Numerical Investigation of Heat Transfer Characteristics of a Double-Pipe Heat Exchanger using Inner Corrugated Tube and Perforated Elliptical Turbulator

Submitted By

Mahia Afrin

200011117

Mir Tasnuva Sara

200011130

Supervised By

Prof. Dr. Md. Rezwanul Karim

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

Department of Mechanical and Production Engineering (MPE)

Islamic University of Technology (IUT)

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Candidate's Declaration

This is to certify that the work presented in this thesis, titled, "**Numerical Investigation of Heat Transfer Characteristics of a Double-Pipe Heat Exchanger using Inner Corrugated Tube and Perforated Elliptical Turbulator**", is the outcome of the investigation and research carried out by me under the supervision of Dr. Md. Rezwanul Karim, Professor, MPE Dept., IUT, Board Bazar, Gazipur-1704, Bangladesh.

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.

Mahia Afrin
Student No: 200011117

Mir Tasnuva Sara
Student No: 200011130

Recommendation of the Thesis Supervisors

The thesis titled "**Numerical Investigation of Heat Transfer Characteristics of a Double-Pipe Heat Exchanger using Inner Corrugated Tube and Perforated Elliptical Turbulator**" submitted by MAHIA AFRIN, Student No: 200011117 and MIR TASNUVA SARA , Student No: 200011130 has been accepted as satisfactory in partial fulfillment of the requirements for the degree of B Sc. in Mechanical Engineering **on 29th September, 2025.**

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   Dr. Md. Rezwanul Karim (Supervisor)
   Professor
   MPE Dept., IUT, Board Bazar, Gazipur-1704, Bangladesh.

CO-PO Mapping of ME 4800 -Theis and Project

COs	Course Outcomes (CO) Statement	(PO)	Addressed by
CO1	Discover and Locate research problems and illustrate them via figures/tables or projections/ideas through field visit and literature review and determine/Setting aim and objectives of the project/work/research in specific, measurable, achievable, realistic and timeframe manner.	**PO2** **Problem analysis**	Thesis Book Performance by research Presentation and soft skill
CO2	Design research solutions of the problems towards achieving the objectives and its application. Design systems, components or processes that meets related needs in the field of mechanical engineering	**PO3** **Design/ development of solutions**	Thesis Book Performance by research Presentation and soft skill
CO3	Review, debate, compare and contrast the relevant literature contents. Relevance of this research/study. Methods, tools, and techniques used by past researchers and justification of use of them in this work.	**PO4** **Investigation**	Thesis Book Performance by research Presentation and soft skill
CO4	Analyse data and exhibit results using tables, diagrams, graphs with their interpretation. Investigate the designed solutions to solve the problems through case study/survey study/experimentation/simulation using modern tools and techniques.	**PO5** **Modern tool usage**	Thesis Book Performance by research Presentation and soft skill
CO5	Apply moral values and research/professional ethics throughout the work, and justify genuine referencing on sources, and demonstration of own contribution.	**PO8** **Ethics**	Thesis Book Performance by research Presentation and soft skill
CO6	Perform own self and manage group activities from the beginning to the end of the research/work as a quality work.	**PO9** **Individual work and teamwork**	Thesis Book Performance by research Presentation and soft skill
CO7	Compile and arrange the work outputs, write the report/thesis, a sample journal paper, and present the work to wider audience using modern communication tools and techniques.	**PO10** **Communication**	Thesis Book Performance by research Presentation and soft skill
CO8	Recognize the necessity of life-long learning in career development in dynamic real-world situations from the experience of completing this project.	**PO12** **Life-long learning**	Thesis Book Performance by research Presentation and soft skill

Student Name /ID:

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Signature of the Supervisor:
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Name of the Supervisor:
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<u>K-P-A Mapping of ME 4800 -Thesis and Project</u>

COs	POs	Related Ks								Related Ps							Related As				
		K1	K2	K3	K4	K5	K6	K7	K8	P1	P2	P3	P4	P5	P6	P7	A1	A2	A3	A4	A5
CO1	PO2	√	√	√	√					√											
CO2	PO3					√				√	√	√				√	√	√	√		
CO3	PO4								√	√	√	√					√	√			
CO4	PO5						√			√				√							
CO5	PO8							√													
CO6	PO9																				
CO7	PO10																				
CO8	PO12																				

Student Name /ID:

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Signature of the Supervisor:
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Name of the Supervisor:
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List of Sustainable Development Goals (SDGs) Addressed in this Project

SDG No.	Goals	Targets	Relevance to the Thesis (put √ if valid)	Remarks
7	Affordable and Clean Energy	7.1 Ensure universal access to affordable, reliable, modern energy services.		
		7.2 Increase substantially the share of renewable energy.	√	
		7.3 Double global rate of improvement in energy efficiency.	√	
		7.a Enhance international cooperation on clean energy research/technology.		
		7.b Expand infrastructure and upgrade technology for sustainable energy.		
9	Industry, Innovation, and Infrastructure	9.1 Develop quality, reliable, sustainable infrastructure.		
		9.2 Promote inclusive and sustainable industrialization.		
		9.3 Increase access of SMEs to financial services and integration into value chains.		
		9.4 Upgrade infrastructure for sustainability and resource efficiency.	√	
		9.5 Enhance scientific research and technology development.	√	
		9.a Facilitate sustainable infrastructure in developing countries.		
		9.b Support domestic tech development and value addition.		
		9.c Increase access to ICT and internet.		
12	Responsible Consumption and Production	12.1 Implement 10-Year Framework on sustainable consumption/production.		
		12.2 Achieve sustainable management and use of resources.	√	
		12.3 Halve per capita global food waste.		
		12.4 Manage chemicals and waste sustainably.		
		12.5 Substantially reduce waste generation.	√	
		12.6 Encourage companies to adopt sustainable practices.		
		12.7 Promote sustainable public procurement.		
		12.8 Ensure people have relevant information for sustainable development.		
		12.a Support developing countries' scientific and technological capacity.		

		12.b Develop tools to monitor sustainable tourism impacts.		
		12.c Rationalize inefficient fossil-fuel subsidies.		
13	Climate Action	13.1 Strengthen resilience and adaptive capacity to climate-related hazards.		
		13.2 Integrate climate measures into national policies.	√	
		13.3 Improve education and awareness on climate change.	√	
		13.a Implement UNFCCC commitments (mobilize $100 billion annually).		
		13.b Promote mechanisms for capacity-building in least developed countries.		

Acknowledgment

We would like to express our deepest gratitude to our esteemed supervisor, Prof. Dr. Md. Rezwanul Karim for his invaluable guidance, patience, and unwavering support throughout this research. His insightful feedback, constructive criticism, and constant encouragement were instrumental in shaping this thesis.

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We sincerely appreciate the time, effort, and mentorship both of you have devoted to our academic growth.

Abstract

This study presents the numerical analysis of thermal–hydraulic properties in a double pipe heat exchanger fitted with an inner corrugated tube and perforated elliptical strip turbulators. The three-dimensional CAD model was developed in SolidWorks, while the simulations were performed using ANSYS Fluent with the Realizable k–ε turbulence model and enhanced wall treatment. A total of eight geometric configurations were studied by changing the corrugation pitch: 15 mm, 20 mm, 25 mm, turbulator width: 3.6mm, and 4.4 m, with changing inclination angle between 15° and 20°. The computational approach has been validated against the Gnielinski and Blasius correlations for smooth tubes and experimental data from literature, ensuring accuracy below 10% for both the Nusselt number and friction factor.

It was observed from the results that the heat transfer enhancement with the insertion of turbulators is very significant compared to a plain tube. For approximately the same value of Re≈18,000, the Nusselt number for Case 8 (p = 15 mm, W = 4.4 mm, α=20) is over 260, while for the smooth tube it is only 110. However, this enhancement in heat transfer is associated with higher frictional losses, since friction factor values increase up to 0.42 against 0.03 for the plain tube. As far as the thermal performance factor is concerned, in terms of an effective trade-off, the best values were at low Reynolds numbers, where Case 8 gave the maximum value of 1.57 at Re=6000, although at higher flow rates its effectiveness reduces to about 1.0-1.1. In summary, the results clearly show that reducing the pitch and increasing the turbulator inclination significantly enhances the heat transfer but at the price of higher pumping power requirements. Among all the geometries tested, Case 8 gives the best configuration for maximum improvement in thermal characteristics, hence promising applicability in high-performance heat exchanger systems.

Summary

(for non-technical audience)

Heat exchangers are devices which allow the flow of heat from a hot fluid to a cold one without mixing the two. They form an essential part of systems encountered in daily use, including cooling units, power generation plants, and car engines. The improvement in performance reduces energy consumption, lowers operating costs, and minimizes environmental impact. In this work, two main enhancements that were given to a double-pipe heat exchanger include an inner pipe with a waved pattern and thin, metal strips perforated with holes, also called perforated elliptical turbulators, inserted inside. These alterations cause the water to swirl as it moves, enhancing the efficiency of heat transfer.

To understand how geometry can affect performance, three structural features of this design were modified. The pitch is the distance between two consecutive waves along the corrugated pipe, and smaller spacing creates stronger turbulence in the fluid. Height indicates how far the corrugations extend into the flow area, while the width regulates how wide the turbulator strips are inside the pipe. And finally, the angle of inclination provides information on how much the strips are tilted relative to the flow direction. By changing these parameters, eight different design variations were developed and then tested by computer simulations.

Each model was tested based on three different criterions. The first one was the value of the Nusselt number, which is a dimensionless group that gives an idea on the level of the occurring heat transfer. The second was the friction factor, which means how much power is required in sending the fluid inside the pipe. And the third one was effectiveness, a measure of how well a design balanced heat transfer gains against the extra pumping cost. The reduction of pitch and the increment of turbulator width resulted in far better heat transfer, more then twice that of a smooth pipe. These same models, however, increased the flow resistance and developed higher pressure drops.

Among the presented eight cases, Case 8 was the most promising design, with a pitch of 15 mm, height of 1.1 mm, turbulator width of 4.4 mm, and an inclination angle of 20°. This case gave the highest Nusselt number, which is over 250 compared to about 110 for a plain tube, and reached an effectiveness value of 1.57 at lower flow conditions. It also showed the highest friction factor, but was still the best design in applications where a high thermal performance is necessary rather than low pumping effort. The result emphasizes that carefully designed corrugations or turbulators can make heat exchangers much more efficient and can provide significant opportunities for industries operating with compact and energy-saving thermal systems.

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Nomenclatures and Symbol

Abbreviation	Meaning
TEF	Thermal efficiency factor
LMTD	Logarithmic mean temperature difference
DPHE	Double Pipe Heat Exchanger
PEC/PEF	Performance Evaluation Criterion/ Factor
TKE	Turbulent Kinetic Energy
TDR	Turbulence Dissipation Rate
CFD	Computational Fluid Dynamics
LES	Large Eddy Simulation
RSTM	Reynolds Stress Transport Model
SST	Shear Stress Transport

List of symbols:

h	Convection heat transfer coefficient
ρ	Density(kg/m^3)
μ	Dynamic viscosity($kg/m-s$)
ϵ	Energy dissipation rate
Cp	Heat capacity ($kJ/kg.K$)
Q	Heat flux(W/m^2)
k	Kinetic energy
L	length (mm)
Nu	Nusselt number
U	Overall heat transfer coefficient($W/m^2.K$)
Re	Reynolds number

K	Thermal conductivity$(W/m.K)$
μ	Viscosity$(N.s/m^2)$
D_i, D_o	Inner and Outer diameter of pipe
ΔT	Temperature difference
f	Friction Factor
η	Effectiveness/ Thermal performance factor

Chapter 1: Introduction

1.1 Background of the study

Heat exchangers play a vital role in many engineering applications related to industrial processing, power production, refrigeration, and air conditioning. Their fundamental role is to transfer heat among two fluids without mixing them, and the effectiveness of such units directly affects the energy efficiency of these systems. In traditional smooth-tube heat exchangers, the heat transfer is often limited due to a permanent thermal boundary layer established along the tube wall. Passive augmentation techniques like inserts, baffles, and geometrical surface changes are thus quite often employed to enhance fluid mixing and turbulence without any need for additional power consumption.

