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# Numerical Investigation of Thermal Performance for Turbulent Water Flow through Dimpled Pipe

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**ABSTRACT**

Dimples are tiny indentations on the surfaces of the body that enhance heat transfer and alter fluid flow characteristics on or within the body. Numerical investigations were conducted to analyse the heat transfer and flow characteristics in a cross-combined dimple tube in the range of Reynolds numbers from 6000 to 14000. The finite volume method recognised a novel enhancement model utilising methods for composite-form surfaces. Compared to a smooth tube working similarly, the effects significantly improve the heat transfer index, performance evaluation criteria, and friction factor. A three-dimensional simulation was conducted to clarify the underlying process by which dimples affect thermal performance. The simulation findings suggest that the dimples effectively enhance heat transfer by altering the temperature distribution and increasing the temperature gradient within the central area of the dimple section. A concave surface profile disrupts the flow and prevents the formation of a stable boundary layer, promoting the mixing of hot and cold fluids. Furthermore, the study investigates how geometric characteristics impact thermal and hydraulic efficiencies, emphasising that larger dimples improve overall thermo-hydraulic performance. Specifically, the heat transfer enhancement achieved an average increase of 17.3%, ranging from 18.03% to 38.6%, surpassing that of the traditional smooth tube.

## 1. Introduction

Heat transmission is an inherent process that occurs naturally. Heat exchangers are utilised to transport heat efficiently and cost-effectively. Studying the performance of thermal devices is a crucial area of focus in engineering. Enhancing heat transport is crucial for sustained energy expansion. Researchers have persisted in their investigations of heat transfer enhancement in the literature. Several researchers have focused on improving heat transfer by utilising intricate geometric designs to induce secondary fluid motion and boost the velocity of heat transmission as shown in the previous studies [1-4].

From a financial and industrial standpoint, a significant imperative exists to enhance the efficiency of a heating and cooling switch system that may be installed by Al-Qalamchi and Adil [5]. Augmenting

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the heating switch entails facilitating ease of use and mitigating excessive heat production to avert the dangerous ramifications of combustion and overheating was mentioned by Amit and Dingare [6]. Temperature stage in application: The only factor that may differ is the floor area corresponding to the unit size. Several strategies for doubling a hot switch include a formal change of the existing shape or inducing a perturbation mutation studied by Ehan [7]. Disturbance is needed to suppress the thermal boundary layer's increase or to reduce its thickness. It is often discovered that enforcing these strategies regularly may also lead to reduced stress. Accordingly, a larger layout is usually beneficial. The design must also be good enough to address issues such as surface scaling and fouling, which generally tend to lower heat exchange charges in the case of hulls' use of lower-grade fuels Zunce *et al.,* [8]. There are several strategies to increase key hotness, such as twisted strip inserts [9-11], screw fins, slot fins, slotted fins, ribs by Ansam *et al.,* [12], ridges, and dimple tubes by Falah [13] and Abdul *et al.,* [14]. Of the approaches listed, the dimple tube offers the most improvements in warmth switching costs, with the lowest pressure drop compared to others. Engineering versions, in particular, include long decks, but much research is being done thanks to the call for light, compact, and more cost-efficient discovery. Plus, this era could be beneficial in various petrochemical heat exchange methods because it provides a minimal pressure penalty and the ability to reduce fouling charges investigated by Harsh *et al.,* [15].

Tappet tubes can be used in many applications, including evaporators, condensers, oil radiators, and heat exchangers. In addition, it is frequently applied in biomedical equipment, aerofoil internal cooling for fuel turbines, gasoline additives for nuclear power plants, and microelectronic cooling investigated by Li 16]. In a sample, drainable dimples were often used on the inner face of tubes to make more muscular surfaces. This allowed researchers like Wenguang and Zhibin [17] to look at how the cooling changed on the inverted concave floor of the digital chip assemblies and found that the change in the near Nusselt number on the velvet is not the main factor affecting heat distribution.

A computational and experimental investigation of a set of system and operational parameters was done to assess the influence of the split rib on the fluid flow and heat transfer behaviour in tube heat exchangers [18]. A new enhanced tube with slotted dimples has been introduced to facilitate heat transfer. The ETSD's flow and heat transfer properties are digitally calculated and compared to those of spherical or oval dimples [19]. Temperature, velocity, pressure, Nusselt number, and rheology distributions remained calculated to characterise the mechanisms of heat transfer and fluid flow. It was determined that dimples increased friction pressure losses and heat transfer rates. The findings above, prevalent across various geometric configurations and Reynolds numbers [20], have been supported by other independent investigations [21-24]. Some famous works have also shifted their recognition to enhance the design [25,26].

Similarly, dimples affect float styles locally (each adjacent to and close to the dimples). Heat transmission intensifies because of the destruction of the thermal and pace boundary layers resulting from the dimples placed over the tube surface. It can cut up the float and bring a two-dimensional float to its pinnacle [27,28].

The heat transfer zone on the surface of these channels exhibited a drop in the top portion of the soft cavity and then an increase in the transmission of heat in the lower half. Additionally, additional regions of significant heat transfer were observed downstream of the border of the blob. Using both experiments and CFD, test and study a concentric tube heat exchanger using standard and spherical junction tubes with different mass flow rates [29]. It is recommended to use a dimpled line to replace the inner tube of a concentric tubular heat exchanger. At various flow rates, the influence of the dimple tube on heat transfer rate, heat transfer coefficient and efficiency matches that of the traditional tube. The efficiency heat transfer coefficient and heat transfer rate are compared to a standard tube's. Agreeing with experimental and CFD data, using water in a spherical, dimpled line increases the total heat, heat transfer rate, transfer coefficient, and effectiveness compared to a standard tube [30].

Other ways to enhance the thermal performance is using nano-based fluids as the heat transfer fluid (HTF) inside heat exchangers to enhance Nusselt number, distribution of velocity and temperature, pressure drop entropy generation, and friction factor [31,32].

This article's primary aim is to quantitatively study how the inner surface of a circular tube's inverted ribs impacts convective heat transfer and fluid flow in a turbulent environment. A new type of improved tube known as a double tubular tube with different diameters has been constructed as a result of the application of shape composite surface techniques; additionally, a numerical model is run on the tube to generate an estimation of fluid flow in the range of 6000–14000 Reynolds number (Re) and its effects on heat transfer. Overall, the article's novelty lies in combining the unique tube design, the application of inverted dimples, numerical modelling, and the focus on quantitatively studying convective heat transfer in a turbulent flow regime.

