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Original Article

A Comparative Study on Heat Transfer Augmentation in Corrugated Plate Heat Exchangers Using Zinc Oxide Nanoparticles

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Abstract *- Heat Exchangers (HEs) are essential for facilitating heat transfer between fluids at different temperatures. HEs use passive, active, or mixed heat transfer enhancement strategies to increase efficiency, reduce surface area, and reduce pumping power. Heat exchangers are proposed for energy savings. Among these, passive techniques are more cost-effective as they do not require external power input. These methods primarily aim to enhance the surface area, fluid residence time, and thermal conductivity, often through the use of nanofluids. Better heat transmission and turbulence are provided by Corrugated Plate Heat Exchangers (CPHEs) when compared with flat plate heat exchangers at low Reynolds numbers because of their corrugationsand occupy less space because of their high surface area to volume ratio. The Wavy Corrugated Plate Heat Exchangers (WCPHEs) that are the subject of this study provide increased turbulence and surface area. Zinc oxide nanofluids (ZnONF) at volumetric concentrations of 0.01%, 0.03%, 0.05%, 0.07%, and 0.09% were used in experiments to assess the heat transfer capability, with water serving as the base fluid. Three corrugation angles—10°, 30°, and 50°—as well as 0.5,1,1.5 lpm nanofluid flow rates were used for the studies. The findings demonstrated that, in contrast to H_2O as the test fluid, the transmission of heat rate rose to all volume fractions and flow rates of the nanofluid. A 30° corrugation angle, a 1.5 lpm flow rate, and a 0.01% volume fraction of ZnO nanoparticles produced the greatest heat transfer of 2430.55W. Additionally, it was shown that heat transfer rose from 10° to 30° as the corrugation angle increased but reduced at 50°.*

Keywords *- Augmentation of heat transfer, Corrugation plates, Heat exchanger, Nanofluids, Reynolds number, Turbulence.*

1. Introduction

With the global shortage of energy and fossil fuels, optimizing energy consumption has become increasingly important. One of the most effective strategies for reducing energy losses and improving heat exchangers' thermal efficiency is efficient energy utilization. HEs are mostly used in different industries because of their durability, compact size, and efficient heat exchange rates. Consequently, enhancing the performance of heat exchangers is a key challenge. Significant efforts have recently focused on increasing heat transfer rates using nanofluids. Among HEs, PHEs are very efficient due to their enormous surface area, enabling faster heat transfer than other designs. Corrugated Plate Heat Exchangers (CPHEs) represent a recent advancement, offering an impressive surface area-to-volume ratio of over 700 m²/m³.

The design of corrugated plate channels induces turbulent flow at low Reynolds numbers (generally between 50 and 200), resulting in a convective coefficient of heat transfer that is three to five times greater than that of traditional tube and shell heat exchangers. A WCPHE is illustrated in Figure 1.

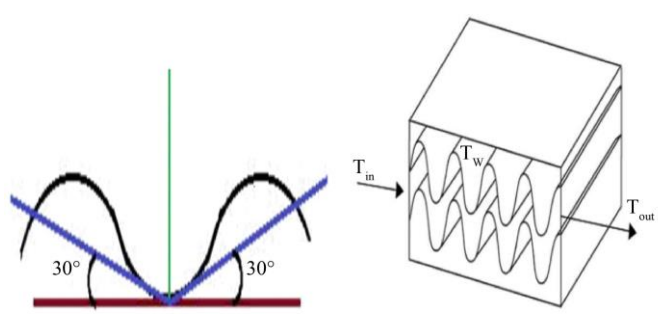

Fig. 1 Corrugated plate with corrugation angle

2. Literature Review

CPHEs offer a higher heat transfer coefficient than other types of heat exchangers due to the increased contact area between the fluids. They are easy to repair, maintain, and service because of their small size and low maintenance space requirements. Additionally, these heat exchangers are self-cleaning due to their corrugations, which reduce fouling and lead to better performance and longevity. Various studies have explored different approaches to optimize HEs performance. Grekova et al. [1] optimized the geometry of the adsorbent heat exchanger for adsorption chilling application by using methanol-LiCl/SiO_2 as a working fluid by considering several fins and channels. They found that the best geometry balances adsorbent granule volume, plate channels, and fins with optimal condensing, evaporating, and regeneration temperatures of 100°C, 350°C, and 800°C. A numerical study by Montazerifar et al. [2] examined a unique fractal fin design in Oil/MWCNT nanofluid multi-stream plate-fin heat exchangers at various angles of attack and Reynolds number and realized higher fluid mixing and deflection with the rise in attack angle of fractal fins. Jassim et al. [3] showed that nanofluids improved heat transfer more than base fluids. Illan-Gomez et al. [4] tested evaporator-type PHEs with various refrigerants under transient settings and found robust and accurate heat transfer coefficient correlations. Baris Gurel et al. [5] studied plate heat exchangers with latent heat energy storage (LHTES) and several phase transition materials to determine optimal geometry for various plate geometries numerically.

