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Research Article

Influence of different geometrical dimple configurations on flow behaviour and thermal performance within a 3D circular pipe

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ABSTRACT

Numerical analysis has been performed to evaluate the heat transfer characteristics and performance of a circular pipe with geometrical dimple patterns. Using computational fluid dynamics (CFD) codes, we examine the effects of geometrical configurations on the flow and thermal behavior of circular pipes with concavity (dimple) diameters. Fluid mixing and flow perturbation are facilitated by perforations across the pipe core and wall regions, thereby improving thermal efficiency. In addition, a concavity with a diameter of 4 mm enhances heat transfer. Based on the results of the study, the disrupted pipe wall and pipe core region produce swirls and transverse vortices in the flow that provide superior heat transfer compared to conventional (smooth) pipes. In an increasing Reynolds number (Re), mixing, secondary, and separation flows become larger. Performance evaluation factor (PEF) values increased at low Reynolds numbers when dimple diameter was 1mm. As a result of these improved pipes, heat exchanger efficiency may improve in industrial applications, a key factor for energy conservation.

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INTRODUCTION

Various industrial applications of heat exchangers, such as chemical processing, reactors, and power plants use geometrical dimple configurations to improve thermal performance through reduction of energy consumption and costs, improve the efficiency of the heat exchanger by using two types of solutions, i.e., passive techniques [1]. In the latter approach, expanded surfaces or roughening surfaces are provided utilizing various configurations such as grooves and ribs [2-5]. In recent decades, research has been conducted on enhancing the thermal efficiency of heat exchangers by employing twisted tape and corrugated designs. For example, Naphon et al. [6] in research on the increase of heat transfer with twin tubes with helical ribs used in an experimental investigation. The height to diameter (h/d) ratios of helical ribs was used, including 0.12, 0.15, and 0.19, as well as varied rib pitch to diameter ratios

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including 1.05, 0.78, and 0.63. In addition to examining the impacts of the corrugated pipes with various pitches of corrugation such as 5.08, 6.35, and 8.46 mm. Laohalertdecha and Wongwises [7] also examined the impacts of corrugated tubes made of aluminum. The corrugated depth remained constant at 15 mm regardless of Reynolds numbers ranging from 6000 to 27,000. In comparison to smooth tubes, they concluded that corrugated tubes have heat transfer coefficient ratios of up to 50 and 70 %. The experimental results from Pethkool et al. [8] showed that helically corrugated pipes showed improved heat transfer and friction metrics. Various corrugated pitches were used, with diameters ranging from 0.18 mm to 0.27 mm, as well as various corrugated tube rib heights, with diameters ranging from 0.02, 0.04, and 0.06. According to these researchers, heat transfer increased by 232% compared to smooth tubes, and the friction factor and Nusselt number were respectively 2.14 and 3.0 times higher. Using several geometries of the dimpled tube in turbulent flow, Chen et al. [9] studied the friction factor, coefficient, and enhancement of the heat transfer providing reliable correlations for the pressure drop and heat transfer. As Reynolds number was maintained constant, the heat transfer rate was increased from 25 % to 137 % while as steady power of pumping increased from 15 % to 84 %. Researchers found that reducing the heat exchanger weight and size by almost a factor of two may make the heat exchanger more compact and lighter without affecting other system variables, and a higher heat transfer coefficient can result from using dimpled surfaces. Dong et al. [10] presented another experiment for evaluation of the heat transfer properties and drop of the pressure of several corrugated tubes, with test-performed runs at Re ranging from 6000 to 93000 for water and Re ranging from 3200 to 19000 for oil. They cla10.18186/thermal.1429444imed to have improved the efficiency and friction factor of heat trans-fer by up 160 %. Heat transfer was improved by the cor-rugated ribs, according to the findings. Furthermore, their findings demonstrate that heat exchanger performance is affected by Re, Pr, and corrugation shape. Use of corru-gated tubes with a constant corrugation ratio and surface temperature. Elshafei et al. [11] investigated the corru-gated tube's heat conduction and pressure drop properties. When compared to smooth tubes, the findings showed a considerable improvement in heat transfer i.e., by a factor of up to 3.2 and a drop of pressure by 2.6. Furthermore, as channel spacing grew, the friction factor rose. They found that the corrugated tube has a spacing ratio of less than 3.0 and a phase shift of fewer than 90º will increase the heat transfer performance. According to Harleb et al. [12], an experimental investigation of 18 corrugated tubes for heat exchangers was performed, ranging from 0.283 to 1.117, and 0.024 to 0.087 in height, as well as corrugation angles ranging from 14.7° to 48.8° and Re numbers ranging from 500 to 23000. In comparison to smooth tubes, the results revealed that the thermal performance was improved. Numerical analyses were performed by Wang et al. [13] to explore the increase in thermal conductivity for different corrugated tubes with varying Re ranging from 3800 to 43800. They found that rotating flow on heat transfer improved by 1.4 and Nu improved by 1.77. Simparov et al. [14] investigated heat transfer improvements for various spirally corrugated tubes ranging from 0.44 to 1.18 mm in pitch heights and 6.5 to 16.9 mm in corrugation pitches and Re of 104 to 6,104. In comparison to smooth tubes, the thermal performance improved by 177−273 %, while the factor of friction increased by 400%. In experimental research on heat transfer and flow properties. The heat exchanger was fitted with louvered strips to create turbulent flow, as demonstrated by Eiamsa-ard et al. [15]. There is a wide range of Reynolds numbers for the fluid used in this process, ranging from 6,000 to 42,000. The findings confirmed that the louvred strip provides a heat transfer of 285% in comparison to the plain tube. Numerical study for the impacts of various parameters of the corrugated tube was performed by Al-Obaidi and Alhamid [16], and as working fluid, water is utilized with Re ranging from 4000 to 12,000 under conditions of turbulent flow and found that Nusselt number and friction factor are in direct proportion to the ring angle of the corrugated tube. In a simulation of embedded conical strips in helical tubes, Fan et al. [17] found that the friction factor and Nusselt number Nu were 10 times and 5 times better than smooth tubes, respectively, with a value of 2.6 times higher than smooth tubes. Subasi et al., Dang et al., and Azmi et al. [18-20] investigated the type of twined coil with type inserts. Heat transfer performance and pressure drop in the pipe can be improved by combining both techniques,

This work investigates the combination of dimple, corrugated and twisted tape topologies for improving the thermal efficiency and heat transfer of heat exchanger systems. Thermal flow in customized pipes is also examined in terms of how it affects heat transfer improvements. A heat exchanger pipe with various geometrical designs fitted with different dimple diameters is subjected to numerical analysis to determine the characteristics of flow, heat performance, and Nusselt number (Nu) friction factor (f). The calculations are motivated by the above analysis. By using 3D models, it will be possible to design heat exchangers that offer better thermal performance.

