Numerical investigation on double pipe heat exchanger for augmentation of heat transfer having twisted inner pipe with conical ring turbulator

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A thesis submitted in partial fulfillment of the requirement for the degree of Bachelor of Science in Mechanical Engineering

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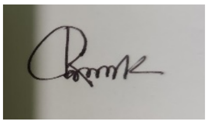
Declaration

This is to certify that the work presented in this thesis, titled, "Numerical investigation on double pipe heat exchanger for augmentation of heat transfer having twisted inner pipe with conical ring turbulator", is the outcome of the investigation and research carried out by me under the supervision of Dr. Md. Rezwanul Karim, Associate Professor, Islamic University of Technology (IUT)

It is also declared that neither this thesis nor any part of it has been submitted elsewhere for the award of any degree or diploma.



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RECOMMENDATION OF THE BOARD OF EXAMINERS

The thesis titled "NUMERICAL INVESTIGATION ON DOUBLE PIPE HEAT EXCHANGER FOR AUGMENTATION OF HEAT TRANSFER HAVING TWISTED INNER PIPE WITH CONICAL RING TURBULATOR" submitted by RIZVI AREFIN RINIK, Student No: 180011241 has been accepted as satisfactory in partial fulfillment of the requirements for the degree of B Sc. in Mechanical Engineering on **19 May,2023**

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ABSTRACT

Double pipe heat exchanger is made of two concentric pipe where one carries hot fluid and another one carries cold fluid inside the pipe. The walls of the pipes allow heat to pass between the fluids. Enhancing the transfer of heat is crucial for effective heat exchangers. This study focuses on evaluating the impact of a fully twisted inner pipe along with conical ring turbulator by conducting a numerical analysis of the double-pipe heat exchanger (DPHE). The analysis considered both parallel and counterflow configurations for fluid-to-fluid heat transfer. Additionally a straight elliptical pipe, two other pipes with three twists and full twists along their lengths are investigated to compare heat transfer rate, pressure drop, and turbulence. The realizable k-turbulence model is used, and a grid independency study has conducted using a 3-dimensional structured and unstructured mesh approach. The fully twisted effect generates a swirling motion, and conical rings are inserted inside the outer pipe as a passive turbulator to guide the flow towards the inner pipe, where the hot fluid passes. The regime of turbulent flow is studied within the range of Reynolds numbers from 5000 to 26,000 with water is simulated as the model fluid. Comparing the results, the inner twisted pipe with the conical ring exhibits a significant improvement in the Nusselt number, reaching 445 in the counterflow direction with the six rings. The Performance Evaluation Factor (PEF) is also greater than 1 for both parallel and counterflow flow, indicating that enhancement of the rate of heat transfer, outweighs the decrease in pressure drop. Particularly in counterflow directions the PEF is 2.3 which is impressive. Overall, full twist along the pipe lengths enhances the heat exchanger's performance and full twist with six conical ring fortify most in both flow directions.

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NOMENCLATURE

L = Length of the tube (mm)

d_a = major axis (mm)

d_b = minor axis (mm)

T = temperature (k)

t = thickness (mm)

ρ = Density ($\frac{kg}{m^3}$)

k = thermal conductivity ($\frac{w}{mk}$)

c = specific heat ($\frac{j}{kg.°C}$)

t_i= inlet temperature (k)

t_o= outlet temperature (k)

μ= Dynamic Viscosity ($\frac{kg}{m.s}$)

<u>Acronym</u>

TOT = Twisted Oval Tube

DPHE = Double Pipe Heat Exchanger

HE/HX = Heat Exchanger

PEF = Performance Effective Factor

DTHE = Double Tube Heat Exchanger

SST = Shear Stress Transport

Chapter - 1: Introduction

For industrial and many other purposes we need to transfer heat from one medium to another medium, for that we need a device which is a heat exchanger. Several types of heat exchanger is designed in the field of engineering but most simple design has done with two pipes are nested within one another to create an annular gap. As the fluids move through the inner and outer pipes, heat is transferred between them. Typically, people will call the inner pipe the "tube" or "inner tube," whereas the outside pipe will be called the "shell" or "outer pipe." The fluid that requires heating or cooling passes through the inner tube, while the heat transfer or absorption fluid circulates in the space between the tubes, known as the annular area, on the outer shell side. [1]–[3]

Conduction across the tube wall is the mechanism through which heat is transferred in a double-pipe heat exchanger. Depending on the temperature difference, the fluid in the inner tube will either heat or cool the fluid in the outer ring. The fluid then transfers the heat to its surroundings in the outer pipe. Depending on the needs of the procedure, the flow may be parallel or counter. [3], [4]

1.1 Background

Heat exchangers are crucial parts of many different kinds of thermal processes. Heat exchangers are essential components of many modern conveniences, refrigeration, chemical processing, electricity generation plants, and even automobiles.

Due to their simple design and minimal maintenance needs, double pipe heat exchangers (DPHE) find application in various technical fields. The overall performance of the DPHE is a crucial metric throughout the design phase. Double-pipe heat exchangers are small despite their great surface area for heat transfer for compact space. This is achieved through the use of multiple tube passes, where the fluid flows back and forth between the tubes, increasing the contact time and enhancing heat transfer efficiency. The size and running costs of a HX are directly affected by its thermal and fluid flow characteristics, both of which make up its overall performance. [1] This explains the extensive research and development that has gone into discovering new ways to enhance performance. In this research, it is recommended that the heat component be enhanced. Increasing the efficiency of heat exchange between the working fluids has a significant bearing on the system's performance. However, if you make a proposal to improve one component or the whole system's performance, you'll need to make similar adjustments to the other component as well. As an example, increasing the heat transfer rate could cause an increase in pressure drop, which would necessitate more pumping power and increase the operational cost. [5]

Many efforts have been done in the past few decades to increase the effectiveness of these HE’s. Increasing turbulence in the flow and decreasing laminar separation are the two main goals of the adjustments. how high the fluid's thermal substrates are. To improve the mixing process, one can use either an active or passive technique, or a hybrid of the two. Some examples of active methods that improve the HE's performance by applying an external force are increasing fluid vibration with ultrasound waves, increasing mixing by rotating the inner tube and employing magnetic nanoparticles in the fluid, as well as moving the magnets around the pipes. Deforming the HE Structure or introducing new, either permanent or removable, components into the interior are examples of passive strategies that can be used to promote mixing. Because of how simple it is to the passive method, which has been used by numerous researchers recently. [6]

1.2 Objectives of the study

Currently a significant amount or research work has done on DPHE. Most of the work is based on different types fluid, turbulator and different geometrical shape modification on pipe. Nano fluid is widely used for the enhancement of performance but they have also some drawbacks.[7] There are two types of turbulator, active turbulator and passive turbulator. Active turbulator requires energy supply from the outside source like rotating magnetic field, Vibration on the fluid all of them requires energy supply that will cost additionally. Passive turbulator is a good option for the enhancement of the heat transfer but here pressure drop increases as well as more pumping power required. [8]

The specific objective of this research paper is to perform a numerical approach on an experimental work with addition of some modification on the geometry of the inner pipe and using some conical ring turbulator inside the outer tube to produce more converge flow to the hot fluid surface.

The goal of this research is to validate the numerical work with the experimental work and compare the performance of the modified model with the existing model to achieve better result in terms of Nu, h, PEF and q.

Chapter – 2: Literature Review

Two basic strategies for improving overall performance are readily discernible in the literature. These approaches can be active or passive. Active approaches use external power to carry out the suggested improvement. Electronic fields are common examples. Fluid vibration, injection, and suction. The total increase in performance from using active methods relies mostly on how much power is used and may drop by a lot. However, no external power source is needed for passive approaches, which has the disadvantage that the introduced augmentation technique causes drops in pressure.[9] Multiple attempts have been made, all using passive approaches, to improve heat transfer rates in DPHEs while minimizing pressure losses. According to a review of the relevant literature, increasing the DPHE's passive heat transfer rate can be accomplished through the incorporation of various items within the DPHE's inner tube (e.g., twisted tapes, coils, fins) or through the use of nanofluids in conjunction with turbulators.[10] Agrawal and Sengupta[11] thermal convection in a DPHE that is numerically close to laminar. There were a variety of inner tubes used, each with its own special periodic booster. The improved heat transfer was measured in the redesigned tube, but at the price of somewhat larger pressure decreases. And now we turn in a new direction, Chiang and Yang [12] The effectiveness of a DPHE using water as the working medium was examined through experiments using a serpentine inner pipe. They showed that an increase of just 40% in pressure drop might potentially result in a 100% increase in heat transfer efficiency. In another work, Dung and Chen [13]The efficiency of a DPHE with counter and parallel flow was analyzed numerically, and different configurations pipes within that are oval in shape, running both horizontally and vertically were tested.

A modest performance boost in heat transfer was seen in the parallel flow arrangement. Using twisted oval tubes instead, Tan x et al.[14] provide a comprehensive analysis in efficient transfer of heat as well as drop in pressure, both experimentally and analytically. Pressure drop and rate of heat transfer increases as result of the experiments. In addition, Heat transfer in twisted oval tubes with varying geometric characteristics was studied. Anbu et al[15] When the tape was introduced into the twisted tube, the experimental findings revealed that the turbulence and friction factor both rose dramatically. When the inner tape was also a component of the twist, the pressure loss and heat transmission were greatly amplified. Nu was also improved in both the straight and twisted tubes when nano additions were added to the basic fluid. Overall, At low Reynolds numbers, the twisted tube's thermal performance is greatly enhanced by the inner tape than that of the plain stube. Indurain et al. [16] To study how well the running fluid on the shell side cooled, a number of TOT were put into an industrial shell and tube HE. The recovered oil had a laminar flow regime (Re = 50–600) because it was thick. The results showed that the heat transmission and pressure drop were both improved when the tubes were twisted rather than utilizing straight pipes. The effect on heat transmission and pressure loss was studied after changing the oval tube section to a three-lobed one. Gao et al.[17] Experiments were conducted observe flow resistance and transfer rate of heat through water as a medium over twisted pipe in different flow regimes. Nusselt number and friction factor for twisted tubes are calculated to be 1.3-2 and 1.2-1 higher than for a smooth tube, respectively. Bhadouriya et al.[18] square tube twist ratios of 11, 16, and 18 were calculated and experimentally explored. The greatest mean Nusselt number was found with an energy input of H = 2.5 at Re = 3000. Many authors have researched fluid flow inside twisted pipes with square and elliptical cross sections, and the analysed literature shows that there are many different perspectives on heat transmission and flow characteristics. This week saw the debut of a unique solid-liquid blend delivery system: the twisted tri-lobed tube (TTT). Ievlev et al.[19] demonstrated the effects of air as the working fluid on heat transfer and hydraulic resistance across longitudinal and cross-flowing bundles of elliptically twisted tubes. Twist ratios between 6.5 and 35 were employed. The Reynolds numbers that were looked at were between 3000 and 10000. Over the wide range of investigated parameters, the use of twisted tubes resulted in a statistically significant improvement in heat transmission and a substantial decrease in heat exchanger size. Córcoles et al.[20] helically corrugated inner tubes were the focus of both an experimental research and a computational study of the thermal dynamics of a DPHE in three dimensions. Kamel et al.[21] Complex fluid dynamics using numerical simulation and sensitivity analysis. Davood et el.[22] Flow field and heat transfer properties in a tube with twisted-tape inserts and nanofluid.

All of the analyses focused on the 25,000 Re region of turbulent flow. The general rule is that rate of heat transfer may increases due to irregular surfaces, but at the expense of higher pressure drops as compared to a smooth pipe.

Chapter – 3: Description of The Model

For the computational model a part of the DPHE has been selected which is defined in the fig 1. The outer pipe is the simple circular tube and the inner pipe is fully twisted tube. Keeping the wall temperature constant while varying the fluid and wall temperatures greatly, a computational fluid dynamics investigation was conducted. Table 1 contains necessary information needed to design the heat exchanger, including geometric parameters and fluid properties. At the below figure that delineate a counter flow type heat exchanger where both of the working fluid is water.