## 2. Mathematical Model

### *2.1 Physical Domain*

This study presents a schematic representation of a concentric twin-tube heat exchanger where water is used as the working fluid, as depicted in Figure 1. The inner tube was used to pass hot water, while the annulus was used to flow cold water. The double-pipe heat exchanger consists of an internal tube made of copper with a thickness of 2 mm. The inner tube has an inside diameter of 16 mm and a length of 2470 mm. The outer tube, conversely, is constructed of G.I. by a thickness of 3 mm. The outer tube has an interior diameter of 32 mm and a length of 2400 mm. The dimples' effectiveness in enhancing the heat transfer rate has been estimated based on measurements of the internal tube surface, outlet temperatures, and pressure drop. This research investigates the variations in physical properties of dimples, specifically those with diameters of 7mm, 5mm, and 3mm. The range of Reynolds numbers for the hot water passing through the inner tube was adjusted between 6000 and 14000. The Reynolds numbers employed in the study are determined by the tube's hydraulic diameter and the working fluids' mean input velocity. The average inlet velocity boundary is determined using the specific Reynolds number, and a pressure exit boundary is employed. So, the fluid in motion enters the tube, which is oriented in the z-direction, and a nonslip boundary condition characterises the tube walls.

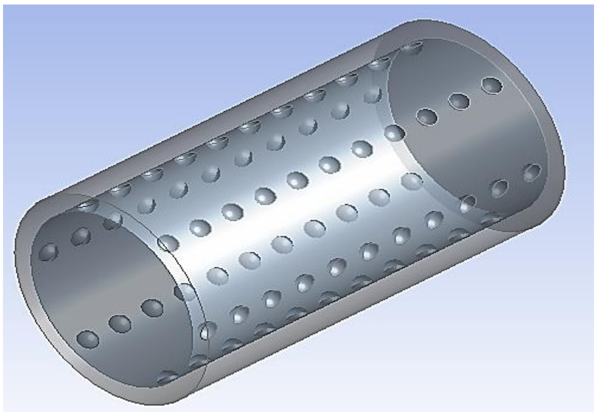


**Fig. 1.** Design Model - Double pipe heat exchanger

### *2.2 Governing Equations*

This study assumes that the fluid flow is characterised by continuity, steadiness, and incompressibility while also considering the insignificant influence of gravity. The simulation strategy that yielded the most favourable results was implementing a second-order k- ε turbulence model executed on a finely discretised grid consisting of 791,200 elements. The evaluation of convergence in the calculation is performed by observing continuity residuals, velocities, and energy, with the convergence criterion set at $10^{-6}$. The subsequent conservation equations for Navier stock's are formally expressed as follows by Ahmed *et al.,* [33]:

$$\frac{(\partial(\rho\rho \mathrm{ui}))}{(\partial xi)} = 0 \tag{1}$$

Where ρ= Density (kg/m$^3$)

The momentum is represented as

$$\frac{\partial(\rho\overrightarrow{u_i u_j})}{\partial(x_j)} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right) - \frac{\partial\left(\rho\overrightarrow{u_i{}^{\surd}u_j{}^{\surd}}\right)}{\partial(x_j)} \tag{2}$$

Where $\underset{u}{\rightarrow}$ = Velocity Vector, μ = Viscosity (Pa ·s), p = Pressure (Pa.)

The energy equation is presented as

$$\frac{\partial}{\partial \mathrm{x_i}}[u\_j\ (\rho e + p)] = \frac{\partial}{\partial \mathrm{x_j}}\left(\lambda\frac{\partial \mathrm{T}}{\partial \mathrm{x_j}}\right) \tag{3}$$

Where λ = Thermal Conductivity (W/m. K), T = Temperature (K)

In this numerical study, the Re Normalization Group RNG form of the (k-ε) model is rummage-sale. The Turbulent kinetic energy k is present by

$$\frac{\partial}{\partial \mathrm{x_j}}(\rho \mathrm{kuj_j}) = \frac{\partial}{\partial \mathrm{x_j}}[(\mu + t\mu/\sigma\_k\ )\lambda\ \partial k/(\partial x\_j\ )] + \mathrm{G_k} - \rho\varepsilon \tag{4}$$

$$\frac{\partial}{\partial \mathrm{x_j}}(\rho\varepsilon \mathrm{u_j}) = \frac{\partial}{\partial \mathrm{x_j}}\left(\left(\mu + \frac{\mu_\mathrm{t}}{\sigma_\varepsilon}\right)\lambda\frac{\partial\varepsilon}{\partial \mathrm{x_j}}\right) + \rho \mathrm{C_1 S_\varepsilon} - \rho \mathrm{C_2}\frac{\epsilon^2}{\mathrm{k}+\sqrt{v\epsilon}} \tag{5}$$

$$\mathrm{G_k} = \mu_\mathrm{t}\mathrm{S}^2 \tag{6}$$

$$\mu_\mathrm{t} = \rho \mathrm{C}_\mu\frac{\mathrm{k}^2}{\varepsilon} \tag{7}$$

k is Turbulent Kinetic Energy, $G_k$ = Turbulent kinetic energy and Cμ is not constant popular k-ε realizable model. It is intended from empirical relation. The calculation of Cμ is given by Vegnish *et al.,* [34]. The model constants are:

$$\mathrm{C_1} = max\left[\frac{43}{100}, \frac{\eta}{\eta+5}\right], \eta = \mathrm{S}\frac{\mathrm{k}}{\varepsilon}, \mathrm{S} = \sqrt{2\mathrm{S_{ij}S_{ij}}}, \mathrm{C_2} = 1.9, \sigma_\mathrm{k} = 1, \sigma_\varepsilon = 1.2 \tag{8}$$

$$S_{ij} = \frac{1}{2}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)$$

Where S= Pitch (mm) and Φ = Energy Dissipation from the Viscosity (W/m$^3$)

Using the SIMPLE algorithm with ($10^{-15}$) convergence to analyse the higher-order partial differential governing equations. Upwind scheme was used in this model. The initialization was selected as a hybrid for twenty iterations to save the initialization values used for running as shown in Figure 2.

