JOURNAL OF ADVANCED INDUSTRIAL TECHNOLOGY AND APPLICATION
e-ISSN: 2716-7097

JAITA

Vol. 6 No. 1 (2025) 70-82
https://publisher.uthm.edu.my/ojs/index.php/jaita

Using Ansys Fluent to Study Flow Characteristics and Heat Transfer Enhancement by Inserting Different Sizes of Dimple in Three-Dimensional Horizontal Single Pipe Heat Exchanger

Sarmad A. Ali[1]*

[1] *University of Babylon, Department of Automobile Engineering, College of Engineering-Al Musayab, Province of Babylon, 51001, IRAQ*

*Corresponding Author: sarmad.ahmed96@uobabylon.edu.iq
DOI: https://doi.org/10.30880/jaita.2025.06.01.006

Article Info

Received: 22 September 2024
Accepted: 24 January 2025
Available online: 30 June 2025

Keywords

Heat transfer, nusselt number, horizontal pipe, numerical simulation, dimples

Abstract

In many engineering applications, the relevance of heat transfer improvement has grown, and a lot of work has gone into using various ways to enhance the hydraulic thermal performance of fluids running through pipes. This study uses numerical analysis to examine the thermal performance of turbulent flow properties and heat transfer under a uniform 30000 W/m^2 heat flux in dimpled pipes. For several designs of dimpled pipes, the friction factor, heat transfer rate, and performance assessment criterion were calculated and compared with smooth pipes. The examples under consideration are within the range of 8000 to 14,000 Reynolds numbers. For this, the ANSYS Fluent 2023 R2 is employed. The governing flow equations are modeled using the Reynolds-averaged Navier-Stokes equations (RANS). To simulate turbulent flow next to the inner wall surface, the realizable k-ε turbulence model is applied with increased wall conditions. The results of the current study showed the presence of dimples on the surface of the pipe significantly enhances the rate of heat transfer represented by the Nusselt number compared to the normal smooth pipe. Also, the analyzed models indicated by the results of the numerical investigation have high average Nusselt counts, low pressure, overall performance criterion, and low average thermal resistance due to the increase in the Reynolds number. The dimpled pipe with different radii (2,3, and 4 mm) has an increased percentage of enhanced heat transfer (79.91, 86.77, and 94.47%) compared to the smooth pipe.

1. Introduction

In many devices for industrial applications, such as cooling the blade of a gas turbine, cooling microelectronic devices, as well as heat exchangers, improving heat transfer is of great importance and an important role. In many previous literature, attempts have been made to increase heat transfer rates using surface modification methods in applications where temperature control is of paramount importance in the maintenance of their performance and operation due to the small size and large energy consumption being electronic devices. The use of protruded surfaces, pointed surfaces, rib beaters, and pin fins is one of the techniques to improve the heat transfer process. Many previous research articles have shown the effect of different shapes of dimples, including square, rectangle, triangle, circular, and rhombus on the thermal performance of heat transfer and fluid flow inside the tube. The results of their study showed that the dimples contribute to improved heat transfer compared to not using them on the surface of the pipe [1-6]. Xiang Zhang et al. [7] a numerical study focusing on the use of elliptical dimples in three different cases, the first horizontal, the second vertical, and the third inclined at an angle to the tube wall for heat transfer by forced convection of single-phase flow for a Reynolds number range (4080-24480) to improve the flow characteristics of a car radiator. The results of the study indicate that the use of dimples improves the heat transfer process compared to a regular pipe. The coefficient of thermal performance of the third case is higher than 1.39, so it gives the best improvement compared to the other two cases. Abeer H. Falih et al. [8] numerical investigation by using a program of the finite element method (Ansys Fluent 2022 R1) to solve the governing equations of single-phase fluid flow and the Reynolds number range (3000-8000) to study the effect of changing the diameter of the dimples of the heat transfer pipe and also the distribution of the dimples in two ways, the first on one line of the pipe surface and the second in staggered arrangements. The results of the numerical study indicated that the process of heat transfer by forced convection improved with the use of dimples compared to the smooth tube. Recently, many publications of researchers have studied the effect of various geometric parameters of the dimples on the surface of the tube to indicate their effect on the thermal response, including the diameter of the dimples [9-11], the distance between the dimples [12-14], the shapes of the dimples [15-17] and the angle [18 and 19]. The results of the investigations when comparing the conventional smooth tube with the use of dimples showed a clear difference in enhancing the heat transfer in the tube for heat exchangers. Ming Li, et al. [20] numerical and experimental study of forced convection heat transfer and water flow as a working fluid in a pipe with a dimple with single-phase turbulent flow for the Reynolds number range (500-8000). Several parameters were studied, including the Nusselt number, the coefficient of thermal performance, and the friction factor with the presence of dimples on the surface of the pipe to compare it with a conventional smooth empty pipe. Experimental relationships were derived for the Nusselt number and friction factor based on experimental data. The results of numerical simulations showed the dimples lead to the generation of secondary flows that improve the level of turbulence. Toygun Dagdevir and Veysel Ozceyhan [21] an experimental study of the improvement of heat transfer and water flow characteristics in a single-phase turbulent flow heat exchanger of the Reynolds number range (5217-22754) exposed to a constant heat flux. The method of adding twisted tape was used in three different cases, the first is empty, the second is perforated, and the third is a dimple inserted into the tube for comparison. The results of the study recorded a sign of improvement in heat transfer characteristics using twisted tape with a dimple compared to perforated and smooth. S. Vignesh et al. [22] experimental and CFD analysis of a concentrated tubular heat exchanger in which water is used as the working fluid in two cases, the first is an empty tube and the second contains spherical dimples with a variable mass flow rate. A different method is used to increase the heat transfer rate without affecting the performance of the overall system. Some parameters were studied, including the heat transfer rate, the heat transfer coefficient, and the efficiency of the heat exchanger. The results of the study recorded an improvement in heat transfer using the dimpled tube compared to the usual one. S. Chokphoemphum et al. [23] an experimental study of improving heat transfer using a V-Wing at different angles (30,45, and 60 degrees) in a tube of a heat exchanger for comparison with a regular tube. Air was used as a working fluid with turbulent single-phase flow for the Reynolds number range (5300-24000). Sarmad A. Ali [24] presented a numerical study to improve the properties of heat transfer by forced convection and turbulent flow in Reynolds number ranges (9000-18000) in a two-dimensional channel by inserting ribs of various configurations, including a quarter circle, square, and a triangle. The results indicated the ribs improved heat transfer compared to the empty channel and the higher quadrant configuration better improved. Moreover, the number of Nusselt is gradually increasing by increasing the Reynolds number. Falih, Abeer H. et al. [25] numerically studied using (Ansys Fluent 2022 R1) program to improve the hydrothermal performance of a mandrel pipe with different diameters as a means to enhance the heat transfer of water (working fluid). To treat the turbulent flow with Reynolds number ranges (3000 - 8000), the finite element method was used in the numerical simulation in addition to solving the partial differential equations related to the flow. The results indicated that the Reynolds number plays an important role in changing the parameters, as the Nusselt number increases with increasing Reynolds number, in addition to decreasing the friction factor, decreasing pressure, and thermal resistance. Finally, the models used in the study showed a clear improvement compared to the regular model.

