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Numerical investigation of heat transfer enhancement in a double pipe heat exchanger using tangential perforated ring turbulators

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Abstract

This research investigates the heat transfer and fluid flow characteristics of a double pipe heat exchanger enhanced with perforated turbulators. The study focuses on the effects of varying Reynolds numbers and geometric configurations, particularly the number of perforations in the inserts, on thermal performance. Using the finite volume method and Ansys Fluent simulations, the heat exchanger was analysed under different conditions, comparing the results with a smooth tube configuration. The findings reveal that the pitch ratio of 2.5 has shown the highest heat transfer capacity followed by pitch ratios of 4.5 and 6.5. Further, irrespective of relative pitch ratio, the ring with no perforations has shown the highest value of average Nusselt number and in the case of perforation, the open area ratio of 0.068 has yielded the best thermal performance.

Keywords: Computational fluid dynamics; Double pipe heat exchanger; Perforated ring turbulators; Tangent placed turbulator; Thermohydraulic performance

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1. Introduction

Heat exchange between flowing fluids is one of the most important physical processes of concern, and a variety of heat exchangers are used in different types of installations, like process industries, power plants, HVACs (heating, ventilation, and air conditioning systems), etc. The purpose of constructing a heat exchanger is to get an efficient method of heat transfer from one fluid to another, by direct contact or by indirect contact. The heat transfer occurs by three principles: conduction, convection and radiation. In a heat exchanger, the heat transfer through radiation is not taken into account as it is negligible in comparison to conduction and convection [1].

There are generally two methods observed in literature to augment heat transfer, i.e. active and passive techniques [2]. Active techniques require an external source of energy to improve heat transfer efficiency. Examples of such methods include the

Nomenclature

C_p – specific heat, J/(kg K)
d, D – diameter, mm
D_h – hydraulic diameter, mm
Fr – friction factor
h – ring thickness, mm
H – heat transfer coefficient, W/(m^2 K)
I – uniform heat flux, W/m^2
K – thermal conductivity, W/(m K)
L – length, mm
Nu – Nusselt number, $=H D_h/K$
P – roughness pitch between the rings, mm
p – pressure, Pa
Pr – Prandtl number, $= \nu/\alpha$
PR – pitch ratio, $= p/D_h$
Re – Reynolds number, $= \rho v D_h/\mu$
T – temperature, K
u, v, w– velocity components, m/s
x, y, z– Cartesian coordinates, m

Greek symbols
α – thermal diffusivity, m^2/s
λ – open area ratios
μ – dynamic viscosity, Pa·s
ν – kinematic viscosity, m^2/s
ρ – density, kg/m^3

Subscripts and Superscripts
i – inner
h – hydraulic
o – outer

Abbreviations and Acronyms
CFD – computational fluid dynamics
PTT – twisted tape with perforations
PCR – perforated ring

use of cams and reciprocating plungers to create induced pulsations, employing a magnetic field to disturb light particles in a fluid stream, and implementing mechanical aids. Other techniques involve surface vibration, fluid vibration, electrostatic fields, as well as methods like suction, injection, and jet impingement, all of which depend on an external power source to achieve enhanced heat transfer [3].

Passive techniques involve altering the flow channel's surface or design through various modifications, or insertion of different-sized particles (particularly nano or micro) in fluid [4]. These can include adding components like inserts or swirl flow devices, modifying the surface with treatments or roughness, extending surfaces, using displaced enhancement devices, employing coiled tubes, etc. [5]. Over time, various advanced designs have been introduced to balance acceptable pressure levels while improving heat transfer efficiency. Enhancements in heat exchanger performance have been achieved by optimizing surface area, incorporating turbulence-inducing elements, minimizing the fluid's overall thermal resistance, and lowering pumping power requirements for a given heat load [6,7]. The flow pattern within a heat exchanger's passage is inherently intricate, with the natural formation of vortices. This complexity increases significantly when the geometric configuration is altered, as such modifications amplify interruptions in the flow. By introducing elements like vortex generators or louvers, it is possible to harness this phenomenon to create intentional, large-scale vortices along the flow direction. Commonly used vortex generators include devices like twisted tapes, ring-shaped inserts, baffles, turbulators, winglets, and pins, which are widely implemented to improve the efficiency of heat exchangers. Detailed investigations into heat transfer enhancements have been conducted by researchers, such as Khoshvaght-Aliabadi et al. [8], Sinha et al. [9], and Hu et al. [10].