PHYSICAL PIPE MODEL DESCRIPTION

ANSYS CFD was used for the development of the fluid flow domain. There were two types of pipes that were selected: plain pipes and modified pipes. Simulations have been conducted for four different cases of modified pipes. Figure 1 illustrates the dimensions of the concavity diameters of 1, 2, 3, and 4 mm as well as the dimensions of the twisted tape. It has the same pipe length as shown in figure 1. Using a wide range of Reynolds numbers i.e., 1000 to 15000. The fluid at Re = 1000 is laminar, and the fluid at Re higher than 2300 is turbulent flow. The modifications are analysed with the expectations of improved heat transfer in the pipes due to swirl flow and increased level of mixing. By solving the flow boundary conditions, they are imposed for modified pipe and smooth pipe.

In non-slip conditions, the inlet temperature of 293 K is matched to the intake velocity for hydrodynamic boundary conditions velocity input and pressure output. The results are applicable to smooth pipes and pipes of various configurations. Computational calculations are simulated with the wall pipe at constant heat flux. The round 3D smooth pipe is used as a comparison foundation model. Furthermore, the experimental data were collected comprised the temperatures at the inlet and outlet of the pipe and the pressure difference through the tube. The pressure data were got at the pipe inlet and outlet by using a manometer.

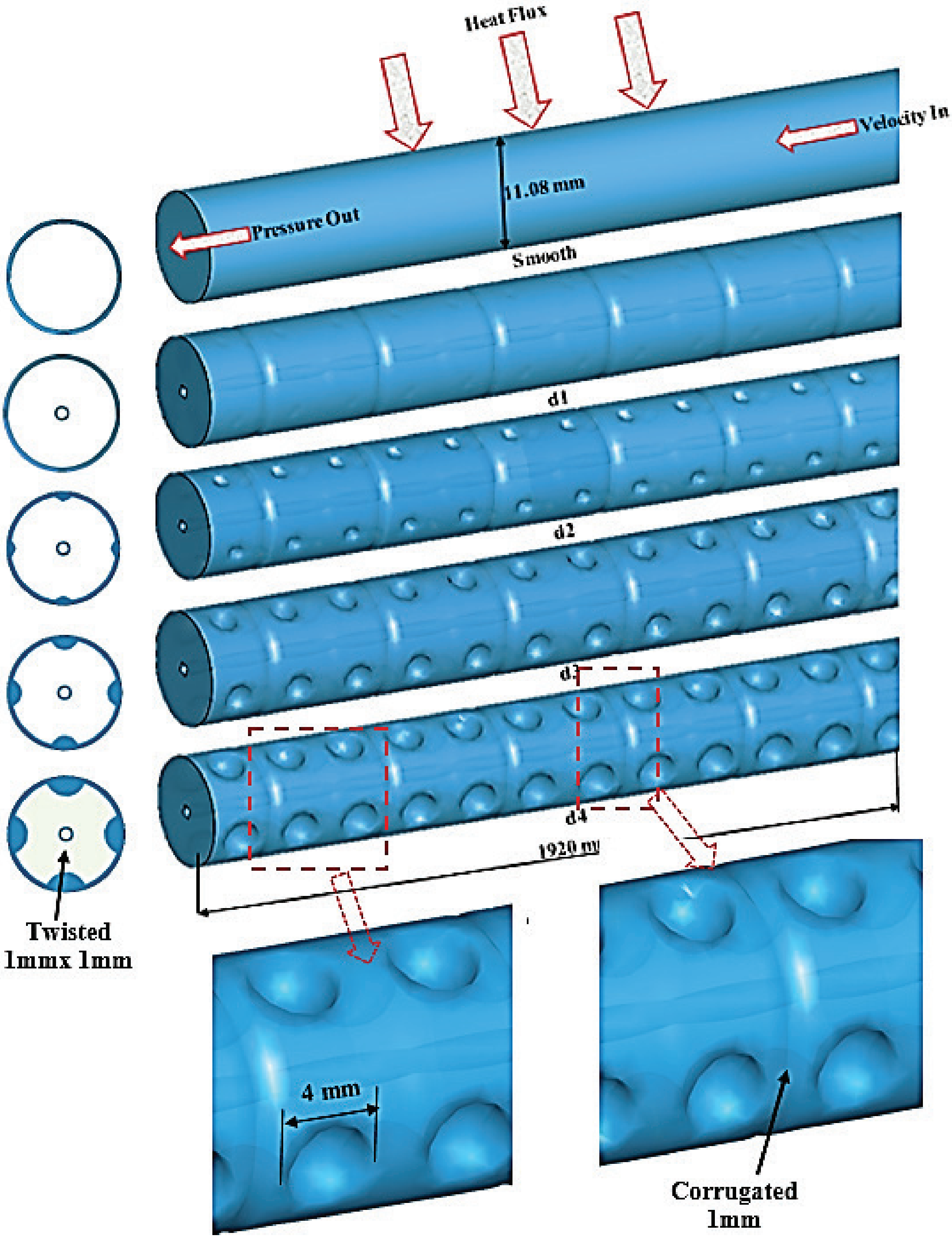


Figure 1. The schematic representation of the 3D domain of plain and modified pipes.

