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Research article

Computational Analysis of Dimples Surface Tube on Heat Transfer Forced Convection Using Turbulence Model of Low Reynolds Number with Different Cases

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Abstract: The researchers aimed to improve the heat transfer process using various industrial engineering techniques to highlight the importance of maximum heat transfer and improvement. In the current numerical study, the Ansys Fluent R23 program was used to enhance the heat transfer rate with a laminar flow in the Reynolds number range of (500-1500) steady state single-phase inside a dimpled tube with three different states heated with a uniform constant heat flux (8000 W/m^2) along the flow axis. The effect of the dimples' arrangement on the tube's surface was studied to compare the thermal performance against the empty tube. Comparing the three cases (1, 2, and 3) to the standard tube without a dimple, the heat transfer rates increased by (86.99, 87.22, and 87.47%), respectively. Compared to tubes with two or three dimples, the heat exchanger tube with four dimples on its cross section and linearly towards the flow axis performs much better in terms of thermal-hydraulic performance.

Keywords: Laminar flow; Horizontal tube; Thermal performance factor; Dimples; Friction factor

1. Introduction

Around the world, heat exchangers are widely used in the most important engineering industrial applications. The most appropriate option is necessary to make these devices more convenient than others, reduce their size and cost, and improve their hydrothermal performance and efficiency. The methods of enhancing the heat transfer rate are generally divided into two main parts: the passive and active methods. The current numerical study is focused on using the first method, which has a low cost because it does not require external devices compared to the second method [1]. To achieve the best enhancement of heat transfer by the passive method and using different configurations of the tube surface, such as ribs, dimples, and corrugations, increasing the surface roughness is one of those methods. All the mentioned configurations cause an increase in turbulence, mixing of the flow, and development of the boundary layer. Therefore, the heat transfer rate improves significantly [2, 3]. The pressure loss and heat transfer rate of tubes with a rough surface is significantly increased compared to smooth ones [4-7]. For example, many researchers in their experimental and numerical studies in the recent past have managed to move away from increasing the surface area of heat transfer and have turned to using dimples on the surface of tubes, causing mixing and disturbance of the boundary layer, thereby improving the thermal efficiency of the heat exchanger Juin Chen et al. and Yu Wang et al. [8 and 9] presented a study of the effect of inserting dimple protrusions to the tube surface to improve thermal performance, the effects of different ribs have been studied by Dennis Leroy Gee and R. L. Webb, J. C. Han and J. S Park [10 and 11] while A. García et al. [12 and 13] presented the realization of the effect of using spiral wires and dimples with axial corrugations to enhance heat transfer and the development of the thermal-hydraulic performance of tubes using CFD in numerical simulation and experimental approach. Ming Li et al. [14] performed numerical simulation using the computational fluid dynamics method to improve the thermal-hydraulic performance of a three-dimensional tube with stable single-phase turbulent flow incompressible. The technique known as the Semi-Implicit Method for Pressure Equations Consistent (SIMPLEC) solves the pressure-velocity coupling. The numerical results showed that the dimpled tube surface in linear order gave a thermal performance and improved high heat transfer compared to the graded configuration. Various geometry parameters such as dimple shape, depth, pitch, and diameter have been shown to affect enhancing heat transfer. M. Z. U. Khan et al. [15] presented a numerical investigation to solve the problem of enhancing heat transfer by the effect of a dimple protrusion in a microchannel of rectangular cross-section using water as the working fluid with a constant heat flux shed on the channel wall as a boundary condition. The coefficient of thermal performance is investigated by calculating the number of Nusselt and the friction factor at a laminar flow range (100-900) of the working fluid inside the channel. Ganesh V. Wafelkar and L. V. Kamble [16] presented an experimental study to overcome the length of the double tube by designing a triple tube for a heat exchanger because it provides a larger transfer heat area than the double tube. The purpose of the study is to evaluate the performance of triple-tube heat exchangers using dimple tubing. Different flow rates of hot and cold fluid are investigated experimentally. The relationship between several performance metrics, including heat exchanger efficacy, friction factor, Nusselt number, and Prandtl number, is also discussed. Farah Nazifa Nourin and Ryoichi S. Amano [17] conducted an experimental and numerical study using six different states of dimples based on the underlying smooth surface to investigate the characteristics of the heat transfer rate of a gas turbine blade inserted with a channel along the turbulent flow axis in the Reynolds number range (6000-50000). The results of the maximum Reynolds number's heat transfer enhancement and friction factor indicated that the leaf-dimpled surface is the optimal cooling channel. On the other hand, half-spherical dimes with a depth-to-diameter ratio of 0.25 had the best thermal performance at the lowest Reynolds number. Kanit Aroonrat and Somchai Wongwises [18 and 19] experimentally studied the condensation process of a two-phase tube flow with a dimpled surface for the development of Thermo-hydraulic performance. The results indicated the heat transfer rate represented by the number of Nusselt improved by (84%) compared to the empty tube. Sarmad A. Ali et al. [20] studied numerically using the finite-volume method of the program (Ansys Fluent) to improve the properties of the working fluid and the forced convective heat transfer of a turbulent flow at Reynolds ranges (3500-7000) single-phase stable three-dimensional. The improvement process involved using three different roller configurations (circular, square, and triangle) with equal hydraulic diameter to achieve the effect on the heat transfer rate represented by the number of Nusselt and the friction factor. The results indicated dimples enhance heat transfer by a ratio of (13, 18, and 21 %) to (square, circular, and triangle) respectively. Sarmad A. Ali [21] presented a numerical study of the optimization of heat transfer by forced convection of fluid flow within a two-dimensional horizontal channel using three different configurations of ribs (quarter circle, square, and triangle) to enhance heat transfer. Several parameters have been studied numerically, including Reynolds numbers, friction factor, pressure drop, and the Nusselt number. The results showed heat transfer improved by (72, 70, and 68%) for configurations (quarter-Circle, Square, and Triangle), respectively. Moreover, as the Reynolds number increases, the number of Nusselt gradually increases, and the friction factor decreases.