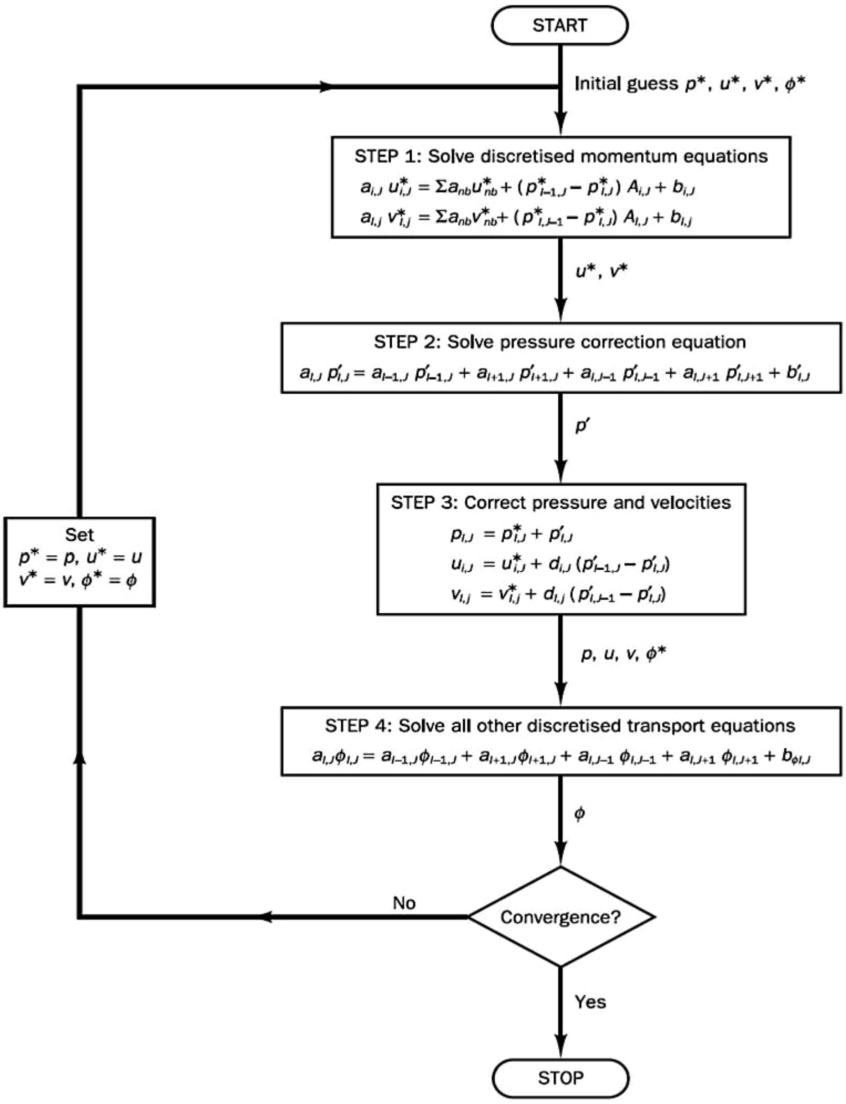



**Fig. 2.** SIMPLE algorithm flow chart [35]

*2.3 Boundary Conditions*

Steel circular pipes are used, dimples, and the working fluid is water as illustrated in Figure 3.

At the inlet, the temperature is 300 K, and the velocity inlet is 0.0200961731 m/s.

At the outlet, the backflow temperature is 300 K, pressure outlet.

At the wall, and dimples no-slip boundary condition, and the temperature is 400 K,

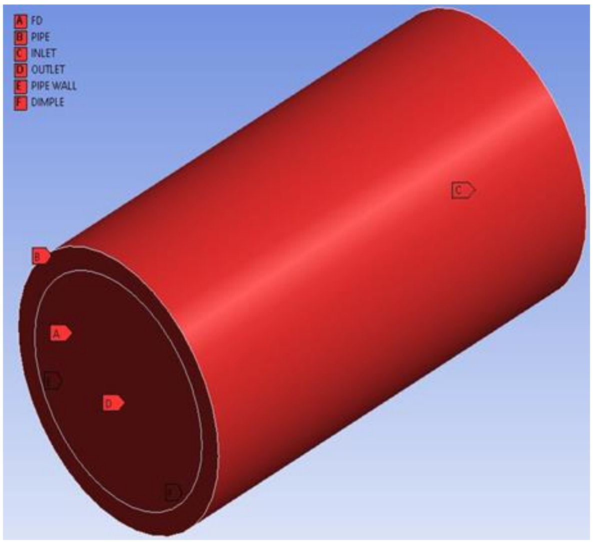



**Fig. 3.** Boundary condition

## 3. Validation

The computational fluid dynamic (CFD) analysis has utilised the ANSYS FLUENT 2022R2 software. A study involved analytical validation of the heat exchanger model. Using the Gnielinski correlation, the Nusselt number was verified, and the friction factor was certified using the Filonenko correlation as shown in Figure 4 and Figure 5. The current Nu values correspond closely to those derived from Gnielinski through ±6.7% as the maximum deviation. Similarly, the friction factor values align with those obtained from Filonenko *et al.,* [36], with ±7.0% as the maximum discrepancy. The suitability of the k-ϵ realisable turbulence model for the present study's calculation should also be recommended. This methodology is suitable for predicting the present investigation's heat transfer and flow characteristics. Gnielinski correlation

$$Nu_f = \frac{(f/8)(Re-1000)Pr_f}{1+12.7(f/8)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}}-1\right)}\left[1+\left(\frac{d}{L}\right)^{\frac{2}{3}}\right]C_1 \tag{9}$$

For liquid $C_1 = 0.01\,(\frac{Prf}{Prw})x$, If $\frac{Prf}{Prw} = (0.05\ to\ 20)$ , f = Friction Factor, Pr = Prandtl Number, L = Length (mm) and Re = Reynolds Number

Correlation by Filonenko gives:

$$f = (1.82logRe - 1.64)^{-2} \tag{10}$$

$$PEC = \left(\frac{Nu/Nu0}{(f/f0)}\right)^{1/3} \tag{11}$$

PEC (Performance Evaluation Criteria) is the ratio between heat transfers within improved pipes to the friction coefficient in the enhanced pipes, compared to that in regular pipes. This ratio exposes the significance of heat transfer about pressure drop in the heat exchangers and is a crucial factor in the heat transfer for the domain.

