



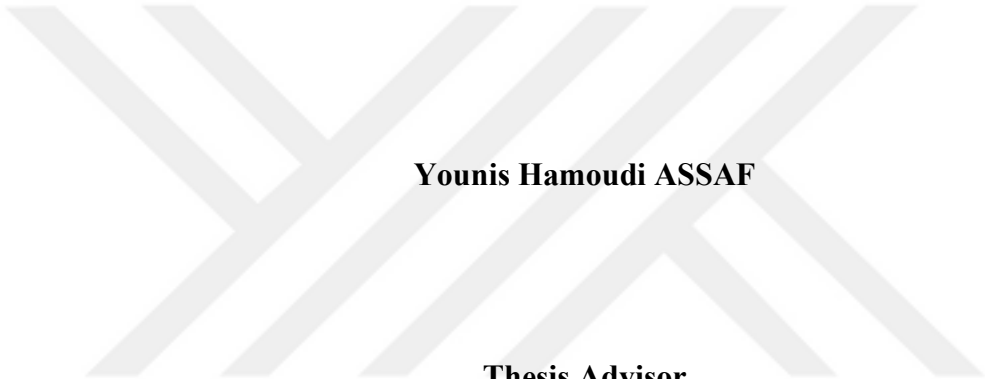
**EFFECT OF NANOMATERIAL ON THE
PERFORMANCE OF HEAT EXCHANGER WITH
INSERT RINGS AND TAPES**

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PhD Thesis
MECHANICAL ENGINEERING**

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“I declare that all the information within this thesis has been gathered and presented in accordance with academic regulations and ethical principles and I have according to the requirements of these regulations and principles cited all those which do not originate in this work as well.”

Younis Hamoudi ASSAF

ABSTRACT

Ph. D. Thesis

EFFECT OF NANOMATERIAL ON THE PERFORMANCE OF HEAT EXCHANGER WITH INSERT RINGS AND TAPES

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Department of Mechanical Engineering

Thesis Advisor:

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In this study, Improving heat exchanger performance remains a key focus in thermal engineering, particularly for applications requiring compact design, energy efficiency, and low operating costs. Passive optimization techniques, such as twisted ribbons and continuous rings, have been widely explored due to their ability to enhance turbulence, disrupt boundary layers, and enhance convective heat transfer without external energy. However, most previous studies have addressed these techniques independently and using conventional geometries, without comprehensively evaluating integrated continuous rings and advanced working fluids. This thesis investigates the thermo-hydraulic performance of an improved tubular heat exchanger with novel internal continuous rings and twisted ribbon configurations, using both water and an Al_2O_3 –Cu/water hybrid nanofluid.

The study is divided into three phases. In the first phase, a detailed numerical simulation was performed using water as the working fluid under turbulent flow conditions ($Re = 4000\text{--}14000$). Specially designed ring gaskets featuring oval holes, radial openings, and internal fins were independently evaluated. Ring gaskets were tested at four diameter ratios ($d/D = 0.55, 0.65, 0.75$, and 0.85) and four pitch ratios ($P/D = 4.5, 5.5, 6.5$, and 7.5). The results demonstrated a significant improvement in heat transfer.

The highest Nusselt number was achieved at a Reynolds number of 14,000 using water as the working fluid, with a ring gasket configuration of $d/D = 0.75$ and $P/D = 7.5$. Under these conditions, a Nusselt number of 214.5 was recorded, with a friction coefficient of 0.172. This combination achieved a maximum thermal performance factor (TPF) of 1.15. Optimum thermal performance was achieved at a Reynolds number of 4000 using water as the working fluid, with an annular packing configuration of $d/D = 0.65$ and $P/D = 5.5$. Under these conditions, the Nusselt number was recorded at 184.54, with a friction coefficient 0.191. This formulation achieved a maximum thermal performance factor (TPF) of 2.88, demonstrating an excellent balance between improved heat transfer and acceptable pressure drop.

In the second phase, the optimal geometry determined by hydrolysis was examined by introducing a twisted tape at twist ratios of 5.5, 10.5, and 15.5. The most significant improvement in Nusselt numbers and thermal performance coefficient was achieved at a twist ratio of 5.5. In the third phase of this part of the study, the replacement of the test fluid from water to a hybrid nanofluid composed of $Al_2O_3\text{--}Cu$ (1:1) dispersed in water at volume concentrations of 0.1%, 1%, and 2% was investigated. The addition of twisted

tape at a twisted ratio (5.5) and nanoparticles enhanced thermal conductivity and thermal performance. At a 2% concentration, an additional 42% increase in heat transfer was observed compared to a water-only baseline for the same inlet configuration, with only a moderate increase in pressure drop. Based on the combined results, the recommended optimal design is a ring-inlet configuration ($d/D = 0.65$, $P/D = 5.5$), a twisted ribbon ($TR = 5.5$), and a 2% $Al_2O_3\text{--}Cu$ /water hybrid nanofluid. This

system achieved the best thermo-hydraulic performance, making it suitable for industrial applications requiring compact and efficient heat exchangers. The proposed inlet designs and fluid compositions provide a reliable and cost-effective strategy for next-generation thermal systems.

Key Word : Internal ring insert, Twisted tape, Hybrid nanofluid, Heat transfer enhancement, Al_2O_3 –Cu/water, Turbulent flow..

Science Code : 91412



ÖZET

Doktora Tezi

NANOMALZEMENİN GEÇME HALKALI VE BANTLI ISI DEĞİŞTİRİCİ PERFORMANSINA ETKİSİ

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Bu çalışmada, ısı eşanjörü performansının iyileştirilmesi, özellikle kompakt tasarım, enerji verimliliği ve düşük işletme maliyetleri gerektiren uygulamalar için termal mühendislikte önemli bir odak noktası olmaya devam etmektedir. Bükülmüş şeritler ve sürekli halkalar gibi pasif optimizasyon teknikleri, türbülans artırma, sınır katmanları bozma ve harici enerji olmadan konvektif ısı transferini artırma yetenekleri nedeniyle yaygın olarak araştırılmıştır. Bununla birlikte, önceki çalışmaların çoğu, entegre sürekli halkaları ve gelişmiş çalışma sıvılarını kapsamlı bir şekilde değerlendirmeden, bu teknikleri bağımsız olarak ve geleneksel geometriler kullanarak ele almıştır. Bu tez, hem su hem de Al_2O_3 -Cu/su hibrit nanoakışkan kullanarak yeni iç sürekli halkalar ve bükülmüş şerit konfigürasyonlarına sahip gelişmiş bir borulu ısı eşanjörünün termo-hidrolik performansını araştırmaktadır.

Çalışma üç aşamaya ayrılmıştır. İlk aşamada, türbülanslı akış koşulları altında ($Re = 4000-14000$) çalışma sıvısı olarak su kullanılarak ayrıntılı bir sayısal simülasyon gerçekleştirilmiştir. Oval delikler, radyal açıklıklar ve iç kanatçıklar içeren özel olarak tasarlanmış halka contalar bağımsız olarak değerlendirilmiştir. Halka contalar dört çap oranında ($d/D = 0.55, 0.65, 0.75$ ve 0.85) ve dört hatve oranında ($P/D = 4.5, 5.5, 6.5$ ve 7.5) test edilmiştir. Sonuçlar ısı transferinde önemli bir iyileşme olduğunu göstermiştir.

En yüksek Nusselt sayısı, $d/D = 0.75$ ve $P/D = 7.5$ 'lik bir halka conta konfigürasyonu ile çalışma sıvısı olarak su kullanılarak 14.000 Reynolds sayısında elde edilmiştir. Bu koşullar altında, 0.172 sürtünme katsayısı ile 214.5 Nusselt sayısı kaydedilmiştir. Bu kombinasyon 1.15 'lik maksimum termal performans faktörüne (TPF) ulaşmıştır. Optimum termal performans, çalışma sıvısı olarak su kullanılarak 4000 Reynolds sayısında, $d/D = 0.65$ ve $P/D = 5.5$ halka şeklindeki salmastra konfigürasyonu ile elde edilmiştir. Bu koşullar altında, Nusselt sayısı 184.54 ve sürtünme katsayısı 0.191 olarak kaydedilmiştir. Bu formülasyon 2.88 'lik maksimum termal performans faktörüne (TPF) ulaşarak gelişmiş ısı transferi ve kabul edilebilir basınç düşüşü arasında mükemmel bir denge olduğunu göstermiştir.

İkinci aşamada, hidroliz ile belirlenen optimum geometri, $5.5, 10.5$ ve 15.5 büküm oranlarında bükülmüş bir bant eklenerek incelenmiştir. Nusselt sayıları ve termal performans katsayısındaki en önemli iyileşme 5.5 'lik büküm oranında elde edilmiştir. Çalışmanın bu bölümünün üçüncü aşamasında, test akışkanının su yerine $\%0.1, \%1$ ve $\%2$ hacim konsantrasyonlarında su içinde dağılmış Al_2O_3 -Cu'dan ($1:1$) oluşan hibrit bir nanoakışkanla değiştirilmesi incelenmiştir. Bükülmüş su ilavesi

bükümlü bir oranda (5.5) bant ve nanopartiküller termal iletkenliği ve termal performansı arttırmıştır. 2 'lik bir konsantrasyonda, aynı giriş konfigürasyonu için sadece su içeren bir taban çizgisine kıyasla ısı transferinde $\%42$ 'lik ek bir artış gözlenmiş ve basınç düşüşünde sadece orta düzeyde bir artış olmuştur. Birleşik sonuçlara dayanarak, önerilen optimum tasarım bir halka giriş konfigürasyonu ($d/D = 0.65, P/D = 5.5$), bükülmüş bir şerit ($TR = 5.5$) ve $\%2$ Al_2O_3 -Cu/su hibrit nanoakışkandır. Bu sistem en iyi termo-hidrolik performansı elde ederek kompakt ve

verimli ısı eřanjörleri gerektiren endüstriyel uygulamalar için uygun hale gelmiştir. Önerilen giriş tasarımları ve akışkan bileřimleri, yeni nesil termal sistemler için güvenilir ve uygun maliyetli bir strateji sağlamaktadır.

Anahtar Sözcükler :İç halka ekleme, Bükülmüş bant, Hibrit nanoakışkan, Isı transferi geliştirme, $\text{Al}_2\text{O}_3\text{-Cu/su}$, Türbülanslı akış.

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“Whoever walks the path of conviction, Allah will guide him to his destination.

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SYMBOLS AND ABBREVIATIONS

SYMBOLS

CFD	computational fluid dynamics [-]
C_p	heat capacity [kJ/kg.K]
D_o	outer diameter [cm]
d_i	inner diameter [cm]
f	friction factor
h	heat transfer coefficient [w/m ² .K]
k	thermal conductivity [w/m.K]
L	length [m]
Nu	nusselt number [-]
Pr	prandtl number [cm]
Q	heat transfer rate [watt]
Re	reynolds number [-]
T	temperature [°C]
TPF	thermal performance factor
u	velocity components [m/s]

GREEK SYMBOLS

Δp	Pressure drops [Pa]
β	The coefficient of thermal expansion
μ	Dynamic viscosity [Pa.s]
ν	kinematic viscosity [m ² /s]
Φ	volume fractions
A	Area m ²
Q'	Heat flux W/m
V	Kinematic Viscosity m ² /s

ρ	Density Kg/m^3
V	The velocity m/s
P	the pressure pa

Φ_{hnf}	The volume fraction of the hybrid
Φ_{np1} and Φ_{np2}	The volume fractions of the first and second nanoparticles, respectively.
ρ_{hnf}	The density of the hybrid nanofluid is denoted (kg/m^3).
ρ_{hnf} , and ρ_{np1}	The densities of nanoparticles 1 and 2.
$C_{p_{\text{hnf}}}$	The specific heat capacity of the hybrid nanofluid ($\text{J/kg}\cdot\text{K}$).
p_{np1} and $C_{p_{\text{np2}}}$ ($\text{J/kg}\cdot\text{K}$)	The specific heat capacities of nanoparticles 1 and 2.
K_{hnf}	The thermal conductivity of the hybrid nanofluid is given ($\text{W/m}\cdot\text{K}$).
k_{np1} and k_{np2}	The thermal conductivities of nanoparticles 1 and 2.
μ_{hnf}	The viscosity of the hybrid nanofluid is denoted as

PART1

INTRODUCTION

1.1. BACKGROUND AND MOTIVATION

Heat exchangers are fundamental components in a wide range of industrial processes, including power generation, chemical processing, HVAC (heating, ventilation, and air conditioning), and refrigeration systems [1]. By facilitating the transfer of thermal energy between two or more fluids at different temperatures, heat exchangers play a critical role in improving energy efficiency and reducing operational costs. Over the decades, considerable research has focused on enhancing the performance of conventional heat exchangers through both active and passive techniques. Among passive methods, the introduction of internal flow disturbances, such as twisted tapes, wire coils, and insert rings, has proven effective in augmenting convective heat transfer without requiring external power input [2].

Over the last few decades, as industries have moved toward compact, cost-effective, and energy-efficient solutions, the performance expectations for heat exchangers have grown significantly. Traditional shell-and-tube and plate heat exchangers, although widely used, often fall short in meeting the high heat transfer rates required in modern applications, especially when space and energy consumption are critical factors. Enhancing the rate of heat transfer without significantly increasing the size or energy input of the heat exchanger remains one of the fundamental challenges in thermal system design [3,4].

Two general strategies have emerged to enhance heat exchanger performance: active techniques and passive techniques. Active techniques, such as the use of electric fields, magnetic fields, or mechanical agitation, require external power input. While effective, they are often expensive to implement and maintain, especially in large-scale or continuous operations. On the other hand, passive techniques, which modify the internal geometry of the heat exchanger or utilize special working fluids, do not require additional energy input and are generally simpler, more economical, and more robust for industrial applications [5].

Among passive methods, insertion of turbulence promoters, such as twisted tapes, wire coils, baffles, and ring inserts, has shown significant promise. These components disturb the boundary layer of the flow, induce secondary swirling motions, and increase the effective heat transfer surface area [6]. Twisted tape inserts, in particular, are well-known for their ability to promote strong tangential and axial mixing, improving convective heat transfer without moving parts [7]. Similarly, ring inserts positioned inside the tube can act as flow obstructions that create wake regions and recirculation zones, enhancing heat transfer by disrupting the thermal boundary layer and promoting flow mixing [8].

Parallel to structural modifications, the development of advanced working fluids, most notably nanofluids, has opened new avenues for thermal performance enhancement. Nanofluids, engineered by dispersing nanoparticles (e.g., metal oxides, carbon nanotubes) into conventional base fluids, exhibit superior thermophysical properties, including higher thermal conductivity and modified viscosity profiles. These enhanced properties translate into improved convective heat transfer coefficients when nanofluids flow through heat exchangers [9,10]. Despite their promise, practical deployment of nanofluids is hindered by issues such as particle stability, potential erosion of surfaces, and increased pressure drop due to higher viscosity. Research efforts have thus focused on characterizing the trade-offs involved and identifying optimal nanoparticle types, concentrations, and dispersion methods [11].

Combining passive structural enhancements with nanofluid technology presents a particularly attractive strategy for maximal heat exchanger performance gains. Recent

studies have explored hybrid approaches, such as twisted-tape inserts with nanofluid coolants and wire coil-inserted tubes conveying nanoparticle suspensions [9,12]. Yet, there remains a significant gap in understanding the synergistic effects of complex insert geometries, like specially designed fixed rings, and advanced working fluids under turbulent and laminar flow regimes. Moreover, the long-term stability of nanofluids when interacting with insert surfaces, as well as their impact on overall system reliability and maintenance requirements, have received limited attention.

The motivation for the present study stems from the urgent need to develop compact, high-efficiency heat exchangers that can meet increasingly stringent energy and environmental regulations. By investigating the combined use of nanomaterials and novel insert-ring/tape configurations, this research aims to:

- Quantify the heat transfer enhancement and pressure drop characteristics of hybrid systems under varying operational conditions.
- Determine the optimum combination of nanoparticle concentration, insert geometry, and flow parameters that maximizes thermal performance while minimizing pumping power.
- Assess the thermal-hydraulic performance trade-offs and provide design guidelines for reliable, cost-effective heat exchanger applications in power generation and process industries.

1.2. IMPORTANCE OF HEAT EXCHANGERS IN ENERGY SYSTEMS

Heat exchangers lie at the very heart of modern energy systems, serving as crucial enablers of efficient heat transfer between process streams. Their widespread application spans power generation, industrial processing, HVAC (heating, ventilation, and air-conditioning), refrigeration, and increasingly, renewable and waste-heat recovery systems. Understanding their importance requires examining how they influence system efficiency, operational flexibility, environmental impact, and economic viability [13,14].

With the integration of heat exchangers in these systems, we can enhance energy conversion efficiency, minimize system footprint, and improve long-term sustainability. Enhanced designs using nanofluids and innovative inserts, as investigated in this thesis, can further augment the thermal efficiency of renewable energy systems without the need for costly structural redesigns [15,16].

Figure 1 illustrates the broad spectrum of applications where heat exchangers are essential for thermal regulation and energy efficiency. These devices serve as fundamental components in building heating and cooling systems, ensuring optimal indoor climate control across residential, commercial, and industrial environments. In the renewable energy sector, heat exchangers are integral to systems like solar thermal collectors and wind energy setups, where efficient heat transfer is vital for energy conversion and storage. Aviation and automotive applications rely heavily on compact, high-efficiency heat exchangers to manage engine temperatures, cabin climate, and electronic systems within constrained spaces. The cooling of electronics, particularly in high-performance computing and power electronics, demands advanced heat exchanger designs to dissipate concentrated thermal loads while maintaining component reliability. In oil and gas industries, large-scale heat exchangers are utilized in refining, separation, and processing operations to optimize thermal energy usage and enhance system productivity. This wide-ranging applicability underscores the importance of continued innovation in heat exchanger technology to meet the growing demands for thermal efficiency, environmental sustainability, and performance optimization across multiple sectors.

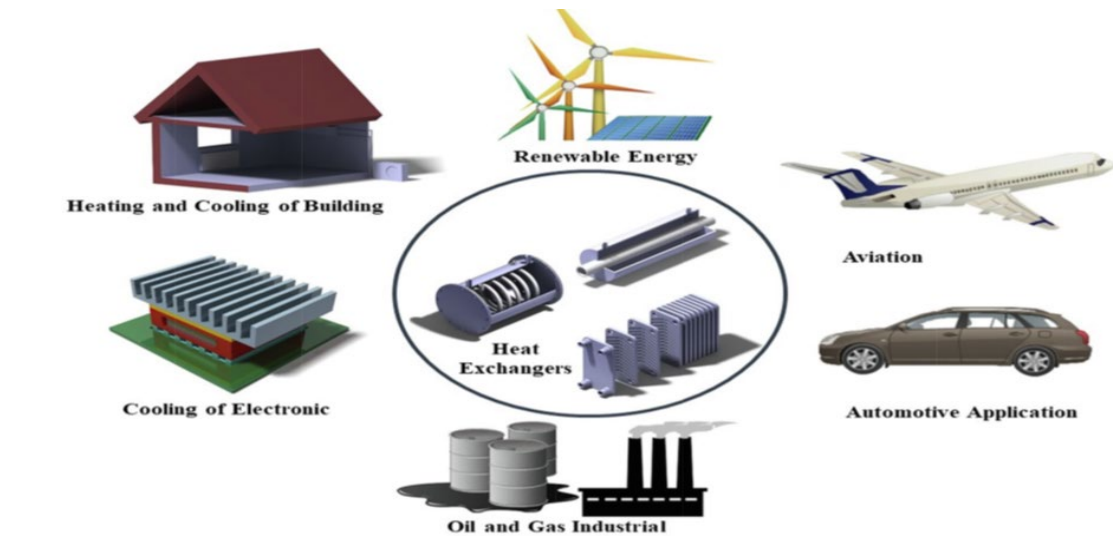


Figure 1.1. A few applications for heat exchangers in various devices

Figure 1.2. provides a schematic overview of the central role heat exchangers play across various engineering domains. Positioned at the heart of the diagram, heat exchanger applications are depicted as integral to multiple sectors, each connected through directional arrows. In the oil and gas industry, heat exchangers are utilized in refining, separation, and liquefaction processes to optimize thermal energy transfer and enhance operational efficiency. Cooling of electronic components, such as CPUs and power electronics, is another critical domain where compact and high-performance heat exchangers prevent overheating and ensure device reliability. In aviation, these systems are embedded within engines, environmental control systems, and avionics, enabling thermal management under high-altitude and variable load conditions. The automotive industry also relies on them for engine cooling, battery thermal regulation in electric vehicles, and passenger cabin climate control. Renewable energy systems, including solar thermal collectors and geothermal units, use heat exchangers to convert and store energy efficiently. Lastly, in heating and cooling of buildings, they are key to HVAC systems, ensuring energy-efficient indoor temperature control. This figure underscores the wide applicability and technical importance of heat exchangers in achieving sustainability, performance, and thermal reliability in modern engineering systems.

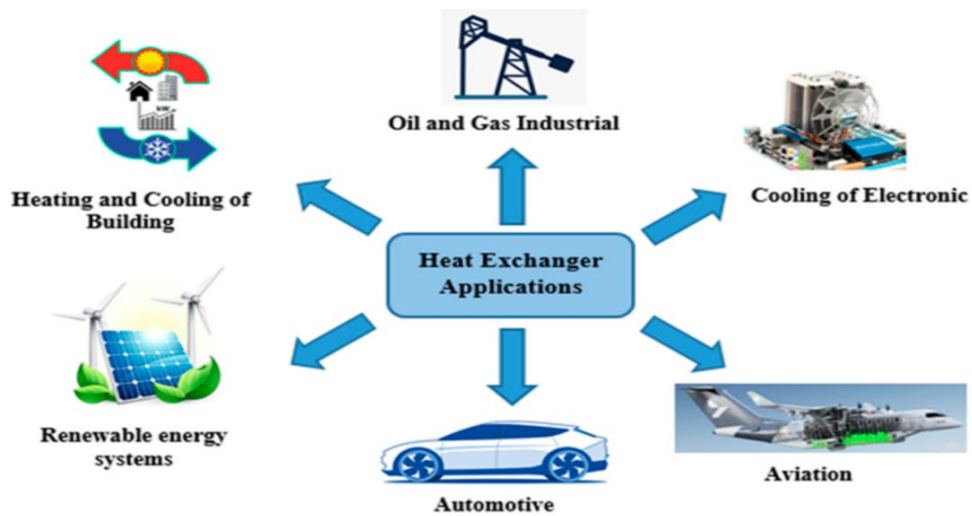


Figure 1.2. A few applications for heat exchangers

1.2.1. Applications in the Transportation Sector

Another critical application of heat exchangers lies in the transportation industry, particularly in automotive and aerospace engineering. In conventional vehicles, heat exchangers are employed in engine cooling systems, exhaust gas recirculation, and transmission oil cooling. With the advent of electric vehicles (EVs), their role has expanded into battery thermal management systems, where maintaining the battery pack within safe temperature ranges is essential for longevity and performance [17].

In aviation, where weight, space, and efficiency are tightly constrained, compact and high-performance heat exchangers are indispensable. They are integrated into environmental control systems (ECS), fuel systems, and onboard electronics. Given the extreme operating conditions, such as high altitude and large thermal gradients, heat exchangers must maintain thermal balance without compromising on weight or safety standards. The development of lightweight, high-efficiency heat exchangers, such as those using aluminum and nanofluid technology, has significantly improved thermal regulation in modern aircraft [18].

1.2.2. Importance in Renewable Energy Systems

As the global energy paradigm shifts toward renewables, heat exchangers have also gained prominence in solar thermal systems, geothermal installations, and biomass conversion technologies[1]. In solar thermal energy, flat-plate and evacuated tube collectors use heat exchangers to transfer the absorbed solar energy to a fluid, which can then be used for domestic heating, industrial processing, or electricity generation through Organic Rankine Cycles (ORCs) [19]

In geothermal systems, underground heat is extracted using working fluids circulated through heat exchangers, often under varying pressure and temperature conditions. Here, the durability and efficiency of the heat exchanger play a critical role in determining the system's reliability and overall coefficient of performance (COP) [20]. Additionally, in waste-to-energy plants and bioenergy facilities, heat exchangers allow for the safe and efficient transfer of heat from combustion gases to water or other secondary fluids.

1.2.3. Use in HVAC and Building Systems

In residential and commercial buildings, heat exchangers are used in HVAC systems to control temperature, humidity, and air quality. They play a crucial role in reducing energy consumption in buildings, which account for nearly 40% of global energy use [21]. By recovering waste heat from exhaust air and redirecting it to incoming fresh air (in air-to-air heat exchangers), or by transferring heat between fluids in boilers and chillers, these systems improve energy efficiency and reduce operating costs [22].

1.2.4. Cooling of Electronics and High-Performance Systems

As electronic devices and computational systems become more powerful, their thermal loads increase significantly. Servers, data centers, power electronics, and high-frequency components generate substantial heat that must be dissipated quickly and effectively to prevent system failure. Compact heat exchangers, including

microchannel and mini-fin types, are employed in these contexts to maintain thermal stability [23,24].

The introduction of nanofluid-based cooling and micro-structured inserts can drastically improve heat dissipation capabilities, as shown in recent research [25,26] and highlighted in this thesis. This is particularly important in sectors such as telecommunications, defense, and artificial intelligence (AI) computing, where system reliability and performance are closely tied to thermal management [27,28].

1.2.5. Environmental and Economic Impacts

Beyond their technical function, heat exchangers contribute to environmental conservation by reducing greenhouse gas emissions, fuel consumption, and thermal pollution. The use of high-efficiency exchangers in industrial processes minimizes the need for fossil fuel combustion and enhances the effectiveness of heat recovery systems [29].

Economically, improved heat exchanger design reduces operational costs through lower energy bills and maintenance requirements. When advanced fluids and insert technologies are applied, as proposed in this thesis, the performance-to-cost ratio of thermal systems can be significantly optimized. These cost savings can be particularly impactful in energy-intensive industries, municipal infrastructure, and developing countries where energy access and affordability remain critical concerns [30,31].

1.3. PASSIVE ENHANCEMENT TECHNIQUES

The continuous quest to improve heat exchanger efficiency has driven the exploration of various heat transfer enhancement methods. These techniques are generally categorized into active and passive approaches [32]. While active techniques require external energy input (e.g., vibration, mechanical agitation, or electric or magnetic fields), passive techniques enhance heat transfer through geometric or surface modifications without any additional power consumption [33]. Due to their simplicity,

low cost, and ease of integration, passive enhancement techniques have received substantial attention in industrial and academic research [6].

Among the diverse passive methods developed, internal inserts—such as twisted tapes, wire coils, baffles, and ring turbulators- have been widely employed due to their ability to disturb flow patterns, promote turbulence, extend flow paths, and reduce thermal boundary layer thickness [34,35]. These inserts modify the internal geometry of the heat exchanger tube, causing the fluid to follow a more complex trajectory, which leads to improved heat transfer rates.

There are many advantages of passive insert techniques, which are summarized as [36]:

- The widespread application of insert-based passive enhancement techniques is driven by several advantages:
- Inserts work solely by altering the flow dynamics, making them highly energy efficient.
- Twisted tapes and rings can be fabricated using conventional manufacturing processes and can be inserted into existing heat exchangers.
- Inserts can be used with various working fluids, including nanofluids, refrigerants, oils, and air.
- The approach is applicable across a range of sizes and heat exchanger types (shell-and-tube, double pipe, microchannels).

Despite these benefits, the challenge lies in optimizing geometry to achieve a high TPF, especially in systems with long-term operating demands where pressure drop and maintenance costs become significant [37].

The current research builds upon the fundamental understanding of passive insert performance by integrating advanced hybrid nanofluids with innovative insert geometries. This dual approach aims to unlock new performance thresholds in heat exchangers. The synergy between the fluid's enhanced thermal conductivity and the

inserts induced turbulence leads to significantly higher heat transfer rates than either method alone.

1.3.1. Twisted Tape Inserts

Twisted tape (TT) inserts are among the most extensively studied passive devices. These are thin metal strips twisted to form a helical shape and inserted into the flow path of the heat exchanger [38]. When fluid flows through a tube with a twisted tape, it is forced to rotate along the axis of the tube, which increases turbulence intensity and promotes mixing between the fluid layers. This swirling motion increases the residence time of the fluid near the tube wall, enhances the temperature gradient, and delays the growth of the thermal boundary layer, all of which contribute to a higher Nusselt number and convective heat transfer coefficient [39,40].

Figure 3 illustrates a single twisted-tape insert mounted coaxially within a circular tube of inner diameter D , where a helical strip of width d and thickness spanning the tube's cross section is twisted through a pitch length H (one full 360° turn) over a total insert length L . As fluid enters the tube, the narrow gap between tape and wall induces a strong swirl that continuously renews the thermal boundary layer, enhancing the local Nusselt number; reducing H (tighter twist) intensifies this swirl but also increases the pressure drop, so d , H , and L must be optimized to balance heat-transfer gains against pumping power. Adequate L ensures that the induced swirl persists throughout the heated section, while maintaining a clearance of $(D-d)/2$ prevents mechanical interference with the tube wall.

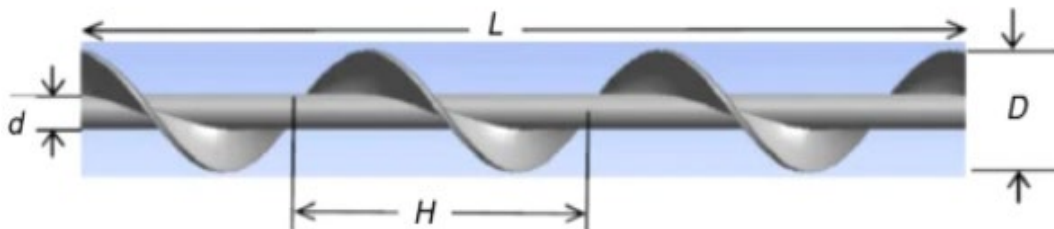


Figure 1.3. Schematic of Twisted-Tape Insert in a Circular Tube [41].

The working principle of twisted tape inserts relies on the generation of secondary flows and centrifugal forces that intensify the mixing of fluid particles. Unlike active techniques, twisted tapes do not require external power and can be easily retrofitted into existing systems, making them cost-effective and energy-efficient solutions [38].

1.3.1.1. Mechanisms of Heat-Transfer Enhancement in Twisted-Tape Inserts

The enhancement achieved through twisted tape inserts is primarily due to [42]:

- The helical structure forces the fluid to follow a longer, spiral path. This motion increases residence time and promotes axial and radial mixing, which is especially effective in the near-wall region where the temperature gradient is high.
- The swirling motion thins the thermal boundary layer, which reduces resistance to heat transfer at the tube wall. This increases the Nusselt number (Nu), a dimensionless indicator of convective heat transfer.
- As the fluid takes a longer route through the tape-induced spiral path, it has more opportunity to exchange heat with the wall.
- The interaction between the swirling flow and the tube wall results in secondary vortices that further intensify turbulence and mixing.

1.3.1.2 Variants of Twisted Tape Inserts

To further improve performance or reduce pressure losses, many modeling twisted tape geometries have been proposed [40,43]:

- **Oblique or Alternate Twist Tapes:** These change the twist direction periodically to generate complex secondary flow patterns.
- **Perforated Twisted Tapes:** These include punched holes or slots that allow partial fluid bypass and reduce pumping power while sustaining turbulence.
- **Tapered or Wavy Twisted Tapes:** These alter the swirl intensity along the length of the tube and can be optimized for flow conditions.