Turbulators have proved to be one of the most successful passive techniques for the enhancement of convective heat transfer. It is important to find an optimum between the gain in heat transfer and flow resistance because increasing the turbulence often leads to larger frictional losses and pressure drops. The inner corrugated tube inserted with perforated elliptical turbulators can enhance the performance of a double-pipe heat exchanger. This research covers CFD analysis of the variation in geometric configuration affecting thermal performance and flow characteristic by varying parameters like pitch, width, and inclination angle.

1.2 Present state of the problem

The performance of conventional smooth-tube heat exchangers is limited by the development of a thermal boundary layer, which restricts efficient heat transmission. Very many techniques for improvement have been devised, including dimpled or corrugated surfaces, twisted tapes, and inserts; most of these, however, present excessive pressure drops or are difficult to install and therefore are not very well suited for practical application. Improving the heat transfer efficiency without appreciably raising the flow resistance or energy consumption remains a challenge.

Recent studies have shown that surface corrugation with perforated turbulators can enhance thermal performance significantly by inducing secondary flow and increasing fluid mixing. However, the influence of geometric parameters such as pitch, width, height, and inclination angle on heat transfer and hydraulic behavior in double-pipe geometries has not been well established. The present work identifies, for the first time, the most efficient design with enhanced thermal performance for an acceptable pressure loss penalty through a comprehensive numerical analysis.

1.3 Objectives of the Study

The main goal of the study is to numerically investigate how modifications inside a heat exchanger can improve its performance. Heat exchangers play an essential role in almost every industrial and domestic system that involves heating or cooling, and even small improvements in their efficiency can lead to significant energy savings. In this work, the inner pipe of a double-pipe heat exchanger was redesigned with corrugations and equipped with perforated elliptical turbulators. The main objectives of this study were to understand how these changes affect the heat transfer and flow characteristics, and to find out which design provides the most optimum balance between heat transfer enhancement and pumping requirements. The specific aims include:

- To develop an accurate model using SolidWorks and ANSYS Fluent.
- To compare the outcomes with experimental results.
- To design a novel turbulator and analyze its heat transfer performance.
- To identify optimal design parameters for enhanced thermal efficiency.

By achieving these objectives, the study seeks to demonstrate the potential of novel turbulator designs in improving heat exchanger performance. The findings are expected to provide useful guidance for future research and for engineers working to design more compact, efficient, and sustainable thermal systems.

1.4 Structure of the Thesis

The structure of this thesis has been designed to carefully investigate the thermal characteristics of a novel designed turbulator. Following the introductory chapter, Chapter 2 is the literature review. An extensive review of relevant literature has been discussed. Different studies which cover basic heat transfer principles, various turbulator and double pipe heat exchanger geometries used in different thermal systems, recent innovations, advancements in heat transfer and their performance and lastly, various research gaps have been mainly addressed. This chapter will give an overall picture of the present state of various turbulator designs and their enhancement capabilities in heat transfer.

In chapter 3, the detailed description of the model used in the study has been explored. The preparation of the CAD model, the design parameters, the different design cases to carry out CFD simulations and the specific parameter that needs to be analyzed have been briefly discussed.

The computational methodology chapter follows Chapter 4. It explains the governing equations, turbulence modeling strategy, meshing approach, and boundary conditions used in the simulations. This chapter also includes the grid independence test and validation procedures to ensure the reliability and accuracy of the numerical model.

Chapter 5 presents the results and discussions of this simulation. The velocity and temperature contours, pressure distribution patterns, and the comparative evaluation of the Nusselt number, as well as, the friction factor and also effectiveness are discussed for all cases. Particular attention is paid to the trade-off between heat transfer enhancement and the increase in hydraulic losses, besides aiming at identifying an optimum case configuration.

Chapter 6 summarizes the major conclusions from this work and discusses their engineering implications. Practical heat exchanger design optimization for industrial applications was presented, including recommendations on how to implement turbulators. Additionally, promising directions have been indicated for future work to further develop turbulator geometries and maximize their thermal performance potential, thereby presenting a complete research cycle and pointing out a direction for further advancement in the field.

Chapter 2: Literature Review

2.1 Introduction to the purpose of different heat transfer enhancement methods

Improving the performance of heat exchangers has always remained one of the central objectives in thermal sciences, and lately, several innovative techniques have been developed to further enhance their effectiveness. Among these, passive techniques that involved the addition of turbulators, inserts, baffles, or geometrical interruptions in the flow passages have proved most effective. These operate on the simple principle of fluid flow disturbance, which results in the breaking of the boundary layer; this promotes better heat exchange between the surfaces and the flowing fluid-all this without any extra energy input. Researchers have looked into many ways the shape and structure of the components can be modified in heat exchangers in order to study the changes in the way heat and fluid move through it. The modifications involve changing geometry, such as the addition of grooves, twists, perforation, or curved surfaces, done to make the fluid mix effectively and stay in contact with the heat transfer surfaces for a longer period. These modifications often lead to better heat transfer because they create more turbulence and encourage increased contact and energy transfer between the wall surface and the working fluid. These designs are continually being tested and fine-tuned by engineers through experimental studies and simulations in order to get the best compromise between enhanced thermal efficiency and acceptable flow resistance within the system.

2.2 Overview of Active and Passive Methods Purpose

A thorough review of active and passive heat transfer augmentation methods was performed by Battacharya et al.[1]. The study states that while active methods usually gives superior performance, passive techniques, such as, vortex generators, inserts, and duct surface modifications (e.g., ribs, dimples) are more widely investigated due to their simple operation and lower cost. The review shows that duct modification alone can enhance thermal performance by over 200% compared to a plain channel.

2.3 Vortex Generators and Winglet Inserts Purpose

Sun et al. [2] has performed experimental study on rectangular winglet vortex generators at 45° inclination and observed maximum thermal enhancement (TEF=1.27) at pitch ratio 1.57 with eight winglets. They found that by increasing the height ratio and winglet numbers multidirectional vortex flow increases, which caused the Nusselt number and friction factor to increase. Zhai et al. conducted an investigation examining transfer of heat behavior within circular tube equipped with delta-shaped vortex-generator pairs. As the attack angle and winglet height increased, so did the Nusselt number and friction factor. When compared to a smooth tube, the greatest improvement in heat transmission (Nu/Nu_0) with DWVG pairs was 73%, while the biggest increase in friction factor (f/f_0) was 2.5 times higher.[3]

In another similar study, Modi et al [4] has investigated the thermal-fluid behaviour of fin-and-tube compact heat exchangers with sinusoidal and elliptically curved rectangular winglet vortex generators using a two-dimensional computer study. The wavy and curved designs significantly increased the fin-and-tube heat exchanger's capacity to transport heat when compared to the flat and winglet-free baseline. Although it had the lowest London goodness factor (j/f) ratio, the up-wavy RWVG configuration produced the greatest heat transfer. On the other hand, according to the London goodness factor, the curved-down RWVG was the most appropriate since it displayed the highest j/f.

2.4 Twisted Tape Inserts Purpose

Twisted tape innovations have shown particularly better results. Chaurasiya et al. [5] developed a rectangular V-notched twisted tape that enhanced the Nusselt number by 18.27% while causing merely a 2.97% increase in friction losses. Song et al.[6] further advanced this concept with alternating-twist oval tubes, achieving a 173.6% Nu improvement and thermal performance factor (JF) of 2.18 at Re=2600, though with a 96.4% higher friction factor than conventional tubes [5], [6]. Noorbaksh et al.[7] systematically studied twisted tape geometries, showing that 4-fin tapes with aspect ratio 1 hollows improved the coefficient of performance (COP) by 63.9% compared to single-fin designs. The significance of geometric improvement ensured a favorable balance between the gain in heat transfer and the accompanying rise in pressure loss [7]. Ranjith et al. [8] further investigated computationally how twisted-tape inserts affected the double-pipe heat exchanger's friction factor and heat transmission. Findings demonstrated that a twist ratio of three produces the most heat transfer and boosts swirl flow. Similarly, Nakhchi et al. [9] numerically assessed how a Cu-water nanofluid influences entropy production in systems employing transverse-cut twisted tape inserts. They observed twisted tapes with width ratio b/c= 0.7 generated more turbulence and stronger swirl motions than modified tapes with thinner edges, leading to higher overall energy dissipation. Nakchi et al. [10] investigated the various cut ratios experimentally using twisted tapes with double cuts in heat exchangers. They reported that DCTTs cause swirl flows, which promote greater mixing between central flow; consequently, enlarging the rectangular cut ratio from 0.25 to 0.9 led to a rise in the Nusselt number by as much as 77.4%. Abolarin et al. [11] did experimental testing to evaluate the thermal behavior of twisted tape inserts arranged in both clockwise and counter-clockwise directions. Alternating clockwise or counter-clockwise twisted tapes (CCCTT) enhanced heat transfer in a circular tube, with connection angles ranging from 0° to 60° and heat flux, 1.35 to 4 kW/m², significantly affecting transitional flow. Higher angles improved transitional-regime performance, while increased heat flux delayed transition onset. New empirical relationships for the flow conditions ranging from laminar to turbulent zones, the Nusselt number and friction factor were developed. between Re = 300 and Re = 11,404. A DNA-shaped twisted tape turbulator was developed by Luo et al.[12] to enhance heat transfer in tubular systems. With an ideal pitch ratio of 2 mm, the segmented design improved heat transfer by 125% compared to a simple tube by allowing fluid to pass between gaps. The arrangement raised the heat transfer coefficient by 142% when paired with a helical coiled wire. Nevertheless, this led to a notable 960% increase in frictional losses, resulting in a thermal performance factor of 1.31 to 20% greater than that of a typical twisted tape.