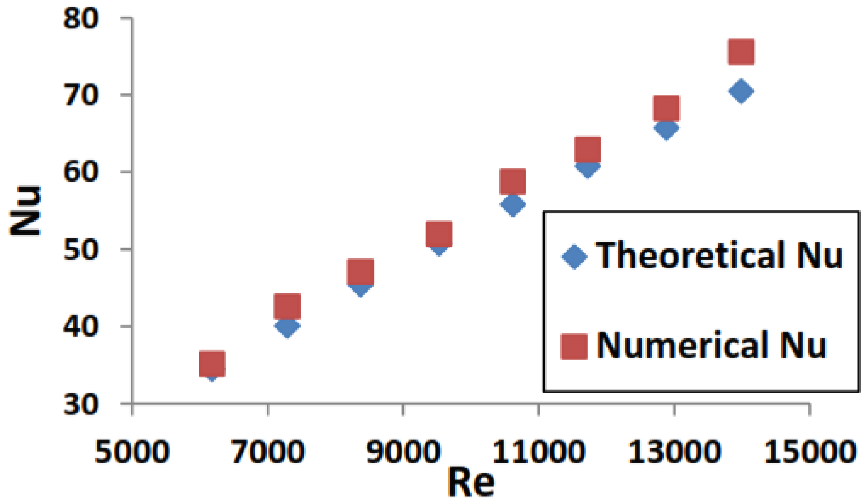



**Fig. 4.** Comparison of the numerical and theoretical Nusselt number vs Re

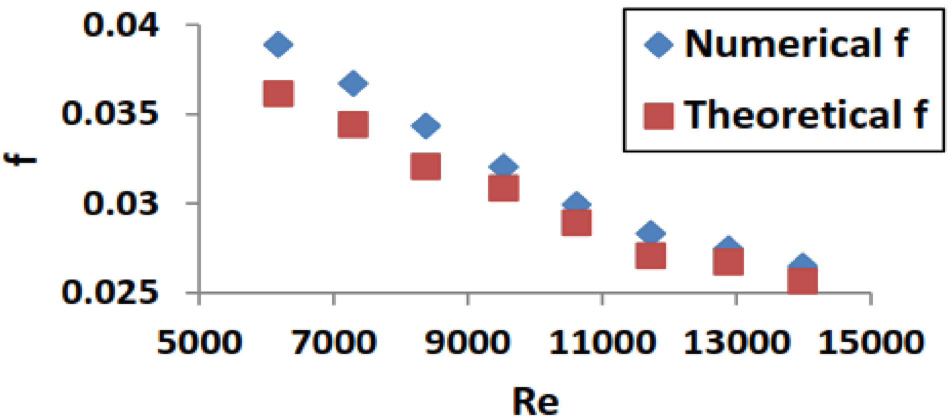



**Fig. 5.** Comparison of the numerical and theoretical friction factor vs Re

## 4. Mesh Independence

Hybrid meshes, combining hexahedral and tetrahedral elements, are universally employed to enhance computational accuracy across all scenarios. Specifically, tetrahedral meshes are utilized to capture the intricate features of dimples and protrusions and achieve superior mesh quality. To strike a balance between computation precision and resource demands, a grid independence test was conducted, as depicted in Figure 6 and Table 1. Mesh six demonstrates a difference of 0.73% and 0.8% in Nu and f values, respectively, compared to the reference mesh, indicating its suitability and reliability.

