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Enhancing Heat Transfer Performance in Heat Exchangers using Nanoparticle-Infused Fluids: A Computational Approach

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ABSTRACT

The study examines the impact of hybrid nanoparticle-based materials on the flow performance in a tube with a consistent surface temperature of 373 K. The simulation of the flow was conducted using Ansys 2024 and two-dimensional governing equations for partial differentials. This simulation utilized three distinct concentrations of hybrid nanomaterials Al_2O_3–Cu to observe their effects. Al_2O_3–Cu nanoparticles are dispersed in water at volume ratios of 1%, 5%, and 100%. The fluid velocity varied between 0.1 and 1.5 m/s, while the input temperature was 25 °C (~298 K). The current study examines three cases of dimples: 15, 30, and 45 concave/convex dimples. Increasing the concentration of Al_2O_3-Cu nanoparticles in water is shown to improve certain physical properties while decreasing others. Moreover, when the concentration of nanoparticles grows, the Nusselt number (Nu) also rises. The thermal performance factor is enhanced with an increase in the number of dimples. The study shows that the heat transfer coefficient increases with hybrid nanoparticle concentration, achieving enhancements of 2.14%, 10.8%, and 21.82% for 1%, 5%, and 10% concentrations at 1 m/s. The highest increase of 22.22% compared to pure water occurs at 1.5 m/s with a 10% concentration. Compared to a smooth pipe, improvements are 15.67%, 30.01%, and 37.72% for three cases, with case 2 (30 concave/convex dimples) exhibiting the best thermal performance factor.

1. Introduction

The primary role of a heat exchanger is to transfer heat energy among different fluids separated by a solid. Heat exchangers have an extensive collection of usages in many sectors, where they are utilized for processes such as crystallization, concentration, distillation, fractionation, pasteurization, sterilization, and process fluid management. Standard heat exchangers include air preheaters, cooling towers, condensers, evaporators, shell and tube exchangers, and vehicle radiators. Heat exchangers may be categorized based on numerous factors, for example, the method of heat transfer, the quantity of fluids involved, and the compactness of the surface, the flow arrangements, the structure, and the heat transfer mechanisms [1].

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Two primary methods are used to improve heat transfer: Active and passive. Procedures such as passive heat transfer augmentation can enhance heat flow without any need for additional power or control instruments. These methods involve changes in surface fluid or fluid properties, such as adding fins or using porous media with surface roughness. Passive approaches are simple, reliable, and cost-effective, making them suitable for applications where ease of maintenance and operation is a priority. Although not as high as the active levels, passive methods are easy to implement and less complex. On the other hand, actives use external sources like mechanical devices or electric fields— they reach more significant degrees of augmentation by controlling the system from outside [2].

One of the most significant methods in passive technique is geometry change. This method is typically employed to alter the tube's outer and inner surfaces to correct the heat transfer rate of the several tubes inside the heat exchanger [3]. Industrial heat recovery equipment in many different sectors may use this method extensively, which works with various tube shapes and sizes, including corrugated, ribbed, dimples, helical, spiral, and other cross-sections. Enhancing boundary layer fluid mixing, flow turbulence level, and the tube heat transfer surface area boosts the rate of heat transfer [4,5].

Córcoles *et al.,* [6] examined the validity of spirally corrugated tubes on the improvement of heat performance of the heat exchanger numerically and experimentally. The analysis is conducted in a Reynolds number range of 25,000 to 50,000. The main finding of this study is that this modification improved the rate of heat transfer by 23%. Moreover, the reduced helical pitch and the rise in helical height significantly impact the heat transfer rate. Han *et al.,* [7] mathematically calculated the effect of inner corrugation on heat rate performance. The findings implied that the heat transfer improved because of a decline in corrugation height, and the Nusselt number improved by up to 81%.

Ahmed *et al.,* [1] examined the effect of rib depth (e/d) and pitch between two successive ribs (p/d) on the heat exchangers' thermal performance. The studied parameters are e/d and p/d equal to 0.025 to 0.069 and 0.29 to 5.8 [8]. The results indicated that this modification can upgrade the heat rate by up to 40%. Furthermore, Nusselt number values are more remarkable in ring-type ribbed tubes than those with rectangular, trapezoidal, or triangle ribs. In contrast, friction factor values are higher in triangle-ribbed tubes than in the other types. Pethkool *et al.,* [9] mentioned that raising the height of the rib and pitch ratios increased the friction factor, thermal performance factor, and Nusselt number.

Conversely, Eiamsa-ard *et al.,* [10,11] inspected the incorporation of several twisted tapes into heat exchanger tubes. Using several twisted tapes improves heat transfer more than a single one. It was mentioned that using twin twisted tapes results in a heat progress rate twice as high as using a single twisted tape. Additional research was conducted on implementing twisted-tape adjustment to find optimal performance. The study investigated the impact of twisted tape with peripherally, trapezoidal, and V cuts on heat transmission, as documented in references [12,13]. Asgar Ali *et al.,* [14–18] investigated the effect of nanofluid on enhancing heat transfer and entropy generation.