- **V-Cut and Double-V Twisted Tapes:** These have periodic V-shaped notches along the edge, enhancing turbulence through repeated flow disturbance.

Each of these modifications is designed to either increase heat transfer, decrease pressure loss, or both, depending on the application requirements.

1.3.1.2. Performance Metrics of Twisted Tape Inserts

The effectiveness of twisted tapes is typically evaluated using [44,45]:

- Nusselt Number (Nu), which measures convective heat transfer enhancement.
- Friction Factor (f) indicates the resistance imposed by the insert, which correlates with the pressure drop.
- Thermal Performance Factor (TPF), which is a ratio that balances Nu enhancement against the increase in friction factors. $TPF > 1$ means effective enhancement.

While twisted tapes significantly increase Nu (sometimes by over 50%), they also cause a rise in the friction factor due to the flow resistance. Thus, optimizing the twist ratio and geometry is critical to achieving a high TPF.

1.3.2. Ring Inserts

Ring inserts are a class of passive heat transfer enhancement devices used inside tubes of heat exchangers to promote flow disturbance, intensify turbulence, and improve thermal performance [46]. These devices are typically installed along the axial direction of the tube in a repetitive pattern and act by periodically interrupting the axial fluid flow, generating recirculation zones and vortex structures that increase convective heat transfer near the tube walls [47].

Unlike twisted tapes, which induce continuous swirling motion, ring inserts work by creating discrete flow obstructions, encouraging localized boundary layer renewal, increasing heat transfer surface area, and prolonging the fluid's contact time with

heated walls. Their effectiveness lies in their ability to enhance heat transfer significantly with minimal modification to the existing heat exchanger structure [48].

Figure 1.4 illustrates how inserting a perforated annulus into a circular tube passively modifies the flow to enhance heat transfer: When the blue streamlines meet the annulus's mesh, portions of the fluid are forced through its perforations and around its bars, creating local separation at the leading edges. Downstream, these separated flows form recirculation zones that draw the cooler core fluid toward the hot wall, thinning the thermal boundary layer. By repeating this turbulence in each annulus, the device maintains high turbulence and significantly increases the local Nusselt number, albeit at the cost of additional pressure drop.

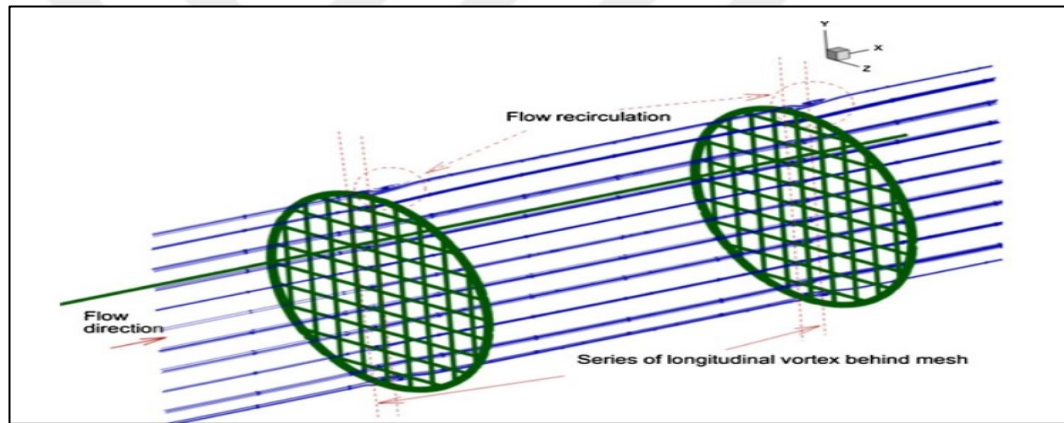


Figure 1.4. Schematic of Ring Insert in a Circular Tube [49]

1.3.2.1. Working Mechanism of Ring Inserts

The enhancement achieved using ring inserts is a result of several interacting phenomena [48,50]:

- Rings serve as baffles that periodically interrupt the boundary layer, allowing it to redevelop repeatedly. This delays the thermal boundary layer thickness and improves heat transfer.
- Recirculation and wake formation cause the separation of flow and the formation of low-pressure recirculation zones downstream of each ring, enhancing mixing between fluid layers.

- Rings can increase the surface area available for heat transfer.
- Like other passive devices, rings promote radial and transverse mixing, especially when combined with turbulence promoters like fins or perforations.

1.3.2.2. Geometrical Parameters Influencing the Performance of Ring Inserts

Several parameters determine the effectiveness of ring inserts [51,52]:

- The ratio of the inner diameter of the ring to the inner diameter of the tube (d/D Ratio). Smaller d/D means more blockage and higher turbulence, but also increased pressure drop.
- The distance between two consecutive rings to the tube diameter (Pitch Ratio (P/D)). Closer spacing (low P/D) increases mixing but may raise flow resistance.
- Thicker rings provide more surface area but can increase hydraulic loss.

1.3.2.3. Variants of Ring Inserts

To further improve performance or reduce pressure losses, many modiling ring designs have been proposed [53,54]:

- **Solid Rings:** Provide the highest obstruction but also the largest pressure drop.
- **Perforated Rings:** Have multiple small holes to promote turbulence while controlling resistance.
- **Slotted Rings:** Include longitudinal or radial slots that allow partial flow-through, reducing pressure losses.
- **Finned or Ribbed Rings:** Include attached fins or ridges to extend the surface area and enhance turbulence further.

1.3.2.4. Thermal–Hydraulic Performance of Ring Inserts

Ring inserts are evaluated based on [52,53]:

- Nusselt Number (Nu), which reflects convective heat transfer enhancement. The Nu generally increases with the use of ring inserts, indicating enhanced heat transfer.
- Friction factor (f), which indicates the increase in flow resistance.
- Thermal Performance Factor (TPF), which combines both Nu and f to assess the overall efficiency of the enhancement technique.

1.3.2.5. Advantages and Limitations of Using Ring Inserts

There are many advantages to using ring inserts, which can be summarized as follows [55,56]:

- Rings can be made of metal or polymer and inserted into tubes without the need for complex manufacturing processes.
- They can be added to existing systems without redesigning the heat exchanger casing.
- Engineers can adjust the temperature, diameter, perforation, and thickness to meet specific thermal requirements.
- Excellent compatibility with nanofluids. Their ability to disrupt the flow of suspended nanoparticles improves thermal performance when used with nanofluids.

There are also some limitations to using ring inserts, which can be summarized as follows [57,58]:

- The flow obstruction leads to additional pressure loss. This must be accounted for in the pumping system design.
- In certain environments (e.g., chemical plants), the rings can accumulate deposits, requiring regular cleaning.
- Unlike twisted tapes, rings do not induce axial swirl, which may limit mixing in laminar flow unless used in conjunction with other inserts.

1.3.3. Combined Insert Configurations

As global energy demands rise and system efficiency becomes more critical, single-method heat transfer enhancements, such as twisted tapes or ring inserts alone, may no longer suffice to meet advanced thermal performance requirements [32]. A growing body of research is now focusing on combined insert configurations, wherein two or more passive techniques are employed simultaneously to generate synergistic thermal enhancement. The integration of twisted tape inserts and ring turbulators is a leading example of such compound passive methods, aiming to combine their unique flow modification mechanisms to maximize convective heat transfer while keeping pressure drop within acceptable limits [59].

Figure 5 shows a hybrid passive device combining a twisted-tape and ring inserts: the helical tape (solid sinusoidal line) spans the length of the tube, continuously imposing swirl, while discrete ring inserts (dashed cones) are mounted at regular intervals. As the inflow enters, the twisted tape forces the fluid into a sustained helical motion, thinning the thermal boundary layer along the entire length; each ring then acts as a reset point, creating local separation and recirculation zones before the swirl re-energizes downstream clearances.

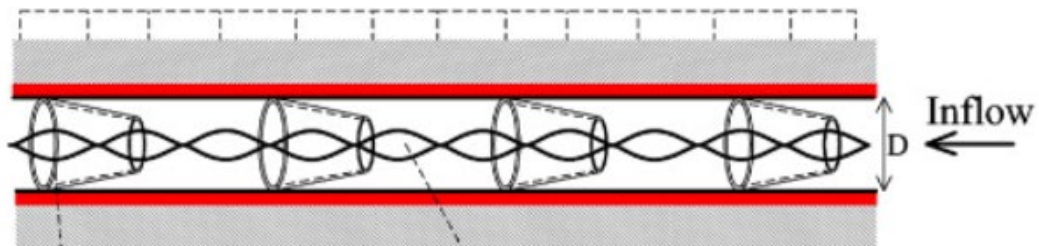


Figure 1.5. Hybrid Insert Configuration: Twisted Tape Integrated with Ring Inserts in a Circular Tube [60]

1.3.3.1. Thermal Enhancement Mechanism of Combined Inserts

In a heat exchanger tube fitted with both twisted tape and ring inserts, the combined configuration enhances heat transfer by engaging multiple flow disruption mechanisms [61]:

- The twisted tape induces helical motion in the flow, promoting strong axial and radial mixing and thinning of the thermal boundary layer near the tube wall.
- The rings act as baffles that obstruct and periodically disturb the flow, creating localized turbulence and continuously interrupting the reformation of boundary layers.
- When these two mechanisms are combined, the swirl flow induced by the tape intensifies the recirculating zones formed by the rings. This leads to enhanced fluid agitation, thermal homogenization, and better interaction with the wall surface.
- As the flow moves downstream, the swirling motion of the fluid from the twisted tape combines with the ring-induced wake structures to periodically renew both the hydrodynamic and thermal boundary layers.

1.3.3.2. Design and Parametric Considerations for Combined Inserts

For optimal performance, several geometric and operational parameters must be considered when designing combined insert configurations [61,62]:

- **Twist Ratio (TR):** The degree of twist affects swirl intensity. Lower TR (e.g., 5) enhances mixing but increases pressure drop. Higher TR (e.g., 15) provides milder enhancement with lower hydraulic resistance.
- **Ring Pitch (P/D):** The spacing between consecutive rings (relative to tube diameter) influences the frequency of flow interruption. Small P/D increases turbulence but also raises flow resistance.
- **Ring Diameter Ratio (d/D):** Affects the blockage area. Lower d/D ratios produce stronger vortex shedding; higher ratios reduce resistance but also reduce thermal impact.
- **Insert Alignment:** Rings can be placed upstream or downstream of the twisted tape, or interspersed along its length. Each arrangement produces different turbulence patterns.
- **Insert Shape:** Perforated and slotted rings combined with notched or V-cut twisted tapes allow for turbulence modulation and reduced drag.

1.3.3.3. Expected Performance Benefits of Hybrid Ring–Twisted-Tape Inserts

Multiple studies have indicated that combined configurations outperform single-insert designs in the following ways [63,64]:

Heat transfer enhancement is often 30–70% higher in combined configurations compared to a plain tube, and 10–25% higher than using twisted tapes or rings alone.

The balance between increased heat transfer and pressure drop leads to TPF values typically greater than 2.0, and in optimized designs, up to 3.5.

With superior performance per unit length, the required size of the heat exchanger can be reduced, making systems more compact and cost-effective.

Combined inserts improve the suspension and distribution of nanoparticles in nanofluids, preventing settling and improving overall thermal conductivity.

1.3.3.4. Challenges in Implementing Hybrid Ring–Ring–Twisted-Tape Inserts

Despite their advantages, combined insert configurations are not without drawbacks [65]:

- The additive nature of resistance from both inserts can significantly raise the pressure drop. This must be managed via perforations, spacing optimization, or higher-efficiency pumps.
- The interaction of swirl and wake can sometimes lead to non-uniform flow fields or stagnant zones, requiring careful design and CFD validation.
- Designing and aligning both inserts correctly inside a narrow tube may increase fabrication and assembly costs.

1.4. ROLE OF NANOFLUIDS IN HEAT TRANSFER ENHANCEMENT

The performance of thermal systems such as heat exchangers, radiators, and electronic cooling devices depends heavily on the properties of the working fluid. Traditionally, fluids such as water, ethylene glycol, and engine oil have been used as heat transfer media due to their favorable availability and thermophysical properties. However, these conventional fluids possess inherently low thermal conductivities, which limits their ability to meet the growing thermal demands of modern energy systems. In response to this challenge, nanofluids have emerged as a promising solution capable of significantly enhancing heat transfer performance [66].

1.4.1. Introduction to Nanofluids

Nanofluids are engineered colloidal suspensions of nanometer-sized particles (1–100 nm) in a base fluid. The nanoparticles are typically metals (e.g., Cu, Ag), metal oxides (e.g., Al_2O_3 , TiO_2 , CuO), or carbon-based materials (e.g., CNTs, graphene). These particles possess higher thermal conductivity than the base fluids and, when dispersed uniformly, enhance the fluid's overall thermal conductivity, heat capacity, and convective heat transfer characteristics [12].

The concept of nanofluids was first introduced by Choi in 1995 at Argonne National Laboratory, sparking a wave of experimental and computational studies that confirmed their superiority over conventional fluids [67]. With only a small volume fraction of nanoparticles (typically 0.1%–1.0%), nanofluids can offer considerable improvements in thermal performance without significantly increasing fluid viscosity or pumping power requirements [68].

The high surface-to-volume ratio of nanoparticles enhances heat transfer by increasing the contact area and promoting energy exchange between the particles and the fluid molecules. Nanofluids have since emerged as a key tool for engineers aiming to develop compact, high-efficiency, and environmentally friendly thermal systems [69].

1.4.2. Mechanisms of Heat Transfer Enhancement in Nanofluids

Nanofluids improve thermal performance through several physical mechanisms, many of which act simultaneously [70]:

- Nanoparticles have higher thermal conductivity than the base fluid. Their uniform dispersion enhances the fluid's overall thermal conductivity, promoting better conduction.
- Nanoparticles near the tube wall disturb the insulating boundary layer, reducing thermal resistance.
- The random motion of nanoparticles increases micro-convection and disrupts thermal boundary layers, improving energy exchange.
- In some applications, nanofluids with phase-change materials or encapsulated particles add to heat capacity through absorption or release of latent heat.
- Nanoparticle clustering and thermophoresis generate complex flow fields and additional energy transport modes, especially under temperature gradients.

The combination of these effects allows nanofluids to significantly boost convective heat transfer, especially in turbulent and transitional flow regimes commonly encountered in practical heat exchanger systems.

1.4.3. Types of Nanofluids

Nanofluids are typically classified into two main categories:

- **Mono Nanofluids:** Contain a single type of nanoparticle (e.g., Al_2O_3 -water, Cu-ethylene glycol) [71].
- **Hybrid Nanofluids:** Consist of a combination of two or more types of nanoparticles dispersed in a base fluid [71]. For example, Al_2O_3 -CuO/water is a hybrid nanofluid used in this thesis.

Hybrid nanofluids aim to combine the favorable properties of different nanoparticles, such as high thermal conductivity from one and chemical stability from another. This

synergy results in superior thermal and rheological properties compared to mono-nanofluids.

1.4.4. Parameters Affecting Nanofluid Performance

The performance of nanofluids in heat exchangers depends on multiple interrelated parameters [72,73]:

- **Volume Fraction (ϕ):** Increasing nanoparticle concentration generally enhances thermal conductivity but also increases viscosity and pressure drop. Optimal values are typically between 0.3% and 1.0%.
- **Base Fluid:** Water, ethylene glycol, and oil behave differently depending on thermal load, pressure, and temperature.
- **Temperature and Flow Regime:** Nanofluids often exhibit greater enhancement under turbulent flow and at higher temperatures.
- **Particle Material and Size:** Materials with higher thermal conductivity and smaller particle sizes offer better performance.
- **Stability and Dispersion:** Uniform dispersion is critical. Sedimentation and agglomeration reduce effectiveness and can damage equipment over time.

In this thesis, three volume fractions of Al_2O_3 -Cu hybrid nanofluids (0.1%, 1%, and 2%) were tested under turbulent flow conditions with Reynolds numbers ranging from 4000 to 14,000.

1.4.5. Performance of Nanofluids Benefits in Heat Exchangers

Numerous studies have validated the benefits of nanofluids in different heat exchanger configurations [9]:

- Enhancements of 20–60% Nusselt Number compared to base fluids have been reported under similar flow conditions.
- Nanofluids help reduce temperature gradients across the system, increasing the reliability and lifespan of components.

- Due to their superior thermal properties, nanofluids allow for downsizing heat exchangers without compromising efficiency.
- In systems with small temperature differences, nanofluids maintain high heat transfer rates where traditional fluids underperform.

In this thesis, three volume fractions of Al_2O_3 -Cu hybrid nanofluids (0.1%, 1%, and 2%) were tested under turbulent flow conditions with Reynolds numbers ranging from 4000 to 14,000.

1.4.6. Nanofluid Interaction with Passive Enhancements

The true potential of nanofluids is often realized when they are combined with passive heat transfer techniques such as twisted tapes and ring inserts. This hybrid enhancement strategy maximizes the benefit of both fluid and geometric modifications [11,74,75]:

- Ring inserts create wake regions that further disrupt the boundary layer and promote enhanced mixing.
- Inserts like twisted tapes induce turbulence and swirl, keeping nanoparticles suspended and evenly distributed.
- The synergy of enhanced flow dynamics and superior fluid properties leads to remarkable improvements in heat transfer performance.

1.4.7. Practical Implementation Issues and Challenges of Nanofluids

Despite their potential, nanofluids present several practical challenges that must be addressed for real-world applications [9]:

- Nanoparticles can settle over time, reducing effectiveness. Use of surfactants or ultrasonication is often necessary.
- High-purity nanoparticles and stabilization processes can increase operational costs.
- Some nanoparticles may accelerate wear in metallic components.

- Lack of universal models and inconsistent experimental data hinder industrial adoption.

Research continues into developing cost-effective, stable, and highly conductive nanofluids that can be reliably integrated into commercial systems.

Despite their promise, nanofluids also present challenges [76,77]:

- Without proper dispersion methods, particles can settle, clogging channels or reducing performance.
- Increased viscosity leads to greater pressure drop and pumping power, especially at higher concentrations.
- Nanoparticles and preparation methods can be expensive or difficult to scale.
- The interaction between nanoparticles and the tube material needs to be considered in the design.

Overcoming these challenges requires careful material selection, fluid formulation, and system integration strategies.

1.5. RESEARCH GAP AND NOVELTY OF THE CURRENT WORK

In recent decades, numerous studies have explored passive heat transfer enhancement methods in an effort to improve the performance of conventional heat exchangers. Devices such as twisted tape inserts, ring turbulators, and other geometric modifications have been widely investigated, each demonstrating the ability to enhance convective heat transfer by disrupting laminar sublayers, inducing turbulence, and increasing residence time near the heat transfer surfaces. Similarly, the use of nanofluids has gained significant attention due to their potential to improve thermal conductivity and convective heat transfer performance. Despite these advances, substantial gaps remain in understanding the combined effects of these enhancement strategies under realistic operating conditions, particularly with hybrid nanofluids and novel insert geometries.

1.5.1. Limitations in Existing Literature

A review of the existing literature reveals several critical limitations:

- Many studies evaluate either twisted tape inserts or ring turbulators in isolation. While these approaches provide valuable insights into individual performance characteristics, they fail to address the interactive behavior of multiple passive inserts when combined in a single flow system.
- The majority of compound enhancement studies focus on active-passive combinations, often requiring external energy input. There is relatively little systematic work on fully passive compound techniques, such as integrating twisted tapes with ring inserts, particularly in turbulent flow regimes.
- While mono-nanofluids like Al_2O_3 –water or CuO –water are well-documented, hybrid nanofluids, which combine the thermal and stability advantages of two nanoparticles, remain underexplored. Studies that do include hybrid nanofluids often overlook how their performance varies with different insert geometries.
- Many investigations focus on laminar or transitional flow regimes ($\text{Re} < 5000$), whereas practical applications in industry, especially in HVAC, power generation, and automotive cooling, predominantly operate in turbulent conditions ($\text{Re} > 6000$).
- In several studies, the effects of key geometrical parameters such as twist ratio (TR), ring pitch (P/D), and diameter ratio (d/D) are evaluated only partially or not at all in combination. As a result, there is a lack of comprehensive design guidelines for engineers seeking to implement these strategies effectively.
- Some previous works rely solely on experimental methods or simplified analytical models, which may not capture detailed flow structures or allow for extensive parametric variation. Modern computational fluid dynamics (CFD) tools remain underused for multi-factorial analysis involving both inserts and nanofluids.

1.5.2. Novelty of the Present Work

This thesis addresses the aforementioned gaps through a systematic and integrated investigation of heat exchanger performance under the influence of:

- Custom-designed ring inserts featuring perforations, fins, and varying pitch/dimension ratios.
- Twisted tape inserts with varying twist ratios ($TR = 5.5, 10.5, \text{ and } 15.5$).
- Hybrid nanofluids (Al_2O_3 –Cu/water) at different volume concentrations (0.1%, 1%, and 2%).
- Turbulent flow regimes within a Reynolds number range of 4000 to 14,000.
- The novelty of this research lies in its multi-dimensional approach, combining fluid innovation (nanofluids) with geometrical enhancements (compound inserts), evaluated through CFD-based simulations. This integration enables a deeper understanding of:
 - The effect of combining swirling and obstructive flow patterns on the Nusselt number, friction factor, and thermal performance factor (TPF).
 - How hybrid nanofluid properties interact with turbulence-inducing geometries.
 - Optimal configurations that balance heat transfer augmentation and pressure drop minimization, providing actionable insights for thermal system designers.

Moreover, the proposed ring geometries are innovative, incorporating elliptical perforations, sharp teeth, and radial fins—a combination not previously studied in the open literature. These features are intended to exploit multiple flow disturbance mechanisms simultaneously, offering superior enhancement over standard ring types.

1.6. OBJECTIVES OF THE THESIS

In light of the growing demand for compact and energy-efficient thermal systems, this study aims to investigate the combined effects of novel passive insert geometries and hybrid nanofluids on the thermal and hydraulic performance of tubular heat exchangers under turbulent flow conditions. The specific objectives of the research are as follows:

- To design, simulate, and evaluate geometrically advanced ring inserts featuring elliptical perforations, radial slots, and integrated internal fins, and assess how varying diameter ratios (d/D) and pitch ratios (P/D) affect flow disruption, secondary vortex generation, and overall thermal–hydraulic enhancement.
- To analyze the impact of twisted tape inserts with varying twist ratios ($TR = 5.5, 10.5, 15.5$) on heat transfer rate, friction factor, and thermal performance factor (TPF) under turbulent flow regimes.
- To evaluate the thermal–hydraulic behavior of Al_2O_3 –Cu/water hybrid nanofluids at four volume concentrations (0%, 0.1%, 1%, and 2%), prepared with equal proportions (1:1) of Al_2O_3 and Cu nanoparticles, and to analyze their effects in the presence of internal inserts.
- To assess the combined effects of twisted tape and ring inserts used with hybrid nanofluids on critical thermal-hydraulic indicators—namely, the Nusselt number, pressure drop, and thermal performance factor—to identify the optimal configuration for enhanced performance with minimal energy loss.

1.7. THESIS STRUCTURE OVERVIEW

This thesis is organized into six main chapters, each designed to build progressively toward the understanding and evaluation of the proposed heat transfer enhancement system.

Part 1: Introduction

This chapter outlines the background, motivation, and significance of the study. It introduces the research objectives, questions, and scope, along with the underlying rationale for investigating hybrid nanofluids and complex insert geometries.

Part 2: Literature Review

This chapter presents a comprehensive review of previous work related to passive heat transfer enhancement techniques, including twisted tapes, ring inserts, and nanofluids.

It also highlights the limitations in existing studies and identifies the research gaps that justify the need for this investigation.

Part 2: Methodology

Here, the geometrical configuration of the heat exchanger, the properties of the working fluid, and the boundary conditions used in the CFD simulations are explained in detail. The chapter also covers the meshing strategy, numerical approach, and validation process.

Part 4: Results and Discussion for Insert Ring Designs

This chapter presents the thermal and hydraulic performance of several newly proposed internal ring insert designs. It analyzes the effects of different d/D ratios and P/D ratios under turbulent flow conditions using water.

Part 5: Results and Discussion for Twist Ratio and Special Ring Inserts with Hybrid Nanofluids

In this chapter, the combined impact of twist ratio (TR) and specially engineered internal ring inserts is investigated. The results explore how different hybrid configurations affect heat transfer enhancement and pressure drop. The analysis includes the effect of hybrid nanofluid concentration.

Part 6: Conclusions and Recommendations

The final chapter summarizes the key findings, discusses the study's implications, and provides recommendations for future research and potential applications in heat exchanger design.

PART 2

LITERATURE REVIEW

2.1. OVERVIEW OF PASSIVE AND ACTIVE HEAT TRANSFER ENHANCEMENT METHODS

Heat transfer enhancement techniques vary, including active, passive, and combined methods, which are used depending on the nature of the thermal system and performance requirements. Active methods use external energy sources to stimulate heat transfer, such as mechanical vibrations, pulsating fluid pumping, or electric and magnetic fields to alter the heat flow behavior internally. Passive methods, on the other hand, rely on improving the system design without using additional energy. These include using fins to increase surface area, inserting twisted tapes into tubes to create turbulence that enhances convection, or using nanofluids containing fine metal particles to improve thermal conductivity without altering the system's underlying structure. Combined methods are advanced solutions, incorporating techniques from both types to maximize efficiency. For example, combining twisted tapes and nanofluids in a tube with fixed loops improves thermal mixing and conductivity. Studies have shown that combined systems achieve the best thermal performance but require careful design to balance cost, energy efficiency, and complex engineering.

Passive methods enhance heat transfer without requiring external energy input. Instead, they rely on geometrical or surface modifications, insert devices, or additives to improve fluid mixing, disrupt boundary layers, and extend the heat transfer surface area. Passive techniques are widely adopted due to their simplicity, reliability, low operational cost, and ease of integration into existing systems. Several researchers have contributed to the development and evaluation of passive heat transfer enhancement methods. Selvan et al. [78] focused on improving the thermal

performance of bus duct conductors by modifying the spacing of metal fins used as heat sinks, elements used in high-current electrical junction boxes to dissipate Joule-generated heat. The researchers relied on three-dimensional numerical simulations using ANSYS Fluent, supported by laboratory experiments according to IEC 60439, to study five different fin spacing models (1.0 mm, 1.5 mm, 2.0 mm, 3.0 mm, and 4.0 mm). The results showed that reducing the fin spacing increases the conductive resistance within the fin base, reducing the benefit of some fins located far from the heating center. It was observed that a 1 mm spacing gave the best thermal performance, achieving the lowest envelope surface temperature (80.5°C) and the highest fin efficiency of 1.054, with a significant reduction in the overall thermal resistance. In this case, the system recorded a maximum heat flux density of 375 W/m² and the highest average Nusselt number ($Nu = 14.8$). On the other hand, larger spacings (3 mm and 4 mm) showed greater development of natural convection currents due to the appearance of the "chimney" phenomenon between the fins. The Nusselt number at 4 mm reached 16.74, but the number of fins was smaller, which limited the overall performance. The actual results also showed that practical thermal performance was best at 1 m.

Wang et al. [79] presented a detailed experimental analysis of the influence of various geometric factors on the thermal and hydrodynamic performance of pin-fin tube bundles, which are widely used in heat recovery systems such as heat exchangers and boilers in power plants. The researchers tested ten different tube bundle models, all with a fixed outer diameter of 45 mm, while adjusting four key geometric parameters: the transverse tube spacing (S_1), the longitudinal tube spacing (S_2), the fin spacing (Ph), and the fin height (H). The experiments were conducted in a thermal wind tunnel under turbulent flow conditions (Reynolds number between 8,000 and 30,000) using 150°C hot air passing through the water-cooled tubes. The analysis focused on the behavior of the system within a Reynolds numerical range of 8,000 to 30,000, values that represent the turbulent flow that enhances heat transfer capability. The results showed that the ideal distance was found to be $S_2 = 95$ mm, where the highest value of the Performance Evaluation Criterion was achieved. On the other hand, reducing the distance between fins (Ph) from 28 mm to 16 mm resulted in an improvement in heat transfer by up to 15%, but with a huge increase in pressure loss of 85.5%, making

$Ph = 20$ mm best suited for thermal and hydrodynamic balance. Reducing the transverse distance between pipes from 123 mm to 113 mm increased aerobic turbulence, improving the heat transfer rate, but at the expense of high aerodynamic resistance. As for the longitudinal distance, increasing the $S2$ to 95 mm improved thermal performance by 14.8%, while raising it to 100 mm led to a gradual decline in the efficiency of the total area. As for the height of the fin (H), the findings showed that increasing it from 96 to 108 mm increased the surface area exposed to air and thus improved the rate of heat exchange.

Nitturi et al. [32] discussed the passive techniques for enhancing heat transfer, which are reliable and cost-effective and do not require additional power compared to active methods. The discussed techniques include inserts (such as conical strips, twisted tapes, winglets, and baffles), porous materials, extended surfaces (fins), coil/helical/spiral tubes, rough surfaces, and nanofluids (mono and hybrid). These methods improve the thermal performance of energy preservation systems by altering the flow path of the working fluid. Key findings indicated that insert components such as twisted tapes, baffles, conical strips, and winglets improve heat transfer rates when inserted into the flow path. Lower twist ratios, winglet angles, and pitches result in higher heat transfer rates but also slightly increase the friction factor. Porous materials provide a larger surface area for heat transfer and can be used to improve thermal performance in certain applications. Fins extend the surface area for heat exchange, enhancing the overall heat transfer rate without requiring additional power input. Rough surfaces increase turbulence in the fluid flow, leading to higher heat transfer rates at the expense of increased pressure drop. Coil/helical/spiral tube designs promote turbulent flow, which enhances heat transfer compared to straight tubes. Adding nanoparticles to base fluids improves their thermal conductivity. While nanofluids slightly enhance pumping power requirements, they significantly boost heat transfer efficiency.

Ali et al. [80] reviewed the impact of various design factors on improving the performance of heat exchangers using passive heat transfer enhancement techniques, which rely on geometric modifications without the need for an external energy source. These techniques are widely used in industrial applications to improve heat transfer

efficiency, including vortex generators, fins, surface roughness, and destabilized internal structures such as twisted ribbons and spiral tubes. The results indicate that adding wavy or overlapping fins expands the surface area for heat exchange without significantly affecting pressure loss, especially when combined with other elements such as calculated baffles or perforations. Surface roughness has proven effective in stimulating internal turbulence but requires careful balancing to avoid excessive flow resistance. Roughnesses between 0.95 and 2.75 microns were found to significantly improve the heat transfer coefficient, especially in systems operating with turbulent flows. The study also demonstrated that the use of various vortex generators—such as perforated and tilted fins or spiral ribbons—generates turbulent flow and reduces boundary layer thickness, increasing the heat transfer coefficient by more than 40% in some designs, particularly when combining more than one shape (such as a perforated fin with a ring or a multidirectional curved fin). The review also focused on using spiral tubes or twisted ribbons within heat exchangers, which stimulate spiral flow and increase the fluid residence time within the tube. The results showed that introducing an internal wire with a diameter equal to half the tube diameter improved thermal performance by more than 10%, making it a practical and cost-effective solution. A notable aspect of this review is the integration of multiple technologies. Combining fins with twisted ribbons, or baffles with spiral tubes, achieved a greater synergistic effect than using each technology alone.

