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Numerical Study of Thermal-Hydraulic Performance of Forced Convection Heat Transfer in Dimple Surface Pipe with Different Shapes using Commercial CFD Code

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ABSTRACT

Various technologies have been used, including a regular tube with the inclusion of dimples, which are widely used to enhance the thermo-hydraulic performance of engineering applications. A numerical analysis involving this work of the thermo-hydraulic performance of water flow in a scarred pipe. By numerical solution using the Ansys Fluent 2023 R2 program to model the flow through the pipe, the effect of different dimple shapes (square, circular, and triangle) has been studied. The finite volume method identified a new enhancement model that makes use of composite-form surface techniques. To assess the effect of scarring on the field of velocity and turbulent flow, the continuity, momentum, and heat energy equation represented by the three-dimensional governing differential equations was studied using the k-epsilon flow model for the Reynolds number range of (3500-7000). The numerical results indicated the dimple surface of the pipe significantly improves the heat transfer rate and also increases the friction factor compared to the smooth pipe without dimple. Percentage (12.808, 17.987, and 20.978%) of the increase in heat transfer rate enhancement (Nusselt number) for the three dimple shapes of the pipe (square, circular, and Triangle) respectively, moreover dimpled pipe has a higher coefficient of friction compared to the normal pipe.

1. Introduction

The production of devices with high efficiency and efficiency of heat transfer as a heat exchanger is a difficult task, so methods and ways must be found to increase the rate of heat transfer. Some find new forms with low cost and lower material consumption to study heat transfer manufacturers and raise the heat-related factor, for example, heat exchangers and heat transfer rates. In our study, we mainly focus on these insights. There are three main ways to enhance heat transfer that should be considered: the active method, the passive method, and the combination method. To increase

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the heat transfer rate, some engineering changes were made by placing a dimple on the surface of the pipe. To improve the process of heat transfer by the active method some engineering changes are being made. While some external sources are used in the second (passive) method of the process of increasing heat transfer. The third method uses engineering changes with the addition of external sources to enhance heat transfer in industrial and engineering applications [1-5]. Many studies by researchers include numerical or experimental, or both, using dimpled surfaces in various shapes, including oval, triangle, square, and circular, or using various angles to improve the thermal performance of the pipe or channel compared to the absence of dimples [6-10]. Zhou and Acharya [7] presented an experimental and computational study using the naphthalene sublimation method to conduct mass and heat transfer measurements in a square inner passage of one wall containing one dimple. Four different types of dimples were studied, including triangular, square, circular, and teardrop at the Reynolds number (21000) at the same conditions for flow calculations were performed. The study obtained satisfactory agreement when comparing the computational results against the experimental data. Numerical and experimental results indicated the teardrop shape of the dimples gave the best improvement in heat transfer compared to the other forms studied. Harish *et al.,* [11] conducted a numerical analysis of the flow field to enhance the heat transfer process of a three-dimensional circular tube by stabilizing the heat flow along the wall of the tube embedded with/without dimples on the twisted strip. The effect of the use of twisted tape with dimples on hydraulic performance was studied numerically with laminar flow for the range of the number of Reynolds (800-2000). Nusselt number variations of 2.5, 5.75, and 9.5% are obtained from the simulation results of plain, plain twisted, and dimple twisted tapes against their Reynolds numbers. Also, the value of the friction factor using a twisted strip with dimples is (6 to 13) times that of an ordinary pipe. Kiselev *et al.,* [12] reviewed an experimental study to determine the drag and heat transfer coefficients of spherical and oval smooth and dimpled surfaces in a cylinder with a diameter of (8 mm) placed in an opening channel with a height of (30 mm). The direct order of spherical dimples and the overlapping order of spherical and ellipsoidal dimples were considered. The Reynolds number range ranges (10.9 -53300) to determine the drag coefficients based on the cylinder diameter. Depending on the specific surface and the distance between the cylinder and the wall, the Reynolds analogous factor for the aforementioned models varied from 0.45 to 7.73. Li *et al.,* [13] provided numerical analysis to obtain optimal Thermo-hydraulic performance to improve the engineering design of the pipe. Using experimental data, the simulation is validated with a non-compressive single-phase turbulent flow of the k-epsilon method. The results of the study showed that the diameter of the dimples has little effect on thermal performance compared to other geometric parameters such as shape and depth. Xie *et al.,* [14] and Vignesh *et al.,* [15] numerically conducted a study of the presence of dimples on the surface of pipes to examine their effect on thermal performance. The results indicated smooth surfaces do not improve heat transfer compared to the use of dimples as a means of improvement. Xie *et al.,* [16] investigated the impact of the spherical and plain dimples on the hydrothermal reaction of the dimpled tube. Using the turbulence model (k-epsilon) and the Reynolds number range (5000-30000) in numerical simulations the effect of dimple depth, pitch ratio, and axis on thermal performance was discussed. Ali and Shehab [17] presented a numerical and experimental analysis to examine the thermal properties of water in a heat exchanger containing two different types of dimples, the first with a padded distribution and the second with an overlapping distribution and two angles (60 and 90 degrees) of variable diameters (4 and 6 mm). The research findings demonstrate that, in comparison to a plain pipe, the heat transfer rate as measured by the Nusselt number improved by 7.55 to 11.2 times in the case of a staggered dimpled inner pipe. Gürdal *et al.,* [18] and Zhang *et al.,* [19] according to previous numerical or experimental investigations, many researchers have shown that when compared with a smooth pipe, a relatively good thermo-hydraulic response is shown by the type of dimpled pipe. Several effects have been studied for geometric parameters including shape, distance, diameter, and angle. Mohammed *et al.,* [20] conducted a numerical study to investigate the properties of single-phase turbulent flow of the Reynolds number range (6000-14000) and the properties of forced heat transfer in a circular horizontal tube. Dimples with a circular cross-section are inserted on the surface of the pipe to improve the heat transfer rate compared to a smooth pipe. To clarify the basic process of the effect of dimples on thermal performance, a three-dimensional simulation was used. The numerical results indicated the temperature distribution changes by the presence of dimples that promote heat transfer by forced convection effectively within the central area of the dimple section. In heat exchangers, there are other ways to enhance the heat transfer process, including the inclusion of nanomaterials with the base fluid to improve the number of Nusselt, temperature distribution, fluid velocity, as well as pressure and friction factor reduction [21,22]. Ahmad *et al.,* [23] presented a numerical study to analyze the effect of spherical dimples on the properties of heat transfer and turbulent fluid flow for the Reynolds number range (10000-30000) in a double tube in two cases, the first is the addition of nanoparticles with pure water (Al_2O_3, TiO_2, and CuO) and the second is hybrid nanomaterials (Al_2O_3+CuO and Al_2O3 + TiO_2). many researchers have improved heat transfer using effective methods, including nanomaterials with basic fluids or dimples on the surface of pipes, thereby developing thermal hydraulic properties. The results of numerical simulations showed that the heat loss increases when the fluid flow inside the pipes is obstructed and the thermophysical properties improve when mixing different nanoparticles with the basic working fluid [24-28].