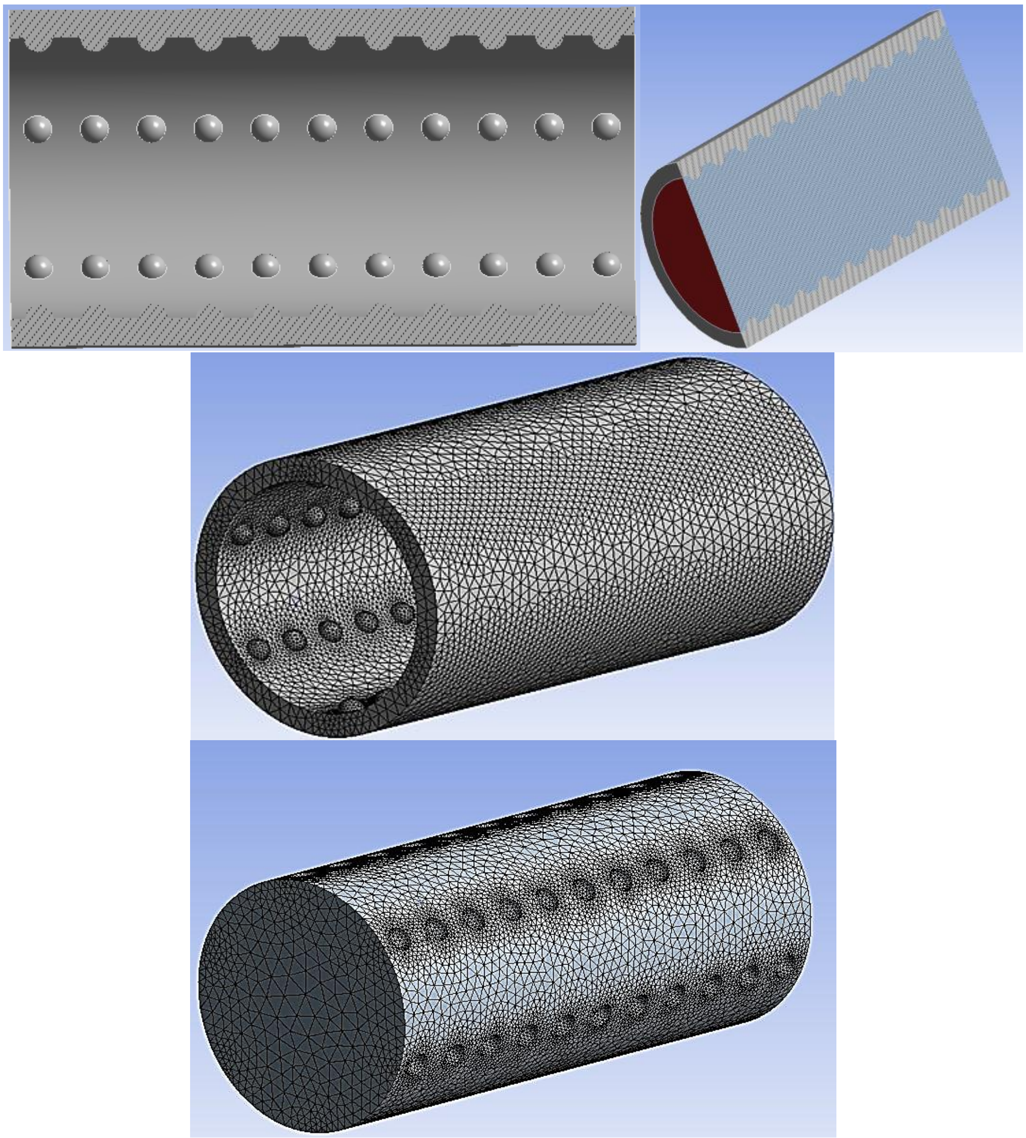


**Fig. 6.** Geometry sample of used in CFD with associated meshes

**Table 1**
Grid independents study

| Mesh | Number of element | Nu | Difference % | F | Difference % |
|---|---|---|---|---|---|
| Case 1 | 159730 | 78.33 | 9.27 | 0.0072 | 4.55 |
| Case 2 | 205443 | 79.98 | 7.35 | 0.0071 | 2.9 |
| Case 3 | 266863 | 85.94 | 0.44 | 0.0071 | 3.7 |
| Case 4 | 409202 | 86.35 | 0.02 | 0.0070 | 1.1 |
| Case 5 | 571402 | 86.96 | 0.73 | 0.0069 | 0.8 |
| Case 6 | 791200 | 86.33 | Reference | 0.0069 | Reference |

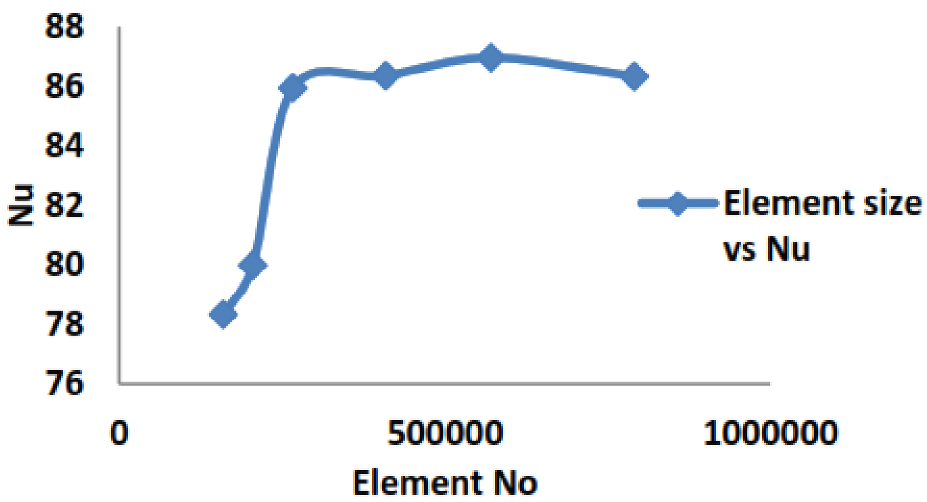



**Fig. 7.** Mesh independence

## 5. Result and discussion

*5.1 Comparisons between Smooth and Dimpled Tubes*

Figure 8 and Figure 9 shows the performance indicators of smooth and dimpled tubes at different diameters under the same condition. f represents the average friction factor of the dimple tube, and the methods for defining the Nusselt number are the same as f. It should be noted that all f index values are smaller than one, and Nu are more than 1. This means that heat transfer performance on all dimple pipes is enhanced compared to plain smooth pipes, but the flow resistance is more excellent. Considering the overall thermal-hydraulic performance, round tubes are better than soft tubes. Matched with a larger diameter dimple tube, while in Figure 10, the dimple's overall performance and heat transfer performance are remarkably and steadily enhanced; however, the hydraulic version is reduced. Dimples with larger diameters can react to flow added vigorously and have an additional healthy ability to extinguish boundary layer. In addition, the dimple tubes can reduce overlapping.

In addition, higher diameter raster tubes can reduce thermal resistance, with an increase in thermal performance of 18.03-38.6%, an average of 17.3%. Conversely, the performance indicators show a complex joining when comparing a 3mm dimple tube with a 7mm dimple tube ( d= diameter). About heat transfer, the Nu number of the dimple tube at (7 mm) is significantly larger than that on the other dimple tube of the different diameters at 5296 < Re < 8574 and 10000 < Re < 12888 otherwise, the Nu is faintly larger. For flow resistance, while Re is from 8574 to 12888, f for a dimple tube with a diameter of 7 mm is more significant than a dimple tube with 5 mm and 3 mm diameters. Compared with a smooth tube with less resistance at the same rate of Re, the hydraulic performance of a dimple tube increases with the increase in diameter. Generally, in the end, the PEC value of a 7 mm dimpled tube is always greater than that of a line of other diameters. In conclusion, the dimple tube has improved heat transfer and thermal-hydraulic performance.

