SSRG International Journal of Mechanical Engineering
ISSN: 2348-8360/ https://doi.org/10.14445/23488360/IJME-V12I9P103

Volume 12 Issue 9, 29-36, September 2025

Original Article

Optimization of Heat Flow Parameters of Corrugated Plate Heat Exchanger with CuO Nanoparticle/Water as Working Fluid

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Received: 04 July 2025 Revised: 05 August 2025 Accepted: 06 September 2025 Published: 30 September 2025

Abstract *- The research focuses on the thermal performance of Wavy Corrugated Plate Heat Exchangers (WCPHEs), which offer superior heat transfer capabilities and more efficient heat recovery than conventional flat designs. A comparative study of heat flow parameters at different corrugated angles is done. The study focuses on and emphasizes the effect of 0˚,30º, 40 °, 50 °, and 60 ° corrugation angles. Less research has been conducted regarding the impact of corrugated angles on thermal flow parameters of nanofluids. This study experimentally investigates the optimal angle with CuO nanofluid exhibiting enhanced thermal conductivity and minimal pressure drop. A comparative study of heat flow parameters at different corrugated angles is carried out.*

Due to their effectiveness, these exchangers are commonly used in pharmaceuticals, brewing, dairy, and food processing sectors. The corrugation is the key geometric factor influencing heat transfer due to enhanced surface area and the ability to promote flow turbulence. Experimental investigations were carried out by varying corrugation angles from (0° to 60°) and flow rates (2,3,4 lpm) and Copper Oxide nanoparticle concentration in water (Φ). An attempt is made to find the optimum condition for maximum heat transfer coefficient and minimum pumping power. Maximum convective heat transfer (W/m^2 K) is 4004W/m^2K at 50 ˚ angle,0.09 phi, at 4 lpm. The corresponding convective heat transfer using water under the same conditions is 1686.62/m^2K. The percentage increase when compared to a flat plate (0˚) is 110%. Pumping power (pressure drop) in pascal is minimum at 2 lpm flow rate at 0 and 0 phi. Minimum being 7.3Pa at 2 lpm. And maximum 58Pa at 40,60,4 lpm,0.09 concentration. Using MINITAB software, the optimum parameters that strike a deal between the coefficient of convective-heat transfer, pressure drop were noted to be 40,4 lpm,0.05 concentration with 'h' equal to 3196/m^2K and pressure drop equal to 42Pa.

Keywords *- Corrugated plate, Flow rate, Nanoparticle, Optimum, Pumping power, Turbulence, Wavy.*

1. Introduction

Heat exchangers are essential in various processes requiring efficient thermal energy exchange. Among different types, Wavy Corrugated Plate Heat Exchangers (WCPHEs) have gained significant attention due to their compact design, enhanced surface area, and natural ability to induce turbulence. The corrugations on the plates promote better fluid mixing and higher heat transfer compared to conventional flat-plate designs. However, with the rising demand for even greater thermal performance, traditional working fluids like water or oil often fall short. To overcome these limitations, nanofluids-engineered by dispersing nanosized particles into a base fluid-appeared as a viable option. Due to their superior thermal conductivity and enhanced energy transport capabilities, the Brownian motion of nanofluids can significantly boost the heat transfer rate within WCPHEs. When combined with the already turbulent environment created by corrugated surfaces, nanofluids further thin the thermal boundary layer and improve convective heat transfer. This synergistic effect makes the use of nanofluids in corrugated plate heat exchangers a powerful strategy for achieving higher thermal efficiencies in applications such as chemical processing, refrigeration, and power generation. Corrugated plates create complex flow channels, which promote turbulent flow even at low Reynolds numbers. The heat transfer coefficient is three to five times higher than that of traditional systems with minimum fouling effect. Metals have higher thermal conductivity than engine oil, glycol, and water. Hence, nanosized metal particles are mixed to improve the K of fluids [1]. Energy and energy flow efficacy have led to the use of nano-fluidics in industrial applications [2]. Nanosized material disrupts natural convective heat transfer, which results in the Nu number. At low velocity, the movement reduces and hence the decline in convective coefficient [3]. Nusselt number and Rayleigh number vary proportionally in natural convective heat transfer conditions for titanium oxide/H_2O nanofluid. Due to viscous forces, Nusselt no declines with titanium oxide percentage in H_2O [4]. Aluminium oxide, Titanium Oxide, Zirconium Oxide in base fluid at equal Reynolds No have shown efficacy in transfer of heat by 23,8,15 percent when flow is laminar, while for turbulent, efficacy is 51.13.41 percent at the cost of pressure drop [5]. More experimentation on the topic is the requirement [6]. For Silver in H_2O, the Overall Heat Transfer Coefficient is inversely related to phi and directly proportional to the velocity of nanofluid [7]. N_2 in Graphene have shown improvement in the coefficient of heat transfer in comparison with pure H_2O [8]. For various nanofluids, thermal performance was studied using the same experimental setup. [9, 10].