2.5 Perforated and Baffled Configurations Purpose

Perforation strategies have been developed as effective solutions since it increases turbulence and hence improves thermal characteristics of heat exchangers. Rajendra et al.[13] performed experimental investigation on rectangular ducts on perforated baffles. The study reflected significant heat transfer enhancement with half-perforated baffles (26% open area) achieving 133–274% higher Nusselt numbers compared to smooth ducts, albeit at 4.42–17.5 times higher friction. Fully perforated baffles (46.8% open area) offer lower thermal-hydraulic performance compared to half-perforated designs at smaller relative pitches (e.g., 7.2), which yield a 51.6–75% performance gain under equal pumping power. While baffles enhance heat transfer, a critical limitation is forming of Lower Heat Transfer Areas (LHTAs) baffles, particularly near baffle-wall corners. To address this, Djamel et al.[14] proposed perforated baffles with strategically positioned pores (Pores Axis Ratio, PAR = 0.190–0.660) to disrupt LHTAs. At Re = 10^4–10^5, a PAR of 0.190 optimally suppresses LHTAs, achieving heat transfer improvement of about 2–65% relative to solid baffle designs, with numerical results validated against experimental data. This design refinement complements earlier findings on half-perforated baffles[13] demonstrating that pore geometry and placement are pivotal performance of thermo-hydraulics. Numerical investigations of heat exchanger tubes equipped with perforated conical ring inserts, performed by Nakchi et al.[15] reveal that hole configuration significantly influences thermal performance. For Re=4000-14000, increasing hole count (N=4 to 10) reduces Nu by 35.48% despite enhancing turbulence and core-wall mixing. The optimal design attained thermal performance value of 1.241 at a Re= 4000, showing that carefully positioned perforations can enhance heat transfer while controlling pressure penalties. Song et al.[16] demonstrated that helical fins with 8 mm perforations increased Nu by 122.3% over smooth tubes while maintaining a manageable 15.6% friction factor increase. In another study, Nakhchi and Esfhani [17] carried out numerical study assessing how perforated hollow cylinder inserts affect the overall efficiency of a heat exchanger. They analyzed effects of the perforation index, variable like as the diameter ratio (d/D) and Reynolds number. Their findings indicated that the highest thermal efficiency occurred with a diameter ratio of 0.74 and a perforation index of 24%. Similarly, Wang et al.[18] found that perforated sinusoidal vortex generators at 15° attack angle and 60 mm spacing achieved the best PEC (1.22) by improving vortex strength and flow resistance [16], [18]. Sheikholeslami et al.[19] experimentally studied the influence of perforated circular turbulators on a heat exchanger's thermal efficiency, pressure drop and Nusselt number. They investigated the influence of design variables such as pitch ratio, open-area ratio, and perforation index for the perforated turbulators and found that maximum thermal efficiency of 1.59 will occur under flow conditions corresponding to Re = 6000, an open-area ratio of 0.07, and a pitch ratio of 1.06. Experimental work by Vaisi et al.[20] on a double-pipe heat exchanger showed that perforated turbulators, which are not continuous, show a better compromise between pressure drop and thermal improvement. Compared to a non-perforated discontinuous turbulator of identical hydraulic diameter, the perforated variants with circular and square holes simultaneously increased heat transfer by 20.8% and 15% and decreased the pressure drop coefficient by 27.7% and 22.8%. Since a common drawback of turbulators is their associated pressure penalty, the study concludes that evaluating such enhancements requires a thermal performance parameter that accounts for both heat transfer and friction loss. Dagdevir and Ozceyhan [21] recently tested how perforated and dimpled twisted tapes affect a heat exchanger's thermal efficiency. Twisted tape inserts containing perforations and a pitch ratio of 0.25 delivered the maximum thermal performance index of about 1.42. A numerical study, performed by Al-Hldari et al.[22], showed that surface dimples greatly improve heat transfer in turbulent flow in dimpled tubes, with improvements ranging from 28% to 38% when compared to a smooth tube. Nevertheless, this improvement was accompanied by a significant increase in the friction factor, which exceeded 65%. With a maximum Performance Evaluation Factor (PEF) of 1.3, which indicates a net positive thermal-hydraulic advantage, the ideal design had a smaller dimple diameter (3 mm) and a particular pitch. The numerical results were reasonably accurate when compared to experimental data. A combined experimental and numerical study performed by Li et al. [23] evaluated a dimpled tube in a pipe-in-pipe heat exchanger. Experiments with water (500 < Re < 8000) and water/glycol (150 < Re < 2000) were used to establish correlations between Nusselt number and friction factor. CFD simulations confirmed that dimples enhance heat transfer by disturbing boundary layers and generating secondary flows. The numerical results showed good agreement with experiments, falling within a ±10% error band. The study concluded that the dimpled tube provided a net thermal-hydraulic benefit, as indicated by a Performance Evaluation Criterion (PEC) greater than 1.

2.6 Dimpled and Surface-Modified Tubes Purpose

Sabir et al.[24] performed a computational analysis of dimpled tubes featuring various shapes, such as 0° ellipsoidal, teardrop, and 45° ellipsoidal, and a range of pitches between 3.2 mm and 13.2 mm, for Reynolds numbers spanning 9,000 to 40,000. The 45° ellipsoidal and teardrop dimpled configurations at an optimal pitch of 3.2 mm achieved the best results, improving overall thermal-hydraulic efficiency by about 45.7% and 31.2%, respectively, within their optimal Reynolds number ranges of $9{,}000 \leq \mathrm{Re} \leq 30{,}000$ and $14{,}000 \leq \mathrm{Re} \leq 40{,}000$. In contrast, the ellipsoidal 0° tubes showed limited improvement. The authors also introduced new empirical equations for estimating the Nusselt number and friction factor.

2.7 Novel Turbulators and Nanofluid-Assisted Tubes Purpose

Hollow cylindrical turbulators are another novel insert used by Nakchi et al. [25] to examine the effect of CuO nano powder on entropy production. They found that variations in CuO nanoparticle concentration and the perforation index strongly influence the resulting heat transfer characteristics. Maintaining a constant diameter ratio (DR) of 0.7, the PI was increased from 8% to 16% and 24%, the corresponding average Nusselt numbers were recorded as 133.2%, 112.5%, and 97.4%, respectively. Agham et al. [26] conducted a numerical study examining how chamfered circular ring inserts influence heat exchanger behavior. Chamfered Circular Rings (CCRs) enhanced heat transfer via a 30° chamfered edge that creates a nozzle jet effect, inducing wall turbulence. At optimal spacing (SR=1) and diameter ratio (DR=0.7), CCRs achieved a peak thermal performance factor of 1.23 at Re=6000, outperforming conventional turbulators.

2.8 Conical and Fusiform Inserts Purpose

While metallic inserts enhance thermal performance, conical rings remain understudied due to high pressure drops. Hassan et al. [27] performed a computational investigation analyzing the hydro-thermal properties of heat exchangers containing both solid and perforated conical inserts. This study revealed that solid divergent rings achieve superior heat transfer with Nu up to 360.2 at Re=26000 but at high friction factor, f=5.04 at Re=6000, whereas perforated rings offered balanced performance (TEF=1.06) with lower pumping power. Circular holes showed better results compared to triangular ones, demonstrating shape-dependent optimization potential for efficient heat exchanger designs. Xiong et al. [28]. carried out computational study to assess how conical and spindle-shaped turbulators perform thermally within double-pipe heat exchangers featuring varied inner-tube geometries. Their study examined 21 different configurations and found that a 9 mm fusiform insert in a rectangular shaped inner tube having 0.72 aspect ratio delivered best performance at Re=4000, while 12 mm fusiform insert performed best in circular inner tubes. Ibrahim et al. [29] explored how conical-ring-type turbulators influence heat-exchange efficiency. Conical-ring turbulators, including convergent, divergent, and standard forms, greatly enhance heat transfer, with the divergent-ring configuration producing the strongest effect. Optimal performance ($\eta = 1.291$) occurs under operating parameters of $d/D = 0.4$ and pitch ratio $PR = 2$ at a Reynolds number of 6000, though entropy generation rises with Re and lower d/D. Rinik et al. [30] executed a computational investigation using conical ring inserts. The study's numerical results show that integrating six convergent conical rings with a fully twisted elliptical inner pipe significantly improves heat transmission in a double-pipe heat exchanger. The design showed that the improvement in heat transmission greatly outweighed the accompanying pressure loss, with a Nusselt number of 445 and a PEC of 2.3 in counterflow.

2.9 Advanced and Hybrid Enhancement Methods Purpose

A recent study introduced an innovative woven-wire turbulator of low weight and large surface area. Khedher at al.[31] performed experiments which showed improvement in thermal-transfer rate of as much as 236 %, accompanied by only slight rise in pressure loss, yielding a high Thermal Enhancement Factor (TEF) value reaching 2.45. Furthermore, coupling it with a helical-coil wire turbulator produced a synergistic rise in performance, elevating the heat-transfer coefficient by 545 % and attaining a TEF as high as 3.13, demonstrating significant potential for heat exchanger efficiency. A newly introduced enhancement approach called the Electromagnetic Vibration (EMV) technique, was investigated by Mashoofi et al.[32] In this study, a coaxial string or strip oscillator inside a heated tube was made to vibrate longitudinally. The superior strip geometry was shown by the experimental findings, which increased the heat transfer coefficient by up to 2.5 times. The substantial potential of the EMV method was demonstrated by a numerical model that predicted a notable enhancement at high frequencies (1 kHz), producing heat-transfer coefficients and thermal-enhancement factors that were approximately 32.2 and 14.2 times higher than those of plain tube, respectively. Another study performed by Pan et al. [33] showed the use of multiple perforated magnetic turbulators, vibrated via an external AC magnetic field, to enhance heat transfer. Findings indicated that both solid and perforated turbulators enhanced heat transfer by roughly 150–156 %, though at the cost of a higher pressure drop. The optimal configuration was a perforated turbulator featuring 2 mm holes and 12 mm pitch yielded a TEF of 2.06, demonstrating the effectiveness of perforations in maximizing overall thermal-hydraulic performance.

2.10 Spherical, Rotating and Wire-Based Inserts Purpose

Manjunath et al. [34] had experience with spherical turbulence generators. He noticed that the relative roughness pitch decreased as the spheres' diameter increased. The average peak improvement in thermal efficiency over the standard arrangement for D=25mm and P/D=3 was around 23.4%. At Re=23560 for D=25mm and P/D=3, the Nusselt's highest growth number is found to be 2.5 times higher than in the standard model. In another study Goh et al. [35] demonstrated that heat exchanger performance could be substantially increased through the use of self-rotating turbulators. These specialized inserts intensify turbulent convection and entropy generation. The greatest enhancement to the Nusselt number, a key heat transfer metric, was a 360% increase from stationary inserts, while the rotating versions still delivered a robust 240% improvement. In a study on shell and coiled tube heat exchangers, Panahi et al. [36] used helical wire turbulators. Results showed that such devices markedly boosted thermal performance, as measured by average heat-transfer rate, though they also introduced additional pressure losses. Bashtani et al.[37] conducted a numerical investigation on the thermal performance of gear disc turbulators in conjunction with water–Al_2O_3 nanofluid. Their results showed that the greatest improvement in heat performance occurred at a nanoparticle loading of 6%. The presence of turbulators significantly improved heat transfer by inducing fluid-surface collisions and Mohsen et al.[38] experimentally examined the influence of various fin geometries—

including rectangular, circular, and helical ribs—on heat-exchanger performance. The study found that rectangular-fin configurations yielded the greatest improvement in Nusselt number, with values increasing from 68% to 168% compared to a smooth tube. Numerical simulation performed by Fan et al. [39] that louvered strip inserts significantly enhance heat transfer, increasing the Nusselt number increased 2.75 to 4.05 times compared with a smooth tube. While larger slant angles and smaller pitches boost heat transfer, they also increase flow resistance. The performance evaluation criterion (PEC) ranged from 1.60 to 2.05, confirming a strong overall thermo-hydraulic performance. Optimal results were achieved with a moderate slant angle and small pitch.

Maakoul et al.[40] carried out numerical analysis on continuous helical baffles for thermohydraulic performance, finding 5–45 % heat-transfer gains but 2–21 times higher pressure drop than conventional setups compared to a conventional design. The peak thermal performance occurred under laminar-flow conditions, where the enhancement factor was 19–35% higher than in turbulent flow. Nakchi et al. [41] experimentally examined how rotating elliptical inserts influence double-pipe heat-exchanger performance, where the rotation angle of the inserts was systematically varied. The study demonstrated that the rotary elliptic turbulator generated additional recirculation zones and flow disturbances, significantly enhancing thermal performance. The optimal configuration was found at a rotation angle of 90°, using a slant angle of 25° and diameter ratio 0.25 achieved the top thermal-efficiency value of 2.23.