A small number of studies focus on reducing pressure and increasing the heat transfer rate at the same time, although previous research literature has shown that the introduction of dimples or protrusions in pipes may enhance the heat transfer rate. The current numerical study focuses on the use of different types of dimples in a horizontal three-dimensional pipe subjected to heat flux on the outer surface with the single-phase turbulent flow with Reynolds number range of (3500-7000) to improve the Nusselt number and pressure drop, where the flow results were recorded in the dimpled and normal pipe such as velocity, pressure, friction factor, and temperature.

2. Methodology of Numerical Solution

2.1 Description of Computational Model

A numerical investigation to improve the hydrothermal properties of a three-dimensional horizontal circular pipe using different dimple configurations is the focus of the current study. Improvement of hydrothermal performance and investigation of fluid flow inside the dimpled pipe is the main objective of this study. Figure 1 shows the smooth three-dimensional pipe with a wall made of aluminum material with a diameter of (0.03 m) and a length of (0.15 m) with a uniform heat flux of (40kW/m^2) placed on the outer surface of the pipe. Figure 2 shows the dimpled tube of various types (square, circular, and triangle) with a distance between the dimples of (0.01 m) and the diameter of the dimple of (0.04 m). The initial temperature of the working fluid is constant for all numerical cases at (300 K).