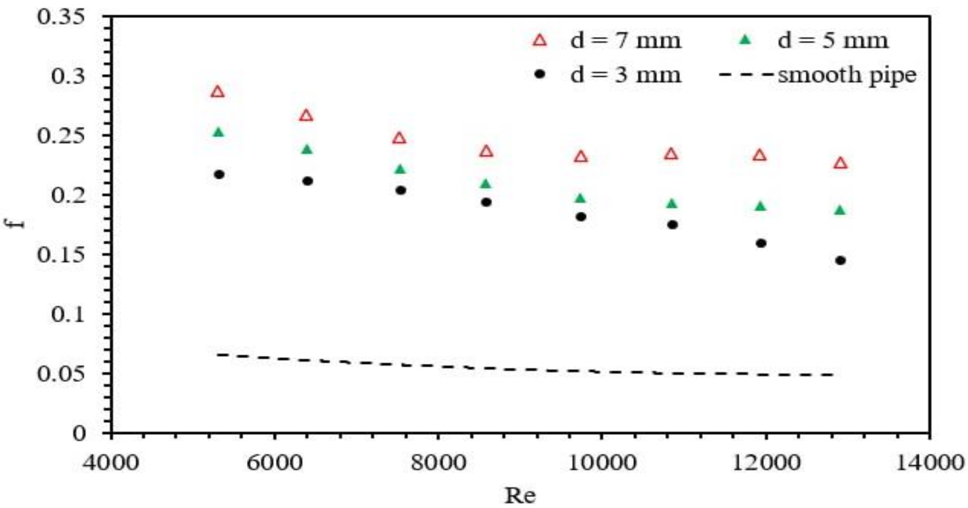



**Fig. 8.** Comparison of variation of friction factor with Reynolds number for various dimple diameters and plain tubes

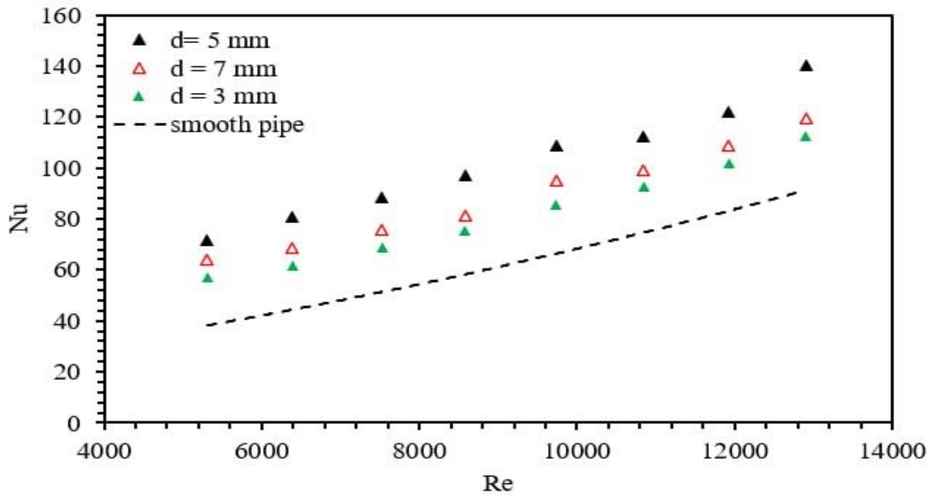



**Fig. 9.** Comparison of the variation of the Nusselt value with the Reynolds number for various dimple diameters and standard tubes

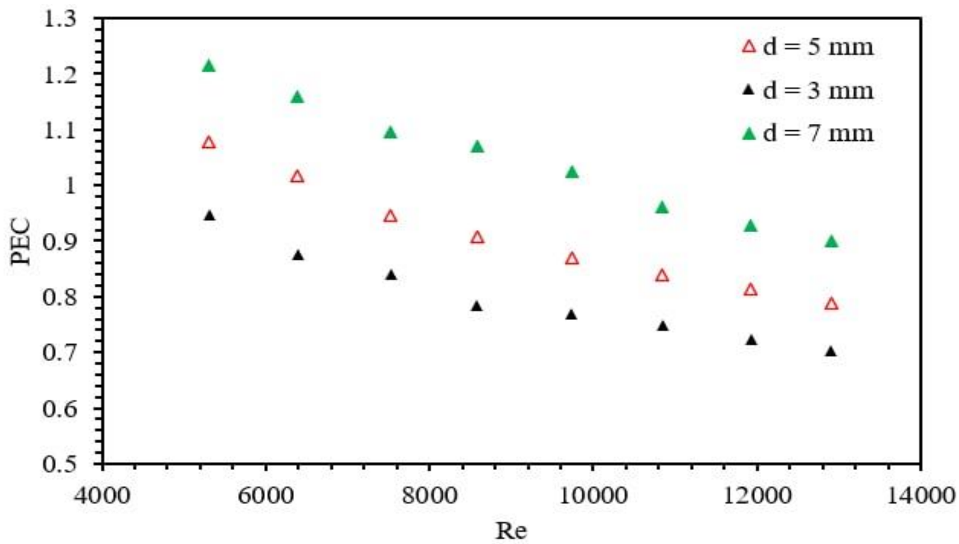



**Fig. 10.** Variation in the Reynolds number with varying dimple diameter in the Performance Evaluation Criteria (PEC)

From general knowledge of heat transfer, dimples can create one or other favourable situations to improve heat transfer: Evolution stops the boundary layer increasing turbulence, increases the effective heat transfer area, and generates cyclic and secondary flows.

Here, the focus is on an additional aspect, the friction factor and the Nusselt number, to achieve improved heat transfer in any design you should be sensitive to. Friction and heat resistance transmissions are the two vital factors in optimising piping. This paper mainly discusses Flow results and boundary conditions information regarding velocity distribution, temperature distribution and their effect on Nusselt number fracture modulus and performance.

*5.2 Thermal Performance*

Figure 10 compares the overall thermal performance characteristics that depend on the Reynolds number of smooth and dimpled tubes. It is clear that the Nusselt number increases with the increase of the Reynolds number, according to the equation

$$PEC = ({}^{Nu}/_{Nu_o}) / \left({}^{f}/_{f_o}\right)^{\frac{1}{3}}$$

We notice through the thermal performance in the presence of the dimple an increase in the heat transfer rate by increasing the thickness of the thermal boundary layer, thus giving us a higher temperature difference, which leads to an increase in the Nusselt Number and friction factor compared to the smooth tube.

*5.3 Friction and Heat Transfer Characteristics*

Figure 11(a) displays the temperature distribution along the axial cross-section of the section, revealing lower temperatures in the core compared to the areas near the walls. The temperature profiles depicted in Figure 11(b) illustrate the temperature variations in four distinct regions along the radial cross-section. Additionally, Figure 11(c) showcases the temperature gradient at z = 0.075 mm, z = 0.055 mm, z = 0.025 mm, and z = 0.095 mm after the dimple sector, highlighting the essential role of the junction displayed in Figure 8(a). These observations can be deduced from the temperature parameters presented in Figure 7. After passing through the dimple section, the temperature gradient in the core of the cross-section is relatively small, while the temperature difference near the wall is more pronounced. However, at z = 0.095 mm, the temperature gradient remains somewhat higher compared to z = 0.025 mm, z = 0.055 mm, and z = 0.075 mm due to the transition of fluid flow from laminar to turbulent inside the tube. In the dimple sector at z = 0 mm, the core temperature gradient is significantly higher than in the smooth region. The smooth cross-section between adjacent rows of dimples is referred to as a "slippery section" to distinguish it from the non-dimpled section. It is worth noting that the smooth cross-section at z = 0.025 mm exhibits the same temperature distribution as the pre-dimple cross-section at z = 0.095 mm, indicating that the presence of the dimples.