Kumar et al. [81] highlighted the critical role of heat-transfer enhancement in improving the economic and environmental performance of thermal power plants. It explains that augmentation techniques fall into three categories: passive (no external power), active (using external forces or fields), and compound (combining two or more methods). The focus is on passive compound heat-transfer augmentation (PCHTAT), where configurations such as helical-ribbed tubes with double twisted-tape inserts, dimpled tubes with tape-swirl generators, or non-uniform wire coils are used in tandem. The paper's review of recent passive heat-transfer enhancement methods reveals a variety of cost-effective strategies for improving thermal performance without external energy input. Porous materials further augment transfer rates by creating complex internal flow pathways and maximizing surface contact. Devices such as twisted tapes, conical strips, baffles, and winglets act as inserts to increase

turbulence and boost convective heat transfer within conduits, while extended surfaces or fins expand the available area for heat exchange. Similarly, coiled, helical, or spiral tubes generate swirl flow that significantly outperforms straight-tube designs, and deliberately roughened surfaces, whether corrugated or ribbed—disrupt boundary layers to increase fluid–surface interaction. Finally, suspending nanoparticles in base fluids yields nanofluids with enhanced thermal conductivity, and hybrid nanofluids combining multiple particle types can achieve even greater improvements.

Active techniques involve the use of external power to enhance heat transfer. While they can offer significant improvements, they also require additional energy input and may increase system complexity. Numerous studies have advanced and assessed methods for active heat transfer enhancement. Thapa et al. [82] reviewed various active methods and materials for enhancing heat transfer in heat exchangers, aiming to minimize energy consumption. The applications include automobile radiators, solar air heaters, condensers, refrigeration units, evaporators, and steam turbines. The study highlighted new techniques such as electroactive and magnetic materials, which enhance heat transfer rates when combined with passive methods. The electrohydrodynamic (EHD) method increased heat transfer by 20 times compared to a 3 times increase in the friction factor. Vibration enhanced heat transfer by 97%. Also, magnetic materials, used with passive techniques, increase the Nusselt number by 10%. Overall, the research highlights how these cutting-edge active techniques can boost heat-transfer efficiency while keeping frictional losses and energy use to a minimum.

Agrawal et al. [83] investigated the use of a rotating mechanical stirrer to overcome the low thermal conductivity of D-mannitol in a shell-and-tube latent heat thermal energy storage system. By mechanically agitating the melting PCM at 200 RPM under a constant 185 °C heat-transfer fluid flow, they reduced total melting time by approximately 5 % (≈ 1 200 s), increased charging efficiency from 68.6 % to 72.0 %, and achieved a net energy saving of 52 kJ despite the stirrer’s own power consumption. Their results demonstrate that active stirring can markedly accelerate phase change and improve overall system performance in solar-driven thermal storage applications.

Li et al. [84] addressed improving evaporator performance through active vapor–liquid adjustment (AE) instead of the conventional design without adjustment (CE). The goal is to increase the heat transfer coefficient (α) and reduce the pressure loss (Δp) in the internal boiling flow of the refrigerant tubes. The liquid–vapor separation process is carried out in a passage called the “separator” inside the left distribution header, where a portion of the liquid is drained (Liquid Drainage) and then re-injected into the final passage (Liquid Refilling) to extend the optimal vapor quality (x) range (0.60–0.80) along the evaporator tubes. The experimental evaporator is designed with three paths, with 7, 8, and 9 tubes, respectively, with a tube diameter of 7 mm and a length of 450 mm. The results demonstrate that tuning the vapor and liquid distribution enables a simultaneous improvement in α and a reduction in Δp up to $\eta \approx 40\%$, avoiding exceeding this value to prevent superheating. At $\eta = 40\%$, the TTP decreases by approximately 14.4%, the heat transfer efficiency improves, and the pressure loss decreases by up to 15.3%–22.5% depending on m and n . The largest local improvement in α occurs in the second pass ($\approx +31.7\%$), and the largest reduction in Δp occurs in the first and second passes. The design is simple and scalable by optimizing the number of tubes, joint positions, and fin shapes, and it can be applied to evaporators with various types of refrigerants.

Staats et al. [85] developed active cooling techniques using integrated centrifugal fans to improve heat sinks' thermal performance in high-density electronic devices. The goal is to reduce the thermal resistance per unit pumping power compared to conventional heat sinks. The research relied on a system called PHUMP, which consists of a heat pipe, multiple parallel laminate capacitors, and integrated centrifugal fans operating in an interdigitated mode. The performance of a single integrated fan was experimentally studied using 12 different blade designs. The finding proved that integrated centrifugal fans significantly enhance the thermal performance of air-cooled systems. Fan geometry significantly impacts efficiency, particularly the number of blades, exit angle, and fill ratio. The proposed system outperforms commercial systems and is an effective alternative to liquid or complex cooling. The proposed thermal model can be used in design without expensive CFD. The integrated fans improved heat transfer by a factor of three compared to natural cooling at the same flow rate. The coefficient of performance (COP) exceeded 1, indicating a high heat transfer

capacity compared to the mechanical power consumed. At an 80°C surface temperature, 25°C inlet air temperature, and 9 krpm fan rotation, the thermal resistance dropped to less than 0.08 kW. As the number of blades increased (up to 15 or 17), performance improved, but with diminishing marginal benefit beyond a certain point. A fill ratio (FR) > 0.6 significantly enhanced performance.

Damanafshan et al. [86] explored the performance of Near-Wall Active Vortex Generators (NWAAGs) in enhancing heat transfer in a rectangular channel while minimizing friction losses. Three distinct motion patterns of vortex generators (VGs) were simulated numerically at a constant Reynolds number ($Re = 1000$). The results showed that positioning vortex generators (VGs) in close proximity to the heated surface dramatically enhances convective heat transfer by generating strong, periodic vortices within the thermal boundary layer. These near-wall vortical structures continually disrupt the boundary layer, thinning it and promoting more efficient fluid mixing, which accelerates heat removal from the surface. The three motion patterns exhibit markedly different heat-transfer enhancements and efficiencies. Pattern 1, featuring continuous motion, delivers a moderate improvement, raising the Nusselt number by about 25 % ($Nu/Nu_0 = 1.25$) and achieving a peak efficiency (η) of roughly 1.10. In contrast, Pattern 2's reversible intermittent motion underperforms, providing only a 9 % increase in Nu ($Nu/Nu_0 = 1.09$) and barely improving efficiency ($\eta \approx 1.04$) due to cancellation of vortical structures. Pattern 3, however, stands out: its unilateral intermittent actuation yields an 85 % boost in Nu ($Nu/Nu_0 = 1.85$) and a maximum efficiency of 1.26, making it by far the most effective strategy of the three.

Hybrid approaches integrate both active and passive strategies to achieve optimal heat transfer enhancement. For instance, pairing nanofluids with extended-surface configurations can produce synergistic gains in thermal performance. A broad body of research has since advanced and assessed these combined enhancement techniques.

Sayoud et al. [87] improved the performance of latent thermal energy storage (LHTES) systems using tube-in-shell systems by incorporating multiple passive enhancement techniques, including the use of dual phase change materials (Dual PCMs), modifying the shape and orientation of the inner heat-trapping fluid (HTF) tubes, and adding aluminum oxide (Al_2O_3) nanoparticles. A two-dimensional

numerical simulation was performed on six different geometric configurations to evaluate the system performance in terms of full charge time, stored energy, and improvement ratio compared to the reference case based on only one PCM. The results showed that using a double PCM with a low melting point at the bottom of the storage module reduced the charging speed by 37.06% compared to the conventional case. Adding Al_2O_3 nanoparticles to the front of the module, where melting is slower, significantly improved the performance. At a concentration of 6%, the PCM could be charged in 14.14 minutes, an 83.36% improvement compared to the reference case. Furthermore, replacing the orientation with one for each of the two coils, particularly the elliptical and vertical ones, reduced the charging time by up to 73.59%. However, increasing the differential radiation concentration resulted in a slight decrease in power due to the lower core temperature.

Ke et al. [88] designed and analyzed the performance of an integrated thermal protection system that combines active (fluid cooling) and passive (insulation materials) approaches, with the aim of protecting supersonic aircraft exteriors from high aerodynamic heat, while reducing thickness and coolant consumption. The proposed model is designed to include four main layers: a radiation layer (high-emissivity zirconia ceramic), an upper protection layer (high-thermally conductive metal alloy), a lower protection layer (low-conductivity silica gel), and finally a bearing layer made of aluminum alloy. The protection system was simulated in two scenarios: without a coolant (passive system) to calculate the maximum operating time before the temperature reaches the permissible maximum, and with active cooling to calculate the minimum possible thickness while maintaining thermal integrity. The results showed that when coolant was introduced into the tubes, its flow in opposite directions in adjacent tubes effectively reduced heat buildup and distributed temperatures more evenly, improving cooling performance. The shape and number of tubes also had a significant impact, with triangular tubes outperforming square and circular tubes. Increasing the number of tubes from 5 to 8 increased the heat transfer coefficient from 8612 to 13347 $\text{W/m}^2\cdot\text{K}$, and the internal temperature decreased by 23 K. The passive system alone could provide protection for up to 214 seconds with ideal layer distribution ($l_2 = 12$ mm and $l_3 = 4$ mm), compared to 146 seconds with the initial distribution, while maintaining a total thickness of less than 20 mm. The

minimum possible thickness of the integrated system was determined at $l_2 = 6$ mm and $l_3 = 2$ mm, giving a total thickness of only 12 mm, while maintaining all temperatures within safe limits under a heat flux of 500 kW/m^2 and a fluid velocity of 0.1 m/s .

Bukar et al. [89] evaluated active and passive cooling techniques used to improve the efficiency of photovoltaic (PV) panels by reducing the high temperatures caused by solar radiation. It is based on reliable quantitative studies, focusing on reducing temperature, increasing electrical efficiency, and increasing energy production. Passive cooling techniques include the use of phase change materials (PCMs), metal fins, and evaporative cooling using media such as cotton, heat transfer tubes, radiant coatings, and immersion in liquids. These methods require no external energy. Active cooling, on the other hand, includes systems that rely on external energy to activate cooling, such as forced air or liquid cooling, as well as hybrid systems that combine active cooling with passive technologies. Results have shown that phase change materials such as paraffin wax and calcium chloride can reduce temperatures by more than 23°C , provided they are properly designed with regard to melting point and material distribution. The results also demonstrated that various cooling techniques achieve an average panel temperature reduction of up to 15°C , with reductions as high as 38°C using forced air cooling and 29°C using active liquid cooling. In terms of electrical efficiency, improvements of up to 16% were recorded using liquid immersion techniques, while efficiency increased by 13% using PCMs and 10.6% using metal fins. Fins of various shapes (rectangular, hollow, multi-level) have been found to offer high thermal performance, especially when combined with PCM, but this can increase mass and cost. Heat pipes, on the other hand, have achieved effective temperature reductions of up to 35°C and have been prominent in concentrated phase ventilation (CPV) systems, while evaporative cooling using natural materials such as cotton has reduced temperatures by up to 26°C , especially in arid regions. Radiant coatings such as PDMS and TiO_2 have offered slight temperature reductions ($1\text{--}3.5^\circ\text{C}$) but are cost-effective and easy to integrate. Active systems have demonstrated outstanding performance, especially when combined with radiative cooling or PCM technologies, making them suitable for systems that require high cooling despite their energy consumption.

El-Nagar et al. [90] presented an experimental and practical evaluation of a novel system that combines active and passive cooling to improve the performance of building-integrated photovoltaics (BIPV), with a focus on increasing electricity generation efficiency and utilizing the heat generated during operation. The innovative system is designed to immerse cylindrical PCM capsules within a cooling water channel behind the PV panel, creating a dual-effect system known as BIPV/T-PCM. Five different systems were tested: a reference system (BIPV) without cooling, a BIPV/T system with only water cooling, and three BIPV/T-PCM systems containing different proportions of PCM capsules (20%, 30%, and 40%) within the cooling channel, named BIPV/T20, BIPV/T30, and BIPV/T40. The results showed that the BIPV/T40 combined system performed best, reducing the panel surface temperature by 29.6°C compared to the reference system, a reduction of 37.4%. Furthermore, electrical efficiency increased from 11.95% (for the uncooled system) to 22.1% in the BIPV/T40 system, with an 85.1% improvement in daily electrical energy production. While water cooling alone helped reduce temperature and improve performance, the inclusion of PCMs resulted in better thermal stability during peak solar periods. The material absorbed excess heat during melting, delaying the panel's arrival at critical temperatures and maintaining stable electrical performance throughout the day. Water cooling alone reduced temperature compared to the reference system, but the combination with PCMs yielded even better results due to the material's ability to absorb latent heat during the solid-to-liquid phase transition. The study also showed that increasing the amount of PCMs delayed the panel's arrival at peak temperature and improved temperature stability throughout the day, supporting stable system efficiency throughout its operational life. In terms of economic feasibility, the new system has been shown to be able to recover its cost in just 1.6 years, while reducing carbon dioxide emissions by 32%, which enhances its environmental feasibility in the context of the transition to sustainable energy in smart buildings.

Zhao et al. [91] focused on the comprehensive evaluation of the thermal protection systems for the struts of supersonic breathing aircraft engines, particularly those operating in the RBCC propulsion system. This critical structural component is exposed to extreme heat flow rates due to high-speed air friction and internal combustion, requiring a thermal design that can withstand extreme conditions while

maintaining minimal weight, size, and cost. Three different strut designs were compared: the first is made from GH3625 alloy and uses an active hydrocarbon-fueled cooling system; the second is a composite design combining a TA-10W super-resistant head and a GH3625 structure; and the third is a fully passive design made from a C/C–SiC composite material. The results showed that the actively cooled strut offered the best thermal performance, maintaining the wall surface temperature at ~ 1300 K, which is below the permissible limit for the material. The TA-10 W panel-maintained temperatures of approximately 1500 K, well within its tolerances, and reduced the amount of heat absorbed by 8% compared to the first panel. However, it was 33% heavier and six times more expensive. The passive panel made of C/C–SiC reached temperatures exceeding 2100 K, exceeding the material's capacity, and exhibited significant corrosion and loss of thermal stability, making it unsuitable for long-term applications. The authors recommended using the actively cooled panel in situations where sufficient cooling is available and using the TA-10 W panel when cooling is limited.

2.2. REVIEW OF TWISTED TAPE INSERTS HEAT EXCHANGER APPLICATIONS

Twisted-tape inserts rank among the most widely adopted passive enhancement methods for heat exchangers, especially when higher thermal performance is needed without extensively modifying the core design. A substantial body of research has focused on developing and testing different twisted-tape geometries and arrangements to maximize their heat-transfer benefits. Shelare et al. [92] summarized methods used to enhance heat transfer using twisted tape inserts (TTIs), one of the most effective and popular passive methods in heat exchangers. The authors discussed various types of TTIs, both fixed and self-rotating, such as perforated, winged, double, and triple-twisted, highlighting the influence of geometric characteristics such as twist angle, length-to-diameter ratio, and width on thermal performance and pressure drop. It also highlighted the effectiveness of TTIs in quiescent flow compared to turbulent flow and reviews their applications in cooling and heating systems, including their use in condensers and evaporators. The results showed that the insertion of TTIs significantly improves the heat transfer coefficient, especially in quiescent conditions, despite a

relative increase in pressure loss. Additionally, the efficiency of this technology is greater under laminar flow conditions than under turbulent flow due to the ribbons' ability to improve internal mixing and reduce thermal resistance at the walls. The results also indicated that the performance of twisted ribbons begins to deteriorate as the Reynolds number increases due to increased flow resistance. Furthermore, the study highlighted the potential for future development of twisted ribbons in heat exchangers, especially when combined with other active technologies or nanofluids.

Abu-Khader et al. [93] evaluated the effect of introducing internal twisted strands into tubes on thermal and hydraulic performance in the transition flow region, the region between laminar flow ($Re < 2300$) and turbulent flow ($Re > 10000$). The twisted ribbon concept relies on creating a vortex flow inside the tube, forcing the fluid to rotate and enter a continuous intersection with the hot walls of the tube. This weakens the thermal boundary layer, enhances internal mixing, and thus increases the heat transfer coefficient (h). The authors studied the effects of multiple design factors, such as the twist ratio (y), tube diameter, and flow velocity. The findings showed that the improvement in thermal performance using ribbons was most pronounced for tubes with small diameters (≤ 0.04 m). In these cases, ribbons with $y = 2$ doubled the Nusselt number compared to a smooth tube, while improving heat distribution along the tube wall. The smaller the diameter, the higher the surface area-to-volume ratio. Tapes with a small twist ratio ($y = 2$) generate stronger vortices and thus higher thermal performance, but at the same time increase pressure loss. Tapes with $y = 5$ or greater provide lower thermal performance but are more hydraulically balanced, making them preferred in systems requiring low pumping energy consumption. The author noted that the transition region begins at $Re \approx 2800$ in some cases when using twisted tapes, which is lower than the known value ($Re \approx 4000$). Therefore, using these tapes not only improves thermal performance but also helps accelerate the transition from turbulent flow to laminar flow. It was also observed that using $y = 2$ resulted in sharp increases in ΔP , which may be up to 2–3 times the loss in the smooth tube. Even in turbulent flow, where self-mixing is high, twisted ribbons have shown additional performance. Nu increases of about 15–20% have been reported in some cases, but f increases of 40–60% have been reported.

Shijie et al. [94] investigated circular tube heat exchangers by proposing a geometrically unique L-shaped twisted tape (L-TT) design aimed at further improving thermohydraulic performance. The proposed L-shaped twisted tape includes a vertical and horizontal segment that forms an “L” shape, which is positioned near the tube wall. Parameters such as twist pitch (P), tape width (W), inner tube diameter (D), and the number of tapes (N) were systematically varied to investigate their effect on performance. The findings presented that when the tape widths (W) of 0.5 mm, 1 mm, and 2 mm were analyzed. The highest Nu was recorded for W = 2 mm, with improvements up to 231%, and friction increased by up to 795%. However, sensitivity analysis later showed that W had the least influence among all parameters. Lower twist pitch (P = 60 mm) generated the highest heat transfer enhancement due to more frequent swirl generation. Nusselt number (Nu) increased by 232–241% over the smooth tube, but this also came with a friction factor (f) increase of 894–897%. Increasing N from 4 to 6 enhanced both Nu and f due to increased surface interaction and turbulence. For N = 6, the Nu improved by up to 238%, and f rose by up to 824%. As D increased from 6 mm to 10 mm, the heat transfer performance generally declined due to a larger central flow region that avoided interaction with the tape. The highest Nu (up to 293%) was achieved at D = 6 mm, and the lowest f (469%) at D = 10 mm, demonstrating that smaller diameters result in stronger interaction with the swirl generators. All test cases showed $PTC > 1$, indicating better performance than smooth tubes. The best PEC was observed for D = 6 mm ($PEC \approx 1.78$), and values decreased with increasing diameter. The final optimal design parameters were twist pitch (P): 65.59 mm, tape width (W): 0.95 mm, and inner diameter (D): 8.80 mm. Under these conditions, the system achieved a Nusselt number of 66.62, a friction factor of 0.414, and $\eta = 160.78$.

Barzegar et al. [95] improved the thermal performance of a double-tube heat exchanger (DPHE) by incorporating a new passive element, rectangular cross-sectioned twisted bands (RCTs), into the inner tube to evaluate how geometric modifications to these bands, specifically the rectangular cross-sections, affect the exchanger's performance metrics. Using ANSYS Fluent, the authors constructed a detailed three-dimensional numerical model of a counter-flow DPHE. The working fluid is water, with hot water flowing through the inner tube and cold water in the annulus. Various configurations

of rectangular cut twisted tapes were studied under different Re ranging from 6000 to 18,000. The TR of the recordings was kept constant at 3.0, while the cut DR and cut WR were varied across three values each: 0.15, 0.24, and 0.33. The study's findings reveal that including RCTs significantly alters the temperature and velocity fields within the heat exchanger. Temperature contours showed that RCTs facilitated the movement of lower-temperature fluid from near the tube wall towards the center, creating symmetrical S-shaped thermal fields not observed with traditional twisted tapes. This enhanced flow behavior translated into a marked improvement in the Nu, indicating higher heat transfer. The simulations demonstrated that DR = 0.33 and WR = 0.15 for the most effective configuration, the Nu increased by up to 192% over the plain tube and by 118% compared to the typical TT. The maximum friction factor observed was approximately 8.84 times higher than that of the plain tube. Across all cases, RCTs delivered higher TPF values than traditional twisted tapes. The peak performance was again observed in the case of DR = 0.33 and WR = 0.15, yielding a TPF of 1.46 at a Re of 6000.

Samutpraphut et al. [96] presented an innovative experimental analysis of the effect of using Sawtooth Twisted Tapes (S-TTs) inside heat exchanger tubes under turbulent flow conditions using air as the working fluid. Twisted strips were designed with different tooth angles ($\alpha = 20^\circ, 30^\circ, 40^\circ, 50^\circ, 60^\circ, \text{ and } 70^\circ$) and a constant twist ratio ($y/w = 3.0$) and were tested over a Reynolds number range of 6,000 to 20,000 under constant heat flux. The results showed that increasing the tooth angle (α) led to a significant enhancement in heat transfer, with the Nusselt number improvement reaching a maximum of 186.2% compared to the smooth tube and 113.8% compared to the conventional twisted strip. Although the friction coefficient also increased significantly, especially at larger angles, the aero-thermal performance index remained above 1 in most cases, demonstrating the feasibility of using this technology at low Reynolds numbers ($Re \leq 8,000$). Despite the increased thermal performance, the study revealed a significant increase in the friction coefficient (f), exceeding its value for $\alpha = 70^\circ$ by 381.2% compared to the smooth tube, reflecting an increase in pressure loss. However, all cases of serrated tape were found to offer better Aerothermal Performance Index (API) than conventional tape, with the highest API value reaching 1.33 at $\alpha = 70^\circ$ and at the lowest Reynolds number ($Re = 6000$). This improvement is

attributed to the combined effect of vortex flow and intermittent flow turbulence resulting from the serrated tape design.

Durán-Plaza et al. [97] focused on the energy upgrading of tubular heat exchangers by introducing tube inserts without changing the exchanger design or increasing its surface area. A total of 25 different tube insert geometries were analyzed based on experimental correlations from previous studies. For each insert, heat transfer and friction were modeled using equations of the form: $Nu = a \cdot Re^b$ and $f = c \cdot Re^d$. The following parameters were calculated: thermal Length (LT), which is the required length to achieve the target heat load, and hydraulic Length (LH), which means the maximum allowable length considering pressure drop limits. The insert is considered feasible only if $LT < LH$. Entropy generation and total irreversibility (I) were also computed to assess the energy degradation due to heat transfer and friction losses. Five realistic retrofit scenarios were tested, including cases with single-phase and two-phase flow conditions. The results showed that among all tested inserts, Promoter 1 (Perforated Twisted Tape) and Promoter 2 (V-cut Twisted Tape) provided the best balance between thermal enhancement and pressure drop. In one case, Promoter 1 delivered a heat duty of 5250 kW compared to 5040 kW for the base case, with only 2.14 kPa pressure drop. Only 9 out of 25 promoters satisfied the thermal and hydraulic constraints across all tested cases. Some inserts had high Nusselt numbers but failed due to excessive pressure drop or poor entropy performance. Promoter 1 showed the lowest irreversibility at 48.9 kW, compared to 55.7 kW for Promoter 7 and over 70 kW for Promoter 18. The study results showed that improving heat exchangers does not require structural changes or the use of additional surfaces, but can be achieved with high efficiency by selecting the appropriate internal catalyst. The entropy analysis also indicated that some catalysts, despite achieving high thermal performance, may be unsuitable if they cause significant energy loss due to friction or exceed pressure limits.

2.3. REVIEW OF INSERT RINGS IN HEAT EXCHANGER APPLICATIONS

Ring inserts represent a widely utilized passive enhancement strategy in heat exchanger design, offering improved thermal performance with minimal alterations to

the primary configuration. Extensive research has explored various ring geometries, placements, and material choices to optimize their effectiveness in boosting convective heat transfer. Banihashemi et al. [98] investigated the effects of circular ring obstacles (stationary and rotating) on thermal performance and pressure drop in a turbulent pipe flow. This study showed that using 90° rotatable inlet rings is an effective option for improving the performance of heat exchangers, providing an ideal balance between heat transfer and pressure drop. The authors analyzed different diameter ratios ($DR = 0.6, 0.7, \text{ and } 0.8$) and angular configurations (complete ring vs. 90° segment) to determine the optimal design that balances improved thermal performance and manageable flow resistance. Laboratory experiments were conducted using an inner-outer tube heat exchanger, with air flowing in the inner tube and water in the outer shell. Stationary and movable inner metal rings were used at various angles (full ring – 90° section) and rotation speeds ranging from 10 to 3000 rpm. Inner tube diameter: 0.073 m, length: 1.6 m. This design surpassed many other optimization techniques, such as twisted ribbons or spiral fins, in terms of thermal performance. While most previous designs achieved η between 0.9 and 1.8, this study recorded up to 2.23. Metal rings are also simpler to manufacture and less expensive than complex components such as twisted ribbons. Numerical simulation results showed the formation of vortices and a re-adhesion zone after each loop. The higher rotation speed resulted in better thermal distribution, but at the expense of increased energy loss. The turbulent energy was concentrated near the wall, which contributed to thinning the static heat layer and increasing conductivity. Inlet rings, particularly those with a 90° angle and rotatable inserts, showed a significant improvement in Nu . At varying Re numbers between 5,000 and 25,000, Nu increased by 3.15 to 4.2 times compared to a smooth tube. The best results were achieved with a diameter-to-wall ratio of $DR = 0.6$ and the use of a segmented ring (90°). Fixed full rings improved thermal performance but caused greater flow resistance. The friction factor (f) increased significantly with the introduction of rings, especially as the distance between the rings decreased or the rotation speed increased. The coefficient of friction ratio (f/f_s) to the smooth tube ranged from 5 to 75. However, the 90°-cut rotary rings demonstrated better performance in terms of the balance between heat transfer and pressure drop than the full rings. η reached its highest value of 2.23 with the 90° rotating ring, compared to

1.32 for the fixed rings. The best performance was achieved at low Reynolds numbers (e.g., 6000) and moderate rotation speeds (300–600 rpm).

Anvari et al. [99] presents a comprehensive and accurate analysis of the effect of inserting special conical rings inside a horizontal circular tube to improve heat transfer, using both numerical modeling (CFD) and laboratory experiments. The focus was on water as the working fluid under steady-state heat flow conditions. Two geometric arrangements of these rings were tested: convergent (CR), where the diameter gradually decreases in the direction of flow, and divergent (DR), where the diameter gradually increases. The tube used was copper, 1100 mm long, with an inner diameter of 13.8 mm and an outer diameter of 15.8 mm. The system was equipped with 35 conical rings made of aluminum, 13.8 mm long, with end diameters of 13.6 mm and 6.8 mm. This study provided accurate approximate equations for calculating Nu and f for each type of ring. The CFD numerical analysis results also showed good agreement with experimental data in terms of Nusselt number but overestimated the pressure loss by 10–15%. The results showed that the use of conical rings resulted in a significant increase in the Nu compared to the smooth tube: Nu_0 increased from 4.8 (in the smooth tube) to 10.61 in the CR case and 17.48 in the DR case, indicating that the spaced arrangement is more thermally efficient. However, this increase in heat transfer came at the expense of a significant increase in the friction coefficient (f); it increased from $f = 93.202 \cdot Re^{-0.62}$ for the smooth tube, to stable values of $f = 12.56$ (CR) and $f = 16.141$ (DR).

Promvonge [100] focused on analyzing the effect of inserting conical ring inserts inside a circular tube heated at a constant heat flow, using air as the working fluid, to improve thermal performance under turbulent flow conditions. Three ring-to-tube diameter ratios ($d/D = 0.5, 0.6, \text{ and } 0.7$) were tested, each with three different geometric arrangements: Converging Rings (CR) which its diameter decreasing in the direction of flow, Diverging Rings (DR) which its diameter increasing in the direction of flow, and Combination of CR and DR (CDR) which alternating in direction (one CR, followed by one DR, and so on). A 1250 mm long tube with an inner diameter of 48 mm was used, and a fixed number of rings was installed. The Re varied between 6000 and 26000, and the heat flux was constant along the outer wall of the tube. The

results revealed that the introduction of conical rings leads to a significant enhancement in the heat transfer coefficient compared to the smooth tube. The DR was found to be thermally more efficient because it generated stronger vortices due to the expansion of the rings, which enhanced interlayer mixing. The Nu was measured for all cases and found to be increased by 197% in the CR case, reaching its highest value of 333% in the DR case, while the increase was 237% in the CDR case, at the lowest diameter ratio ($d/D = 0.5$) and the lowest Reynolds number. A significant increase in the coefficient of friction (f) was also recorded due to the reduced flow cross-section and the formation of strong vortices behind the rings. The increase in f was between 5 and 20 times compared to the smooth tube, and it was found that DR caused the highest-pressure loss. The highest efficiency was recorded at $d/D = 0.5$, and was as follows: DR= 1.80, CDR= 1.41, and CR= 1.15. This efficiency gradually decreases with the increasing Re, due to the increasing negative effect of pressure loss.

Mohamed et al. [8] improved the heat transfer efficiency in tubular heat exchangers by combining three effective strategies: perforated ring inserts, tube rotation, and using single and hybrid nanofluids (Cu, Al_2O_3 , and Cu- Al_2O_3 alloys). In this study, a double-tube heat exchanger is designed, consisting of a European-style adjustable copper inner tube with an outer diameter of 25 mm and an inner diameter of 15 mm, and a 35 mm outer tube. It contains 11 circular rings connected by an inner conductor, each with eight radially arranged holes, with varying void ratios. Pitch ratio between rings is 2.2, inner conductor rotation speed from 0 to 500 rpm, and Re varying from 1000 to 5000. Three types of liquids were used: pure water (as reference), copper-water nanofluid, aluminum-water nanofluid, and hybrid copper-aluminum-water nanofluid with a constant volume concentration of 3% and a nanoparticle diameter of 50 nm. Temperature distribution plots revealed a significant improvement in the thermal distribution when the rings were combined with rotation, with the transformation of thermal stagnation zones into active exchange zones due to local vortices. Numerical flow analysis results also showed that the inclusion of perforated rings generates strong turbulence at the perforation edges, weakening the thermal boundary layer, especially when they coincide with tube rotation. The results showed that the inclusion of perforated rings alone increased the heat transfer coefficient from $7500 \text{ W/m}^2\cdot\text{K}$ to $9500 \text{ W/m}^2\cdot\text{K}$ at a Reynolds number of 4946 and a rotation speed of 500 rpm. When

the Cu–H₂O nanofluid was added at a concentration of 3%, the heat transfer coefficient increased to over 12,100 W/m²·K. Meanwhile, the use of the Cu–Al₂O₃ hybrid nanofluid achieved the highest thermal performance recorded, with a 51% improvement compared to the conventional case without additives. An increase in pressure loss of up to 235 Pa, an additional pumping energy requirement of 0.1 kJ/kg and a maximum value of TPF is 1.67 were recorded when using the rotation with the Cu–Al₂O₃ nanofluid and the ideal ring arrangement.