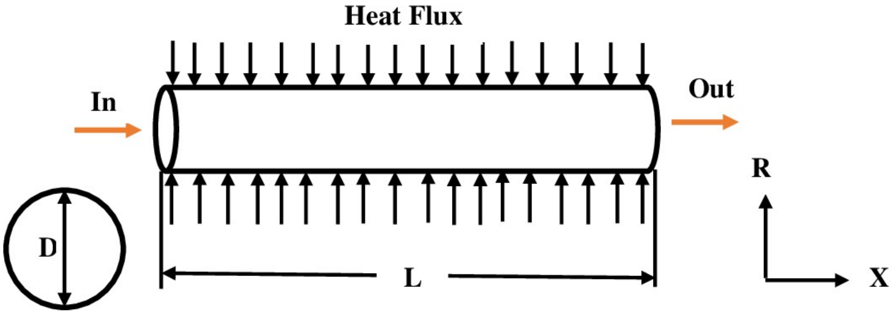

Fig. 1. Geometrical problem of numerical modeling

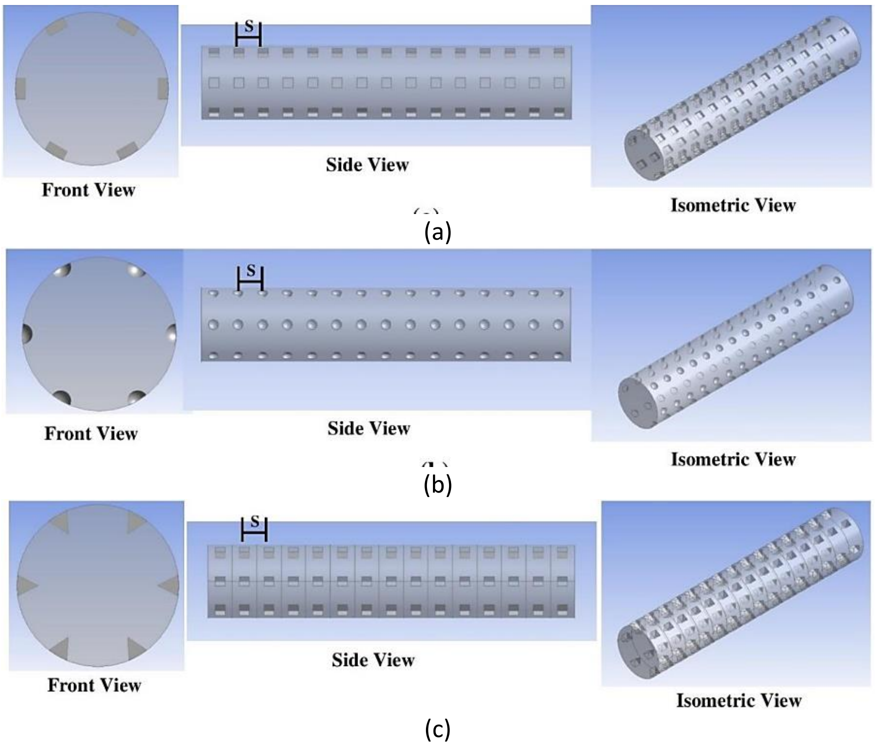

Fig. 2. Dimple pipe with different configurations (a) square, (b) circular, and (c) triangle

2.2 Thermophysical Properties

With an initial density of 998 kg/m^3, a specific heat of 4179 J/kg·°C, a dynamic viscosity of 0.000855 kg/m· s, a thermal conductivity of 0.613 W/m·°C, and a thermal expansion rate of 276.1x10^{-6} K^{-1}, water was the working fluid required at an initial temperature of (300 K). Aluminum had the following properties when it came to construction: it was 36.0 W/m·°C thermally conductible, had a specific heat of 765 J/kg·°C, and a density of 3970 kg/m^3. The operating pressure and gravity acceleration applied to each model in the investigation were 101.3 kPa and 9.81 m/s^2, respectively [29].

2.3 Governing Equations

Using the Reynolds Average Navier-Stokes the flow governing equations of continuity, momentum, and energy are solved numerically. Under many assumptions and simplifications as follows [30]:

i. The fluid flow is assumed to be incompressible, turbulent, and steady-state.

ii. The thermophysical properties of the working fluid and the pipe-making material were assumed to be independent of temperature.
iii. Ignoring and not mentioning the contribution of heat transfer by the radiation method.
iv. Apply a constant heat flux with the imposition of a smooth outer tube surface.
v. In this numerical analysis, the effects of vibration that occur are not considered.

The partial differential equations represented by the governing equations of fluid flow and convection based on the previous assumptions can be expressed as follows:

Continuity (mass) equation:

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

Momentum equation:

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

Energy equation:

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

To simulate turbulence for this study, a turbulence model (k-epsilon) with enhanced wall treatment was used. Below are the typical transport equations for both k and epsilon based on the current numerical model [31]:

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

$$\frac{\partial}{\partial x_j}(\rho \varepsilon u_j) = \frac{\partial}{\partial x_j}[(\mu + \frac{\mu_t}{\sigma_\varepsilon})\frac{\partial \varepsilon}{\partial x_j}] + C_3\Gamma\varepsilon - C_4\frac{\varepsilon^2}{k + \sqrt{\nu\varepsilon}} \tag{5}$$

where Γ is the kinetic energy (k) generation rate of turbulence.

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

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

The above-mentioned realizable κ −ε equations have the following coefficients [32]:

$$C_3 = \max[0.43, \frac{\mu_t}{\mu_t + 5}], C_4 = 1.0, \sigma_k = 1.0, \sigma_\varepsilon = 1.2 \tag{8}$$

2.4 Boundary Condition and Obtaining Data

Regarding homogeneous temperature and velocity profiles at the three-dimensional horizontal pipe entry, the following hypotheses are made:

$$v = 0, w = 0, u = u_{in}, T = T_{in} \tag{9}$$

The velocity of the fluid is highlighted to the left of the pipe and the pressure is from the outside, while the outer wall of the pipe is highlighted by a constant, uniform heat flux along the flow axis of the fluid.