Therefore, in comparison to plain pipe, the current work investigates turbulent forced convective heat transfer flow in 3D horizontal pipe with a dimple for various radii (2, 3, and 4 mm). Under a constant heat flux condition of 30000 W/m^2 on the outer pipe wall surface, the flow is termed a turbulent-single-steady state, with Reynolds numbers in the range of $8{,}000 \leq Re \leq 14{,}000$. Important findings include the average surface temperature, friction factor, thermal performance factor, and Nusselt number. The difference in the diameters of the dimples distributed in order around the pipe wall with the difference in operating conditions included in the values of heat flow, diameter, and length are considered factors for the originality and modernity of the current study compared to previous literature.

2. Materials and Methods

2.1 Physical and Mathematical Numerical Model

A three-dimensional horizontal pipe made of aluminum with an interior radius of D is used to generate the computational domain using commercial Computational Fluid Dynamics (CFD) code (ANSYS Fluent 2023 R23) for steady incompressible turbulent flow. The pipe's whole length measured (L = 150 mm). Furthermore, the outer surface of a pipe is subjected to a constant uniform heat flux of 30,000 W/m^2. Different dimple radii (2, 3, and 4 mm) will be studied, with the pipe diameter (D) being 30 mm. Figures 1 and 2, respectively, provide a schematic diagram of the dimple pipe parameters and computational domain. To examine the impact of dimples on turbulent flow and temperature field for improved pipes, the realizable k–ε model was utilized. The Semi-Implicit Method for Pressure Equations Consistent (SIMPLEC) is used to solve the pressure–velocity coupling in a steady state. Table (1) provides a summary of the boundary conditions that were developed. The listed dimples for improving heat transfer have specifications where the cross-section of the tube is (6) while the number of target bodies towards the depth of the (z) axis is (15). In addition, the total number of dimples on the surface of the tube is (90).