The current numerical analysis focuses on increasing the heat transfer rate by forced convection and improving the flow properties of a horizontal circular tube with a dimpled surface in three different cases compared to a smooth tube. A constant heat flux (8000 W/m^2) subjected on the outer wall of the tube along the flow axis in the Reynolds number range (500-1500) is a steady state incompressible single-phase flow. The effect of the change in the number of dimples on the number of Nusselt, the friction factor, the velocity distribution, temperature, and pressure of the fluid was studied numerically. Many previous studies have dealt with improving heat transfer by turbulent flow, so the current study is concerned with laminar flow because of its importance as a type of fluid flow in various pipes and heat exchangers.

2. Physical Numerical Model

Figure (1) shows the computational domain of a three-dimensional horizontal tube with a length and outer diameter (160 and 30 mm), respectively. Dimples with a diameter of (4 mm) are inserted in three different cases: the first, second, and third cross-section of the tube contains dimples number (2,3, and 4) respectively, as shown in Figure (2). The dimples were linearly distributed over the surface of the tube with an equal spacing (10 mm). In addition, the outer wall is also provided with a uniform constant heat flux of (8000 W.m-2) along the fluid flow axis.

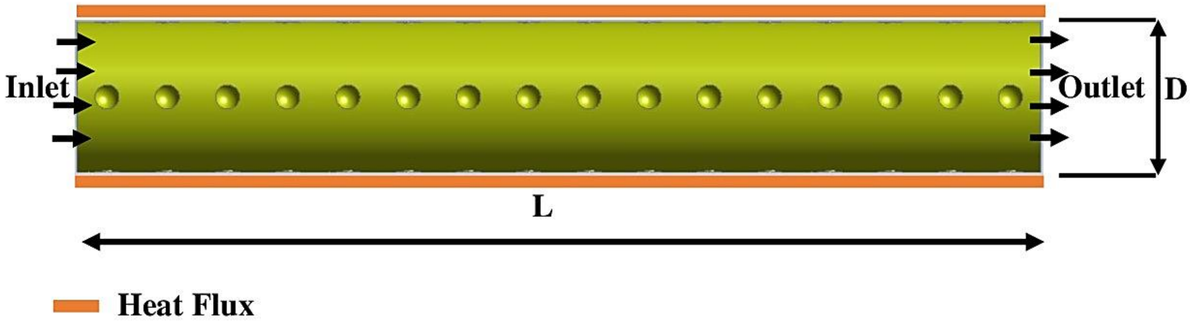

Figure 1. Schematic diagram with dimensions of dimple tube

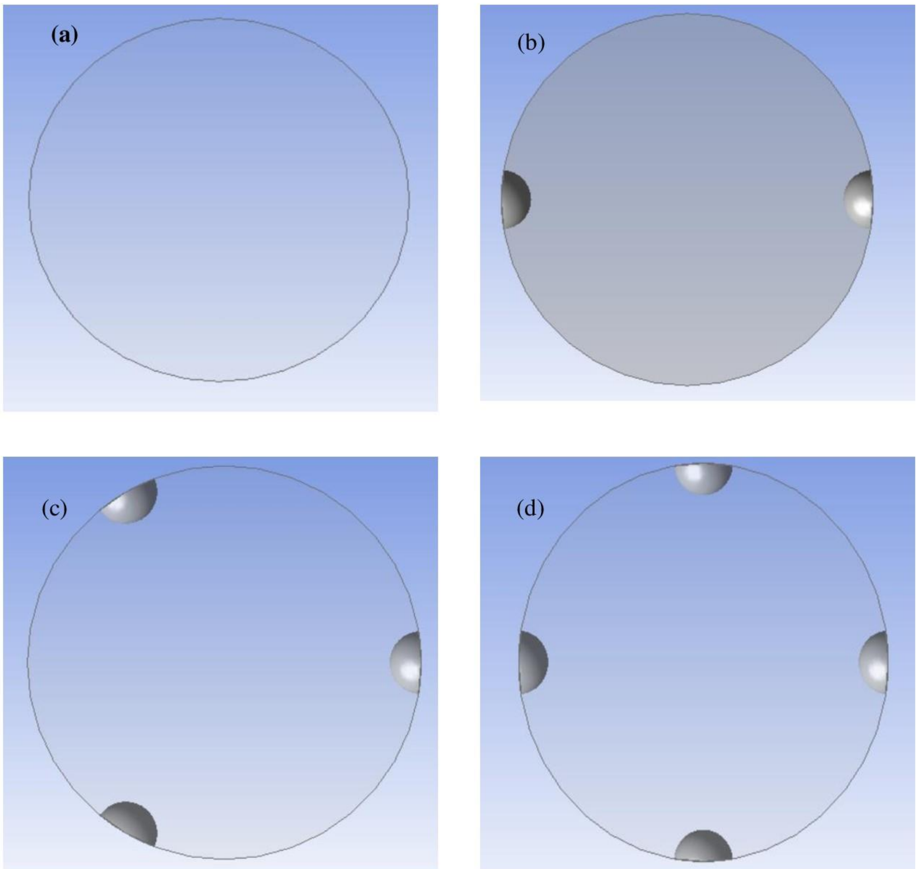

Figure 2. Three different cases of dimples included around the circumference of the tube cross-section: (a) plain, (b) two dimples, (c) three dimples, and (d) four dimples

2.1 Equations of Governing Fluid

