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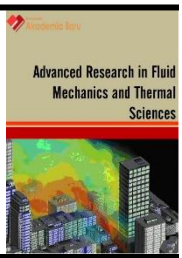

Investigation of Heat Transfer Through Dimpled Surfaces Tube with Nanofluids

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ABSTRACT

Nowadays, the energy resources are almost depleted due to the tremendous demand. Hence, the efforts were paid worldwide to innovate new heat transfer technique to get satisfying outcomes with less cost. Heat transfer enhancement by surface modification is commonly practiced throughout the last decades because it helps to reduce both the size and cost of heat transport devices. The numerical investigation of forced convective heat transfer inside circular tube with dimpled surface of inline arrangement, the dimple size was denoted in term of d/D, it was in range of (1/6).under internal turbulent flow conditions has been studied with ZnO water nanofluids of (0.2, 0.6, 0.8 and 1) % volume fractions were used. The size and arrangement of dimples were conducted. The gained outcomes showed the heat transfer coefficient is directly proportional with both dimples size and nanofluids volume fraction. It was concluded a maximum enhancement achieved is 2297 at Reynold number of 12000 and volume fraction of 1. While the poorer performance was 1013 at Reynold number of 6000 and volume fraction 0.2.

1. Introduction

The growth of technology forced other fields to get along with this progress. The conventional ways cannot provide the required results anymore. That is why the finding out a suitable solution for the emerged problems is a crucial issue. Regarding the heat transfer field, heat transport devices play a vital role in all aspects of life, especially in industry. A compact heat exchanger is required and traditional heat exchanger ae disposal due to the physical and functional depreciation of the old techniques. The new trend in the industry is to saving time and effort in each step of production processes. Many researchers examined the use of new ways to attain maximum results in heat exchangers, especially the use of nanofluids, surface modifications. Nowadays, ZnO has received mended attention because of its potential advantage over the group III-nitrides due to its merits as a bulk single crystal having larger excitation binding energy and excellent radiation hardness [1].

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Metal oxide nanostructures have attracted considerable attention for their potential applications in many technologies, such as solar cells, electroluminescent devices, electrochromic windows and chemical sensors [2]. There are many reports on the synthesis, growth kinetics, crystallization of ZnO, and its applications. Among various nanoforms, one-dimensional oriented nanostructures, such as nanorods, nanowires, nanotubes, and nanopins, are very important because of enormous commercial applications [3–11]. To achieve the high performance of these devices, metal oxides are used in these applications. The nanostructured ZnO is found to be one of the metal oxides, possesses high-surface area as well electrical, electrochemical and structural properties. The ZnO powders at nanoparticles size possess medical applications such as antibacterial agent [12]. Recently, sound propagation through random media has also been a subject of great interest. Such media include colloidal suspensions, porous material, magnetorheological medium, and nanofluids. The nanofluid is a colloidal suspension of material nanoparticle in the carrier fluids. The nanofluids have great interest due to a broad application in different fields [13-14]. The effect of both attacked angle and nanofluid was examined numerically under turbulent flow conditions [15]. The reported results showed that the maximum heat transfer enhancement was achieved at an angle attacked by 90°. This angle of attack can increase the thermal performance remarkably but there will be an appreciable increase in pressure loss. Turbulent flow in a tube having perforated conical ring was examined numerically under constant wall flux conditions [16]. The study primarily conducted to reveal the combined effect of both Cu-water nanofluid and insert on heat transfer characteristics. The reported results showed that the tube of quaternary perforation rings exhibits about a triple enhancement that smooth tube. Xie *et al.,* [17] conducted a series of experimental and numerical test in an elliptical dimpled tube to show the effect of such surface modification on heat transfer characteristics. The reported results showed that the Nusselt number increase as the dimple depth and axis ratio increase. Piper *et al.,* [18] investigate the effect of a dimpled surface tube in a heat exchanger, the outcomes revealed that this surface geometry representing by dimples form regular turbulence in the secondary flow region, this turbulence help to break the hydraulic boundary layer and enhance the heat transfer.