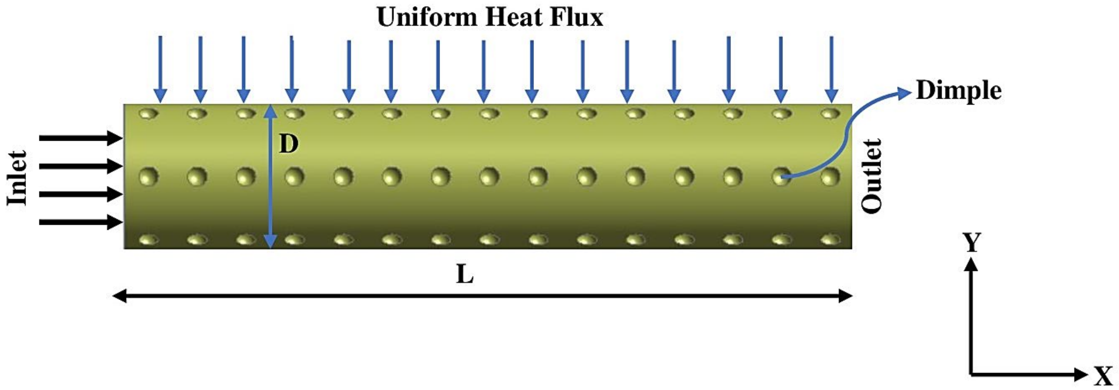


Fig. 1 *Schematic illustration showing the size and parameters for dimple pipes*

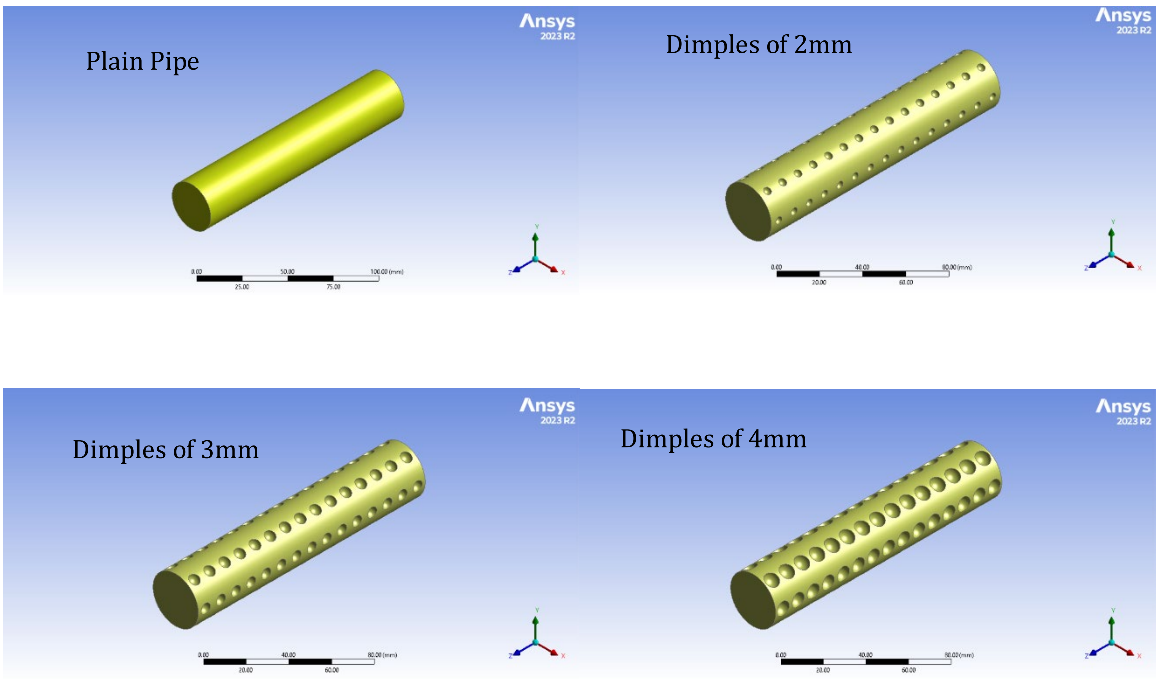


Fig. 2 *Computational domain of plain pipe and dimpled pipe for different radii*

Table 1 *Summary of computation domain boundary condition*

Surface	Boundary Condition	
	Thermal	Momentum
Inlet	293 K	Inlet of Velocity
Outlet	----	Outlet of Pressure
Wall	Heat flux of 30000 W/m²	No Slip Condition

2.2 Equations of Governing

The governing equations of fluid flow through the tube and heat transfer can be expressed as follows [26 and 27]:
For the equation of continuity:

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

For the equation of momentum:

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

For the equation of energy:

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

Also, the equations of turbulence kinetic energy (k) and dissipation of energy (ε) can be expressed as follows [26-28]:

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

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

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

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

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

The following coefficients can be found in the realizable κ −ε equations mentioned above [29].

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

Here, C_1, C_2, C_3, and C_4 are constant of turbulence model (dimensionless), Cp is specific heat at constant pressure (J/kg. K), k is turbulent of kinetic energy (m^2/s^2), T is the fluid temperature (K), ε is the turbulence of kinetic energy dissipation rate (m^2/s^2), ρ is the density of fluid (kg/m^3), μ is the dynamic viscosity of fluid ($N.s/m^2$), λ is the thermal conductivity of fluid (W/m. K).

2.3 Boundary Condition and Specification of Data

The following assumptions are used regarding uniform temperature and velocity profiles at the three-dimensional horizontal pipe entrance:

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

Both the turbulent dissipation rate and the turbulent kinetic energy entry profiles are computed from:

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

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

The following boundary conditions apply upon exit:

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

At the exit, the boundary condition of a model (k-ε) is as follows:

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

Since there is no slip condition at the walls and all velocity components are zero:

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

The boundary condition of turbulence model (k) is:

$$k = 0 \;,\; \frac{\partial \varepsilon}{\partial r} = 0 \tag{15}$$

The following boundary condition will apply for the constant heat flux: $\frac{\partial T}{\partial r} = \frac{q''}{\lambda}$ (16)

The Reynolds number is a representation of the ratio between viscous and inertial forces. This is how the Reynolds number is expressed [30]:

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

The definition of the coefficient of heat transfer is [31]:

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

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

The Nusselt number is used to calculate the amount of convective heat transfer. The description of the Nusselt number is [32 and 33]:

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

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

The following is the friction factor, as determined by the Fanning factor [34]:

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

Here, p is the pressure of fluid (N/m^2), Nu and Re are the Nusselt number and Reynolds number (dimensionless) respectively, L is pipe length (m), D is the diameter of the pipe (m), u_{in} is the inlet fluid velocity (m/s), h is the heat transfer coefficient (W/m². K), q" is the heat flux (W/m²), u, v, and w are the velocity of the fluid in x, y, and z direction (m/s) respectively.

A SIMPLE approach was employed in Ansys fluent software to analyze all Partial Differential Equations (PDE) of the highest order. Figure 3 illustrates the application of the upwind strategy in the numerical simulation model.