Researchers have explored various inserts to enhance heat transfer in heat exchangers. Thianpong et al. [11] introduced a novel approach by incorporating twisted tapes with perforations (PTT) into heat exchanger tubes. These tapes are designed with holes along their entire length, which significantly boosts the heat transfer rate. Their findings revealed an impressive improvement of 36−85% compared to plain tubes without PTT. Additionally, they discovered that reducing the pitch ratio and twist ratio of the tapes further enhances heat transfer efficiency. Eiamsa-ard [12] expanded on this concept by employing multiple twisted tapes within the same tube, ranging from two to four tapes. This configuration resulted in thermal performance factors superior to those achieved with a single twisted tape, with an observed efficiency increase of 0.94−1.4%. Murugesan et al. [13] explored the use of twisted tape inserts with trapezoidal cuts and found that these modified inserts significantly improved thermal performance. In another study, Murugesan et al. [14] investigated twisted tapes with U-shaped cuts, discovering that this design achieved a thermal enhancement factor of 1.22 with a twist ratio of 2.0. Promvonge [15] conducted an experimental study combining wire coils and twisted tapes inside a heat exchanger tube. The wire coil was wound around the twisted tape along its entire length, creating a swirl flow within the tube. This combination nearly doubled the efficiency compared to using twisted tape alone. Eiamsa-ard et al. [16] extended this concept by experimenting with variable-pitch coils, finding that adjusting the coil pitch resulted in the highest efficiency.

Arulprakasam Jothi et al. [17] focused on using conical strip inserts as turbulators inside a circular tube. They experimented with conical strips having varying twist ratios in both staggered and non-staggered arrangements. Their research showed that the staggered arrangement of conical strips with a twist ratio of 3 achieved the highest Nusselt number. Promvonge and Eiamsa-ard [18] conducted a study where they utilized a combination of conical-ring and twisted-tape inserts within a circular tube. Their findings revealed that this hybrid configuration significantly improved the heat transfer rate, achieving an enhancement of over 10% compared to using conical-ring inserts alone. This demonstrates the synergistic effect of combining different types of inserts in optimizing thermal performance in heat exchangers.

This research paper focuses on double pipe heat exchangers, and the following are the highlights of relevant work previously conducted in this field. Saud Ghani et al. [19] experimentally demonstrated that double pipe heat exchangers improve air conditioning performance by reducing compressor work and increasing efficiency. Rennie and Raghavan [20] numerically analysed double pipe helical heat exchangers, providing key design insights for enhancing temperature control and uniformity in liquid food processing. Structural modifications in double pipe heat exchangers primarily involve the incorporation of elements such as fin inserts [21,22], wire inserts [23,24], and tape inserts [25], as well as alterations to pipe designs [26,27]. Additionally, techniques like employing porous media [28] and integrating generators and turbulators [29] are also utilized. Sheikholeslami and Ganji [30] used circular perforated rings turbulators on air side in their double pipe heat exchanger. They placed turbulators in radial direction of the tube and found that thermal performance enhances with an increase in open area ratio. Salem et al. [31] performed comparative experimental study on flower and conventional segmental baffles in double pipe heat exchanger and observed that flower design achieve better characteristics than traditional baffles. Banihashemi et al. [32] investigated the impact of using moving turbulators in heat exchangers, comparing them to stationary turbulators. They found that rotating turbulators, particularly those with smaller angle ratios, greatly improved thermal efficiency and performed better than the stationary versions. Nakhchi et al. [33] studied double-pipe heat exchangers that utilized perforated, inclined elliptic turbulators. Their findings revealed a 217.4% rise in the Nusselt number and a 39.4% improvement in heat transfer. They achieved a thermal efficiency of 1.85 without notable increases in friction loss. Kumar [34] conducted a study on the impact of hemispherical turbulators in a double pipe heat exchanger. The study showed that while the turbulators enhanced heat transfer, they also led to an increase in the friction factor. The highest performance indicator recorded was 1.41.