Hassan et al. [54] focused on understanding the behavior of heat transfer and turbulent flow in circular tubes with internal inserts in the form of conical rings, some of which are solid and some of which are perforated. Four types of inserts were studied within the tube: solid conical close-round (SCR), perforated conical close-round (PCR), solid conical close-round (SDR), and perforated conical close-round (PDR). The tube used was 1440 mm long, with an internal diameter of 64 mm, and the rings were 2 mm thick and 60 mm long, with an end diameter ratio of 0.5. Three perforation shapes were also tested: circular, square, and triangular, with 6, 8, and 10 perforations per ring. The simulation revealed that the solid rings caused local velocity increases around the edges, creating secondary vortices that help weaken the thermal boundary layer. Perforated rings showed reduced resistance due to reduced flow blockage, as the perforations created side currents that weakened the thermal boundary layer. Sparsely spaced rings (SDRs) caused flow acceleration at the walls and a decrease in center velocity, resulting in localized flow backflow zones that increased internal mixing. The results showed that the use of SDR resulted in the highest Nusselt number values, reaching over 360 at a Reynolds number of 26,000, more than 20 times higher than that of the smooth tube at Re = 6,000, where the Nu did not exceed 17.6. SCR also achieved a good improvement, but to a lesser extent. On the other hand, introducing holes within the rings led to a gradual decrease in Nusselt number values, with the decrease being more pronounced when the number of holes per ring increased from 6 to 10. The highest Nu (Nu = 360.2) was recorded at a Re of 26,000 for the SDR configuration, whereas the lowest value (Nu = 17.66) was observed at Re = 6,000 in the smooth tube. The spaced solid rings recording the highest value, exceeding 5.0 at the lowest Re, compared to 0.037 for the smooth tube. Perforated rings, on the other hand, gradually decreased the coefficient of friction as the number of holes increased. The holes

reduced blockage and improved flow, thus reducing pressure loss. The highest thermo-hydraulic efficiency factor (TEF) of 1.1 was found when using SDR at low Re(6000). In other cases, PDR, especially those with circular holes and 6 perforations, provided the best balance between thermal performance and pressure drop, with a TEF of ≈ 1.06 .

2.4. REVIEW OF NANOFLUID DEVELOPMENT TO ENHANCE HEAT TRANSFER

Nanofluids have emerged as a central area of study in heat-transfer enhancement, since suspending precisely engineered nanoparticles within conventional fluids significantly boosts thermal conductivity and convective heat transfer without greatly complicating system design. Consequently, extensive research efforts have been devoted to formulating, characterizing, and optimizing various nanofluid mixtures to achieve maximum thermal performance. Aksoy et al. [101] presented a new method based on the use of sprays of TiO₂-based nanofluids and water to enhance thermal performance by depositing nanoparticles on hot surfaces. The study focused on evaluating the effect of nanoparticle concentration, size, and the number of deposition layers on thermal performance across the boiling range, from 190°C to 100°C. Experiments were conducted on an aluminum block heated to 190°C and then exposed to nano-aquatic sprays at different concentrations (0.05%, 0.1%, and 0.2%). Experiments were conducted on clean surfaces and surfaces covered with deposits resulting from previous sprays. The study used two types of TiO₂: Rutile (larger particles, 30–50 nm), and anatase (smaller particles, 5–30 nm). Thermal performance was evaluated over a cooling rate between 190°C and 118°C. The results showed that increasing the concentration of nanoparticles in the fluid led to a gradual and significant improvement in thermal performance. At a concentration of 0.2%, the highest cooling rate was achieved, with a 32.9% increase compared to using pure water. Surprisingly, however, repeated spraying without surface cleaning resulted in an additional performance increase, indicating that continuous dynamic deposition regenerates the active layer over time. The study indicated that the smaller particle size (anatase) did not provide significant improvement compared to the larger size (rutile), suggesting that concentration and number of sedimentation layers are more important than size alone.

The formation of a nano-layer with an irregular texture, containing cracks and bulges, leading to an increase in local boiling points (hotspots) and an optimal heat distribution was observed. When using plain water on a TiO₂-coated surface, no similar improvement was achieved, and the improvement was significantly less than when using active TiO₂ spray. This demonstrates that the improvement is not solely related to the pre-existing nano-coating, but rather to the continued deposition of particles during operation.

Praveenkumara et al. [102] evaluate the performance of a counterflow tube-in-tube heat exchanger (TTCFHE) using four low-concentration nanofluids: CuO–H₂O, MgO–H₂O, GO–H₂O, and RGO–H₂O, at two low volume concentrations (0.01% and 0.05%) to improve the efficiency of the heat exchangers without significantly increasing the hydraulic loss. The nanofluids were prepared using nanoparticles with an average size of 50 nm, with 0.25% SDS and distilled water added as a dispersant. The mixture underwent mechanical mixing and then ultrasonic vibration to ensure stability. Zeta potential tests showed that the CuO nanofluid had the best stability (37 days at 0.05%), followed by RGO (30 days), while the stability of MgO and GO was intermediate (22–29 days). It was found that adding nanoparticles improved thermal conductivity compared to pure water, increasing from 0.6513 W/m·K (for water) 0.7698 W/m·K for MgO at 0.05%, 0.7742 W/m·K for GO and RGO at 0.05%, and 0.7677 W/m·K for CuO at 0.05%. The density and viscosity also increased slightly, affecting the coefficient of friction. The results showed that the use of nanofluids resulted in a Nu by 4.99% and 7.69% for RGO at 0.01% and 0.05% concentrations, 5.42% and 9.52% for CuO, 13.25% and 19.7% for MgO at 0.01% and 0.05% concentrations, and 8.12% and 11.55% for GO. The highest Nu value was achieved with MgO at a concentration of 0.05%, especially at Re = 18000, where the improvement reached 25% compared to pure water. The friction coefficient also increased by 1.1% and 2.7% for MgO and 4.65% and 8.49% for CuO, while it decreased by 1.7% and 2.5% for GO and 1.9% and 2.9% for RGO. The decrease in GO and RGO is attributed to the lubricant-like nature of carbon, while the increased density and viscosity of CuO and MgO led to increased pressure loss. The thermal enhancement factor (TEF) showed the highest value for MgO, reaching 1.24 at Re = 18000, and the overall average TEF was 1.13 and 1.19 for MgO at 0.01% and 0.05%,

while it ranged from 1.04 to 1.12 for the other fluids. This indicates that MgO is the most efficient choice in terms of improving heat transfer versus hydraulic loss. The study concluded that the 0.05% aqueous MgO nanofluid is the best in terms of stability, thermal improvement (Nu increase), and balance between efficiency and pumping energy consumption.

Abdelmagied [103] explore the heat transfer and flow behavior in annular curved tubes using an Al_2O_3 /Water nanofluid, within a laminar flow system as a passive means to enhance heat transfer and reduce the size of the heat exchangers. The study aims to determine the effects of Cross-sectional shape (Circular, Oval, Rectangular, Square, Triangular), Tube shape (Helical, Spiral, Conical), Cross-sectional area. Three levels of nanoparticle concentration were: 1%, 3%, and 5%. The study focused on Reynolds numbers ranging from 850 to 6400, within the linear flow range, with $\text{Re}_{\text{crit}} = 7570$ as the transition limit. All this under a constant internal wall temperature (50°C) and internal flow conditions with an initial temperature of 20°C . Numerical results revealed that the introduction of nanoparticles and increased curvature lead to generation of a strong secondary flow manifested in the form of radial vortices, improved temperature distribution across the tube surface by up to +18% uniformity, and reduction in the thickness of the thermal boundary layer. Nanoparticle movement via Brownian motion also enhances local turbulence, which positively impacts the Nu value. Three cross-sectional areas were evaluated (0.000128 m^2 , 0.000256 m^2 , and 0.000385 m^2), with the following results: Nu increased by up to +107% when using the larger area compared to the smaller one, at $\text{Re} = 4000$, while f increased by +90% for the larger area, due to the increased pressure differential and hydraulic diameter. When comparing the three tube configurations (helical, spiral, and conical), it was found that the helical tube offers the best thermal performance, due to the secondary flow distribution caused by centrifugal force. At $\text{Re} = 4000$ Nu = 38.45 for helica, Nu = 36.6 for Spiral, and Nu = 34.84 for conical configuration. The improvement in Nu for the helical tube compared to the conical tube was +10.4%. The difference in the coefficient of friction was only 5%, making the increase in thermal performance more significant from an engineering perspective. The impact of Al_2O_3 nanoparticle concentration on thermal and hydraulic performance was clearly observed. At a 1% volume concentration, the Nu increased by 6.5%, while the pressure drop rose by

12.5%, resulting in a performance index of 1.05. When the concentration increased to 3%, the heat transfer enhancement became more significant, with a 20.4% increase in the Nu and a 62.5% rise in pressure drop, yielding a performance index of 1.07. At the highest concentration tested, of 5%, the Nu improved by 31.7%, accompanied by a 137.5% increase in pressure drop, leading to a performance index of 1.10.

Dagdevir et al. [104] improve the thermal and hydraulic performance of a tubular heat exchanger using the rod inserted twisted tape (RITT) and the rod inserted straight tape (RIST) structures, incorporating CuO/Water nanofluid at different concentrations (0%, 1%, 2%). The study was conducted using turbulent flow ($Re = 5000\text{--}29000$) and under a constant heat flux. A closed-loop experimental system was designed, consisting of a 17 mm inner diameter steel tube measuring 1,000 mm in length. Four main configurations were tested: plain tube (PT), typical twisted tape (TTT), rod inserted straight tape (RIST), and rod inserted twisted tape (RITT). Each insert was tested at three different geometric ratios $L/p = 10, 20$, and 40 , where L is the effective length of the rod and p is the twist pitch. Experiments were conducted over a wide range of Reynolds numbers between 5000 and 29000 to cover the entire turbulent flow region. Uniform heat was generated on the outer wall of the tube using a 1.5 kW electric heater. The nanofluid was prepared using CuO particles with diameters ranging from 15–45 nm. A two-stage mixing method was used, followed by ultrasonication to ensure homogeneous dispersion. The fluid achieved high stability, showing that viscosity increased by 8–12% at 2% concentration, Density increased slightly, and thermal conductivity increased by 22–28% compared to pure water. The results showed that using RIST and RITT inserts significantly improved heat transfer. In the RITT case at $L/p = 40$ and with a 2% CuO/water fluid concentration, the Nu reached to 353.6 at $Re = 6572$. In comparison, the Nu was only 174.2 for the smooth tube at the same Re, equivalent to a +103% improvement. The use of the CuO/water nanofluid also demonstrated a significant additional effect, with Nu increasing by 7.4% to 14.8% when moving from pure water to the 2% nanofluid. At $L/p = 10$, the RITT showed the highest recorded friction coefficient of $f = 0.19514$, more than 5.4 times the f value of the smooth tube at the same Re. The use of nanofluid also added additional resistance: at 1% CuO concentration, the f coefficient increased by 2.15%, while at 2% concentration, the increase reached 5.31%. This increase is attributed to the higher

dynamic viscosity and increased density of the nanofluid. The highest PTC value was achieved when using RITT, $L/p = 40$, and CuO 2%, with a $PEC = 1.377$. This demonstrates that the high thermal performance reasonably outweighs the impact of hydraulic loss. The results also presented that the thermal entropy generation rate reduced by 70.1% in the RITT case with 2% CuO compared to the smooth tube. Mechanical entropy generation rate increased due to turbulence and pressure loss, and was highest at $L/p = 10$. The lowest of Exergy destruction rate value was 0.0337 W in the best case, and the second-law efficiency reached its maximum at 0.439.

Liu et al. [105] evaluate and improve the performance of ground-mounted heat exchangers (GHEs) using a highly conductive CuO/Water nanofluid within a spiral tube buried in sand. Three numerical methods for simulating heat transfer were compared: the Single-Phase Model (SPM), Thermal Dispersion Model (TDM), and Discrete Phase Model (DPM) to evaluate the effect of nanoparticles on improving thermal performance under realistic conditions that mimic natural soil (wet sandy soil). A three-dimensional numerical model was developed for a 100-meter-long, 32-mm outer diameter polyethylene spiral pipe, distributed over vertical coils embedded in a 3-meter-deep soil layer. The study relied on a nano-aqueous solution composed of CuO particles with a diameter of 25 nm, a density of 6500 kg/m^3 , and a thermal conductivity coefficient of $33.6 \text{ W/m}\cdot\text{K}$. Three volume concentrations (0.5%, 1.0%, and 2.0%) were studied. The results showed the following: the effective thermal conductivity of the nanofluid increased by 21–38% compared to pure water, depending on the concentration, the bulk density also increased by 4.8% compared to water, resulting in a significant effect on flow resistance and pressure loss, and the dynamic viscosity increased by up to 16.5% at a 2% concentration. Simulations showed that the use of nanofluids resulted in an improvement in the internal heat transfer coefficient (h_i) of 11.7% to 19.3% compared to pure water, depending on the particle concentration. When using the TDM model at a concentration of 1%, the heat exchange rate per unit length was found to be 18.9% higher than the conventional reference. However, the DPM model took over 627 computational hours to run for a single simulation, while the SPM model took less than 0.2 hours. The TDM model took 9.5 hours and achieved a relative error of 3.94%, making it the most engineeringly feasible choice. With increasing CuO concentration, the pressure drop inside the tube

increased significantly, as ΔP was found to increase by 8.3% at 1% concentration, and 18.5% at 2% concentration, due to the increase in the overall viscosity of the nanofluid, in addition to the increase in density.

Abolarin et al. [106] developed a new U-shaped twisted strip (PUCTT) and introduced an additional geometric modification using internal rings (PUCTTR) to achieve early transient flow stimulation and reduce pressure loss without compromising thermal efficiency. The strips were made of 1 mm thick and 18 mm wide copper, with semicircular side sections of two different depths representing two depth-to-width ratios (DWR) of 0.105 and 0.216. These strips were twisted to form a constant twist ratio (YWR) of 5 (where $YWR = H/W$). 15.9 mm diameter metal rings were then added to the strips to form the PUCTTR composite type, with three ring-to-bandwidth ratios (RWR) of 1.25, 2.5, and 5 mm. A large number of rings were distributed along the strip (234, 117, and 58 rings, respectively), enhancing the thermal boundary layer shearing and increasing the flow turbulence inside the tube. The Re's ranged from 315 to 11,404, covering the three flow ranges: laminar, transitional, and turbulent. The study demonstrated that the addition of U-cuts to the tape accelerated the onset of transitional flow compared to a fully twisted tape (CTT), where the cut broke up the laminar flow structure at lower speeds. In addition, the inner rings created localized flow disturbances, increasing fluid mixing and enhancing heat transfer at low Re values. The results showed that the use of PUCTT tapes resulted in a faster onset of the transition phase compared to conventional tapes (CTT). With the addition of rings (PUCTTR), the transition started at lower Reynolds numbers, with the onset of the transition flow recorded at $Re = 576$ for the PUCTTR21 case versus $Re = 796$ for the PUCTT case without rings. At points with low Rr (smaller ring spacing), the Nu coefficient increased by up to 53% compared to the tape case without rings at $Re = 1000$. In the turbulent case, the increase reached 103% at $Re = 2000$. Increasing the lateral cutting ratio (Rd) from 0.105 to 0.216 resulted in Reducing the onset of the transition flow (from $Re = 689$ to 640), reducing the width of the transition zone (ΔRe) (from 592 to 585), and increasing the Colburn coefficient by 27% at $Re = 1000$. Reducing the inter-ring spacing ($Rr = 1.25$) resulted in reducing the onset of the transition to $Re = 576$, enhancing heat transfer by up to 53%, and reducing the total width of the transition zone. Significant increases in the coefficient of friction (f) were

observed, especially when using rings. In cases involving greater depth of cut and higher ring density, the f coefficient increased by 40–60% compared to a smooth tube, but this was compensated by double the thermal performance.

2.5. REVIEW OF COMBINED ENHANCEMENT TECHNIQUES

Hybrid enhancement techniques combine two or more passive methods, such as twisted tapes and ring inserts, to exploit their joint impact on turbulence and fluid mixing. These synergistic pairings can elevate local heat-transfer rates while limiting friction increases, delivering a well-balanced improvement in thermal performance. Extensive studies have evaluated a variety of such hybrid configurations to optimize convective heat transfer. Singh et al. [107] analyzed the thermal and hydraulic performance of a circular tube heat exchanger equipped with a combination of multiple twisted tapes (single, double, triple, and quadruple twisted tapes) with solid rings as internal inserts to enhance heat transfer in a turbulent flow regime using air as the working fluid and under constant wall convection. The effect of step ratios ($PR = 1, 2$) and twist ratios ($TR = 2, 3$, and 4) was tested with a wide range of Reynolds numbers (6300–22500). Temperature distribution analyses on the inner surface of the tube showed that the ribbons and rings generate strong axial vortices and secondary radial vortices, which weaken the thermal boundary layer and increase internal fluid mixing. Increasing the pitch ratio also has a negative effect on heat gain, as wider spacing reduces turbulence frequency and weakens the mixing effect. Reducing the twist ratio (TR) results in stronger vortices and greater pressure losses but provides the highest thermal efficiency, especially at low Reynolds numbers. The results showed that introducing just one twisted ribbon with a solid ring can significantly improve heat transfer. However, increasing the number of ribbons to two, three, or even four resulted in a significant increase in the Nusselt number (Nu), reaching a maximum of 293.7, compared to only 98 for the smooth tube. The coefficient of friction (f) ranged between 0.93 and 0.99 for all composite cases, with significant increases in pressure loss recorded when using more twisted ribbons. However, the thermal performance factor (TPF), which represents the ratio of heat transfer improvement to pressure loss, remained above 1 for all tested configurations, reaching a maximum of 1.61 when four bars were used at $TR = 2$ and $PR = 1$.

Kumar et al. [108] investigated experimentally the thermal performance improvement of a circular tubular heat exchanger using combined passive enhancement techniques, combining Perforated Conical Rings (PCR) and Twisted Tapes (TT), with CuO–H₂O nanofluid as the working fluid at a volume concentration ranging from 0.25% to 1.0%. The researchers focused on determining the combined effects of these three techniques on the Nu, f, and TPF under constant heat flux conditions (10 kW/m²). The effect of a range of geometric parameters was investigated, such as the ring spacing to edge diameter ratio (P/D) from 1.41 to 2.51, the ring inlet diameter to ring base diameter ratio (Di/Do) from 1.33 to 2.33, and the twist ratio (T/W) of the twisted ribbons from 3.33 to 4.38. The ring height was fixed at D/W = 2.33, and the nanoparticle diameter was fixed at 30 nm. The results showed that the Nu increased significantly with increasing nanoparticle concentration and Re. The maximum Nu value was 156.88 at Re = 18000, ϕ_{np} = 1.0%, Di/Do = 1.83, T/W = 3.50, and P/D = 1.76, compared to Nu = 61.92 in the conventional case (pure water and a smooth tube). This represents a 153.3% improvement in heat transfer. The friction coefficient (f) increased significantly when perforated rings and twisted bands were combined, especially at low T/W ratios (e.g., 3.33), resulting in increased turbulence. The highest value of the friction coefficient (f) was 0.0641, while the lowest value was 0.0318 at a T/W ratio = 4.38. This increase is attributed to the complex interactions between the twisted band, which generates long-period vortices along the tube, and the perforated ring, which generates intermittent axial flow. The composite TPF was evaluated to determine how effective the heat enhancement was compared to the hydraulic losses. The highest TPF value was found to be 1.45 when using ϕ_{np} = 1.0%, T/W = 3.5, Di/Do = 1.83, and P/D = 1.76. When using lower droplet ratios or nanoparticle concentrations below 0.5%, TPF was less than 1.2, demonstrating that the heat transfer improvement does not always justify the pressure loss.

Marzouk et al. [109] evaluated a double tube heat exchanger (DTHX), by integrating a perforated twisted tape insert (PTTI) and an aluminum oxide nanofluid Al₂O₃, under turbulent flow to improve the thermal and hydraulic efficiency. A perforated steel strip with a length of 1000 mm, a thickness of 2 mm, a width of 16 mm, a hole spacing of 100 mm, , a twist ratio of 6.25, and a hole diameter of 3 mm were used. The tape was

inserted into the inner tube of the exchanger, while the Al_2O_3 /water nanofluid was prepared at three volume concentrations: 0.1%, 0.2%, and 0.3%. Multiple experiments were conducted under Reynolds number conditions ranging from 4800 to 7800. The nanofluid was prepared using nanosized Al_2O_3 particles (20–30 nm) using a two-stage mixing method, including mechanical stirring followed by ultrasonic processing. The dynamic viscosity increased by 5–9% depending on the concentration. The thermal conductivity of the nanofluid increased by 12.5% compared to water at 0.3%, while the density increased slightly, but this increased the pumping power required. These changes helped enhance convective heat transfer but also contributed to increased pressure loss. Thermal and flow maps showed that Nanoparticle motion (Brownian motion) helped increase turbulence and velocity distribution, especially near the walls. Introducing PTTI reduced the thickness of the thermal boundary layer, thereby increasing the efficiency of wall-to-fluid heat transfer. The findings showed that the inclusion of the perforated PTTI strip alone increased the Nu by 127–146% compared to the smooth tube. When combined with the nanofluid, the Nusselt number increased by 146–162% at $\phi = 0.1\%$, 155–168% at $\phi = 0.2\%$, and 158–175% at $\phi = 0.3\%$. This indicates that the best thermal enhancement was achieved at the highest nanofluid concentration. The study recorded also an increase in pressure differential due to the introduction of the perforated tape. The pressure differential increased as follows: 176–186% when using PTTI alone, and 196–201% when using PTTI with $\phi = 0.3\%$ nanofluid. The results for the exergonic efficiency showed 21% increased using PTTI alone at $\text{Re} = 7800$, while the highest increase was 36% when using PTTI with Al_2O_3 at a concentration of 0.3%.

Qudah et al. [63] introduced a new geometric design for a spiral tape with rings within a circular tube aims to enhance heat transfer by generating enhanced vortices and periodic interruption of the thermal boundary layer. The study was implemented using the response surface methodology (RSM) to determine the interactive relationship between design variables and their impact on thermal and hydraulic performance indicators. 93 experiments were conducted in a specialized thermal laboratory using a circular aluminum tube with inserts of spiral ribbons embedded with metal rings at specific points. This configuration aims to enhance turbulence, velocity distribution, and heat transfer coefficient without creating excessive energy vortices. Four

independent variables were carefully analyzed: air speed (17.5 to 23.5 m/s), pitch (distance between ribbon turns): 90, 180, 270 mm, ribbon length (800, 1000, 1200 mm), and ribbon height (18, 21, 24 mm). The results showed that at the highest speed, pitch = 90 mm, length = 1200 mm, and height = 24 mm, the maximum Nu value was 43% higher than the smooth tube. The integrated rings enhanced performance by an additional 13% by creating local turbulence and stimulating microscopic air evaporation. A significant increase in the coefficient of friction (f) was recorded when the tape was inserted with the rings, with the increase ranging from 200% to 700%, depending on the twist density. The highest-pressure loss was observed at higher speeds and lower pitch. The rings added local resistance to flow, but their thoughtful geometric placement helped prevent flow stagnation. The results also showed that the lowest entropy generation rate was achieved at a strip height of 24 mm and pitch of 180 mm, with a length of 1000 mm and a velocity of 21 m/s. The entropy rate was reduced by 34% compared to the tube equipped with a strip without rings.

2.6. IDENTIFIED GAPS AND NEED FOR CURRENT STUDY

Although passive heat transfer enhancement techniques have been widely studied, most existing research focuses on conventional methods such as twisted tapes or simple ring inserts. Very few studies have explored the impact of complex or non-traditional ring designs, especially those with geometric variations that could disrupt the flow in a more controlled and efficient manner. This oversight leaves a significant gap in understanding how specially designed internal rings—with adjusted diameters, spacing, or cross-sectional shapes—might influence both the thermal and hydraulic performance of heat exchangers.

Another key gap lies in the limited investigation of how these complex ring structures interact with hybrid nanofluids, such as Al_2O_3 -Cu/water, under turbulent flow conditions. While mono-nanofluids have been explored extensively, the combination of hybrid fluids with advanced insert designs remains largely unexplored. This limits the practical applicability of prior findings to real-world thermal systems that operate at high Reynolds numbers and demand optimal performance with minimal pressure losses.

Moreover, existing studies often isolate variables—either focusing on geometry or fluid properties—but rarely examine their combined effects. Additionally, while some rely on experimental methods, they tend to avoid the use of detailed computational simulations that can capture nuanced flow behaviors and optimize configurations.

This study is therefore positioned to fill these gaps by evaluating a novel and more intricate ring design integrated with twisted tape and hybrid nanofluid, under turbulent flow. Using CFD analysis, the aim is to provide a deeper understanding of how these innovative geometries influence heat transfer and friction, and to develop practical insights for next-generation heat exchanger systems.

PART 3

MATERIALS AND METHODS

This chapter presents the materials and methodologies employed in the design, modeling, and analysis of the enhanced heat exchanger system investigated in this study. Given the complexity of convective heat transfer in turbulent flows augmented by hybrid nanofluids and passive inserts, a systematic and detailed methodological framework was essential. The chapter begins by describing the geometric configuration of the heat exchanger, including dimensions and physical characteristics, followed by the design parameters of the twisted tape and ring inserts used to intensify thermal performance. The preparation and thermophysical characterization of the $\text{Al}_2\text{O}_3\text{--Cu}$ /water hybrid nanofluids are also outlined to support the simulation accuracy. Furthermore, a comprehensive explanation of the numerical approach using Computational Fluid Dynamics (CFD) is provided, including mesh generation, boundary conditions, and turbulence modeling. Validation procedures and analytical metrics are then detailed to ensure the reliability of the numerical results. The methodological rigor adopted in this chapter provides a robust foundation for analyzing the heat transfer enhancements explored in the subsequent sections.

Figure 3.1 presents the flowchart outlines the comprehensive process of designing and analyzing a computational fluid dynamics (CFD) model. It begins with selecting the object, defining physical properties and boundary conditions, and conducting mesh generation. A grid independence test is performed to ensure result reliability, followed by validation against smooth tube data and literature values. If validation fails at any stage, the model is revised and adjusted. The process includes evaluating various geometric configurations and hybrid models to identify the optimal setup. The iterative nature of the workflow ensures accurate, validated, and optimized simulation results.

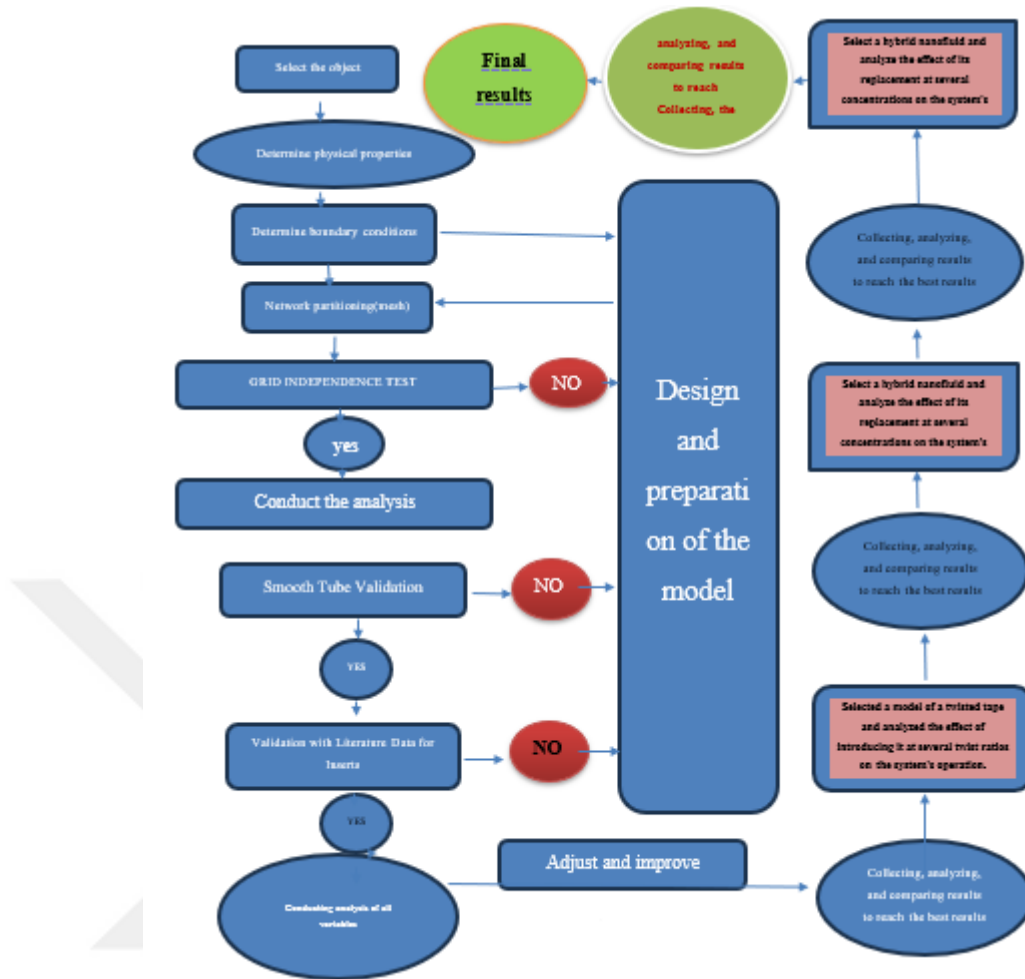


Figure 3.1. Flowchart of Model Design, Simulation, and Validation Process.

3.1. DESCRIPTION OF HEAT EXCHANGER GEOMETRY

The heat exchanger utilized in this study is a circular tube-type model, specifically designed to evaluate the performance improvements derived from hybrid nanofluid utilization and combined passive insert technologies. The geometry has been carefully selected to replicate realistic configurations typically used in industrial thermal systems such as process heaters, power plants, and HVAC equipment. The chosen tube geometry allows the incorporation of multiple internal modifications, including twisted tape inserts and ring turbulators, while maintaining compatibility with standard numerical simulation frameworks in CFD environments.

The main tube of the heat exchanger is a straight circular duct with an inner diameter (**D**) of 50 mm, an outer diameter of 53 mm, and a length (**L**) of 2000 mm. The variable design parameters in this study include the ratio of the inner diameter of the ring inserts (**d**) to the inner diameter of the tube (**D**), explored at values of $d/D = 0.55, 0.65, 0.75$, and 0.85 . Additionally, the axial spacing between successive rings (**L**) is varied at 225 mm, 275 mm, 325 mm, and 375 mm. These spacings correspond to non-dimensional pitch ratios (**P/D**) of 4.5, 5.5, 6.5, and 7.5, respectively, where **D** denotes the tube's inner diameter, fixed at 50 mm. The main objective of these configurations is to investigate the influence of d/D and P/D ratios, along with the Reynolds number (**Re**), on thermal and hydraulic performance, aiming to maximize heat transfer efficiency while minimizing pressure losses. A comprehensive summary of the geometric and flow parameters, including both dimensional and non-dimensional values, is provided in Table 1. The tube wall material is assumed to be steel due to its widespread application in industrial heat exchangers and its suitable thermal conductivity (approximately 45 W/m·K).