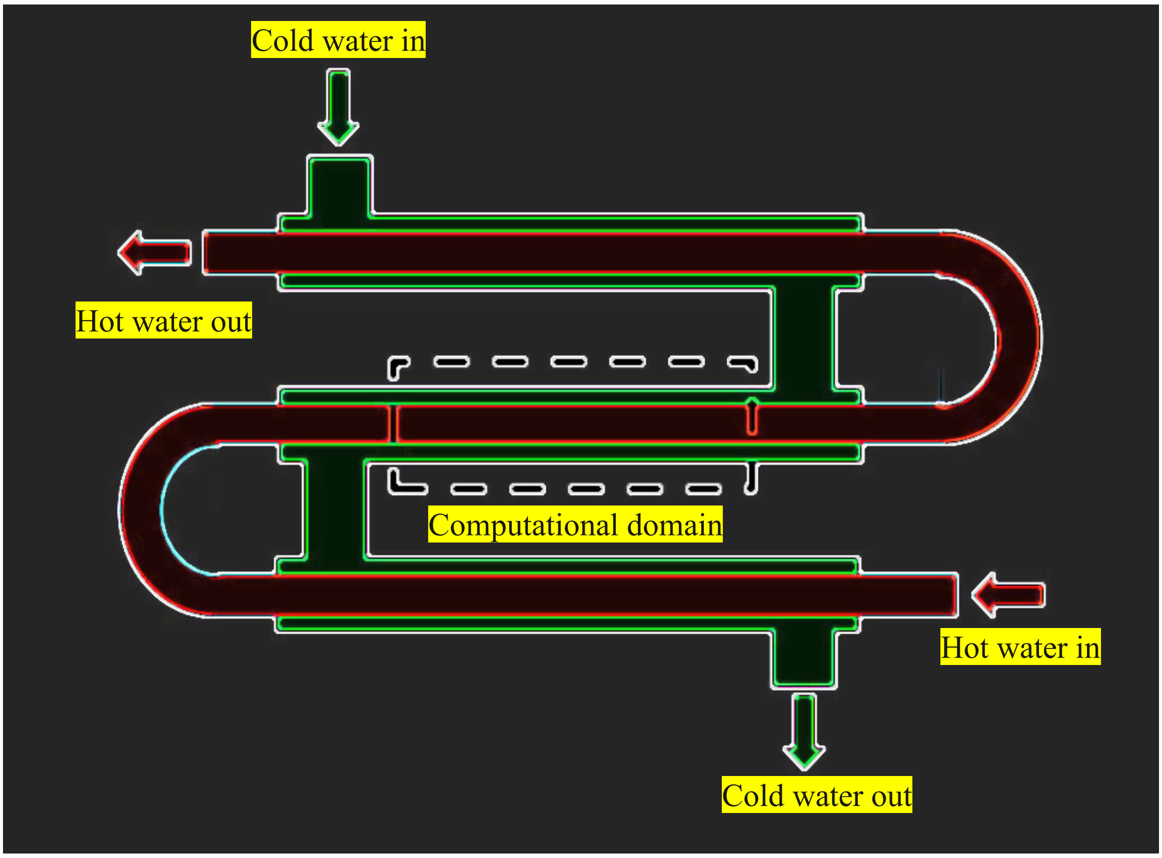

Figure 1: Schematic diagram of DPHE

3.1 Physical Model

In fig. 1 the simulation domain of the twisted annulus HX is shown. As can be seen in fig. 2, the inner oval tube twists in a counterclockwise direction. A tube of length 1000 mm can be expected. For outer tube, dimensions are kept always 38 mm in diameter and thicjness of the wall is 6 mm. The inner tube has a fixed major axis d_a and fixed minor axis d_b and an aspect ratio (e = d_b/d_a) of 0.42. The P twist pitch inner tubes are twisted in a certain way. With s = P/da, it can calculate the twist ratio. The various geometric parameters are summarized in Table 1.

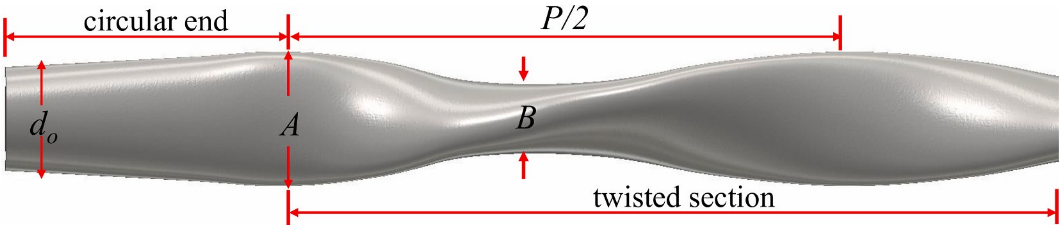

Figure 2: Inner pipe

Inner section of the pipe is twisted over the length where 50 mm pitch is used. The normal shape of the pipe is plane oval pipe and further it is twisted maintaining a certain pitch. Due to the twisting modification on the inner tube vortex motion will be generated on the flow.

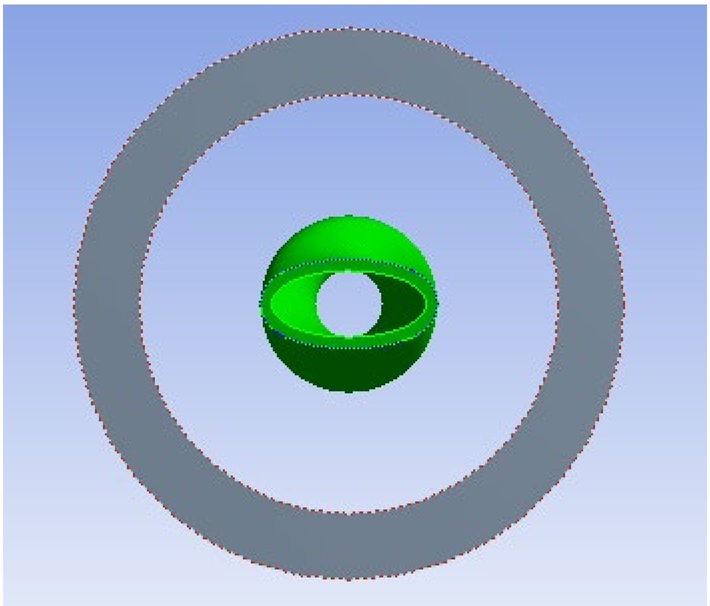

Figure 3: Face view of the computational portion

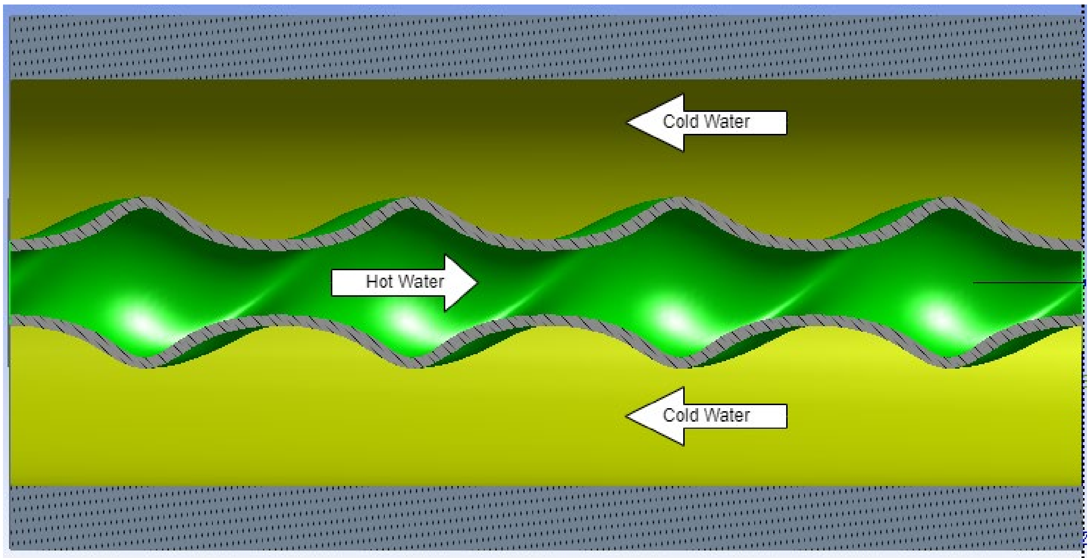

Figure 4: Cross section of the computational domain

In this fig. 4 Cross sectional view of the pipe is presented having counter flow of the fluid. The working medium is fluid and more specifically water. Throughout the analysis both parallel and counter flow are investigated.

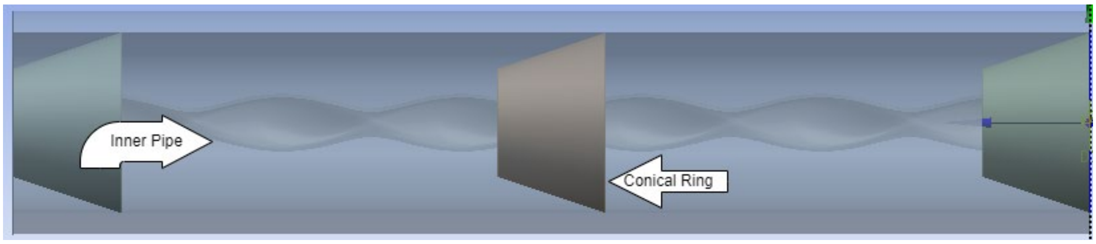

Figure 5: Conical ring turbulator inside outer pipe

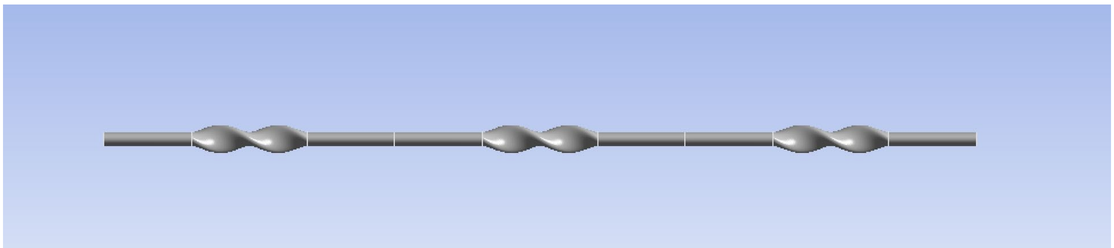

Figure 6: Inner pipe with three twist

The outer pipe consist of some conical ring 2,4,6 to observe the effect of turbulence produced by the conical ring. The flow is more likely to be converged due to the conical portion inside the pipe. The hot and cold fluid will have better surface contiguity due to the convergence.

3.2 Heat Exchanger Specification & Operating Parameters

The heat exchanger Consist of two parts mainly into the computational domain. The outside is a smooth, circular, 6 mm-thick PVC pipe with a 38 mm-diameter inner core. The inner part Consist plane elliptical tube having eccentricity of 0.42 and major axis diameter is 14 mm and minor axis diameter of 6 mm, further this tube is twisted fully to observe the performance of the DPHE. For the twisted portion the pitch is maintained approximately 50 mm.

Table 1: Heat Exchanger Specification

Geometrical Parameters	Value	Material
Tube Length (L)	1000 mm	-
Major Axis Dia (d_a)	14 mm	Copper
Minor Axis Dia (d_b)	6 mm	Copper
Inner Tube Thickness (t_i)	1 mm	Copper
Outer Tube Dia (d_O)	38 mm	PVC
Outer Tube Thickness (t_o)	6 mm	PVC
Eccentricity(e)	0.42	-
Pitch (p)	50 mm	-

In this study, main focuses on a specific application of DTHE as condenser along the x axis. The inner tube consist of hot water and the outer shell has cold water to maintain a constant flow of water. From 2 to 20 liter per minute water can enter the system. A computational model is developed to gain further insights into the characteristics of heat transfer and frictional hydraulic resistance in the shell side of Double Tube Heat Exchangers (DTHEs). The model is designed in SOLIDWORKS to facilitate a comprehensive understanding of properties of the heat transfer over it's length.

3.3 Boundary Conditions

In this numerical study only water medium is used as fluid. Heat transfer has taken place in between hot and cold fluid. Here is the boundary conditions for inlet and outlet for computational domain.