Titanium Oxide in H_2O exhibited better results when compared to Aluminium Oxide in H_2O [9,10]. The efficiency of the system increased by 85% for all Reynolds No with Graphene nanofluid [11]. The coefficient of heat transfer improved by 0.75 with an increment in pumping power with Carbon in CH_3COCH3 [12]. In all regimes of flow, the friction factor and the pressure drop multiply with the nanofluid [13]. An increment of the overall heat transfer coefficient (U) is observed with the rise of the rate of flow in multi-walled carbon nanotubes in H_2O [14]. Copper Oxide in H_2O revealed the highest pumping power, and Ethylene Glycol in H_2O showed the least, with Aluminium Oxide and Silicon Oxide in between [15]. Transfer of thermal energy improves with nanofluids [16]. Due to improved thermal-conductivity K (W/m K), Nanofluids provide an effective alternative for the improvement of HXs. But the role of Viscous effects is to be experimented with [17]. Flow regime also determines the ability of nanofluids in the case of Flat plate HXs. [18]. Less research related to the impact of corrugations on thermal flow parameters of nanofluids has been conducted. The optimized condition of angle, which can be accomplished with the utilization of CuO nanofluid of higher thermal conductivity, has been studied experimentally. A comparative study of heat flow parameters at different corrugated angles is done**.**

2. Materials and Methods

The core element of the setup is a Wavy Plate Heat Exchanger with Corrugations (WCPHE) designed with wavy-corrugation angles of 0°, 30°, 40°, 50°, 60°. It is fabricated by welding 3 SS plates in a precise arrangement to create 2 contiguous flow channels. The top channel, designated for nanofluids, maintains a spacing of 5 mm, while the bottom channel, intended for hot fluids, has a spacing of 15 mm. To support the fluid circulation, auxiliary components such as fluid storage tanks for both the nanofluid and hot fluid are integrated into the system. The experimental setup is equipped with thermocouples at the entrance and exit of each channel for recording fluid temperatures and seven thermocouples installed on the heat transfer plate surface to monitor wall temperatures at different points. These thermocouples are linked to digital temperature indicators for real-time data acquisition. The flow configuration adopted for the experiment is counter-current flow, ensuring efficient heat transfer between the two streams. The experimental setup, illustrated in Figure 1, was designed to study the performance of a corrugated plate heat exchanger.

Fig. 1 Experimental setup

Courtesy: *'International Journal of Engineering and Technology Innovation, vol. 5, no. 2, 2015, pp. 99-107'*

It comprises a test box, a tank to collect nano (test) fluid, a tank for hot water, motors-2, and rotameters-2. The test box is constructed from three stainless steel sinusoidal plates welded together, ensuring durability and resistance to corrosion during experiments. The test box includes a corrugated plate featuring corrugation angles of 0°, 30°, 40°, 50°,60°fabricated by precisely welding three stainless steel plates to create two distinct flow channels. Figure 2 details the schematic.

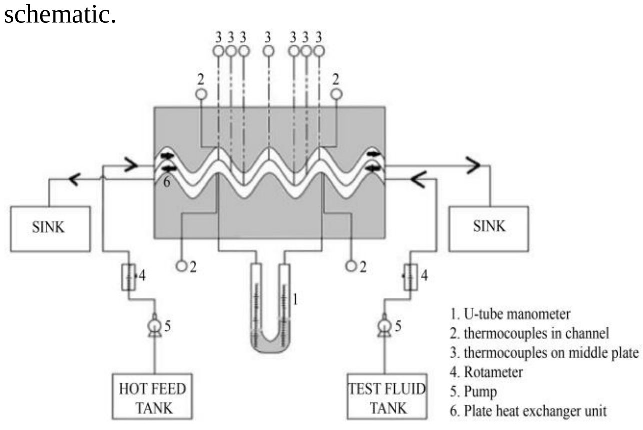

Fig. 2 'Wavy CPHE schematic of experimental setup'

Courtesy:" *http://www.ijrmee.org"*

The configuration of these corrugation angles relative to the horizontal is depicted in Figure 3.

