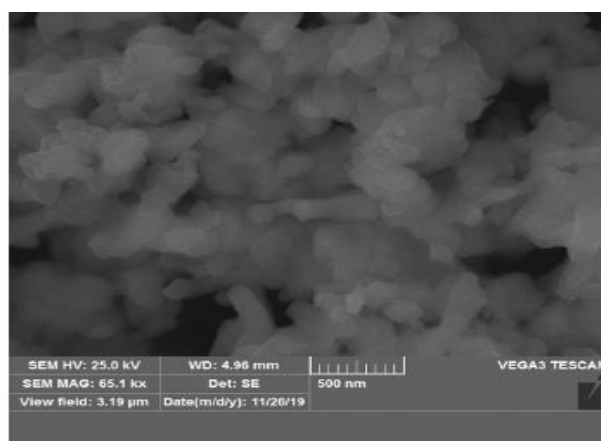
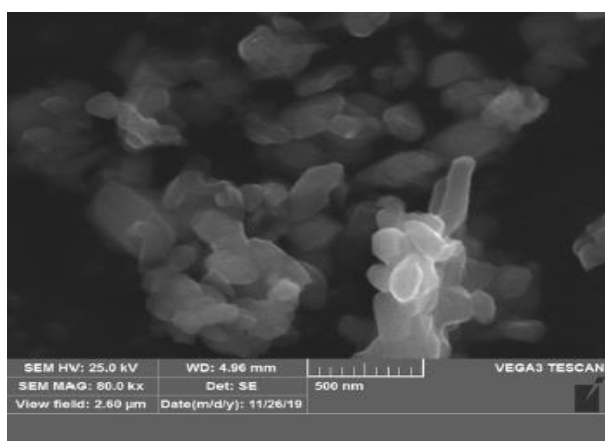