Table 2: Boundary conditions for flow

Property	Value
Mass flow rate of cold water	20 L/min
Mass flow rate of hot water	2-10 L/min
Inlet temperature of cold fluid	26-29 °C
Inlet temperature of hot fluid	50 °C

Table 3: The properties of water [23]

Property	Value
Thermal conductivity, k	0.642 W/m (°C)
Specific heat, c	4.1 kJ/kg (°C)
Density, ρ	990 kg/m^3
Prandtl number, Pr	3.71
Dynamic viscosity, μ	5.72 × 10−4 Pa·s

3.4 Numerical Scheme

Fluent can be utilized to solve the Reynolds-averaged Navier-Stokes equations. The current numerical computations employ the SST k-ε turbulent model, and its verification is demonstrated in Section 4.4. The turbulent model was also employed to investigate heat transfer in TOTs and understand the mechanics of frictional hydraulic resistance. [24]

The partial differential equations for mass, momentum, and energy are solved using the finite volume method and a second-order upwind system is implemented using the SIMPLE algorithm. The velocity and energy residuals are iteratively adjusted until they reach a convergence value of 10^{-6}. [25] Extensive simulation testing has determined that for effective flow development and the elimination of backflow in the shell side of Double-Pipe Heat Exchangers (DPHEs), it is crucial to have inlet and outlet sections with a length ranging from 2.5 to 4.0 times the length of the twisted pitch. This established guideline ensures optimal flow conditions within the heat exchanger.

In order to simulate the laminar viscous sub-layer and take into account the y^+ parameter, a finer mesh was created adjacent to the tube walls. In order to implement this improvement, the area of computation was discretized. By employing an improved wall treatment approach utilizing the k turbulence model, the boundary layer's of viscous sub-layer was successfully resolved. This was achieved by placing the initial cell normal to the wall at y^+=1. [26]

The following assumptions are made to streamline the calculations:

(1) A single, incompressible, and continuous phase of cold water makes up the system.

(2) Since the intake and output temperatures are so close, it is expected that water's characteristics do not change. (3) thermal radiation's consequences are overlooked. [27]

Chapter – 4: Computational Methodology

4.1 Computational Domain

The model is developed by ANSYS design modeler and the mesh for the domain is generated in the simulation software ANSYS 2022.

There are four key divisions in the computational domain: the hot fluid domain, the cold fluid domain, the inner pipe section, and the outer pipe section, which is made up of a conical ring turbulator. There are two types of mesh that are used across the four distinct domains. Due to the increased curvature of the inner tube's irregular surface, both structured and unstructured meshes are seen here. Both hexahedral and tetrahedra are prevalent in the mesh's element shape. The cold fluid zone is where the majority of the unstructured mesh forms. To improve the computational results, different element sizes are utilized in various fields. A summary of the domain's mesh characteristics may be found in Table 2.

The total number of elements are 3.5 million and the number of nodes are 1.95 million in the computational domain. Maximum skewness is 0.79 and average skewness is 0.20 and the greater skewness is available in the twisted section of the pipe. The orthogonal quality of the domain is 0.82 which is good enough.

Table 4: Mesh Properties

Domain	Element Shape	Element Size	Inflation Type	No. of Layers	Growth Rate
Cold fluid	tetrahedron	10 mm	Total layer thickness	8	1.2
Hot fluid	hexahedral	5 mm	Total layer thickness	6	1.1
Inner pipe	hexahedral	3 mm	-	-	
Outer pipe	hexahedral	5 mm	-	-	

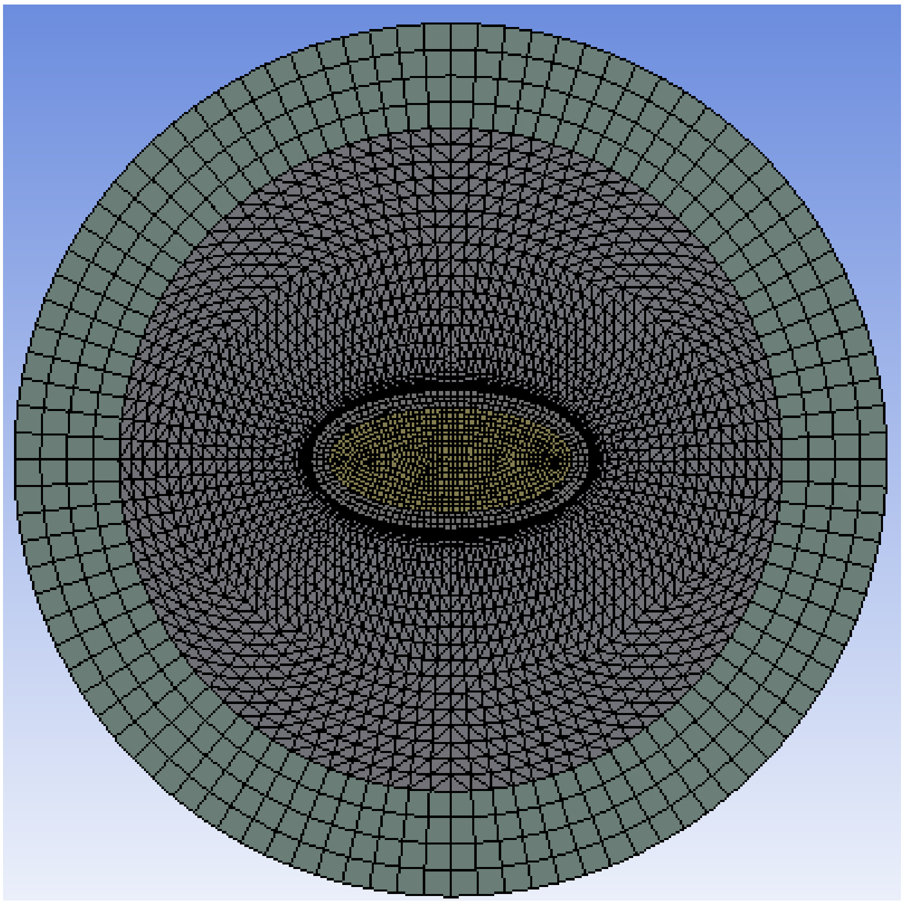

Figure 7: Facial grid layout of mesh domain

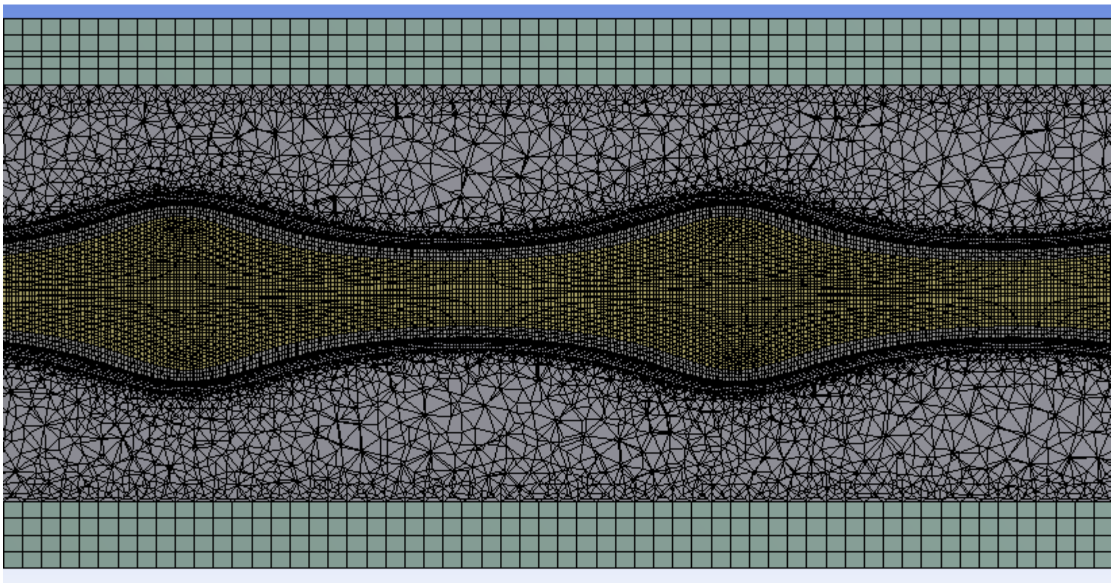

Figure 8: Cross section grid layout of mesh domain

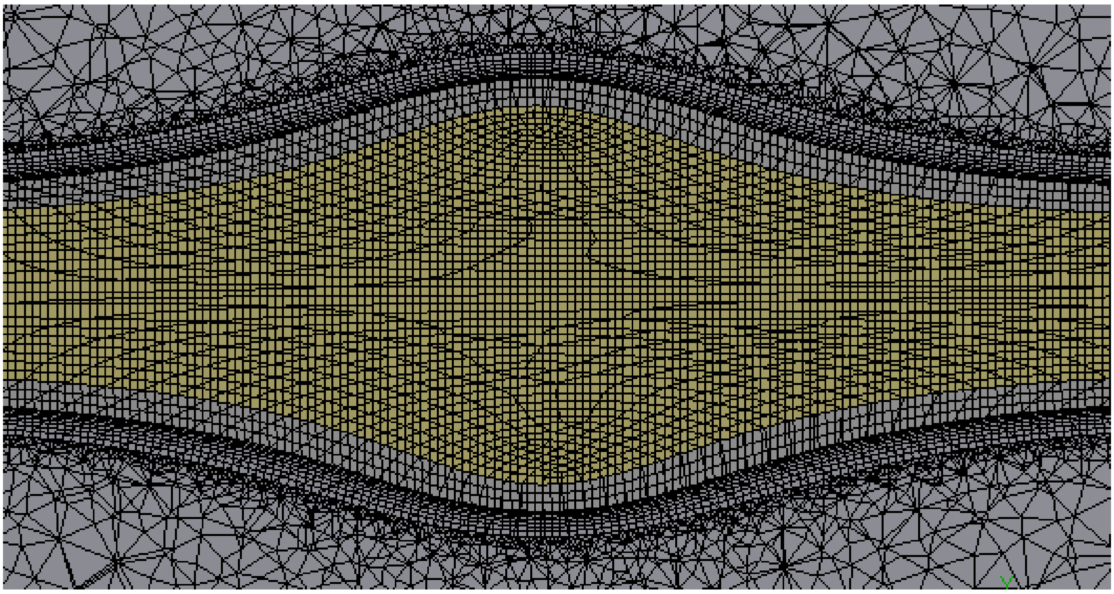

Figure 9: Inner twisted tube cross section grid layout of mesh domain

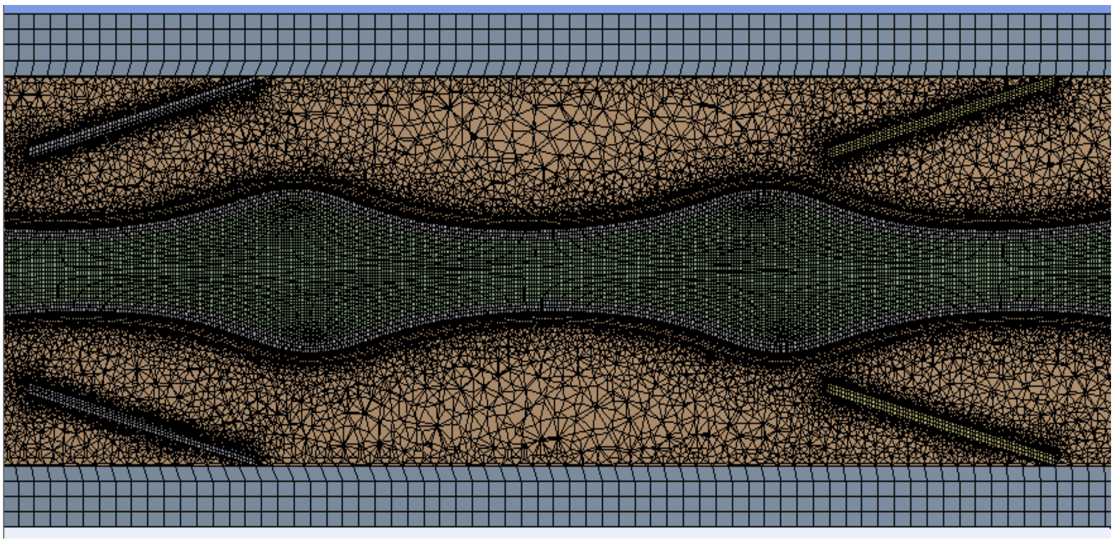

Figure 10: Cross sectional grid layout of the mesh domain with conical ring turbulator

4.2 Governing Equations

In CFD simulations, the K-epsilon turbulence model is commonly employed. It is based on the Reynolds-averaged Navier-Stokes equations (RANS), which explain the average motion of fluids. In turbulent flows, the dissipation rate (epsilon) and kinetic energy (K) are calculated using a two-equation model, which is the K-epsilon model. [28]

The K-epsilon model assumes that turbulent flow is isotropic, which means that it acts the same in every direction. It also assumes that the turbulence has reached a steady state and is not changing over time. The model tracks how the turbulent flow changes by using two transport equations, one for K and one for epsilon.