2.11 Louvered Strips and Elliptic Turbulators Purpose

Researchers commonly employ louvered strips and elliptic turbulators as effective turbulators in both numerical and experimental studies. Mohammed et al. [42] numerically studied louvered strip inserts in circular DPHEs using nanofluids. Their study revealed that backward-oriented strips at 30° slant angle and 30 mm pitch increased the Nusselt number (Nu) by 367–

411% compared to smooth tubes, achieving a peak performance evaluation criterion (PEC) of 1.54. However, the skin friction coefficient increased significantly, focusing on the basic compromise between flow resistance and heat transfer enhancement. Yaningsih et al. [43] investigated louvered strips with slant angles of 15°, 20°, and 25°. They established empirical links based on their testing results, which showed that these inserts improved heat removal by 77% and friction reduction by 335%. Backward and forward louvered strips were compared by Eiamsa-ard et al [44]. According to their findings, the forward-facing inserts performed better, increasing the friction loss by 413% and the Nusselt number by 284%. An experimental investigation performed by Rivier et al. [45] explored novel elliptic turbulators for improving pipe flow heat transfer. The setup with a 60° bend angle produced the best thermal efficiency, according to the study, which covered both laminar and turbulent flow regimes. Kim et al.[46] employed numerical simulations with the SST turbulence model to optimize the design of elliptic dimple turbulators. Their analysis concluded that the optimal shape boosted thermal performance, yielding a 32.8% higher Nusselt number compared to a conventional circular design. Research by Rajaseenivasan et al. [47] revealed that circular turbulators outperformed V-shaped ones in a solar air heater application. The circular vortex generators produced the highest thermal output, attaining an air temperature of 66°C. In another study, Shirvan et al. [48] has examined the efficiency and heat transfer of a double pipe heat exchanger using a porous material. The sensitivity analysis shows that the mean Nusselt number can be maximized by using Re=5000, Da=10-5, and δ=1/3. However, the efficiency of the heat exchanger alone is optimized at δ = 1, Re = 5000, and Da = 10-3. On the other hand, Nu and E can be maximize, if Re=5000, Da=10-5, and δ=1 are utilized.

2.12 Porous-Media and Geometry Based Purpose

Apart from adding different metallic inserts inside heat exchanger tubes, the geometry of these tubes has proved to play a crucial role in heat transfer enhancement of double pipe heat exchangers. Zahrani [49] explored several geometric modifications. Hybrid oval-wavy configurations improved Nu by 28% over conventional designs, while pure oval geometries reduced pressure drop by 55%. The oval configuration showed an 83–85% improvement in thermal performance factor (η), suggesting that cross-sectional geometry optimization can significantly enhance thermo-hydraulic performance. Wang et al. [50] performed investigation on novel dimpled tube. He demonstrated that ellipsoidal dimpled tubes enhance heat transfer more effectively than spherical dimples, showing 38.6–175.1% higher Nusselt numbers than smooth tubes while maintaining lower friction penalties (26.9–75% increase). The alternating major-axis orientation of ellipsoidal dimples promoted early flow transition at $Re < 1000$ and achieves up to 87% thermal performance gain at comparable friction costs, showing greater optimization potential for enhanced heat exchangers. Kaood et al. [51] performed computational study with conical tubes with dimples. Dimpled conical tubes demonstrated superior thermal-hydraulic performance, with convergent dimpled tubes (DR=1.5) achieving a 121.4% Nusselt number increase and 29.54% PEC enhancement compared smooth tubes at Re=3000. The novel design showed particular promise for energy-efficient applications at lower Reynolds numbers, offering both improved heat transfer (up to Nur=2.72) and potential CO_2 reduction benefits. Chehragi et al. [52]investigated a novel "deep dimpled tube," analyzing 27 configurations with varying dimple pitch, diameter, and depth at Reynolds numbers of 500, 1000, and 2000. Results showed that deeper dimples with larger diameters and smaller pitches could enhance the heat transfer rate by up to 600%, though with a significant increase in friction factor. The Performance Evaluation Criterion (PEC) ranged from 1.15 to 3.3, indicating a net positive thermal-hydraulic benefit. The maximum PEC of 3.3 was achieved at Re=2000 with configuration consisting of 18 mm diameter, 2 mm dimple depth, and pitch four times the diameter. Xie et al.[53] numerically compared heat transfer tubes with compared teardrop-shaped dimples with spherical and elliptical counterparts. The teardrop geometry was found to reduce the strength of recirculation flow and pressure loss, leading to a better combined thermal and hydraulic behavior. Both Nusselt number and friction factor grew with dimple depth and radius but declined as the pitch increased. The optimum configuration (depth=2 mm, pitch=15 mm, radius=4 mm at Re=5,000) achieved a highest PEC value of 2.06. In another study, Xie et al. [54] proposed an innovative tube featuring helical dimples that promoted fluid swirling and mixing, thereby increasing thermal performance by 120–270 % relative to a smooth tube. The Nusselt number and friction factor rose with additional dimple starts, deeper cavities, and larger radii. The optimal configuration reached a maximum PEC of 1.93 for dimples 2 mm deep and 2 mm in radius.

2.13 Corrugated Tube Design Purpose

A numerical study analyzed by Al-Obaidi [55] thermal transfer and fluid flow characteristics in corrugated pipes with different geometric parameters such as corrugation ratio, ring count, and diameter. Results showed that higher corrugation ratios and Reynolds numbers raised Nusselt values but simultaneously caused a substantial growth in friction factor, with a 100% PCR configuration showing a friction factor up to 67.1% higher than other models. Despite the increased pressure drop, the performance evaluation factor remained above unity, indicating a net positive thermal-hydraulic benefit. The correlations were also formulated for estimating Nusselt and friction-factor behavior. In another experiment, Al-Obaidi et al.[56] performed numerical investigation of corrugated tubes which revealed that heat transfer enhancement is primarily due to the transformation of convective heat transfer into jet impingement and boundary layer redevelopment caused by flow pulsation. While increasing the Reynolds number (Re) and corrugation geometry (ring angle, diameter, and number) raised the Nusselt number (Nu), it also increased the friction coefficient and increased pressure loss resulting from induced spiral motion. all corrugated designs yielded PEF values exceeding 1, confirming superior thermo-hydraulic efficiency, confirming their overall thermal-hydraulic superiority over smooth tubes, although the PEF declined progressively as the Reynolds number increased. Corcoles et al. [57] investigated how spiral-shaped inner corrugated tubes affect double-pipe heat-exchanger behavior. The largest pressure drop resulted from the highest corrugation height and low pitch. It boosted the heat-transfer rate by about 29 % over the comparison case. In another study Garcia et al. [58] carried out a study among three different tube configurations. She demonstrated that corrugated and dimpled tubes showed better results than wire coils for Re > 2000, offering comparable heat transfer with lower pressure drops, while wire coils were optimal for 200 < Re < 2000. Roughness shape significantly impacted pressure drop and transition to turbulence, with smooth tubes remaining preferable for Re < 200. The study provided clear regime-specific recommendations for enhanced tube selection. Again, Kaood et al. [59] performed numerical study of thermal characteristics in various corrugated tubes. It was found that inward corrugated inward corrugations improved heat transfer more efficiently than outward forms; trapezoidal profiles gave the highest Nusselt increase and roughly 52.6 % beyond smooth-tube performance. Curved and triangular corrugations offered superior thermal-hydraulic performance beacause to increased turbulence kinetic energy along with velocity homogeneity. The study demonstrated that corrugation direction and shape strongly affected both the heat-transfer efficiency and the accompanying pressure-drop behavior. A numerical study performed by Cauwenburge et al.[60] employing a verified Large-Eddy-Simulation computational framework, found that helicoidally corrugated tubes enhanced convective heat transfer by 83% to 119%. However, this improvement came with a significant penalty, causing pressure losses approximately 5.6 to 6.7 times higher than those in smooth tubes. To extend these findings to industrial applications, the LES data was used to develop a more accurate Reynolds Stress Transport Model (RSTM) for predicting performance. Han et al. [61] conducted numerical analysis of flow and heat transfer within symmetric and asymmetric convex corrugated tubes; the observed 8–18 % heat-transfer gain in the asymmetric case was linked to altered turbulent-velocity distributions near the wall surfaces.

2.14 Research Gap

Few studies have examined the combined impact of inner corrugation and perforated elliptical turbulators in double-pipe heat exchangers, despite the fact that numerous studies have examined passive heat transfer enhancement employing twisted tapes, inserts, and corrugated tubes. There is still a lack of research on how important geometric factors like pitch, width, height, and inclination angle affect heat transfer and pressure drop. By using a thorough numerical analysis to find an ideal arrangement that strikes a balance between high heat transfer and low flow resistance, this study fills that gap.

Chapter 3: Numerical Model Description

3.1 Numerical Model

SolidWorks was used to create a computational model of the double pipe heat exchanger shown in Figure 1, which has an elliptical perforated turbulator and an inner corrugated tube. Figure 2's cross-sectional view of the heat exchanger demonstrates the corrugated tube's height, width, and pitch were changed in order to investigate thermal properties numerically. The overall dimensional parameters of the inner and outer pipe are presented in Table 1.

Figure 1: Double pipe heat exchanger

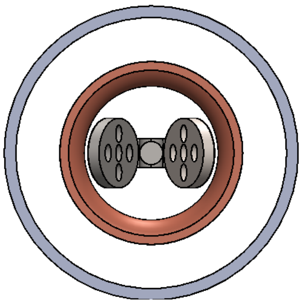


Figure 2: Cross section view of DPHX

Inner pipe	
Material	Copper
Thickness	0.75 mm
Diameter	14.3 mm
Length	855 mm
Outer pipe	
Material	Steel
Thickness	1.00 mm
Diameter	23.4 mm
Length	840 mm

Table 1: Primary Parameters of DPHX

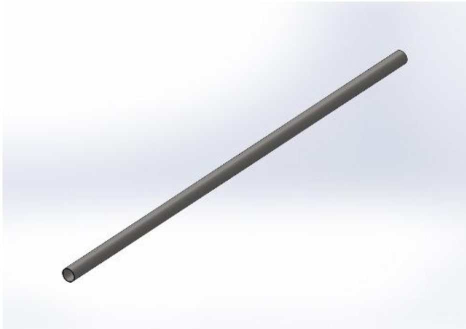


Figure 3: Outer pipe

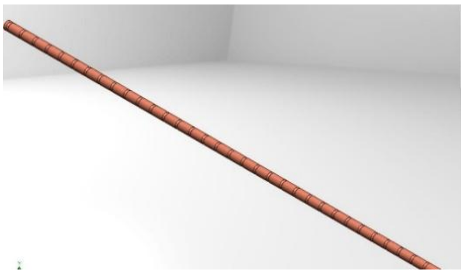


Figure 4: Corrugated inner pipe

Figure 5: Turbulator

The turbulator represented in Figure 5 is made up of stainless steel and a thickness of 2mm and the pitch between vortex generators is varied according to the pitch of the corrugated tube. The design specifications of the turbulators are selected according to the study done by of Yaningsih et al.[43]

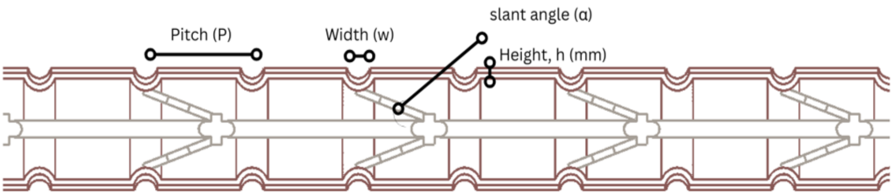
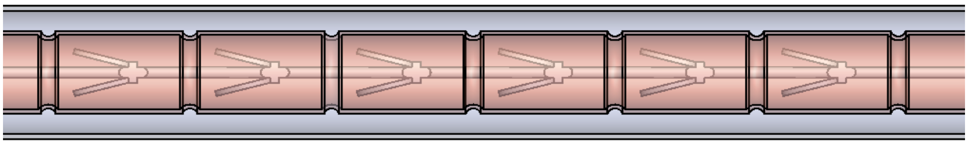
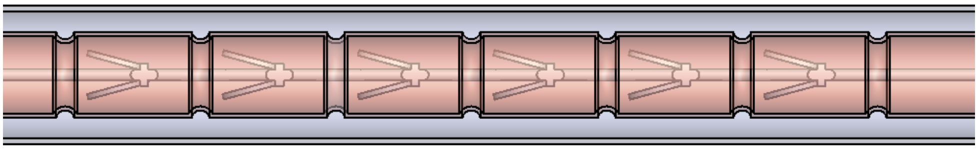


Figure 6: Detail of the lengths that define the inner pipe

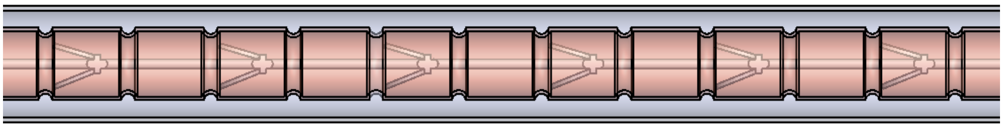
By changing the width, height and pitch of the corrugated tube and the angle of the turbulator illustrated in Figure 6, a total of eight cases have been numerically studied.