Fig. 1. Scanning electron microscope (SEM) images CuO Nano Particle

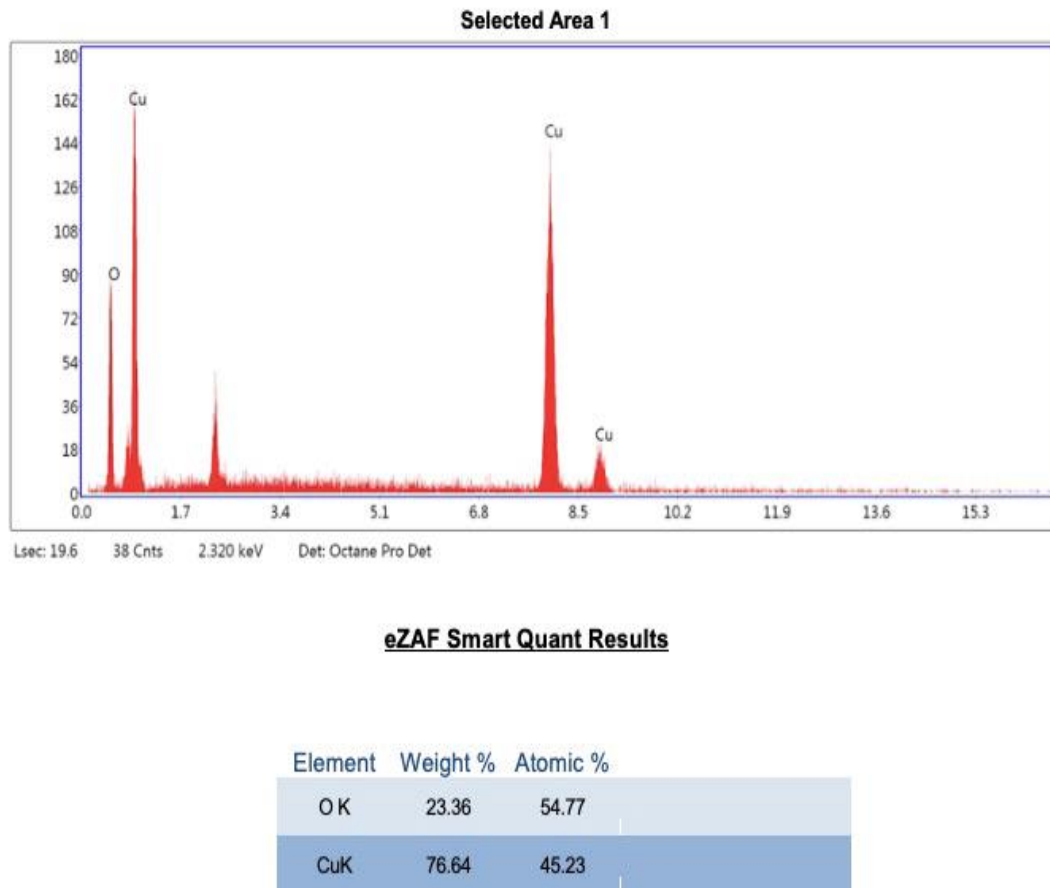


Fig. 2.eZAF smart Quant Results CuO nano particle

3. THERMOPHYSICAL PROPERTIES OF COPPER OXIDE (CUO) WATER FLUID

wt. in gm	Ψ_v	$C_p(\text{J/Kg-K})$	$\mu (\text{mPa. s})$	$\rho(\text{kg/m}^3)$	$K(\text{W/m.K})$	Re	Pr	Nu
water	Nil	4180	0.789	995.7	0.615	29543.6	5362.634	8446.829268
70	0.0259	3950	0.9503447	1044	0.69	25717.65	5440.379	7528.695652
140	0.0518	3960	1.0165766	1045	0.698	24065.12	5767.398	7442.406877
210	0.0777	3970	1.0894555	1046	0.699	22476.78	6187.608	7431.759657
280	0.1036	3980	1.169864	1047	0.6998	20951.89	6653.414	7423.26379
350	0.1295	3990	1.2588329	1049	0.7	19508.3	7175.347	7421.142857