The K-epsilon model is known for a wide range of applications which is also validated with significant number of dispensation, such as turbulent boundary layers, wakes, and jets. Nonetheless, it has a few drawbacks. Flows with severe shear or turbulence are out of the question.[29] Complex turbulence models may be necessary to explain the fluid flow in such situations.

From the specified initial and boundary conditions, K and epsilon are often computed using a numerical technique like the finite volume or finite element approach. The mean velocity and pressure fields of the fluid flow can be obtained by solving the RANS equations once K and epsilon have been determined by the eddy viscosity hypothesis. [30]

In general, the K-epsilon turbulence model is a straightforward and efficient method for simulating turbulent flows. In more complex flows, however, more sophisticated turbulence models may be required

The governing equation of CFD for 3-D incompressible flow for Newtonian fluid are [31]

Continuity Equation:

The continuity equation for mass conservation has the following form

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

Momentum Equation:

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial P}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} + \rho g_i \qquad (2)$$

Energy Equation

$$\frac{\partial}{\partial t}(\rho e) + \frac{\partial}{\partial X_j}(\rho e u_j) = -\frac{\partial}{\partial X_j}(Q_j) + \frac{\partial}{\partial X_j}(\tau_{ij} u_i) + \rho g_i u_i \qquad (3)$$

Here,

u_i= velocity in x direction in tensor form

t= time

v_j= velocity in y direction in tensor form

p= pressure

ρ= density

τ= viscous shear stress

e= energy dissipation

4.3 Data Processing

It is possible to determine the Nusselt number (Nu) and the friction factor (f) based on the temperatures at the heat exchanger's inlet and exit, as well as the surface temperature of the inner pipe. The Performance Enhancement Factor (PEF) is calculated using these numbers. The PEF is a performance improvement statistic used to examine the effectiveness of various internally twisted pipe layouts.

The transfer of heat from the cold fluid to hot fluid can be calculated by using equation below:

$$q_c = \dot{m}_c C_{p,w}\left(T_{c,out} - T_{c,in}\right) \tag{4}$$

whereas the temperature drop of the hot water is as follows:

$$q_h = \dot{m}_h C_{p,w}(T_{h,out} - T_{h,in}) \tag{5}$$

where the mass flow rates of cold and hot water, $m_c$and m_h, $C_{p,w}$ is the specific heat of water. The cold and hot water input and output temperatures are denoted as $T_{c,in}$, $T_{c,out}$, $T_{h,in}$, and $T_{h,out}$, respectively.

Between these two extremes, find the average heat rate (q_{avg}) as

$$q_{avg} = \frac{(q_c + q_h)}{2} \tag{6}$$

whereas the inner pipe's average surface temperature ($T_{s,avg}$) is found by taking the average of the k number of iterations temperatures taken at various points inside the tube. [32]

$$T_{s,avg} = \frac{1}{n}\sum_{i=1}^{n=k} T_{s,i} \tag{7}$$

Bulk temperature ($T_{b,h}$) is also known to be the mean of the hot water stream's input and outflow temperatures, as shown in the following equation:

$$T_{b,h} = \frac{(T_{h,in} + T_{h,out})}{2} \tag{8}$$

The hot water's coefficient of convective heat transfer (h_h) can be calculated as:

$$h_h = \frac{q_{avg}}{A_{s,in}(T_{b,h} - T_{s,avg})} \tag{9}$$

Nusselt number is calculated by using

$$Nu_h = \frac{h_h d_h}{k} \tag{10}$$

where$A_{s,in}$, d_h, and k represent the inner pipe's surface area, hydraulic diameter, and thermal conductivity, respectively.

Reynolds number is calculated by using

$$Re = \frac{\rho U d_h}{\mu} \tag{11}$$

Subsequently, the Reynolds number is employed in combination with either empirical correlations or theoretical equations to determine the friction factor. For fully developed turbulent flow in twisted pipes, the Swamee-Jain equation is a commonly used empirical correlation. [33]

$$f = \frac{0.25}{[(\log(\frac{\frac{\varepsilon}{D}}{3.7}) - \frac{5.74}{Re^{0.9}})^2]} \tag{12}$$

Here ε is the surface roughness of the pipe, D is the equivalent diameter of the pipe.

The mean velocity within the inner pipe is determined by dividing the measured flow rate of hot water by the cross-sectional area, denoted as A, of the inner pipe, in the following way [34]

$$U = \frac{\dot{m}_h}{\rho A} \tag{13}$$

Any increase in heat transfer rate achieved through passive means will, as noted above, cause a noticeable drop in pressure. Therefore, proposals to increase heat transfer rates in heat exchangers must also consider the effects of pressure drops. To measure the benefits of increased heat transmission against the costs of lower pressures, researchers have developed a dimensionless indicator called the pressure effectiveness factor (PEF). The performance efficiency factor is calculated by using

$$PEF = \frac{\frac{Nu_h}{Nu_{h,0}}}{(\frac{f}{f_0})^{\frac{1}{3}}} \tag{14}$$

where the Nusselt number and friction factor for the original pipe (plain) are denoted by $Nu_{h,o}$and f_o respectively. Simply put, when PEF>1, pressure losses due to the increased heat transfer rate are tolerable. But if PEF<1, then the pressure reduces significantly despite the improved heat transfer rate. [35]

4.4 Grid Independency Test

In order to solve complex partial differential equations on fluids case studies, several computational methods rely on grid formation or meshing of the physical domain. Both structured and unstructured mesh domains have been used for the grid independency study that is being conducted here. The outcomes of the grid independence test are presented in the following at fig. 11.

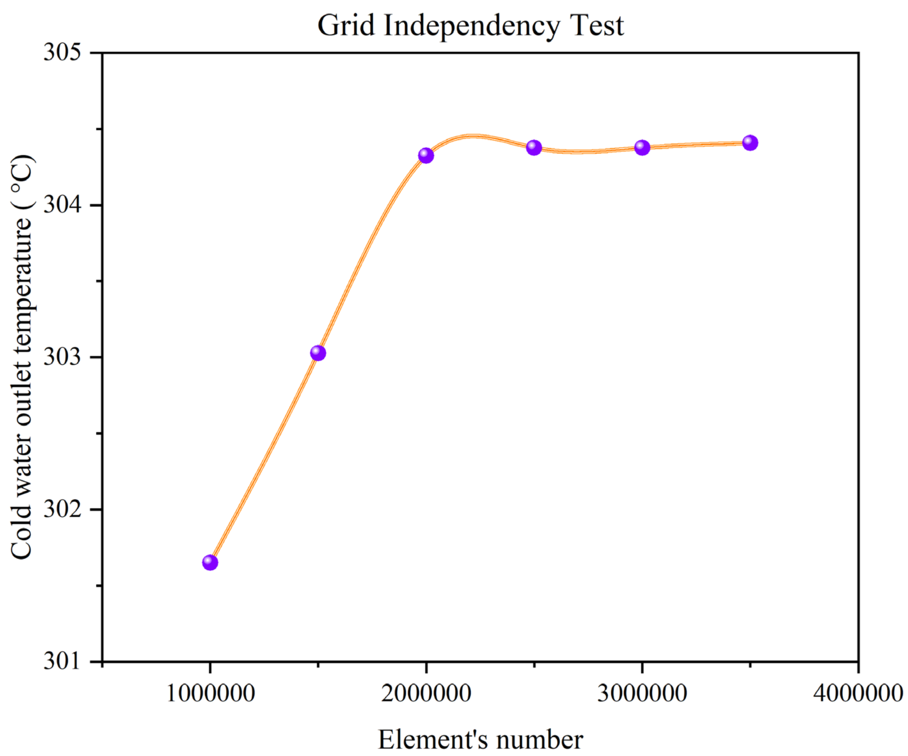

Figure 11: An evaluation of the grid independency using the cold-water outflow temperature for the number of elements.

Total number of elements on the grid domain is 3.5 million and there are 1.95 million of nodes. The grid independency test has done by focusing of the cold water outlet temperature and it is found that after 2.5 million elements the result is almost unchanged. This shows that, having more element beyond 2.5 million does not have any effect on the temperature of cold water outlet. So, in this study, 2.5 million elements was used for simulation.

4.5 Model validation

A validation study is performed in relation to the experimental study that was carried out by M. H. A. and R. E. Jalal [1] in order to guarantee the accuracy of the existing numerical model. Their work is based on same turbulent region with water as an working fluid which satisfy the working condition of this numerical study.

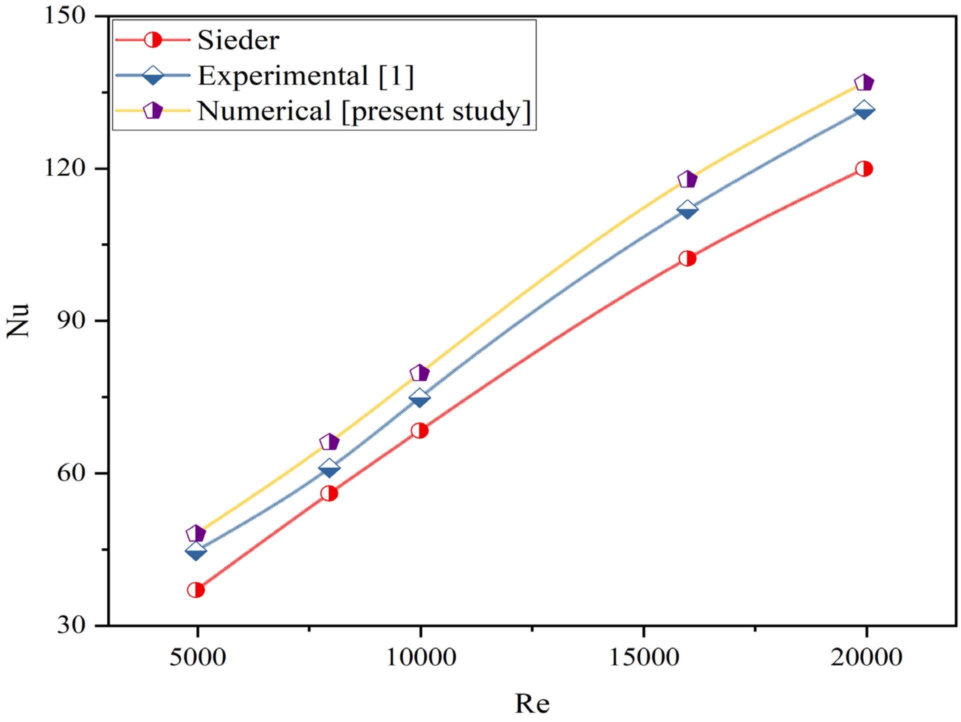

Figure 12: Validation of the numerical value with experimental data

This numerical study is validated with an experimental work. In fig. 12 it is shown that the Nusselt number of the hot fluid is increasing gradually with the increase of its corresponding Reynolds number that means the turbulence of the flow is getting higher. There is acceptable error founded between the numerical and experimental work. The average deviation from the experimental data is 2.1%.