A nanofluid is a type of fluid that has particles with dimensions in the nanometer range that are dispersed throughout it [19]. Nanofluids, consisting of nanoparticles dispersed in a base fluid, mainly liquids, enhance thermal conductive and convective heat transfer capabilities. Nanofluids are classed depending on the type of nanoparticles used in the liquid or base fluids [20]. Two broad categories of nanofluids may be defined according to the nanoparticles that make them up: metallic nanofluids and non-metallic nanofluids. Metallic nanoparticles are used to create metallic-based nanofluids, which can consist of metal oxides or metals such as zinc (Zn), aluminum (Al), zinc oxide (ZnO), copper (Cu), or copper oxide (CuO) [21].

Chun *et al.,* [22] displayed that the heat transfer rate is boosted by up to 25% when nanofluid (Al_2O_3) is added to water with a 0.25% to 0.5% concentration. Introducing nanofluid (Al_2O_3 with CuO) with the primary working fluid (ethylene glycol) and the concentration of Al2O3 from 0.1 to 1% and CuO concentration of 0.1 to 1% have the ability to improve heat transfer rate by up to 37.2 % [23]. The Nusselt number improved by 19% and 22.8 % when Al_2O_3 was added to water with concentrations of up to 1% [24,25]. The Magnesium Oxide (MgO)–water nanofluid improved the Nusselt number by 36%, but the friction factor increased by 19% [26]. TiO2 nanofluid boosted the heat transfer rate by 72%, but the friction factor increased by 52% [27].

Louis *et al.,* [28] review the impact of various nanoparticles on the thermophysical properties of nanofluids, highlighting how factors like concentration, size, and shape of nanoparticles influence heat transfer coefficients and pressure drops in double-pipe heat exchangers. The study indicates that using metallic oxide nanoparticles can enhance heat transfer coefficients by up to 30% compared to conventional fluids.

This study aims to analyze the impact of hybrid-nano (Al_2O_3–Cu/water) on enhancing heat flow in a circular conduit. The analysis will be conducted utilizing a one-phase mathematical approach. Furthermore, the effect of concave/convex dimples under three cases is examined to investigate their impact on thermal performance. The three cases are 15, 30, and 45 concave/convex dimples integrated into the tube surface. The study investigated the effect of nanofluid (Al_2O_3–Cu /water) on the thermal and physical characteristics of the fluid at different volume concentrations.

The current research tries to answer the following questions: the effect of nanofluid integrated in surface modification on the thermal performance of heat exchangers.

2. Model Description

Figure 1 illustrates the geometry and dimensions of the test section. As shown in this figure, the test section length is 1080 mm with a diameter of 13.8 mm. Moreover, the smooth tube is equipped with concave/convex dimples along with the inner surface of the tube. These concave/convex dimples have a diameter of 6.9 mm, and they are distributed in 3 cases: Case 1 is 15 concave/convex dimples, Case 2 is 30 concave/convex dimples, and Case 3 is 45 concave/convex dimples.

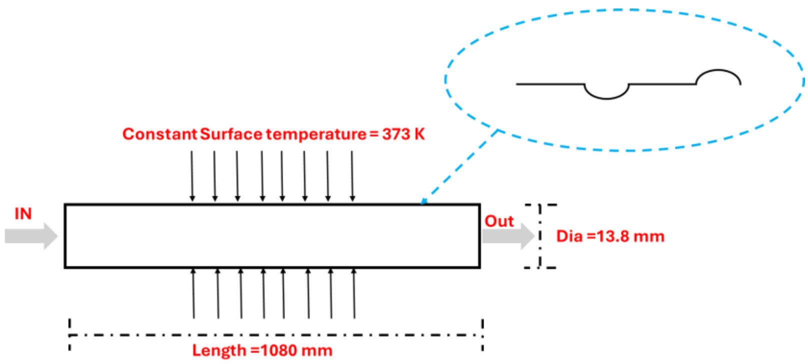


Fig. 1. Schematic diagram for the test section

3. Data Reduction

The current part shows the procedures for calculating factors used to estimate thermal performance through the tubes that were tested will be given for the variables heat transfer coefficient by convection (h), Nusselt Number (Nu), Reynolds Number (Re), and Friction factor (f) respective. The following correlations are used to get the numerical findings that are obtained [1,29]:

$$\mathrm{Re} = \frac{\rho \times u \times d}{\mu} \tag{1}$$

$$\mathrm{Nu} = \frac{\mathrm{h \times d}}{\mathrm{k}} \tag{2}$$

$$\mathrm{h} = \frac{\mathrm{Q}}{\mathrm{A(T_s - T_b)}} = \frac{\dot{m}\mathrm{c_p(T_o - T_i)}}{(\pi \mathrm{dL})(\mathrm{T_s - T_b})} \tag{3}$$

$$f = \frac{2\Delta \mathrm{P}}{\rho(\mathrm{L/d})\mathrm{u^2}} \tag{4}$$

Where ρ is the density of working fluid (kg/m^3), μ is the fluid viscosity (Pa.s), d is the tube diameter (m), u is the mean velocity of the fluid (m/s), Q is the rate of heat transfer (Watt), k is the thermal conductivity (W/m.K), A tube cross-section area (m^2), T_s and T_b are the surface and bulk temperature (ºC), L is the tube length (m), T_o and T_i are the outer and inner temperature (ºC), and ΔP is the pressure drop across the tube.