4. EXPERIMENTAL SETUP

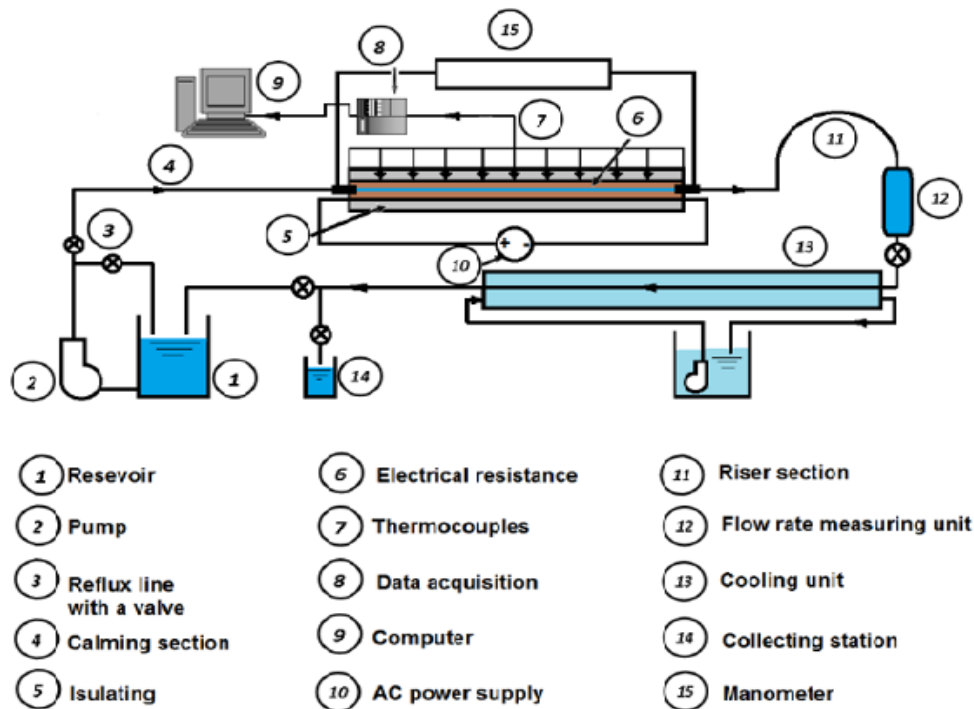


Fig .3 Schamatic view of the Exprimental set up

The picture of experimental set up are fabricated in 22-gauge GI SHEET. In this experiment the two different cases are applicable in the form of arrangement of set up parallel flow and counter flow of the experiment includes the water loop of the measurement system comprises a water tank are containing heater and flow rate are measure thermally insulated material like wood dust and hot fluid are flow in central channel and cold fluid are flow in upper and lower channel. This Is very important role of industrial application and widely used to significant increase their thermal conductivity and eventually their heat transfer coefficient. their where efficient cooling is a strong need to reduced cost, and energy consumption and environmental impact of the system. Specifications of the experimental setup. Set up Length 100 cm, set up Width 10 cm, the gap between corrugated plates 5 cm and angle of the plate is 45° , the plate material is GI of 22 gauges.

5. EXPERIMENTAL PROCEDURE

Experimental procedure Hot and cold water are two different cases in inlet and outlet temperature of water provided in the different channel first in inlet hot water temperature and outlet. The Cold-water loop compromise a water tank is considerable 40° to 70°C inlet temperature of hot water in parallel and counter flow arrangement. varying at 0.50 to 1 LPM and 0.50 to 1 LPM. The flow rate is measure by noting done in time for fixed and different volume of the fluid. In the hot and cold channel loop in a measurement system, thermocouple device is used for temperature measurement flow through the central corrugated channel to maintain the channel surfaces at approximately constant temperature the hot water loop comprises in a water tank, and heater and water tank with pump. In the whole system are thermally insulated with wood dust particle, and water flow are in measure LPM unit base.

6. NUMERICAL METHODOLOGY

Eqs. (1) and (2), respectively. Q_{ave} represents the average heat power of the hot fluid and the cold fluid and is calculated by Eq. (3)

$$Q_h = m_h C_{ph} (T_{h, in} - T_{h, out}) \quad (1)$$

$$Q_c = m_c C_{pc} (T_{c, out} - T_{c, in}) \quad (2)$$

$$Q_{ave} = (Q_h + Q_c) / 2 \quad (3)$$

The overall heat transfer coefficient (U) can be calculated by eqn (4)

$$U = Q_{ave} / A \cdot \text{LMTD} \quad (4)$$

where A represents the total heat transfer area (0.369m^2). LMTD represents the logarithmic mean temperature difference, and it is calculated by in this eqn (5)

$$\text{LMTD} = \theta_m = \frac{\theta_2 - \theta_1}{\ln \frac{\theta_2}{\theta_1}} \quad (5)$$

The heat transfer coefficient of the nanofluid (h_{nf}) can be calculated using Eq. (6):

$$\frac{1}{U} = \frac{1}{h_{nf}} + \frac{\delta}{\lambda} + \frac{1}{h_w} \quad (6)$$

where δ and λ represent the width and thermal conductivity of the corrugated plate, respectively. h_w is the convective heat transfer coefficient of the water. In this work, the heat transfer performance of the hot water was obtained by using the following equation

$$\text{Nu} = 1.615 \left[(f \cdot \text{Re} / 64) \text{Re} \cdot \text{Pr} \cdot D / L \right]^{1/3}$$

where Nu, Re, and Pr are Nusselt number, Reynolds number and Prandtl number, respectively. f is related to the flow characteristics and structure of the corrugated plate. D is the equivalent diameter, i.e., the plate depth for this work. The Reynolds number (Re) and Prandtl number (Pr) can be calculated by the following equations.

$$\text{Re} = \rho v D / \mu$$

$$\text{Pr} = \mu c_p / k$$

velocity (v), viscosity (μ) and thermal conductivity (k). The Nusselt number of the hot water can be calculated as follows:

$$\text{Nu} = h D / k$$

In this paper, density and specific heat of the nanofluid are calculated as suggested in Refs. [15,16]:

$$\rho_{nf} = (1 - \phi) \rho_w + \phi \rho_p$$

$$(\rho c_p)_{nf} = (1 - \phi) (\rho c_p)_w + \phi (\rho c_p)_p$$

7. RESULTS AND DISCUSSION

In this experiment the effectiveness of heat exchanger is calculated by 1-1 passes through the corrugated plate heat exchanger by using nanofluids.