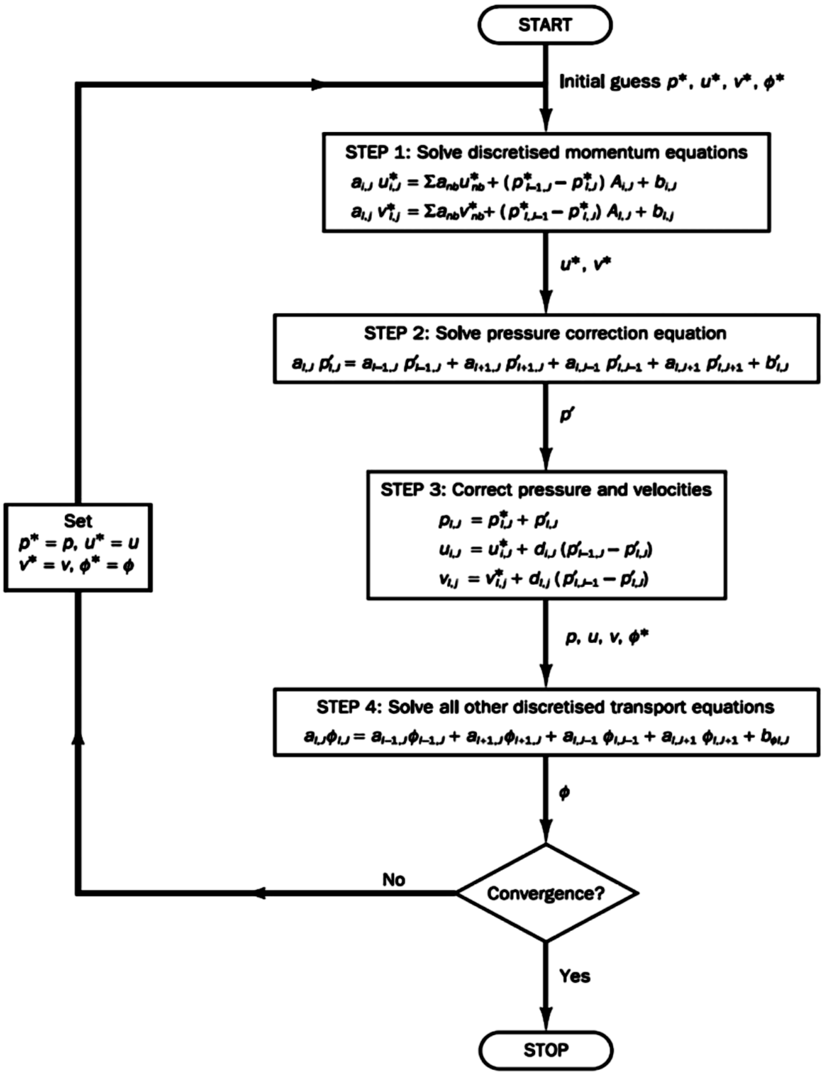
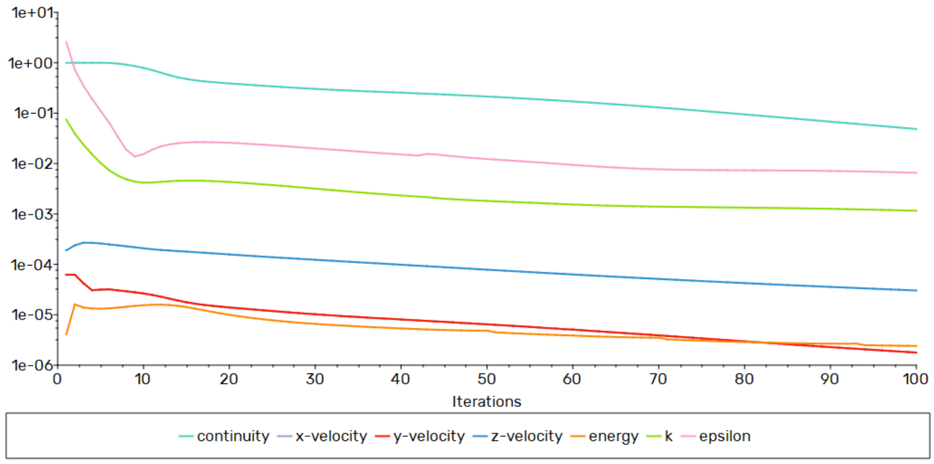


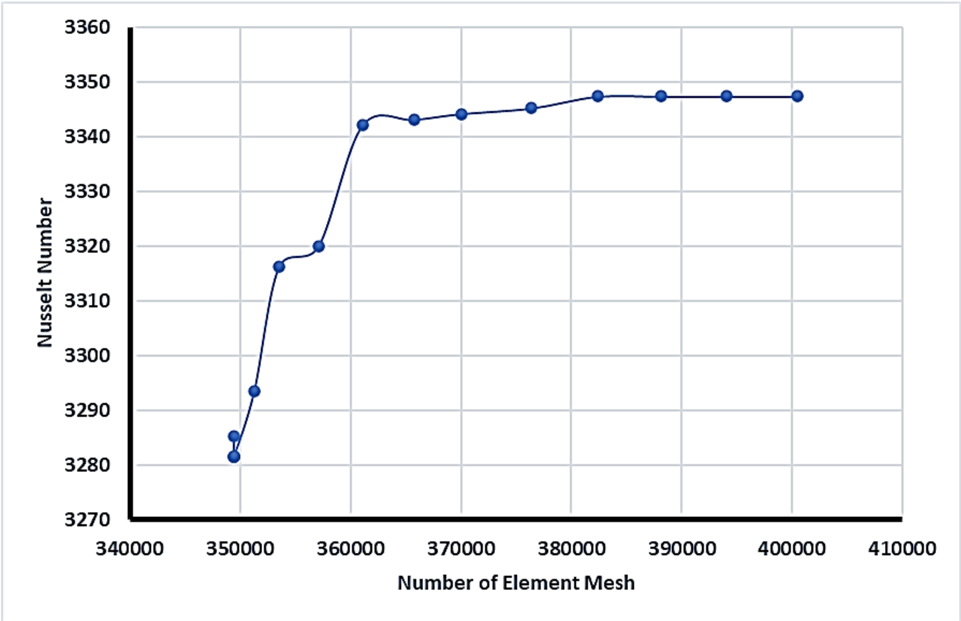
Fig. 3 *The SIMPLE method flow chart for the computational numerical model [35 and 36]*

2.4 Mesh of Modeling and Flow Structure

In the current numerical investigation, the Ansys Fluent 2023 R2 program was used following the finite volume approach to solve the differential conservation equations of continuity, momentum, and energy. Using the method of direct approach and the second-order discrete technique the problem of speed coupling and pressure equalization was solved, respectively [34]. A perturbation model with an improved wall function was used to solve the computational domain problem. To solve the equation of momentum, turbulent kinetic energy, and turbulent dissipation rate a separate technique was used to reverse the direction of flow. The convergence criteria for the numerical solution of all variables are 10^{-6} and to improve the solution results, the mesh independence test is used by changing the number of divisions and thus increasing the number of mesh elements to reach the stable state of heat transfer as shown in the figure (4). The generation of the mesh grid for the smooth horizontal pipe with different directions and also for the dimpled pipe with variable radius are shown in Figures (5 and 6) respectively.