The effects of heat transfer by radiation, natural convection, and viscous dissipation these factors are eliminated. The fluid flow in the present study is subject to the steady state with an incompressible three-dimensional laminar flow for all governing equations, including continuity, momentum, and energy. According to the Cartesian system, the governing equations can be written as follows [22]:

$$\frac{\partial u_j}{\partial x_j} = 0 \tag{1}$$

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}[(\mu + \mu_t)(\frac{\partial u_i}{\partial x_i} + \frac{\partial u_j}{\partial x_i})] \tag{2}$$

$$\frac{\partial u_i T}{\partial x_i} = \frac{\partial}{\partial x_j}((\frac{\mu}{\mathrm{Pr}} + \frac{\mu_t}{\mathrm{P}_t\mathrm{r}})\frac{\partial T}{\partial j})) \tag{3}$$

Using two transport equations, this two-equation model broadly explains turbulence. Equations for the turbulent kinetic energy dissipated (ε) and turbulent kinetic energy (κ) in turbulence are provided as [23]

$$\frac{\partial}{\partial x_j}(\rho k u_j) = \frac{\partial}{\partial x_j}[(\mu + \frac{\mu_t}{\sigma_k})\frac{\partial k}{\partial x_j}] + \Gamma - \rho\varepsilon \tag{4}$$

$$\frac{\partial}{\partial x_j}(\rho \varepsilon u_j) = \frac{\partial}{\partial x_j}[(\mu + \frac{\mu_t}{\sigma_\varepsilon})\frac{\partial \varepsilon}{\partial x_j}] + C_{1\varepsilon}\Gamma\varepsilon - C_{2\varepsilon}\frac{\varepsilon^2}{k + \sqrt{\nu\varepsilon}} \tag{5}$$

$$\Gamma = -\overline{uu}\frac{\partial u_i}{\partial x_j} = \frac{\mu_t}{\rho}(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i})\frac{\partial u_i}{\partial x_j} \tag{6}$$

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{7}$$

The empirical constants in the equations of the turbulent model are:

$$C_\mu = 0.09; C_{1\varepsilon} = 1.47; C_{1\varepsilon} = 1.92; \sigma_k = 1.0; \sigma_\varepsilon = 1.3 and\ \mathrm{Pr}_t = 0.85 \tag{8}$$