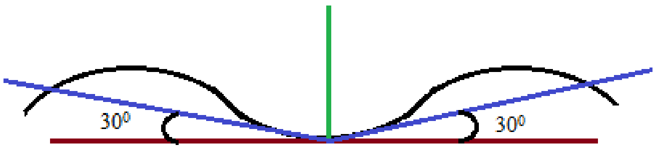

Fig. 3 Corrugated angle

The upper flow channel's specified distance is 0.5 cm. The corrugated plate dimensions are specified as 30 cm (length) × 10 cm (width) × 0.5 cm (plate spacing). Figure 4 illustrates the arrangement of thermocouples placed on the corrugated plates. Thermocouples T1 and T11 are positioned at the hot fluid's entrance and exit, respectively, monitoring the temperature as hot fluid flows from left to right (T1 to T11). Conversely, the cold fluid flows from right to left in the lower channel, monitored by thermocouples T3 and T2, placed at the inlet and outlet points, respectively. Arrangement of thermocouples is revealed in Figure 4.

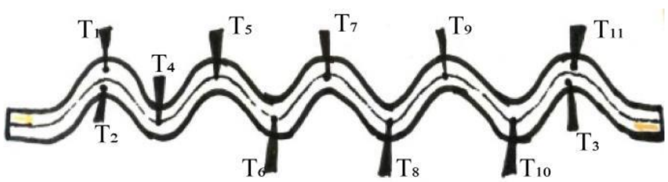

Fig. 4 Arrangement of thermocouples on corrugated plates

The experiment was conducted using a counter-current flow pattern, with CuO nanoparticles/water serving as the working fluid (test or cold fluid). Pure hot water was utilized as the hot liquid. Pressures of both liquids are recorded by pressure gauges with a 200 mm dial. Rotameters were incorporated to record the flow rates of liquids accurately. Ball valves were installed to control the flow of liquids through the system. Two tanks, each having a measurement of 300 mm × 300 mm × 600 mm, are used to store the respective liquids. A collection tank was placed to collect the cold fluid after it passed through the system. A heater with a 3000 W capacity was fitted in the hot water tank to maintain the required temperature of the hot fluid. The test section consisted of three identical corrugated channels, each fabricated with varying corrugation angles. These channels formed the core area where heat exchange between hot and cold fluids took place. Additional components like the fluid storage tanks (for hot and cold fluids) and other auxiliary parts supported the flow and operation of the system.

Hot water maintained at 70°C to 73 ˚C and at a constant flow rate of 3lpm, and the flow rate of cold fluid was varied from 2 to 4 lpm for 0.01,0.03,0.05,0.07,0.09% concentrations of CuO nano particles. For every experimentation trial, the inlet and outlet temperatures of the fluids and temperatures at seven different locations on the heat exchanger plates were recorded. The readings were obtained through a digital thermometric display with a precision of± 0. 1°C.The internal film heat transfer coefficient (hi) was determined by applying the energy balance method, utilizing the Logarithmic Mean Temperature Difference (LMTD) approach. Pressure drop in cold fluid is recorded by a pressure gauge. This setup enabled a consistent heating environment for accurate experimental analysis. Density, thermal conductivity ('K'), and specific-heat-capacity ('C_p') of CuO nanoparticle are 6400Kg/m^3, 32.9W/mK, 540J/J/kg K, respectively. 0.1% of Sodium Dodecyl Sulfate (SDS) surfactant is added to the cold nanofluid to prevent agglomeration, and a mechanical stirrer is used to stir the fluid. Thermal physical properties of cold water are estimated at a bulk temperature of nano (test fluid), and according to the concentration of percentage volume of nanoparticle by mass (phi), thermophysical properties of nanofluid are calculated. Amount of nanoparticle (Φ) is calculated by Equation (1), and (2).

$$(\Phi) = \frac{volume_{np}}{volume_{np} + volume_{water}} \tag{1}$$

$$= \frac{W_{np}/\rho_{np}}{W_{np}/\rho_{np} + Volume_{water}} \tag{2}$$

Specific heat, thermal conductivity, and density of nano(test)fluid are determined using Equation (3), (4), and (5), respectively.

$$C_{nf} = \frac{[\emptyset(\rho_{np}c_{np}) + (1-\emptyset)(\rho c_w)]}{\rho_{nf}} \tag{3}$$

$$K_{nf} = \frac{k_{p+2k_w+2\emptyset(k_p-k_w)(1+2.5\emptyset)}}{k_p+2k_w-\emptyset(k_p-k_w)} K_w \tag{4}$$

$$\rho_{nf} = \emptyset\rho_p + (1-\emptyset)\rho_w \tag{5}$$

T1 and T11 are hot liquid (H_2O) temperatures at the entrance and exit, while T3 and T2 are temperatures of cold CuO nanofluid at the inlet and outlet, and T4 to T10 are wall temperatures. The average wall temperature is given by Eq. (6). Heat transfer rate and LMTD are given by Equatiobn (7) and (8), respectively.

$$Tavg = \frac{T4+T5+T6+T7+T8+T9+T10}{7} \tag{6}$$

$$Qavg = (MFRh.Cph.\Delta Thot + MFRnf.Cpnf.\Delta Tcold \tag{7}$$

$$\text{LMTD} = \frac{(T_{avg}-T_{c,in})-(T_{avg}-T_{c,out})}{ln\frac{T_{avg}-T_{c,in}}{T_{avg}-T_{c,out}}} \tag{8}$$

Convective- heat- transfer coefficient 'h' is calculated by Equation (9).

$$Qavg = \text{h. A. LMTD} \tag{9}$$

Pressure drop was drawn from the pressure gauge.