In the view of the previous literature, many authors focused on using a solo enhancement techniques and a certain nanofluid, that is the motivation behind the using combined enhancement techniques representing by both dimpled surface along with ZnO nanofluid to reveal the effect of this combination on heat transfer enhancement.

2. Thermophysical Properties of the Nanofluids

The nanofluids were prepared by suspending the 50 nm ZnO nanoparticles in the deionized water for a different range of volume fraction (0.2, 0.6, 0.8 and 1) %. The prepared nanofluids were used to measure the thermophysical properties. The thermal conductivity of nanofluid was measured by galvanized transient hot wire as it was used in the study [14]. It was measured at a temperature range of (30-60) °C for various volume fractions. On the other hand, the dynamic viscosity was measured by sine wave Vibro Viscometer SV-10 after the dispersion of Cu and Zn nanofluid in a water bath. While the other thermophysical properties (heat capacity and density) were evaluated as follow [19].

$$Cp_{nf} = Cp_f + \varphi Cp_{np} \tag{1}$$

$$\rho_{nf} = \rho_f + \varphi \rho_{np} \tag{2}$$

3. Governing Equations and Boundary Conditions

Navier-Stockes is the main equation which governs the turbulent, steady-state flow in tube, it could be expressed as follow

Momentum:

$$\frac{\partial(u_i u_j)}{\partial x_i} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[(\mu_t + \mu)\left(\frac{\partial u_i}{\partial x_i} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\frac{\partial u_l}{\partial x_l}S_{ij}\right)\right] + \rho g_i$$

Equations of k and ε are as follow

$$\frac{\partial(\rho u_i k)}{\partial x_i} = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_t}\right)\frac{\partial k}{\partial x_i} + G_k + G_b - \rho\varepsilon\right]$$

$$\frac{\partial(\rho u_i \varepsilon)}{\partial x_i} = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_t}\right)\frac{\partial \varepsilon}{\partial x_i} + C_1\frac{\varepsilon}{k}(G_k + C_3 G_b) - C_2\rho\frac{\varepsilon^2}{k}\right]$$

where $G_b = \beta g_i \frac{\mu_t}{Pr_t}\frac{\partial T}{\partial x_i}$, $G_k = \mu_t \frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j}\frac{\partial u_j}{\partial x_i}\right)$, $\mu_t = C_\mu \rho \frac{k^2}{\varepsilon}$

Energy equation can be formulated as follow

$$\frac{\partial(\rho u_i h)}{\partial x_i} = -\frac{\partial}{\partial x_i}\left[C_p\left(\frac{\mu}{Pr} + \frac{\mu_t}{Pr_t}\right)\frac{\partial T}{\partial x_i}\right]$$

Mass conservation equation could be expressed as follow

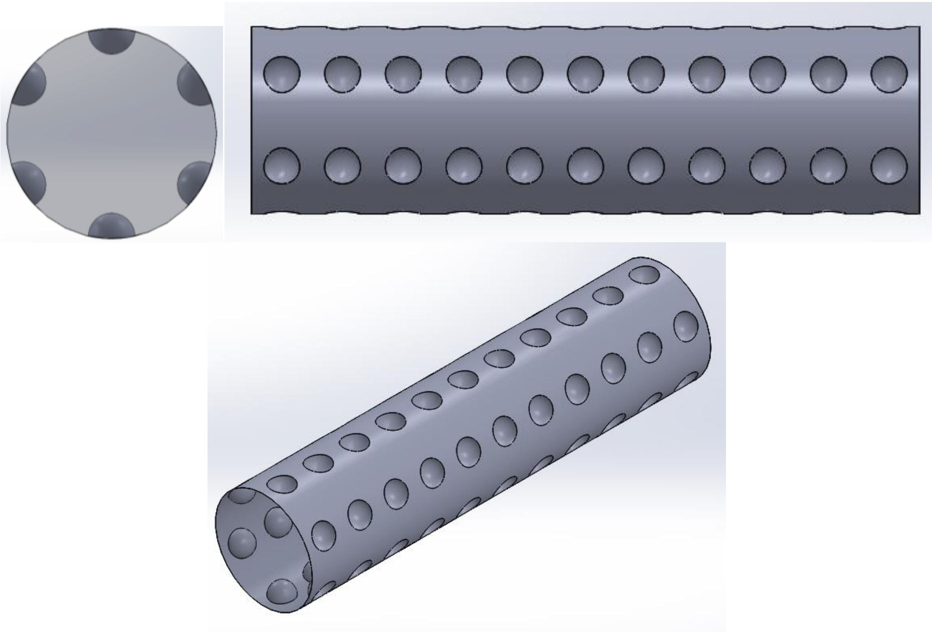
$$\frac{\partial}{\partial x_i}(\rho u_i) = 0$$