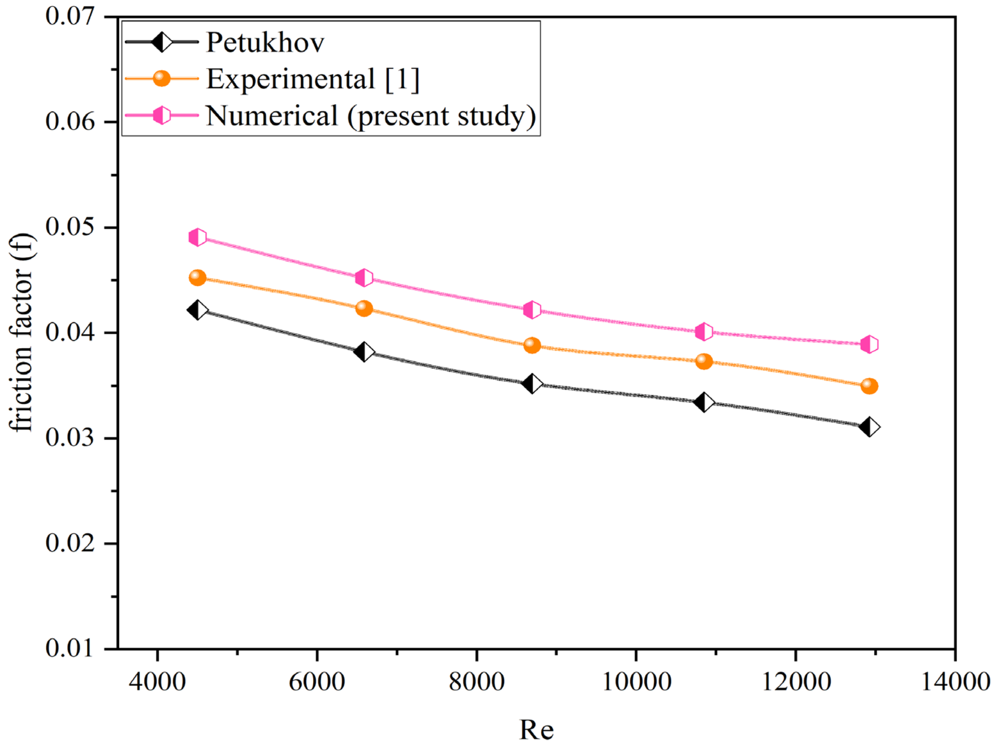

Figure 13: Validation of numerical value with experimental & empirical data

Here is the comparison between numerical investigation of present study and experimental data along with empirical data showed in fig. 13. Friction factor decreases with the increase of Reynolds number within the range. The average deviation from experimental data compared with numerical one is about to 2.8%.

Chapter – 5: Results and Discussions

Once the inner elliptical pipes of a DPHE are designed with a twist, heat exchanger's over all performance is investigated and analyzed through numerical simulations. The simulations are conducted using water as the working fluid, considering both parallel and counter flow configurations. Five different designs for the inner pipe are examined, including a complete twist, three twists, and the inclusion of two, four, or six conical rings per unit length with a full twist in the inner pipe. The conical rings are introduced in the outer pipe to guide the flow towards the inner pipe. The effect of whole twisting and inserting the conical ring in the annular section can be evaluated by comparing the value of heat transfer and also the change in pressure drop across the pipe.

To get a Reynolds number between 5000 and 26,000, The hot water flow rate can be adjusted between 2 to 10 L/min while the cold water flow rate is kept constant at 20 L/min.

The input temperature for the hot water stream was similarly maintained constant at 50°C. The steps involved in confirming and interpreting the results are outlined below.

5.1 Effect of full twist and conical ring on heat transfer performance

The variation of heat transfer and the change in hot fluid outlet temperature is clearly visible from the simulation. It can be observed that all the conical ring configurations provides better heat transfer rate than full twist with compared to plane elliptical tube. Cold fluid entering from the right side and hot fluid is entering from the left side through the inner tube. In case of (a) the temperature at hot outlet reduces by 4 °C for case (b) it is reduces by 5.2 °C further at full twist (c) temperature reduces by 7.2 °C. At the section of conical ring insert for each pair of conical ring, the temperatures dropped by 1 °C. ultimately the DPHE with six conical ring gives a decent temperature reduction at hot outlet which is 314 °C.

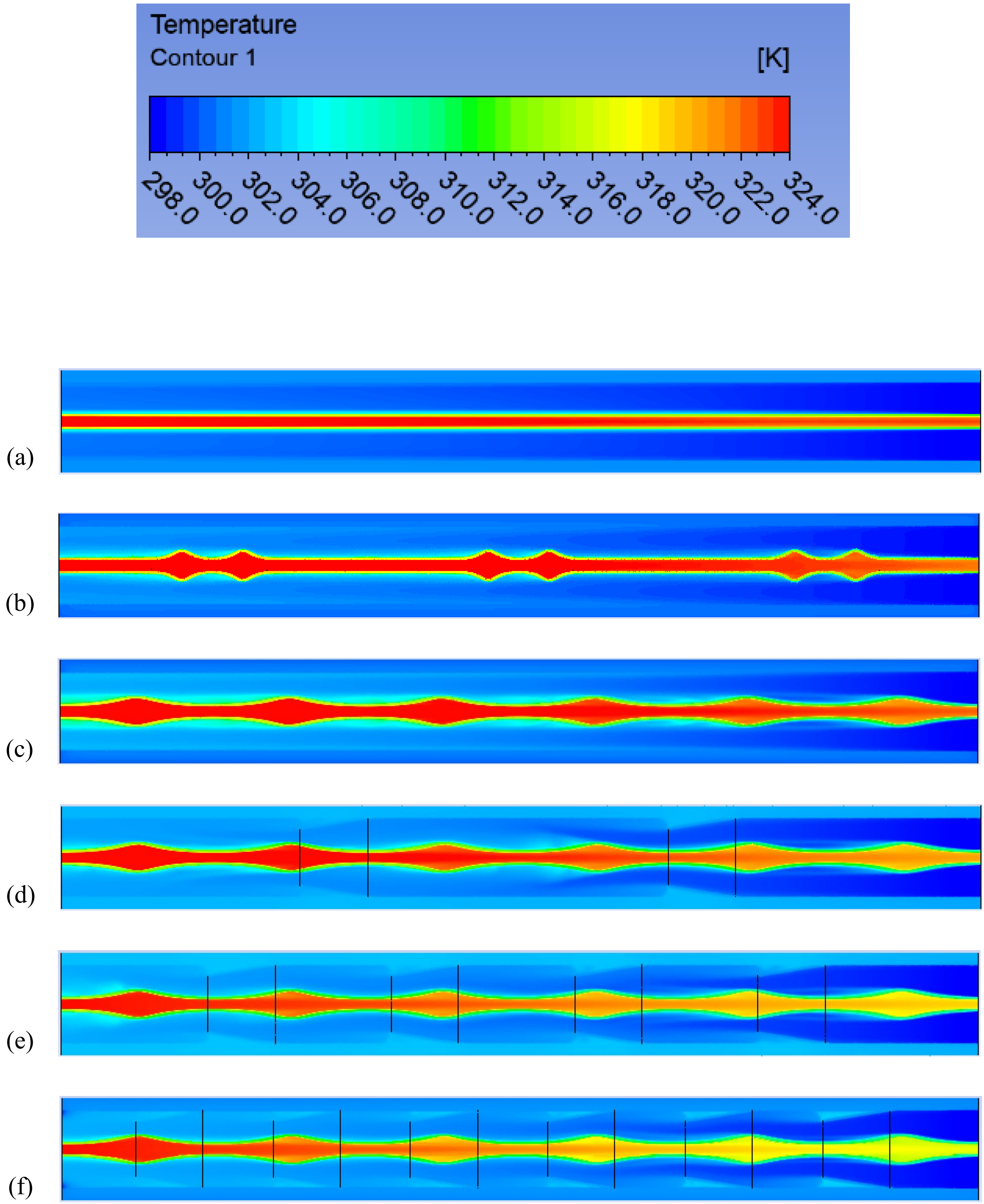

Figure 14: Temperature Contours of DPHE Computational domain

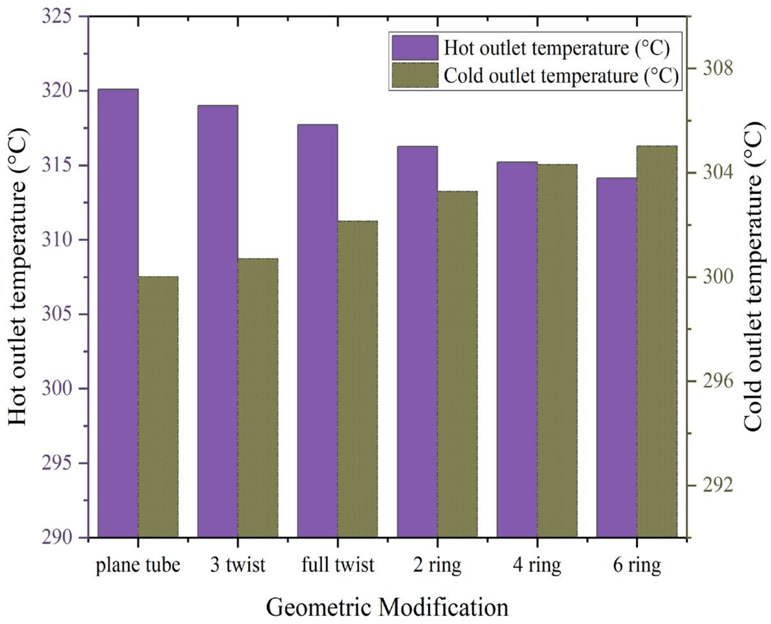

Figure 15: Variation in outlet temperature of hot and cold fluid

The temperature of outlet hot fluid reduces with the increase of twist and conical ring simultaneously and the cold fluid temperature also increased. Having six conical ring with full twist(f) showed maximum temperature reduction at outlet of the hot fluid that is 10 °C and the temperature of the cold fluid is increased by 7 °C for the same geometrical modification at fig. 15.

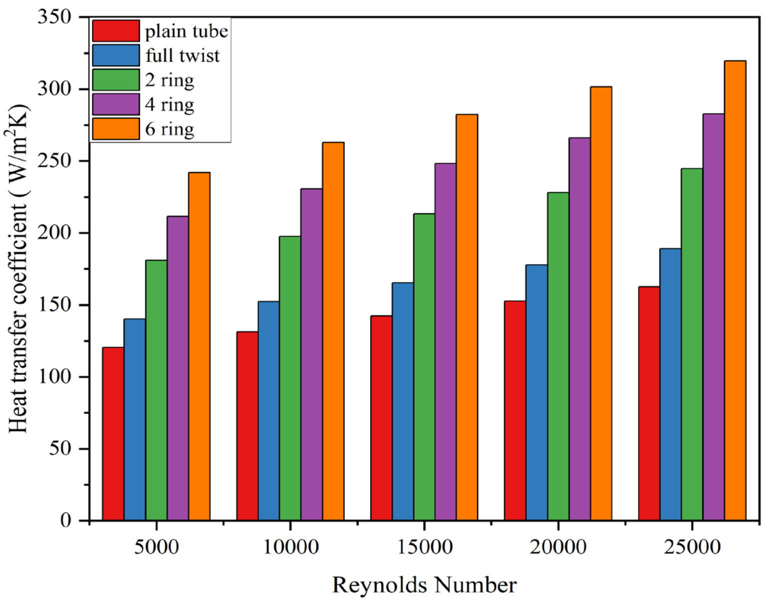

Figure 16: Heat transfer coefficient for corresponding Reynolds number

At fig. 16 the coefficient of heat transfer has calculated with corresponding Reynolds number, With the increase of the Reynolds number the heat transfer coefficient also increased. Nu rises as a direct outcome of more rapid rates of convective heat transfer. The twisting of the pipes causes an increase in the flow's swirl and turbulence, which in turn contributes to the increased convection. This For all Reynolds numbers, the pipe with complete twists depicts the bigger Nusselt number, and the same holds true for the six conical rings. Maximum heat transfer coefficient is 310 w/m^2k per unit length.