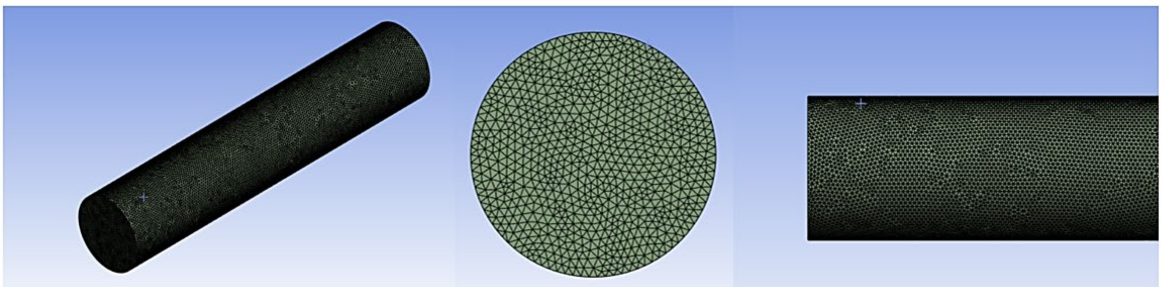


(a)



(b)

Fig. 4 *Numerical solution of computational domain (a) Standard convergence; and (b) Mesh test*



(a) **(b)** **(c)**

Fig. 5 *Mesh grid of pipe without enhancement technique (a) Isometric view of 3D pipe; (b) Front view; and (c) Side view*

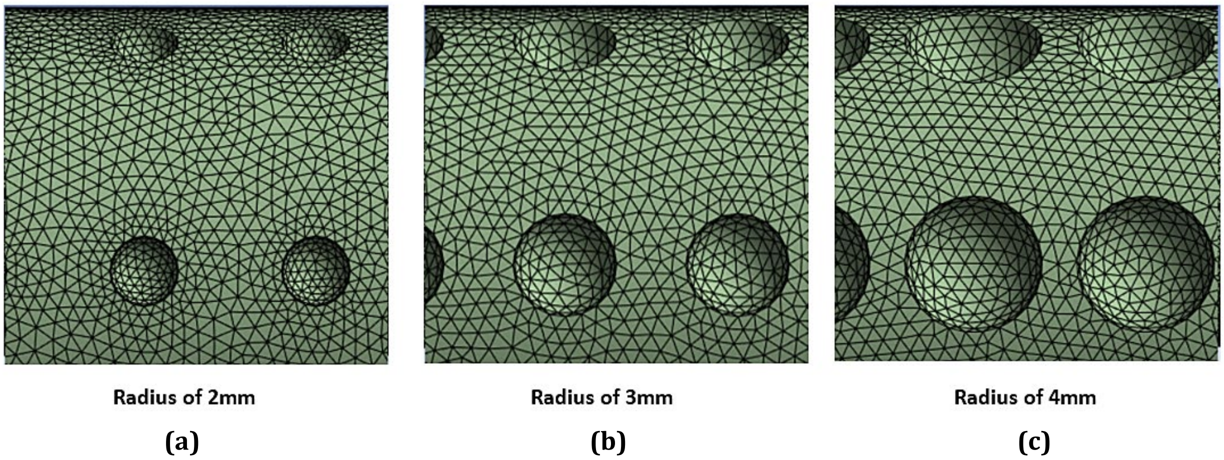


Fig. 6 *Grid of mesh for different dimples: (a) 2mm; (b) 3mm; and (c) 4mm*

3. Results and Discussions

3.1 Contour of Temperature

Figures (7-10) show the temperature distribution of a smooth pipe wall for comparison against a dimpled pipe of different radii (2, 3, and 4mm) at a range Reynolds number of (8000 -14000). The rate of fluid temperatures can be observed increasing towards the flow axis. The dimpled pipe with the smallest radius of (2 mm) gave the highest average temperature compared to other radii and the pipe without dimples.