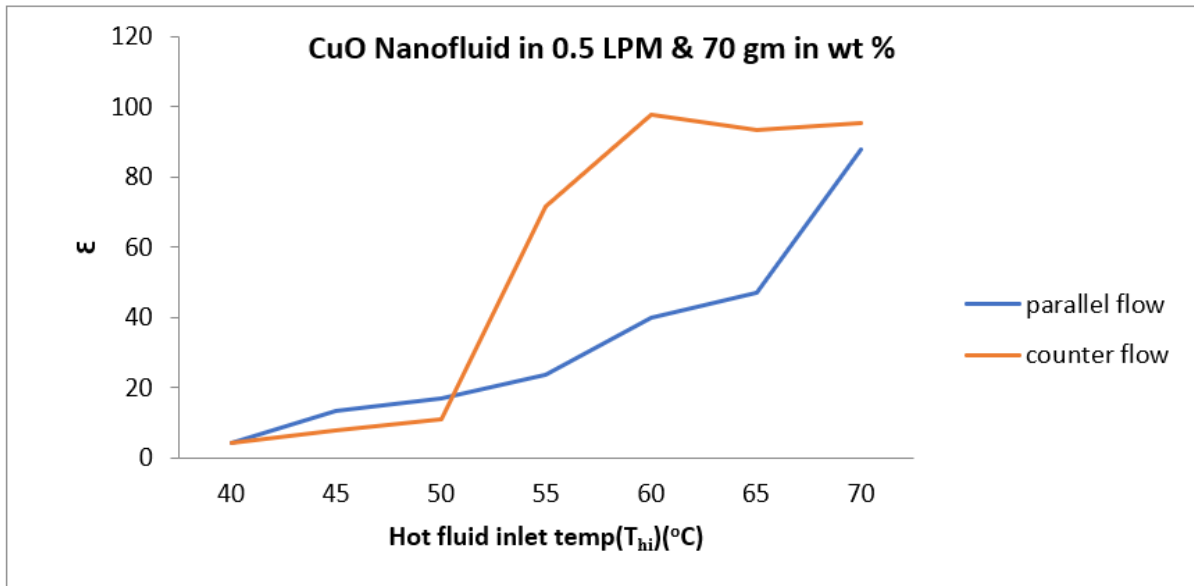


Fig :4 Variation of hot fluid inlet temperature at fixed mass flow rate at 70gm CuO.

In above figure fixed mass flow rate at different inlet hot fluid temperature observed that the effectiveness of heat exchanger is increased 25% in counter flow arrangement with minimum loss of energy.