The density (kg/m^3), viscosity (Pa.s), heat capacity (J/kgK), and thermal conductivity (W/m.K) are defined as follows [30]:

Hybrid nano density

$$\rho_{hnf} = (1 - \emptyset_t)\rho_f + \emptyset_{P1}\rho_{P1} + \emptyset_{P2}\rho_{P2} \quad \text{Where} \quad \emptyset_t = \emptyset_{P1} + \emptyset_{P2} \tag{5}$$

Hybrid nano viscosity

$$\mu_{hnf} = (1 + 2.5\emptyset_t)\mu_f \tag{6}$$

Hybrid nano heat capacity

$$\mathrm{Cp_{hnf}} = [(1 - \emptyset_t)\rho_f\, Cp_f + \emptyset_{P1}\rho_{P1}Cp_{P1} + \emptyset_{P2}\rho_{P2}C_{PP2}] / \rho_{hnf} \tag{7}$$

Hybrid nano thermal conductivity

$$\frac{\mathrm{K_{hnf}}}{\mathrm{K_f}} = \frac{\frac{\emptyset_{P1}K_{P1} + \emptyset_{P2}K_{P2}}{\emptyset_t} + 2K_f + 2(\emptyset_{P1}K_{P1} + \emptyset_{P2}K_{P2}) - 2\emptyset_t\, K_f}{\frac{\emptyset_{P1}K_{P1} + \emptyset_{P2}K_{P2}}{\emptyset_t} + 2K_f - (\emptyset_{P1}K_{P1} + \emptyset_{P2}K_{P2}) + \emptyset_t\, K_f} \tag{8}$$

Where ρ_f is the water density, ρ_{P1} is the Al_2O_3 density, and ρ_{P2} is the Cu density. $\emptyset t$ is the volume fraction of total nano, $\emptyset_{P1}$ is the Al_2O_3 volume fraction, and $\emptyset_{P2}$ is the Cu volume fraction. Table 1 shows the properties of water Al_2O_3 and Cu.

Table 1
Properties of different materials at 27 ºC [31]

Properties	Water	Al2O3	Cu
Density (kg/m3)	998.2	3970	8933
Heat capacity (J/kgK)	4182	765	386
Thermal conductivity (W/mK)	0.6	40	401

The Thermal performance Factor (TPF) is applied to measure the thermal efficiency of the enhancement conducted in the circular tube and calculated as follows:

$$TPF = \frac{\left({}^{h_{mod}}/_{h_{smooth}}\right)}{\left({}^{f_{mod}}/_{f_{smooth}}\right)^{1/3}} \tag{9}$$

Nu_{mod} is the Nusselt number, and f_{mod} is the friction factor of the modifications.

3.1 Governing Equations

In this computational analysis, the continuous, turbulent flow that is flowing through the tested tubes is applied in a two-dimensional (axis-symmetric) configuration. In the course of this simulation investigation, the following equations are solved: momentum, continuity, energy, and turbulent model of RNK (K- ε). It is possible to state these equations as follows [32]:

Momentum equation $$\frac{\partial}{\partial x_j}\left(\rho U_i U_j\right) = -\frac{\partial P}{\partial x_i} + \frac{\partial \tau}{\partial x_j} \tag{10}$$

Continuity equation $$\frac{\partial}{\partial x_i}\left(\rho U_i\right) = 0.0 \tag{11}$$

Energy equation $$\frac{\partial}{\partial x_i}\left[U_i(\rho E + P)\right] = \frac{\partial}{\partial x_i}\left(k\frac{\partial T}{\partial x_i}\right) \tag{12}$$

The viscous stress tensor is τ_{ij}, k is the effective thermal conductivity of the fluid, and i is used to indicate the tensor and take 1 and 2.

The RNK (K- ε) turbulence model, which uses improved wall functions to estimate turbulent viscosity μ_t for near-wall treatment, is used in this work.

$$\frac{\partial}{\partial x_i}\left(\rho k u_i\right) = \frac{\partial}{\partial x_j}\left(\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right) + P_k - \rho\varepsilon \tag{13}$$

$$\frac{\partial}{\partial x_i}\left(\rho \varepsilon u_i\right) = \frac{\partial}{\partial x_j}\left(\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right) + C_{1\varepsilon}\frac{\varepsilon}{k}P_k - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{14}$$

Where k stands for the turbulent fluid kinetic energy and ε for the rate at which that energy dissipates. In the model, the constants σ_k=0.7194, $C_{1\epsilon=}$1.42, $C_{2\epsilon=}$1.68, σ_ϵ=0.7194 and C_μ=0.0845, are known.