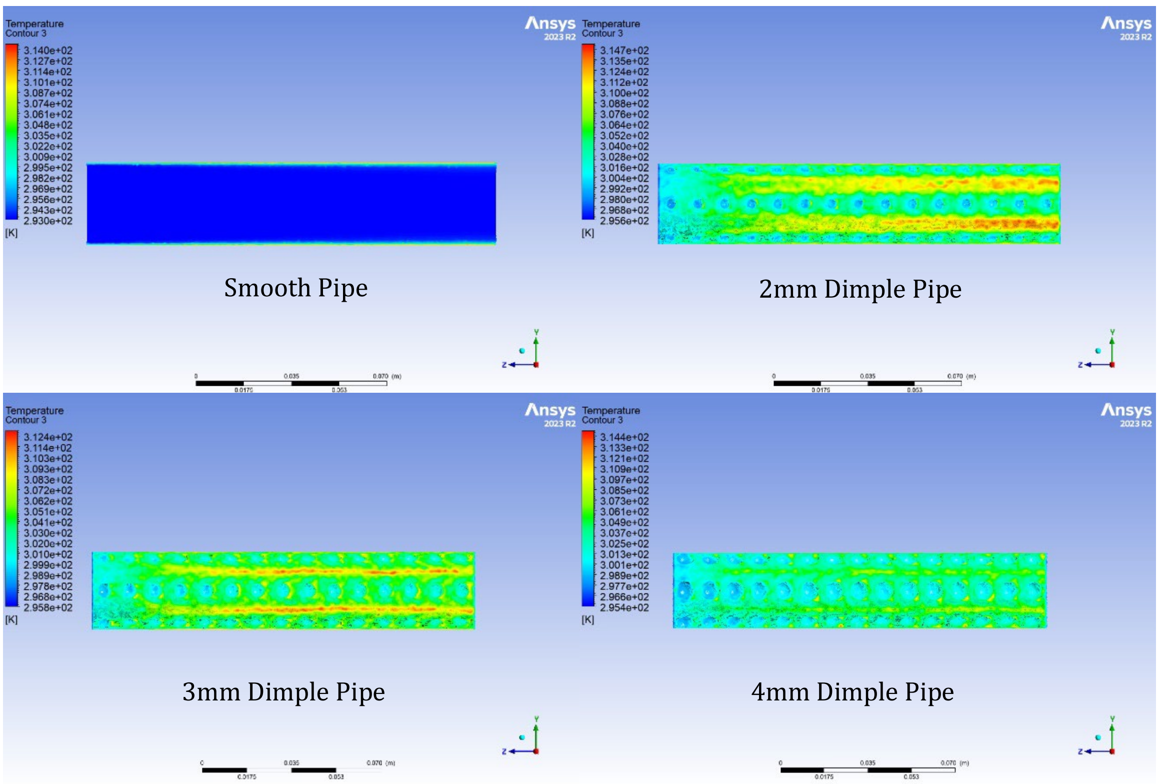


Fig. 7 *Variation of contour temperature for smooth and dimple pipe at (Re=8000)*

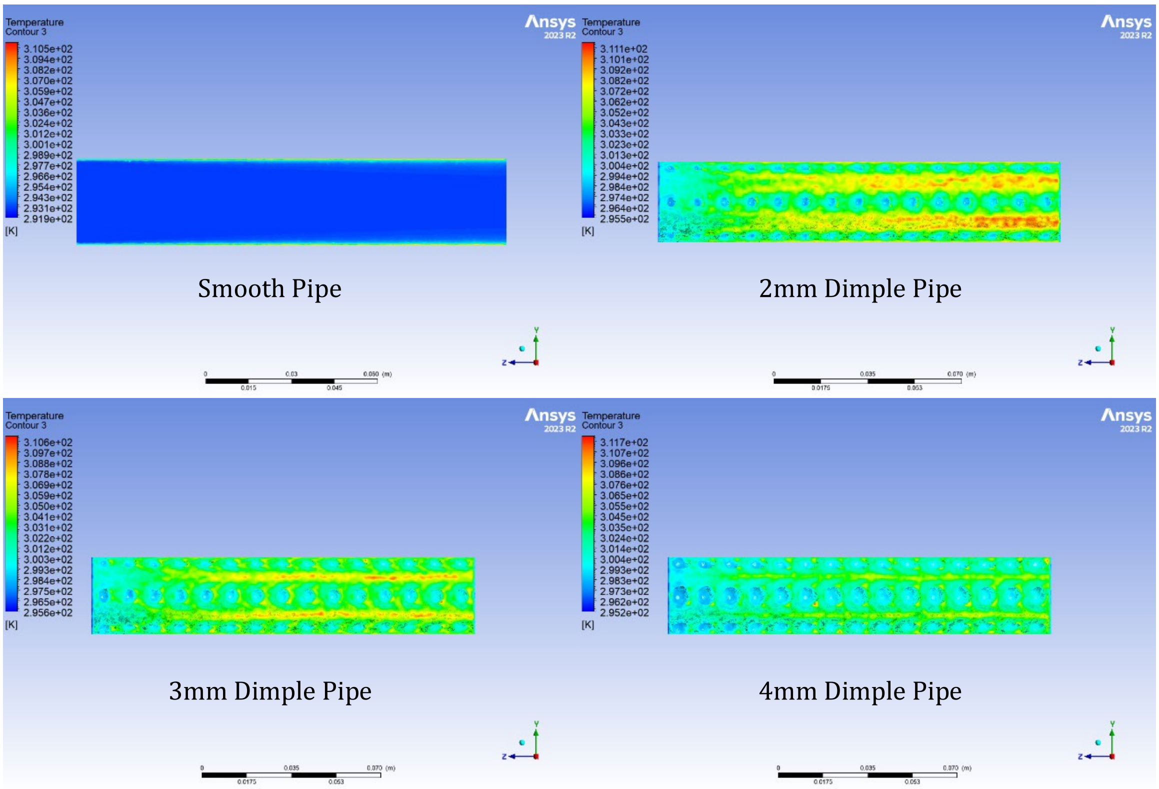


Fig. 8 *Variation of contour temperature for smooth and dimple pipe at (Re=10000)*

