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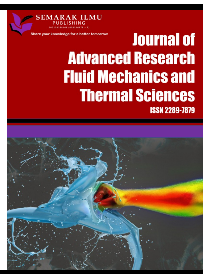

Experiencing the effect of dimple diameter on heat transfer characteristics of a tube using (Al_2O_3 and CuO)/H_2O nanofluids (NFs)

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ABSTRACT

Enhancing the heat transfer in heat exchangers is crucial for various industrial applications. One promising approach is the use of dimpled tubes containing nanofluids (NFs). This study investigates the effect of different dimple diameters on the heat transfer characteristics of dimpled tubes using NFs, aiming to improve the thermal performance of heat exchangers. Numerical simulations were conducted in ANSYS Fluent 17.2 to analyze the friction factor and Performance Evaluation Criteria (PEC), assessing the feasibility of employing dimpled tubes with NFs. Water was selected as the base fluid, with CuO and Al_2O_3 nanoparticles (NPs) of 30 nm diameter. The simulations were performed under low turbulent flow conditions, with Reynolds numbers ranging from 2500 to 6000. The study examined the impact of three dimple diameters (1 mm, 2 mm, and 4 mm) and varying NF concentrations (0.1%, 0.3%, and 0.5%) for both CuO/H_2O and Al_2O_3/H_2O NFs. Results indicated that increasing dimple diameter enhances both the Nusselt number and friction factor. Although PEC also increased, higher Reynolds numbers led to reduced heat exchanger efficiency. Among the NFs, Al_2O_3/H_2O demonstrated superior heat transfer properties and overall performance. Increasing NFs volume fraction up to 0.5% further improved the Nusselt number and PEC. The optimal configuration was achieved with a 4 mm dimple diameter and 0.5% Al_2O_3/H_2O at a Reynolds number of 2500, yielding the highest PEC value of 3.77.

1. Introduction

Enhancing the heat transfer of thermal management systems such as heat exchangers is one of the major challenges facing high-tech industries in maintaining their thermal reliability and ideal performance. The use of plain tubes in heat exchangers suffers several disadvantages reducing their thermal performance. This motivated the researchers to improve the heat transfer through alternative ways that could enhance the efficiency of the heat exchanger at low costs [1, 2]. Heat transfer enhancement could be done in two ways one through active and another through passive. The active methods of enhancing heat transfer include pulsation and vibration of working fluids whereas the passive methods mostly involve the surface or shape modification techniques through

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extending the heat transfer area or creating turbulence of fluid flow. This could be done through displacing the external inserted systems such as altering the fluid pathway, or through the surface roughness of fins, ribs, protrusions, and dimples. Geometrical modifications through "tarbulization" is another effective approach to getting low pressure-loss penalty applied in a wide range of heat transfer applications [3, 4].

The use of dimple tubes is the most widely used method for enhancing heat transfer in many applications including heat exchangers. Dimple tubes have depressions on their external surfaces through which flow can separate, creating turbulence by destroying the laminar boundary layer which improves the heat transfer by improving the convective heat transfer. By adjusting geometric properties, such as the giving diameter to the dimples, the heat transfer efficiency can be optimized to achieve maximum enhancement [5, 6]. Rao *et al.,*[7] reported the enhancement in heat transfer through spherical and tetrapod dimples on the plate by creating the formation of vortical patterns, which noticeably affects more areas enhancing the heat transfer. Garcia *et al.,*[8] compared three kinds of surface modification on tubes such as dimpled tube, corrugated tube, and through inserting wire coil. They concluded that the favorability of each type is dependent on the Reynolds number, as the shape of the roughness has a greater impact on pressure loss than on the increase in heat transfer. Wang *et al.,*[9] experimentally investigated both ellipsoidal and spherical dimples on the inner surface of a heat exchanger tube, establishing correlations for the Nusselt number and friction factor. The results demonstrated that tubes with ellipsoidal dimples had significantly higher Nusselt numbers and friction factors. Additionally, their empirical study highlighted the accelerated transition caused by the roughness of ellipsoidal dimples.