The constants C1, C2, C3, C_μ, σ_k, σ_ε have a value of 1.44, 1.92, tanh $|v_p/v_n|$, 1.0. and 1.3 respectively as already stated in previous studies [14, 20].

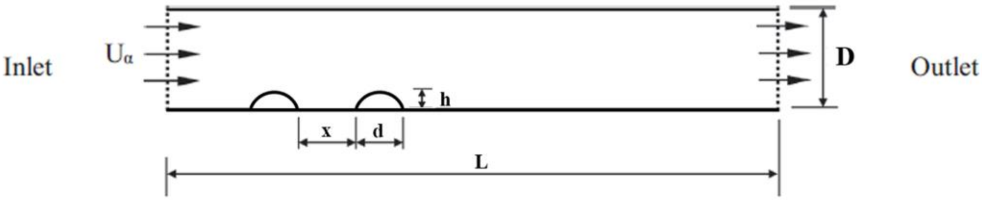
A constant heat flux of 5000 kw/m^2 was applied on the outer surface of the wall, and the inner wall temperature was calculated by a one-dimensional conduction equation. The working fluid is considered as a single phase, Newtonian, and incompressible for the ease of calculations.

4. Mathematical Models

Regarding the numerical simulation of the circular tube which has dimpled surface, the tubes (one smooth and another one is dimpled) were created and modelled by software package. The mesh was performed with the aid of Gambit 2.4.6 environment and ANSYS FLUENT 15.0 was used to simulate the cases. The tubes geometry and specifications are shown and listed as in the Figures 1 and 2 and Tables 1 and 2.



Arrangement of inline dimples on the tube



Computational domain and the dimples geometry
Fig. 1. Dimpled Tube

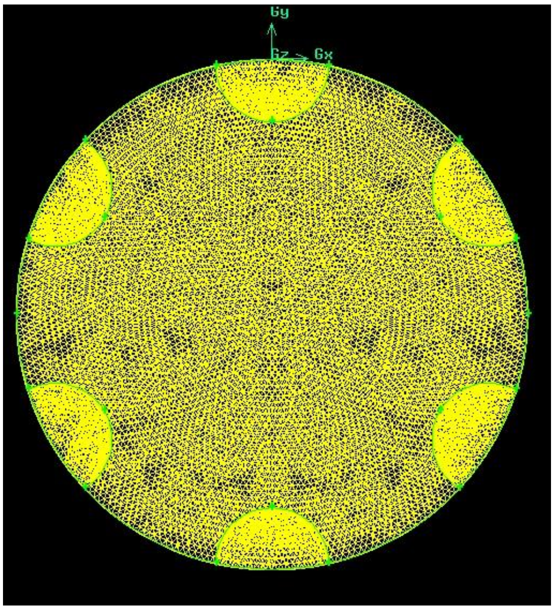


Fig. 2. Geometry grid of dimples tube

Table 1
Specifications of Pipe Heat Exchanger

Specifications	Inner Tube
Unit	Mm
Inner diameter(D1)	25
Length(L)	2000
Thickness(T)	1.5
Layout	Circle
Material	Copper