Mikhaeil et al. [6] discovered that evaporation heat transfer coefficients in an asymmetric plate heat exchanger varied from 1330 to 160 W/m^2K, depending on the wetted surface and operating circumstances. Water and PCM were used to test plate-type heat exchangers by Rami Saeed et al. [7], and PCMs saved costs. Nitesh K. Panday and Shailendra N. Singh [8] examined multi-pass plate heat exchanger thermo-hydraulic performance and proposed Nusselt number, effectiveness, and friction factor correlations for various configurations. Giraud et al. [9] studied water vaporisation in small plate-type evaporators at different working pressure, fluid temperatures, and filling ratios. Using an ANN model, Longo et al. [10] predicted refrigerant condensation heat transfer coefficients in herringbone-type Brazed PHEs. Wang et al. [11] computationally compared plate-type heat exchangers to a chevron-type plate HE using H_2O as the working fluid.

Depth pitch and corrugated angle were important. Li et al. [12] tested bubble flow in a dimple-type plate heat exchanger using an air/H_2O combination under different inlet and output circumstances. Infrared thermography was used by Berce et al. [13] to study carbonate crystallisation fouling in corrugated plate heat exchangers. Lee et al. [14] applied an aqueous Lithium Chloride (LiCl) solution to a plate-type heat exchanger to study heat and mass transfer during dehumidification. Taghavi et al. [15] studied Plate-Type Thermal Energy Storage Systems' (PTESs') effectiveness and performance when charging and discharging. Sopian et al. [16] examined the forced convective turbulent flow of SiO_2-water nanofluid in various corrugated channel designs, such as straight, semicircular, and trapezoidal. L. Syam Sunder et al. [17] tested nickel-water nanofluids in a corrugated plate HE, focussing on heat transfer, friction factor, entropy, energy efficiency, performance index ratio and pumping power. Nanofluids and phase transition materials improve HE thermal performance. M.A. Khairul et al. [18] found that CuO/water nanofluids improved corrugated plate HE heat transfer coefficients by 27.2%.V.K. Nema et al. [19] found that Al_2O_3-water nanofluids demonstrated improved heat transmission with increased Reynolds and Peclet numbers.

Nanofluids affected pressure drop and heat transfer in double-dimpled pipe surfaces, according to Ehsan et al. [20]. Sunden et al. [21] tested a corrugated plate HE in solar energy systems with varying nanofluid concentrations of Al_2O_3, SiC, CuO, and Fe_3O_4. A computational investigation by Hasnan et al. [22] found that using SiO_2-water nanofluid in a symmetrical semicircle-corrugated channel improved heat transmission and flow. Vafajoo et al. [23] constructed a mathematical model for a plate recuperative HE that recovers energy from refinery flue gases and preheats input air under 2-dimensional, compressible, and turbulent flow compared to flat plate heat exchanger, increasing the Chevron angle increased output air temperature by 18% and flue gas pressure decrease by 63%. Jiang et al. [24] developed a symmetrical capsule-type plate HE(SCPHE) with counter-rotating vortices to reduce flow resistance and increase performance. Lee et al. [25] investigated fluid flow in chevron-type PHEs by unstable numerical analysis with Large-Eddy Simulation (LES). The literature shows that corrugated structures, fractal fins, and nanofluids can improve heat transfer coefficients and thermal efficiency in industrial applications, improving energy efficiency.

Furthermore, studying these systems' thermal performance under different operational situations provides a solid foundation for optimising heat exchange processes in conventional and renewable energy systems. The findings emphasize the need for continuing study in this area to fully utilise sophisticated heat exchanger designs, which improve energy utilisation, operational costs, and environmental effects in engineering applications. This research adds to heat exchanger technological knowledge and offers useful advice for future advancements. Water is a suitable solvent and available adequately, so it is used as base fluid. Various nanoparticles with different dimensions and effects of base fluid other than water need to be studied

3. Materials and Methodology

Zinc Oxide Nanoparticles (ZnONF) were used as the primary material in the study, with base fluid as water. Figure 2 shows an image of nanoparticles. The density of ZnONF is 5600 g/m^3, its thermal conductivity is 13 W/mK, and its specific heat is 494 J/KgK.

Figure 3 shows WCPHE. The measurements of the WCPHE are taken as the length of the plate is taken as 30 cm, the width of the plate is 10 cm, the plate spacing for nanofluid is 0.5 cm, the plate spacing for hot fluid is 1.5 cm, and the corrugation angles were taken as 30^o, 40^o and 50^o.