The turbulent kinetic energy entrance profiles and the turbulent dissipation rate are calculated from:

$$k_{in} = 0.03 u_{in}^2 \tag{10}$$

$$\varepsilon_{in} = C_\mu \frac{k_{in}^{2/3}}{0.03R} \tag{11}$$

Upon leaving, the following boundary conditions are applicable:

$$\frac{\partial u}{\partial x} = \frac{\partial v}{\partial y} = \frac{\partial w}{\partial z} = 0 \quad , \quad \frac{\partial T}{\partial x} = 0 \quad , \quad \frac{\partial p}{\partial x} = 0 \tag{12}$$

When a model (k-ε) exits, its boundary condition looks like this:

$$\frac{\partial k}{\partial x} = \frac{\partial \varepsilon}{\partial y} = 0 \tag{13}$$

Due to the absence of slip conditions at the walls and the zero velocity components,

$$u = v = w = 0 \tag{14}$$

The following is the turbulence model (k) boundary condition: $k = 0$, $\frac{\partial \varepsilon}{\partial r} = 0$ (15)

With a steady heat flow, the following boundary conditions will be met:

$$\frac{\partial T}{\partial r} = \frac{q''}{\lambda} \tag{16}$$

The relationship between viscous and inertial forces is represented by the Reynolds number. This is the expression for the Reynolds number.

$$\mathrm{Re} = \frac{\rho u_{in} D_i}{\mu} \tag{17}$$

The coefficient of heat transmission is defined as [31,32]:

$$h_x = \frac{q''}{(T_{wall(x)} - T_{bulk(x)})} \tag{18}$$

where total heat flow (W/m^2), wall temperature (K), and bulk temperature (K) are represented by the variables q”, T_{wall}, and T_{bulk}, respectively.

$$T_{bulk} = \frac{\int_0^R uT(2\pi r)dr}{\int_0^R u(2\pi r)dr} \tag{19}$$

The quantity of convective heat transport is determined by the Nusselt number. What the Nusselt number is described as is [33]:

$$Nu_x = \frac{h_x D_i}{\lambda} \tag{20}$$

$$Nu = \frac{1}{L}\int_0^L Nu_x dx \tag{21}$$

Based on the Fanning factor, the friction factor is as follows [34-36]:

$$f = \frac{2\Delta p D_i}{\rho L u_{in}^2} \tag{22}$$

where the pipe length (m), pressure differential (N.m^{-2}), fluid density (kg.m^{-3}), and fluid inlet velocity (m.s^{-1}) are, respectively L, Δp, ρ, and u_{in}.

The thermal performance criterion (TPF) was developed by Gee and Webb and is computed as follows. It takes friction loss and enhanced heat transmission into consideration [37]:

$$TPF = \frac{Nu_{with\_enhancement} / Nu_{without\_enhancement}}{(f_{with\_enhancement} / f_{without\_enhancement})^{1/3}} \tag{23}$$

To analyze all Partial Differential Equations (PDE) of the highest order, a SIMPLE algorithm was used. In the numerical model, the upwind scheme was used, as shown in Figure 3.

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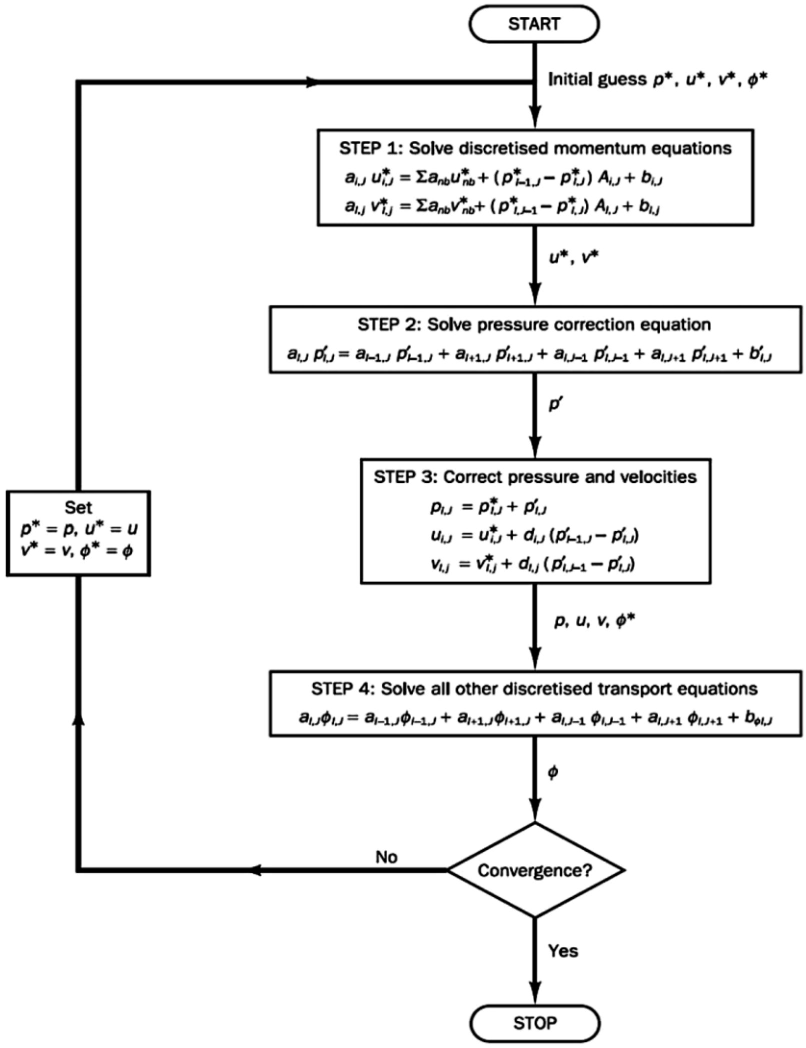