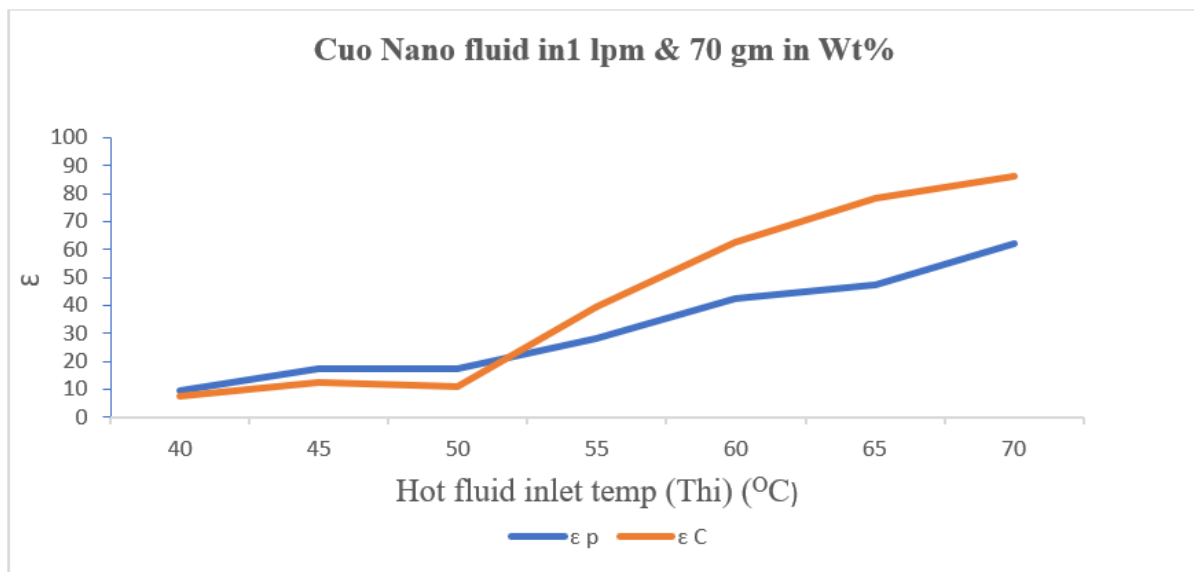


Fig :5 Variation of hot fluid inlet temperature at fixed mass flow rate at 70gm CuO.

In the above figure the variation of a graph between hot fluid inlet temperature the effectiveness will be increased in 32 % maximum temperature in 60°C in parallel and counter flow arrangement.

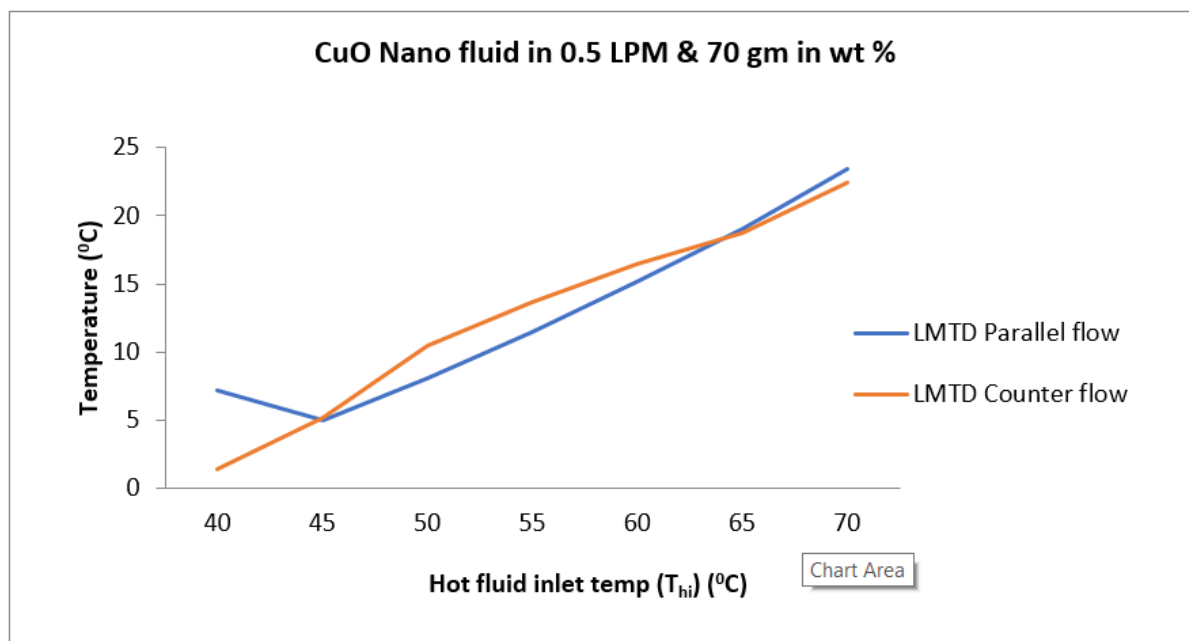


Fig :6 Temperature V/s Hot fluid inlet temperature at fixed mass flow rate at 70gm CuO.

In above figure temperature will be maximum in parallel flow arrangement with comparison to counter flow arrangement.