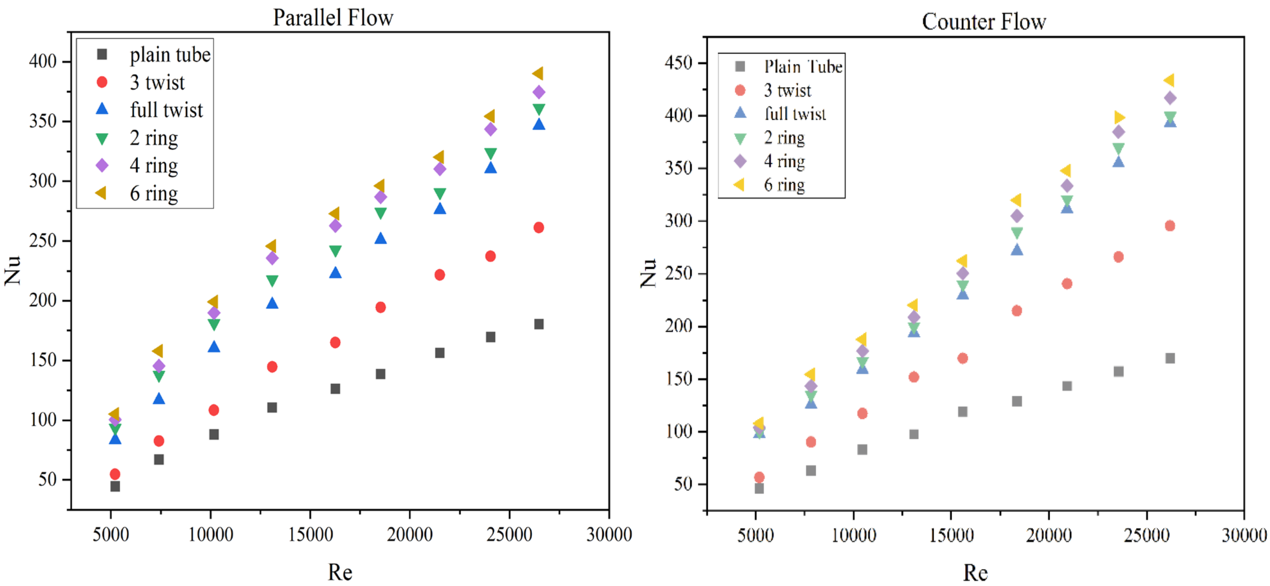

Figure 17: Effect of full twist on Nu and its variation with Re for parallel and counter flow.

The effect of nusselt number has showed for the both direction of flow over the change of Reynolds number. The results are displayed in fig. 17 that illustrates that for all investigated pipe designs, Nusselt number increases with increasing of Reynolds number for both flow directions. Nu rises as a direct outcome of more rapid rates of convective heat transfer. The twisting of the pipes increases the swirl and turbulence in the flow, which in turn increases the convection.For the same rationale, the pipe with complete twist demonstrates the bigger Nusselt number for all Reynolds values. Nusselt number values are similarly close between the 3-twist and full-twist pipes over the whole range of Reynolds numbers. Fig. 17 depicts the proportional increase in heat rate due to the presence of twisting as well as conical ring. For parallel flow, the Nu is increased by around 1.4 to 1.9 times for the 3 twist and full-twist pipes, respectively compared to the plain pipe. For the counter flow that is onward to 1.6 to 2.3 times with compared to plain tube. Maximum value of the nusselt number is 390 for the parallel flow and 445 for the counter flow observed on the full twist with conical ring turbulator.

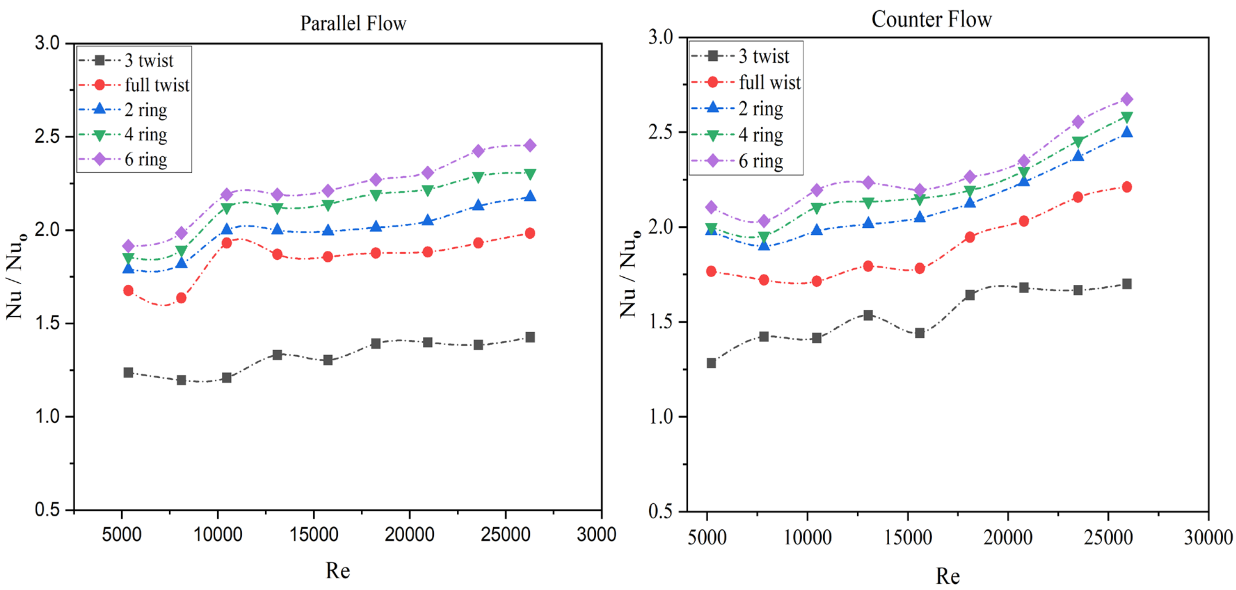

Figure 18: Effect of increment on twist and conical ring on ${Nu}/{Nu_0}$ at different Re for counter & parallel flow

The value of the nusselt number from each type of flow is compared with the nusselt number of the plain tube(Nu_0) to see the ratio enhancement for the DPHE. Nusselt number enhancement ratio for counter flow ranges between 1.40 and 2.20 for Re > 22,000. and 2.48, with highest values around 2.65 that is founded in full twist with 6 conical ring model. The results confirm that a DPHE can benefit from a fully twisted inner pipe along with some conical ring turbulator for increasing the rate of heat transfer. Maximum improvement of nusselt number ratio has noticed for the fully tiwsted inner pipe with six conical ring for counter flow at fig. 18.

5.2 Effect of conical ring on friction factor

For all pipes examined, the frication factor is highest with lower Re and decreases with increasing Re. No matter how many turns are put into a twisted pipe, the friction factor will always be higher than it would be with a straight pipe of the same Re.

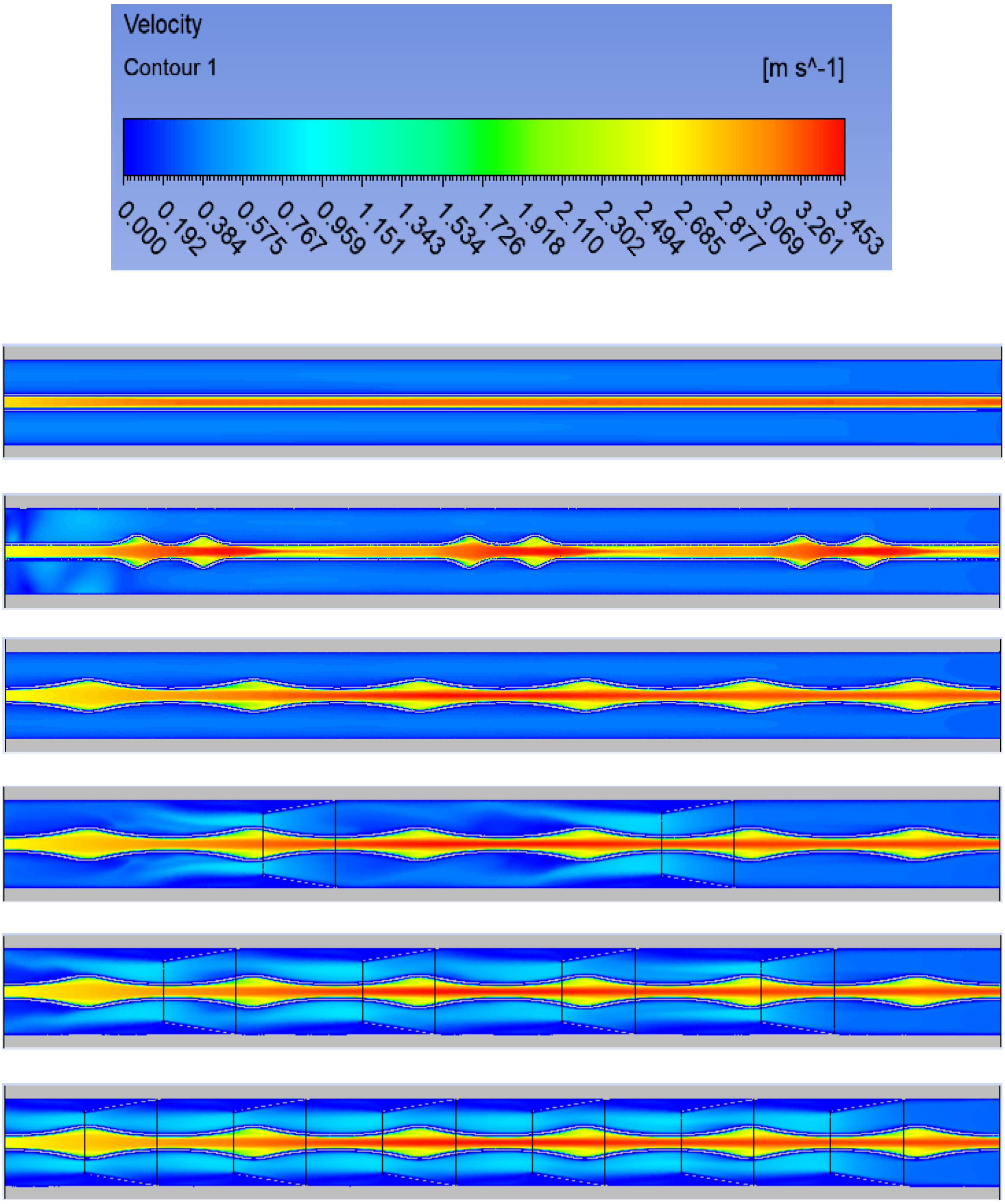

Figure 19: Velocity contours of DPHE computational domain

From velocity contour it shows that the maximum velocity arises in the region of twisting side inside inner tube and the overall maximum average velocity increases in the full twist with six conical rings (last case) in Fig 19. From the darcy weisbach equation of turbulent flow [36]

$$h_l = \frac{fLv^2}{2gd}$$

From this equation it is found that if the velocity of the flow is increases then the friction factor decreases as well as when the velocity increases of the flow then the Reynolds number also increases which is verified by [37]

$$R_e = \frac{\rho VD}{\mu}$$

so with the increase of velocity the flow becomes more turbulent and friction factor decreases with it.

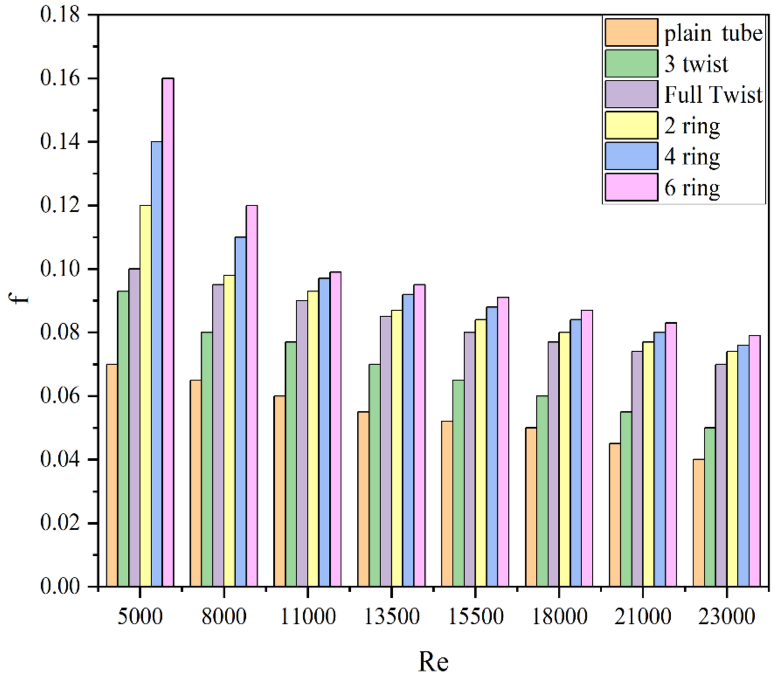

Figure 20: Effect of twist and conical ring on friction factor at different Reynolds number

The value of friction factor is 0.16 which is maximum for the six ring fully twisted pipe with Reynolds number 5000 and the minimum value of the friction factor is noticed for the Reynolds number 23000 that is shown at fig. 20.