The improvement in the properties of fluid is another passive technique that contributes to enhancing heat transfer. So combining this technique with the surface modification of external surfaces leads to enhancing the heat transfer of the whole system. The use of nanofluids (NFs) as a working fluid is reported to be the best media for enhancing heat transfer due to their higher thermal conductivity, thermal stability, and better dispersion to transfer heat through particle-to-particle interaction [10, 11]. Li *et al.,*[12] achieved an enhancement in heat transfer to about 39% using a 2% volume fraction of CuO/H_2O NFs. Similarly, Suresh *et al.,*[13] investigated the hydraulic performance of CuO/H_2O NFs varying their concentrations (0.1%-0.3%) flowing in helically dimpled tubes under low turbulent conditions. They found an increase in Nusselt number by 19.49%, 627.06%, and 39.50% using NFs in volume fractions of 0.1%,0.2%, and 0.3% respectively. Similarly, Nguyen *et al.,*[14] reported a 40% enhancement of the Nussle number using 6.8% of Al_2O_3/H_2O NFs for enhancing the cooling performance of electronic components. Awis *et al.,*[15] achieved good thermo-hydraulic efficiency with a low entropy generation rate using Soyabean oil/MXene-0.075wt.%. The literature reported the positive career of NFs towards enhancing the heat transfer of devices.

Despite the extensive research on heat transfer enhancement techniques, the combined effects of dimpled tube geometries and NFs remain underexplored, particularly in the context of varying dimple diameters and NF concentrations. Addressing this gap is crucial for optimizing the design of heat exchangers to achieve higher efficiency and performance. This study is significant as it provides a comprehensive analysis of how different dimple diameters and NFs concentrations affect heat transfer characteristics, friction factor, PEC, and Nusselt number. This study aims to investigate the effect of three different dimple diameters (1 mm, 2 mm, and 4 mm) with two different NFs (Al_2O_3/H_2O, CuO/H_2O) at various concentrations (0.1%, 0.3%, and 0.5%) on the heat transfer characteristics of tubes using ANSYS Fluent. The heat transfer characteristics, friction factor, PEC, and N_u were examined to understand the influence of different dimple diameters and NF volume fractions on the performance of the tubes.

2. Methodology
2.1 Model Description and Meshing

Figure 1a shows a dimpled tube model for heat exchangers. It is 75 mm long with a 20 mm hydraulic diameter. The dimples, spaced 15 mm apart with 10.5 mm between them, form five rows of six dimples each. The study varies the dimple diameters to see their impact. Figure 1b depicts the 3D model, consisting of a dimple section and a 300 mm extension section to create turbulent flow. The inlet of the tube is kept before the dimple section of the tube and the outlet is after the dimpled section. Figure 2c presents the tube's mesh, which is finer near the dimples for better accuracy. Figure 2d shows additional mesh refinement along the tube wall to analyze fluid behavior and heat transfer more precisely.

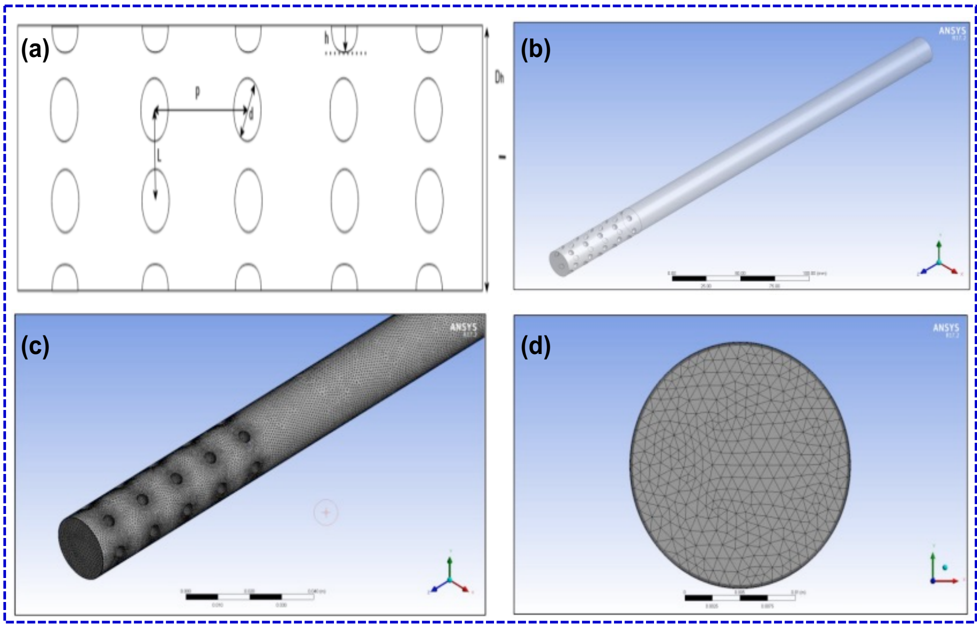

Fig. 1. (a) Dimple tube model (b) 3D model of dimple tube Meshing (c) Isometric View (d) front view of Dimpled Tube