Fig. 3. Flow chart of SIMPLE algorithm for computational model [38]

3. Mesh Modeling of Flow Structure

The Ansys Fluent program was used in the current numerical study to solve the equations governing the continuity, momentum, and heat energy of the fluid flow inside the pipe. Using a separate second-order technique for solving the pressure equation and also using the method of direct approach the problem of coupling of the pressure velocity was solved [39]. To solve the turbulent dissipation rate and equations of both momentum and turbulent kinetic energy a separate second-order technique upwind is used. In the numerical solution of all variables, the convergence criteria are (10^{-8}) and using the turbulence model (k-epsilon) with the improved wall function [40]. In every situation, hybrid meshes that combine hexahedral and tetrahedral components are used to improve computational model accuracy. Tetrahedral meshes are specifically used to provide excellent mesh quality and capture the complex characteristics of protrusions and dimples as shown in Figure 4 and Figure 5 respectively.

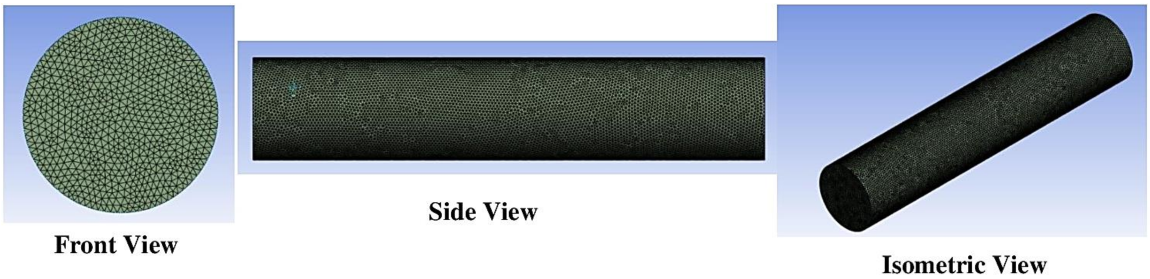
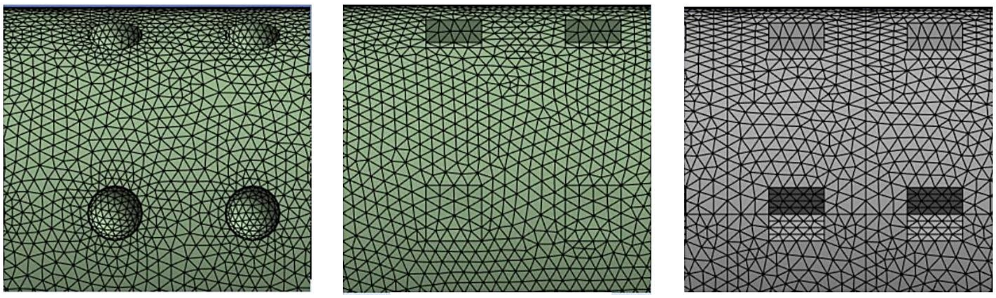


Fig. 4. Three patterns of grid pipe without enhancement technique

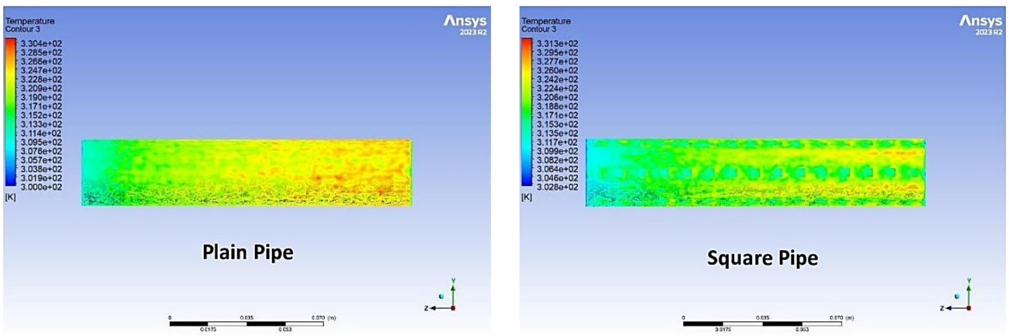


(a) (b) (c)