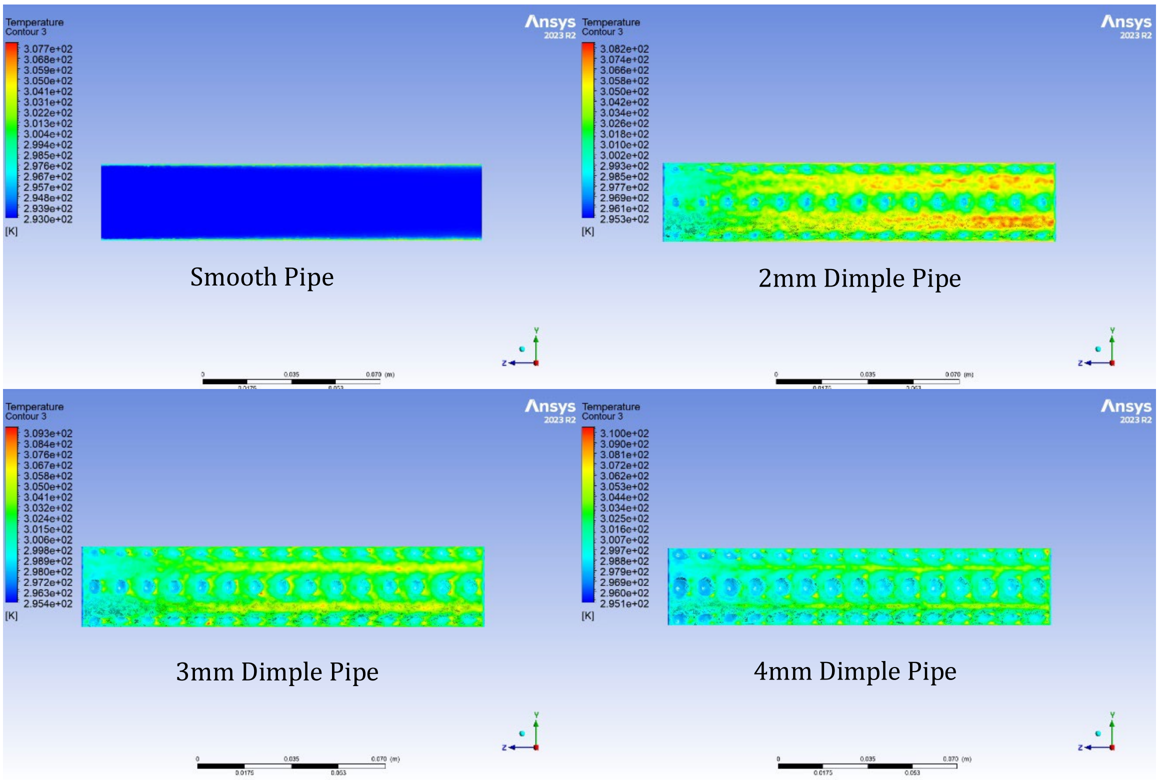


Fig. 9 *Variation of contour temperature for smooth and dimple pipe at (Re=12000)*

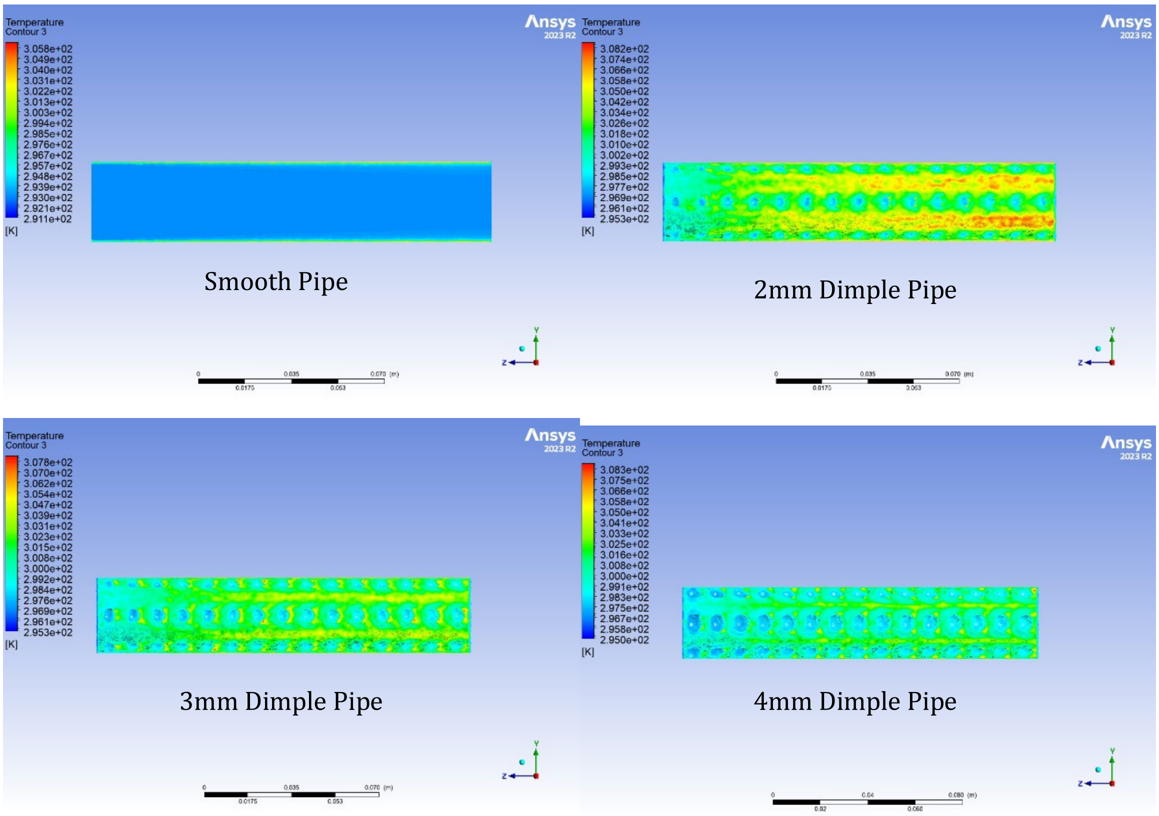
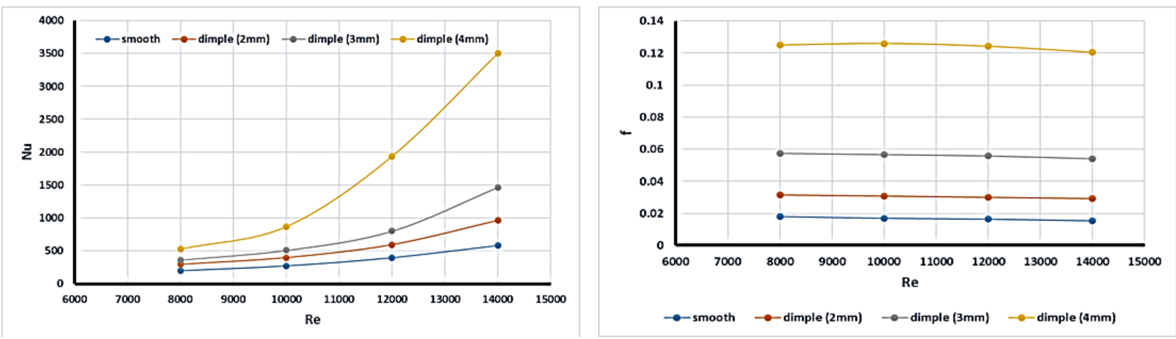