Grid Generation Checking

The flow domain numerical modeling is conducted for both the modified and plain pipes, as illustrated in Figure 2. Elements of tetrahedral mesh with different face widths of mesh cells, such as 1mm, 2mm, and 3mm, were created. The layers of inflation are applied for the border in this mesh generation to check the sensitivity of the mesh. The tests of cell independence are conducted for smooth and dimple pipes. Varying of the cell is done for both friction factor and Nusselt number parameters. Table 1 shows the

Table 1. Examination of cell independency for both pipes

No. of cells	Smooth pipe		No. of cells	Dimple pipe	
	Nu	f		Nu	f
	(%)	(%)		(%)	(%)
514353	4.2	6.3	635571	5.5	6.8
812682	4.1	5.5	996270	4.9	5.6
1991742	3.7	3.8	2022488	3.6	3.7
2588682	2.5	2.6	2855662	2.7	2.6
3046570	1.9	2.3	3122432	2.2	2.3

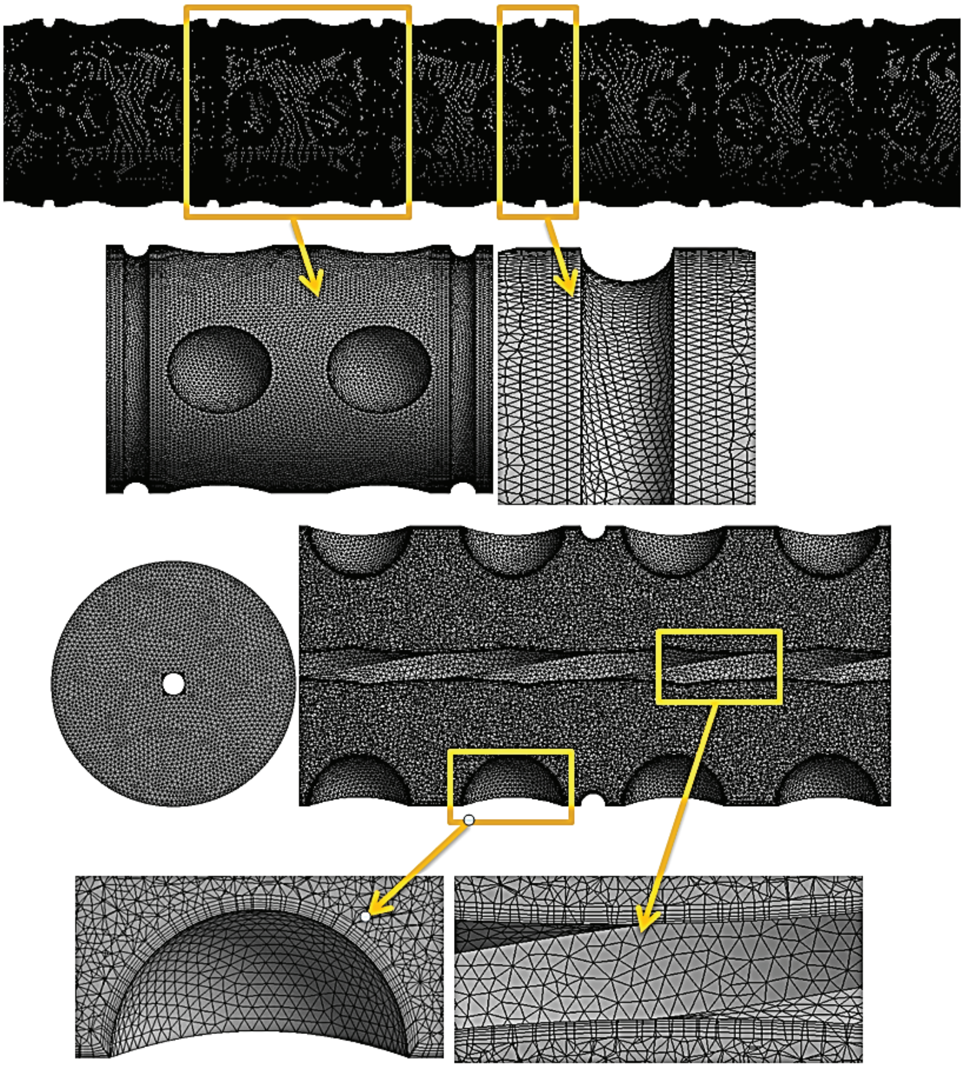


Figure 2. Pipe grid generation in various geometrical configurations.

values of percentage errors. As noted that good compatibil-ity as compared with experimental results with an average deviation was around 2.4% for Nu and 3.7 % for f. Also, the ten layers of inflation are selected near the pipe walls with the value of y+ was lower than 5. The boundary conditions are the velocity and pressure at the inlet and outlet regions.

Numerical Procedure

ANSYS FLUENT (CFD) software was used to study the effects of different configurations on hydraulic, thermal flow, and thermal performance. This system experiences a pressure drop when an average mass-weighted pressure is applied at its entrance. Using the turbulent model type Shear Stress Transport (SST), the Navier-Stokes equa-tion and the energy equation is solved and modeled. To solve and calculate the equations in this section, ANSYS FLUENT 19 computational fluid dynamics (CFD) software is used. This pipe model was evaluated in three dimensions and in steady-state (since it operates horizontally). In con-junction with pressure velocity, coupling, numerical simu-lations utilize the Semi Implicit Pressure Linked Equations (SIMPLE). To study the variables within each cell in the flow domain, 2nd order upwinding is applied to the solu-tion of the energy equations continuity and momentum. Numerical calculations can be performed when the energy and continuity residues exceed 10^{-6}.

Governing Equations

With respect to the simulations, the governing equa-tions were incompressible, turbulent, in the Cartesian tensor using Naiver stokes equations, neglecting viscous dissipation and gravity, with fluid thermal properties con-sidered constant in the simulations.

Equations are formulated as follows [21] based on the above assumptions:

Continuity equation:

$$\frac{\partial}{\partial x_i}(u_i) = 0 \tag{1}$$

Momentum equation:

$$\frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_i}{\partial x_j}\right)\right] + \frac{\partial}{\partial x_j}(-\rho\tau_{ij}) \tag{2}$$

Energy equation:

$$\frac{\partial}{\partial x_j}(u_i(\rho E + p)) = \frac{\partial}{\partial x_j}\left[\left(\lambda + \frac{C_P\mu_t}{Pr_t}\right)\frac{\partial T}{\partial x_j} + u_i(\tau_{ij})_{eff}\right] \tag{3}$$

Here
ρ = fluid density,
E = total energy,
r_t= turbulent Prandtl number,
Stress tensor = $u_i(\tau_{ij})_{eff}$

$$E = C_P T - \left(\frac{P}{\rho}\right) + \frac{u^2}{2} \tag{4}$$

$$(\tau_{ij})_{eff} = \left[\mu_{eff}\left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j}\right) - \frac{2}{3}\mu_{eff}\frac{\partial u_i}{\partial x_j}\delta_{ij}\right] \tag{5}$$

The turbulent dissipation (X) and Kinetic Energy (k) is given as

$$\frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left(\Gamma_k\frac{\partial k}{\partial x_j}\right) + \widehat{G_k} - Y_k + S_k \tag{6}$$

$$\frac{\partial}{\partial x_j}(\rho\omega u_j) = \frac{\partial}{\partial x_j}\left(\Gamma_\omega\frac{\partial \omega}{\partial x_j}\right) + G_\omega - Y_\omega + D_\omega + S_\omega \tag{7}$$