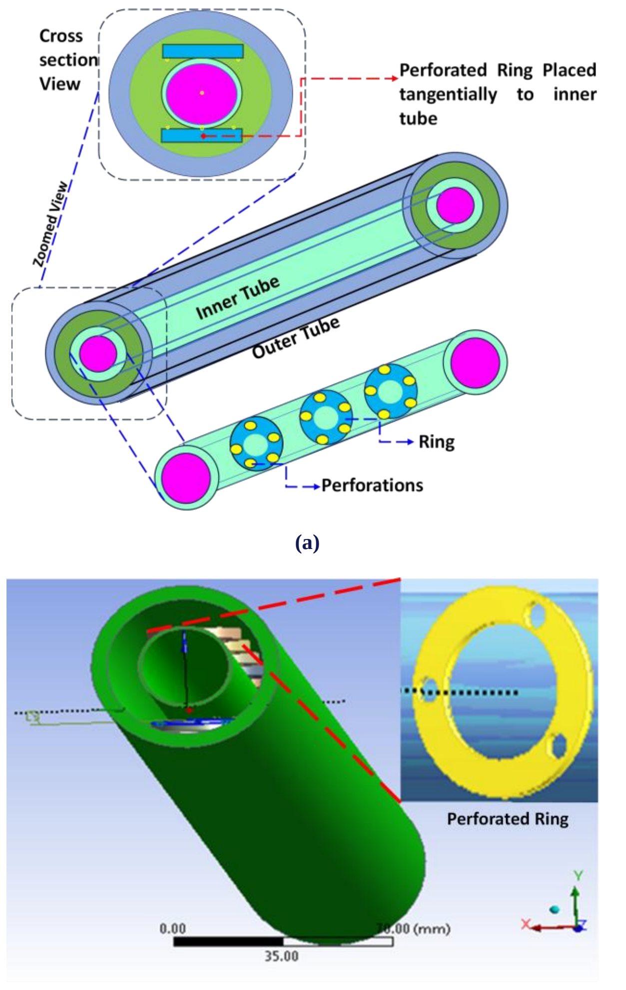
In present research work, heat transfer and fluid flow properties in a double pipe heat exchanger that uses perforated turbulators inserts were explored. The turbulators were placed tangentially to the inner tube. The novelty of this research lies in the use of tangential perforated ring turbulators to enhance heat transfer in a double pipe heat exchanger, offering a unique design innovation compared to the perforated turbulators studied by Sheikholeslami and Ganji [30]. The tangential orientation of the turbulators induces swirling flows and enhanced turbulence near the heat transfer surfaces, leading to improved fluid mixing and thermal performance. Unlike [30], which focused on axial perforations, this study investigates the effects of open area ratio on both heat transfer and pressure drop. Additionally, demonstrating superior heat transfer rates and overall performance, it provides new insights into optimizing heat exchanger design. The heat exchanger performance were evaluated for varying operating condition, i.e. Reynolds number, and varying geometric condition, i.e. number of perforations in the inserts.

2. Materials and methods

The double-tube heat exchanger featuring perforated rings was numerically simulated using finite volume method. Using commercial code Ansys [35], the heat exchanger was analysed for different operating and geometric parameters. Further, comparison was also made with smooth tubes in heat exchanger. The working fluid for this analysis is air, which flows through a constant heat flux tube equipped with perforated rings.

The present model consists of two concentric tubes: a larger outer tube with an inner diameter (D_i) of 50 mm and an outer diameter (D_o) of 60 mm, and a smaller inner tube with an inner diameter of (d_i) 28 mm and an outer diameter (d_o) of 30 mm. The smaller tube is equipped with perforated rings, which serve to ensure a steady flow of hot air in the outer tube. The perforated rings are made of copper and vary in open area ratios and depths, affecting the fluid flow behaviour. Figure 1a depicts schematic diagram of double-tube heat exchanger with perforated rings. Two views were zoomed out from Fig. 1a. Firstly, the inlet cross section, which shows the location of perforated ring, i.e. tangentially to the inner tube.