To assess local thermal and hydraulic variations along the tube, a uniform circular cross-section is maintained throughout the test section, and no curvature or fin extension is added externally.

Table 3.1. Geometric and Flow Parameters for the Tubular Heat Exchanger with Ring Inserts

Boundary classification	values	Unit
Length of pipe	2000	mm
The inner diameter of the pipe	50	mm
The outer diameter of the pipe	53	mm
Pitch	225,275,325,375	mm
Reynolds number (Re)	4000-14000	
Velocity	$v = 0, w = 0$, and $u = u_{in}$	
Diameter ratio (d/D)	0.55,0.65,0.75,0.85	
Inlet temperature of water	25	$^{\circ}C$
Flux of heat	constant	
Pipe Metals	Steel	

Water Density (ρ)	1000	kg/m ³
Water Specific heat capacity (C_p)	4180	J/kg·K
Water Kinematic viscosity (ν)	0.4127	(m ² /s)
Water Dynamic viscosity (μ)	0.891×10^{-3}	N·s/m ²
Water Thermal Conductivity (k)	0.608	W/(m·K)

3.1.1. Wall Characteristics and Heat Flux Application

The inner wall of the heat exchanger is subjected to a constant heat flux condition, mimicking typical applications in solar thermal systems, electronic cooling devices, or heated pipe loops. A uniform constant heat flux was applied along the outer surface of the test section in the simulation to promote a steady-state thermal environment within the tube. The wall is modeled as thermally conductive, and conjugate heat transfer between the wall and the fluid is solved using coupled thermal-solid interaction modeling in CFD.

The thickness of the pipe wall is 1.5 mm, which provides sufficient mechanical integrity while ensuring effective conduction from the external heat source to the working fluid. In addition, the thermal resistance offered by the tube material is accounted for in the simulation to accurately estimate temperature gradients across the wall-fluid interface.

3.1.2. Turbulence and Flow Behavior Considerations

Given that the operational Reynolds numbers exceed 4000, the flow regime within the heat exchanger is fully turbulent. This influences both the fluid behavior and thermal boundary layer characteristics along the pipe. The smooth pipe surface, coupled with uniform circular geometry, provides a consistent baseline for comparison against cases that include insert modifications. The wall roughness is set to zero in the baseline case to isolate the impact of internal inserts. To accurately represent turbulence effects, the CFD model utilized the RNG k - ε turbulence model with enhanced wall treatment.

3.1.3. Segmenting for Insert Configurations

To accommodate the integration of ring-type inserts within the tubular heat exchanger, the computational domain is systematically segmented. The total pipe length is set at 2000 mm, with ring inserts installed along the central test section. The inserts are axially distributed at fixed intervals, corresponding to pitch lengths (P) of 225 mm, 275 mm, 325 mm, and 375 mm. These values yield non-dimensional pitch ratios (P/D) of 4.5, 5.5, 6.5, and 7.5, respectively, based on the pipe's inner diameter (D) of 50 mm.

Each ring insert is designed to span the internal cross-section of the tube, and its inner diameter (d) is varied to evaluate different blockage effects. The ratios of ring diameter to tube diameter (d/D) considered in this study are 0.55, 0.65, 0.75, and 0.85. These geometrical configurations are implemented in such a way that they allow for fluid recirculation and turbulence enhancement, while ensuring structural compatibility with the pipe's inner geometry.

3.1.4. Justification of Geometrical Parameters

The selected geometrical parameters are based on both practical design constraints and established heat transfer enhancement strategies. The inner diameter of the pipe (D) is 50 mm, a representative size for medium-scale industrial and experimental setups, while the outer diameter is 54 mm. The axial lengths for pitch spacing between rings ($P = 225\text{--}375$ mm) are chosen to produce a broad range of turbulence intensities and flow disruptions, enabling comparative performance analysis across different non-dimensional P/D values (4.50 to 7.50).

Similarly, varying the d/D ratio from 0.55 to 0.85 allows for a systematic study of obstruction levels and their influence on the thermal boundary layer. Lower d/D ratios introduce higher blockage, promoting greater mixing and heat transfer, while higher d/D ratios reduce flow resistance.

The Reynolds number (Re) range of 4000 to 14000 ensures that the simulations remain within the fully turbulent regime, appropriate for analyzing industrial heat exchanger

conditions. The inlet water temperature is fixed at 25°C to standardize thermal comparisons across all cases. Finally, steel was selected as the pipe material due to its high thermal conductivity ($\sim 80 \text{ W/m}\cdot\text{K}$) and ease of fabrication, making it an ideal candidate for enhanced heat transfer applications.

3.2. RING AND TAPE INSERT CONFIGURATIONS

In order to enhance the convective heat transfer capabilities of the base heat exchanger system, this study introduces two major types of passive inserts: (1) twisted tape inserts and (2) ring-type turbulators. Each configuration is examined individually and in combined forms under various geometrical and operational settings to determine its influence on the thermal and hydraulic performance of the heat exchanger. These inserts are designed and implemented within the inner flow path of the heat exchanger tube, causing flow disturbances that promote turbulence, enhance mixing, and disrupt thermal boundary layer mechanisms central to passive heat transfer enhancement.

3.2.1. Ring Inserts

Ring-type inserts are discrete geometric obstructions placed inside the heat exchanger tube at regular intervals. These rings act as turbulence promoters, causing flow separation, recirculation, and reattachment, all of which contribute to thermal boundary layer disruption and enhanced mixing of the fluid core and wall layers.

Figure 3.2 illustrates the three-dimensional geometric structure of specially designed ring-type inserts used to enhance convective heat transfer within tubular heat exchangers. The figure displays three views of the insert: a frontal view, an isometric angled view, and a perspective internal view positioned inside the tube.

Figure 3.2 (a) shows the geometric design of the annulus from a direct frontal perspective, showing a regular and precise arrangement of several symmetrically distributed openings and protrusions. The oval openings in the annulus allow fluid to flow through them, resulting in a homogeneous fluid distribution. The internal protrusions (fins) are specially and precisely engineered to achieve two main goals:

Increasing the level of turbulence in the flow (turbulence generation) and creating secondary vortices, which enhance heat transfer by increasing the heat exchange rates between the fluid and the tube walls.

Figure 3.2 (b) provides a better understanding of the insert's thickness and three-dimensional shape. It shows that the radial blades and elliptical perforations are not simply surface features but extend through the thickness of the insert. The thickness of the insert (~2 mm) provides structural strength while preserving low axial space consumption, allowing multiple inserts to be placed along the heat exchanger length.

This view also emphasizes the axial alignment of the ring insert with the tube. The z-axis, representing the flow direction, intersects orthogonally with the radial cut-outs, confirming that these inserts are intended to interrupt and restructure axial flow patterns, transforming them into swirling, recirculating, or converging/diverging secondary flows. Such flow modifications are essential for disturbing the thermal boundary layer near the wall, thereby increasing local heat transfer coefficients.

Figure 3.2 (c) depicts the ring insert as it would appear mounted inside the inner wall of the tube. This placement view is critical for understanding the interaction between the insert and the confined cylindrical domain of the heat exchanger. The insert snugly fits against the pipe wall, indicating a press-fit or fixed placement, which ensures stability under turbulent flow and avoids flow leakage between the insert and tube wall.

From this perspective, it can observe how the internal features, blades, and perforations affect the flow path. As fluid enters the insert zone, it is split and redirected by the blades, then partially recombined through the elliptical perforations. The resulting flow field is characterized by strong recirculation zones, increased turbulence kinetic energy, and chaotic mixing, all of which are beneficial for convective heat transfer.

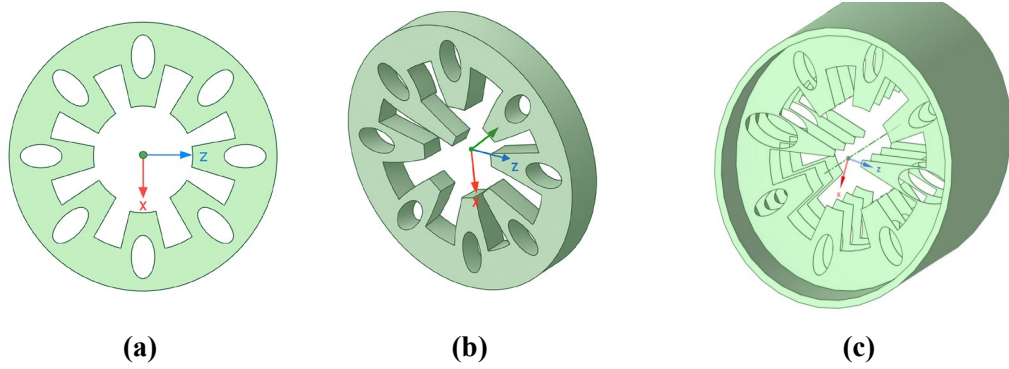


Figure 3.2. Geometric structure of the special ring inserts for heat exchanger enhancement

The special rings investigated in this research are characterized by distinct geometric attributes:

- Each ring has a complete circular shape with a central hole matching the inner diameter, allowing smooth fluid flow along the central axis. This design helps maintain steady fluid momentum and prevents excessive pressure drop within the system.
- Elliptical openings with specific dimensions (12×6 mm and 6×4 mm) are strategically placed between these radial slots. The elliptical holes introduce additional turbulence by generating localized velocity variations and small-scale vortex patterns, which enhance fluid mixing and promote a more uniform temperature distribution across the pipe's cross-sectional area.
- Radial slots arranged uniformly around the ring extend outward from the central opening toward its outer edge. These slots play a crucial role by creating disturbances in the fluid flow, encouraging turbulence and secondary swirling patterns. Such disturbances effectively disrupt the thermal boundary layer at the tube wall, significantly improving convective heat exchange.
- Rings are positioned along the length of the tube at carefully selected intervals (axial spacing of 5 times the pipe diameter). This arrangement allows sufficient distance for turbulent flow structures to develop fully between consecutive rings, balancing improved heat transfer performance against potential pressure losses.

Figure 3.3 provides a detailed illustration of the axial arrangement of a set of specially designed rings inserted inside a cylindrical tube as a technique for improving thermal performance in heat exchangers. These rings feature a special geometric design that includes precisely shaped and dimensional protrusions or fins arranged sequentially at specific intervals called the "pitch between rings."

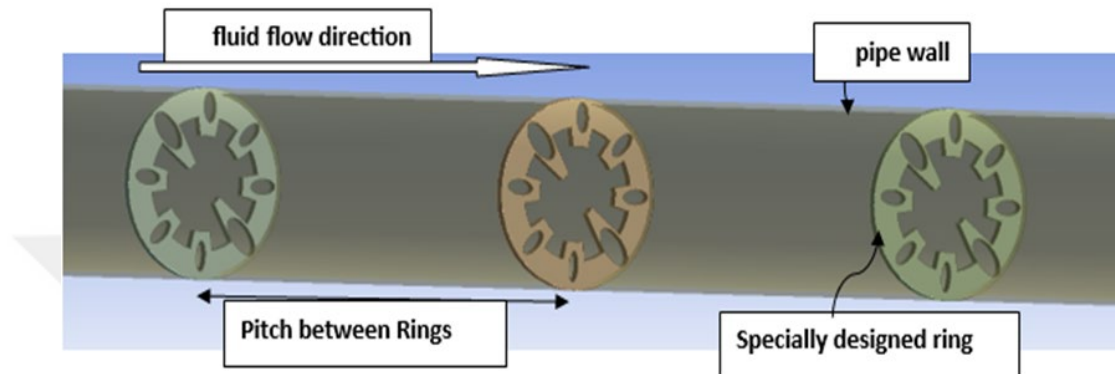


Figure 3.3. Axial arrangement of special ring inserts within the cylindrical pipe used in the CFD model

3.2.2. Twisted Tape Inserts

Twisted tape (TT) inserts are continuous metal strips twisted helically along their longitudinal axis and inserted within the tube to induce swirl flow. These tapes force the working fluid into a spiral motion, enhancing radial mixing and suppressing the growth of thermal boundary layers. The twisted nature of the tape increases the effective path length of the fluid and introduces centrifugal forces that further improve convective heat transfer.

Figure 3.4 illustrates the geometric configuration of a twisted tape with three different twist ratios (TR): 5.5, 10.5, and 15.5. The twist ratio is defined as the ratio of the pitch length to the width of the tape or tube. The use of these twisted tapes within heat exchangers aims to increase flow turbulence and enhance vortex motion, which improves heat transfer rates. Varying the twist ratio (TR) allows designers to achieve a balance between improving heat transfer and controlling pressure losses, enabling them to select the most appropriate one based on the application and operating conditions. Figure 3.4 (a) shows a tape with a relatively short twist, meaning the twists

are more frequent along the tape. This design produces highly turbulent flow due to the frequent bends, which significantly enhance heat transfer but can significantly increase pressure losses. Figure 3.4 (b) demonstrates a medium-twisted ribbon, with a lower twist frequency compared to the previous case ($TR = 5.5$), achieving a balance between enhanced heat transfer and reduced pressure loss. This ratio is considered a compromise widely used in practical applications. Figure 3.4 (c) presents a ribbon with a relatively long twist, meaning less frequent twisting compared to the previous two designs. This results in relatively less flow turbulence, thus moderately improving heat transfer while minimizing pressure losses.

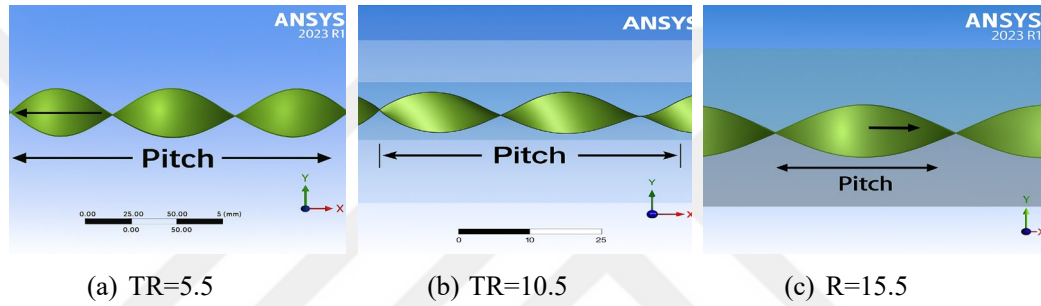


Figure 3.4. Twisted tape configurations with different twist ratios (5.5, 10.5, 15.5)

In this study, twisted tape inserts were fabricated from steel strips with a thickness of (1) mm and a width of 10 mm (approximately 20% of the tube's internal diameter). The tapes were twisted at three different twist ratios ($TR = H/D$) of 5.5, 10.5, and 15.5, where H represents the pitch length required for one 360° twist and D is the width of the twisted tape in the tube. The lower twist ratio ($TR = 5.5$) results in tighter twisting, creating stronger swirl intensity and higher turbulence, while higher twist ratios ($TR = 10.5$ and 15.5) produce milder swirl effects and reduced flow resistance.

3.2.3. Combined Configurations: Twisted Tape + Ring Inserts

One of the most innovative aspects of this study is the incorporation of compound inserts that combine the advantages of both twisted tape and ring turbulators. In these configurations, twisted tapes run continuously through the test section while ring inserts are placed at regular intervals over or around the tape. This approach leverages

two different flow manipulation mechanisms: the swirling action induced by the tape and the flow obstruction caused by the rings.

This hybrid configuration aims to maximize turbulence intensity and enhance thermal performance while controlling the increase in pressure drop. The relative positioning of the inserts was systematically adjusted, placing rings upstream, downstream, or co-aligned with the twisted tape path. Experimental and numerical analyses were conducted to determine the optimal spacing and geometry that balance heat transfer enhancement with manageable hydraulic penalties.

Figure 3.5 provides a schematic representation of the hybrid enhancement strategy employed in the present study, wherein twisted tape inserts and specially shaped ring inserts are used in combination inside a circular tube heat exchanger. This compound configuration is designed to synergistically enhance convective

The schematic shows a cylindrical tube with a transparent wall for visualization. The flow direction is indicated from left to right, simulating the path of the working fluid (e.g., hybrid nanofluid) through the heat exchanger. Inside the tube, two enhancement devices are visible:

- A helically twisted tape extends along the central axis of the tube.
- Multiple ring-shaped inserts are spaced axially with uniform pitch between them.

This setup exemplifies a compound passive heat transfer enhancement technique, where multiple geometric features work in tandem to intensify fluid mixing and surface heat transfer.

What makes this configuration particularly effective is the interaction between the swirling motion created by the twisted tape and the flow disturbances caused by the rings. The twisted tape enhances bulk fluid motion and promotes radial mixing, while the rings introduce axial disturbances and induce secondary flows. Together, these mechanisms create a highly chaotic flow field characterized by:

- Enhanced mixing across the pipe diameter
- Intensity of turbulence intensity
- Improved thermal penetration depth
- Suppressed laminar sublayers near the wall

This synergy significantly boosts the Nusselt number compared to either insert used in isolation, while the thermal performance factor (TPF) remains favorable due to optimized spacing and geometry.

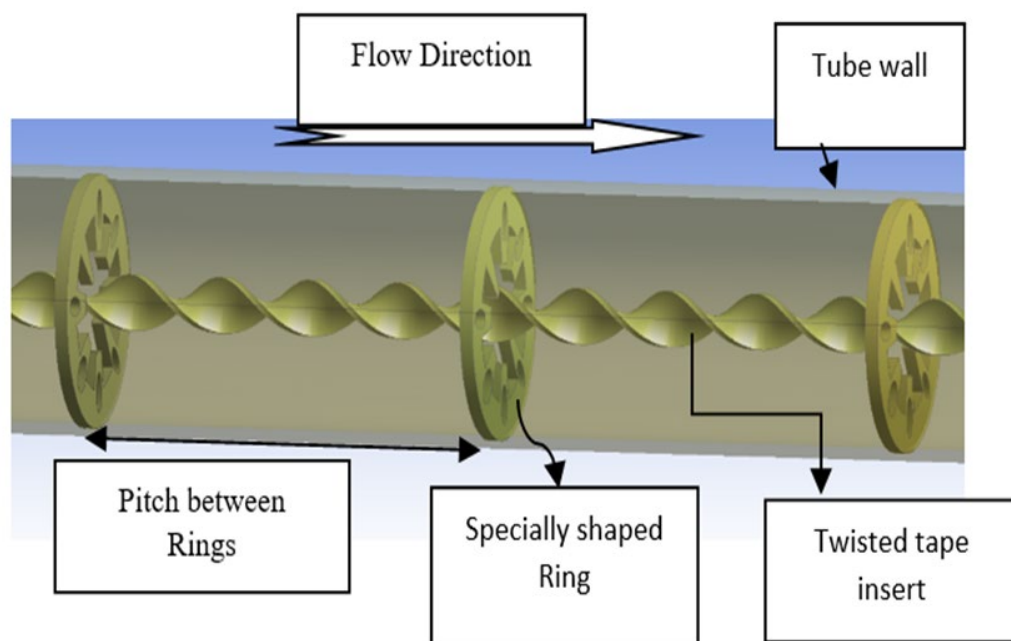


Figure 3.5. Schematic of a Heat Exchanger Tube with Twisted Tape and Specially Shaped Ring Inserts.

3.3. HYBRID NANOFLUIDS FOR HEAT TRANSFER ENHANCEMENT

Improving heat transfer efficiency in heat exchangers is increasingly important to reduce energy consumption and enhance thermal performance in industrial and engineering applications. Nanofluids are one of the most important emerging technologies, gaining widespread attention due to their superior thermal properties compared to conventional fluids. Recently, a more advanced trend has emerged in the

use of hybrid nanofluids, which combine two or more different types of nanoparticles in a single base fluid to further enhance thermal and mechanical properties.

Among the nanoparticles that have been widely studied academically are aluminum oxide (Al_2O_3) and copper oxide (Cu), due to their unique properties and the potential for their integration into a hybrid nanofluid to achieve significant improvements in heat transfer rates.

Figure 3.6 shows high-resolution TEM images demonstrating the nanostructure of aluminum oxide (Al_2O_3) and copper (Cu) particles used in the preparation of the hybrid nanofluid. The figure consists of two parts: These images provide a direct view of the structure of nanoparticles and the nature of interactions and interference between them when forming hybrid nanofluids. They highlight the importance of controlling the stability of these particles to avoid excessive agglomeration, which may negatively affect the heat transfer properties and stability of nanofluids used in heat exchangers.

Part (a) shows the distribution of the Al_2O_3 and Cu nanoparticles together. It is noted that the particles are characterized by very small nanoscale sizes (less than 100 nm) and have a clear tendency to aggregate and cluster, demonstrating the importance of using dispersion methods to ensure good distribution in the base fluid. Part (b) shows a closer view of the nanoparticles, more clearly illustrating the shape and size of the clusters. The Cu particles appear red in the image due to their high atomic density, while the Al_2O_3 particles appear gray. This visual illustration allows for a clear distinction between the two types of particles.

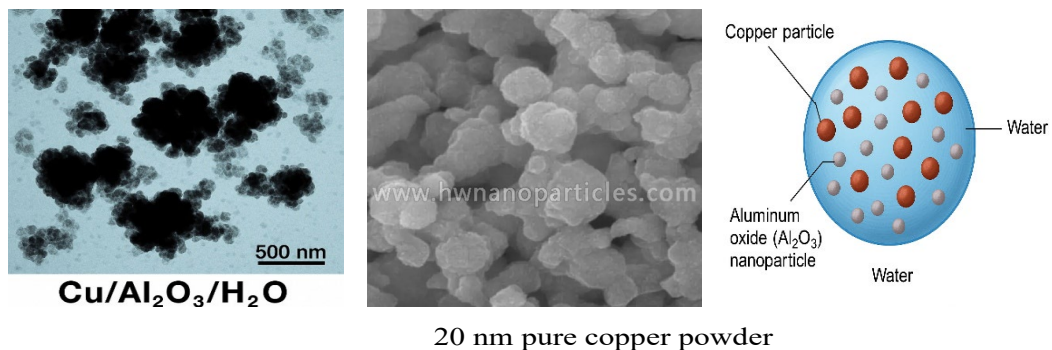


Figure 3.6. Structure of nanoparticles (Al_2O_3 and Cu) used in the hybrid nanofluid

3.3.1. Properties of the Nanoparticles Used

Al_2O_3 nanoparticles are among the most popular materials used in nanofluids due to their distinctive properties:

- Good chemical stability and low reactivity with base fluids such as water and oils.
- High thermal conductivity (approximately $30\text{--}40 \text{ W/m}\cdot\text{K}$).
- Non-toxicity and suitability for environmental and industrial applications.
- Wide availability and relatively low cost.

Studies have shown that adding Al_2O_3 to base fluids significantly improves thermal conductivity, but can lead to an increase in viscosity, which increases pumping energy consumption.

Cu nanoparticles also possess unique properties that make them highly suitable for use in nanofluids:

- Excellent thermal stability, especially at high operating temperatures.
- Very high thermal conductivity (approximately $76.5 \text{ W/m}\cdot\text{K}$).
- Excellent ability to improve heat transfer, particularly in high-performance cooling systems.

However, Cu nanoparticles are more prone to agglomeration than Al_2O_3 nanoparticles, requiring the use of stabilizers or advanced dispersion techniques to ensure their stability in the base fluid. Table 3.1 provides a comparison of the thermophysical properties of aluminum oxide (Al_2O_3) and copper oxide (Cu) nanoparticles.

Table 3.2. Thermophysical Properties of water, Al₂O₃ and Cu Nanoparticles [1][2][3]

Physical properties	Water	Cu	Al ₂ O ₃
CP (J/Kg K)	4179	385	765
$\dot{\rho}$ (Kg/m ³)	997.1	8933	3970
K W/(m·K)	0.613	400	40
μ N. s/m ²	0.000891		
Average Particle Size (nm)		30	40
Purity(%)		>99%	>99%

The idea of using hybrid nanofluids emerged to overcome the limitations associated with each type of nanoparticle individually. Combining Al₂O₃ and Cu particles provides the following advantages:

- Combining high-thermal-conductivity particles, such as Cu, with chemically stable particles, such as Al₂O₃, improves both stability and conductivity.
- Al₂O₃ particles help reduce the tendency of Cu particles to agglomerate, ensuring better stability of the hybrid nanofluid over long periods.
- This hybrid formulation allows for greater heat transfer enhancement with lower energy consumption, thereby increasing the efficiency of thermal and engineering systems.

3.3.2. Thermal Specifications of Hybrid Nanofluids

The following equations describe the fundamental thermophysical characteristics of hybrid nanofluids, including thermal conductivity, density, heat capacitance, and dynamic viscosity [110,111].

Volume fraction of nanoparticles in the nanofluid:

$$\phi_{hnf} = \phi_{np1} + \phi_{np2} \quad (3-1)$$

Density of hybrid nanofluid:

$$\rho_{hnf} = \rho_{np1}\Phi_{np1} + \rho_{np2}\Phi_{np2} + (1 - \Phi_{hnf})\rho_{bf} \quad (3-2)$$

Specific heat of hybrid nanofluid:

$$Cp_{hnf} = \frac{\Phi_{np1}\rho_{np1}Cp_{np1} + \Phi_{np2}\rho_{np2}Cp_{np2} + (1 - \Phi_{hnf})\rho_{bf}Cp_{bf}}{\rho_{hnf}} \quad (3-3)$$

Thermal conductivity of hybrid nanofluid

$$K_{hnf} = K_{bf} \frac{\left(\frac{\phi_{np1}k_{np1} + \phi_{np2}k_{np2}}{\phi_{total}}\right) + 2k_{bf} + 2(\phi_{np1}k_{np1} + \phi_{np2}k_{np2}) - 2\phi_{total}k_{bf}}{\left(\frac{\phi_{np1}k_{np1} + \phi_{np2}k_{np2}}{\phi_{total}}\right) + 2k_{bf} - (\phi_{np1}k_{np1} + \phi_{np2}k_{np2}) + \phi_{total}k_{bf}} \quad (3-4)$$

Viscosity of hybrid nanofluid:

$$\mu_{hnf} = \frac{\mu_{bf}}{(1 - \phi_{np1} - \phi_{np2})^{2.5}} \quad (3-5)$$

where:

Φ_{hnf} is the volume fraction of the hybrid nanofluid.

Φ_{np1} and Φ_{np2} represent the volume fractions of the first and second nanoparticles, respectively.

ρ_{hnf} is the density of the hybrid nanofluid is denoted (kg/m³).

ρ_{hnf} , and ρ_{np1} indicate the densities of nanoparticles 1 and 2.

Cp_{hnf} is the specific heat capacity of the hybrid nanofluid (J/kg·K).

p_{np1} and Cp_{np2} (J/kg·K) correspond to the specific heat capacities of nanoparticles 1 and 2.

K_{hnf} is the thermal conductivity of the hybrid nanofluid is given (W/m·K).

k_{np1} and k_{np2} representing the thermal conductivities of nanoparticles 1 and 2.

μ_{hnf} is the viscosity of the hybrid nanofluid is denoted as (Pa·s).

The thermophysical properties of Cu and Al₂O₃ nanoparticles, along with their corresponding hybrid nanofluids, are provided in Table 3.3.

Table 3.3. Thermal properties of the hybrid nanofluid.[3][4]

Φ (%)	Cp (J/kg·K)	ρ (Kg/m ³)	μ (Pa·s)	K (W/m·K)
0.1	4155.337752	1002.5544	0.000972	0.61998
1	3953.422799	1051.644	0.0016	0.657
2	3750.091153	1106.188	0.001935	0.6849

3.4. NUMERICAL MODELING APPROACH (CFD)

Computational Fluid Dynamics (CFD) has become an indispensable tool in the analysis and optimization of thermal systems, especially for evaluating complex fluid dynamics and heat transfer behavior under varied operating and geometrical conditions. In this study, CFD modeling was employed to simulate turbulent flow and heat transfer characteristics of hybrid nanofluids within a circular tube heat exchanger equipped with twisted tape and ring inserts. The numerical simulations aimed to achieve the following:

- Quantify the enhancement in convective heat transfer due to twisted tape and ring inserts.
- Evaluate pressure drop and flow resistance across different geometries.
- Investigate the synergistic effect of hybrid nanofluids with passive inserts.
- Analyze turbulence behavior and flow structures using streamline, velocity, and temperature contours.

The results were used to calculate the Nu number, the friction factor (f), and the thermal performance factor (TPF) for different Reynolds numbers and insert configurations.

3.4.1. Modeling Assumptions

To simplify the computational process while maintaining physical accuracy, the following assumptions were made:

- The flow is steady, incompressible, and three-dimensional.
- The working fluid (hybrid nanofluid) is treated as a single-phase homogeneous medium.
- The thermophysical properties of the nanofluid are temperature-dependent and implemented via User Defined Functions (UDFs).
- No phase change, chemical reaction, or radiation heat transfer is considered.
- The walls are thermally and mechanically smooth, with no-slip boundary conditions.

These assumptions are widely adopted in heat exchanger modeling literature and validated against experimental data.

3.4.2. Governing Equations

The following equations govern the flow and heat transfer inside the tube:

- **Continuity Equation** (Mass Conservation) is a fundamental principle of conservation of mass in fluid dynamics.

$$\nabla \cdot \vec{u} = 0 \quad (3-6)$$

- **Momentum Conservation Equation** (Navier-Stokes): Physics and engineering that describe the motion of a fluid or object due to forces acting on it

$$\rho(\vec{u} \cdot \nabla \vec{u}) = -\nabla p + \nabla \cdot [\mu(\nabla \vec{u} + \nabla \vec{u}^T)] \quad (3-7)$$

- **Energy Equation** (Heat Conservation): that describes the conservation of energy in a system, it is based on the first law of thermodynamics (energy cannot be created or destroyed, only transformed or transferred)

$$\rho c_p(\vec{u} \cdot \nabla T) = \nabla \cdot (k \nabla T) \quad (3-8)$$

where:

ρ is the density,

V is the velocity.

P is the pressure.

μ is the dynamic viscosity.

C_p is the specific heat capacity, and

T is the temperature.

3.4.3. Turbulence Modeling

Given that the flow operates in the turbulent regime ($6000 \leq Re \leq 14000$), selecting an appropriate turbulence model is essential. After comparing different options, the RNG k- ϵ turbulence model was selected due to its enhanced accuracy in swirling and rotating flows(The K-epsilon model was used because most of the research results to be compared were based on this model. Additionally, the fluid used has a low density, and layers at surfaces, edges, and holes were treated by increasing the concentration of the solvent using the program's available options.), which are characteristic of the twisted tape-induced motion.

The RNG model solves two transport equations:

Transport Equation for Turbulent Kinetic Energy (k)

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho k u_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k - \rho \varepsilon \quad (3-9)$$

Transport Equation for Turbulent Dissipation Rate (ε):

$$\frac{\partial(\rho \varepsilon)}{\partial t} + \frac{\partial(\rho \varepsilon u_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_{1\varepsilon} \frac{\varepsilon}{k} G_k - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} - R \quad (3-10)$$

where:

ρ : Fluid density.

u_j : Velocity component in direction x_j

μ_t : Dynamic viscosity

$\sigma_k, \sigma_\varepsilon$: Turbulent Prandtl numbers for k and ε

G_k : Production of turbulent kinetic energy due to mean velocity gradients

R : Additional term unique to the RNG model accounting for strain-rate effects, defined as:

$$R = C_\mu \rho \eta^3 \left(1 - \frac{\eta}{\eta_0} \right) \frac{\varepsilon^2}{k} \quad (3-11)$$

This model provides better prediction of separation and recirculation zones, especially around inserts, compared to the standard k - ε model.