3.2 Boundary Conditions

At the inlet, the velocity is assumed to be constant across the studied scenarios: 0.1, 0.5, 1, and 1.5 m/s. Additionally, the inlet temperature is set at 298 K. The outlet is presumed to be a pressure outlet. The surface temperature of the pipe is assumed to be constant at 373 K.

4. Numerical Model

The software Ansys Fluent 2024 was used to simulate the flow of (Al_2O_3–Cu/water). Figure 2 illustrates the process of meshing creation. The integral equations were solved using the Finite Volume Technique with pressure-velocity coupling and the SIMPLE algorithm [30]. Furthermore, the second-order upwind strategy regulates partial differential equations for enhanced accuracy. The convergence criterion for dependent variables, such as continuity, energy, and momentum, is defined as 10^{-6}. This CFD research made many assumptions: the hybrid nanofluid is an incompressible fluid and a single-phase model with distinct physical features. The nanoparticles exhibit homogeneous dispersion. The exterior surfaces of walls are maintained in a constant state. In addition, a mesh study was conducted to determine the optimal mesh size; the results show that a 0.2 mm element size is sufficient for a solution for both modified and smooth pipes.

The flow of the current research is divided into three steps: Investigate the effect of nanofluid on heat transfer performance, Investigate the effect of surface modification on thermal performance, and investigate the impact of combining the nanofluid and surface modifications on thermal performance.

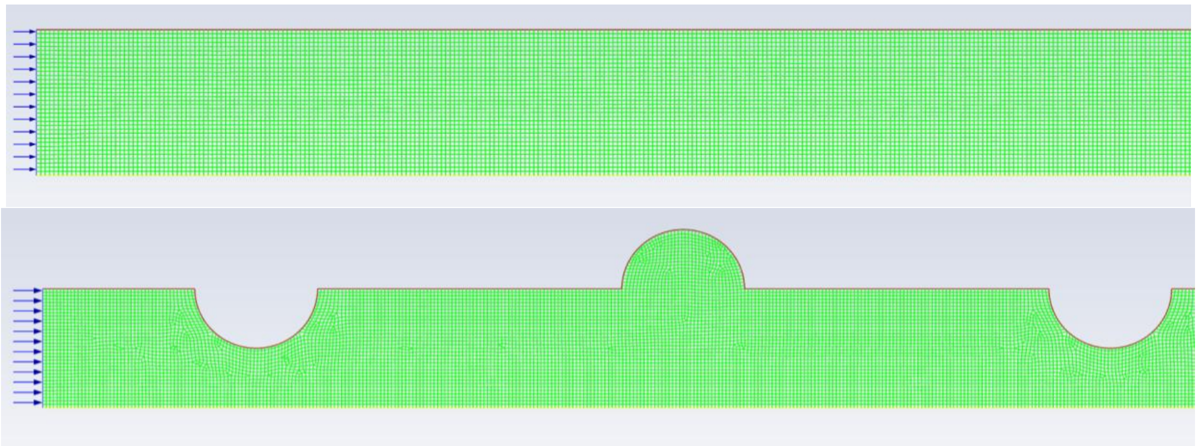


Fig. 2. Schematic diagram for the test section in case of smooth and modified tubes (symmetric pipe)

5. Results and Discussion

5.1 Validation

The accuracy of the numerical results validated by comparing them to the experimental data acquired from Huang *et al.,* [8] based on airflow. Figure 3 displays the numerical and experimental values of the Nusselt number with various Reynolds numbers. The numerical results are in accordance with the experimental data, with an average relative error of ± 3%. As depicted in this figure, the Nusselt number increases as the Reynolds number increases.

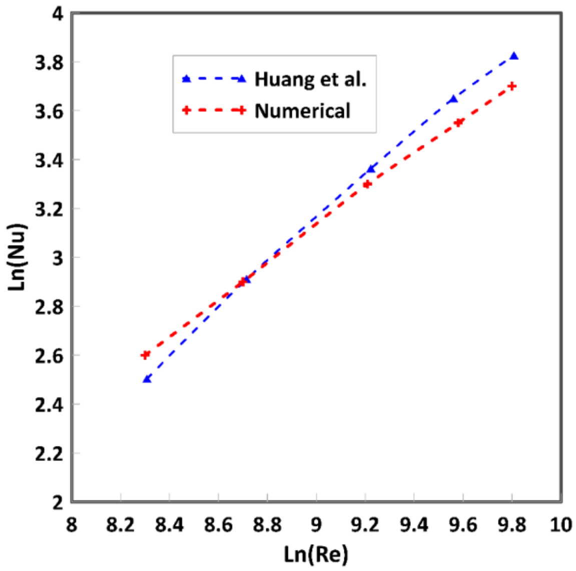
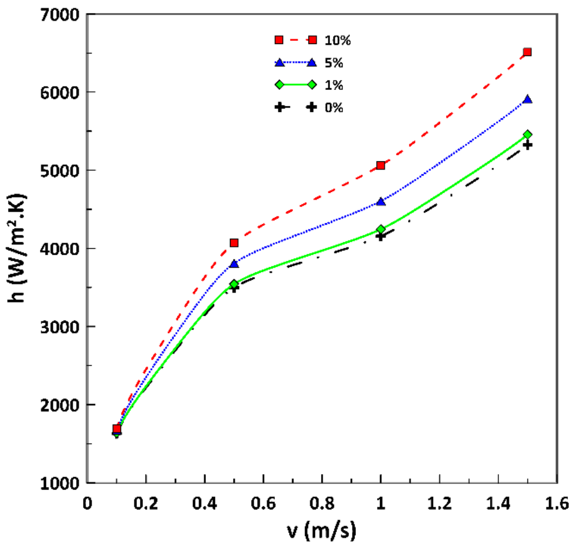