Table 2
Specifications of Pipe Heat Exchanger

Specifications	Inner Tube
Unit	Mm
Inner diameter(D1)	10
Length(L)	2000
Depth of dimple	1.3
Layout	Circle
Material	Copper

5. Grid Study and Code Validation

The reliability and fidelity of the gained outcomes is correlative with the verification steps. That is why the grid independence study should be performed prior to commence the simulation procedure. It was tested three times each case, and the Grid Independent Index were 0.89 and 0.91 for both friction factor and Nusselt number as depicted in Figure 3. These values are close to one and indicate that the simulation results would be acceptable. Validation process shoal follow verification directly to ensure the accuracy of the grid independent study. The validation conducted with the analogous study [21]. Where a tube of the conical ring was modelled and simulated with the same limitations and dimension to ensure a good match.

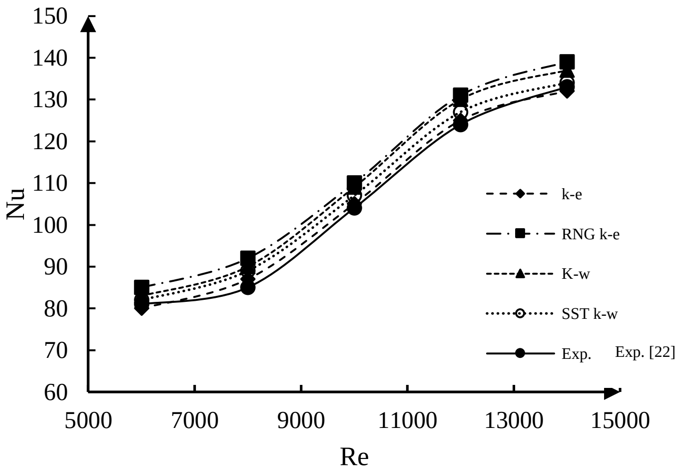


Fig. 3. Comparison results of four different turbulence models with experimental data [22]

6. Grid Independence Test

Grid in-dependency study representing by The Grid Independence Index (GCI) is compulsory for any simulation, and it should be done for any test cases. However. The grid independence has to be proved at any simulation. Anyway, we have to maintain the lowest grid available in order to ease simulation time and keep accuracy. Maybe a non-uniform grid will be the best practice.

The Grid Independence Index (GCI) was checked three times at each case to ensure the fidelity of the study. The current study has five cases, the GCI's values for the case of Reynold number 8000 and volume fraction 1% were 0.89, 0.92 and 0.96. Hence, the case which has GCI 0.96 was selected to the simulation tests.

7. Heat Transfer Coefficient Post Process

There are several ways to measure the thermal heat transfer coefficient. The simplest and most adequate one is the energy balance, which is applicable in the current case if the wall temperature, inlet temperature, and exit temperature are known. The expression of the energy balance could be written as in the following equations:

$$h_x = \frac{q_x}{T_w - T_{bnf}} \tag{3}$$

The Tw holds for local wall temperature at a certain position. While T_{bnf} refers to the bulk temperature of nanofluid.

$$T_{nf(x)} = T_{nf(in)} + \frac{qP\,x}{\dot{m}\,c_p} \tag{4}$$

The P character represents the tube Perimeter, and the Cp_{nf} character denotes to the specific heat of the nanofluid.

The heat flux could be evaluated as follows:

$$q_x = \frac{Q}{A_x} \tag{5}$$

The local Nusselt number can be found by substitute the local heat transfer coefficient in the following formula;

$$\mathrm{Nu}_x = \frac{h_x D}{k_{nf}} \tag{6}$$

The average heat transfer coefficient is:

$$h_{av} = \frac{\int_0^L h_x dx}{x} \tag{7}$$

Eventually, the average Nusselt number could be written as in the following formula:

$$\mathrm{Nu}_{av} = \frac{h_{av} D}{k_{nf}} \tag{8}$$

8. Validation

On set, the validation should be conducted to check the reliability of the current study. Validation is the first step which should be done prior to continuing in tests to save effort and time. The validation of the current study performed with the analogous study [21], by modelling and testing a smooth tube having the same specification and properties as in the comparative study. It also conducted at the same boundary condition to get adequate and fidelity validation as shown in Figure 4. The mentioned figure shows an identical matching between the current outcomes and that reported by Choi [22], with a maximum deviation of 3.6%. This maximum deviation is acceptable by taking the nature of the current study in an account as a simulation study.