Fig. 2 Zinc oxide nanoparticles

Fig. 3 Photographic view of WCPHE

The study was conducted using 0.01%, 0.03%, 0.05%, 0.07%, and 0.09% volumes of zinc nanoparticles in water. Hot water was used as hot fluid. Table 1 shows the Thermo-Physical properties of ZnONF. An experimental setup and its schematic representation are shown in Figure 4.

Flow rates of both nano and hot fluids were measured by rotameters, rotameters calibrated to measure flow rates between 0.5 to 5lpm with an uncertainity of ±1% and flow regulation was achieved through the use of valves. Two motors, each with a capacity of 0.25 HP and 1500 rpm, are employed to pump the test fluids.

The dimensions of the cold and hot fluid tanks are both 300 mm × 300 mm × 600 mm. A collection tank was provided for collecting the cold fluids, and a 3000 W heater was used for heating the hot fluid. The test section consisted of three identical corrugated channels with corrugation angles of 30° and 50°.

The top channel contained zinc nanoparticles in water at concentrations ranging from 0.01% to 0.09% and hot water flowed through the bottom channel. The volume flow rate of hot water was kept constant at 3 litres per minute, and the cold nanofluid's flow rate was varied. Other auxiliary components included the storage tanks for both fluids.

Table 1. Thermo-Physical properties of nanofluid

Zno % in water	Density (Kg/m^3)	Viscosity * 10^{-6} (Pa.s)	Specific Heat (kJ/Kg .K)	Thermal Conductivity (W/m.K)	Prandtl number
0	1000	1006	4.174	0.62	4.8
0.01	1046	1090	4.137	0.64	7.05
0.03	1138	1140	4.063	0.67	6.91
0.05	1230	1190	3.989	0.70	6.78
0.07	1322	1250	3.916	0.74	6.62
0.09	1414	1300	3.842	0.78	6.40

Fig. 4 Experimental setup

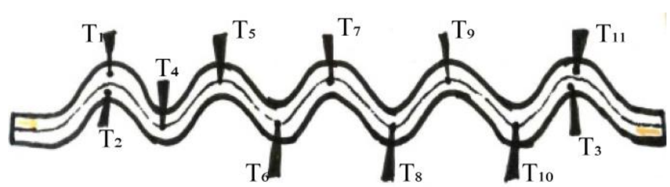

Fig. 5 Arrangement of thermocouples on WCPHE

In all experiments, hot water at 70°C was used to heat the test fluids, with an invariant flow rate. For each experimental measurement, the inlet and exit temperatures of both cold(nano) and hot fluids, along with wall temperatures at seven different points on the heat exchanger plate, were recorded using thermocouples. These thermocouples were welded to the plate and connected to a digital temperature indicator which provided accurate temperature readings with a precision of ±0.1°C., which provided accurate temperature readings with a precision of 0.1°C. Figure 5 shows the arrangement of thermocouples on WCPHE. The recorded temperatures were used to analyze heat transfer. The hot fluid flow rate remained unchanged during the experiments, while the cold nano test fluid flow rate varied from 0.5 lpm to 1.5 lpm. The middle plate was equipped with seven thermocouples distributed along its length and width to gauge wall temperatures. Four additional thermocouples were used to measure both fluids' inlet and outlet temperatures. The inside film heat transfer coefficient (h) was calculated for each flow rate by performing an energy balance using the Log Mean Temperature Difference (LMTD) method. Throughout all experiments, hot water flowed through the top corrugated channels, ensuring that the channel surfaces remained at a nearly constant temperature, which was used to heat the test fluids. The mathematical formulas used are as follows. T_1 and T_{11} were taken as the hot fluid temperature at the inlet and outlet, while T_3 and T_2 are temperatures of cold nanofluid at the inlet and outlet. T_4 to T_{10} are film temperatures. The average wall temperature is given by eq1. LMTD and heat transfer rate are given by eq 2,3, respectively.

$$T_{avg} = \frac{T_1+T_2+T_3+T_4+T_5+T_6+T_7}{7} \tag{1}$$

ΔT_1 = Temperature drop at the inlet = T_{avg} - $T_{c,in}$
ΔT_2 = Temperature drop at exit = T_{avg} - $T_{c,out}$

$$\text{LMTD}= \frac{(T_{avg} - T_{c,in})-(T_{avg} - T_{c,out})}{\ln(\frac{(T_{avg} - T_{c,in})}{(T_{avg} - T_{c,out})})} \tag{2}$$

$$Q=\text{MFR}.C_p.\,T_{cold} \tag{3}$$

$$Q=h.A.(\text{LMTD}) \tag{4}$$

Where Tc, and Tc,ut represent the cold fluid's (test fluid) intake and output temperatures, respectively, where MFR is the water's mass flow rate. Cp is the water's specific heat capacity. Every temperature is expressed in degrees Celsius. The Nusselt number was computed once the heat transfer coefficient was determined 53tilizing equation 4.