Fig. 10 *Variation of contour temperature for smooth and dimple pipe at (Re=14000)*

3.2 Nusselt Number and Friction Factor

Figure (11 a) shows the change of the Nusselt number against different ranges of the Reynolds number for smooth pipe and dimpled tubes of different radii. The Nusselt number can be observed gradually increasing in two cases, the first with an increase in the Reynolds number, the second with an increase in the radius of the dimples compared to the empty pipe, thus the use of dimples significantly improves the thermo-hydraulic performance as a result of the destruction of the boundary layer and causing a mixed flow collision. Figure (11 b) shows the change of the friction factor with the Reynolds number for variable ranges. It can be observed that the friction factor gradually decreases significantly by increasing the Reynolds number as a result of increasing the fluid velocity, causing a decrease in the friction factor, and also noting the increase in the radius of the dimples, causing an increase in friction compared to not using a smooth pipe.



(a) **(b)**

Fig. 11 *Influence of smooth and various dimples radius on Nusselt number and friction factor (a) Nusselt Number; and (b) Friction Factor*

3.3 Surface Pipe Temperature

Figure (12) displays the average surface temperature of the pipe with the change of the fluid velocity represented by the Reynolds number and for different ranges (8000 - 14000). The decrease in the surface temperature of the pipe can be observed gradually by increasing the Reynolds number and also by increasing the radius of the dimples compared to the usual empty pipe. The dimples cause temperature dispersion as well as the fluid not gaining enough heat when flowing inside the pipe.

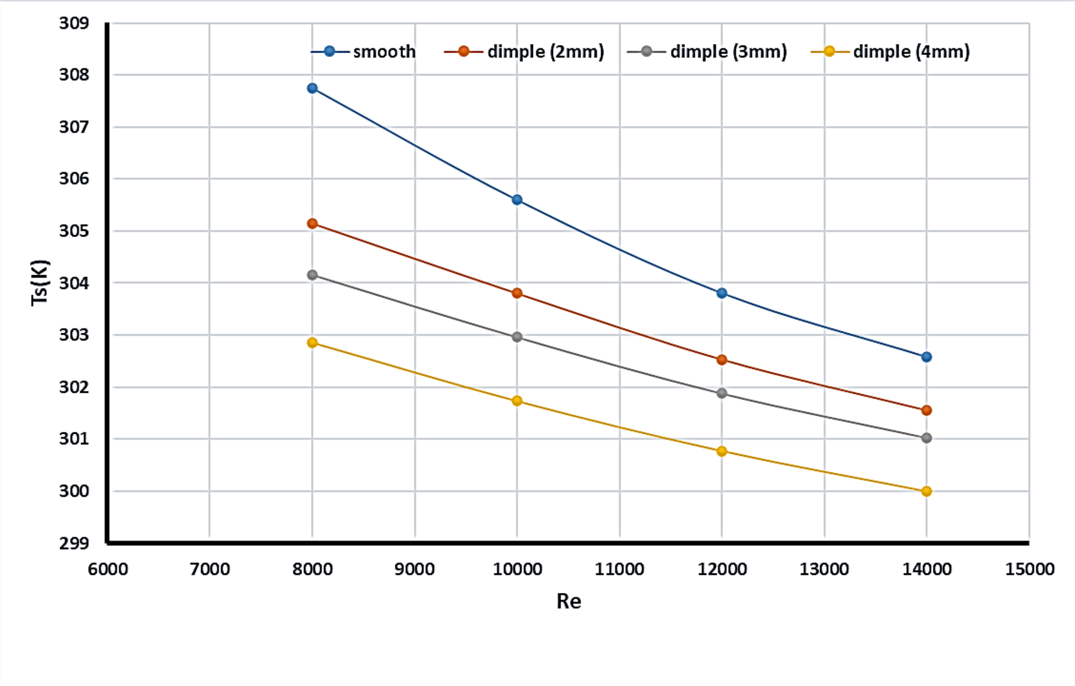


Fig. 12 *Effect of plain and radius dimples on surface temperature of single pipe heat exchanger*

4. Conclusions

A numerical study of forced convection heat transfer in a horizontal three-dimensional pipe with varying radii for single-phase turbulent fluid flow steady-state to improve heat transfer rate by using circular dimple in comparison to a smooth pipe (without dimple). The present investigation's findings show that, in comparison to a smooth pipe, heat transmission is enhanced when a dimple is added to the horizontal pipe's surface. Additionally, as a result of increased turbulent flow, the Nusselt number is progressively raised by raising the Reynolds number of the fluid inside the pipe, and it becomes more uniform when the stream's flow pattern is destroyed. When it comes to adjusting the parameters, the Reynolds number is crucial since it decreases the friction factor, pressure, and heat resistance while also raising the Nusselt number.

Acknowledgments

In this numerical investigation, the author wishes to recognize both the University of Babylon- College of Engineering Al Musayib -Department of Automobile and the University of Technology- Department of Mechanical for their support.

Conflict of Interest

The author declares that there is no conflict of interest regarding the publication of the paper.

Author Contribution

The authors are responsible for the study conception, research design, data collection, data analysis, result interpretation and manuscript drafting.

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