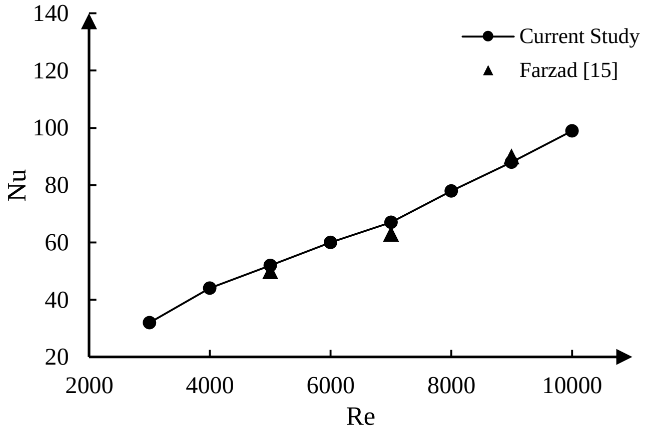


Fig. 4. The comparison of the measured and theoretical Nusselt number for water along the tube at Reynold 5000

A smooth tube was tested on set with a ZnO nanofluid to check the effect of such fluid in a smooth tube and also to separate the effect of each parameter in heat transfer enhancement. The effect of the solo parameter (nanofluid) is shown in Figure 5.

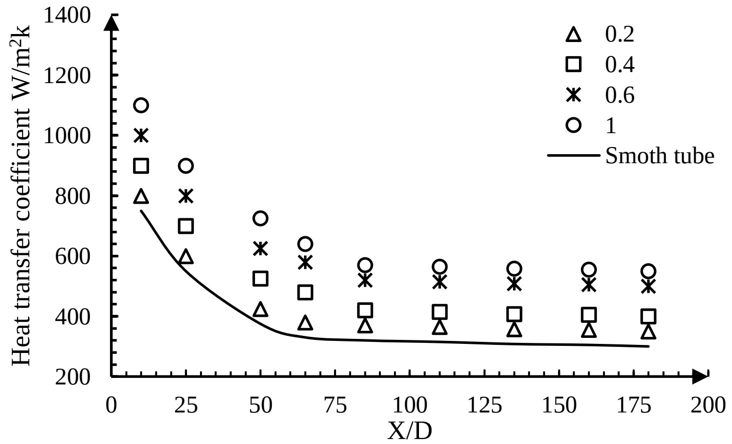


Fig. 5. Heat transfer coefficient for a plain tube at a various volume fraction of ZnO nanofluids and Re 6000

The previous figure shows that the mere effect of nanofluid in a smooth tube, which exhibits a good enhancement in heat transfer coefficient. This figure shows a steady heat transfer at downstream of the tube because the fluid is saturated with the heat at that region and cannot carry any more heat.

9. Results and Discussion

As already mentioned, the current study was planned and conducted under turbulent flow and constant heat flux condition. Two tubes were tested, one is smooth and the other was dimpled. The first one was used in the validation and both were used with ZnO nanofluid. Five volume fractions 0.2, 0.4, 0.6, 0.8 and 1 were tested to check the effect of such nanofluid in both dimpled and smooth tube. As it is commonly known, the heat transfer coefficient is directly proportioned to the surface area and flow rate. Hence, it is obviously concluded that the heat transfer coefficient would increase in a dimpled tube, and the increase would maximize in case of using both nanofluids with a dimple.