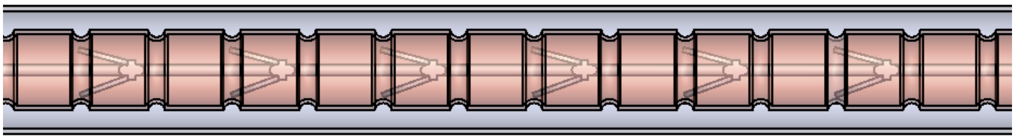
Case 1: H=1.1 mm, p= 25mm, W=3.6mm, α=15



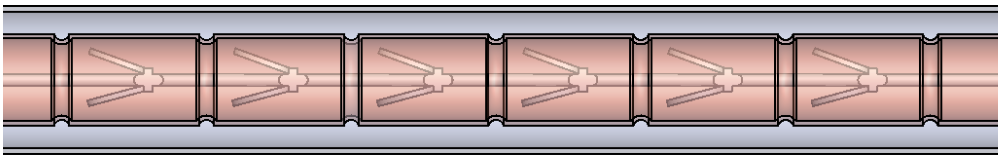
Case 2: H=1.1 mm, p= 25mm, W=4.4mm, α=15



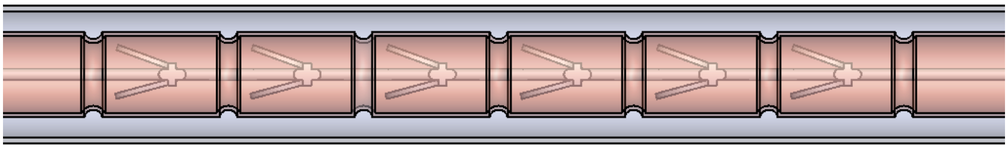
Case 3: H=1.1 mm, p= 25mm, W=4.4mm, α=15



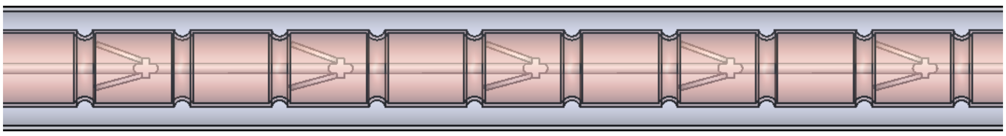
Case 4: H=1.1 mm, p= 15mm, W=4.4mm, α-15



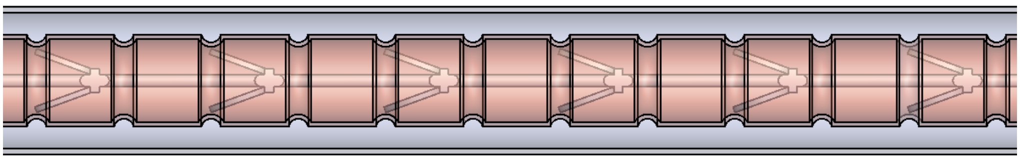
Case 5: H=1.1 mm, p=25mm, W=3.6mm, α=20



Case 6: H=1.1 mm, p=25mm, W=4.4mm, α=20



Case 7: H=1.1 mm, p=20mm, W=4.4mm, α=20



Case 8: H=1.1 mm, p=15mm, W=4.4mm, α=20

Figure 7: Variations of geometry presented in eight scenarios

3.2 Mesh Generation

After preparing the computational model, mesh have been generated in ANSYS FLUENT. The number of nodes during mesh generation was 1560452 and the number of elements was 7480571. The orthogonal quality of the mesh was 0.88 and the skewness of the nodes were 0.2. These parameters primarily indicate that a very good quality mesh has been generated.

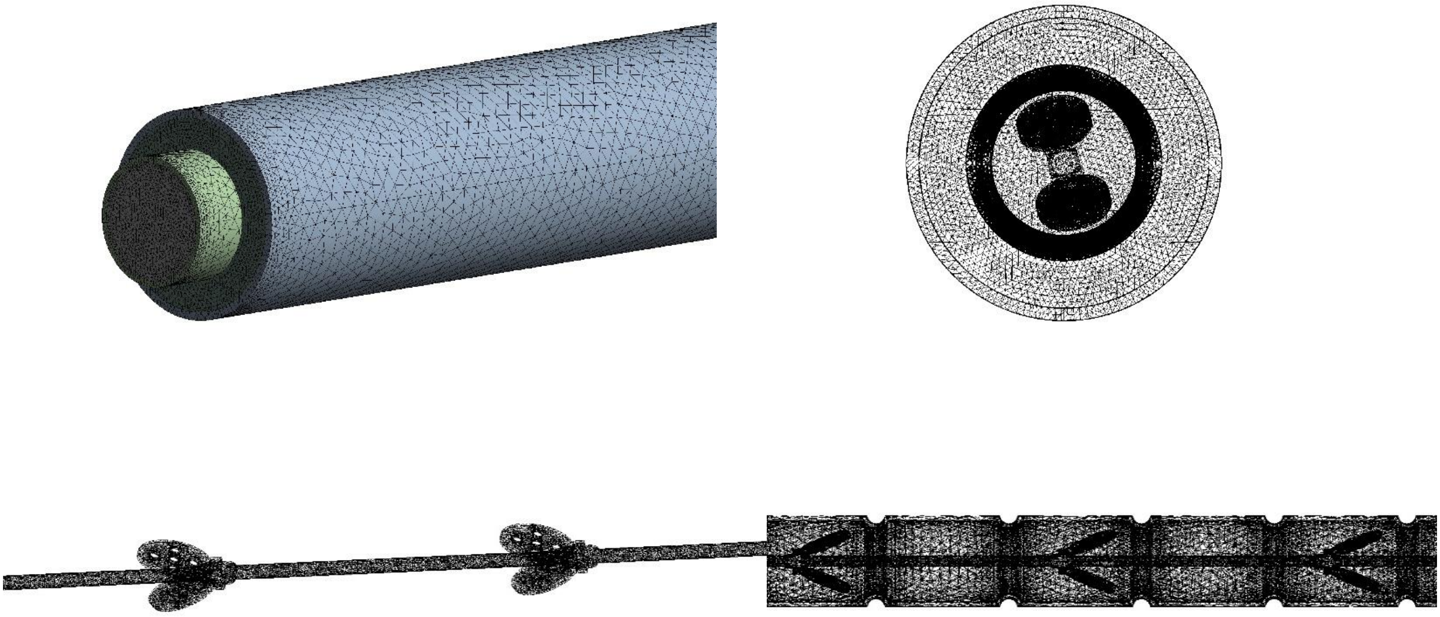


Figure 8: Computational mesh

3.3 Validation Cases

The inclination angle of the elliptical strips is denoted by ‘α’, the pitch and width of the corrugated tubes are denoted by ‘p’ and ‘W’ respectively. Varying these three parameters, there are 8 cases that needs to be validated.

Case	Diameter, D (mm)	Height, H (mm)	Width, W (mm)	Pitch, p (mm)	Slant angle,α
1	14.3	1.1	3.6	25	15°
2	14.3	1.1	4.4	25	15°
3	14.3	1.1	4.4	20	15°
4	14.3	1.1	4.4	15	15°
5	14.3	1.1	3.6	25	20°
6	14.3	1.1	4.4	25	20°
7	14.3	1.1	4.4	20	20°
8	14.3	1.1	4.4	15	20°

Table 2: Dimension of analysed cases

The cold fluid flows in the annular side and the hot fluid flows in the inner pipe of the double pipe heat exchanger. The turbulator is fitted inside the inner pipe where the hot fluid flows. The flow of the hot and cold fluids are opposite in direction implying that the heat exchanger being studied is a counter flow heat exchanger.

Chapter 4: Computational Methodology

4.1 Governing Equations

It is assumed that the system is operating in a steady-state condition, with heat transfer occurring under turbulent flow. The differential form of the governing equations for mass and momentum conservation [62[is as follows:

Continuity Equation

$$\nabla . \vec{v} = 0 \tag{1}$$

where $\vec{v}$ refers to the fluid's velocity vector.

Momentum Equation

$$\nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla p + \nabla \cdot (\mu \nabla \vec{v}) + \nabla \cdot (-\rho \, \overline{\vec{v'} \vec{v'}}) \tag{2}$$

where p stands for pressure, ρ is the fluid's density, dynamic viscosity is denoted by μ and $\vec{v}$ is the turbulent velocity vector.

In this study, turbulence was modelled using the Realizable k–ε turbulence model with enhanced wall treatment. The Realizable k–ε model was selected because of its demonstrated effectiveness in accurately predicting separation of turbulent flow and heat transfer in systems with perforated turbulators. Based on this approach, the transport equations for k and ε are defined as follows [62]:

Turbulence Kinetic Energy Equation

$$\nabla \cdot (\rho \vec{v} k) = \nabla \cdot [(\mu + \frac{\sigma k}{\mu t}) \nabla k] + Gk - \rho \epsilon - YM \tag{3}$$

Turbulent Dissipation Rate Equation

$$\nabla \cdot (\rho \vec{v} \epsilon) = \nabla \cdot [(\mu + \frac{\mu t}{\sigma_\epsilon}) \nabla \epsilon] + \rho C_1 S \epsilon - \rho C_2 k \frac{\epsilon^2}{k + \sqrt{\nu \epsilon}} + C_1 \frac{\epsilon}{k} C_{3\epsilon} G_b \tag{4}$$

where

$C_1 = \max\left[0.43, \frac{\eta}{\eta + 5\eta}\right], \eta = S\frac{k}{\epsilon},\ S = \sqrt{2SijSij}$

Here the turbulent kinetic energy is denoted by k, the loss rate is denoted by ϵ.

Energy Equation

$$\nabla \cdot \rho c_p v \cdot \nabla T = \nabla \cdot [(k + k_t)(\nabla T) + \rho\varepsilon \tag{5}$$

where fluid temperature is denoted by T and specific heat capacity is denoted by c_p.