2.2 Governing Equation

This study used the equation Eq. (1-3) of conservation of energy, momentum, and mass, investigated from the previous study by Khouzestani *et al.,*[16] for the numerical solution of finite volume analysis in ANSYS

$$\frac{\partial \rho U_i}{\partial X_i} = 0 \tag{1}$$

$$\frac{\partial \rho U_i U_J}{\partial X_I} = \rho g_i + F_i - \frac{\partial P}{\partial X_i} + \frac{\partial}{\partial X_i}\left(2\mu S_{ij}\right) \tag{2}$$

$$\frac{\partial \rho U_i E_o}{\partial X} = \rho \mu_i F_i - \frac{\partial q_i}{\partial X_i} + \frac{\partial}{\partial X_i}\left(U_i T_{ij}\right) \tag{3}$$

2.2.1 Thermal Properties of Nanofluids

The study used the following equations Eq. (4-5) for density (ρ_{nf}) and heat capacity (C_p) respectively investigated from the previous studies by Xuan *et al.,*[17] and Ho *et al.,*[18] listed below;

$$\rho_{nf} = (1 - \alpha)\rho_f + \alpha\rho_p \quad (4)$$
$$(\rho C_p)_{nf} = (1 - \alpha)(\rho C_p)_f + \alpha(C_p)_p \quad (5)$$

The correlation for the dynamic viscosity of NFs (μ_{nf}) and effective thermal conductivity (k_{nf}) are used developed by Azmi *et al.,*[19] using the equations Eq. (6-7) given below;

$$\mu nf = \mu w \left(1 + \frac{\emptyset}{100}\right)^{11.3} \left(1 + \frac{T_{nf}}{70}\right)^{-0.038} \left(1 + \frac{d_p}{170}\right)^{-0.061} \quad (6)$$
$$K_{nf} = k_w 0.8938 \left(1 + \frac{\emptyset}{100}\right)^{1.37} \left(1 + \frac{T_{nf}}{70}\right)^{0.277} \left(1 + \frac{d_p}{150}\right)^{-0.0366} \left(\frac{\alpha_p}{\alpha_w}\right) \quad (7)$$

Additionally, the equations Eq. (8-9) is used for determining the thermal conductivity of water (k_w) and dynamic viscosity (μ_w) are given below;

$$kw = 0.56112 + 0.00913 \times Tw - 2.60152749e - 6 \times (TW)^2 - 6.08803e - 8 \times (TW)^3 \quad (8)$$
$$\mu w = 0.00169 + 4.25263e - 5 \times Tw - 4.9255e - 7 \times (TW)^2 - 2.099 - 9 \times (TW)^3 \quad (9)$$

2.2.2 Data Reduction

The Reynolds number (Re) is the ratio of inertial forces to viscous forces, and it predicts whether the fluid flow will be turbulent or steady when passing through a body or duct which could be calculated using Eq. (10);

$$Re = \frac{\rho U_{in} D_h}{\mu} \quad (10)$$

Heat transfer performance can be evaluated by determining the average convective heat transfer coefficient, (h_c). It is calculated using Eq. (11)

$$h = \frac{q}{\Delta T} \quad (11)$$

where $\Delta T = Tw - Tf$

The Nusselt number (is utilized in heat transfer calculations to quantify the convective heat transfer coefficient between the surface and fluid. It can be determined using the Eq. (12);

$$Nu = \frac{hDh}{k} \quad (12)$$

The Fanning friction factor, denoted by (f), represents the resistance of the fluid due to both the internal roughness of the pipe and the velocity of the fluid which could be calculated by using Eq. (13);

$$f = \frac{2(\frac{\Delta P}{L})D_h}{\rho U_{in}} \quad (13)$$

The PEC could be determined by combining the impact of both the heat transfer enhancement with friction factor on the overall thermal performance of the system which could be calculated by using Eq. (14);

$$\mathrm{PEC} = \frac{\frac{\mathrm{Nu}}{\mathrm{Nuo}}}{\left(\frac{\mathrm{f}}{\mathrm{fo}}\right)^{1/3}} \tag{14}$$

2.2.3 Boundary Conditions

To streamline the project and minimize unpredictable factors, the boundary conditions were adopted in this numerical study by Xie *et al.,*[20]. The flow is considered as steady and incompressible without any slip and gravitational effects on the walls which can result in zero velocity (u=0). The temperature conditions at the inlet and wall are specified as T_{in}=300K and T_{wall}=350K. The k-ε turbulence model is used in the simulation of a dimpled tube with the Reynolds number ranging from 2500 to 6000. The convergence is set to 1×10^{-6} for investigating the heat transfer for dimpled tubes using NFs.