$$Nu = \frac{h\,D_h}{k} \tag{5}$$

Here D_h is the channel's hydraulic diameter, which was determined using equation (6), and Reynolds number, which was determined using equation (7).

$$D_h = \frac{4h}{P} = \frac{2\text{Wx}}{\text{W+x}} \tag{6}$$

$$Re = \frac{\rho \vartheta D_h}{\mu} \tag{7}$$

$$\text{MFR (kg/s) = mass flow rate of fluid} = \frac{\rho * flowrate}{60000} \tag{8}$$

V= Velocity of the nanofluid in (m/s) = *Discharge/ Area* ---- eq(9)

4. Result and Discussions

Corrugated heat exchanger performance is investigated for volume flow rate and the Reynolds number impact on heat transfer coefficient. Figures 6 to 10 represent the impact of the volume flow rate of nanofluid on heat transfer for 10°, 30°, and 50° corrugation angles, respectively. From Figure 6, it is observed that h is highest for a 30° angle at Φ=0.01at 2430.55W/m^2 K. For a 10° angle, h is the least at 353.89W/m^2 K at Φ=0.05.

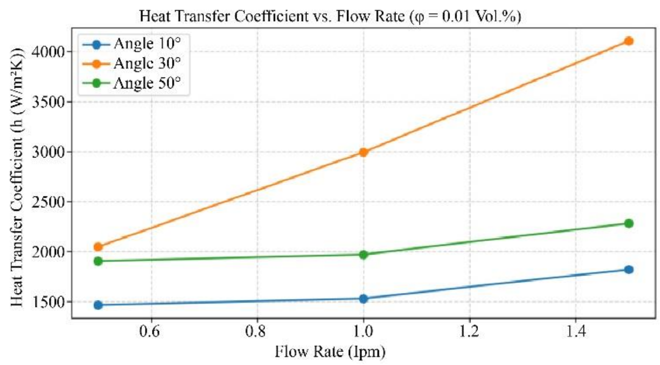

Fig. 6 Volume flow rate Vs h for Φ=0.01%

From Figure 7, it is observed that h increases as the flow rate increases for Φ=0.03%.with % increase of 69% for 30°, 94% for 50° and 116% for 10° angle.