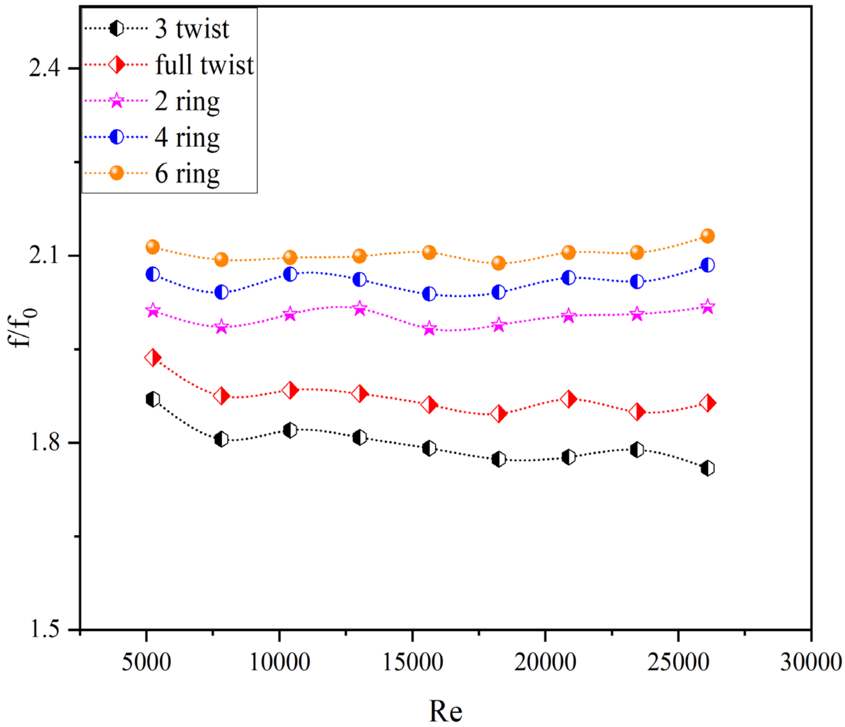

Figure 21: Effect of twist and conical ring on f/f_0 at different Re.

The value of the friction factor from each type of flow is compared with the friction factor of the plain tube(f_0) to see the ratio enhancement. However, in accordance with the results in References [1] [38] [39] [40] [41] find that the friction factor variation at all R_e rises as the number of twist increase gradually from 3 to full. Flow resistance increases and the friction factor rises when the inner pipe of a twisted pipe becomes longer as the twist number rises. For the proper mixing of fluid along with generating more turbulence, which both contribute to increased friction between fluid layers, twisting also creates flow swirl. Up to the point approximately Reynolds number 20,000 the friction factor ratio (f/f_0) exhibits the similar behavior as fig. 21. However, nearly consistent values are seen for Re > 20,000. When comparing the full-twist pipe to the three-twist pipe, the full-twist pipe has a higher friction factor. Friction ratios for both 3 and full-twist pipes are close to 1.9 (it may go lower if the Reynolds number is more than 50,000). For a pipe with six conical rings, the friction factor is consistently greater for full twists, with values ranging from about 1.7 for a Reynolds number of 5000 to under 2.17 for higher Reynolds numbers.

5.3 Effect of full twist with conical ring on pressure drop

Pressure drop is a significant criteria which need to give priority for the design of a heat exchanger. The designer is often concerned in the pressure drop necessary to maintain the flow, since this parameter influences the amount of power that must be supplied to the pump or fan. Fig. 22 illustrates how the pressure drop on the annulus-side (Δp) varies with respect to R_e for the various configuration that were tested.

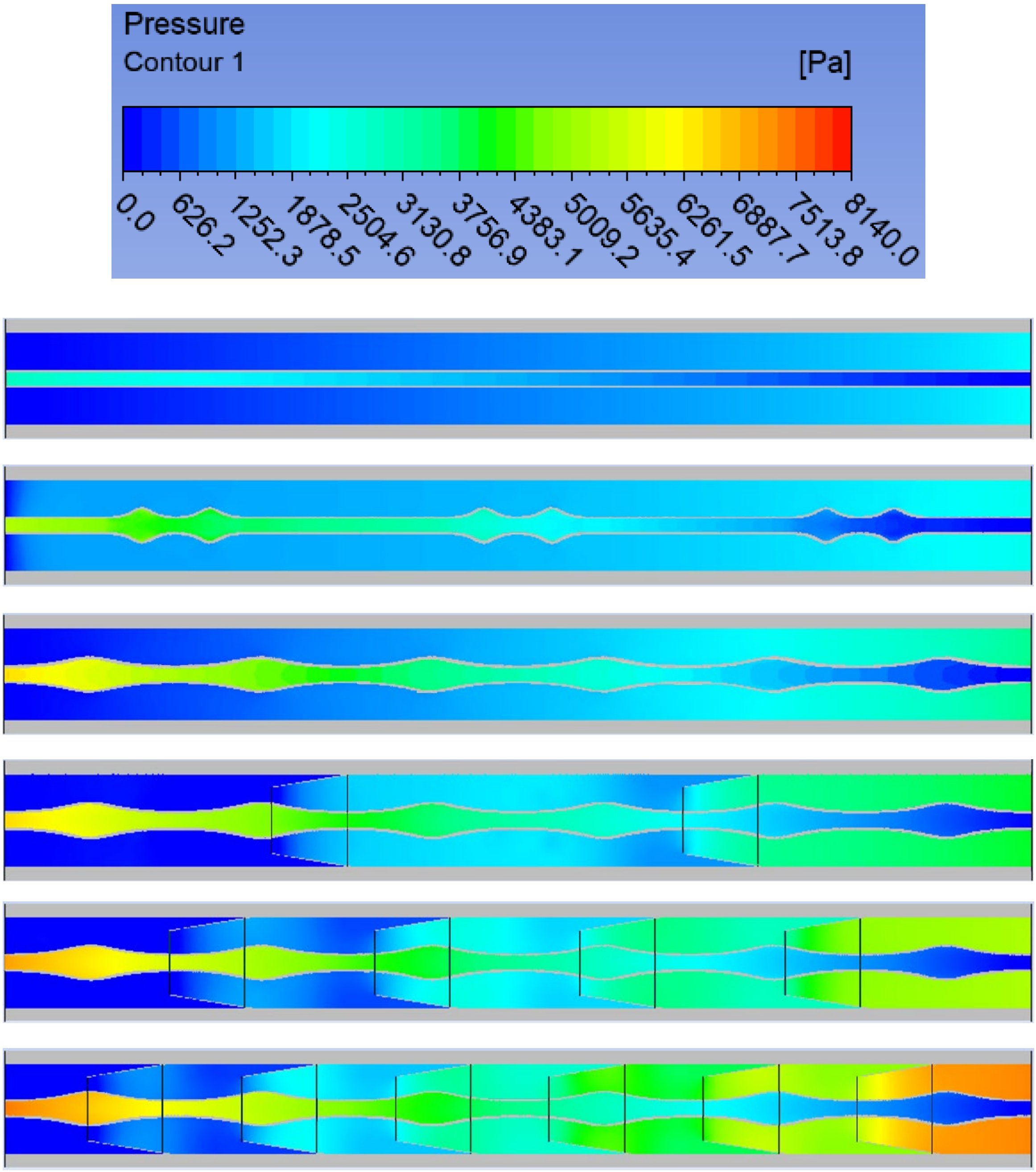

Figure 22: Pressure contour for DPHE computational domain

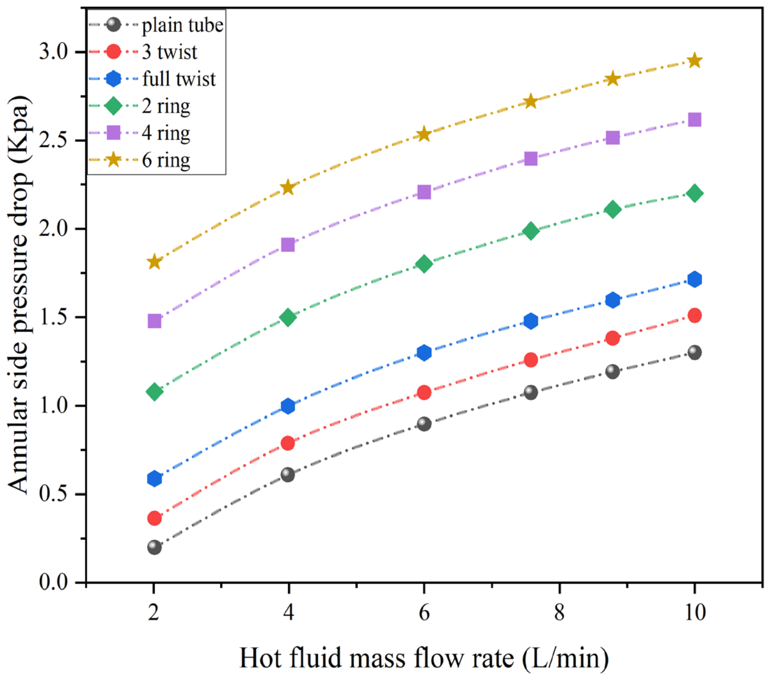

Figure 23: Annular side pressure drop of DPHE.

Annular side pressure drop has gradually increased for the hot fluid mass flow rate. More pressure drop is noticeable for the six ring cases with full twist that is close to 3 kPa and the minimum pressure drop occurs at plane tube. When the flow rate changes from 2 to 10 L/min the pressure drop also rises. It is very common to get this result as the six ring consist more irregular geometry rather than plain tube. The decrease in pressure is proportional to the square root of the Reynolds number (Re) as well as the smoothness of the pipe. The following variables are responsible for this significant reduction in pressure: 1) Increased velocities on the annulus side as a direct result of the twisted tube's shape. 2) an abrupt shift in the flow field and increased peak velocities in the region immediately next to the entry 3) an increased surface area for the flow in the annulus with compared to plain tube

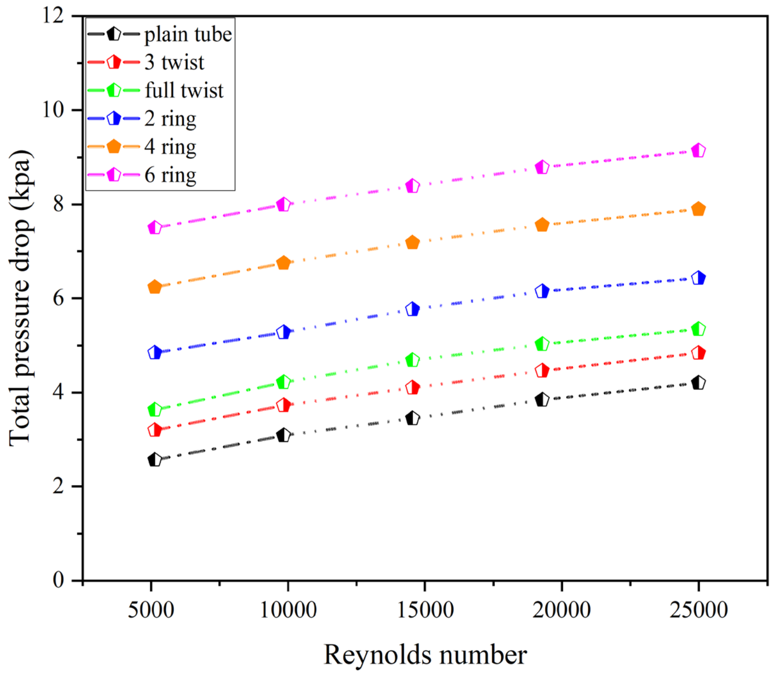

Figure 24: Total pressure drop of DPHE with different Re

Figure 23 shows the pressure drop along the axial flow direction for a range of Reynolds numbers from 5000 to 25,000 along the full length of the pipe. Maximum velocities occur in the inlet zone due to the abrupt shift in the flow pattern, and this is the primary contributor to the large pressure drop. Therefore, in order to maintain the same mass flow rate through the inlet zone, the velocity of the annulus fluid must increase. All of these factors play a role in the significant pressure drop, but the maximal velocities in the entrance region are the main culprit. For the six-ring conical ring with a fully twisted tube, the maximum pressure drop was determined to be around 8.14 kPa at a Reynolds number of 25000 at the input face of the hot fluid and the exit face of the cold fluid. Using a heat exchanger with a twisted inner pipe results in substantially larger reductions in pressure compared to using a plain pipe. As a result, a greater amount of pumping force is necessary to maintain the flow within the pipes. Therefore, in order to establish justification for the use of twisted pipes, a considerable improvement in heat transmission should be accomplished.