Figure 6 to 10 assure the previous expectation, where the heat transfer coefficient is clearly increased as both the Reynold number and volume fraction increase. The maximum increase was noted the tube entrance due to the maximum temperature difference at this region, while the heat transfer coefficient value is drop down the tube end. These figures indicated a maximum enhancement at Reynold number of 12000 and volume fraction 1, and poorer performance was detected at Reynold number of 6000 and volume fraction 0.2.

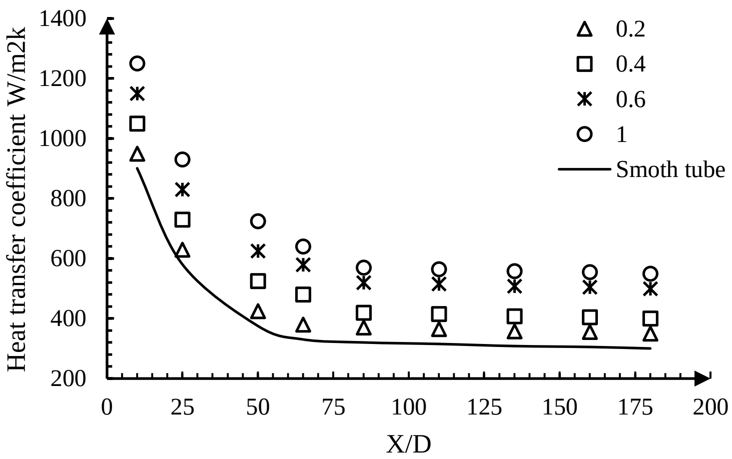


Fig. 6. Heat transfer coefficient for the dimpled tube at a various volume fraction of ZnO nanofluids and Re 7000

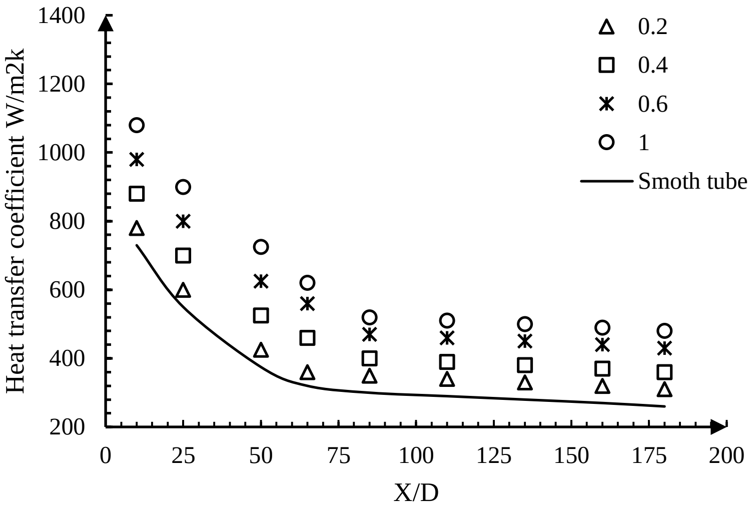


Fig. 7. Heat transfer coefficient for the dimpled tube at a various volume fraction of ZnO nanofluids and Re 8000

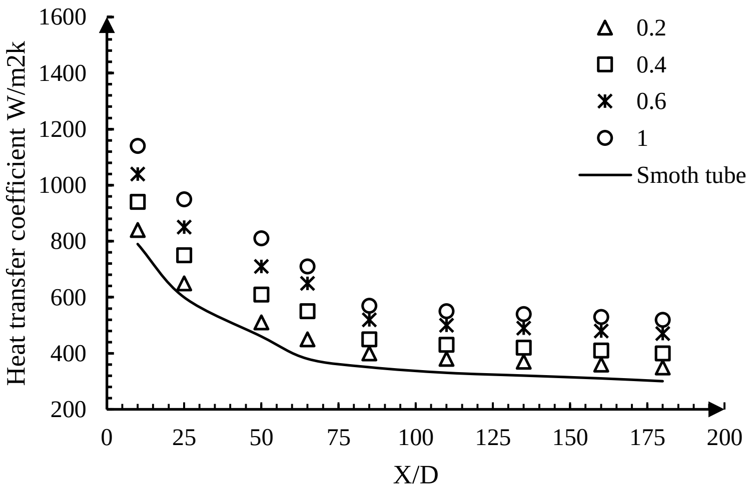


Fig. 8. Heat transfer coefficient for the dimpled tube at a various volume fraction of ZnO nanofluids and Re 8500