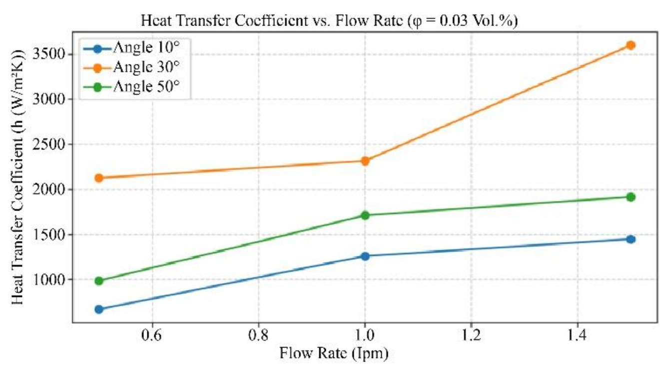

Fig. 7 Volume flow rate Vs h for Φ=0.03%

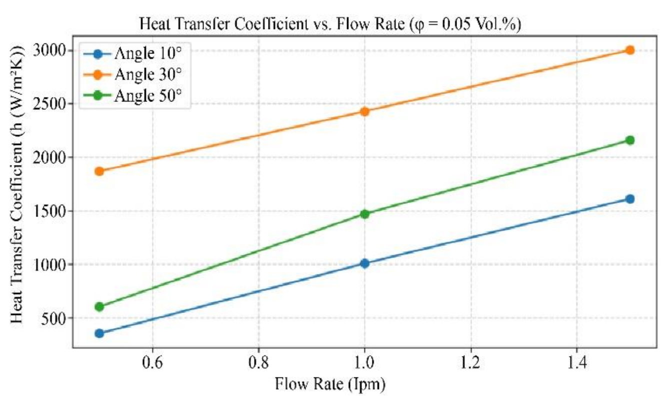

Fig. 8 Volume flow rate Vs h for Φ=0.05%

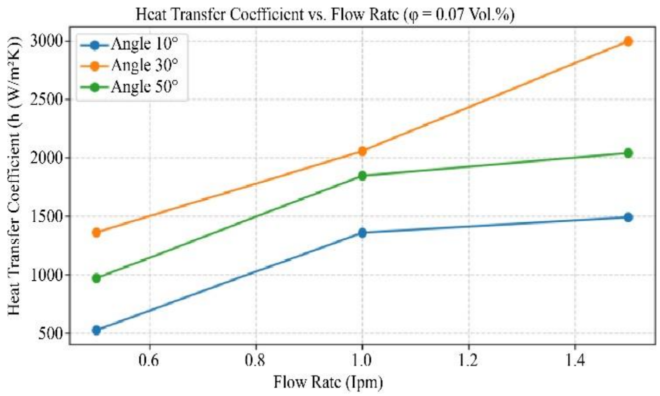

Fig. 9 Volume flow rate Vs h for Φ=0.07%

From Figure 8, it is observed that h increases with an increase in flow rate at Φ=0.05%, with h highest for 30° corrugation angle at 2428.76W/m^2 K and lowest for 10^0 for all values of flow rates at 353.89W/m^2 K.

From Figure 9, it is understood that an increase in nanoparticle concentration to Φ=0.07% and an increase in flow rate enhances h due to enhanced thermal conductivity and turbulence, respectively.

From Figure 10, it is understood that higher flow rates enhance h, nanoparticle concentration Φ enhances heat transfer rate, and 30° angle is effective in heat transfer enhancement with a maximum h value of 4106.09W/m^2 K.

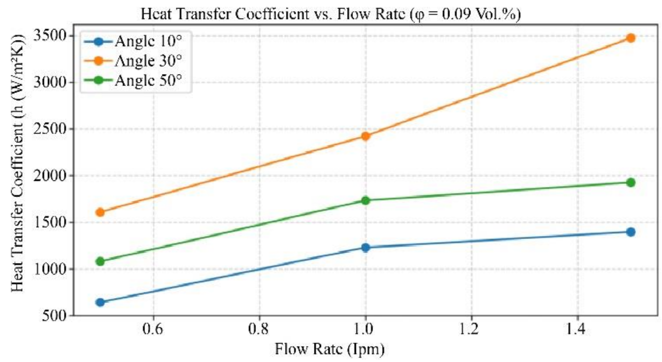

Fig. 10 Volume flow rate Vs h for Φ=0.09%

Figures 11, 12, and 13 represent the variation of the heat transfer coefficient with Reynold's number. With the increase in Reynold's number, the convective heat transfer coefficient increased for all volume fractions of nanofluids. 'h' is enhanced with an increase in Reynold's number for 10°, 30° and then decreases for 50° angle.