4.2 Required equation

The following formula represents the heat absorption from the cold water on the annular side [62]:

$$Qc = \dot{m}_c C_{p,w}(T_{c,out} - T_{c,in}) \tag{6}$$

where $\dot{m}_c$, $T_{c,in}$ and $T_{c,out}$ is the mass flow rate and inlet and outlet temperature of cold fluid respectively. $C_{p,w}$ is the specific heat of water.

The following formula is used to determine the heat rejected by the hot water in the inner pipe [62]:

$$Qh = \dot{m}_h C_{p,w}(T_{h,in} - T_{h,out}) \tag{7}$$

where $\dot{m}_h$, $T_{h,in}$ and $T_{h,out}$ is the mass flow rate, inlet temperature and outlet temperature of hot fluid respectively.

The average heat flow rate is used to determine the average heat transfer coefficient (U) for the inner pipe is calculated as:

$$Q_{avg} = \frac{Q_h + Q_c}{2} = UA_i\Delta T_{LMTD} \tag{8}$$

Here A_i denotes the surface area of the inner tube made of copper in the test section, which is given by:

$$A_i = \pi D_i L$$

where the inner diameter of the tube is given by D_i in the double-pipe heat exchanger.

To compute the internal heat transfer coefficient which is a key objective of this study, the following equation is used [62]:

$$h_i = 1/[\frac{1}{U} - \frac{D_i ln(D_o/D_i)}{2k} - \frac{D_i}{D_o h_o}] \tag{9}$$

In this equation, D_o represents the diameter of the outer annular pipe. The outer heat transfer coefficient h_o is estimated using the Dittus–Boelter correlation[63]:

$$Nu_0 = \frac{h_o D_h}{k} = 0.023{Re_o}^{0.8}Pr^{0.3} \tag{10}$$

Here, $D_h = D_o - D_i$ is the hydraulic diameter, Pr is the Prandtl number, and is the Reynolds number for cold water flowing in the outer tube.

The average Nusselt number for the inner tube equipped with elliptical turbulators is calculated as:

$$Nu_i = \frac{h_i D_i}{k} \tag{11}$$

Frictional losses (*f*) are determined using[62]:

$$f = \frac{\Delta P}{0.5\rho V_{in}^2}\frac{D_i}{L} \tag{12}$$

where V is the inner heated tube's average velocity. The ratio of heat transfer enhancement to dimensionless frictional increase is known as the thermal performance factor (η) and it can be found using [62]:

$$\eta = \frac{Nu/Nu_s}{(f/f_s)^{1/3}} \tag{13}$$

where Nu_s and f_s refers to the Nusselt number and friction factor for a smooth pipe without any turbulator.

4.3 Boundary Conditions

The outlet boundaries are chosen as pressure outlets, whereas the inlet boundary conditions are characterized as velocity inlets. While the cold fluid goes through the outer pipe at a temperature of 17.7°C, the hot fluid flows through the inner pipe at a temperature of 80°C. A turbulent flow regime is indicated by the fluid passing through the pipes at Reynolds numbers between 6000 and 18,000. Assuming that no heat transfer takes place across this boundary, insulation imposes an adiabatic boundary condition on the outer pipe wall. To simulate the interaction between the solid and fluid areas, coupled heat transfer is used to model the inner pipe and elliptical turbulator walls, and the pipe walls are considered to follow no_slip condition.

Thermophysical properties	Hot fluid	Cold fluid
ρ(kg/m^3)	971.8	998.614
Cp($kJ/kg.K$)	4.197	4.183
K($W/m.K$)	0.667	0.594
μ($N.s/m^2$)	0.000354	0.001638

Table 3: Thermophysical properties

4.4 Dimensionless Wall Distance

The computational domain was discretized, paying special attention to the near-wall region, which was guided by the non-dimensional wall distance y^+. In this work, the mesh was refined for values of $y^+ \approx 1$ in order to resolve the viscous sublayer. For the double-pipe geometry, the grid variations mostly included variations in the thickness of the first layer from the walls and the size of the elements on the inner pipe and annulus surfaces. This setup remained consistent with the use of the k−ε turbulence model, combined with enhanced wall treatment. The result of this approach is that wall-bounded effects were appropriately captured while sustaining numerical stability and accuracy throughout the flow domain.

4.5 Solution Method

The governing equations are discretized using the finite volume method for numerical analysis using ANSYS Fluent software. Effects like gravity, thermal radiation, and viscous heating are disregarded, and the fluid is regarded as Newtonian. The Realizable k–ε turbulence model is used to model and simulate turbulent flow. The linked algorithm manages pressure-velocity coupling, guaranteeing accurate interaction between pressure and velocity fields. The momentum, turbulent kinetic energy, and turbulence rate of dissipation equations are spatially discretized using a second-order upwind technique. To increase computational precision, a second-order technique is also used to discretize the pressure gradient. For every residual, the convergence criteria is set at $1e^{-6}$.

4.6 Grid Independence Test

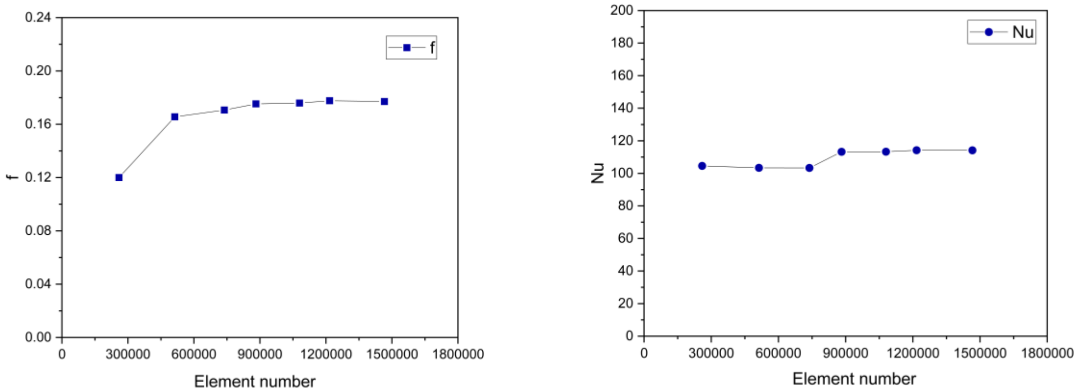


Figure 9: Grid independence test graph between element number against friction factor and Nusselt number

A grid independence study was carried out for case 2 (w = 4.4 mm, p = 25 mm, a = 15°) at a Reynolds number of 10,000 as presented in Figure 9. A structured mesh was applied to the pipe domains, while the hot fluid, cold fluid, and turbulators inside the corrugated inner pipe were discretized using a tetrahedral mesh. To capture the complex flow behavior, the mesh was refined around the turbulators and along the pipe surfaces. The element size was varied between

0.5 mm and 25 mm to evaluate its influence on the numerical results. Both the friction factor and the Nusselt number increased with finer mesh resolution. However, the variations became less significant as the mesh was further refined. Specifically, when the mesh size was increased from 1 million to 1.46 million elements, the changes in the friction factor and Nusselt number were only 0.52% and 0.77%, respectively. Thus, 1 million is chosen to determine fluid performance for precise results as well as fewer computational time.

Chapter 5: Results and Discussions

5.1 Validation

The numerical data for smooth pipe flow were compared with the widely used Gnielinski and Blasius correlations, which serve as reliable benchmarks for turbulent flow in smooth tubes. The Gnielinski correlation [63] $Nu=0.012(Re^{0.07}-280)\ Pr^{0.4}$ for $1.5<Pr<500$ and $3000<Re<10^5$ and Blasius correlation [64] $f=0.318Re^{-0.25}$ are frequently employed to validate numerical and experimental investigations of turbulent heat transfer and friction characteristics in smooth tubes.

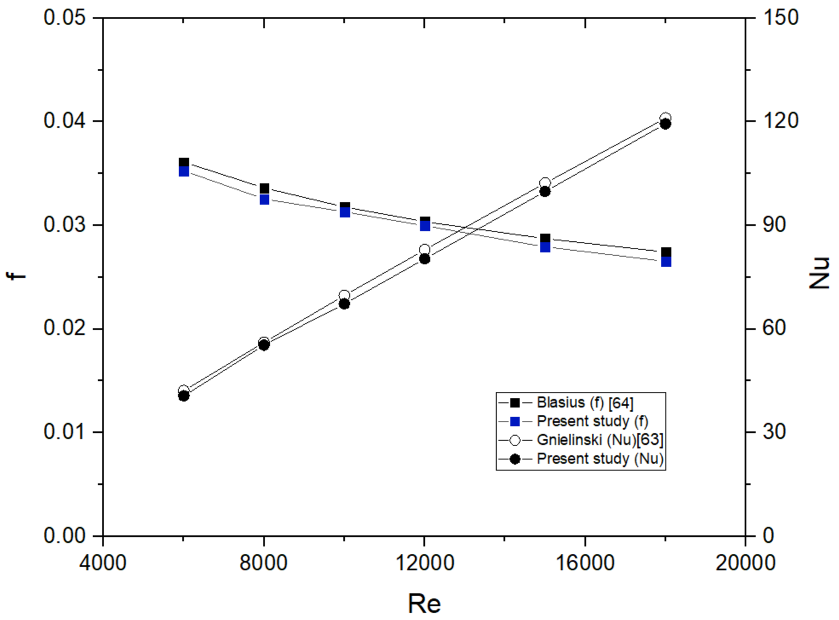


Figure 10: Validation of nusselt number and friction factor of turbulent flow inside a plne tube with Gnielinski and Blasius Correlation

From Figure 10, it can be observed that the numerical results obtained in the present study show a close agreement with the well-established correlations of Gnielinski and Blasius. This consistency indicates that the present computational approach can reliably predict the heat transfer and friction characteristics of turbulent flow in smooth tubes, thereby validating the accuracy of the adopted methodology within the considered Reynolds number range.

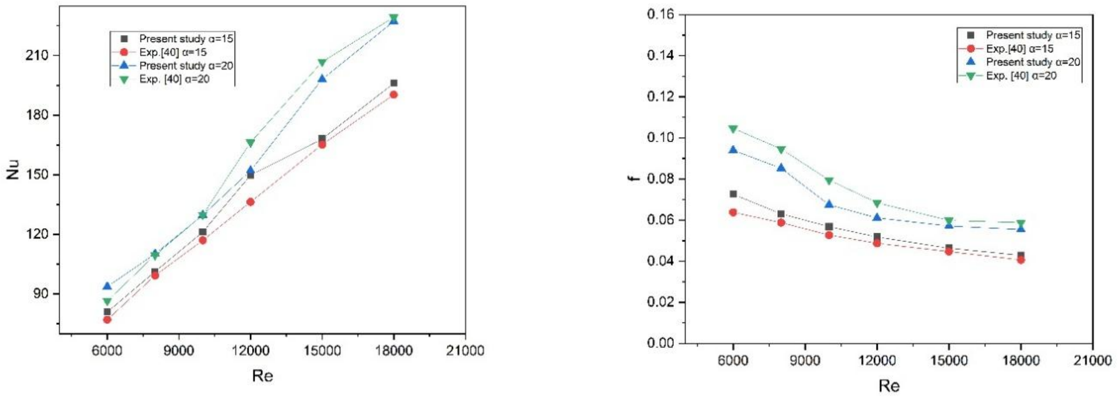


Figure 11: Validation of present study with Nakhchi et al.[41]

Figure 11 shows a comparison between the experimental analysis heat transfer coefficient and frictional loss obtained by Nakhchi et al. [41], for double pipe heat exchanger with perforated elliptical turbulator. The overall percentage error between experimental and numerical friction factor are 5.41% and 9.38% for 15° and 20° turbulator respectively. The error in Nusselt number was 6.28% and 5.218% for 15° and 20° turbulator respectively. As the percentage error is very low, it can be concluded that the numerical investigation performed align with the experimental study and the results produced, from the computational approach, are accurate and valid.