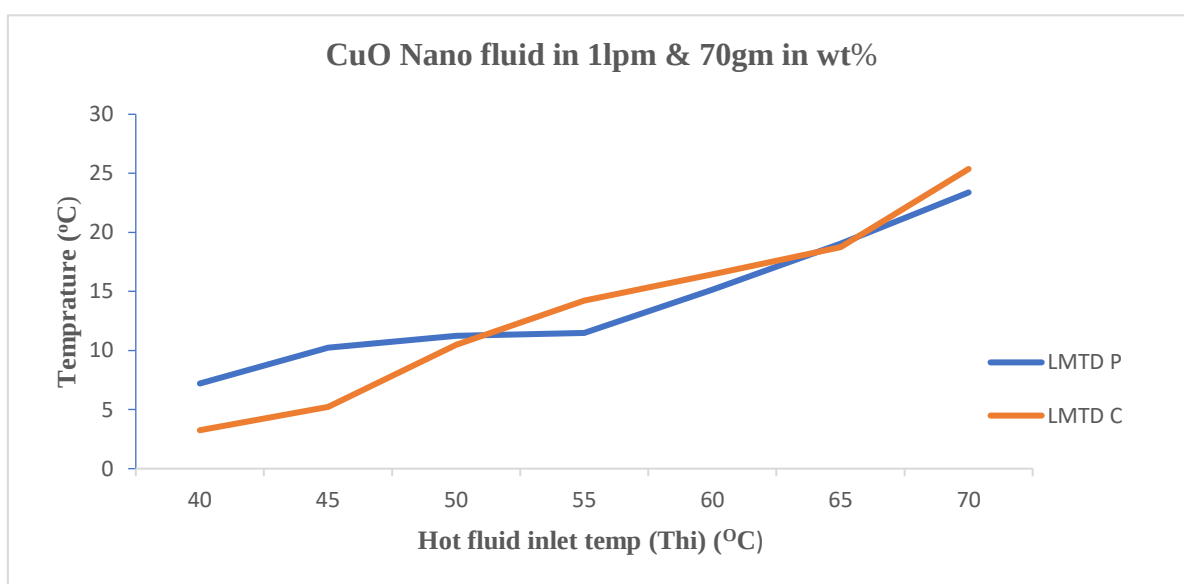


Fig :7 Temperature v/s Hot fluid inlet temperature at fixed mass flow rate 1lpm at 70gm CuO.

In above figure temperature will be maximum in parallel flow arrangement with comparison to counter flow arrangement with minimum loss of energy in 23 % and pressure drop is in maximum in counter flow then in parallel.