2.2 Boundary Conditions of Computational Model

For all boundaries of the computational domain, boundary conditions are provided because the governing equations of the flux are located in spatial coordinates. Various boundary conditions can be described as follows:

- Characteristics of the fluid flow at the tube's inlet at a temperature of (300K) and the axial uniform fluid velocity with laminar flow at the Reynolds number range (500-1500). Based on that, turbulent dissipation (ε) and turbulent kinetic energy (k) are calculated.
- The outer wall of the tube is considered to be in a non-slip state, and it is exposed to a constant uniform heat flux.
- Constant temperature conditions are assumed at the extended areas and the tube inlet.
- As a result of the prevailing pressure in the fluid exit area, the pressure outlet boundary condition is applied. Also, all variables in terms of flow have minimal assumed gradients.

2.3 Definition of Parameter

All parameters used for numerical verification of the current study are described for analysis and presentation of results. Based on the laminar flow to analyze the characteristics of the fluid flow and the pressure drop in the tube as follows, the Reynolds number equation and the friction factor can be expressed respectively [24]:

$$\mathrm{Re} = \frac{\rho u_{in} D_h}{\mu} \tag{9}$$

$$f = \frac{2\Delta p D_h}{L\rho u_{in}^2} \tag{10}$$

The coefficient of heat transfer by forced convection and the average Nusselt number by the following equations are calculated [25 and 26]:

$$h_x = \frac{q''}{(T_w - T_f)} \tag{11}$$

$$Nu_m = \frac{D_h}{\lambda}\int_0^L h_x dx \tag{12}$$

3. Solution of Numerical Model

The governing equations with boundary conditions were solved using a commercial program (Ansys Fluent R23) using the finite volume method to solve laminar flow and heat transfer. The pressure-based model was combined with the simple algorithm for the pressure velocity fields. A second-order downwind scheme was used for the equations of momentum, turbulent kinetic energy, and the rate of turbulence dissipation. An achievable (k-ε) model was adopted with improved wall treatment to provide improved predictions of flows close to the wall. The numerical residuals were reduced to (10^{-6}), and the estimated algebraic finite volume equations were repeatedly solved, as shown in Figure (3). After setting the iteration value to 100, the computations begin. Iterations keep on until convergence is achieved. Figure (4) highlights the unstructured hybrid grid of the three-dimensional tube using the tetrahedrons method.