3. Results and Discussion

Thermal conductivity and nanoparticle boundary layer disruption increase 'h' as phi increases. Brownian motion of nanoparticles and turbulence are affected by the WCPHE angle, which in turn influences 'h'. Figure 5 indicates the effect of WCPHE angle on h, at lpm 2 and at phi equal to 0,0.01,0.03,0.05,0.07,0.09. At 40 $^{\circ C}$- 50 $^{\circ C}$, at Φ between 0.05-0.09 percentage concentration, the convective heat transfer coefficient increases.

Figure 6 indicates the effect of WCPHE angle on h, at lpm 3 and at phi equal to 0,0.01,0.03,0.05,0.07,0.09. h increases for all cphe angles, phi as lpm increases from 2 to 3 due to enhancement of velocity. At 0.09% and 50^0, the maximum convective heat transfer coefficient is observed.

Figure 7 indicates the effect of WCPHE angle on h, at lpm 4 and at phi equal to 0,0.01,0.03,0.05,0.07,0.09. At 50 $^{\circ}$ and phi = 0.09%, improved h is found.

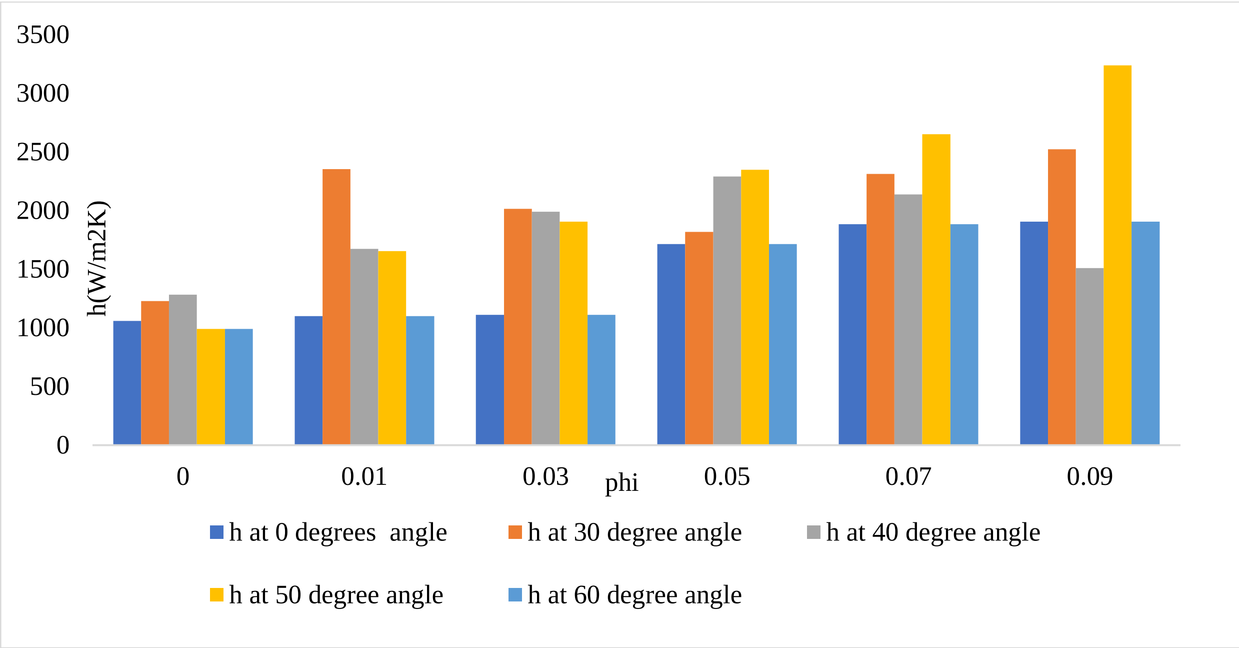

Fig. 5 Effect of cphe angle, phi, on 'h' at 2lpm

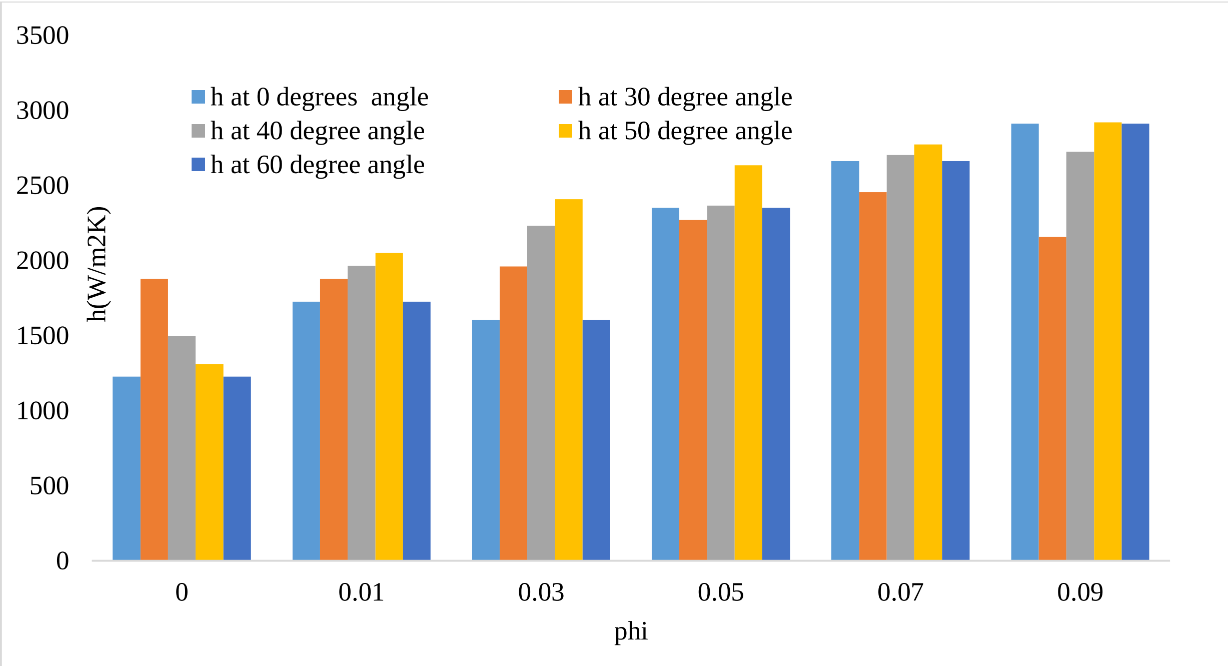

Fig. 6 Effect of cphe angle, phi 0n h at 3lpm

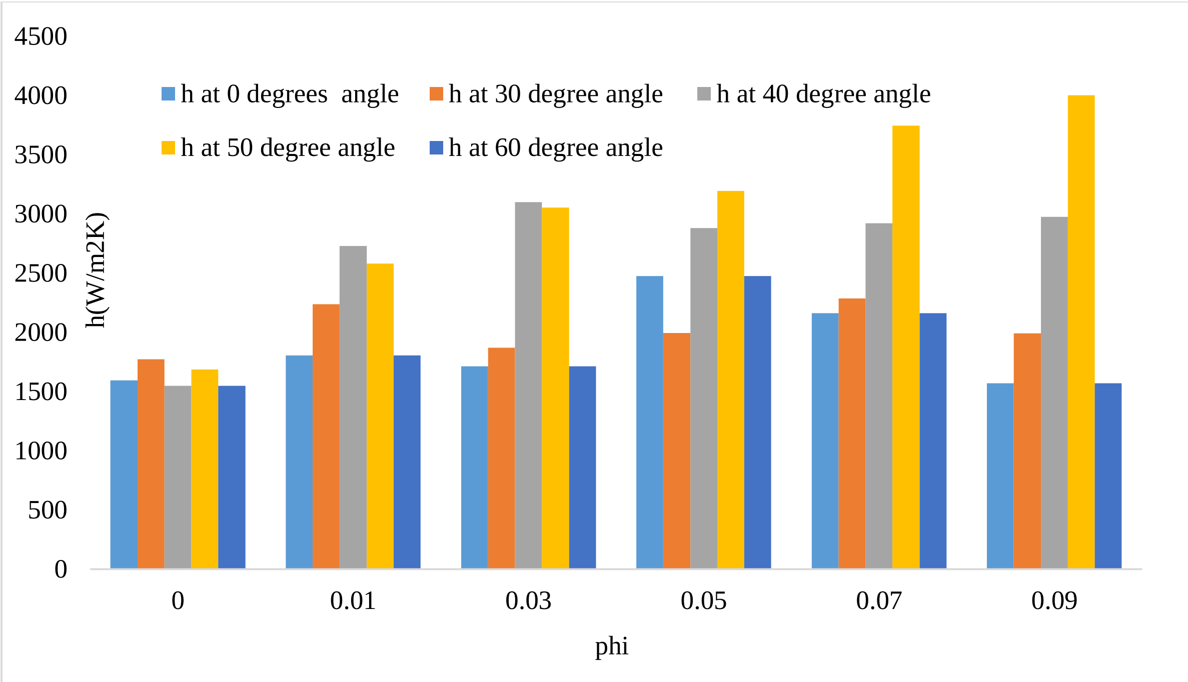

Fig. 7 Effect of cphe angle, phi 0n h at 4lpm

Figure 8 indicates the effect of the cphe angle, phi, on pressure drop at 2lpm. Density and viscosity increase the pressure drop as CuO concentration increases.