The rings were placed at two different locations, i.e. at the top and at the bottom. In the second view, the inner tube is displayed. It shows how perforations were made on the rings, i.e. along the circumference of the circle. Further, for better understanding, 3D geometrical model of present work is shown in Fig. 1b.



(a)

(b)

Fig. 1. (a) Schematic diagram with zoomed view; (b) 3D geometric model of double pipe heat exchanger with tangential perforated ring turbulators.

The study examines different mass flow rates and open area ratios (λ = 0, 0.088, 0.068 and 0.056). A constant heat flux of 1000 W/m² is applied along the entire length of the 1200 mm double-tube system, with turbulent flow conditions analysed in the range of Reynolds numbers from 6000 to 14 000 and pitch ratios (PR) of 2.5, 4.5 and 6.5. Pitch ratio is the ratio of roughness pitch to the hydraulic diameter. The *k–ε* turbulence renormalisation group (RNG) model with enhanced wall treatment for swirl-dominated flow and thermal effects was used for the simulation in Ansys Fluent.

The cold air flowing in the annular gap was heated by the constant heat flux provided on the inner tube wall. The perforated rings (PCR) enhance the total convective area for better heat transfer. Different PCRs characterised by their number of holes and roughness patterns, are installed on the hot water tube to analyse thermo-hydraulic performance. The entrance and exit lengths within this system are not fixed; instead, they vary based on the pitch, which can range from 75 to 195 mm. As the pitch changes within this range, both the entrance and exit lengths adjust accordingly, reflecting the dependency of these dimensions on the chosen pitch value. It was ensured that the flow reaches a fully-developed state. The complete details of the numerical models are provided in Table 1.

Table 1. Details of geometric parameters of the double-tube with perforated rings shaped roughness.

Parameters	Range of value
Total length of tubes, L	1200 mm
Air tube inner diameter	50 mm
Air tube outer diameter	60 mm
Pitch between the rings	75 mm, 135 mm, 195 mm
Pitch ratio (roughness pitch to hydraulic diameter ratio), PR = p/D_h	2.5, 4.5, 6.5
Hydraulic diameter, D_h	30 mm
Perforated ring thickness	4 mm, 5 mm, 6 mm
Uniform heat flux, I	1000 W/m²
Reynolds number, Re	6000, 8000, 10000, 12000, 14000
Number of holes used in perforated ring, N	3, 5
Diameter of holes	4 mm
Open area ratio, λ	0.0833, 0.068, 0.056
Open area ratio for perforated ring without hole, λ	0
Prandtl number, Pr	0.707

3. Numerical modelling

3.1. Governing equations

The governing equations for the flow system have been formulated for three different flow scenarios: steady laminar, unsteady laminar, and turbulent. This section concentrates on the conservation equations applicable to laminar and turbulent flow conditions. The model is very helpful in present geometry as it accurately predicts the swirling flow conditions. The RNG *k–ε* turbulence model is having a quite similar structure to standard *k–ε* but differs in how it manages turbulent viscosity, diffusion, and heat transfer enhancement. The standard *k–ε* model is effective for many turbulence scenarios but less suitable for large adverse pressure gradients.

The mathematical expression of the principle of conservation of mass applied to an elemental control volume within a fluid under motion, known as continuity equation, is given by

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0. \tag{1}$$

The variables u, v and w are velocity components in x-, y- and z-direction, respectively.

The governing equations for momentum conservation in the fluid domain along the three coordinate axes are formulated as follows:

x-momentum equation

$$\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{1}{\rho}\frac{\partial p}{\partial x}+v\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right), \tag{2}$$

y-momentum equation

$$\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)=-\frac{1}{\rho}\frac{\partial p}{\partial y}+v\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right), \tag{3}$$

z-momentum equation

$$\left(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)=-\frac{1}{\rho}\frac{\partial p}{\partial z}+v\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right), \tag{4}$$

where p is the pressure, ρ is the density and v is the kinematic viscosity.

Assuming that the flow is steady and incompressible with constant thermal conductivity and without heat generation and viscous heating, the energy equation is as follows:

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial y^2}\right), \tag{5}$$

where α is the thermal diffusivity and T is the temperature.