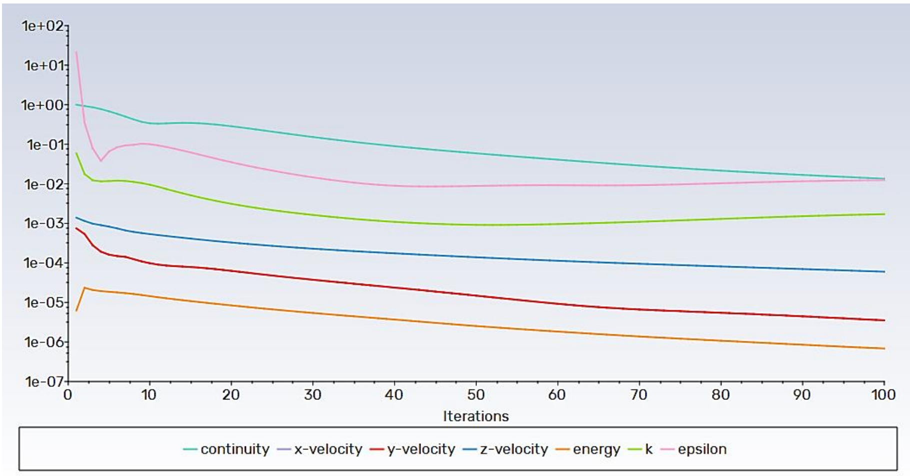

Figure 3. Scaled residuals of the computational model

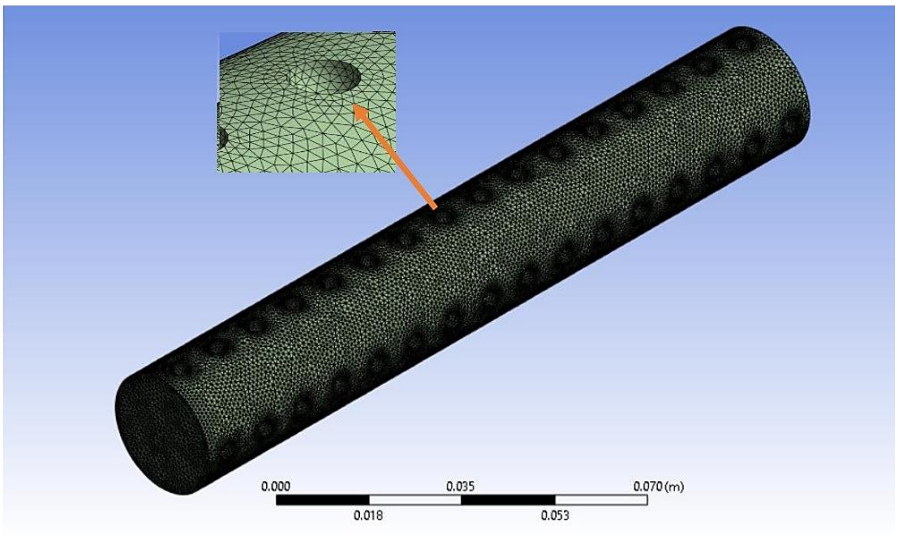

Figure 4. Grid mesh of horizontal 3D tube with a surface dimple

4. Analysis of Grid Independence

Controlling the size and number of elements and cells in numerical calculations is essential to improve numerical results. This use is called grid independence, so the grid is divided into small parts to achieve accuracy in the mathematical solution. The main principle of the ordinary CFD technique is that at the beginning of the solution, a rough grid is formed, gradually improving so that the detection of changes in values becomes small from the previously specified error. This is problematic in two ways. First, there might be issues when using other CFD software to get a single coarse mesh. Second, refining a mesh by a factor of two or more may require extra time. This is inappropriate behavior for software meant to be an engineering tool with limited production capabilities. Furthermore, the other problems have contributed significantly to the idea that CFD is an incredibly challenging, expensive, and time-consuming approach. The Nusselt number was finally recorded and sorted in each case to establish grid independence ,as shown in Figure (5).

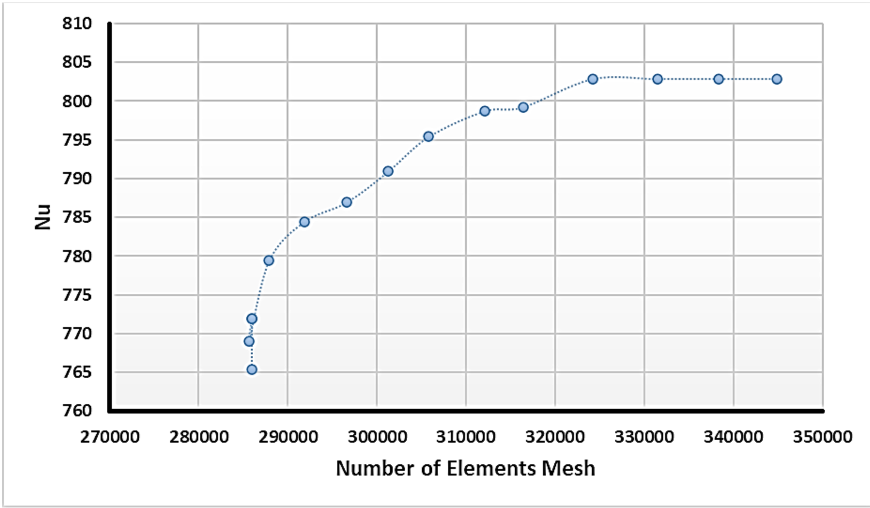

Figure 5. Analysis of grid independence of the computational model

5. Results and Discussions

For variable cases of the surface of a rough, dimpled tube, the change of the Nusselt number against the Reynolds number is shown in Figure (6). As the fluid velocity increases, the Nusselt number gradually increases. Also, by including the dimple on the surface, the Nusselt number increases compared to a tube with a smooth surface. The heat transfer rate of the three cases (1, 2, 3) improved by (86.99, 87.22, and 87.47 %) compared to the ordinary tube without the dimple. The thermal-hydraulic performance of the heat exchanger tube is greatly improved by including four dimples on the tube's cross-section and linearly towards the flow axis compared to the tube with two and three dimples.