G_k and G_ω indicate turbulence kinetic power produc-tion of k and ω, respectively, Y_k and Y_ω reflect turbulence dissipation of k and, and S_k kand S_ω are source terms of k and ω respectively. Furthermore, the constants are. For the inner pipe, the average Nu number is calculated as follows.

$$Nu_{av} = \frac{hD}{\mathrm{k}} \tag{8}$$

Here
k= Thermal conductivity
h = Average coefficient of heat transfer
The Re (Reynold Number) is given as

$$R_e = \frac{\rho D_h u_m}{\mu} \tag{9}$$

Here
u_m = Mean fluid velocity,
D = Tube Diameter,

$$\mathrm{D} = 4\mathrm{A}/P \tag{10}$$

$\mathrm{P_h}$ = wetted perimeter,
A = cross-sectional area.
The factor of friction is given as

$$f = \frac{2}{(\frac{L}{D})}\frac{\Delta_P}{\rho u_m^2} \tag{11}$$

Here
ΔP = Pressure loss
u_m= Mean velocity

$$\Delta\mathrm{P} = P_{av,inlet} - P_{av,outlet} \tag{12}$$

Here
$P_{av,inlet}$ = Inlet average pressure

$P_{av,outlet}$ = Outlet average pressure

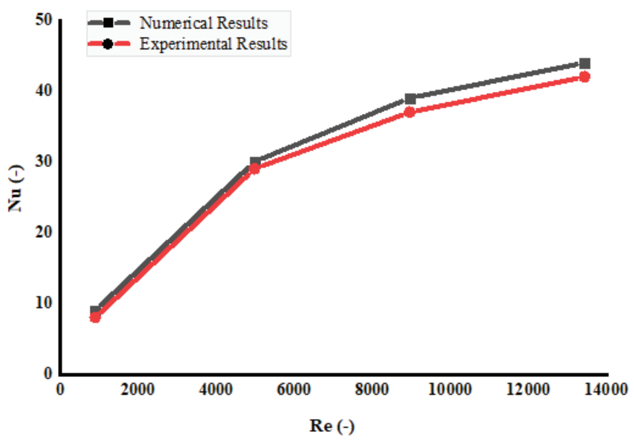
PEC was developed to measure heat transfer performance and defined as follows:

$$\text{PEC} = \frac{N_u/N_{u_o}}{(f/f_o)^{1/3}} \tag{13}$$

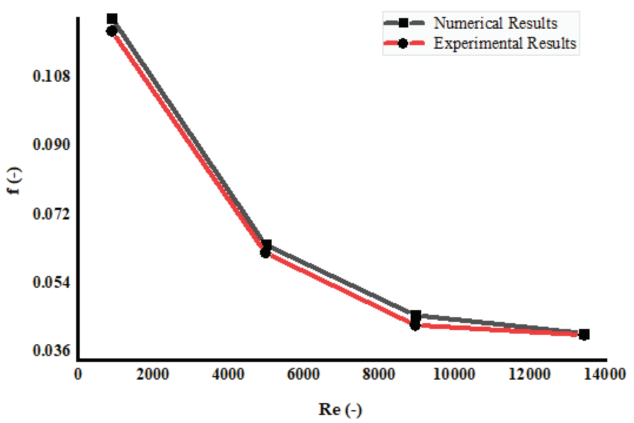
The performance of heat transfer of a fluid is given by Nu (Nusselt number). The fluid's pressure losses are given by f (friction factor), whereas Nu0 and f0 represent the Nusselt value and factor of friction, respectively, under reference parameters [23-24].

Validation

A comparison of Nu values are made with experimental from Albanesi et al. [22], the experimental carried out a test for various dimensions tubes with arrangements to clarify and quantify the heat exchanger performance to study the effect of dimpling tube. as illustrated in figure 3. The average deviation outcomes between Albanesi et al [22] results and the current results are determined for Nu number is 4.5 % ad for friction factor (f) is 3.2. This numerical model is validated based on these findings.



Nu number



Friction factor

Figure 3. Nu number and frication factor validation outcomes comparing with experimental data.

RESULTS AND DISCUSSION

Thermal Flow Analysis

A comparison of different configurations of turbulent viscosity is shown in Figure 4. As noted, the turbulent viscosity intensity is generally higher for the dimple diameter of 4 mm model as compared to the smooth pipe and other tested geometry models as a result of flow perturbation and shear stress. As a consequence, the dimple diameter of a pipe with a higher diameter causes a greater thermal boundary layer disruption than a pipe with a smaller dimple diameter. This is due to higher turbulent intensities in the pipe and further disturbances generated nearby. As a result of the distribution of fluid flow through the pipe using different geometrical parameters, the pipe wall may reattach to the pipe, causing disturbance and mixing of the flow in the pipe core areas, disrupting the thermal boundary layer. Additionally, this type of flow perturbation can significantly improve the fluid flow turbulence intensity and heat transfer.

Hydraulic Flow and Thermal Performance with Various Working Conditions

A comparison is made of the hydraulic and thermal performances under different Re number conditions to show the effects of the geometric parameters. The figure 5 represents the velocity flow streamlines in pipes when flow direction changes due to obstructions in the form of geometric configurations under various Re numbers. When the Re is increased to 15000, the flow streamlines remain approximately the same. In contrast, vortex-like mixing flows formed near dimples and corrugated shapes, resulting in a significant rotation. Hence, the flow resistances caused through the geometrical confirmation are higher than those caused in the smooth pipe.

Generally, these types of geometrical parameters can change the temperature and field flow structure within the pipe. Hence, the heat performance can enhance by the use of latter passive devices. However, the swirl flow and transverse vortices created are quite strong since the wall surfaces of the pipe core are disturbed. Hence, the improvement of heat transfer of the pipes is as good as the conventional pipe (smooth). However, the resistances flow caused by these devices formed a higher drop in pressure more than the smooth pipe.