To simplify the numerical simulation, the following assumptions were made in present work:

- the flow is steady throughout the simulation,
- there is no pressure variation in the y-direction,
- shear forces in the y-direction are assumed to be zero,
- effect of gravity on body forces is neglected,
- the flow is incompressible,
- the flow entering the test section is fully developed,
- axial heat conduction in the fluid is considered insignificant,
- air properties are held constant at standard atmospheric conditions.

3.2. Material properties

In the present study, the inner and outer tube contain water and air, respectively. Copper was chosen as the material for the tube wall and perforated rings due to its superior thermo-physical characteristics, high thermal conductivity, machinability, and low cost. The thermo-physical properties of the working fluid and tube material considered in the simulation are tabulated in Table 2.

Table 2. Thermophysical properties of working fluid and materials considered.

Properties	Copper	Air	Water
Density, ρ (kg/m^3)	8978	1.225	998.2
Specific heat, C_p, (J/(kg K))	381	1006.4	4182
Thermal conductivity, K (W/(m K))	387.6	0.028	0.6
Viscosity, μ (Pa s)	–	1.7894×10^{-5}	0.001003

3.3. Boundary conditions

For the numerical analysis of the double-tube heat exchanger with perforated rings, the boundary conditions are set for the inlet (velocity), outlet (pressure), and wall surfaces of the model. A no-slip boundary condition is applied to all solid surfaces, and a turbulence intensity of 5% is used for these walls. The inlet flow conditions are defined by a Reynolds number range of 6000 to 14 000. The outer wall of heat exchanger holds adiabatic condition. The simulation is performed using velocity-pressure coupling, Green-Gauss node-based methods, and a second-order upwind scheme for calculating momentum, turbulent kinetic energy, and dissipation rate.

3.4. Mesh generation and grid independence test

In computational modelling, the mesh represents a discrete approximation of the geometric model of the double tube. Achieving the ideal balance of automation, validity, accuracy, and efficiency in meshing often requires compromises. The primary goals of meshing include ensuring valid simulations, enhancing accuracy, and optimizing computational efficiency through high-quality graded meshes and flexible control of mesh density. In this study, 3D uniform meshing was employed, specifically using a regular tetrahedron format. This approach involves boundary-based node smoothing and domain boundary recovery. The mesh was generated for the double tube with perforated rings with specific conditions: a span angle centre of fine resolution, smooth transitions, and an element size of 1 mm, as illustrated in Fig. 2. The perforated rings were resolved by mesh and no separate approach was opted for them. This method combines the best aspects of various meshing tools into a single environment to produce high-quality meshes. The maximum skewness observed is 0.8, with an average skewness of 0.25. The highest skewness values are found in the perforated ring. The domain's orthogonal quality is 0.95, indicating satisfactory mesh quality. Additionally, the cell wall distance y (distance of the cell centre of the first layer of elements from the wall) is set to achieve $y^+ > 1$, considering the high Reynolds number in the model. Once computations are completed, if the results remain consistent across different mesh elements, it indicates that the mesh is independent, meaning that further changes to the mesh will not affect the results. This concept is crucial for ensuring the reliability of comparative results.

A grid independence test was performed using a solution-adaptive refinement method to ensure that the mesh is optimized for the simulation. This method adjusts the mesh to be as effective as possible for solving the flow problem while avoiding unnecessary computational costs. According to Table 3, after reaching 939 124 elements, the heat transfer results stabilized and were accurate for the geometric model. At this mesh size, the Nusselt number remained constant regardless of further grid adjustments.

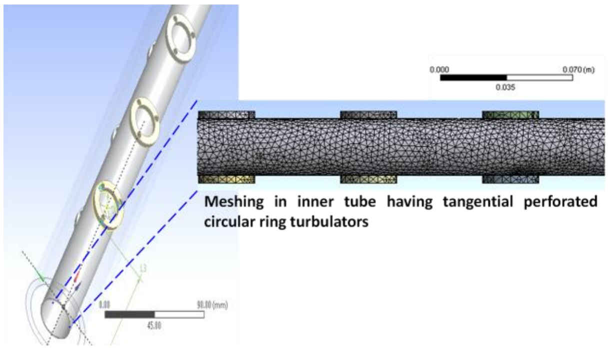


Fig. 2. Meshed image of the inner tube with tangentially placed perforated rings.