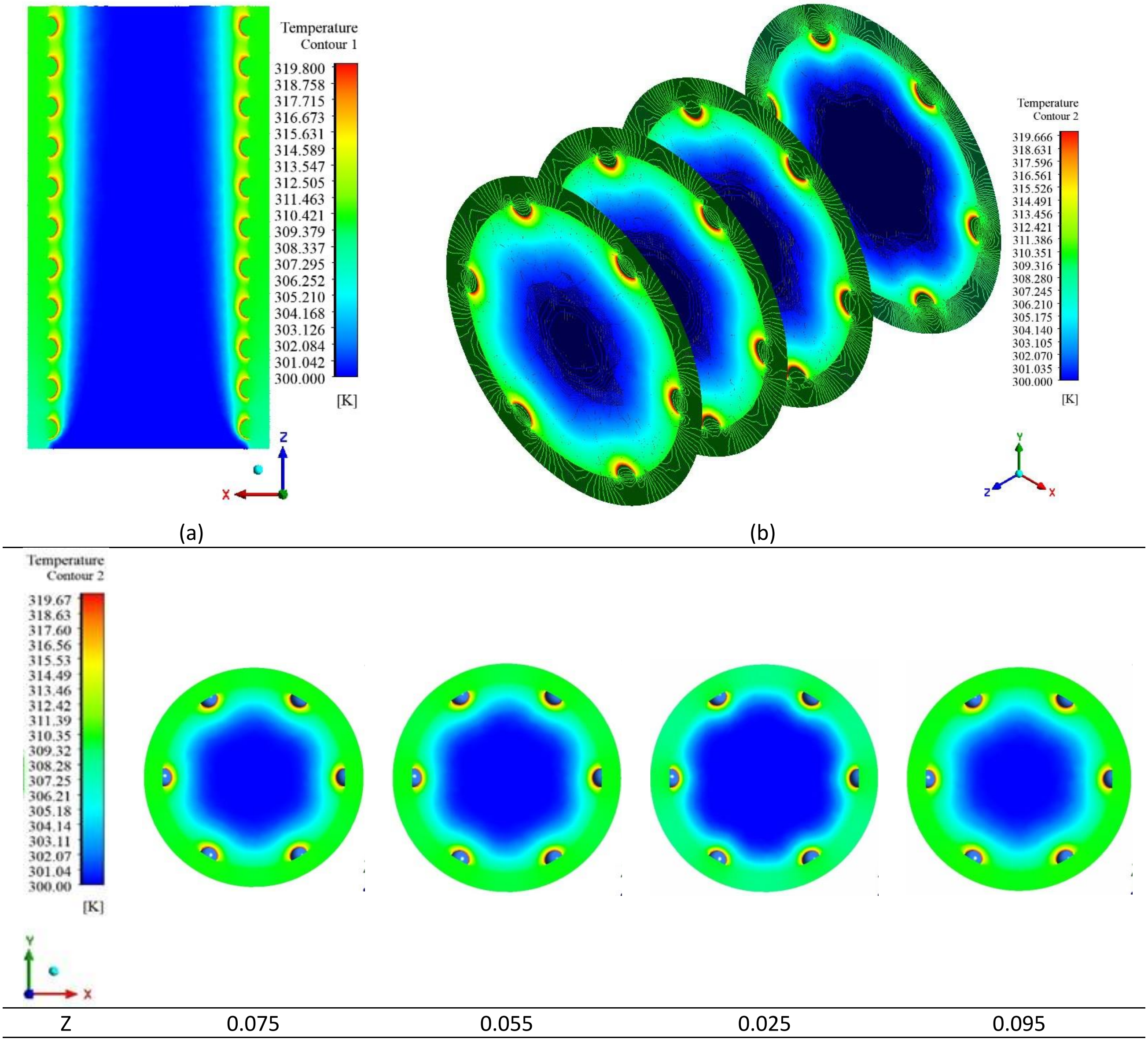


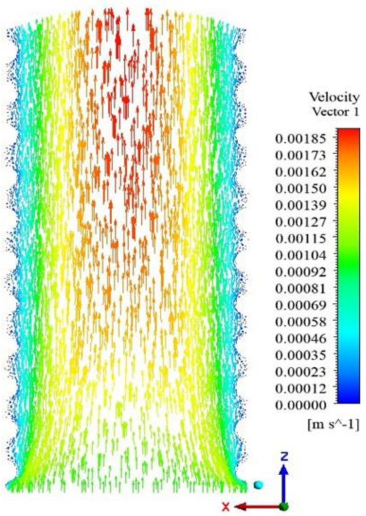

**Fig. 11.** (a) Axial distribution temperature, (b) circumferential distribution temperature contours at different sections and (c) Cross-section temperature contours at other areas

### *5.4 Heat Transfer Enhancement Mechanism*

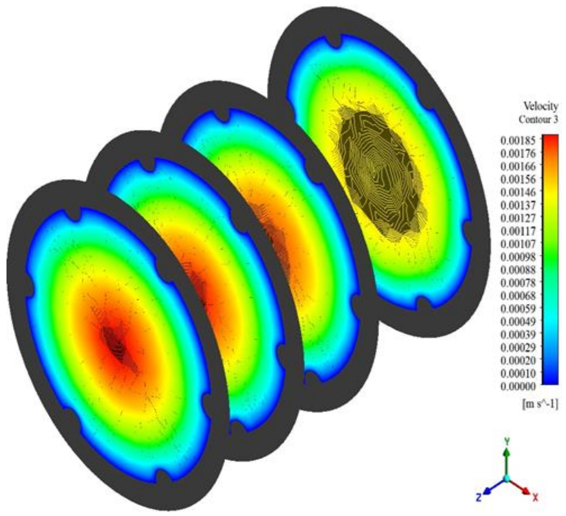
Heat transfer investigation focused only on the thermal and flow behaviours. Through the velocity vector in the x-z plane, we note that the velocity is zero at the wall in the presence of dimpled and begins to increase towards the centre, as illustrated in Figure 12(a). Here, we note that the velocity starts to increase from the beginning of the pipe and then turns into vortices due to the presence of dimpled generated impediment to the movement of the fluid and since the boundary layer resulting from the change of velocity in the x direction towards the centre and along the pipe in the z direction, and as the fluid advances, a boundary layer will be generated, and thus its thickness will increase.

As a result of the change in velocity and the boundary layer and its construction and growth along the length of the pipe towards the flow, which is the z-axis, the temperature distribution begins to increase from the wall, which is the highest temperature since there is a heat flux. Therefore, it will gradually heat the layers near it towards the centre. Here, it will begin to form a thickness thermal boundary layer where we find that dimpled reduces the thickness of the boundary layer compared to the pipe without dimpled because the dimpled is a tool used to increase the thickness of the boundary layer to push it towards the centre more. Therefore, the inference length is more significant than the flat pipe's. As a result, the boundary layer is either hydrodynamic or thermally more critical than its thickness at the flat line, as shown in Figure 12(b). At the same time, Figure 12(c) & (d) represent the temperature distribution in the xy plane through the pipe and fluid flow. These plans were taken at z = 0.075, 0.055, 0.025 and 0.095; it noted that the temperature distribution was less than exit because the dimpled had pushed the thermal and hydrodynamics boundary layer due to its increased thickness.