Volume flow rate increases velocity and frictional losses, which increase the pressure drop. Turbulence and flow disruption increase the pressure drop as the cphe angle increases. At 2 lpm, the minimum pressure drops at 0° angle and 0 Φ.

Figure 9 indicates the effect of the cphe angle, phi, on pressure drop at 3lpm. As lpm increases, the pressure drop increases. Lower pressure drop at 0 phi and flat plate.

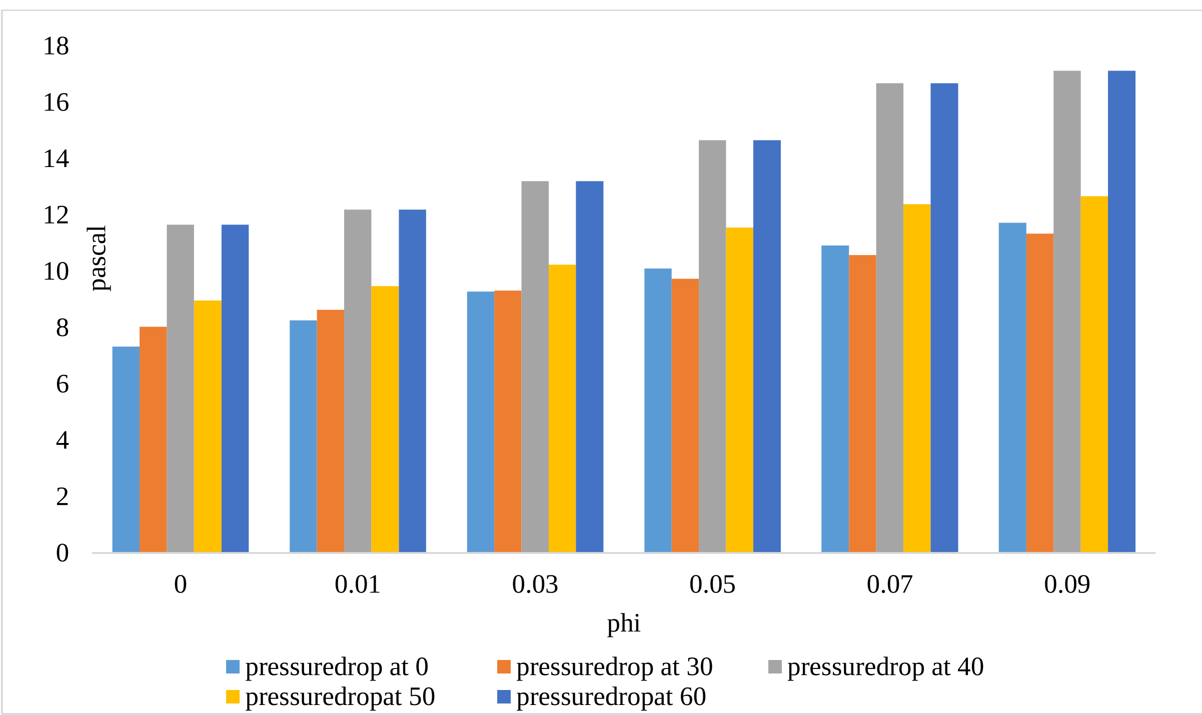

Fig. 8 Effectof cphe angle, phi, on pressure drop at 2lpm

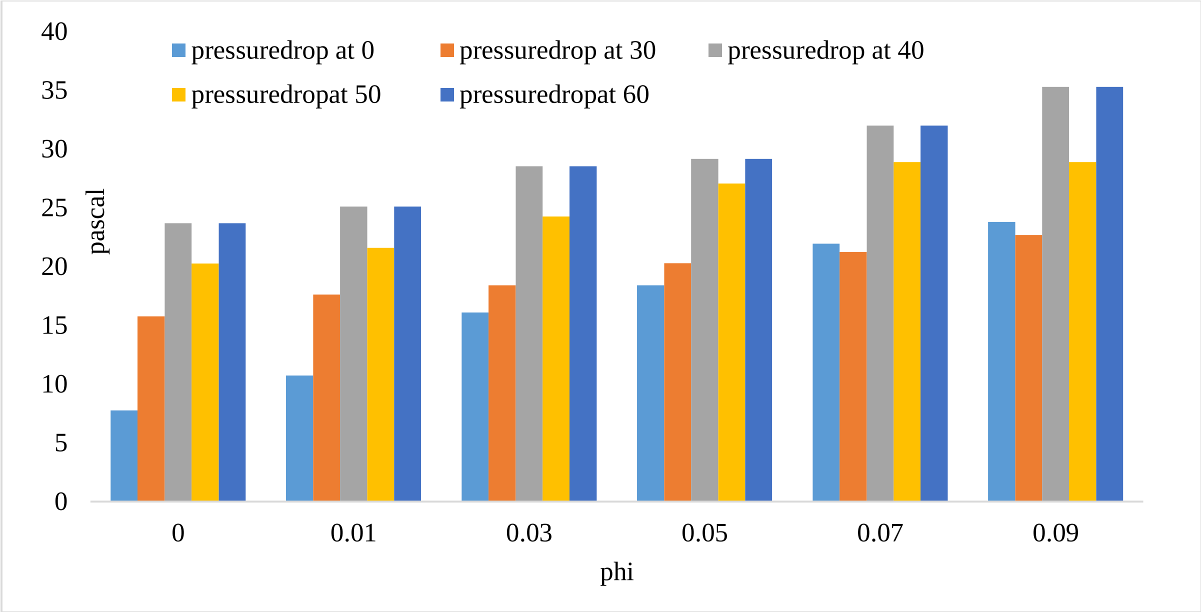

Fig. 9 Effect of cphe angle, phi, on pressure drop at 3lpm

Figure 10 indicates the effect of WCPHE angle, phi, on pressure drop at 4lpm. Due to turbulence and viscosity, steep rise in pressure drops between 40-to 50-degree angles and at phi between 0.05 and 0.09.