Figure 6a shows the distribution in velocity near the pipe wall (dimple side) for various Re numbers. Comparing these figures, the influence of configurations can cause more mixing flow and the vortex. Moreover, the mixing flow extends, the secondary flow becomes larger, and the

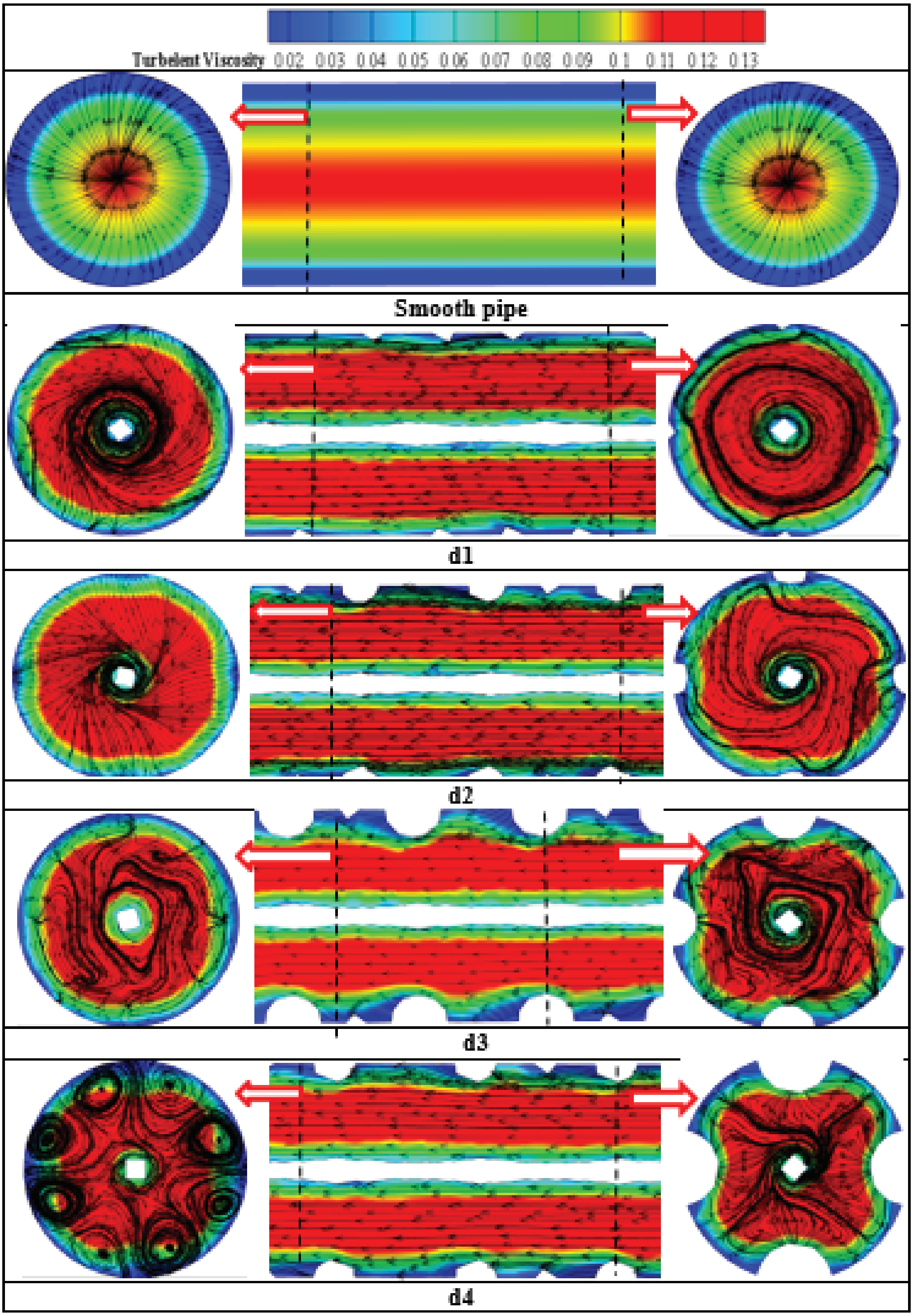


Figure 4. Turbulent viscosity variations with different geometrical parameters.

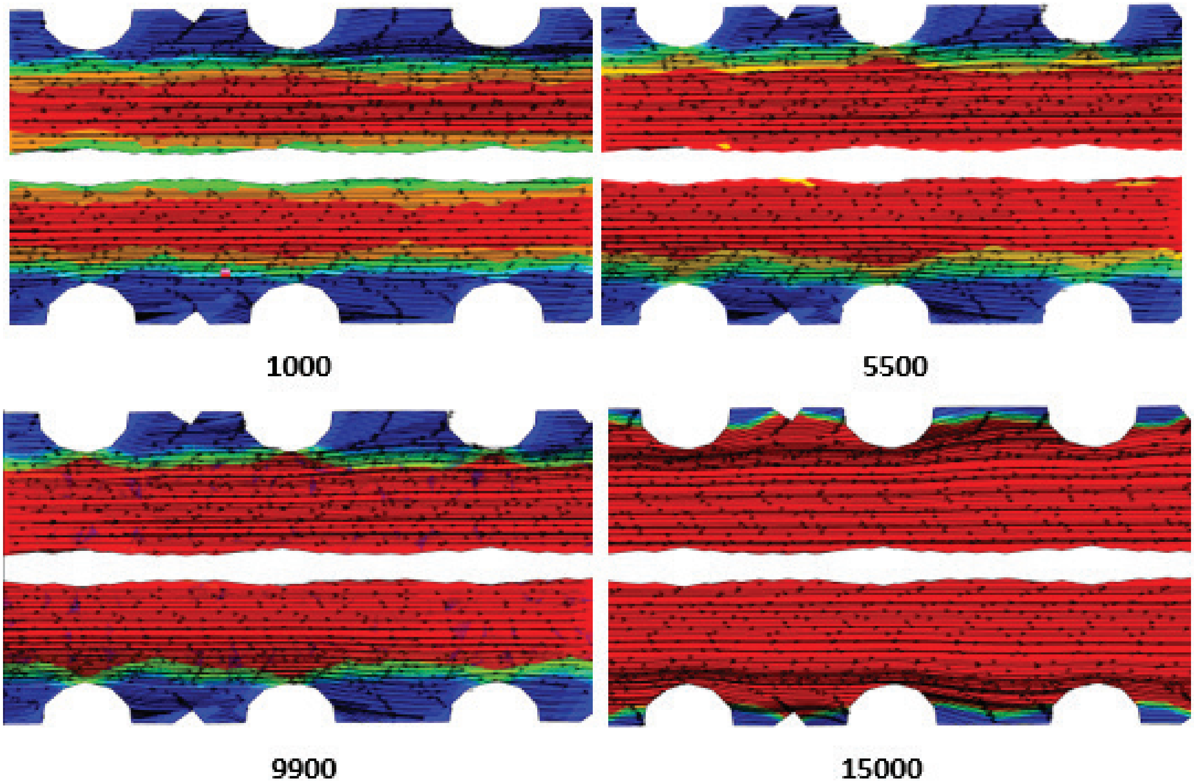


Figure 5. Pipe velocity streamlines with different geometrical configurations under various Re numbers.