2.2.4 Validations

The data validation of simulation for both the simple tube and dimpled diameter tube involved the comparison of Nusselt number values with the values predicted by the Gnielinski correlation and by the Wang *et al.,*[21] respectively. For a simple tube, Figure 2a shows that the Nusselt number has a minimum error percentage of 3.7% and a maximum error percentage of 18.8%. Despite some deviation between the simulation results and Gnielinski's analytical solution, these errors are considered acceptable. For the dimpled diameter tube, Figure 2b indicates a minimum error percentage for the Nusselt number of 19.1% and a maximum error percentage of 25.9%. This variation might be due to some of the parametric differences such as heat loss, length of tube, and fluid properties between the simulation and the experimental setup. Nevertheless, despite these differences, the results of simulations still follow a trend consistent with the experimental, which is deemed acceptable.

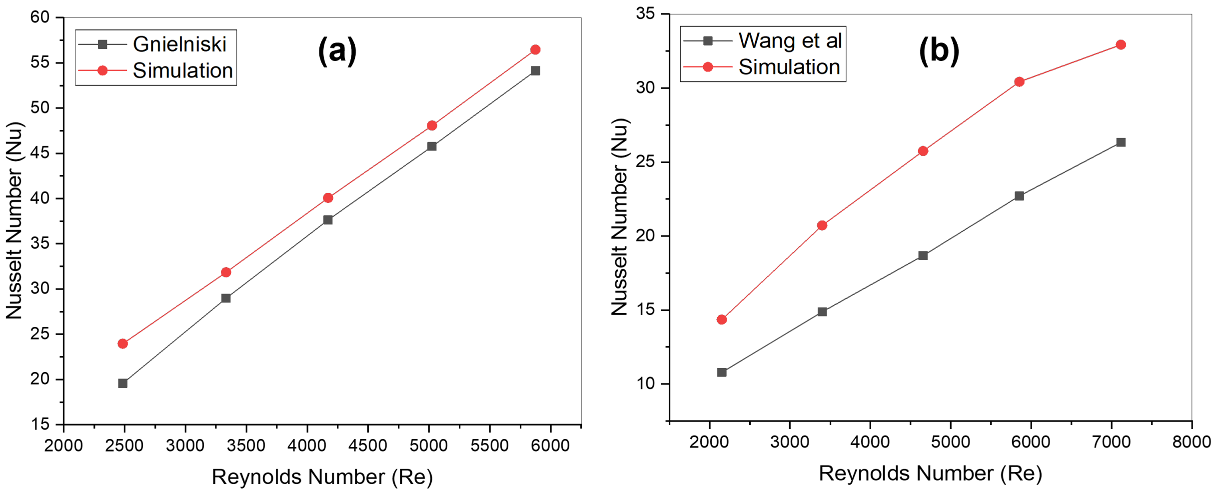

Fig. 2. Validation for a) Smooth tube b) dimple diameter tube

2.2.5 Grid independence test

The Grid Independence Test is vital for evaluating mesh quality and optimizing simulation time during data collection. It ensures the model's accuracy by analyzing how sensitive the observed parameter is to changes in the number of mesh elements. Table 1 illustrates that although Mesh-1 and Mesh-2 differ by around 20,000 elements, their Nusselt numbers differ by only about 0.864%. This slight variation confirms that Mesh 2 is reliable and suitable for use across all models in this study. Implementing the Mesh 2 structure in all models allows for achieving reliable results while minimizing computational effort.

Table 1
Grid independence test data

Mesh (M)	Elements	Nusselt Number (Nu)	%Age difference
M-1	1578996	26.8977	--
M-2	1780177	27.1321	0.864

2.2.6 Properties of Nanofluids

Table 2 shows the properties of CuO/H_2O and Al_2O_3/H_2O NFs used in this study, all of which have a consistent diameter of 30 nm. These properties were calculated for the simulation to accurately represent the fluids, accounting for variations based on the type of NFs used in the study.