Fig. 3. Validation of numerical Nu vs experimental Nu by Huang *et al.,* [8] for airflow within a smooth tube

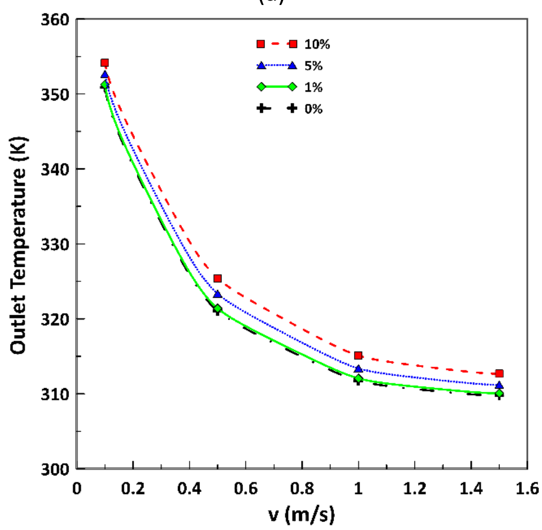
5.2 The Effect of Hybrid Nanoparticles

Figure 4 reveals the variation of various heat transfer characteristics in the smooth pipe with variation fluid velocity for different hybrid nanoparticle concentrations (0% (pure water), 1%, 5%, and 10% volume fractions). An incremental rise in the average heat transfer coefficient is observed as the velocity increases in all scenarios, as shown in Figure 4(a), indicating a strong correlation between the Nusselt number and variations in the fluid velocity (i.e., Re). By including the hybrid nanoparticles and increasing their concentration, the average heat transfer coefficient was elevated compared to the case without nanoparticles (0%). This improvement can be attributed to an improvement in the physical properties of the nanofluid, particularly its thermal conductivity. The results show an average increase in heat transfer coefficient of 2.14%, 10.8%, and 21.82% for hybrid nanoparticle concentration ratios of 1%, 5%, and 10% by volume, respectively, for velocity of 1 m/s. The highest variation is around 22.22% compared to pure water and is observed when the velocity is 1.5 m/s and the concentration is 10%. It is worth mentioning that at a velocity of 0.1 m/s (in laminar flow), the change in heat transfer coefficient due to the addition of nanoparticles is relatively small.

Figure 4(b) represents the effect of hybrid nanoparticles with different concentrations on the outlet fluid temperature at various fluid velocities. The fluid enters the pipe with a temperature of 25 ^{o}C (~298K), as specified in the current study, and the pipe's surface is maintained at a constant temperature of 100 ^{o}C (~373 K). Higher velocity reduces outlet temperature due to insufficient time for heat exchange in the fluid. However, mixing hybrid nanoparticles with water results in an upsurge in outlet temperature, as shown in the figure. For instance, at an inlet velocity of 0.5 m/s, the outlet temperature records an increase of 0.35 K, 2.31 K, and 4.35 K for nanoparticle concentration ratios of 1%, 5%, and 10% by volume, respectively.



(a)



(b)

Fig. 4. Effect of adding hybrid nanoparticles with different concentrations on heat transfer characteristics (a) heat transfer coefficient and (b) outlet temperature

5.3 The Effect of Concave/Convex Surface

Figure 5 illustrates the changes in heat transfer characteristics as fluid velocity varies for a modified concave/convex pipe surface compared to a smooth surface. Numbers 15 for case 1, 30 for case 2, and 45 for case 3 represent the three studied concave/convex configurations. In all study conditions, as Figure 5(a) illustrates, the higher the inlet fluid velocity, the higher the heat transfer coefficient. By utilizing a concave/convex surface and increasing its number, the average heat transfer coefficient is significantly enhanced compared to a smooth pipe. The improved heat transfer area is responsible for this improvement. Compared to smooth pipe at a velocity of 1 m/s, the heat transfer coefficient improves by an average of 15.67% in case 1, 30.01% in case 2, and 37.72% in case 3. The highest divergence, occurring at a velocity of 1.5 m/s and for case 3, is approximately 42.61% relative to a smooth pipe.