Fig. 5. Three patterns of grid dimple surface pipe: (a) circular shape, (b) square shape, and (c) triangle shape

4. Results and Discussions

Figure 6 shows the numerical results of the temperature distribution of a normal pipe against different shapes of dimples (square, circular, and triangle) at the value of the Reynolds number (7000) and the constant heat flux along the flow axis ($40kW.m^{-2}$). Increased deformation and temperature change in the presence of dimples can be observed in comparison with a smooth surface with an increase in the average temperature of the liquid along the direction of motion. In addition, the temperature inside the pipe rises by changing the shape of the dimples on the surface of the wall. Temperatures decrease at the inlet of the pipe, but at the outlet, they gradually increase.



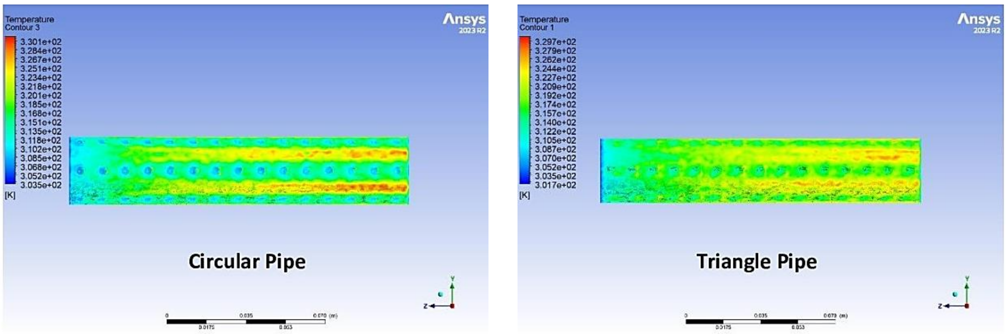


Fig. 6. Axial temperature distribution contour for both plain and dimple pipes at Reynolds number of (7000)

Figure 7 shows the axial and radial pressure distribution of a smooth tube and a tube with a dimpled surface of various shapes (square, circular, and triangular) at a Reynolds number value (7000) and with a constant heat flow at (40 $kW.m^{-2}$). The pressure values at the inlet of the pipe are higher compared to the exit area of the fluid from the pipe, where the higher value of the pressure occurs at the distance between the center and the pipe near the wall. In addition, due to the increased resistance of the fluid during the flow inside the pipe with the presence of dimples, the pressure drop increases compared to the normal pipe.

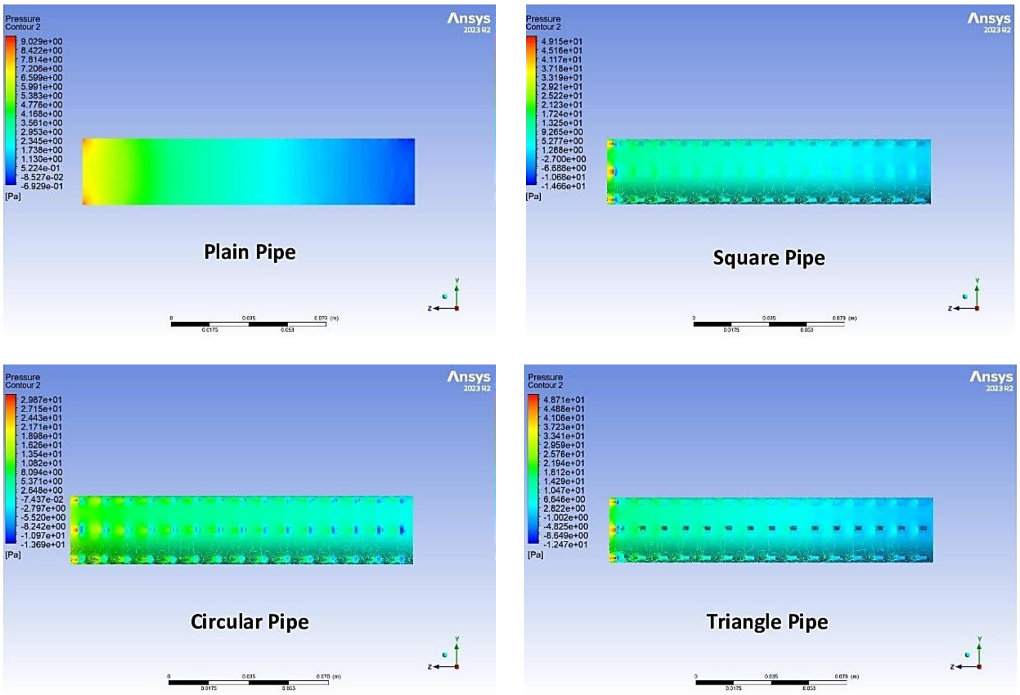


Fig. 7. Axial pressure distribution contour for both plain and dimple pipes at Reynolds number of (7000)