Table 2
Properties of CuO/H_2O and Al_2O_3/H_2O NFs at different weight loadings

Volume fraction	Density (ρ_{nf})		Heat capacity (C_p) J/kg.K		Dynamic viscosity (μ_{nf}) Kg/ms		Thermal conductivity W/mK	
	CuO	Al_2O_3	CuO	Al_2O_3	CuO	Al_2O_3	CuO	Al_2O_3
0.1	1547	1293	2420	2895	0.0022	0.0022	0.76	0.72
0.3	2649	1887	1098	1542	0.0151	0.0151	0.92	0.91
0.5	3752	2482	554	837	0.0764	0.0764	1.12	1.11

3. Results and discussion
3.1 Effect of dimple diameter

This study reported the effects of increasing the dimple diameter (1mm,2mm, and 4mm) on the heat transfer characteristics of the tube affecting the performance of the heat exchanger. For this, the performance evaluation of 0.3% of Al_2O_3/H_2O is examined flowing inside three diameters dimpled tubes. From Figure 3a, it can be shown that the dimple diameter of 1mm with Al_2O_2/H_2O NFs improves the Nusselt number to 33.24%. However, increasing the dimple diameter to 2mm further increases the Nusselt number to 49.86% compared to the simple tube and 17% for the tube with a dimple diameter of 1mm. However, doubling the dimple diameter to 4mm increases the heat transfer to 66.41% and 16.55% compared with a simple tube and a 2mm dimple diameter dimple tube. This referred to the heat transfer becoming less significant on increasing the dimple diameter. The friction factor pattern is shown in Figure 3b, which shows that increasing the dimple diameter also raises the friction factor. When comparing the friction factors of dimpled tubes with diameters of 1mm, 2mm, and 4mm to that of a smooth tube, increases of 4.03%, 4.55%, and 19.49% are observed, respectively. Notably, at Re = 4250, the friction factor for the 2mm dimple diameter drops sharply, eventually falling below that of the 1mm diameter. This suggests that the 2mm dimples have a lower friction factor at higher Reynolds numbers. However, additional research is needed to confirm this observation and to draw definitive conclusions about the friction factor behaviour at different dimple diameters.

Figure 3c illustrates the performance of the heat exchanger based on the PEC for the three different tubes. The data shows that the PEC increases with the dimple diameter. For all dimple diameters, the PEC trend was inversely proportional. However, for the 2mm diameter, starting from a Reynolds number (Re) of 4250, the PEC values increased due to a previously discussed decrease in the friction factor. The highest PEC, at 2.656, was observed for the 0.3% Al_2O_3/H_2O mixture with a 4mm dimple diameter at Re = 2500. Conversely, the lowest PEC of 1.575 occurred with a 1mm dimple diameter at Re = 6000. Within the range of 2500 < Re < 6000, all cases of the 0.3% Al_2O_3/H_2O mixture had PEC values greater than 1, indicating superior performance compared to a smooth tube. Notably, the PEC trend for the 4mm dimple diameter was more erratic compared to the 1mm dimple diameter, suggesting that increasing the dimple diameter at higher Reynolds numbers might reduce the heat exchanger's effectiveness.