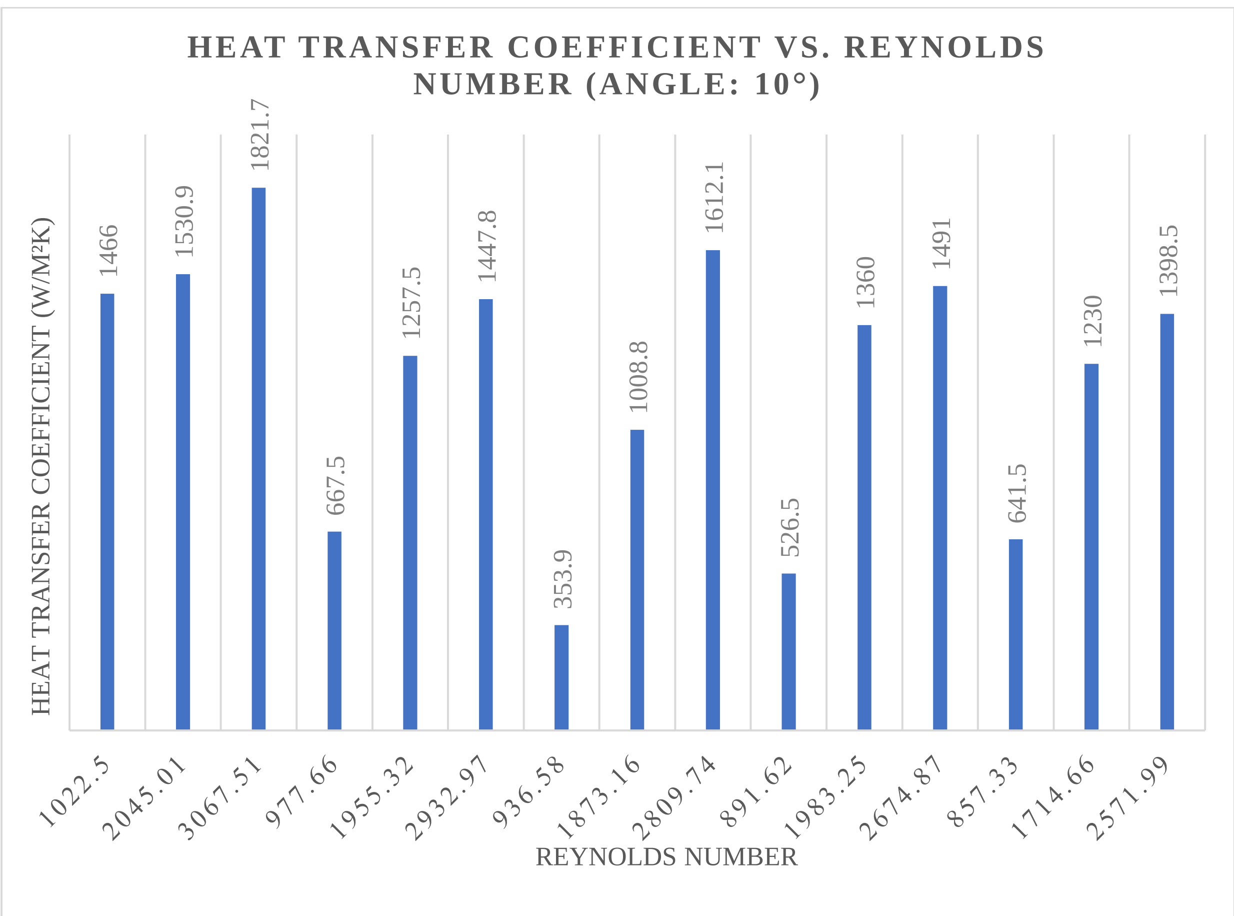

Fig. 11 Reynold's no Vs h for 10° angle

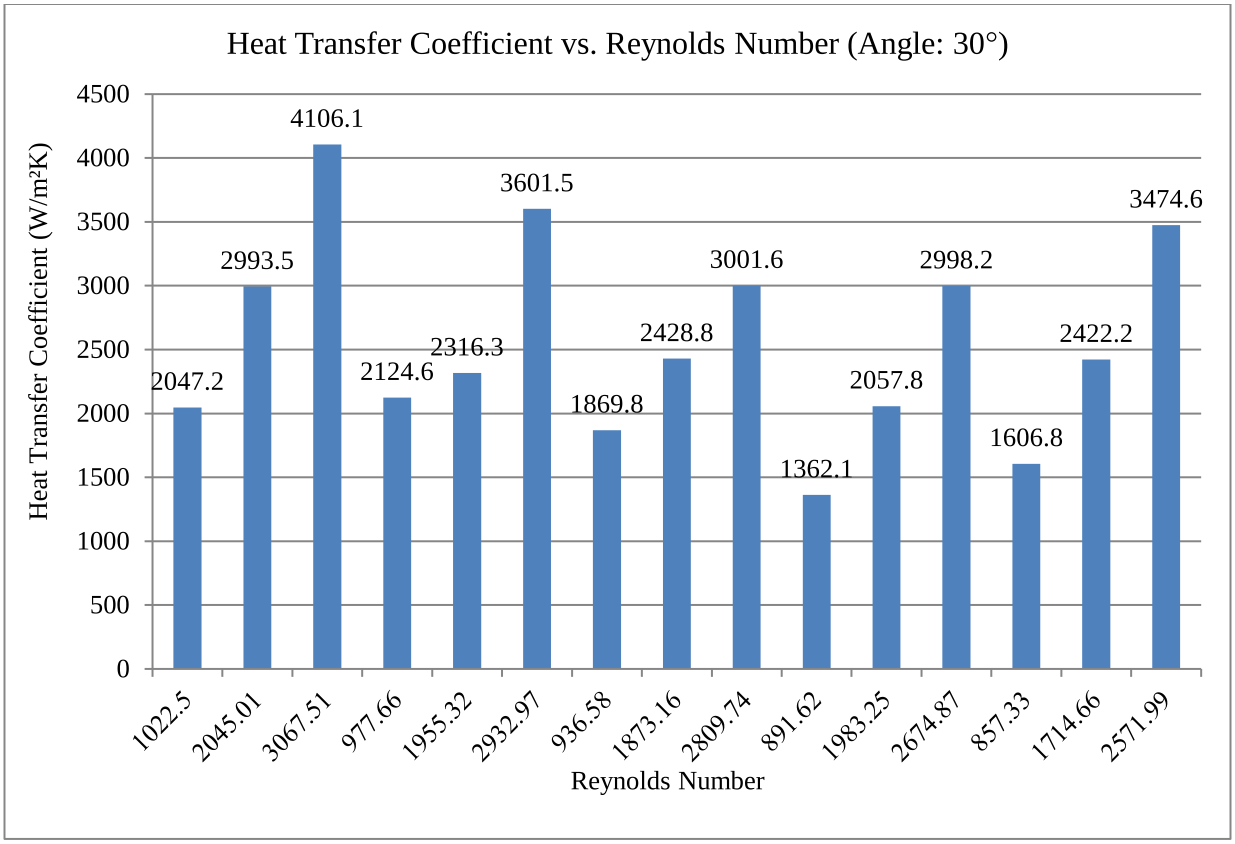

Fig. 12 Reynold's No Vs h for 30º angle

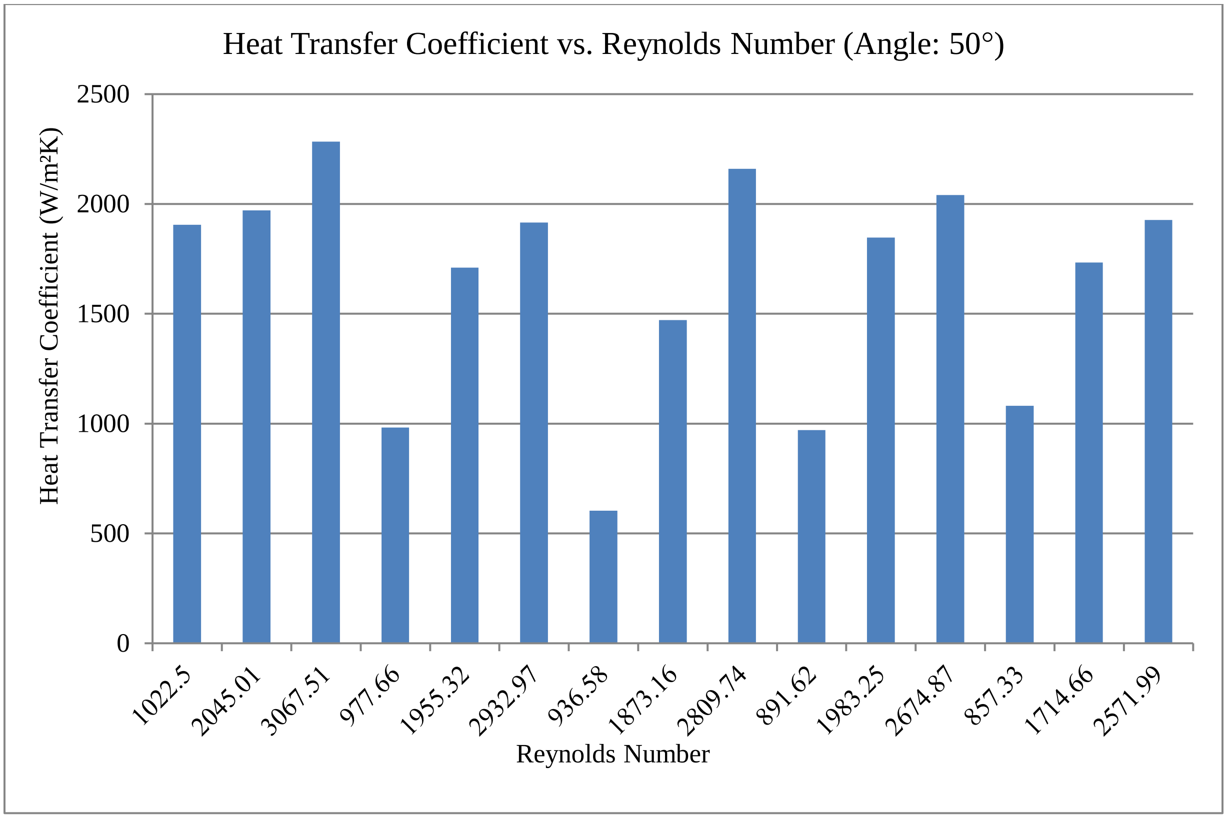

Fig. 13 Reynold's No Vs h for 50º angle

The impact of the percentage concentration of nanofluids on heat transfer rate at 0.5,1, 1.5 lpm flow rates of nanofluids is observed from Figures 14, 15 and 16. From Figure 14, it is observed that for the volume flow rate of 0.5 litres per minute, h is the highest for 30^0 angles for all nanoparticle concentrations. For 10, 50 degrees, there is a drop in h at Φ=0.05%.