separation flows narrowly as the Re is raised, increasing the distance between the reattachment and separation points. However, the flow field structure remains uniform for different Re values representing the development of the mixing and vortex pattern is high based on the Re number at the downstream and upstream of the pipe. The contour of temperature disruptions at the pipe wall for different Re is depicted in Figure 6 b. Comparing these figures, the thermal boundary layer thickness changes as the Re number increases, whether the flow field also changes downstream or upstream. Therefore, these geometrical configurations can affect flow direction, mixing, and thermal resistance, leading to enhanced heat performance. Figure 6 c Turbulent kinetic energy (TKE) distribution, the increase in the TKE is more observable as the Re increase at the downstream and the upstream. The distribution in TKE and the numerical outcomes almost change at dimple positioned at the pipe. Thus, at these locations, the Re stress can considerably increase near the pipe wall surfaces. The fluid parameters associated with the performance of heat transfer increase. This is connected to the resistance performance increases.

To investigate the various thermo-hydraulic performances mechanism caused by the use of different Re numbers with other geometrical parameters, the flow pattern and distribution of temperature in the pipe are analysed. The pipe symmetry plane with three transverse planes (before the corrugated, corrugated, and dimple sections) in the pipe entry section is observed. The viewing plane surfaces are represented in Figure 7.

Figure 8 shows that due to the geometrical configuration disturbance, pairs of counter-mixing flow and rotating vortices are produced, particularly at dimples, twisted tape, and corrugated surfaces. The magnitude of the vortices and the area affected by them increased as Re increased. Further, the flow repeated and generated the previous cycle with flow changes in the position and magnitude of the vortices. The temperature field contour plots in these planes in the pipe with various Re numbers are depicted in Figure 9. As observed, the streamlines of contacting the flow in the transverse flow planes, the vortices flow shaped a transverse mixture flow between the flow in the core area and pipe wall, especially near n the wall surface regions. Hence, the thermal boundary layers for fluid are changed at the flow disturbance. Compared to these figures, the temperature field contours are changed, leading to improved efficiency of heat transfer in the pipe with different geometrical configurations. However, due to the surge in the Reynold number (Re) and constraints of the various geometrical shapes, the hydraulic and thermal performances showed differences.

Figure 10 compares the friction factors (f) and Nusselt numbers (Nu) of various pipes versus the Reynolds numbers (Re). In all pipes, the tendency of f curves is the same, and it decreases with Re due to an increase in gradient

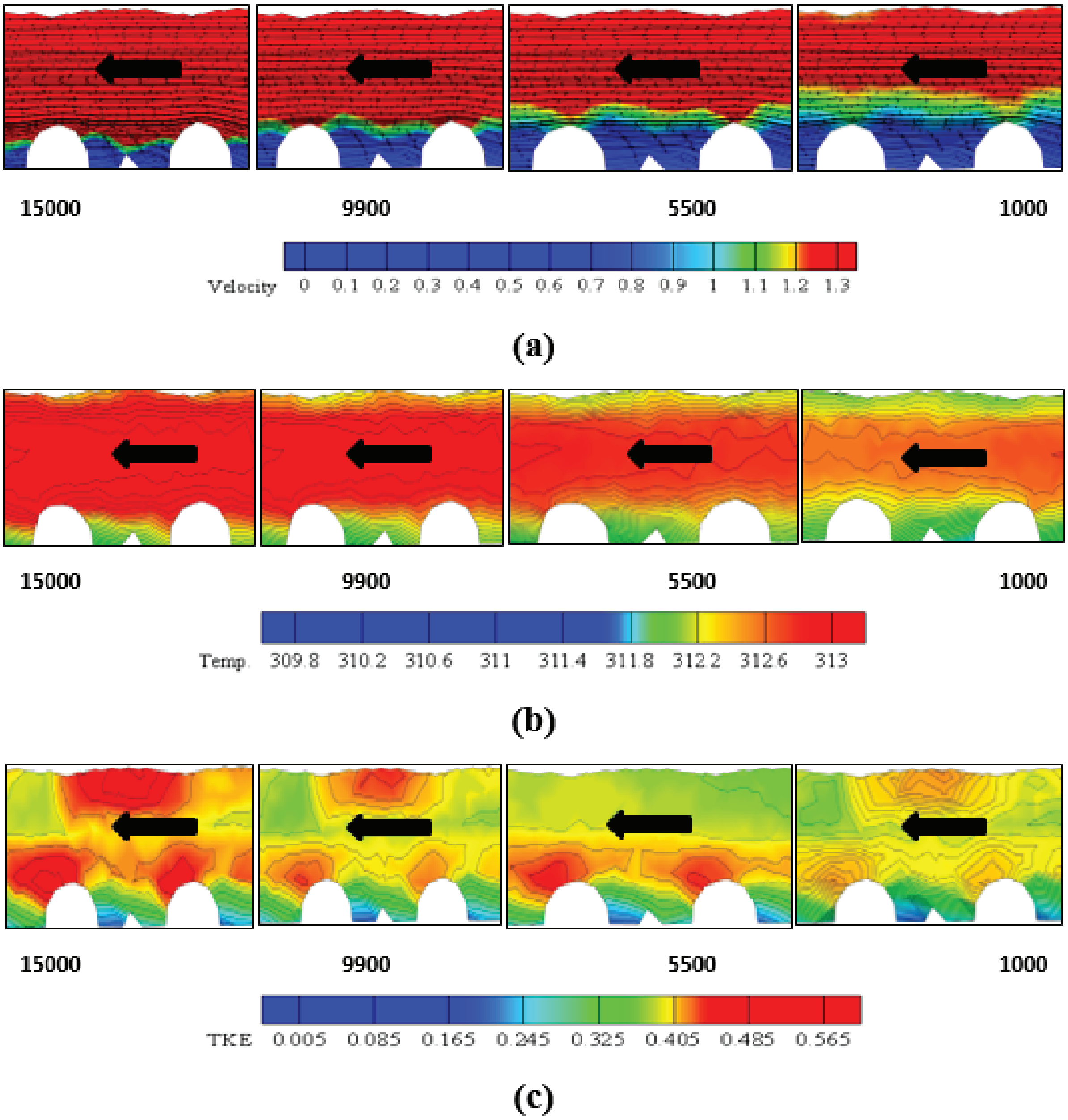


Figure 6. (a) Velocity, (b) temperature, and (c) TKE Distributions under different conditions.