3.4.4. Numerical Solver Settings

The following solver settings were used in ANSYS Fluent:

- **Solver Type:** Pressure-based
- **Time:** Steady state
- **Velocity Formulation:** Absolute
- **Pressure-Velocity Coupling:** SIMPLE algorithm
- **Spatial Discretization:**
 - Gradient: Least Squares Cell Based
 - Pressure: Second Order

- Momentum, Energy, Turbulence: Second-Order Upwind

These settings ensure numerical stability and accurate resolution of gradients near walls and inserts.

3.4.5. Convergence Criteria

Convergence was monitored for residuals and surface monitors:

- **Residuals:** $1.0e^{-6}$ for energy, $1.0e^{-4}$ for continuity, momentum, and turbulence equations.
- **Surface Monitors:** Average outlet temperature, pressure drop, and heat transfer coefficient were tracked to ensure physical convergence, not just numerical.

3.4.6. Inlet Boundary Conditions

Based on the data presented in Table 3.1, the inlet boundary condition for the tubular heat exchanger simulation is defined with a uniform velocity profile, where the axial velocity component is set as $u = u_{in}$, while the transverse velocity components are zero ($v = 0$, $w = 0$). This ensures a fully developed, one-directional flow at the inlet. The inlet temperature of the working fluid (water) is maintained at 25 °C, providing a consistent thermal condition for the start of the simulation. Additionally, the Reynolds number ranges from 4000 to 14000, indicating turbulent flow conditions at the inlet. These inlet parameters ensure realistic thermal and flow behavior that aligns with typical operating conditions in compact heat exchanger systems.

3.5. GRID INDEPENDENCE TEST

A Grid Independence Test (GIT) was conducted to verify the accuracy and reliability of the numerical simulation results by systematically refining and optimizing the mesh density within the computational domain. This procedure was performed using the meshing module of ANSYS Fluent 2023 R2. Mesh optimization involved adjusting

critical parameters, such as mesh sizing, refinement near critical regions, mesh scaling, and applying smoothing techniques, which collectively improved node distribution and enhanced overall mesh quality.

Figure 3.7 shows the mesh generation process and the distribution of mesh elements within a heat exchanger model containing twisted tape and ring inserts. This model was created using ANSYS, a popular tool for computational fluid dynamics (CFD) analysis. The figure shows the importance and accuracy of the network generation design as a basic stage before numerical analysis, to ensure the accuracy and reliability of the results that will be obtained from the simulation.

The left image represents a cross-section of one of the special rings used to enhance heat transfer. The ring contains internal openings and protrusions, and the mesh was generated to capture these fine details. The mesh in this region is denser at the edges and corners, as these areas experience rapid changes in flow properties (such as velocity, pressure, and temperature). Using a fine mesh here is essential to obtain accurate numerical results from the simulation.

The right image shows the 3D structure of the heat exchanger. It is clear that there are several consecutive rings and a twisted tape running through them. The model is divided into multiple regions (each a different color: green, brown, gray, etc.) to distinguish each part of the design within the ANSYS environment. The twisting band generates flow turbulence and internal vortices that help improve heat transfer. The loops, in turn, increase the intensity of turbulence and prevent the formation of a stable thermal boundary layer, increasing heat transfer efficiency.

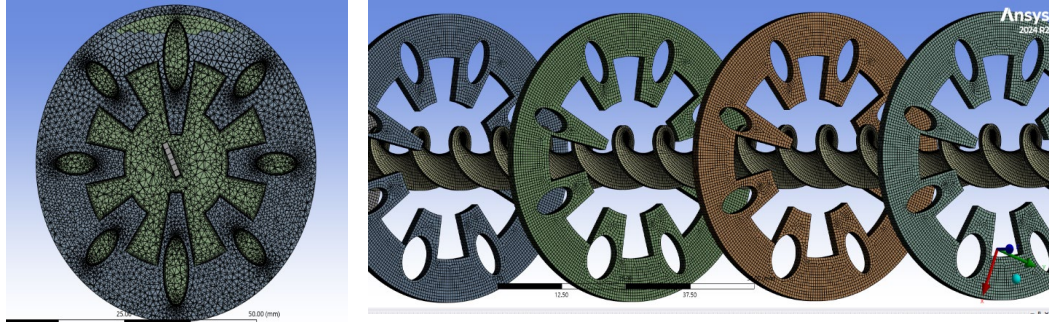


Figure 3.7. Structured Meshing of a Multi-Ring Heat Exchanger with Twisted Tape Inserts

Figure 3.8 shows the results of the grid independence test, a fundamental test performed within CFD analysis to ensure that simulation results do not depend significantly on the number of mesh elements but rather reflect the actual physical behavior of the model. The goal of this test is to determine the optimal number of mesh elements that achieves a balance between accuracy and computational time and ensures that the simulation results are driven by real physical phenomena and not solely by the number of mesh elements. Using a mesh that is too fine is time-consuming and resource-intensive to achieve practical benefits. The curve shows the change in the Nu value as the number of mesh elements increases. Initially, we observe significant fluctuations in the Nu value at small mesh numbers, then the value gradually stabilizes after exceeding a certain number of elements. At a low element count (less than 4×10^5 , the values are relatively unstable, indicating that the mesh is insufficient to accurately represent the heat flow. Once the element count exceeds approximately 6×10^5 , the values begin to gradually stabilize, and changes in Nu become very minor. This stability indicates that increasing the number of elements beyond this limit does not significantly affect the results, and thus the model becomes "mesh independent." The network independence test shows that the optimal number of network elements is within the range of 1×10^6 , where the results become stable and reliable without the need to further increase the elements, which achieves efficiency in calculations and simulations.

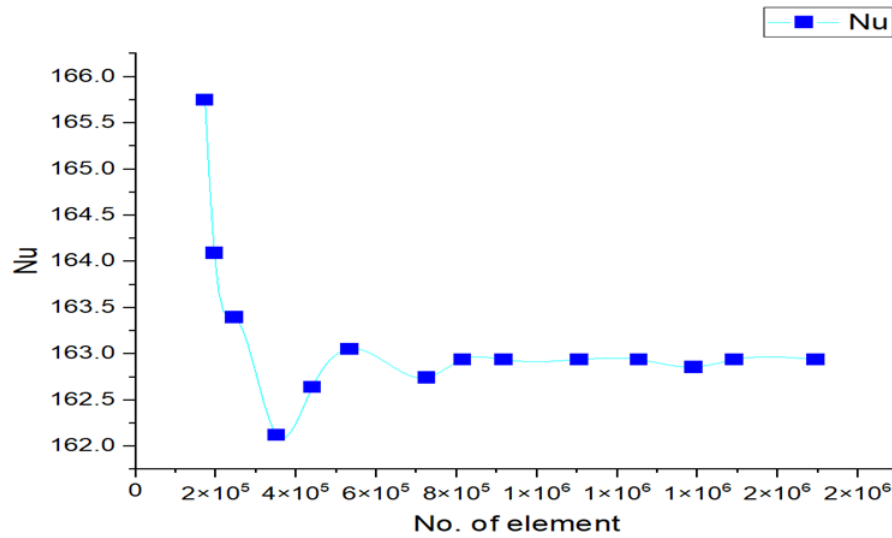


Figure 3.8. Grid independence test results.

3.6. VALIDATION OF SIMULATION MODELS

The validation process was designed to assess the predictive accuracy of the CFD model in the following aspects:

- **Heat transfer characteristics**, primarily through comparison of Nusselt number (Nu).
- **Pressure drop analysis**, through evaluation of friction factor (f).

Validation was carried out in two stages:

- **Smooth Tube Baseline Case** (deionized water, no inserts): Compared against the classical Dittus-Boelter and Gnielinski correlations.
- **Enhanced Configurations**: Compared against experimental and numerical results from peer-reviewed literature involving similar geometries and Reynolds number ranges.

3.6.1. Smooth Tube Validation

A critical validation step involved simulating the flow of pure water (no nanoparticles) in a plain tube under turbulent conditions without any inserts. The Reynolds number

was varied from 6000 to 14,000, and the Nusselt number results were compared with the Dittus-Boelter and Gnielinski empirical correlations:

- **Dittus-Boelter correlation** (for $Re > 2300$, $Pr = 0.7-160$):

$$Nu_s = 0.023 Re^{0.8} Pr^{0.4} \quad (3-12)$$

- **Gnielinski correlation** (for $3000 < Re < 5 \times 10^6$):

$$Nu_s = \frac{\frac{f}{8} (Re - 1000) Pr}{1 + 12 \cdot 7 \left(\frac{f}{8}\right)^{0.5} \left(P_r^{\frac{2}{3}} - 1\right)} \quad (3-13)$$

Where f is the friction factor estimated from the Blasius equation:

$$f_s = 0.316 Re^{-0.25} \quad (3-14)$$

Figure 3.9 compares the Nusselt number (Nu_s) as a function of the Reynolds number (Re_s) for three data sources: Dittus–Boelter, Gnielinski, and the Present Study. All three datasets show a consistent upward trend, indicating that the Nu increases with increasing Re . All three predictions are relatively close in the range ($Re \approx 5000-10000$). The present study slightly underestimates Nu compared to the Gnielinski equation but remains close to the Dittus–Boelter prediction. This indicates good agreement with classical models under lower turbulence conditions. In the mid- Re Region ($Re \approx 10000-20000$), the Nu number from the present study increases more rapidly and approaches or exceeds the Dittus–Boelter values while staying slightly below Gnielinski's predictions. This suggests that the geometry or enhancement method used in the present study (e.g., twisted tape or ring inserts) affects heat transfer performance more noticeably in this regime. In the higher flow rate region ($Re \approx 20000-26000$), the present study closely matches the Gnielinski results and shows noticeable improvement over the Dittus–Boelter model. This confirms the enhanced thermal performance achieved by the system simulated in the current work. The present study effectively validates its simulation model and experimental setup,

showing results within a reasonable range of the two established correlations, especially matching Gnielinski's data at higher Reynolds numbers.

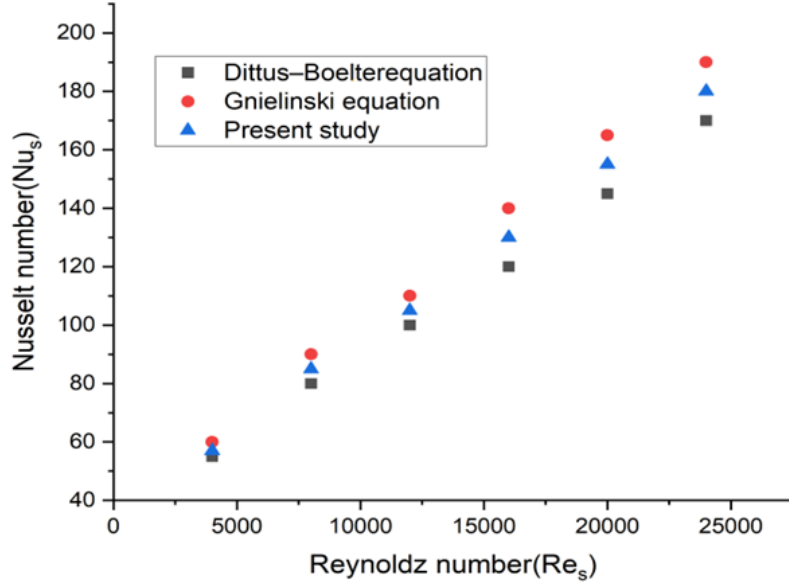


Figure 3.9. Validation of Nusselt Number Predictions: Comparison Between the Present Study and Standard Correlations

For the same smooth tube cases, the simulated pressure drops were used to calculate the Darcy friction factor:

$$f = \frac{2 \cdot \Delta P \cdot D.}{\rho^2 \cdot L \cdot u} \quad (3-15)$$

These results were then compared with the Blasius and correlations for turbulent flow:

- **Blasius equation**

$$f_s = 0.316 Re^{-0.25} \quad (3-16)$$

- **Petukhov equation**

$$f_s = (0.79 \ln Re - 1.64)^{-2} \quad (3-17)$$

Figure 3.10 illustrates a validation of the friction factor (f_s) as a function of the Re (Re_s), by comparing the results of the present study with two well-established empirical correlations: the Blasius equation and the Petukhov equation. The results of

the present study are in good agreement with standard theoretical correlations, particularly at higher Re . This reinforces the credibility of the simulation or experimental setup and confirms that the pressure loss characteristics introduced by the modifications (e.g., inserts or nanofluids) are within acceptable engineering limits. As expected, all curves exhibit a decreasing trend in the friction factor as the Re increases, indicating that the flow becomes more streamlined with higher turbulence intensity. At lower Re (≈ 4000 – 8000), the present study's friction factor is slightly lower than that predicted by the Blasius equation, but still higher than Petukhov's values. As the Re increases (>15000), the gap between the three data sets narrows, and the present study remains between the Blasius and Petukhov models.

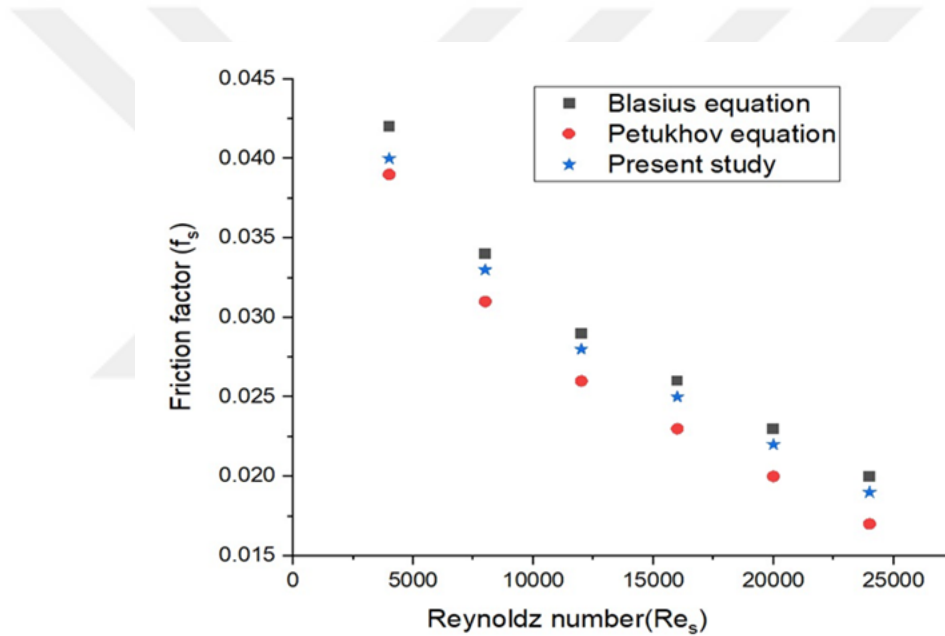


Figure 3.10. Validation of Friction Factor Predictions: Comparison Between the Present Study and Standard Correlations

3.6.2. Validation with Literature Data for Inserts

Table 3.4 presents a comparative analysis between previous studies and the current work for a circular heat exchanger tube equipped with an insert ring. The results show that the heat transfer enhancement ratio (Nu/Nu_s) in the present study ranges from 2 to 5.1, which lies within and slightly exceeds the upper range of many previous studies, indicating a notable improvement in thermal performance. The relative friction factor (f/f_s) falls between 2.3 and 24.7, which is moderate compared to some earlier works,

such as [112], which reported values up to 75. This suggests that the present ring insert design achieves a balanced trade-off between heat transfer enhancement and pressure drop. Most significantly, the thermal performance factor (TPF) in this study ranges from 0.85 to 3.21, representing one of the highest TPF values reported among the referenced studies. This highlights the effectiveness of the proposed ring design in enhancing overall system efficiency with relatively lower frictional penalties.

Table 3.4. Comparisons between the previous studies and the present work results for a circular heat exchanger tube integrated with an insert ring

Ref.	Nu/Nu_s	(f/f_s)	TPF
[113]	3.76–4.9	25.25–36.07	1.46–1.61
[114]	2.2–4.2	5–36	1.03–1.4
[115]	2.62–3.9	7.2–48	0.88–1.47
[112]	1.8–4.3	5–75	0.88–2.23
[116]	2.65–3.3	9–13	1.18–1.48
[117]	2.35–4.06	2.3–24.7	0.82–1.24
Present work	2–5.1	2.3–24.7	0.85–3.21

3.7. DATA ANALYSIS PARAMETERS

In this section, the performance evaluation metrics and analytical expressions used to interpret the results of the CFD simulations are presented. These parameters serve as the basis for comparing the thermal and hydraulic behavior of different insert configurations and nanofluid concentrations. By calculating these performance indicators consistently across all cases, a comprehensive analysis of system efficiency, trade-offs, and optimization opportunities can be achieved.

3.7.1. Overview of Performance Metrics

The evaluation of thermal and hydraulic performance in enhanced heat exchangers typically involves three primary dimensionless parameters:

- **Nusselt Number (Nu)**
- **Friction Factor (f)**
- **Thermal Performance Factor (TPF)**

Each of these parameters is computed using output variables from the CFD simulations, including temperature, pressure, velocity, and fluid properties. The dimensionless parameters in this analysis (Re, Pr, Nu, f , and TPF) are calculated by [118,119] :

- Reynolds Number:

$$Re = \frac{\rho u D}{\mu} \quad (3-18)$$

- Prandtl Number:

$$Pr = \frac{\mu c_p}{k} \quad (3-19)$$

- Nusselt Number:

$$Nu = \frac{hD}{k} \quad (3-20)$$

- Friction Factor:

$$f = \frac{\Delta P}{\left(\frac{L}{D}\right) \left(\frac{\rho u^2}{2}\right)} \quad (3-21)$$

- Thermal Performance Factor (TPF):

$$TPF = \frac{Nu/Nu_s}{(f/f_s)^{\frac{1}{3}}} \quad (3-22)$$

Here, Nu_s and f_s represent the Nusselt number and the friction factor, respectively, for a smooth tube, while TPF denotes the thermal performance of the system.

3.8. UNCERTAINTY ANALYSIS

The uncertainty analysis of the data from Table 3.5 indicates that the overall reliability of the calculated thermal and hydraulic parameters is high. The Nu and heat transfer coefficient (h) exhibit very low error margins of approximately $\pm 0.45\%$. The friction factor (f) shows a slightly higher uncertainty at $\pm 0.66\%$, which remains within acceptable bounds for fluid flow studies. The TPF has the highest relative uncertainty at $\pm 1.51\%$, likely due to its dependence on both thermal and frictional characteristics. To determine the uncertainty, use the relative standard error (RSE) as a percentage.

$$RSE(\%) = \left(\frac{S_{Error}}{X} \right) * 100 \quad (3-24)$$

Where

S_{Error} the standard error.

X the mean of the collected data

The standard error is calculated as:

$$S_{Error} = \frac{\sigma}{\sqrt{n}} \quad (3-25)$$

where:

σ the standard deviation

n the repeatable readings

$$\sigma_k = \sqrt{(\sigma q1)^2 + (\sigma q2)^2 + \dots + (\sigma qn)^2} \quad (3-26)$$

where :

σ_k the total uncertainty and $(\sigma q1)$, $(\sigma q2)$, (σqn) are uncertainty of individual parameters.

Table 3.5. Uncertainty and Error Analysis of Key Parameters in Heat Transfer Study

Parameter	Max Value	Min Value	Range of Uncertainty	Error Range ($\pm\%$)
Nusselt Number (Nu)	503.26	153.77	1.00	0.45%
Heat Transfer Coefficient (h)	6893.64	1867.30	12.56	0.45%
Friction Factor (f)	0.819	0.136	0.0020	0.66%
TPF	4.38	0.72	0.0266	1.51%

3.9 Quality of the network

Table 3.6 presents the mesh element count used for different values of the diameter ratio (d/D) while maintaining acceptable skewness levels, specifically between 0.25 and 0.5. As the d/D ratio varies, the number of elements was adjusted to ensure grid quality and simulation accuracy. The mesh settings were optimized to strike a balance between computational efficiency and numerical stability.

Table 3.6. Number of Mesh Elements for Various d/D Cases with Skewness Between 0.25 and 0.5.

Cases (d/D)	No. element
1 0.55	10842187
2 0.65	10556349
3 0.75	10375360
4 0.85	10194206
5 0.75+ Tr	16771051

PART 4

RESULTS AND DISCUSSION ON THE PERFORMANCE OF NEWLY DESIGNED INNER RINGS

This chapter presents a detailed analysis of the thermal and hydraulic performance of heat exchanger tubes equipped with specially designed ring inserts. The study evaluates the impact of varying geometric parameters, specifically pitch-to-diameter ratio (P/D), and the inner-to-outer diameter ratio (d/D) on key performance indicators such as temperature distribution, velocity distribution, velocity streamline, Nusselt number, heat transfer coefficient, friction factor, and thermal performance factor (TPF). By analyzing a wide range of Reynolds numbers, the investigation provides a comprehensive understanding of how these geometric configurations influence flow behavior, turbulence generation, and heat transfer enhancement. The goal is to identify the optimal configuration that achieves maximum thermal efficiency with minimal hydraulic penalty.

4.1. INFLUENCE OF SPECIAL RING INSERTS ON THERMAL AND FLOW CHARACTERISTICS

The thermal and flow behaviors within the heat exchanger tube equipped with specially designed ring inserts were investigated at a Reynolds number of 10,000 and a pitch ratio (P/D) of 5.5 for various inner-to-outer diameter ratios ($d/D = 0.55, 0.65, 0.75, \text{ and } 0.85$). The results, presented in Figures 4.1–4.3, provide a comprehensive understanding of how geometric variation in insert size affects temperature distribution, axial velocity fields, and streamline patterns.

Figure 4.1 illustrates the temperature distribution (Temperature Contours) inside a heat exchanger tube with specially inserted rings of different sizes (d/D ratio) at a $Re=10,000$ and a step ratio of $P/D = 5.5$. The results confirm that the special ring design has an effective impact on improving heat transfer, particularly at d/D ratios between 0.65 and 0.75. At $d/D = 0.75$, the best balance between improved thermal performance and reduced pressure loss is achieved, making it the optimal configuration in this study.

The findings reveal that $d/D = 0.55$ shows a clear thermal gradient is visible in the center in yellow, surrounded by red on the sides, indicating that the wall temperature is higher than the center temperature of the flow. The internal inlet is relatively small, indicating a moderate effect on flow turbulence and improved heat transfer. The findings present that as the d/D ratio increases from 0.55 to 0.85, heat transfer gradually improves, as demonstrated by the widening of the medium- and low-temperature regions (green and blue). Increasing the d/D ratio means that the internal insert is closer to the tube walls, increasing flow turbulence and reducing the thermal boundary layer. The best thermal performance is observed at $d/D = 0.75$ or 0.85, but this must be balanced against the pressure loss resulting from these modifications. At $d/D = 0.65$, a more pronounced thermal gradient is visible, with increased heat transfer toward the center. A higher d/D ratio increases the overlap between the flow and the inlet, resulting in improved heat transfer. The heat distribution is more uniform across the cross-section at $d/D = 0.75$, and a reduced thermal gradient is observed along the length of the tube. This ratio reflects a more prominent inlet within the flow, causing greater turbulence and greater improvement in thermal conductivity. The heat distribution is more even and diffused at $d/D = 0.85$, with greener and bluer in the center, indicating increased cooling efficiency. However, this may also indicate that the drop in pressure has also become higher, due to the larger inlet size.

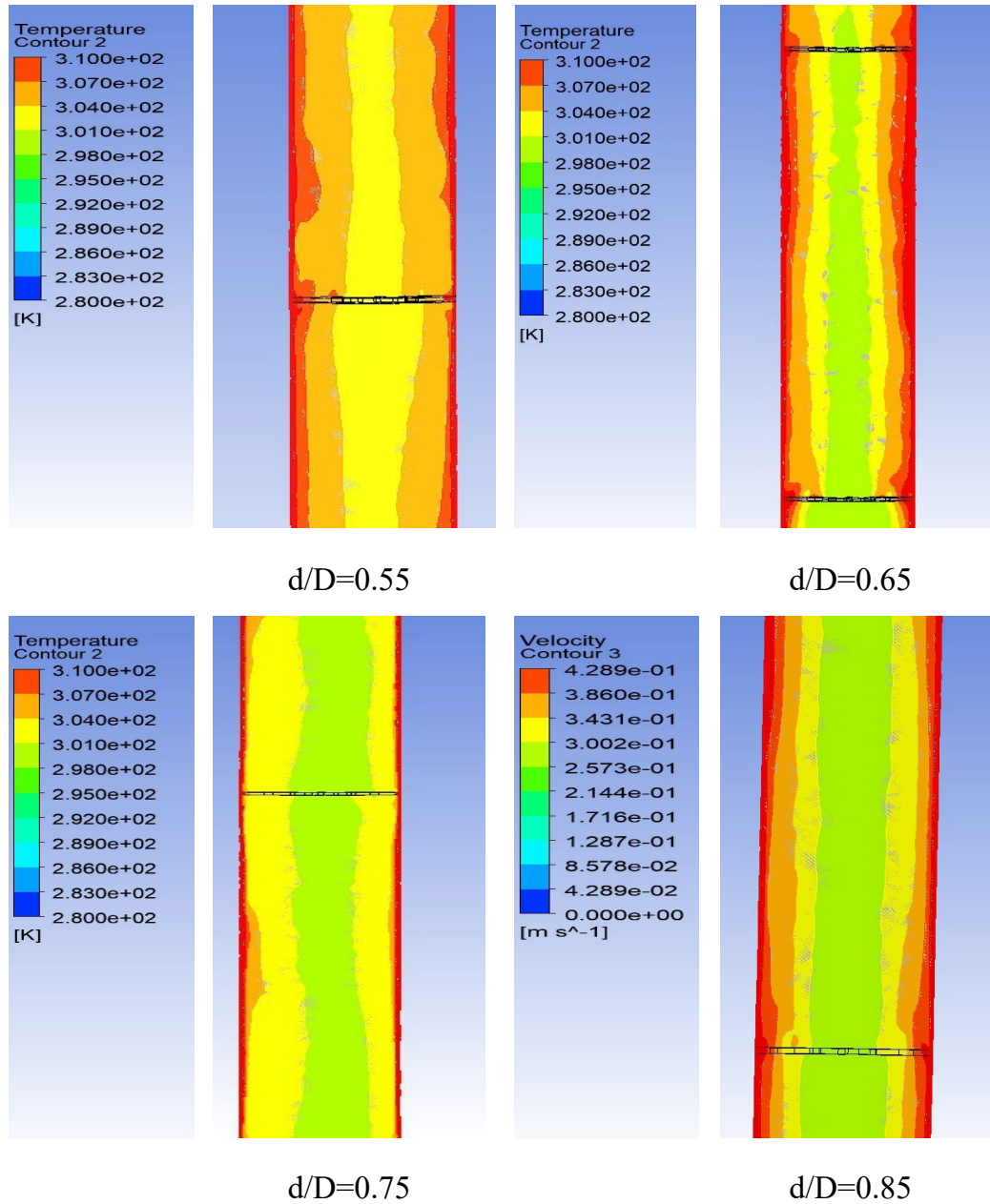


Figure 4.1. Temperature Distributions within the Heat Exchanger Tube with Special Ring Inserts at $Re = 10,000$ and $P/D = 5.5$ for Various d/D Ratio.

The special ring design contributes to changing the flow pattern inside the tube from uniform axial flow to a highly mixed turbulent flow. At $d/D=0.85$, despite the high turbulence, the severe cross-sectional restriction results in a lower overall velocity, which can impair the overall system performance from an economic perspective. At $d/D=0.75$, the best interaction between the rings and the fluid is achieved, resulting in significantly enhanced heat transfer while minimizing the impact on pressure loss.

The findings demonstrate that at a d/D configuration, the velocity map shows a central high-velocity region (in red), surrounded by low-velocity areas near the wall. This indicates that the rings have not significantly affected the central flow, meaning that the resulting turbulence is still weak, and the mixing rate and heat transfer are relatively limited in this case. The axial velocity begins to decrease further near the center of the inlets at $d/D = 0.65$, with larger green and blue areas appearing. This indicates increased flow resistance and the induction of more effective vortices, and an improvement in flow turbulence enhances heat transfer at the tube walls. Strong vortex regions appear behind the rings at $d/D = 0.75$, with a clear decrease in velocity behind them. The velocity distribution indicates that the rings redirect the flow toward the wall and then back toward the center, effectively inducing secondary vortices, and this design strikes an excellent balance between enhanced mixing and reduced pressure loss. Very large areas of low-velocity color appear at $d/D = 0.85$, including the central region, with an irregular flow distribution. This indicates that the rings impede flow, causing significant pressure loss despite enhanced cooling. This achieves maximum turbulence but is likely hydraulically inefficient.

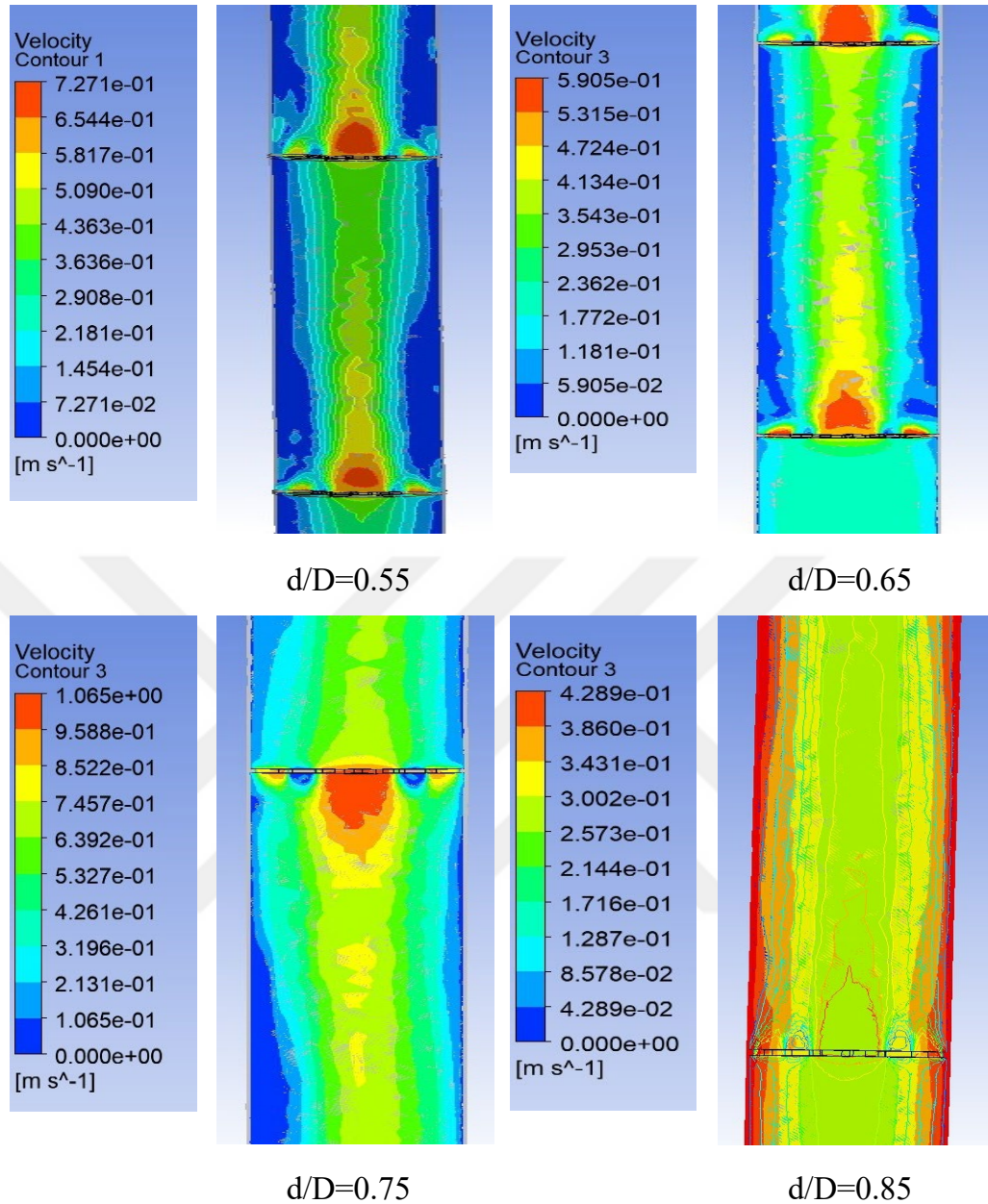


Figure 4.2. Axial Velocity Distribution inside the Heat Exchanger Tube with Special Ring Inserts at $Re = 10,000$ and $P/D = 5.5$ for Various d/D Ratio.