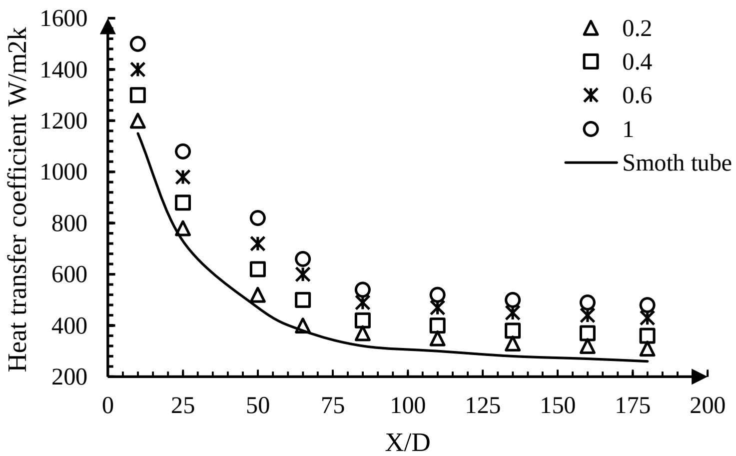


Fig. 9. Heat transfer coefficient for the dimpled tube at a various volume fraction of ZnO nanofluids and Re 9000

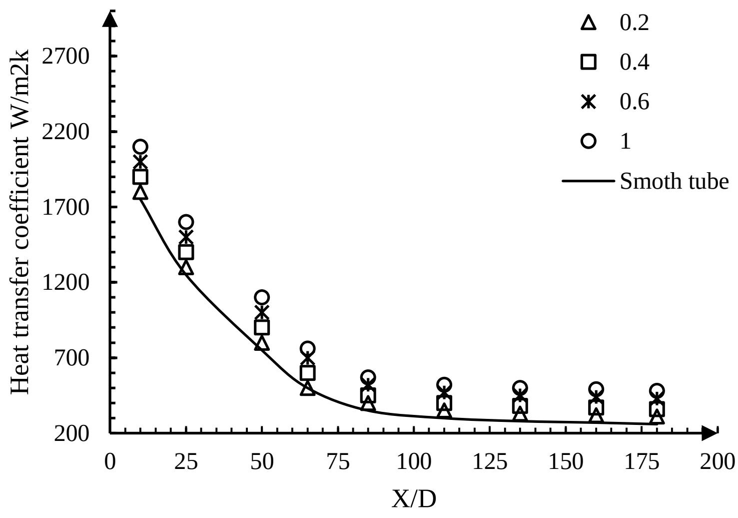


Fig. 10. Heat transfer coefficient for the dimpled tube at a various volume fraction of ZnO nanofluids and Re 12000

10. Conclusion

The current simulation was performed due to the importance of such a field in many aspects of life and industry. The interpretation of the gained outcomes coincides with the fact that the heat transfer coefficient is increased as the flow rate and input heat increase. Moreover, the heat transfer coefficient increases as the dimple increase in number and size with a reasonable increase in pressure. It was inferred that the maximum Nusselt number attained when using the combination of both surface modification (dimple) along with nanofluid. Eventually, it was clearly concluded that the using of surface modification like a dimple and nanofluid prompt the increase of the rate of thermal transport forward due to the increase in the contact surface area between the nanoparticle and base fluid, and also between the nanofluid and the tube wall.

It was concluded a maximum enhancement achieved is 2297 at Reynold number of 12000 and volume fraction 1. While the poorer performance was 1013 at Reynold number of 6000 and volume fraction 0.2.

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