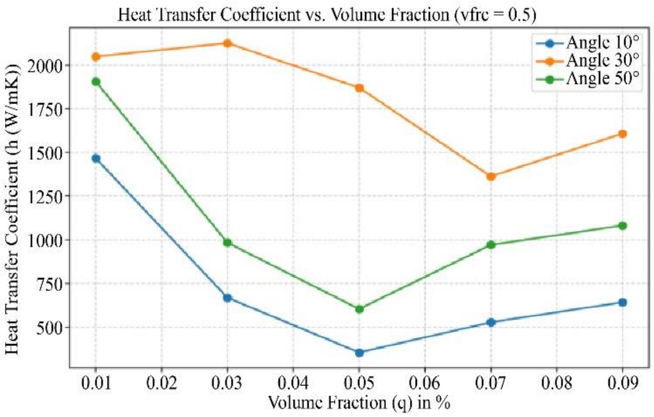

Fig. 14 Percentage concentration of nanofluid on heat transfer rate at 0.5lpm flow rate

Figure 15 represents the effect of Φ on h for 1 lpm flow rate. h for 30 degrees is the highest, and h for 10 degrees is the lowest.

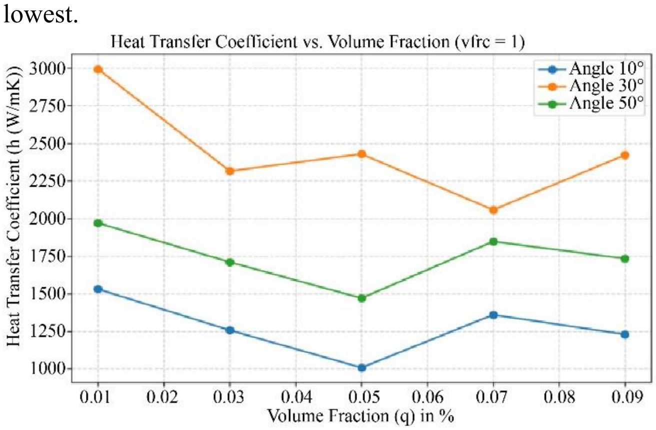

Fig. 15 Percentage concentration of nanofluid on heat transfer rate at 1lpm flow rate

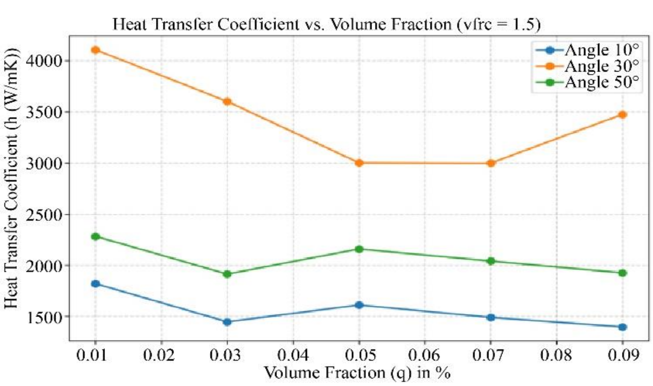

Fig. 16 Percentage concentration of nanofluid on heat transfer rate at 1.5lpm flow rate

5. Conclusion

It can be concluded that with an increase in flow rate, there is an increase in heat transfer coefficient. Maximum heat transfer coefficient is observed for 30º corrugated angle for all flow rates.

With an increase in Reynold's number, the heat transfer coefficient increased for 10, 30, and 50-degree corrugated plates. 30° consistently provides the highest heat transfer coefficients, regardless of nanofluid concentration and flow rate, due to its ability to effectively balance turbulence and viscous effects. Higher flow velocities enhance the heat transfer coefficient, especially at lower nanofluid concentrations.

The improvement is more pronounced at 30°. For 10º angle, the heat transfer coefficient decreased as the nanoparticle volume fraction (phi) increased due to the reduced turbulence and thermal conductivity at smaller angles and higher phi. For 30º angle, the Heat Transfer Coefficient is consistently higher compared to 10° for all values of phi due to better turbulence-enhanced heat transfer efficiency.

For a 50º angle, the Heat Transfer Coefficient is relatively high at phi = 0.01 but drops sharply at phi = 0.05, which indicates that larger angles might increase viscous effects, reducing heat transfer efficiency at higher phi values. The maximum heat transfer occurred at a 30º corrugation angle, a flow rate of 1.5 liters per minute (lpm), and a 0.09% volume fraction of ZnO nanoparticles.

Furthermore, it was observed that heat transfer increased as the corrugation angle increased from 10° to 30° but decreased when the angle was further increased to 50° with maximum heat transfer coefficient 4106.09W/m^2 K at 30º, 1.5lpm, phi=0.01% minimum heat transfer coefficient 353.89W/m^2 K at 10º, 0.5lpm,phi=0.05%.

Future Scope

From Figure 16, it is observed that h is highest for 30 degrees angle for volume flow rate of 1.5 lpm due to balance of turbulence and viscous effects

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The authors declare that there is no conflict of interest regarding the publication of this paper.

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