Figure 4.3 illustrates the flow line patterns inside a coaxial heat exchanger tube equipped with special inlet rings designed to enhance fluid turbulence and thus improve heat transfer. The lines show the fluid flow path and reveal regions of turbulence, recirculation, and vortex formation, all of which result from the interaction between the fluid and the geometric structure of the rings (radial and elliptical slots). The findings confirm that the detailed geometry of the individual annulus plays a pivotal role in shaping the flow pattern within the heat exchanger. At $d/D = 0.85$, the

highest levels of turbulence are recorded, enhancing thermal conductivity but causing high flow resistance that may reduce overall system performance. The best thermal and hydraulic performance is achieved at $d/D = 0.75$, where the annulus generates sufficient turbulence with relatively stable flow.

The results reveal that most flow lines remain uniform along the tube axis at $d/D = 0.55$, with some slight bends around the inserts. No strong vortices are visible, indicating weak turbulence and near-axial flow, and Thermal performance in this case is poor due to insufficient mixing. The flow lines begin to curve more clearly as they pass through the radial slots at $d/D = 0.65$, forming local vortex patterns behind the annulus. This indicates the beginning of hydrodynamic turbulence, which enhances heat transfer efficiency. At $d/D = 0.75$, the flow lines exhibit a complex pattern containing pronounced secondary vortices and high stream dispersion. This indicates maximum annulus effectiveness, as they create controlled turbulence without excessively impeding flow, and this is the best balance between enhancing mixing and maintaining efficient flow. Severe curvature appears in the flow lines at $d/D = 0.85$, with strong turbulence behind the inlets. Dense vortices are produced, which may lead to increased pressure loss despite enhanced mixing. This condition achieves maximum turbulence but may not be optimal regarding hydraulic efficiency.

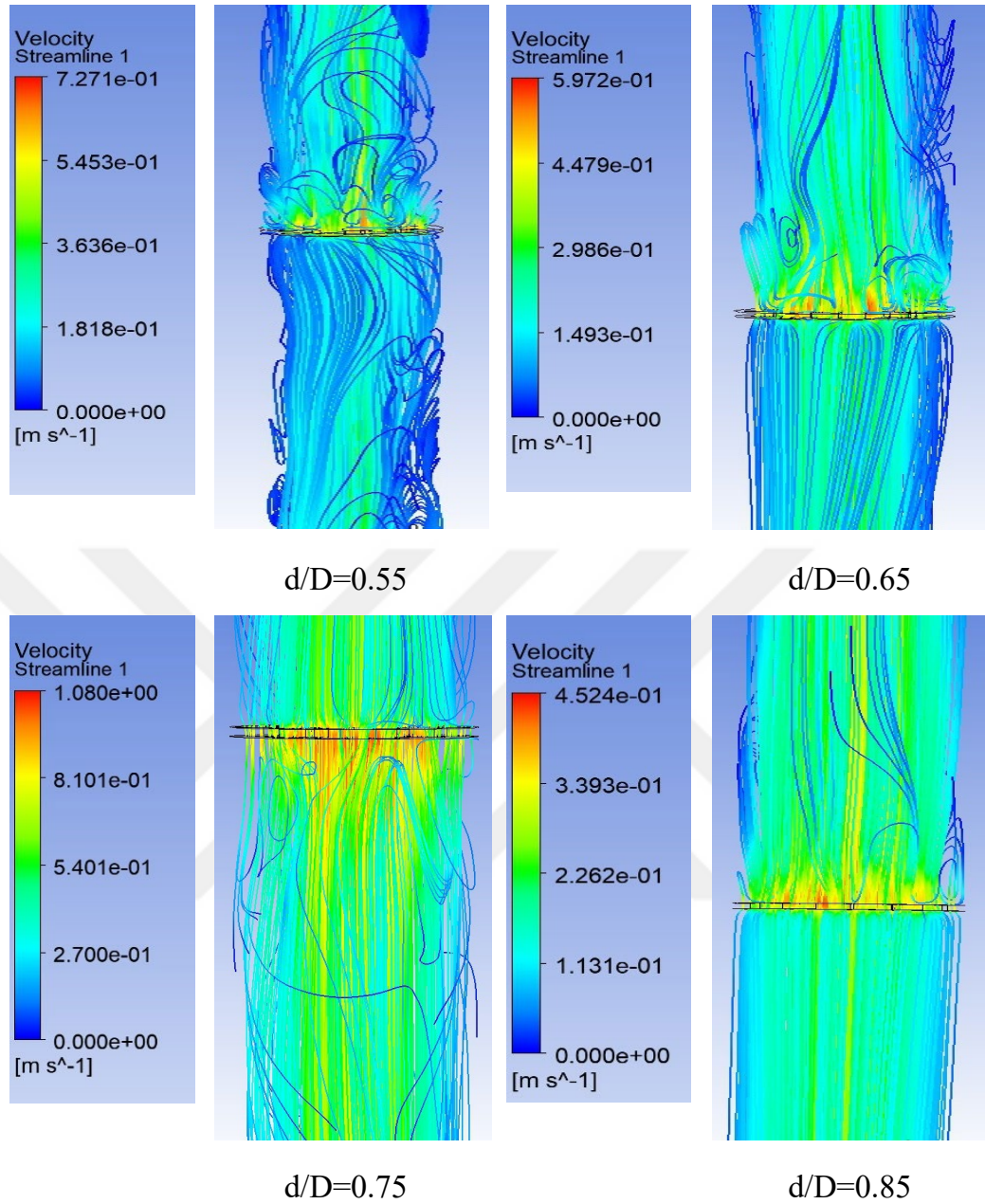


Figure 4.3. Flow Streamline Patterns inside the Heat Exchanger Tube with Special Ring Inserts at $Re = 10,000$ and $P/D = 5.5$ for Various d/D Ratio.

4.2. NUSSELT NUMBER VARIATION WITH RE AND P/D AT DIFFERENT D/D RATIOS INSIDE A HEAT EXCHANGER WITH RING INSERTS

The data in Figures 4 -7 show the variation of the Nu with increasing Re for a variety of special annulus configurations within the tube. Four different inside diameter to outside diameter ratios ($d/D = 0.55, 0.65, 0.75, 0.85$), with four different axial annulus spacings ($P/D = 4.5, 5.5, 6.5, 7.5$), were evaluated. The results confirm that increasing

the Re significantly increases heat transfer efficiency (Nu) in all cases. The results show that increasing the axial spacing between the rings (P/D) consistently improves thermal performance. The configuration $d/D = 0.75$ with $P/D = 7.5$ is optimal, providing the highest possible balance between improved heat transfer and reduced hydraulic resistance within the heat exchanger tube equipped with special rings. These results confirm the importance of carefully tuning the ring distribution and size to suit the required flow conditions.

Figure 4.4 shows the variation of the Nu with the Re when using special insert loops inside a heat exchanger tube with a d/D ratio of 0.55, for four axial loop spacings ($P/D = 4.5, 5.5, 6.5,$ and 7.5). The curves in the figure show that the Nu increases significantly with increasing Re for all studied P/D values, indicating improved convective heat transfer as the flow velocity increases. At each value of Re , it is observed that an increase in P/D is associated with an increase in the Nusselt number. For example, at $Re = 14000$: $Nu = 227.47$ at $P/D = 6.5$ (Pitch = 325 mm) while $Nu = 230.43$ at $P/D = 7.5$ (Pitch = 375 mm). The small difference indicates that increasing the pitch beyond 7.5 may not lead to significant additional improvement. Increasing P/D (i.e., increasing the loop spacing) also results in improved Nu . At $d/D = 0.55$, the loops are less prominent inside the tube, so their design effectiveness is highly dependent on their axial spacing. Increasing the loop spacing (P/D) reduces the negative interference between turbulent regions, allowing for the formation of effective vortices that improve convective heat transfer. The best thermal performance was recorded at $Re = 14000$ and $P/D = 7.5$, reaching $Nu = 230.44$, the highest Nu for this configuration. It is clear that improving heat transfer using special rings is most effective at high speeds and wide axial distances.

The findings of this figure also reveal that at $Re = 4000$ (lowest value), $Nu = 191.24$ at $P/D = 4.5$ (pitch = 225 mm), and Nu increases to 198.88 at $P/D = 7.5$ (pitch = 375 mm). An increase of approximately 4% due to the increased spacing between the rings, allowing for better flow development before encountering another ring. At $Re = 10000$ (mid-range), $Nu = 198.67$ at $P/D = 4.5$, and Nu also increases to 210.63 at $P/D = 7.5$. An improvement of approximately 6% in heat transfer demonstrated the significant effect of the rings in creating vortices and stimulating mixing within the tube. At $Re =$

14000 (highest flow velocity). $Nu = 210.68$ at $P/D = 4.5$, and Nu increases to 230.43 at $P/D = 7.5$. An improvement of more than 9% indicates that the rings' effectiveness increases under high-velocity turbulent flow, provided adequate spacing between the rings.

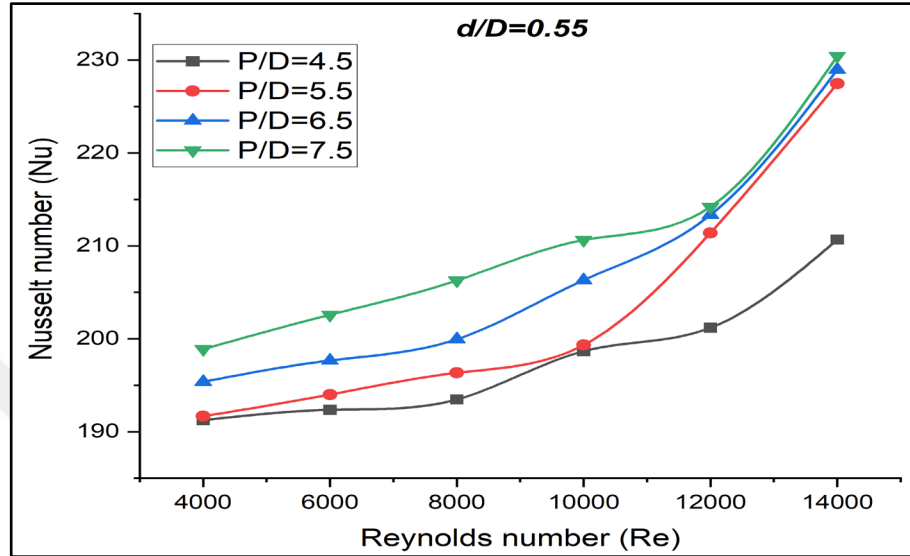


Figure 4.4. Variation of Nusselt Number with Reynolds Number for $d/D = 0.55$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts.

Figure 4.5 demonstrates the relationship between the Nu and the Re at an inner-to-outer diameter ratio of $d/D = 0.65$, with varying axial spacing between the respective rings ($P/D = 4.5, 5.5, 6.5$, and 7.5).

The Nu increases with increasing Re , reflecting improved heat transfer resulting from higher flow velocity and increased turbulence. Although the overall Nu values are higher than at $d/D = 0.55$, the differences between the P/D values remain significant, indicating that the ring geometry is very effective in this configuration ($d/D = 0.65$). The figure also shows that increasing the axial spacing between the rings (P/D) leads to higher Nu values, confirming the effect of the geometric distribution of the rings on enhancing heat transfer efficiency. Thermal performance improves but may be accompanied by a limited increase in pressure loss, which requires further analysis of the pressure results. At $d/D = 0.65$, the heat exchanger tube with the special rings significantly improves heat transfer, especially when the distance between the rings is

increased. The best performance was achieved at $Re = 14000$ and $P/D = 7.5$ (Pitch = 375 mm), where $Nu = 214.38$. These results confirm that larger ring spacing allows the fluid sufficient time to develop turbulence before encountering the next ring, enhancing mixing and improving the thermal gradient in the tube.

The results also showed that at $Re = 4000$, $Nu = 177.06$ at Pitch = 225 mm ($P/D = 4.5$), and it increased to $Nu = 193.98$ at Pitch = 375 mm ($P/D = 7.5$). An increase of 16.9 points ($\sim 9.5\%$) at the same Re indicates that increasing the inter-ring spacing allows for developing more stable vortices that stimulate thermal mixing. At $Re = 10000$, $Nu = 183.26$ at Pitch = 225 mm, while it increases to $Nu = 201.43$ at Pitch = 375 mm. A difference of 18.17 points ($\sim 9.9\%$), demonstrating that the rings at this configuration provide a turbulent environment that enhances thermal performance, especially at medium speeds. At $Re = 14000$, $Nu = 200.47$ at Pitch = 225 mm, and it increases to $Nu = 214.38$ at Pitch = 375 mm with a difference = 13.91 points. This shows that the highest relative improvement is achieved in the mid- Re range and begins to stabilize at high Re , resulting in almost complete turbulence.

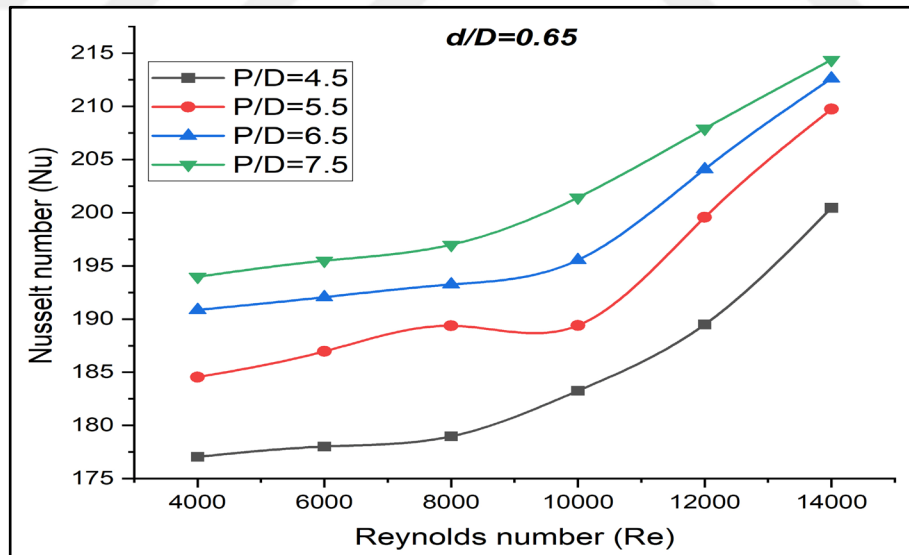


Figure 4.5. Variation of Nusselt Number (Nu) with Reynolds Number (Re) for $d/D = 0.65$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts.

Figure 4.6 shows the effect of Re and inter-ring distance (P/D) on Nu inside a heat exchanger tube with special inlet rings for $d/D = 0.75$. At $d/D = 0.75$, the rings cover

a large portion of the tube cross-section, enhancing turbulence and mixing between the flow layers. The graph clearly shows that an increasing Re and P/D increases Nu , indicating a significant improvement in convective heat transfer efficiency. Linearly, the Nusselt number increases steadily with increasing Reynolds number, resulting from increased flow intensity and the formation of vortices between the rings. At each value of Re , the highest Nusselt value is recorded when $P/D = 7.5$, while the lowest is recorded when $P/D = 4.5$. All curves show an accelerated growth after $Re = 10,000$, consistent with the flow transitioning to a fully turbulent state. Although $P/D = 7.5$ provides the best thermal performance, a large spacing may increase pressure loss, so balancing Nu and ΔP is essential to determine the thermo-hydraulic efficiency. At $d/D = 0.75$, the special rings are very effective in enhancing heat transfer, due to the increased interaction surface area and the formation of regular vortices within the tube. The thermally optimal configuration was achieved at $Re = 14,000$ and $P/D = 7.5$, where the Nusselt number reached approximately 218. These results demonstrate that $d/D = 0.75$ with $P/D = 6.5\text{--}7.5$ is a promising range for achieving high thermal performance without excessive pressure loss.

The numerical results demonstrated that at $Re = 4000$, the Nu increases from 166 to 185, reflecting an improvement of approximately 11.5%, which indicates a significant improvement in heat transfer. At $Re = 10000$, a similar increase is observed, from 179 to 198, confirming the favorable influence of larger pitch spacing in disturbing the thermal boundary layer and promoting mixing. However, at $Re = 14000$, the gain in Nusselt number diminishes to just 6.8% (from 204 to 218), suggesting that the marginal benefit of increasing the pitch ratio declines as the flow becomes fully turbulent and mixing is already well-developed.

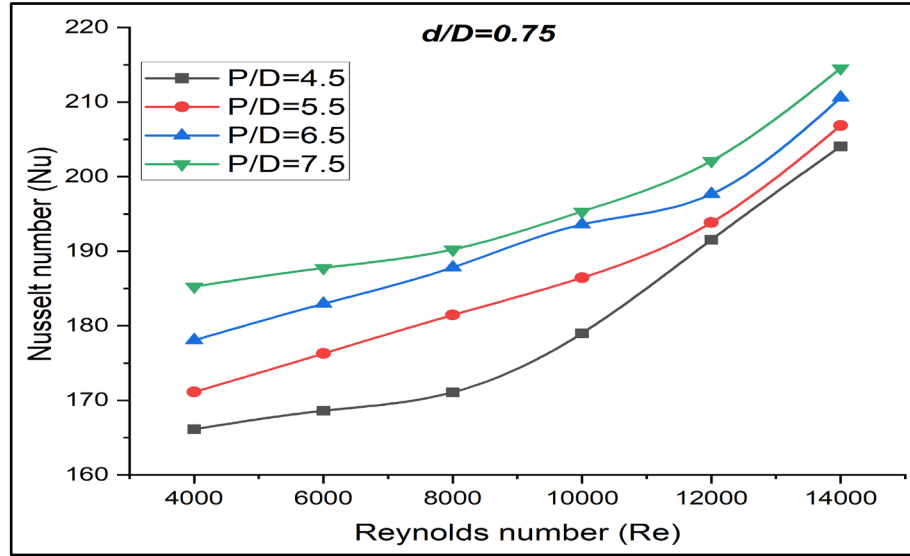


Figure 4.6. Variation of Nusselt Number (Nu) with Reynolds Number (Re) for $d/D = 0.75$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts

Figure 4.7 illustrates the variation of the Nu with Re for different pitch-to-diameter ratios ($P/D = 4.5, 5.5, 6.5$, and 7.5) at an inner-to-outer diameter ratio of $d/D = 0.85$. This figure represents a case where the annulus covers the inner tube diameter more, causing stronger flow turbulence. However, the figure shows that the heat transfer performance of this configuration does not increase as rapidly as with other ratios (such as 0.75 or 0.65) at low and medium speeds, with a significant improvement only occurring at $Re \geq 12,000$. At Re less than $10,000$, the Nu remains relatively constant or grows slowly, indicating that the turbulence generated by the rings at this large ratio of $d/D = 0.85$ impedes flow rather than enhances it. As Re increases to higher values, the flow's ability to overcome the ring resistance increases, and Nu rises more sharply, especially at $P/D = 7.5$. At $d/D = 0.85$, the rings produce strong turbulence in the flow, but they create significant hydraulic resistance, especially at low speeds. Maximum benefit is achieved only at high Re ($\geq 12,000$), where the fluid can penetrate the ring effect and achieve strong heat transfer. The highest thermal performance was achieved at $Re = 14,000$ and $P/D = 7.5$, with a Nu of 232.36.

The findings present that at $Re = 4000$, $Nu = 153.76$ at Pitch = 225 mm ($P/D = 4.5$), and it increases to $Nu = 185.20$ at Pitch = 375 mm ($P/D = 7.5$). The increment is 31.43 points ($\sim 20.4\%$), which is high but does not reflect high efficiency because all values

remain below 190 despite the rings' proximity to the wall. At $Re = 10000$, $Nu = 173.63$ at $P/D = 4.5$, and it increases to $Nu = 187.95$ at $P/D = 7.5$. The difference is 14.3 points ($\sim 8.2\%$), and the increase is limited despite doubling the speed compared to $Re = 4000$. At $Re = 14000$, $Nu = 210.66$ at $P/D = 4.5$, increasing to $Nu = 232.36$ at $P/D = 7.5$. The difference is 21.7 points ($\sim 10.3\%$), and the highest Nu is recorded in all cases studied in this figure. Figures 4.6 and 4.7 show the variation of the Nusselt number (Nu) with the Reynolds number (Re). It is observed that the rate of increase in the Nusselt number decreases beyond $Re 12,000$. This is because, at high Reynolds numbers, the influence of geometry diminishes due to dominant turbulence, leading to convergence of Nusselt numbers across different d/D ratios and ring spacings.

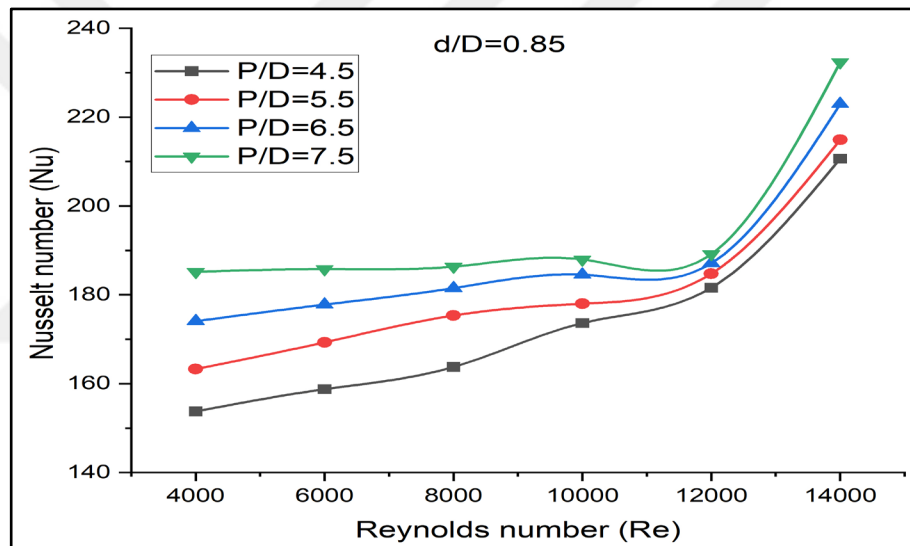


Figure 4.7. Variation of Nusselt Number (Nu) with Reynolds Number (Re) for $d/D = 0.85$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts

4.3. HEAT TRANSFER COEFFICIENT VARIATION WITH RE AND P/D AT DIFFERENT D/D RATIOS INSIDE A HEAT EXCHANGER WITH RING INSERTS

Figures 8-11 illustrate the variation of the heat transfer coefficient (h) with Reynolds number (Re) for four different d/D ratios (0.55, 0.65, 0.75, 0.85) and for four P/D ratio configurations (4.5, 5.5, 6.5, 7.5) inside a heat exchanger tube equipped with special inlet rings. These figures represent the overall thermal performance of a heat

exchanger by tracking the heat transfer coefficient (h) in $\text{W/m}^2\cdot\text{K}$, which is a direct indicator of the system's ability to remove heat from the inner wall of the tube to the fluid passing through it. The results show that h values increase with both the Reynolds number and the annulus spacing ratio P/D , but the effect varies with the d/D ratio. In general, the higher the Re and P/D ratios, the higher the heat transfer coefficient h , but the effectiveness of this effect varies with the d/D ratio. A configuration with $d/D = 0.55$ and $P/D = 7.5$ achieves the highest thermal performance at all Re values. At $d/D = 0.85$, high performance is only achieved at $Re > 12,000$, indicating the importance of high velocity to offset the negative effect of increased annulus area. Therefore, the optimal configuration is $d/D = 0.75$ or 0.65 with $P/D = 6.5$ or 7.5 , as they strike a balance between enhancing heat transfer and avoiding significant pressure loss. The findings indicated that in the case of small loops ($d/D = 0.55$), flow resistance is low, allowing for effective turbulence development and thus excellent heat transfer as Re increases. The effect of the ring resistance begins to become apparent at $d/D = 0.65$, especially at low flow velocities, but at high Re , it is compensated by vortex condensation and better heat distribution. The $d/D = 0.75$ configuration achieves a good balance between creating effective turbulence within the tube and not creating excessive flow resistance, making it suitable in terms of thermo-hydraulic performance. The large rings ($d/D = 0.85$) generate flow resistance that impedes heat transfer at low flow rates, but at high Re , they become very effective due to the development of strong turbulence that compensates for the initial resistance.

Figure 4.8 illustrates the variation of the convective heat transfer coefficient (h) with Re for various axial pitch-to-diameter ratios ($P/D = 4.5, 5.5, 6.5$, and 7.5) at $d/D = 0.55$. This case represents the smallest ring obstruction area among all tested geometries, allowing for relatively free fluid movement while still inducing useful turbulence. At $d/D = 0.55$, all curves show a continuous increase in the heat transfer coefficient (h) with increasing Re . h values increase more markedly with increasing annulus spacing (P/D), especially at higher Re values. This indicates that a relatively small-diameter annulus (low d/D) enhances turbulence without significantly impeding the flow. At $Re \geq 10,000$, the differences in h values begin to widen significantly between small and large P/D values. $P/D = 6.5$ and 7.5 appear to give similar results at $Re = 14,000$, indicating an approach to the "saturation point." Pitch = 375 performs

consistently and outperforms all other values at all Reynolds numbers. The results showed that the special rings with $d/D = 0.55$ allow for efficient turbulence development without creating excessive resistance, enhancing thermal performance under various flow conditions. The best thermal performance was achieved at $P/D = 7.5$ and $Re = 14000$, reaching $h = 2798.4 \text{ W/m}^2\cdot\text{K}$. The geometric configuration at $d/D = 0.55$ and $P/D = 7.5$ provides the most suitable balance between improved heat transfer and reduced potential pressure loss.

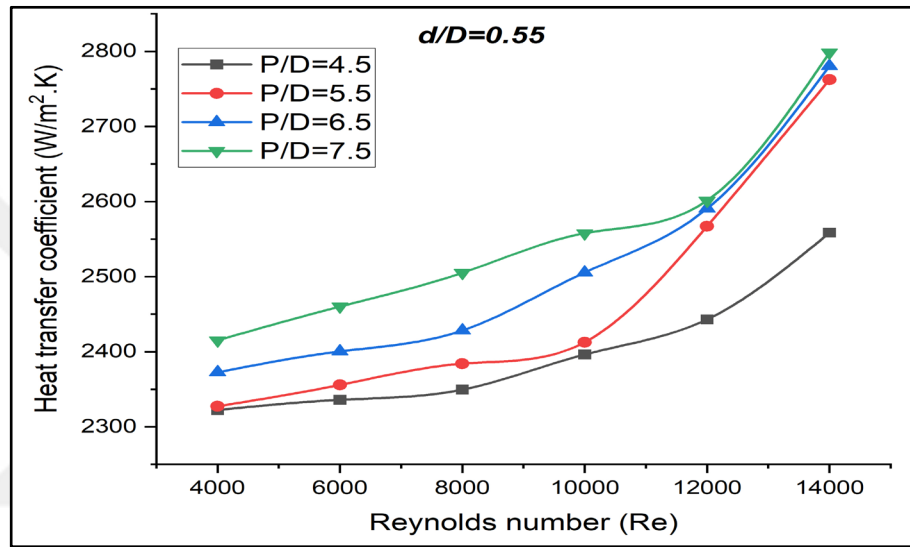


Figure 4.8. Variation of Heat Transfer Coefficient with Reynolds Number (Re) for $d/D = 0.55$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts

Figure 4.9 presents the variation of the convective heat transfer coefficient (h) with Re for various axial pitch-to-diameter ratios ($P/D = 4.5, 5.5, 6.5$, and 7.5) at $d/D = 0.65$. The data showed that the heat transfer coefficient increases significantly with increasing Re , and the values also gradually improve with increasing axial distance between the rings (P/D). It is evident that using rings with a slightly larger inner diameter ($d/D = 0.65$) still provides a good heat transfer enhancement, but the effect is less severe compared to $d/D = 0.55$. The difference between short and long rings ($P/D = 4.5$ and 7.5) is relatively constant across different Re values, indicating a stable design effect. Performance exhibits a more gradual pattern than for $d/D = 0.55$, indicating that increasing d/D relatively reduces the turbulence generated by the rings. The lowest value was recorded at $P/D = 4.5$ and $Re = 4000$, and the highest value was

recorded at $P/D = 7.5$ and $Re = 14000$. At $d/D = 0.65$, the rings continue to perform their role in improving heat transfer, with a noticeable improvement as P/D increases. The best performance was recorded at $P/D = 7.5$ and $Re = 14000$, reaching $h = 2603.4 \text{ W/m}^2\cdot\text{K}$. Although these values are higher than for $d/D = 0.75$ or 0.85 (as will be shown later), the effect of thermal turbulence is greater at lower d/D .

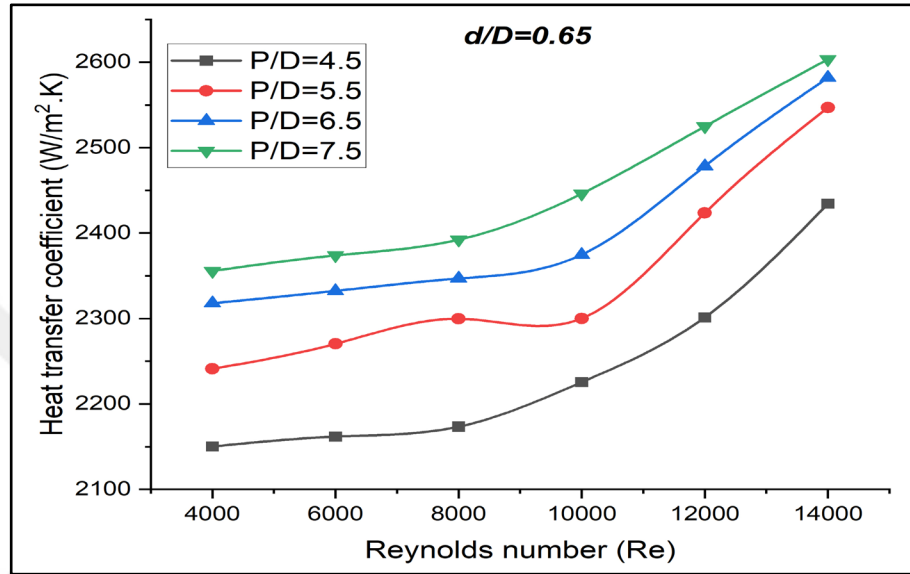


Figure 4.9. Variation of Heat Transfer Coefficient with Reynolds Number (Re) for $d/D = 0.65$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts.