The influence of the concave/convex surface on the fluid outlet temperature at different inlet velocities is shown in Figure 5(b). Previously, it was stated that the fluid inlet was 298K. At the same time, the surface temperature was constant at 373 K. Similarly, because there is less time for heat exchange in a fluid moving at a higher velocity, the temperature at the outflow is lower. On the other hand, as can be seen in the figure, the exit temperature increases when the tube surface is modified to be concave/convex compared to a smooth one. This is because a concave/convex shape results in a higher heat transfer coefficient, as discussed earlier. For example, in case 1, the exit temperature increases by 4.62 K, in case 2, by 8.01 K, and in case 3, by 9.64 K, all at an inlet velocity of 0.1 m/s. Comparing concave/convex configurations, doubling the concave/convex number raises the exit temperature by 3.39 K while tripling it raises the exit temperature by 5.02 K at 0.1 m/s velocity.

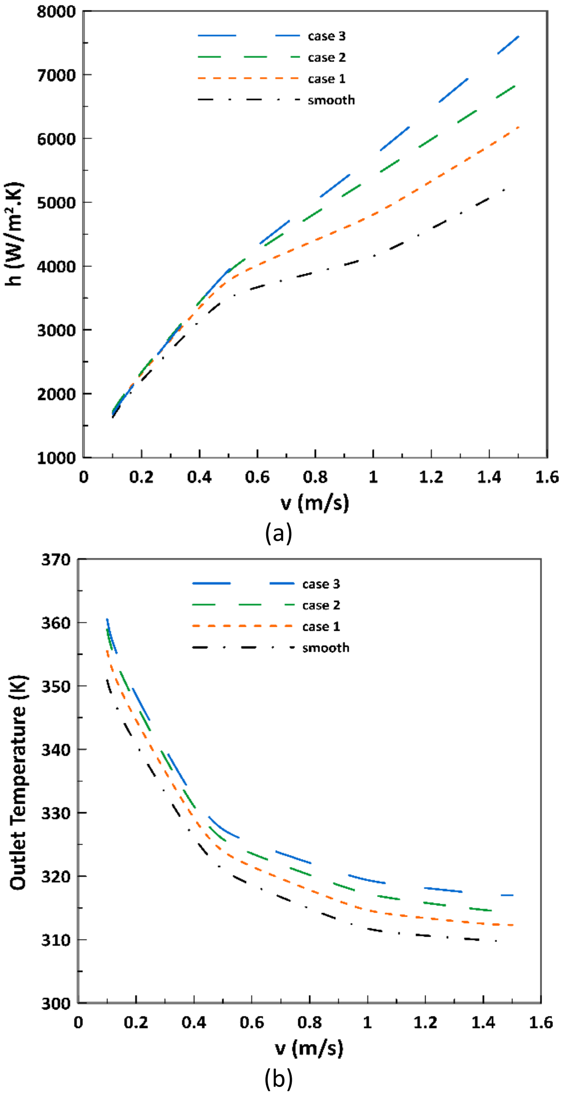


Fig. 5. Effect of concave/convex surface with different numbers on heat transfer characteristics (a) heat transfer coefficient and (b) outlet temperature

5.4 The Effect of Hybrid Nanoparticles and Concave/Convex Surface

Figure 6 illustrates the changes in heat transfer characteristics as fluid velocity varies for a modified concave/convex pipe surface where nanofluid flows: Figure 6(a) heat transfer coefficient and Figure 6(b) outlet temperature. Each concave/convex configuration case is tested with the three different hybrid nanoparticle concentrations: 1%, 5%, and 10%. Augmenting the concave/convex number and hybrid nanoparticle concentration substantially enhances the heat transfer coefficient and the outlet air temperature. As can be seen, the heat transfer coefficient for case 3, with a 10% concentration, is greater than that of case 1, with a 1% concentration, by about half at an inlet velocity of 1.5 m/s. At the same time, the outlet temperature of case 3 (10% concentration) is approximately 10 K higher than that of case 1 (1% concentration). This is related to the above reasons regarding enhancing the fluid's thermal properties and the improved heat transfer area.