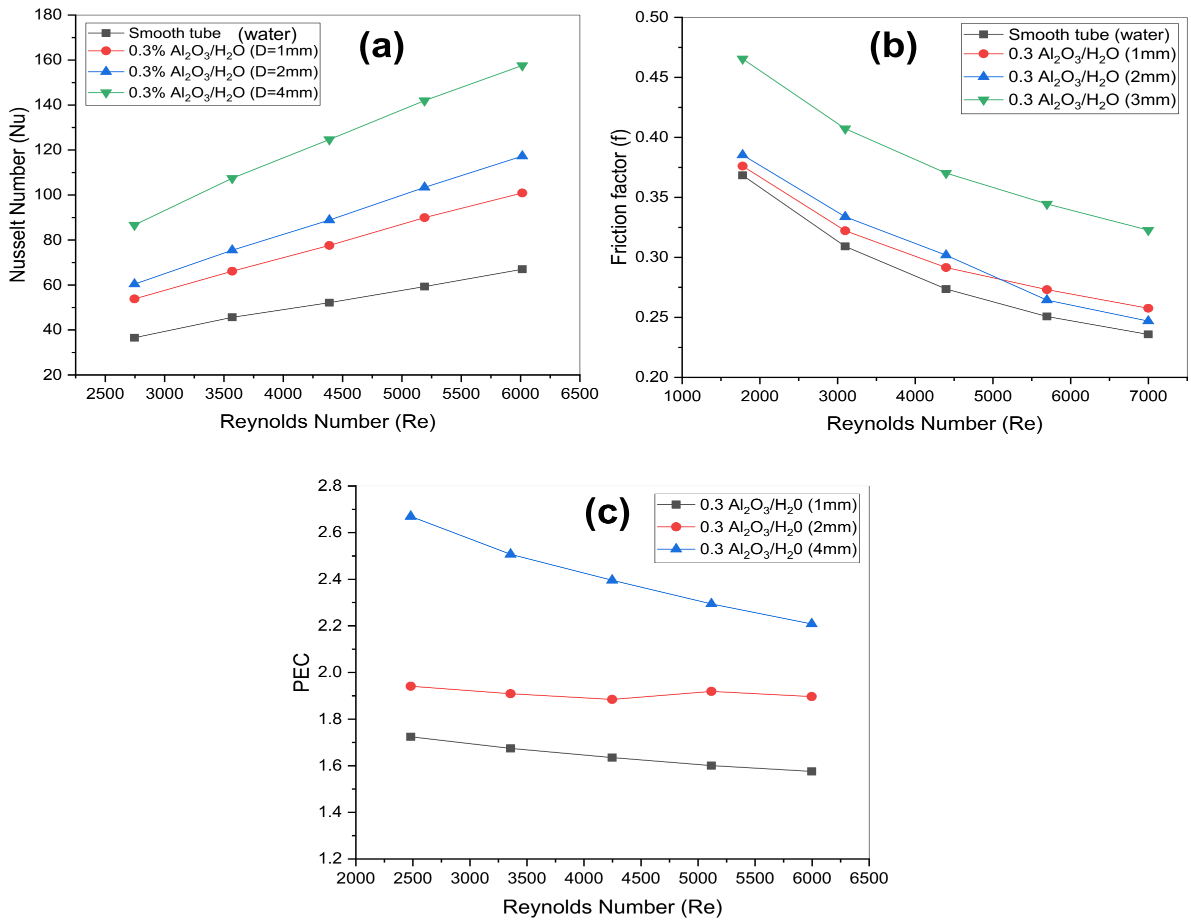

Fig 3. Effect of different dimple diameters on a) Nusselt Number b) friction factor c) PEC

For further support of the results, the temperature distribution counters of NFs flowing inside three different dimple-diameter tubes are shown in Figure 4a. One of the key differences identified in these dimpled tubes was the fluid behaviour behind the dimples. It was noted that heat transfer was highest in the region behind the dimples as compared to other regions. This can be attributed to the fluid colliding with the dimples, which disrupts the flow and generates turbulent mixing. This turbulence reduces the boundary layer near the wall, enhancing convective heat transfer and increasing heat transfer from the wall to the fluid. It was concluded that larger dimple diameters intensified this turbulent mixing. The increased turbulence led to higher convective heat transfer reflected in a rise in the Nusselt Nusselt numbermber showing a correlation between stronger turbulence and higher Nusselt numbers with larger dimple diameters.

Figure 4b illustrates and explains the velocity distributions within the tube. The dimples in the tube narrowed the fluid's flow path, causing an increase in velocity as it moved through the tube, as shown in the velocity contour. Interestingly, as the dimple diameter increased, the velocity was significantly lower than expected. Larger dimples enhanced turbulence mixing, increasing overall turbulence in the tube and disturbing the flow. Consequently, the fluid experienced more random movement, reducing its velocity. This was particularly evident behind the dimples, where the velocity was the lowest. Additionally, when the fluid collided with the dimples, it lost momentum, slowing down the flow. This deceleration was observed by comparing the velocity behind the dimple tips to the velocity before the fluid collided with the dimples, with the former being lower.