Table 3. Variation of Nusselt number with grid size.

Number of nodes	Number of elements	Nusselt number
79390	197 959	23.091230
131 146	404 845	33.771310
197 566	797 580	36.156780
220 502	939 124	36.182411

3.5. Validation of present work

To validate the current numerical method and model, the results were compared with those from a study by Sheikholeslami and Ganji [30]. Their research investigated heat transfer in a double pipe with perforated roughness, 1200 mm in length, with inner and outer diameters of 50 mm and 60 mm, respectively. The study evaluated Nusselt numbers and friction factors across different Reynolds numbers (6000, 8000, 10 000, 12 000, 14 000). The comparison of these results, illustrated in Fig. 3, indicated a deviation of ±9% to 12% from experimental data, which is deemed acceptable for CFD simulations using Fluent 14.0.

Further, comparisons were also made between CFD results for the smooth tube and Dittus-Boelter correlation, and a deviation of 3% was observed.

4. Results and discussion

The research focused on analysing the heat transfer performance and pressure drop characteristics of a double tube heat exchanger featuring perforated rings, comparing these results with those of a smooth tube under identical conditions. A three-dimensional model was developed using Ansys Workbench 14.0 for the numerical simulation.

The study investigated how different configurations of perforations (none, three or five) affected the heat transfer and fric-

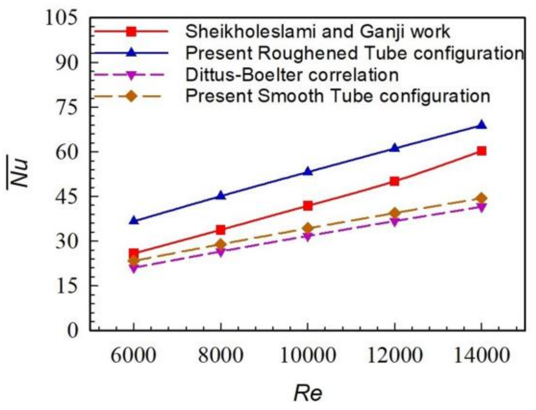


Fig. 3. Validation plot for roughened and smooth tube configuration.

tion characteristics of the heat exchanger. The hydraulic diameter for the annular gap was 30 mm, and the simulations covered Reynolds numbers from 6000 to 14 000. The analysis results, presented through the figures, illustrate the variations in Nusselt number (Nu) and friction factor due to the presence of the PCRs.

In order to examine the heat transfer rate of any fluid flow, the importance of Nusselt number cannot be undermined. In such situation, the study of the effect on Nu due to various modification performed on the existing arrangement becomes imperative. Here, different dimension geometries were examined to analyse the variation in Nu with respect to Re. The change in Nu with the change in Re for different geometries at PR = 2.5, 4.5 and 6.5 is shown in Fig. 4a–c, respectively. It can be seen from Fig. 4 that Nu increases with the increase in Re for all geometries. For PR = 2.5, the highest increase in Nu is observed for the case of open area ratio $\lambda = 0$ followed by with an open area ratio of 0.068, 0.083 and 0.056, respectively. Apparently, the lowest rate of increase in Nu was obtained for smooth tube. The trend of Nu changes with respect to Re at PR = 4.5 and PR = 6.5 was observed to be similar to that observed for the case PR =2.5, as shown in Fig. 4b and c, respectively. Unlike the trends of Nu with respect to Re, the difference in the Nu value noted for 2.5 PR is different from that obtained for the cases PR = 4.5 and 6.5. Largest difference in Nu values was observed between the open area ratio of 0.056 and the smooth tube at PR = 2.5, which is comparable to the difference noted between Nu values for the open area ratio 0.056 and the smooth tube at PR = 4.5. The effect of open area ratio on the variation in Nu is noteworthy at PR = 2.5 and 4.5 but upon examining the case for PR = 6.5, it was found that the value of Nu for the open area ratios of 0.056 and 0.083 are very close to each other at every value of Re. Nu values suggested that the heat transfer is dominated by convection at all three values of PR for all geometries except the smooth surface where the convection effect was not found to be of much significance. The heat transfer between the fluid flow is also a function of friction factor. Since friction ratio indicates the friction loss occurring in the system during the flow, it causes reduction in the overall heat transfer from or to the system.