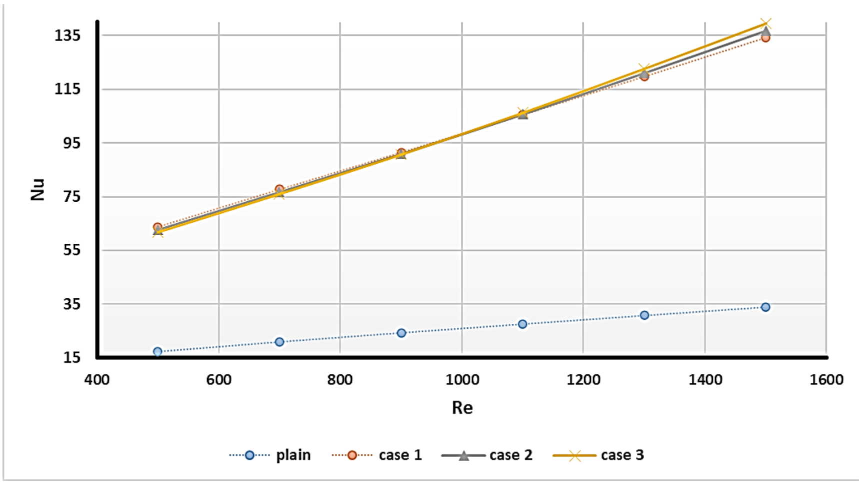

Figure 6. Influence of changing dimples numbering on Nusselt number with Reynolds number

Figure (7) shows the change of the friction factor with the ranges of the Reynolds number. The friction factor can be observed to decrease sharply at first, gradually decreasing by increasing the number of Reynolds using the dimples. The increase in the number of dimples caused an increase in the friction factor, where the third case, including four dimples, gave the highest value of the friction factor compared to the standard tube and the other number of simple cases.

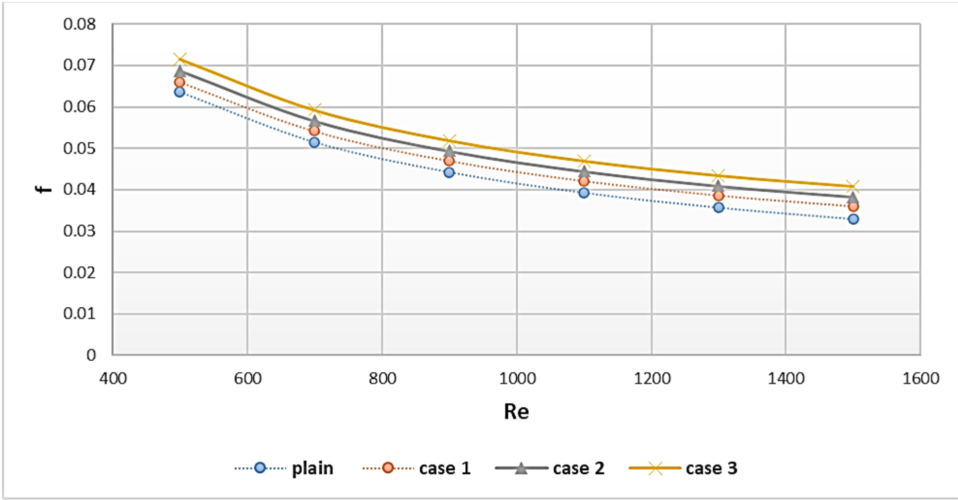

Figure 7. Influence of changing dimples numbering on Nusselt number with Reynolds number

Figure (8) shows the temperature distribution of the water flow with a three-state dimpled tube with an empty smooth tube at a Reynolds number range (1500) and a uniform constant heat flow (8000 W/m²). It can be observed that the average temperature of the fluid gradually increases towards the axis of the fluid flow inside the horizontal tube. Also, the dimples cause a change and deformation of the temperature distribution field. The average temperature is greater in the dimpled tube with the more significant number. It is also evident that the temperature within the tube rises in direct proportion to the number of dimples at the tube wall.