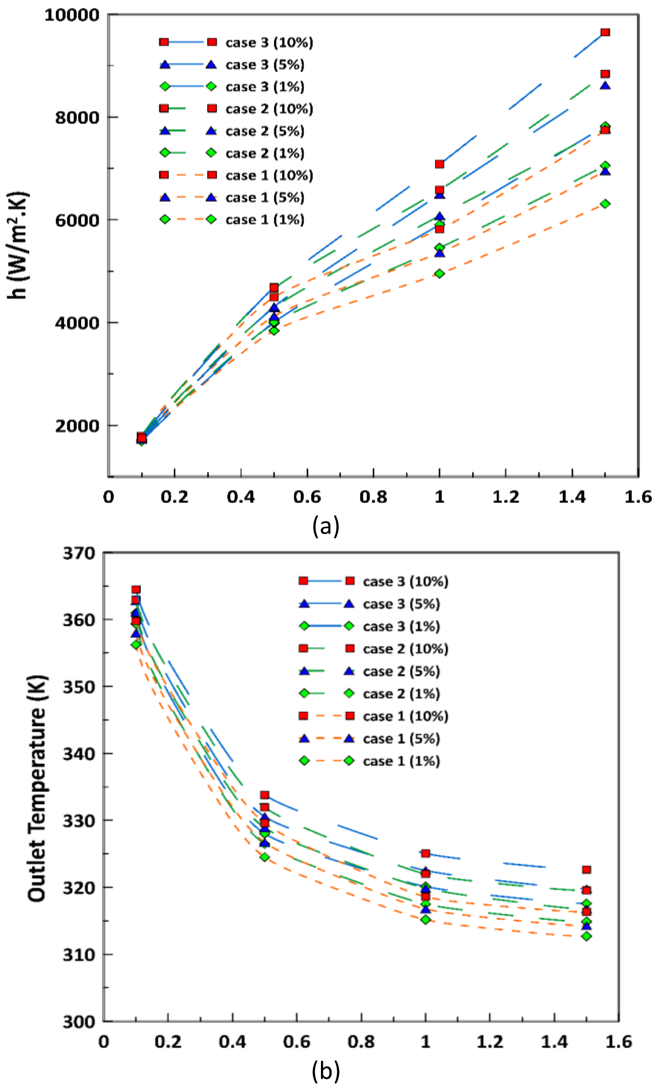


Fig. 6. Effect of concave/convex surface and hybrid nanoparticles on heat transfer characteristics (a) heat transfer coefficient and (b) outlet temperature

5.5 Friction Factor

One of the critical variables in determining the heat exchanger's dynamic performance is the friction factor (f). Several variables, including fluid velocity and pressure drop, determine this component. Figure 7 reveals the impact of adding hybrid nanoparticles into water on f (Figure 7(a)) as long as the impact of utilizing nanoparticles while employing a concave/convex surface on f (Figure 7(b)). As the inlet air velocity increases, the value of f drops in both figures. Figure 7(a) illustrates that when the nanoparticle concentration increases, the f value declines, even while the pressure drop increases. The reason behind that is the enhanced density of the nanofluid; this is because f is directly proportional to density's inverse. At a concentration of 1%, there is a minor variation in the f curve compared to the scenario with pure water. While at 10% concentration, the f drops by about 19.13%, 9.1%, 11.56%, and 12.79% at an inlet velocity of 0.1, 0.5, 1, and 1.5 m/s, respectively, relative to 0% concentration (pure water). These results indicate that the higher reduction in f value occurs when the flow regime is laminar (velocity = 0.1 m/s).

Contrarily, Figure 7(b) shows that a high concave/convex number increases the f value considerably. A higher concave/convex number increases the friction area between the solid and fluid, leading to a higher friction factor. The figure also indicates that as the flow transitions from laminar (v = 0.1 m/s) to turbulent (v = 0.5 m/s), the f value rises. Subsequently, it diminishes as the velocity elevates. Based on case 1 (15 concave/convex), at 10% nanoparticles concentration and one m/s velocity, the f value rises by about 75% and 150% when the concave/convex number is doubled (case 2) and tripled (case 3), respectively.

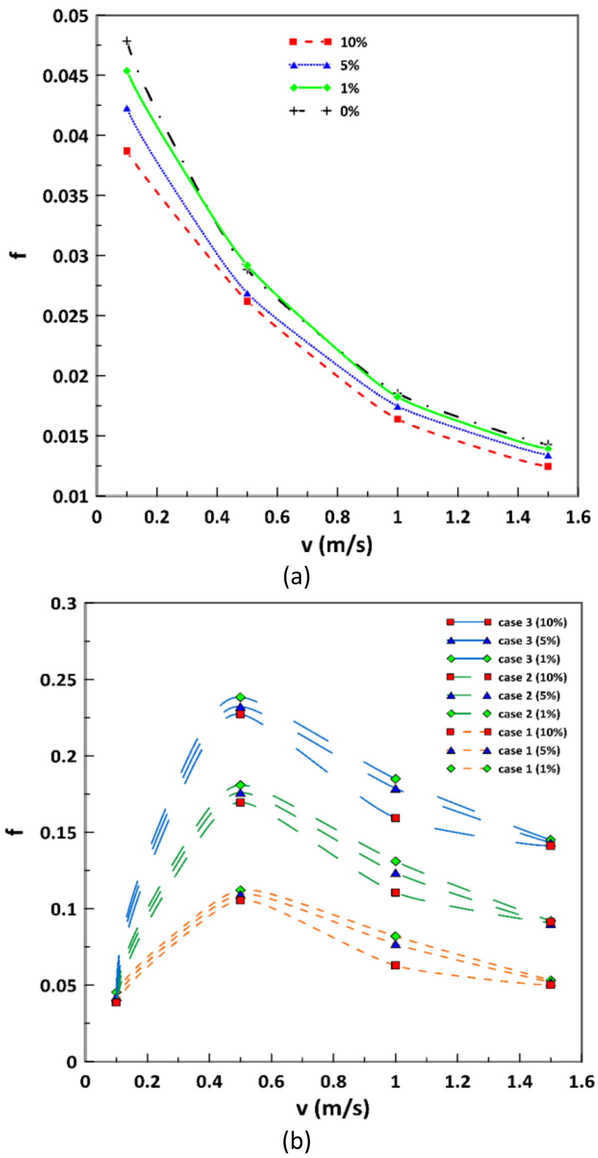


Fig. 7. Effect of (a) hybrid nanoparticles only and (b) concave/convex surface with hybrid nanoparticles on friction factor