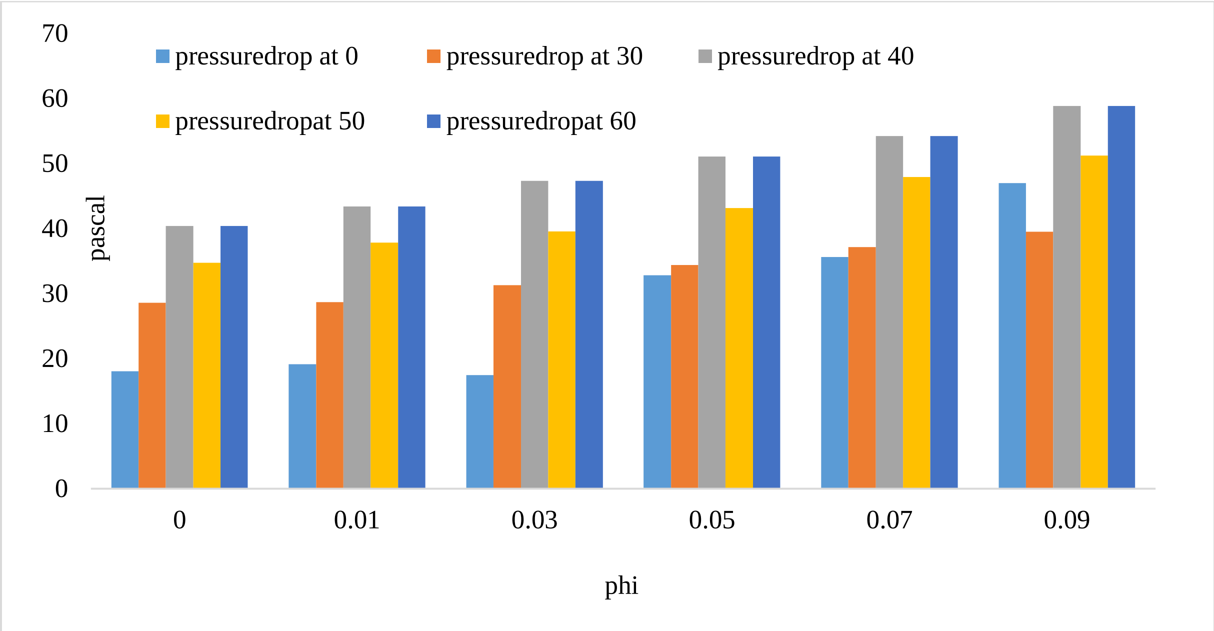

Fig. 10 Effect of cphe angle, phi, on pressure drop at 4lpm

4. Conclusion

Improvement of turbulence and thermal conductivity with uO nanoparticle concentration rises' h'. WCPHE angle affects the heat transfer coefficient as it influences mixing of flow, Brownian movement and boundary layer disruption. For 2lpm, at 0.09 percentage concentration, a maximum heat transfer coefficient of 3200W/m^{2}K is observed at 50° angle. At 3 pm, a max h of 2900W/m^{2}K is obtained at 50° and phi of 0.09. Max h of 4000W/m^{2}K is obtained at 4lpm,0.09 Φ and 50°.With flow rate and increased concentration, WCPHE angle pressure drop increases due to enhanced viscosity, velocity, friction, and resistance to flow. Minimum pressure drop of 7.3 Pa is obtained at 0 Φ and 0 °C at 2 lpm. In case of 3,4 lpm, the minimum pressure drop is 7 and 17.5Pa, respectively, at the same phi and corrugation angle., The optimum value of both responses is found from Minitab. Figure 11 shows the main effect plot for means. Combined main effects plot for h(W/m^{2}K), inverse of pressure drop is taken. In the option, the larger, the better. The optimum condition