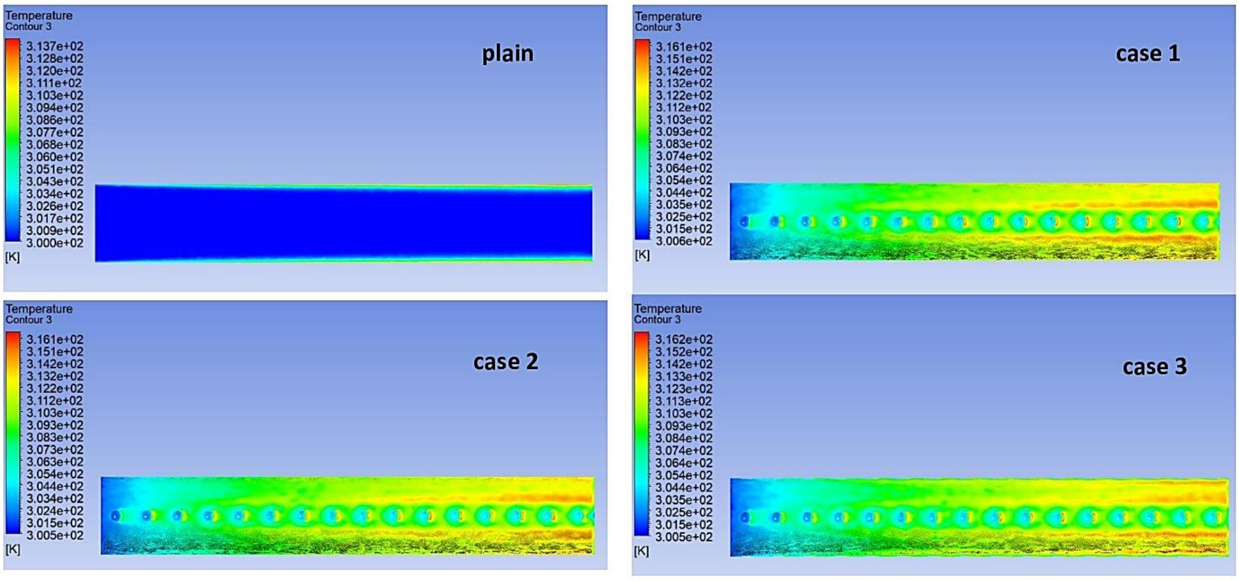

Figure 8. Compassion of Temperature distribution between plain tubes with different case dimples

Figure (9) shows the fluctuations of the fluid velocity distribution with and without dimples on the horizontal tube surface at Reynolds number (1500) and a uniform heat flow along the tube wall. The average velocity of the fluid can be observed to increase using the three states of the number of dimples. The average velocity of water increases by (5.373, 7, and 4.35) for cases (1,2, and 3) respectively.

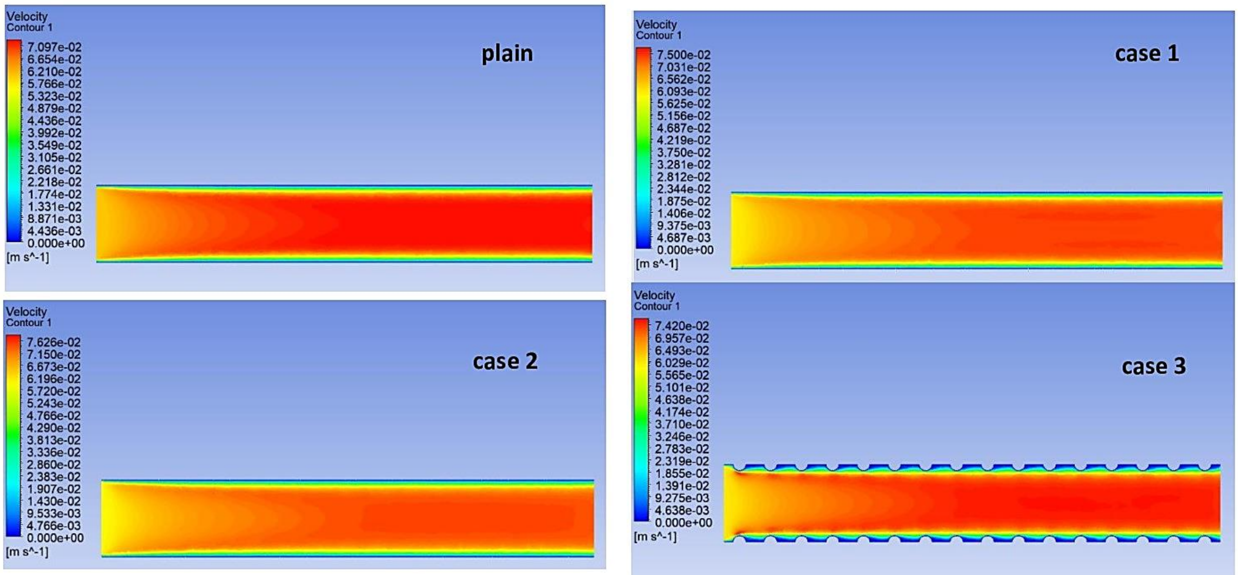

Figure 9. Compassion of velocity distribution between plain tubes with different case dimples

The axial and radial pressure distributions for various numbers of dimpled tubes with a Reynolds number of (1500) and a heat flux of (8000 W/m^2) are shown in Figure (10). The pressure variations were less apparent as the tube length increased. The pressure value is more significant in the input tube zone and lower in the output tube region. This figure allows for the observation of several things. The distance between the near-wall tube and the center is where the more excellent pressure value is found. Furthermore, the pressure is reduced in the direction of the tube wall due to this low-velocity value. It should also be mentioned that the pressure within the tube increases with the number of dimples at the tube wall. The tube's higher resistance is what caused this to happen.