5.4 Effect of conical ring on fluid flow characteristics

Fluid is flowing smoothly in the plain tube where as in twisted tube swirl motion is created over annular tube both inside and outside as a result more turbulence has created there. At outside of the annular ring different number of conical rings has inserted which cause the flow more converged towards the inner twisted pipe that has great cooling effect inside the DPHE.

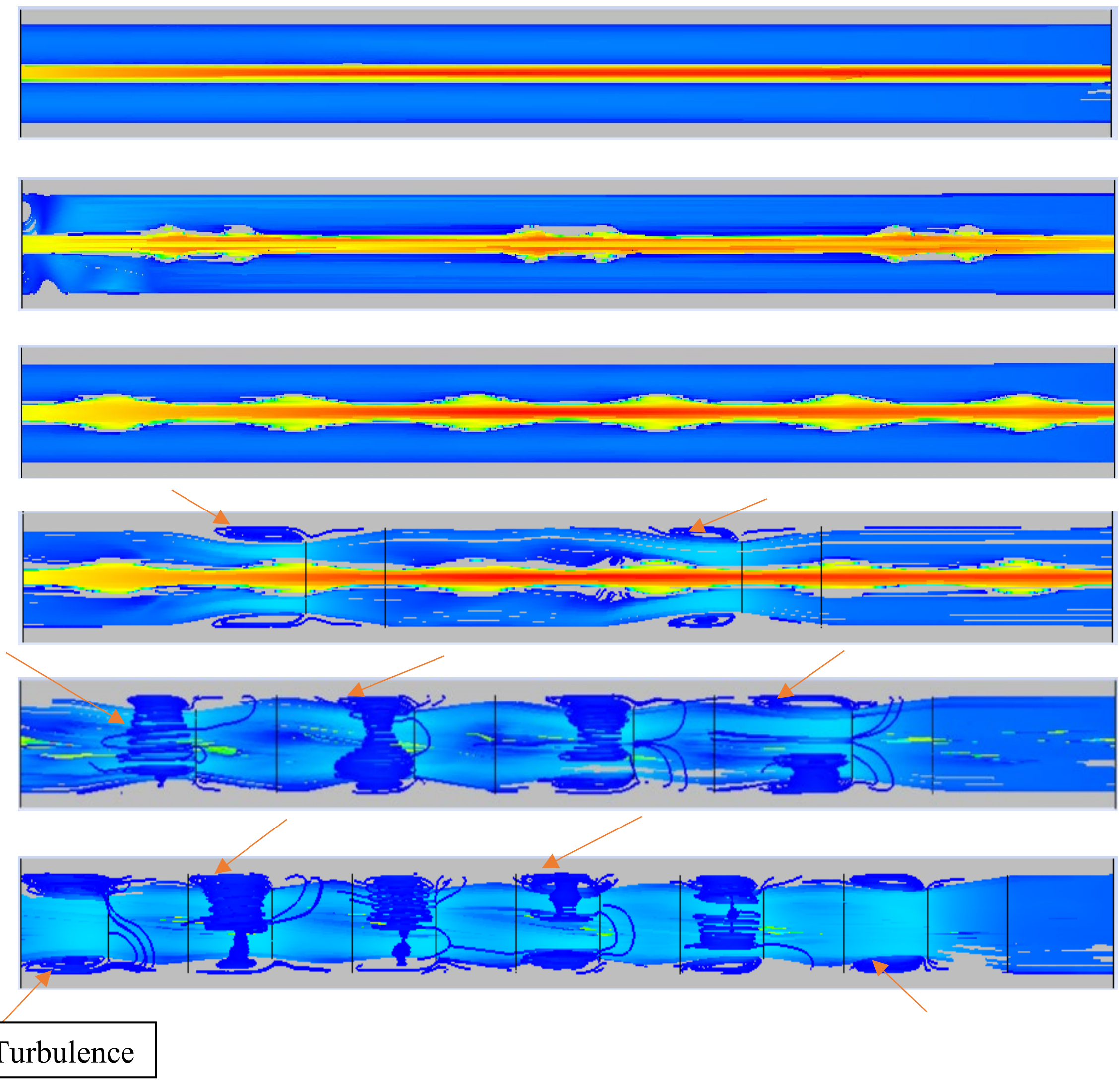

Figure 25: Velocity streamline of the flowing fluid

The presence of eddies, which are intense and quick fluctuations of groups of fluid particles, can be attributed with an increase in the convective heat transfer rates. The primary effects of turbulence on heat transfer is the creation of a turbulent boundary layer near the pipe wall. This layer is thicker and more energetic than the laminar boundary layer, which allows for more heat transfer from the pipe wall to the fluid. Additionally, the turbulence promotes the mixing of fluid particles, which can help to distribute the heat more evenly across the flow cross-section.

Another effect of this turbulence is the generation of secondary flows, such as swirls and eddies, that can enhance the mixing of fluid and further increase the heat transfer rate. These secondary flows are caused by the interaction between the fluid and the wall, and they can lead to significant heat transfer enhancements in this case.

The visible result on the fluid flow having high turbulence is noticed just after the conical ring. In the cold fluid region maximum turbulence has formed due to low pressure at that region. The arrow mark on the fig. 25 indicated the turbulence form after the conical ring, by increasing the number of conical ring indicates more turbulence that provides better heat transfer.

5.5 Effect of conical ring on the overall improvement in performance

PEF stands for "Performance Enhancement Factor", which is a dimensionless parameter used to quantify the effectiveness of a heat exchanger. In a double pipe heat exchanger, the PEF is used here to evaluate the heat transfer enhancement achieved by inserting the twisted pipe and conical ring turbulator outside of the twisted pipe of the heat exchanger. The PEF is the ratio of the total heat transfer coefficient with the insert to the overall heat transfer coefficient when the insert is not present. This ratio is defined as the PEF. Simply said, it is a measurement of how much the twisted pipe and inserted ring turbulator contributes to an increase in the heat transfer rate within the heat exchanger.

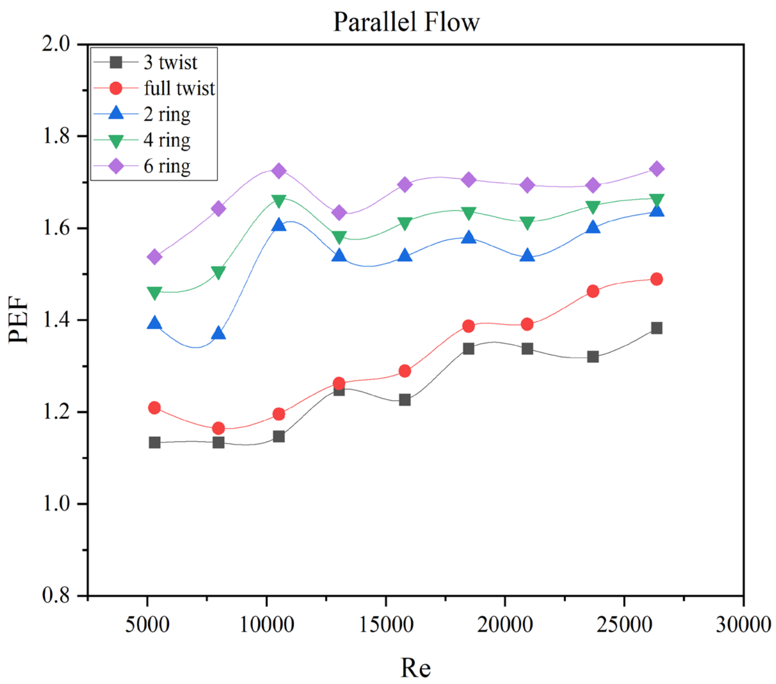

Figure 26: Effect of full twist &conical ring on overall performance for parallel flow

According to the findings of an investigation into the effects of inner twisting pipe on flow characteristics as well as the rate of heat transfer that was carried out and given in the two subsections that came before this one, both the rate at which pressures decrease and the rate at which the Reynolds number increases, heat transfer improves. To demonstrate that pipe twisting was worth the effort, there must be a discernible improvement in the operation as a whole. The total performance index, denoted as PEF, is calculated using Equation (16), and its representation in fig. 26 & 27 is shown for both flow directions. It is demonstrated that for both types of flow, the PEF grows larger as the Reynolds number rises, and that the

PEF is always bigger than 1. This is the case regardless of the flow type. This demonstrates that the rise in the heat transfer rate is far more significant than the decrease in pressure, and as a result, the total performance will improve. Additionally, it can be shown that the pipes with three twists and those with complete twists have a performance that is quite similar for practically for all Reynolds numbers and for both flow directions. In parallel flow full twist with conical ring shows maximum PEF at $R_e = 11000$ that is approximately 1.7

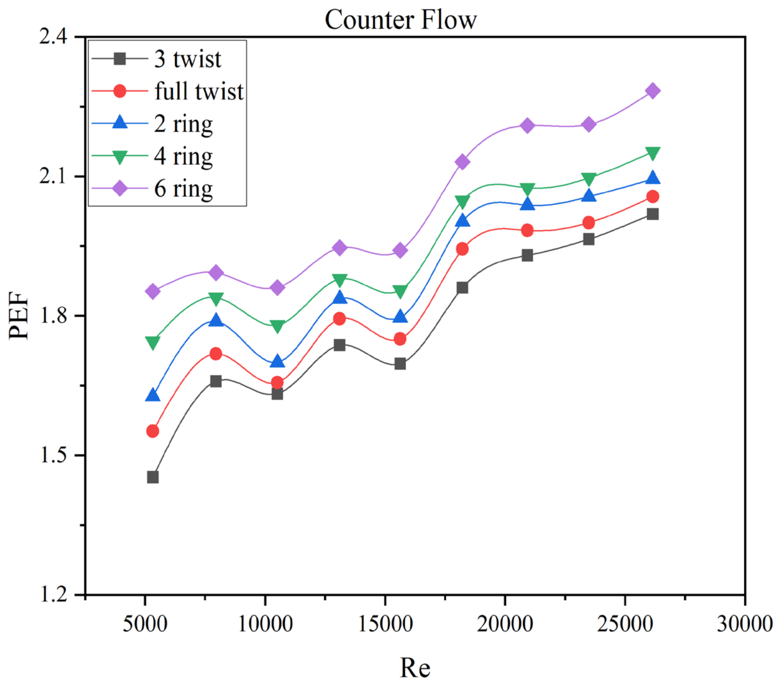

Figure 27: Effect of twist and conical ring on overall performance (PEF) for counter flow

For the counter flow design the value of PEF reduces for $R_e = 11000$ and that increases gradually for the increment of Reynolds number. For this entire study in each case full twist with six conical ring shows maximum performance increment for the both parallel and counter flow type and most PEF is observed at $R_e = 26000$ which is 2.3 at counter flow direction. So it is recommended that pipe with full twist and six conical ring gives maximum performance enhancement along with high heat transfer rate and pressure drop.

Chapter – 6: Conclusion

In this numerical study a double-pipe heat exchanger with an fully twisted inner pipe is studied along with conical ring turbulator for its overall performance. The process is carried out with water as the working fluid through six pipes of varying twist counts per unit of length. Every simulation is compared with one another. when comparing its performance in the turbulent flow zone (Reynolds number ranging from 5000 to 26,000) to the plane tube without any turbulator, with the gradual increment of twist and conical ring turbulators, the highest and most scrumptious level of overall performance is achieved. Consideration is given to both parallel and counter flows. In light of the findings, the following conclusions can be drawn:

1. As Re grows, friction lowers and heat transfer accelerates, independent of the twisting values. The friction factor ratio is nearly constant for the whole twisted pipe.

2. The values of Nu and f grows with the amount of twisting done to the DPHE's inner pipe, which is regarded to be Re. Both Nu and f increase with the number of twists per unit of length. The maximum value of Nu & Re observed at full twist of length.

3. The conical shape ring turbulator helps to converge the flow of cold fluid towards the hot inner pipe which ensures better surface contact than without conical ring.

6.1 Future Recommendation

Different types of fluid can be used as cold fluid like ethylene glycol or nano fluids. Further different shapes of turbulator can be integrated to observe less pressure drop. Different angles of inner pipe setup can be investigated to obtain better PEF. Different arrangements(direction) of conical ring can be used.

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