is found to be 50,4 lpm,0.05 concentration, where h is 3196W/m^{2}K and pumping power is 42 Pa. Need to find optimum is validated by M.N. Pantzali et al.," doi:10.1016/j.ces.2009.04.004"

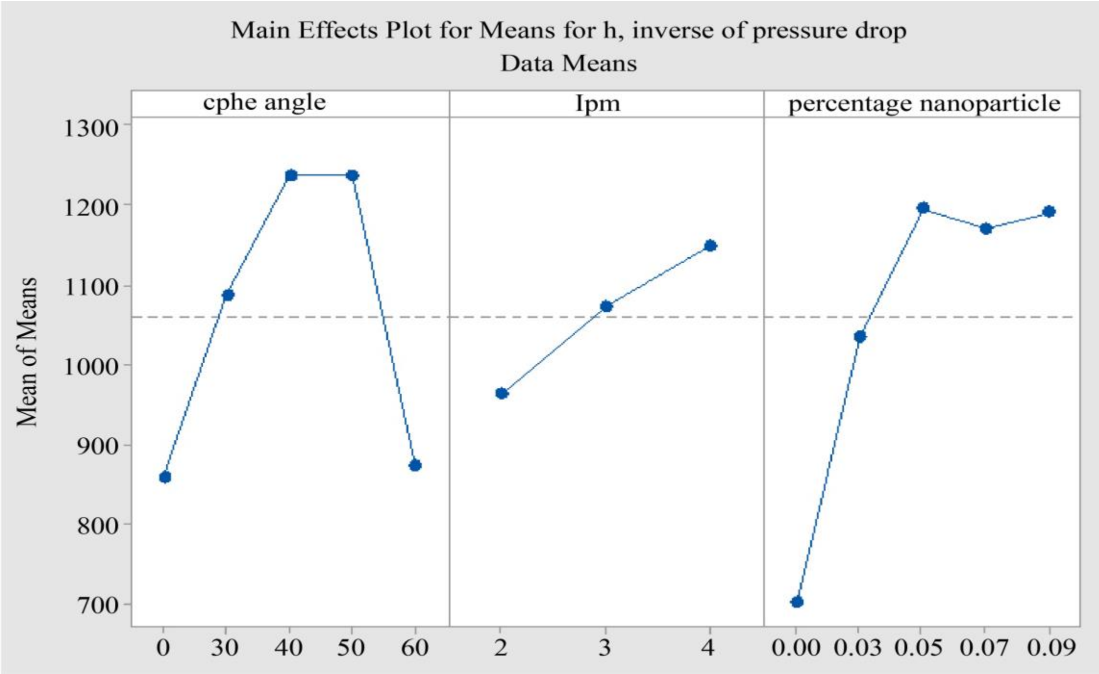

Fig. 11 Main effects plot for h, inverse of pressure drop

4.1. Future Scope

An investigation is required on the impact of magnetic fields on the characteristics of nanofluids. A comparative evaluation of several nanofluids is essential at this time. Practical industrial applications require investigation. The impact of alterations in dimensions and angles on thermal qualities and parameters requires investigation.

Funding Statement

"The work is supported by the Chaitanya Bharathi Institute of Technology, Gandipet, Hyderabad.

Nomenclature

h	Convective heat transfer coefficient (W/m^{2}K)
C_P	Specific- Heat, J. /kg K
g	gravity, m./s^2
K	Thermal- Conductivity, W/m K
Nu	Nusselt No
Q	Rate of heat transfer (Watts)
U	Overall heat transfer rate
ϕ	% volume fraction of nanoparticle
μ	dynamic viscosity, kg/mS
nf	nanofluid
p, np	nanoparticle
h	hot water
w	water
c	Cold fluid
avg	average

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