5.2 Pressure drop contour

The pressure drop contours in the inner corrugated tubes and annulus side for all the cases are shown in Figure 12. By observing the contours, it can be seen that Case 8 has the highest pressure drop when α=20°, p=15mm, w= 4.4 mm and h=1.1mm.

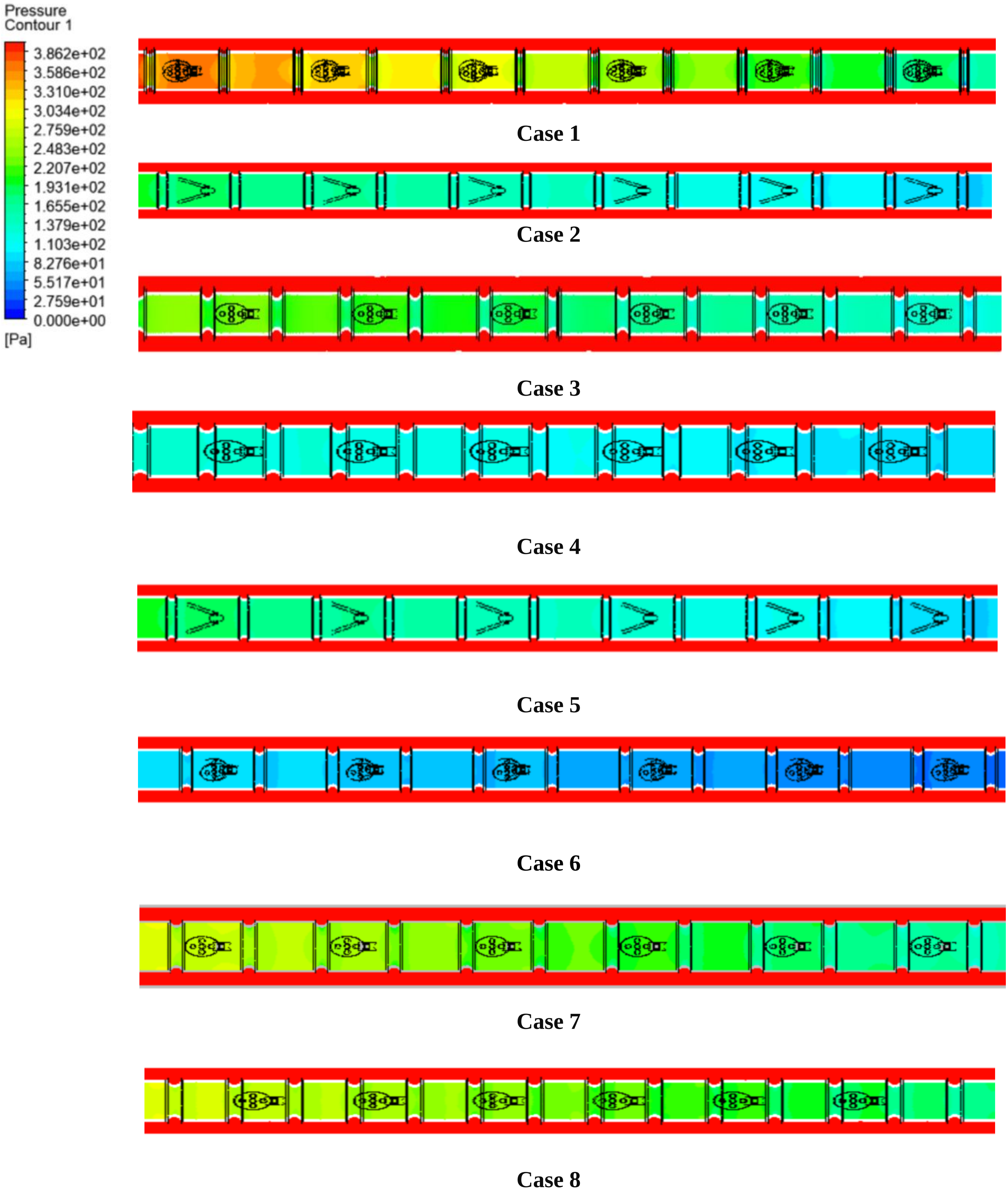


Figure 12: Pressure contour at Re=10000

5.3 Flow pattern and heat transfer

The heat transfer characteristics are strongly governed by the flow behavior throughout the computational domain, particularly in the vicinity of the corrugated surfaces. In this region, the corrugation geometry alters the velocity and pressure fields. Figure 13 shows the temperature distribution inside the double pipe heat exchanger. This explains the fluid flow behaviour due to the corrugated inner pipe and the turbulator. In cases 7 and 8 the temperature difference in the inner pipe is the highest.

Figure 14 shows the velocity distribution in the double pipe heat exchanger. For all cases, the velocity is the lowest at the region near the walls for both inner and outer pipes. On the inner side downstream of the corrugation, flow deceleration near the wall induces vortices and secondary flows, while on the annulus side, separation and reverse flow occur around the corrugation region. These combined flow interactions enhance fluid mixing, thereby improving the heat transfer performance compared to a smooth pipe.