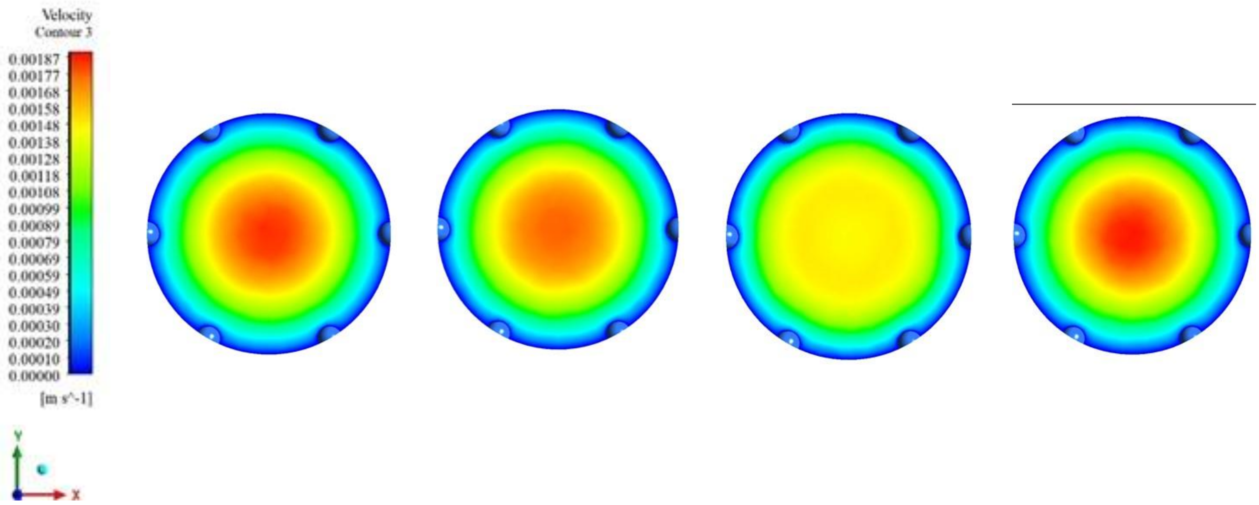
The dimples in Figure 7 and Figure 8, which show all the sections from the entrance to the exit, show that the thickness of the bounder layer increases gradually as it is directed towards the outlet section. At the same time, the speed had its maximum value in the centre because it was away from the side effects of the wall. This is clear in the velocity vector, where it will be noticed that the fluid flow has become vortices. As shown in the velocity contour, the velocity distribution towards the centre was due to the presence of dimpled and for the same reason mentioned earlier.



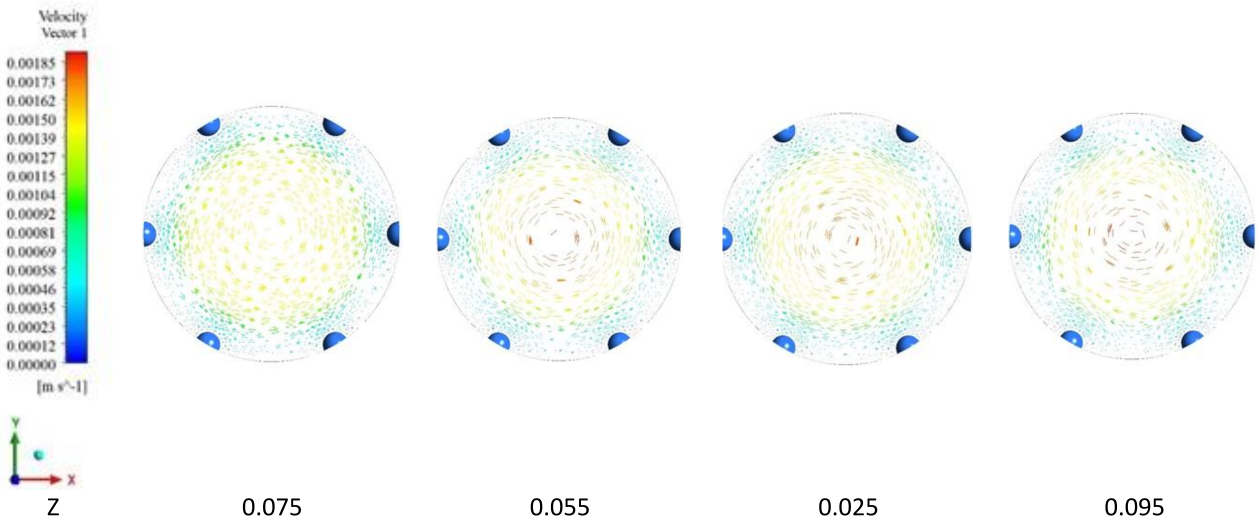

(a)




(b)




(c)




(d)

**Fig. 12.** Axial distribution velocity (a) circumferential distribution velocity contours at different sections (b) Cross-section velocity contours at different sections (c)Velocity counter at planes of z= 0.075,0.055, 0.025,0.095, and (d)Velocity vector at planes of z= 0.075,0.055, 0.025,0.095

## 6. Conclusion

CFD analysis showed a trend that dimples on the tube surface increase heat transfer. A few points were concluded.

i. The presence of dimples affected heat transfer by destroying both hydraulic and thermal boundary layer.
ii. Dimple tube increases heat transfer.
iii. The inclusion of dimples on a tube's surface has a significant impact on the temperature distribution throughout its cross-section. In particular, the dimple section's core temperature gradient is substantially more significant than the smooth section. However, the dimple section variations are more minor than those observed in the non-dimpled region. This phenomenon underscores the enhanced heat transfer efficiency brought about by the presence of dimples, leading to improved overall performance in heat transfer.
iv. The coefficient of friction for dimpled pipes is more significant than that for smooth pipes.
v. The thermal performance of dimpled pipes is higher than that of smooth pipes by (17.3%)
vi. Thermal performance varies proportional to the dimple diameter.

The future scope for enhancing heat transfer for flow through a dimpled pipe numerically involves optimizing dimple geometry, studying different flow conditions and fluids, validating numerical simulations with experiments, and conducting application-specific studies. These efforts can contribute to the development of more efficient heat transfer systems and aid in the design and optimization of dimpled pipe configurations for various industrial applications.

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