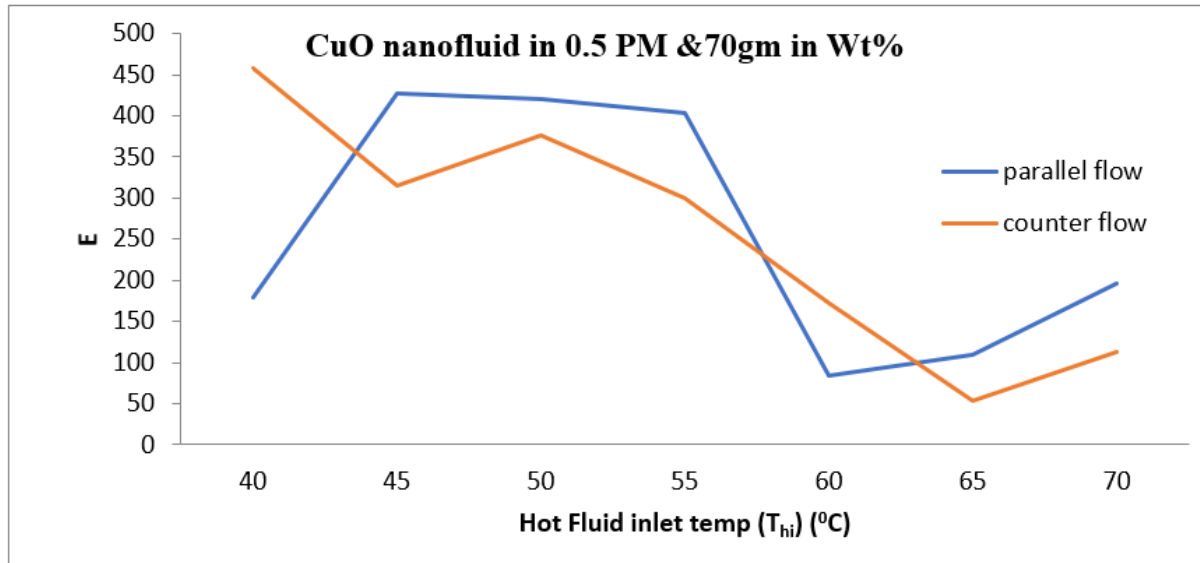


Fig :8 Exergy v/s Hot fluid inlet temperature at fixed mass flow rate 0.5 lpm at 70gm CuO.

In above figure is observed that the exergy losses in maximum 45°C and minimum loss 60°C in parallel flow arrangement with compression to counter flow.

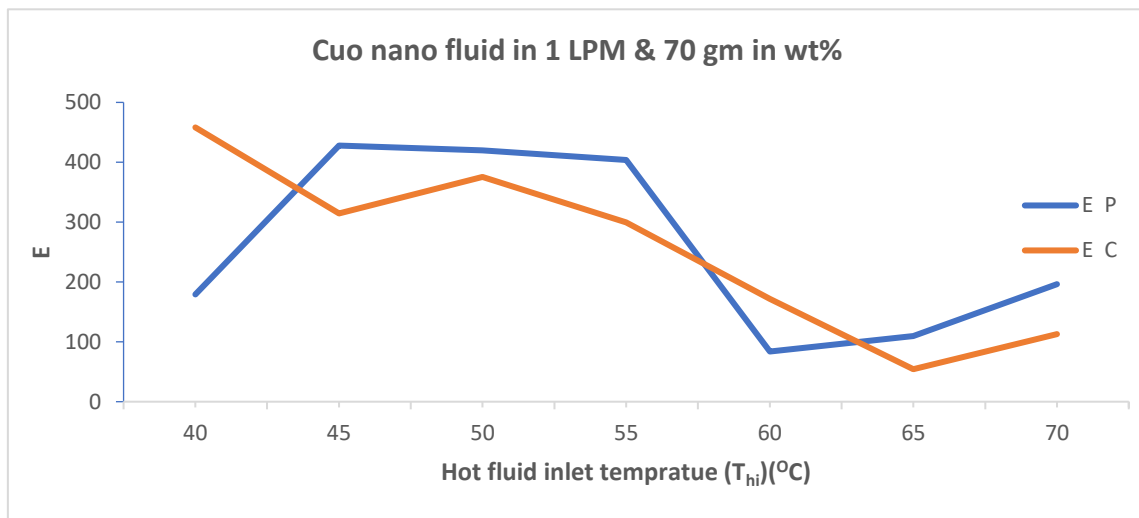


Fig :9 Exergy v/s Hot fluid inlet temperature at fixed mass flow rate in 1 lpm at 70gm CuO.

In above figure is observed that the exergy losses in maximum 42°C and minimum loss 60°C in parallel flow arrangement with compression to counter flow in fixed in 1 lpm in mass flow rate, and weight concentration.

8. CONCLUSION

In this study, in order to increase the performance of a plate heat exchanger by using CuO nano particle in different mass flow rate (0.5 to 1 lpm) and different weight concentration.

In thermo physical properties will be more effective in CuO nano fluid with compression to base fluid, the performance of plate heat exchanger at different input heat load was measured in counter flow and parallel flow arrangement.

Exergy analysis was performed to find its behaviour in parallel and counter flow arrangement and losses is 17.5% in parallel flow then in counter flow arrangement. Effectiveness is more effective in counter flow arrangement like 58%. In effect of temperature T_{hi} from 40°C to 70°C is more significant on T_{h2} when plane cold water is used, as volume % of CuO nano particle in weight basis 70 gm

The initial temperature of hot fluid becomes less or no significant. Exergy analysis was performed to find its behaviour in parallel and counter flow arrangement, exergy loss was found 9.6% more in parallel flow then in counter flow arrangement.

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