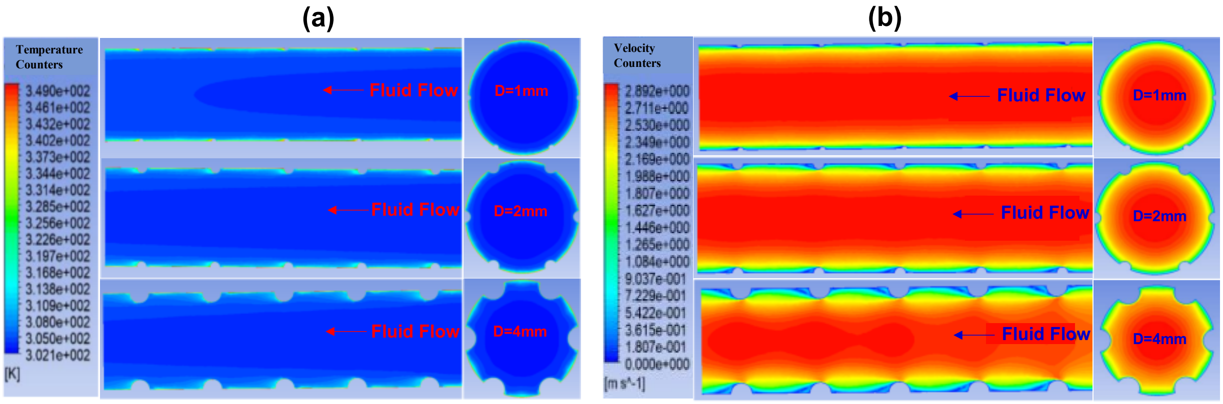

Fig 4. Counters of a) Temperature and b) Velocity for different dimple diameters tube

3.2 Effect of nanofluids and volume fractions

In this study, Al_2O_3/H_2O and CuO/H_2O NFs were used to examine their effects on thermal properties and performance evaluation of heat exchangers. The Al_2O_3/H_2O NFs consistently exhibited high Nusselt numbers in comparison with CuO/H_2O in all working conditions. The smallest variation in Nusselt numbers was observed in the tube with a 0.1% volume fraction at a Reynolds number of 2500, displaying a 7.9% difference. The largest difference was observed in the tube with a 0.5% volume fraction at the same Reynolds number, with a 16.5% difference shown in Fig 5a. When comparing friction factors based on dimple diameter, both NFs showed similar values, indicating that dimple diameter had the most significant impact on friction factors. Both NFs displayed a trend of decreasing friction factors with increasing Reynolds numbers, indicating lower resistance at higher fluid velocities shown in Fig 5b. In terms of the PEC, the Al_2O_3/H_2O NFs has higher values than the CuO/H_2O, which can be attributed to its higher Nusselt numbers. The smallest and largest differences in PEC corresponded to the same conditions as the Nusselt number differences, with the results mainly influenced by Nusselt numbers and only minimally affected by friction factors shown in Fig 5c. Three different volume fractions were used for both Al_2O_3/H_2O and CuO/H_2O NFs to study their effect on heat properties and heat exchanger performance. From Fig 5a It was found that increasing the volume fraction up to 0.5% improved heat transfer properties, as indicated by the Nusselt number, especially within the range of 2500<Re<6000. For a clearer understanding, let's consider an example using CuO/H_2O NFs with a 1mm dimple diameter. With a 0.1% volume fraction of CuO, the Nusselt number increased by 7.85% at Re = 2500, but this enhancement dropped to 2.17% at Re = 6000. With a 0.3% CuO volume fraction, the Nusselt number showed a significant increase of about 35.08% at Re = 2500 and 31.79% at Re = 6000. For a 0.5% CuO volume fraction, the enhancement was still present but less pronounced, with increases of approximately 49.99% at Re = 2500 and 47.61% at Re = 6000. Despite increasing the volume fraction of the NFs, the friction factor values remained almost the same shown in Fig 5b. This consistency is attributed to the dominant influence of dimple diameter on the friction factor, making the effect of volume fraction less significant. The PEC increased with higher volume fractions of NFs shown in Fig 5c. Using the earlier example, the 0.1% CuO-$_{H2O}$ nanofluid in a tube with a 1mm dimple diameter yielded a PEC value of 1.07 at Re = 2500, which dropped to 0.96 at Re = 6000. At a 0.3% volume fraction, the PEC value rose significantly to 1.51 at Re = 2500 and 1.38 at Re = 6000. However, with a 0.5% volume fraction, the increase was less substantial, with a PEC value of 1.96 at Re = 2500 and 1.8 at Re = 6000. In conclusion, increasing the volume fraction of NFs enhances heat exchanger performance, but the extent of this enhancement decreases as the volume fraction increases.