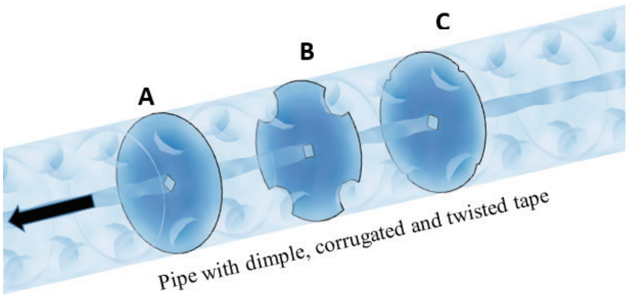


Figure 7. Three different planes (A-C) form a center symmetry plane.

pressure. The shear stress increases because of an increase in the dimple diameter that leads to a rise in the pressure drop in the pipe. Based on the similarity of Nu curves across all pipes, Figure 11 shows changes in Nu number as a bars graph, and it increases roughly linearly with Re as dimple diameters increase. Despite improved heat pipe performance with various configuration options in comparison to smooth pipe, the heat transfer study indicates.

Evaluation of the Overall Thermal Performance

A greater contribution to total heat performance is made by enhancing heat performance by modifying devices

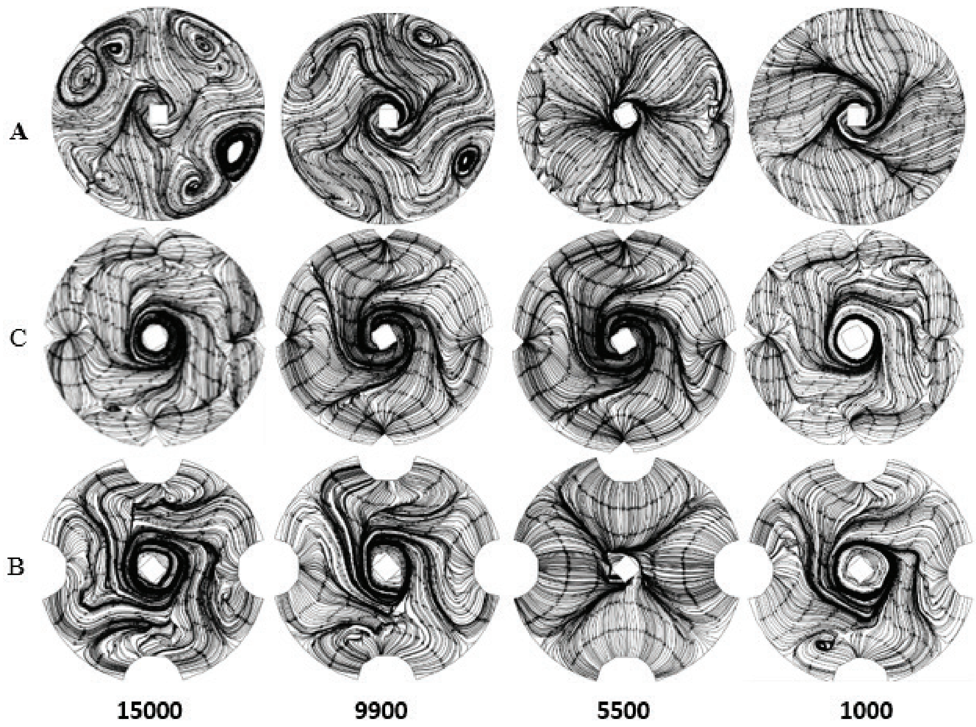


Figure 8. Under different Re number, velocity streamlines in three various transverse planes.

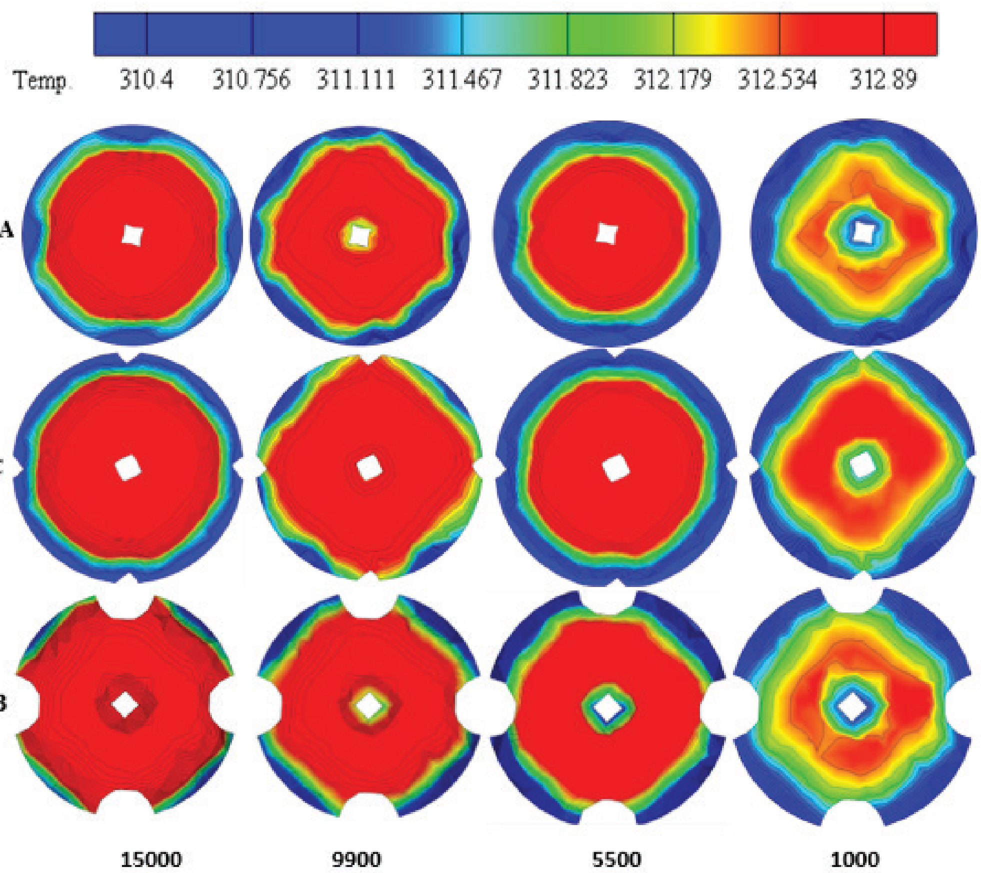


Figure 9. The temperature field contour in three various transverse planes in the pipe under different Re numbers.