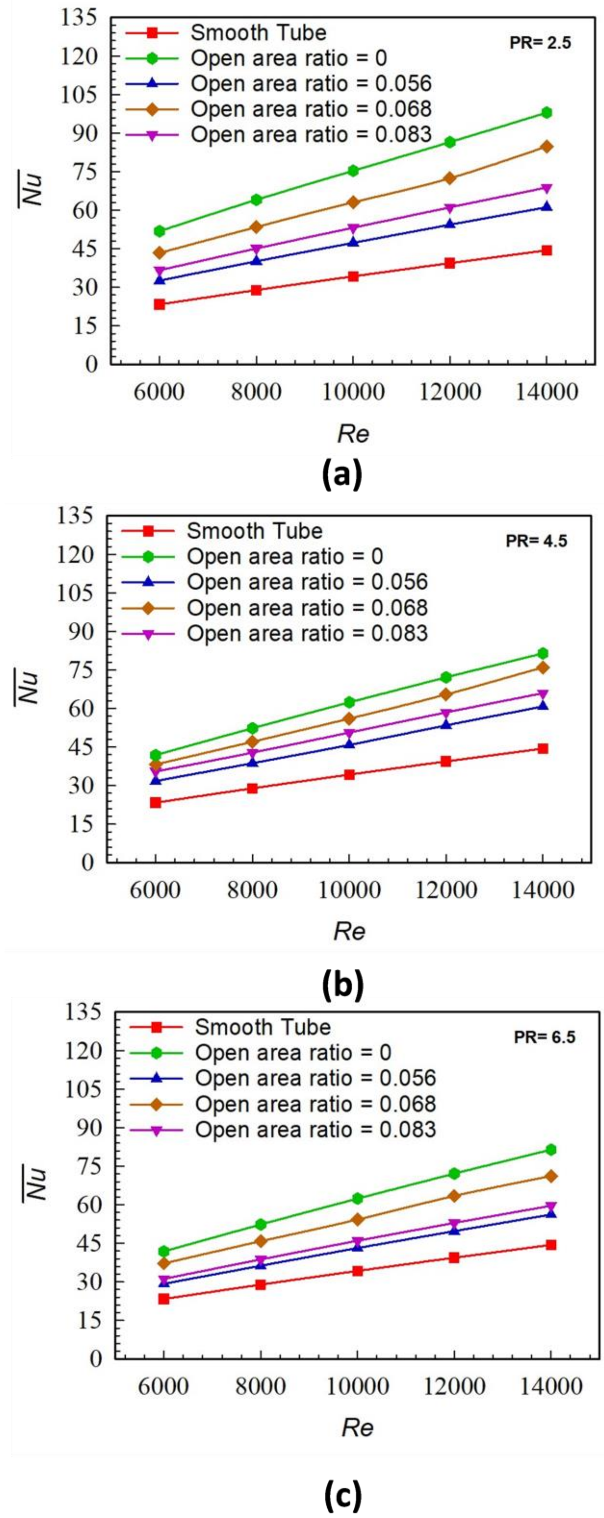


Fig. 4. Variation of average Nusselt number with Reynolds number for different configurations of heat exchanger.

The change in the value of friction factor relative to Re for the considered geometry configurations is plotted in Fig. 5. As can be seen, Fig. 5a–c have been plotted to analyse the effect of change in Re on friction factor for different configurations at PR = 2.5, 4.5 and 6.5, respectively. At PR = 2.5, the change in the friction ratio for smooth pipe with respect to Re is obtained as the lowest among all configurations with a negligible change. This behaviour of friction ratio for smooth pipe is similar to the behaviour obtained at PR = 4.5 and 6.5 with respect to Re values. Among different configurations, open area ratio $\lambda = 0.056$ exhibits low friction factor, followed by open area ratios $\lambda = 0.083$ and 0.068. It was found that configuration with an open area ratio of $\lambda = 0$ showed the highest friction factor at all values of Re. Interestingly, the difference in the friction ratio values at PR = 4.5 and 6.5 is less as compared to the difference in the friction ratio values obtained for PR = 2.5 at all values of Re. Hence, the improvement in the heat transfer can be achieved in the case of smooth pipe after examining the friction factor. But looking at Nu, the overall heat transfer for smooth pipe is somehow reduced when compared to other configurations.