Figure 8 shows the fluctuations of the amount of velocity with/without dimples on the surface of the tube in various forms the average speed increases with dimples. The flow deformation of the dimpled surface and the appearance of eddies can be widely observed, which leads to more effective mixing of the working fluid. It is clear that the local velocity declined downstream along the flow direction and rose upstream of the dimples across the whole dimpled pipe. The development of the boundary layer and consequent improvement in heat transfer efficiency were made possible by the fluid movement on and away from the pipe wall produced by the velocity fluctuation.

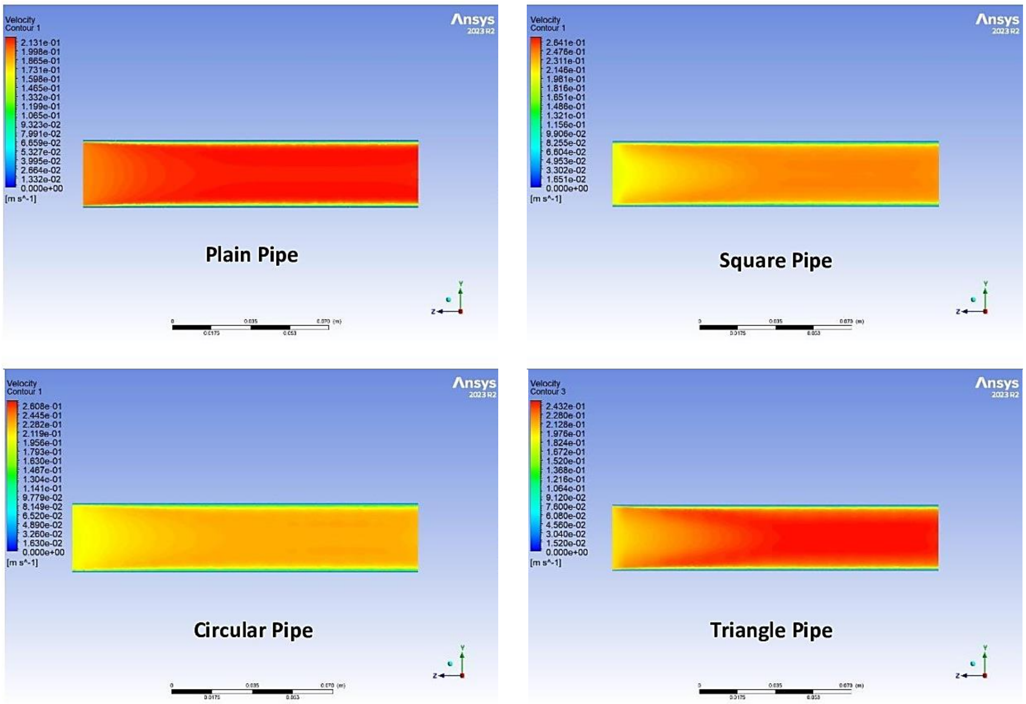


Fig. 8. Axial velocity distribution contour for both plain and dimple pipes at Reynolds number of (7000)

Figure 9(a) shows the change of the Nusselt number with the Reynolds number for variable ranges and different types of dimples on the surface of the tube. In addition, the surface of the dimpled tube in the circular shape gave the highest enhancement of heat transfer compared to the smooth pipe, as well as other dimples (square and triangular), and the reason for this circular dimple generates vortices for fluid flow, better mixing of liquids and periodic collision flows, and therefore circular dimples significantly improved the thermal-hydraulic performance by destroying the boundary layer. Figure 9(b) shows the change of the friction factor against the Reynolds number with variable ranges and for three shapes of dimples (square, circular, and triangle). It can be observed that the friction factor gradually decreases as the fluid velocity increases and also the dimple shape in addition to the circular dimple gave the lowest friction factor compared to the other two forms because it successfully reduces the recirculation of the flow and the loss of low pressure.