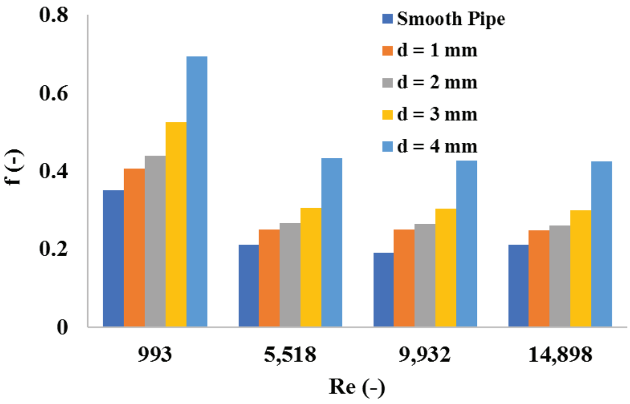


Figure 10. Friction factor variations at different geometrical configurations.

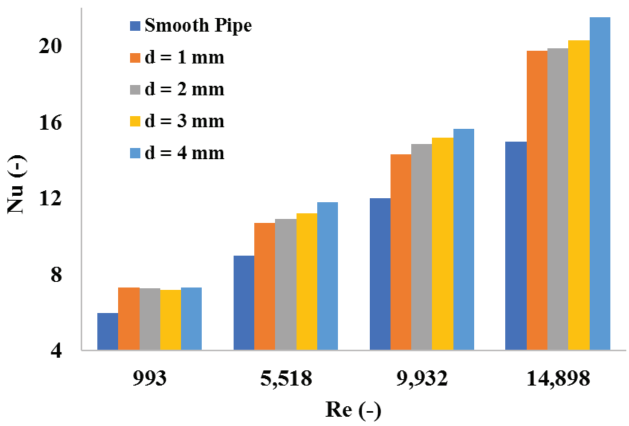


Figure 11. Nu number variations at different geometrical configurations.

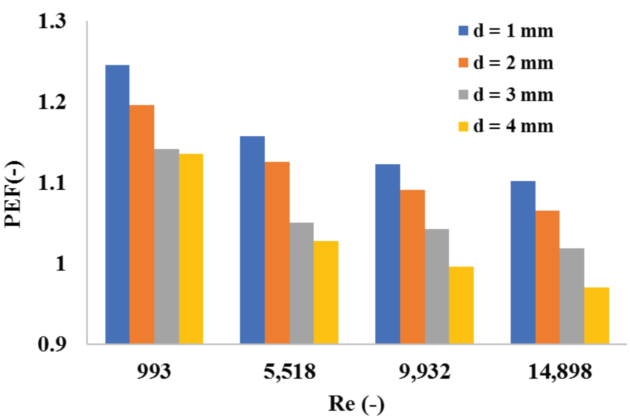


Figure 12. PEF variations with different Re at varying geometrical configurations.

than by increasing friction losses. Figure 12 shows the impact that Re numbers and geometric designs have on the performance factor for heat evaluation. Lower Re numbers and larger dimple diameters result in an increase in PEF. PEF is greater in the low range of Re number with dimple dimensions of 1 mm, as is shown in this figure. Geometrical designs have been found to significantly improve heat flow in pipes, resulting in improved thermal efficiency, as seen in this graph.

CONCLUSION

Various geometrical configurations of heat exchanger pipes (dimpled, corrugated, and twisted tape) were studied experimentally and numerically in this study to investigate their effects on heat transfer, heat performance, and friction loss. In comparison with a smooth pipe, the numerical results show the following main conclusions:

1. Due to streams travelling longer paths than smooth pipes, flow mixing occurs more frequently, causing a greater increase in force, which results in the sublayer breaking near the pipe walls.
2. In areas of the pipe wall and core, recirculation, mixing, and secondary flow can add to the heat performance. By disrupting the thermal boundary layer, better mixing can occur between the pipe wall and core. An enhanced pipe will have a velocity field in the core area, as well as a smooth pipe. It is possible to enrich the heat performance of the pipe wall and core by recirculation, mixing, and secondary flow through the pipe. An improved mixing between the pipe wall and the core can be achieved as a result of disruption of the thermal boundary layer. A high-distributed velocity field is located in the core area of smooth pipes as well as upgraded pipes
3. The model of a one-mm concavity (dimple) exhibited the greatest enhancement in heat conductivity. The turbulent intensity and flow perturbations in the core and pipe walls are primarily responsible for the damage. A corrugated tape's geometrical characteristics strongly influence vortex generation
4. Each pipe has a large velocity field surrounding its core, whether smooth or upgraded. One of the tested geometries with the greatest enhancement in heat transmission was the model with concavity (dimple) of one mm. Turbulent intensity and flow perturbations in the pipe core and wall cause most of the damage.
5. In the low Re number range, PEF is higher at a dimple dimension of 1 mm than 1.3.

NOMENCLATURE

Symbol	unit	Physical Parameter
A	m^2	Cross-sectional area
D	mm	Tube Diameter
E		Total energy

f		Friction Factor
h	$W \cdot m^{-2} \cdot K^{-1}$	Average coefficient of heat transfer
k		Thermal conductivity
$\dot{m}$	$kg \cdot s^{-1}$	Mass Flow Rate
Nu		Nusselt Number
Δp	Pa	Pressure Drop
$P_{av,inlet}$		Inlet average pressure
$P_{av,outlet}$		Outlet average pressure
Ph		Wetted perimeter
PEC		Performance Evaluation Criteria
Pr		Prandtl Number
Qh	W	Input Heating Load
Qloss	W	Heat Loss
Re		Reynolds Number
TKE		Turbulent kinetic energy
ΔT	°C	Logarithmic Mean Temperature Difference
uin	$m \cdot s^{-1}$	Inlet Velocity
u_m	$m \cdot s^{-1}$	Mean fluid velocity

Greek symbols

ϕ		General variable
ρ	$kg \cdot m^{-3}$	density
μ	$Pa \cdot s^{-1}$	dynamic viscosity
λ	$W \cdot m^{-1} \cdot K^{-1}$	conductivity
δ	mm	gap between tape and tube

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AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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