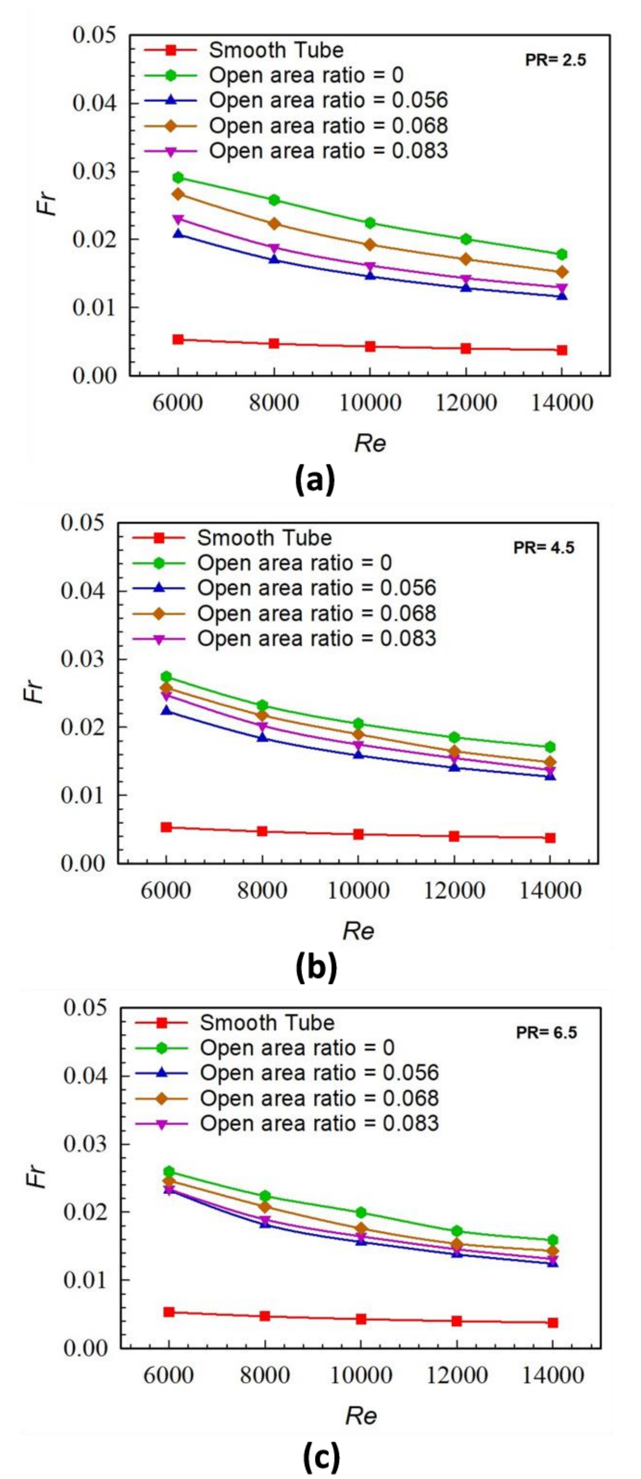


Fig. 5. Friction factor variation with Reynolds number for different configurations of heat exchanger.

5. Conclusions

In this study, the heat transfer and fluid flow characteristics of a double pipe heat exchanger equipped with perforated turbulators were comprehensively analysed. The findings demonstrate that the incorporation of these turbulators significantly enhances the heat transfer performance across various operating conditions and geometric configurations. The Nusselt number increased with Reynolds number for all examined geometries, with the most pronounced improvements observed for configurations with specific open area ratios. Additionally, the friction factor analysis revealed that while the use of turbulators increases frictional losses, it concurrently enhances overall heat transfer, making it a promising modification for improving the thermal efficiency of heat exchangers. This research provides valuable insights into optimizing heat exchanger designs for better thermal performance under diverse conditions.

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