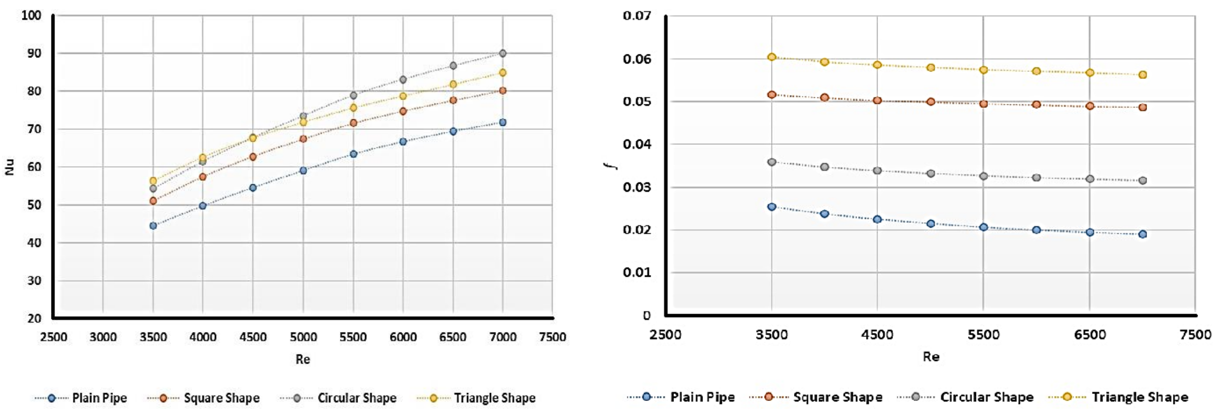


Fig. 9. Influence of dimple shapes on Nusselt number and friction factor (a) Nusselt number, (b) friction factor

Figure 10 displays the change of the thermal friction factor with the Reynolds number ranges at different configurations of the dimples inserted on the surface of the tube. The thermal friction factor can be observed gradually decreasing with increasing fluid velocity. The square shape of the dimples gave the lowest coefficient of thermal friction, while the circular one gave high values.

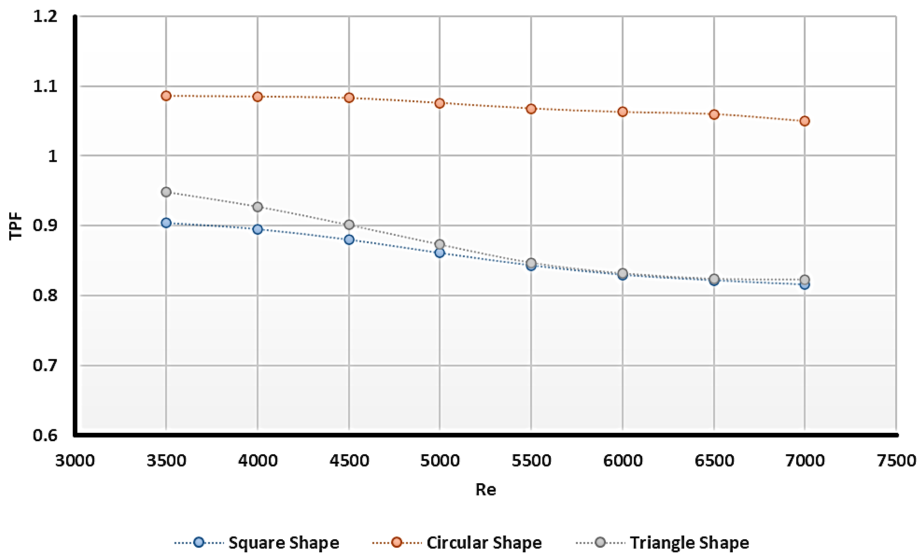


Fig. 10. Change of thermal performance factor of the surface dimpled pipe against the Reynolds number

5. Conclusions

In the current numerical simulations, different types of dimples (square, circular, and triangular) have been used to check the properties of fluid flow and heat transfer and compare them against a three-dimensional empty tube with turbulent flow and steady state. The initial results indicated by the study can be summarized as follows:

i. Percentage (12.808, 17.987, and 20.978%) of the increase in heat transfer enhancement (Nusselt number) between the standard pipe and the three pipe dimple types (square, circular, and triangle), respectively.
ii. The velocity distribution of the fluid changes when using dimples on the surface of the pipe compared to the smooth surface.
iii. At the inlet of the pipe, an increase in the pressure value can be observed, and by increasing the length of the pipe, the pressure drop inside the pipe is reduced.
iv. By increasing the velocity of the fluid inside the pipe and using the dimples on the surface wall, the Nusselt number gradually increases.
v. By eliminating the boundary layer compared to other dimple kinds (square and triangular), circular dimples greatly enhanced the thermal-hydraulic performance.
vi. A noticeable gradual decrease in the friction factor by increasing the Reynolds number of the smooth tube and also the dimpled tube in various shapes.
vii. The hydraulic as well as thermal boundary layer is destroyed as a result of the effect of the dimples on the heat transfer in the pipe.
viii. Gradual decrease of the thermal performance factor by increasing the Reynolds number.

In conclusion, the present study results provide valuable insights for the design and optimization of tube configurations for various engineering and industrial applications that require efficient performance and heat transfer. Various efforts, including dimple engineering, investigation of flow conditions, numerical simulation, as well as conducting an application-specific study, can contribute to enhancing the thermal efficiency of systems, as well as help in improving and designing dimpled pipes of various configurations in different engineering industrial applications.

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