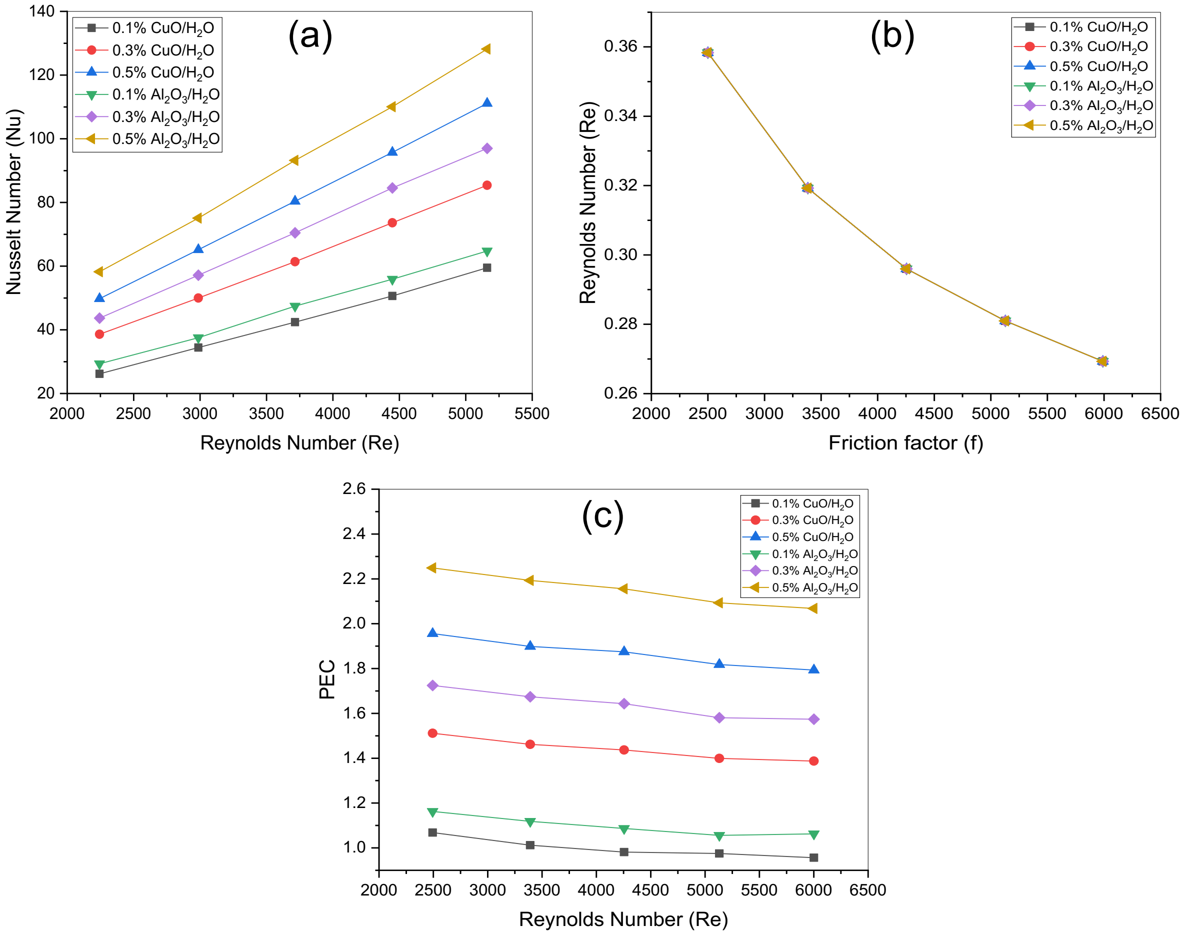

Fig. 5. Effect of nanofluid and their different weight fraction on a) Nusselt number b) Reynolds number and c) PEC

4. Conclusion

This study numerically examined how dimple diameter, NFs type, and volume fraction affect heat transfer and heat exchanger performance using ANSYS Fluent 17.2, validated against analytical and experimental data. Key findings are:

1. Larger diameters enhance heat transfer by increasing the Nusselt number and turbulence but also increase friction. Optimal efficiency was observed at low turbulence, with diminishing returns at higher flow rates.

2. Al_2O_3-H_2O showed better heat transfer than CuO/H_2O NFs, with higher Nusselt numbers and similar friction factors. Al_2O_3/H_2O NFs proved superior within the Reynolds number range of 2500-6000.
3. Increasing volume fraction up to 0.5% improved heat transfer, particularly at lower Reynolds numbers, without significantly affecting friction factors. Higher volume fractions enhanced performance but were less effective at higher flow rates.

The optimal enhancement was achieved with a 4mm dimple diameter and 0.5% Al_2O_3/H_2O NFs at Re = 2500, yielding a PEC value of 3.77, a Nusselt number of 113.9, and a friction factor of 0.425, representing the best balance of heat transfer and fluid resistance within the tested range.

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