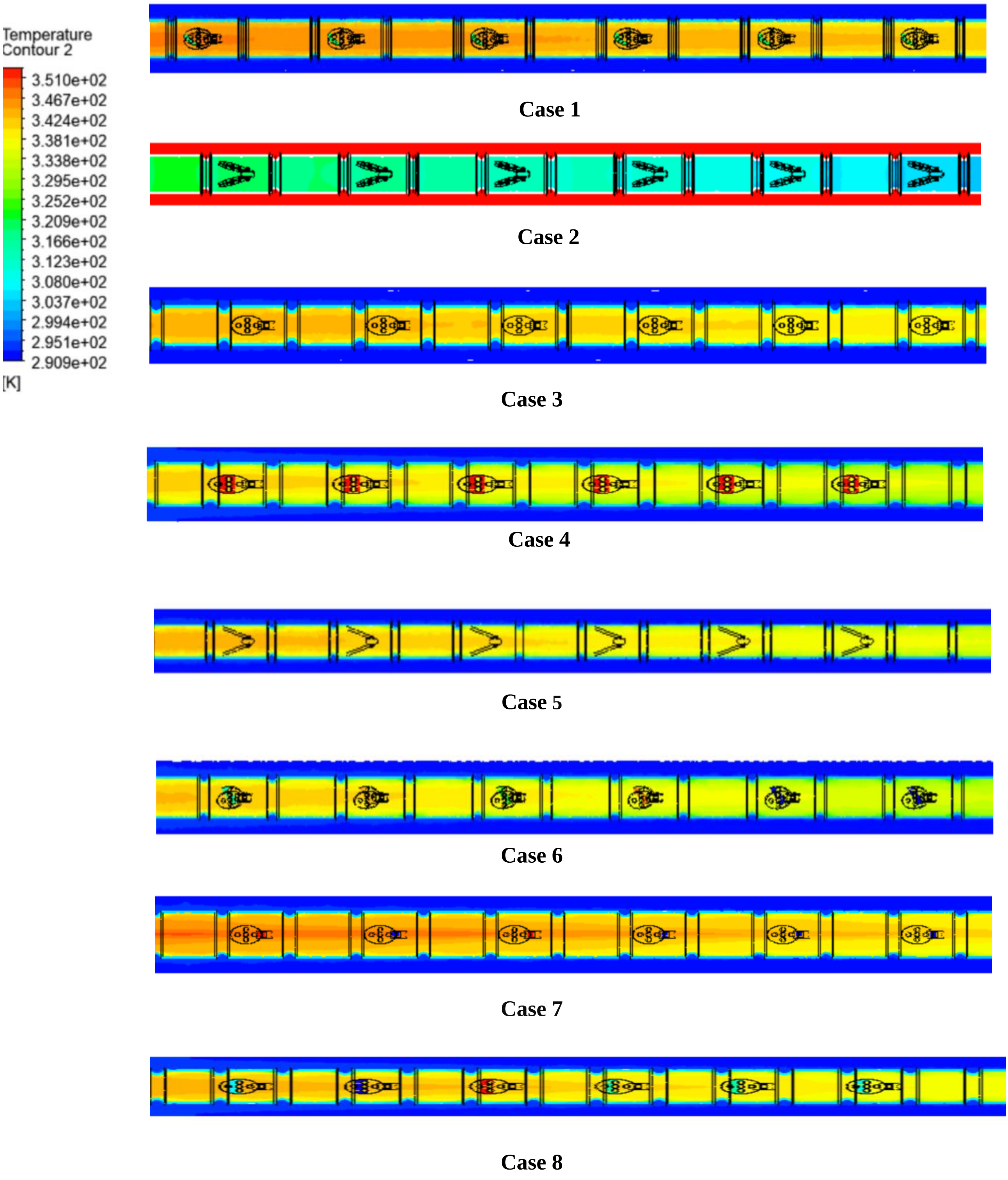


Figure 13: Temperature contour at Re=10000

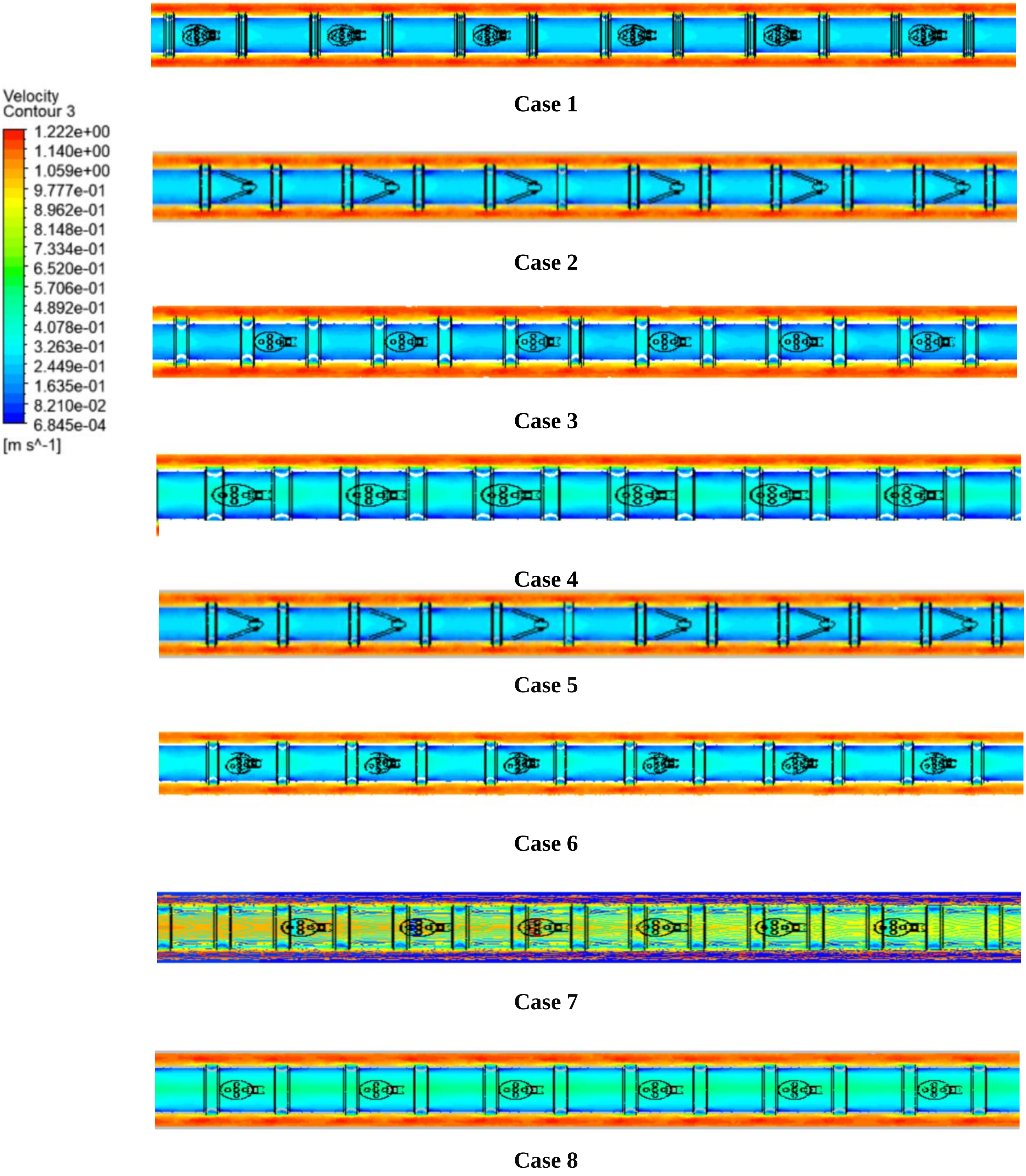


Figure 14: Velocity contour at Re=10000

5.4 Computational results

From the plots shown in Figure 14, the variation of Nusselt number with Reynolds number shows that all corrugated tube configurations perform significantly better than the plain tube. For α=15°, the smallest pitch of p=15 mm yields the highest heat transfer, reaching values above 250 at Re≈18,000 while larger pitches of p=20 and p=25 provide lower enhancement, with Nusselt numbers around 180–200 at the same Reynolds number. The plain tube, in contrast, only achieves a Nusselt number of about 110 at Re≈18,000. A similar trend is observed for α=20°, where Case 8 (p =15 mm, W =4.4 mm) achieves the maximum augmentation, confirming that reducing pitch and increasing inclination angle substantially intensify turbulence and improve convective heat transfer.

Frictional losses represent pressure drop across the tube and is calculation using $f = \frac{\Delta P}{0.5 \rho V_{in}^2} \frac{D_i}{L}$

The friction factor results reveal that this thermal enhancement comes with a corresponding rise in pressure drop. For α=15°, Case 4 (p = 15 mm, W = 4.4 mm) shows the highest friction factor, ranging between 0.27–0.31 across the Reynolds number range, compared to the plain tube which remains near 0.03. At α=20°, Case 8 records the overall maximum friction factor of about 0.28–0.42, making it the configuration with both the highest heat transfer and the highest flow resistance.

The effectiveness (η) results further highlight this trade-off as shown in Figure 13. It is calculated using $\eta = \frac{Nu/Nu_s}{(f/f_s)^{1/3}}$. At lower Reynolds numbers (Re≈6000), Case 8 achieves the highest effectiveness of about 1.57, while Case 5 also performs well at 1.35. As Reynolds number increases, effectiveness decreases for all cases, approaching values around 1.0–1.1 at Re≈18,000. Overall, Case 8 provides the optimal heat transfer performance and effectiveness, though with the largest hydraulic penalty, whereas configurations with larger pitch (e.g., Cases 2 and 3) show reduced thermal gains but lower frictional losses.

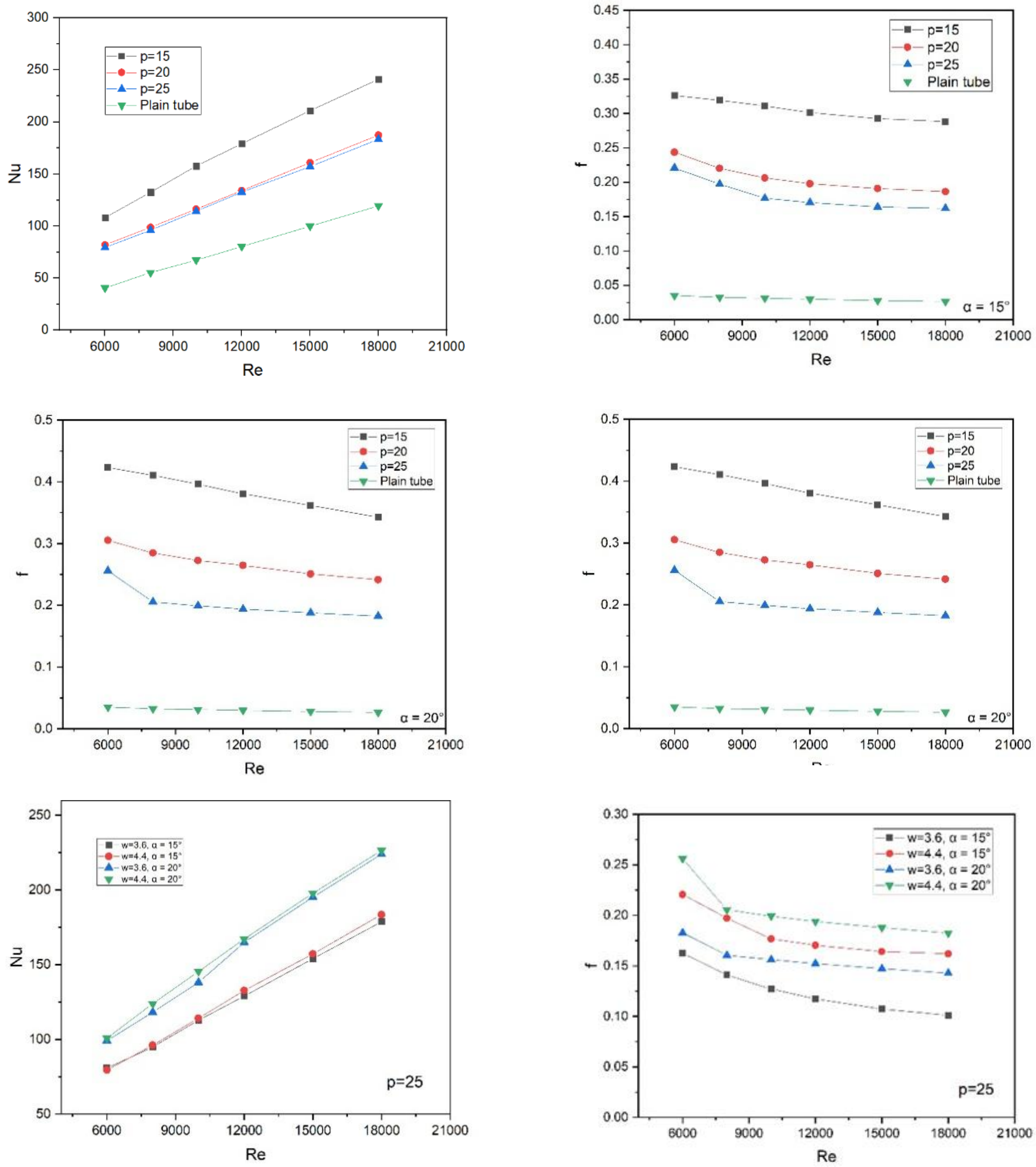


Figure 15: Effects of DPIE inclination and varying corrugation on Friction Factor and Nusselt Number

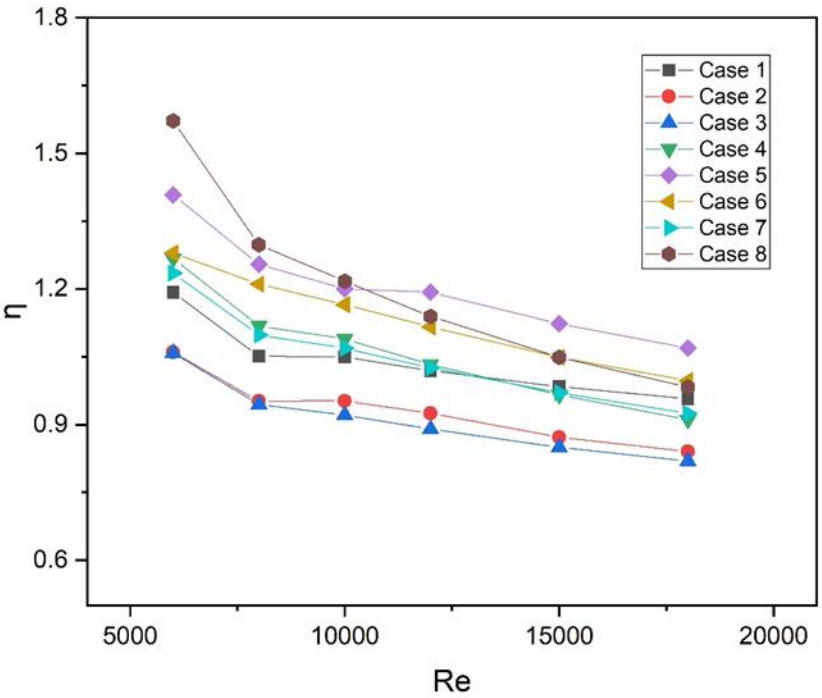


Figure 16: Thermal performance factor against Reynold's number for all cases

Chapter 6: Conclusion

In this study, numerical simulations were performed to analyze the heat transfer characteristics and thermal efficiency of a double-pipe heat exchanger featuring a corrugated inner tube and perforated inclined elliptical turbulators. The flow was considered turbulent, with the Reynolds number ranging from 6000 to 18000. Different configurations of corrugation width and pitch were examined to determine and assess their impact on pressure drop and heat transfer coefficient. Additionally, the inclination angle of the perforated elliptical vortex generators were varied for 15° and 20°.

In case 8, with the lowest pitch (p=15mm), showed the highest pressure drop in the inner pipe, being 11.687 times higher than smooth tubes without any turbulator. The inclination angle of 20° in cases 5, 6, 7 and 8 performed better in contrast to their counterparts resulting in higher pressure drop as well as higher Nusselt number. The results present that heat transfer can be increased by 230.6% in comparison to smooth pipe with no vortex generator. Nusselt number shows increment of 2.93%, 4.81% and 25.23% for inner corrugated pipe with p= 25 mm, 20 mm and 15 mm respectively, compared to typical double pipe heat exchanger with only perforted elliptical turbulator. The largest Nusselt number of 275.519 is obtained for inner corrugated pipe with W=4.4 mm, p=15 mm and vortex generator with inclination angle of 20° at Re=18000.

It is observed that reducing the corrugation pitch in the inner tube of the double-pipe heat exchanger leads to a significant rise in pressure drop. Among the cases studied, Case 8, with the smallest pitch, exhibited the highest friction factor of 0.423. Furthermore, increasing the corrugation width of the inner tube also contributed to higher pressure losses. Specifically, the friction factor increased by approximately 26.3% to 37.6% as the corrugation width was increased from 3.6 mm to 4.4 mm.

The analysis shows that efficiency drops as the Reynolds number increases, but corrugated tubes still perform better than smooth ones. At low Reynolds numbers, Case 8 stands out with the highest efficiency, while Cases 2 and 3 perform the weakest. Design changes such as reducing the pitch and increasing the slant angle to 20° boost heat transfer, though they also cause higher pressure drops. The maximum thermal efficiency (η) of 1.57 is obtained for double pipe heat exchanger fitted with inner corrugated tube and perforated elliptical vortex generator with p= 15 mm, W= 4.4 mm and α= 20° at Re=6000. These results emphasize the fundamental trade-off inherent in corrugated tube designs, wherein enhanced heat transfer performance is accompanied by a corresponding rise in pressure drop and the consequent demand for higher pumping power.

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