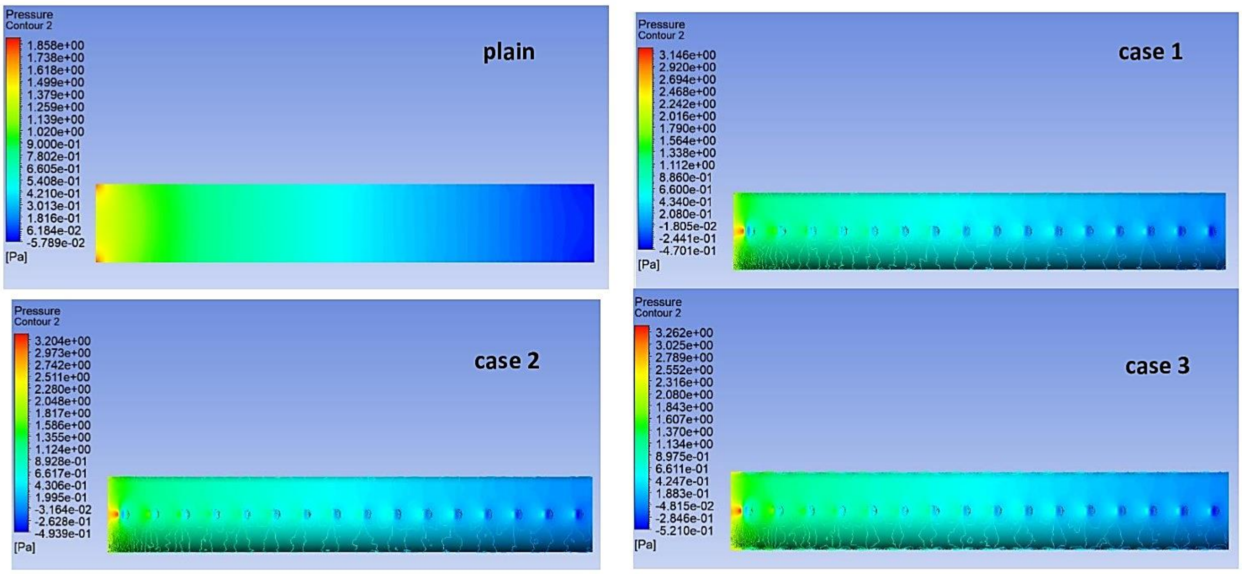

Figure 10. Compassion of pressure distribution between plain tubes with different case dimples

5. Conclusions

In the present study, the effect of increasing the number of dimples on the process of improving the rate of heat transfer by forced convection of laminar flow of water in the Reynolds number range (500-1500) inside a three-dimensional horizontal tube under a constant heat flux along the wall was investigated numerically. In the numerical results, it was mainly concluded that tubes with a dimpled surface are considered an appropriate method to enhance heat transfer, thereby increasing the efficiency and thermal performance of the heat exchanger, making it more compact. Compared to the tube without a dimple, the number of Nusselt increased gradually with the dimple and increasing Reynolds number ranges. At the same time, the friction factor decreased with the increase in the Reynolds number and increased with the inclusion of dimples on the surface of the tube. Compared to a normal tube without a dimple, the heat transfer rates of instances 1, 2, and 3 improved by 86.99, 87.22, and 87.47 percent, respectively. Four dimples on the cross-section of the tube and linearly towards the flow axis provide a significant improvement in the heat exchanger tube thermal-hydraulic performance when compared to tubes with two or three dimples.

Nomenclature

D	Tube diameter (m)
f	Dimensionless friction factor
k	Turbulence kinetic energy (m^2/s^2)
L	Tube length (m)
Nu	Dimensionless Nusselt number
p	The pressure of water (Pa)
Pr	Dimensionless Prandtl number
q	Uniform heat flux ($W.m^{-2}$)
Re	Dimensionless Reynolds number
T	Temperature of water (K)
u	Water velocity ($m.s^{-2}$)
μ	Water dynamics viscosity ($N.s.m^{-2}$)
μ_t	Turbulence of water dynamics viscosity ($N.s.m^{-2}$)
ε	Dissipation of heat energy (m^2/s^3)
λ	Water thermal conductivity ($W.m^{-1}. K^{-1}$)
ϱ	Water density ($kg.m^{-3}$)

Subscript

f	fluid
h	Hydraulic
in	Inlet
o	Outlet
t	Turbulence
w	wall

Abbreviations

PEC	Performance Evaluation Criterion
CFD	Computational Fluid Dynamics

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