5.6 Thermal Performance Factor

Clearly, the findings indicate that a concave/convex surface leads to a simultaneous augmentation in heat transfer and friction factor. Thus, it is necessary to take both factors into account in order to select the most efficient option. The heat exchanger's dynamic and thermal performance is evaluated by using the thermal performance factor (TPF). Figure 8 demonstrates the impact of hybrid nanoparticles as well as the concave/convex configurations on the TPF. The figure shows that when nanofluid is present on a concave/convex surface, the TEF increases to a value greater than one.

Moreover, as the concentration of nanoparticles increases, the TEF also increases. Furthermore, concave/convex case 2 exhibits the best TPF out of the three cases. Nevertheless, using a 10% volume concentration of nanofluid in concave/convex case 2 pipe yields the best thermal/dynamic performance.

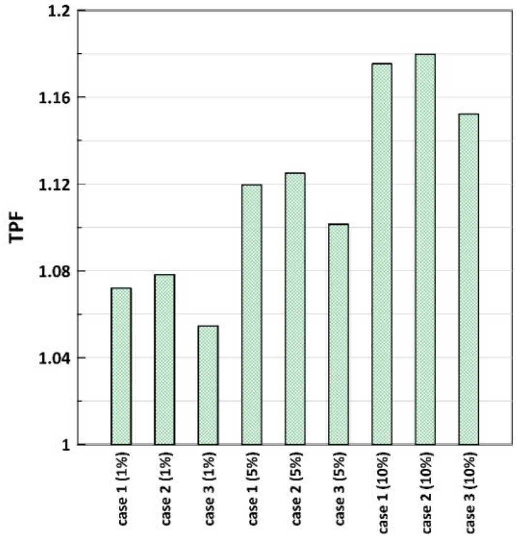


Fig. 8. Effect of concave/convex surface with hybrid nanoparticles on thermal performance factor

This study finally demonstrates that using nanofluids and surface modifications in heat exchangers increases thermal performance, which proves essential to real applications. For example, heat exchangers can be made more effective by increasing thermal conductivity and heat transfer coefficients, achieved in nanofluids (i.e., engineered suspensions of nanoparticles within base fluids). These can increase heat transfer rates by as much as 30% compared to conventional fluids, such as metallic oxide nanoparticles, which have been highlighted in numerous studies. Moreover, fouling can be reduced, and fluid flow rates can be improved through the surface modifications of heat exchanger tubes to optimize their performance and potentially save energy in industrial processes. Collectively, these advancements enable more effective thermal management for a wide array of applications, such as those prevailing in the automotive and energy sectors.

6. Conclusion

A tube has been chosen to work on flow properties due to hybrid Nanoparticle-based materials at constant surface temperature 373 K. By using ANSYS 2024, two-dimensional partial differential equations have been selected as simulating functions with volume ratios of water over Al_2O_3-Cu nanoparticles dispersed at the levels of 1%, 5 %, and 10 %. Three cases for dimples on the tube surface are investigated and demonstrate that fluid velocities vary from 0.1 to 1.5 m/s with an input temperature of 27 °C. The findings suggest that some physicochemical problems would be amplified while others diminished by increasing Al_2O_3–Cu nanoparticle concentrations. In particular, Nu increases with increasing particle concentration.

Furthermore, the thermal performance factor significantly improves with an increase in dimples, underscoring the beneficial effects of nanoparticle concentration and surface modifications on heat transfer efficiency. The principal findings are as follows:

i. An average increase in heat transfer coefficient of 2.14%, 10.8%, and 21.82% for hybrid nanoparticle concentration ratios of 1%, 5%, and 10% by volume, respectively, for velocity of 1 m/s.
ii. The highest variation is around 22.22% compared to pure water and is observed when the velocity is 1.5 m/s and the concentration is 10%.
iii. Compared to smooth pipe at a velocity of 1 m/s, the heat transfer coefficient improves by an average of 15.67% in case 1, 30.01% in case 2, and 37.72% in case 3.
iv. Concave/convex case 2 exhibits the best TPF out of the three cases.

Future research should explore the optimization of hybrid nanoparticle concentrations integrated with inserted shapes inside the tube. This enhancement is expected to increase heat transfer rate and friction; therefore, the thermal performance factor will be the judgment.

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