Figure 4.10 presents the variation of the convective heat transfer coefficient (h) with Re for various axial pitch-to-diameter ratios ($P/D = 4.5, 5.5, 6.5$, and 7.5) at $d/D = 0.75$. The findings presented that the h coefficient continues to increase with increasing Re , but the rate of improvement becomes less dramatic compared to the previous cases ($d/D = 0.55$ and 0.65). The positive effect of increasing P/D is still present, but the differences between the distributions begin to gradually decrease at higher Re values. This indicates that increasing d/D reduces the ability of the rings to disrupt the flow and generate effective turbulence. Although the increase in the heat transfer coefficient is still significant, the rate of improvement decreases as the d/D ratio increases. The overall performance is better at $P/D = 7.5$ in all cases, but the difference from $P/D = 6.5$ is no longer significant at $Re > 12000$. Also, there is relative similarity in h values between some P/D distances at high speeds, indicating saturation of the ring effect. At $d/D = 0.75$, the effect of special rings in enhancing heat transfer is still present but less

effective than in cases with a smaller inner diameter. The best performance is achieved at $P/D = 7.5$ and $Re = 14000$, with $h = 2605.4 \text{ W/m}^2\cdot\text{K}$. Larger ring spacing results in better flow mixing, but the benefit gradually flattens out with increasing Re due to flow stability at high turbulence rates. The results recommend using smaller d/D , such as 0.55 or 0.65, when the priority is to maximize thermal performance.

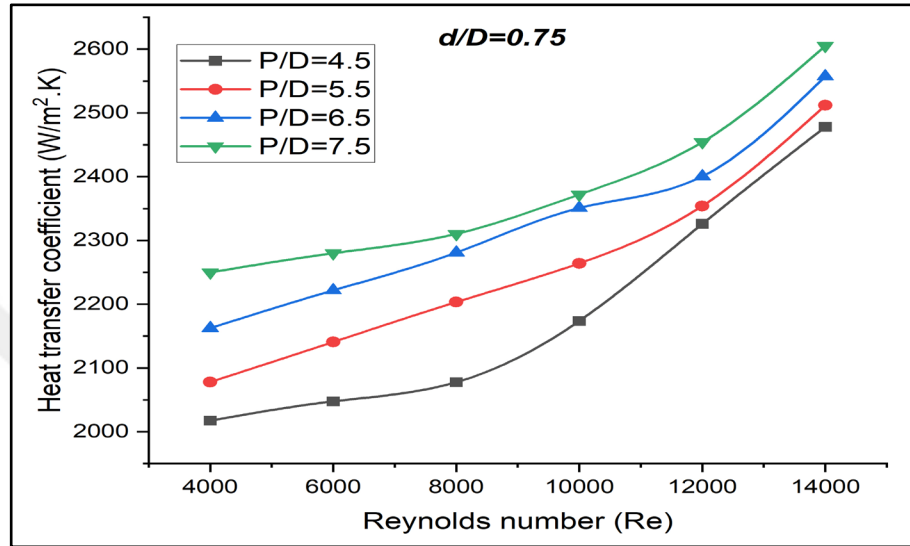


Figure 4.10. Variation of Heat Transfer Coefficient with Reynolds Number (Re) for $d/D = 0.75$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts

Figure 4.11 presents the variation of the convective heat transfer coefficient (h) with Re for various axial pitch-to-diameter ratios ($P/D = 4.5, 5.5, 6.5$, and 7.5) at $d/D = 0.85$. The upward trend in the heat transfer coefficient (h) values continues with increasing Re , but the increase is slower and more consistent compared to cases with lower d/D . The h values appear less affected at low and medium Re values, but a clear spike appears at $Re = 14000$, indicating that a certain flow exceeds the velocity and steady-state flow limit. The results confirm that the special loops remain effective, but their performance becomes less distinct as d/D increases. The differences between different P/D values are less uniform in this case, and a saturation effect is evident in the enhancement effect at Re values between 6000 and 12000. The intersection of the P/D curves at $Re \approx 12000$ indicates that the performance of the distributions is convergent. Although $d/D = 0.85$ allows for smoother central flow, it weakens the effect of the loops on thermal barrier layer disruption. At $d/D = 0.85$, the rings remain

effective but with lower heat transfer efficiency than in the previous cases. The best performance was achieved at $P/D = 7.5$ and $Re = 14000$, where $h = 2821.7 \text{ W/m}^2 \cdot \text{K}$, while the lowest was achieved at $P/D = 4.5$ and $Re = 4000$, where $h = 1867.3 \text{ W/m}^2 \cdot \text{K}$. Using this d/D ratio is suitable for applications requiring lower pressure and low power loss, while lower ratios ($d/D = 0.55\text{--}0.65$) are preferable if the goal is to maximize heat transfer.

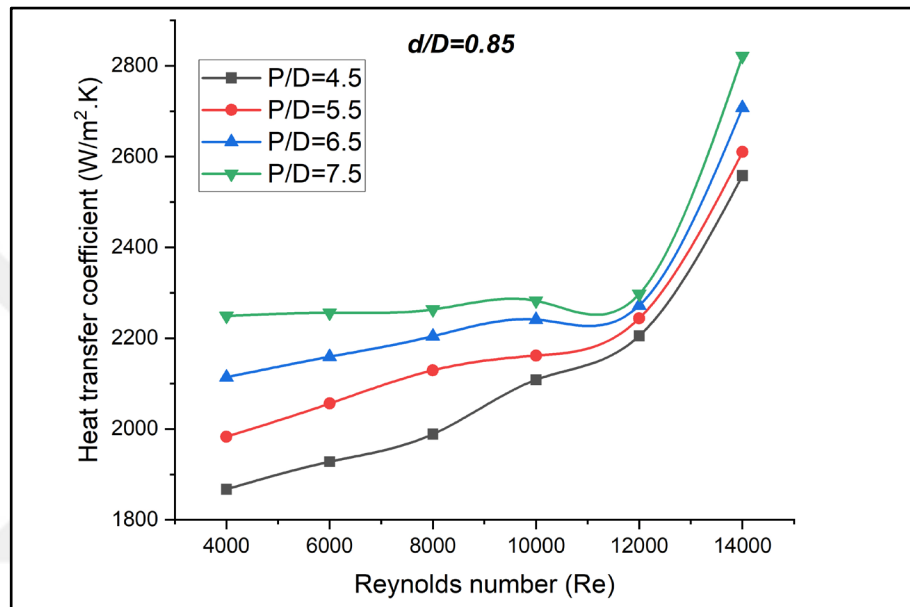


Figure 4.11. Variation of Heat Transfer Coefficient with Reynolds Number (Re) for $d/D = 0.85$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts

4.4. FRICTION FACTOR VARIATION WITH Re AND P/D AT DIFFERENT d/D RATIOS INSIDE A HEAT EXCHANGER WITH RING INSERTS

Figures 12-15 illustrate the variation of the friction Factor (f) with Reynolds number (Re) for four different d/D ratios (0.55, 0.65, 0.75, 0.85) and for four P/D ratio configurations (4.5, 5.5, 6.5, 7.5) inside a heat exchanger tube equipped with special inlet rings. The results presented that the coefficient of friction (f) did not decrease uniformly with increasing Reynolds number (Re), as traditionally expected in smooth pipes. This is due to the effect of special rings that create local turbulence and increase interaction with the wetted surface. Some cases show a rise in the coefficient of friction at low Re due to flow turbulence, followed by a temporary decrease, followed by a

new rise at higher Re due to the formation of new vortices and an increase in the effective wetted surface. The special rings increase the complexity of the contact surface between the fluid and the inner wall, thus increasing the "effective wetted surface." The smaller d/D , the larger the contact area between the fluid flow and the surface, leading to an increase in f due to excessive friction. At $d/D = 0.85$, these effects are reduced, and the flow becomes smoother in terms of resistance. The best hydraulic performance in terms of minimum resistance is achieved when $d/D = 0.75$ or 0.85 and $P/D \geq 6.5$. The rings serve a dual role: enhancing heat transfer and increasing the wetted surface area, but they also increase the coefficient of friction, especially at low speeds.

Figure 4.12 presents the variation of the friction factor (f) with Re for various axial pitch-to-diameter ratios ($P/D = 4.5, 5.5, 6.5$, and 7.5) at $d/D = 0.55$. Figure 4.12 showed the variation of the friction coefficient (f) with the Re at a diameter ratio of $d/D = 0.55$ and for different P/D distributions inside a heat exchanger tube equipped with special inner rings. A sharp increase in the friction coefficient (f) is observed between $Re=4000$ $Re=6000$, followed by a gradual decrease with some undulations after $Re=8000$. This pattern is due to the effect of the special rings that increase interference and vorticity at low flows, while the wetted wall layer begins to settle at higher flows, gradually reducing turbulence. The presence of a clear peak at $Re = 6000$, followed by a gradual decrease in f , is due to the effect of the flow transitioning from a sharp turbulence to a relatively more stable pattern within the tubes. The $P/D = 7.5$ distribution exhibits more stable behavior and lower frictional resistance at all values, indicating that increasing the inter-ring spacing reduces the interaction of the rings with the wetted surface and reduces turbulence. Conversely, the $P/D = 4.5$ distribution produces the highest turbulence at low velocities due to the density of the rings and the increased friction areas with the inner tube surface, which creates high flow resistance and increases f . The $P/D = 7.5$ distribution is the best at reducing the friction coefficient within the heat exchanger at $d/D = 0.55$, especially at high flows ($Re \geq 10000$), making it an excellent candidate for systems that require good heat transfer efficiency with the lowest possible frictional losses.

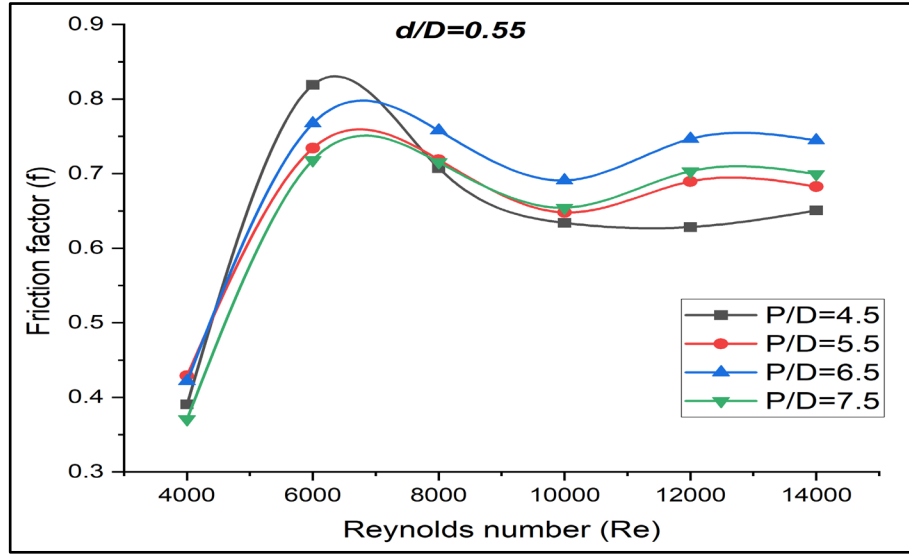


Figure 4.12. Variation of Friction Factor with Reynolds Number (Re) for $d/D=0.55$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts

Figure 4.12 illustrated the variation of the friction coefficient (f) with the Re at a diameter ratio of $d/D = 0.65$ and for different P/D distributions inside a heat exchanger tube equipped with special inner rings. The figure showed that the coefficient of friction (f) increases rapidly as the Re increases from 4000 to approximately 6000, then begins to stabilize or gradually decreases. The highest values of the coefficient of friction are observed at $Re = 8000$ – 10000 for most configurations. Configurations with short axial pitch ($P/D = 4.5$ and 5.5) exhibit higher levels of turbulence than configurations with larger P/D. The smaller the pitch between the rings (small pitch $\rightarrow$ small P/D), the greater the wetted surface length and the effect of turbulence, resulting in improved heat transfer, and increased flow resistance and, consequently, a higher coefficient of friction. At $P/D = 4.5$, the rings are closer together, increasing surface contact with the flow and resulting in a high friction value ($f \approx 0.381$ at $Re = 10000$). At $P/D = 7.5$, the rings are more widely spaced, reducing the effect of local turbulence and reducing friction, especially under high flow conditions. The P/D ratio has a crucial impact on the coefficient of friction, as decreasing it increases the wetted surface area and thus increases friction. The best balance between reducing friction and increasing heat transfer is achieved at $P/D = 6.5$ in this figure.

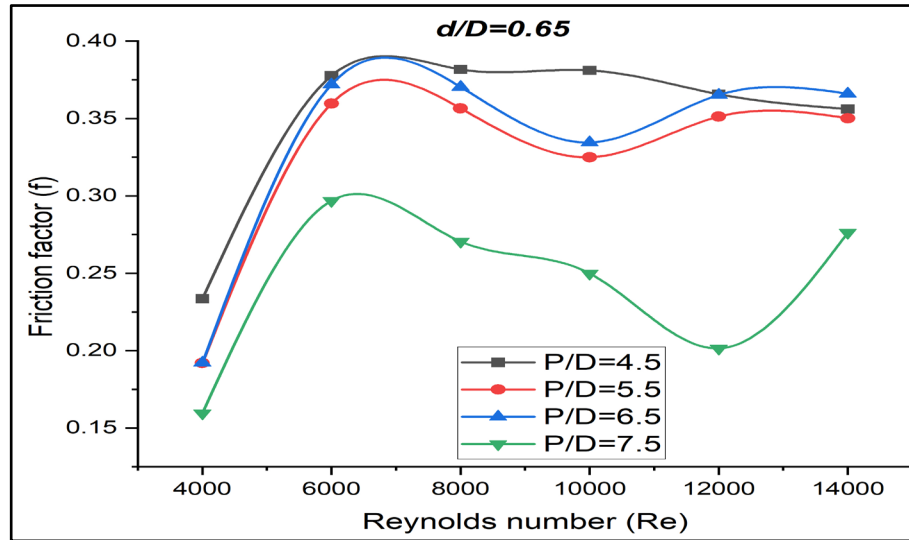


Figure 4.13. Variation of Friction Factor with Reynolds Number (Re) for $d/D = 0.65$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts

Figure 4.14 showed the change in the friction coefficient (f) at the ratio $d/D = 0.75$ inside a heat exchanger tube equipped with special rings, using different configurations of the ratio P/D. The figure showed that the coefficient of friction initially increases rapidly with increasing Re until it reaches a peak, then begins to decrease slightly or stabilize at higher values. The change in the coefficient of friction (f) values is relatively small compared to previous values at lower d/D ratios, indicating relatively stable flow inside the tube at this ratio. The values are very similar at $Re \geq 10000$, indicating stable flow in this range. Increasing the P/D ratio means more ring spacing, resulting in a relatively smaller wetted area and, consequently, less wall friction. Conversely, decreasing the P/D ratio means more ring spacing, increasing the wetted surface area and raising the coefficient of friction due to shear, separation, and secondary flow effects. At $d/D = 0.75$, the amount of turbulence generated by the rings is moderate, so the "wet surface" effect is less severe than at lower d/D ratios.

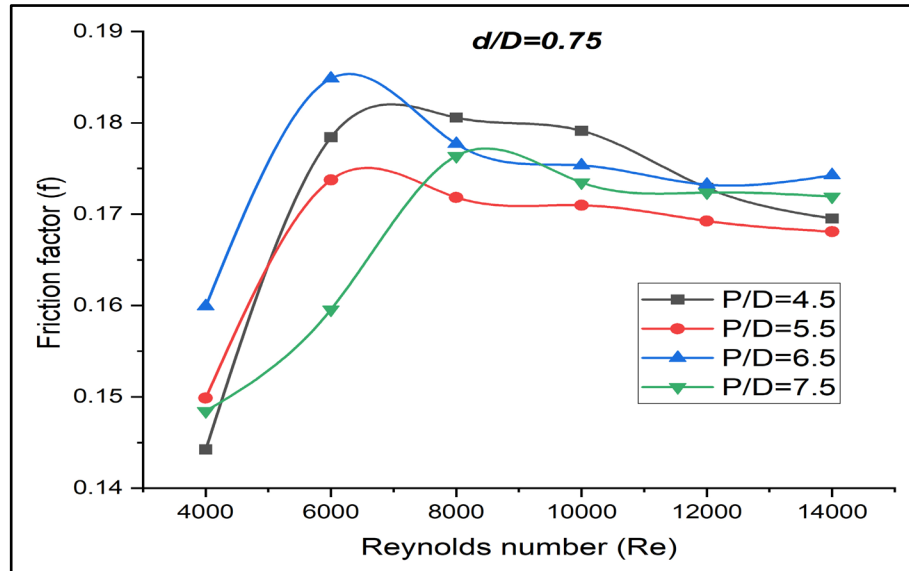


Figure 4.14. Variation of Friction Factor with Reynolds Number (Re) for $d/D = 0.85$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts

Figure 4.15 showed the variation of the coefficient of friction (f) with the Reynolds number (Re) at an inner-to-outer diameter ratio of $d/D = 0.85$ and for different P/D ratios inside a heat exchanger tube equipped with special inner rings. The curves for all ratios show a sharp decrease in the friction coefficient as the Reynolds number increases from 4000 to 6000. Then, a phase of relative stability or a slight increase appears in the range between $Re = 8000$ and 12000 . A sharp increase in values is observed again at $Re = 14000$ in all configurations. It is clearly evident that all configurations tend toward a U-shaped curve, with the lowest point between $Re = 6000$ and 10000 . A P/D value of 6.5 shows the lowest friction coefficient at most mid-Reynolds numbers, indicating good hydraulic efficiency with minimal pressure loss. At a ratio of $d/D = 0.85$, the friction coefficient is strongly affected by increasing Reynolds numbers, especially at the high end. The best balance between friction reduction and thermal stability is often achieved at $P/D = 6.5$ or 7.5 , while $P/D = 4.5$ is more prone to high pressure losses and fluctuating hydraulic performance. A sudden increase at $Re = 14000$ may be due to the reactivation of turbulent flow or a change in the wetted wall distribution due to the effect of special rings. A P/D of 4.5 causes the highest overall frictional resistance and may be undesirable in terms of energy consumption. At $d/D = 0.85$, the wetted surface is very large due to the narrow gap between the ring and the wall and this leads to an early increase in f at low Re. At high

Re, the increase depends on the interaction of the flow with the rings and the turbulence resulting from dense wall contact. Therefore, rings with a small P/D (e.g., 4.5) cause an increase in the wetted surface and lead to higher friction, especially in turbulent flow conditions.

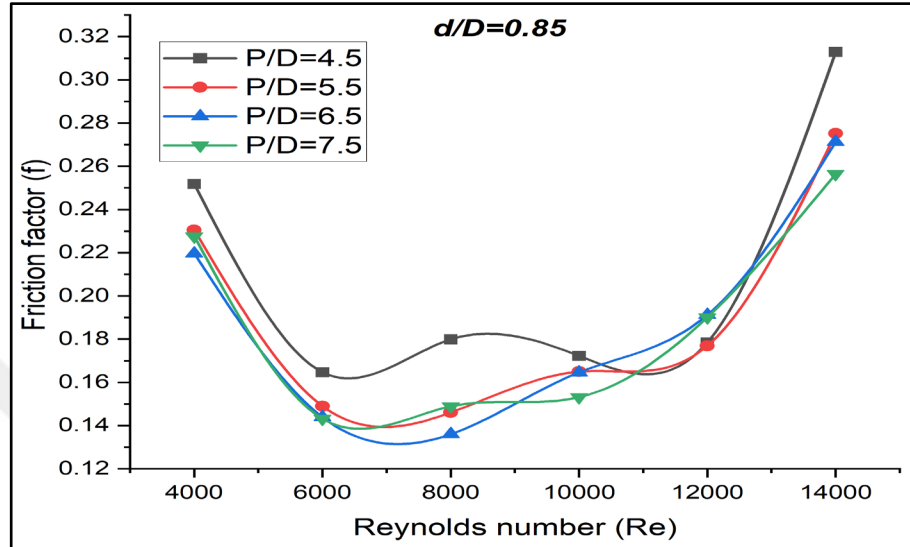


Figure 4.15. Variation of Friction Factor with Reynolds Number (Re) for $d/D = 0.85$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts

4.5. HEAT TRANSFER COEFFICIENT VARIATION WITH RE AND P/D AT DIFFERENT D/D RATIOS INSIDE A HEAT EXCHANGER WITH RING INSERTS

Figures 16-19 illustrate the variation of the Thermal Performance Factor (TPF) with Reynolds number (Re) for four different d/D ratios (0.55, 0.65, 0.75, 0.85) and for four P/D ratio configurations (4.5, 5.5, 6.5, 7.5) inside a heat exchanger tube equipped with special inlet rings. The figures showed that the thermal performance coefficient is optimal at low d/D ratios and low flow velocities. Larger ring spacing (P/D) also improves thermal performance without significantly increasing pressure loss, making it preferable for applications requiring high thermal efficiency and low operating costs. The thermal performance coefficient (TPF) gradually decreases with increasing Reynolds number (Re) in all cases. This is due to the fact that heat transfer enhancement becomes less effective at high velocities compared to increasing pressure

loss. This decrease occurs regardless of the geometric d/D ratio or ring spacing (P/D), indicating that the effectiveness of individual rings decreases at high flow rates. The best thermal performance is achieved at low Re (4000–6000), with the TPF exceeding 2.0 in most configurations. $P/D = 6.5$ and 7.5 showed a balance across most d/D ratios, i.e., good thermal performance with acceptable pressure loss. Performance at $d/D = 0.55$ is the highest for the TPF but is unstable at high Re . At $d/D = 0.85$, the TPF decreases rapidly with Re , reflecting the design limitation in high-velocity flow.

Figure 4.16 showed the relationship between the thermal performance factor (TPF) and the Re at $d/D = 0.55$ for four different P/D ratios (4.5, 5.5, 6.5, and 7.5). The general behavior shows that the TPF gradually decreases with increasing Re , indicating a relative decline in thermal efficiency at higher flows. The findings presented that at $Re = 4000$, all cases achieve relatively high thermal performance factor values. The highest value is at $P/D = 7.5$ (TPF = 2.49278) while the lowest value is at $P/D = 4.5$ (TPF = 2.356). As Re increases to 14000, the TPF values gradually decrease for all cases, reaching 0.77301 for $P/D = 7.5$ and 0.724 for $P/D = 4.5$. $P/D = 7.5$ achieves the best thermal performance at most flow points, especially at the start-up at $Re = 4000$. Although performance decreases with increasing Reynolds number, $P/D = 7.5$ maintains a slight advantage over the other cases. A ring with $d/D = 0.55$ appears to achieve good thermal efficiency in the transition range ($Re < 6000$), but its benefit diminishes with higher flow due to increased viscous effects and greater hydraulic resistance. The special ring configuration at $d/D = 0.55$ is very effective in enhancing thermal performance at low flow velocities, especially at $P/D = 7.5$. However, as flow increases, the thermal benefits gradually diminish, and thermal performance becomes less than ideal compared to flow resistance (pressure loss), necessitating careful balance when selecting the optimal design for heat exchanger applications.

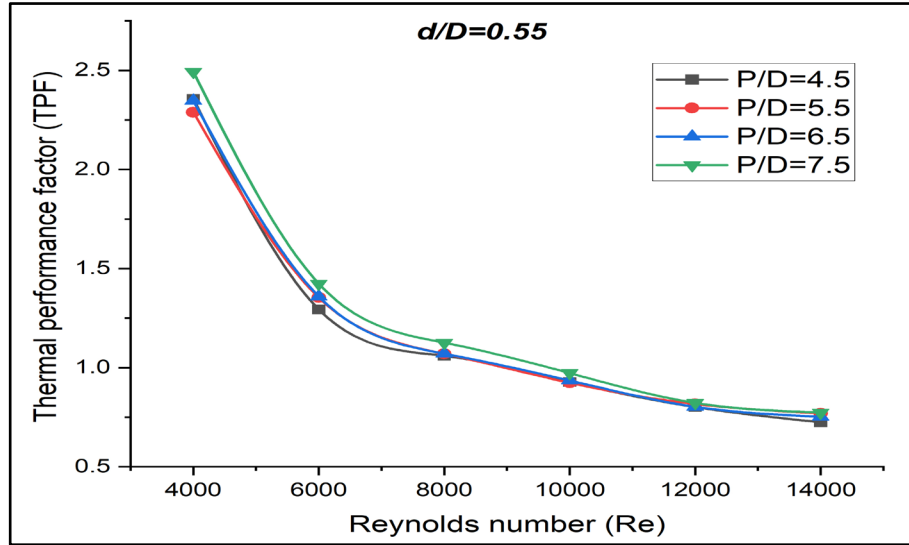


Figure 4.16. Variation of Thermal Performance Factor with Reynolds Number (Re) for $d/D = 0.55$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts.

Figure 4.17 showed the relationship between the thermal performance factor (TPF) and the Re at $d/D = 0.65$ for four different P/D ratios (4.5, 5.5, 6.5, and 7.5). A gradual decrease in the TPF is observed with increasing Re. The differences in thermal performance are most pronounced at low Re values (such as 4000 and 6000). A P/D ratio of 7.5 is the most efficient at all Re values, indicating that increasing the annulus spacing improves flow and reduces pressure. At high Re values (12000–14000), the values converge, indicating that the effectiveness of the thermal performance improvement decreases with increasing velocity. The figure reflects the indirect effect of the wetted surface, as increasing P/D expands the area exposed to heat transfer and reduces the hydraulic resistance. A geometric ratio of $d/D = 0.65$ with $P/D = 7.5$ is optimal for achieving the highest thermal efficiency (TPF), especially in transitional flow (Re from 4000 to 10000). The results show that the benefit of inserting inner loops gradually decreases at high Re, as the values approach Δ for all arrangements.

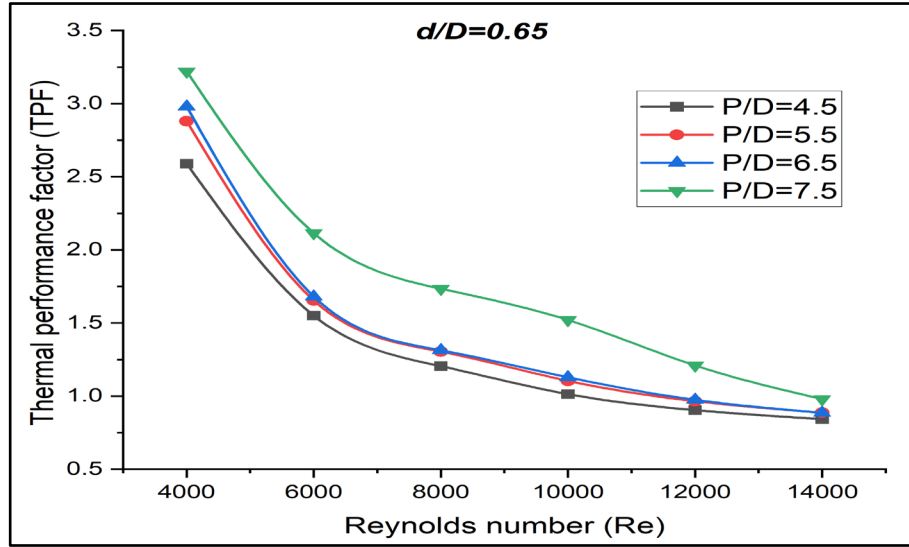


Figure 4.17. Variation of Thermal Performance Factor with Reynolds Number (Re) for $d/D=0.65$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts

Figure 4.18 showed the relationship between the thermal performance factor (TPF) and the Re at $d/D = 0.75$ for four different P/D ratios (4.5, 5.5, 6.5, and 7.5). The TPF decreases steadily as the Reynolds number (Re) increases across all P/D ratios. TPF values are significantly high at low Re (especially at $Re = 4000$) and drop gradually as Re increases. At higher Re values, differences among various P/D configurations become minimal. $P/D = 7.5$ consistently achieves the highest TPF across all Reynolds numbers. Larger P/D ratios are beneficial for heat transfer performance while maintaining reasonable pressure drops and the thermal advantage of large P/D values is more prominent at lower Re (transitional regime) than in the fully turbulent region. The configuration $d/D = 0.75$ with $P/D = 7.5$ delivers the optimal thermal performance, especially at low to moderate Reynolds numbers. While performance differences narrow at high Re, $P/D = 7.5$ remains the superior choice overall, balancing heat transfer enhancement with manageable frictional losses.

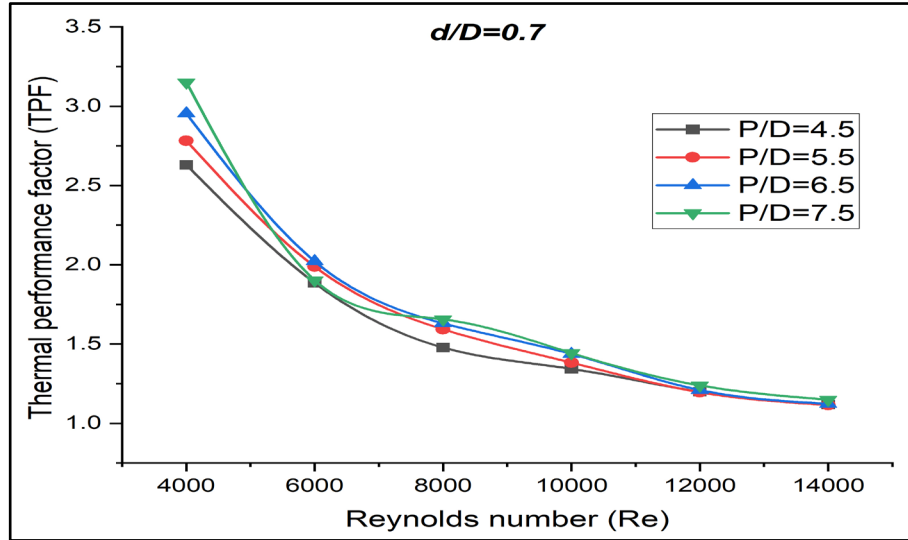


Figure 4.18. Variation of Thermal Performance Factor with Reynolds Number (Re) for $d/D = 0.75$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts.

Figure 4.19 revealed the variation of Thermal Performance Factor (TPF) with Re for $d/D = 0.85$ and various P/D ratios. As Reynolds number (Re) increases from 4000 to 14000, the Thermal Performance Factor (TPF) decreases consistently for all P/D configurations. For $d/D = 0.85$, thermal performance is best optimized at low to medium Reynolds numbers, especially when $P/D = 7.5$ is used. At higher Re (> 10000), although TPF decreases, values still indicate a net positive performance except for $P/D = 4.5$ and 5.5 which fall close to or slightly below 1. Thus, $P/D = 7.5$ with $d/D = 0.85$ is the most thermally efficient and hydrodynamically reasonable configuration for enhancing performance without excessive pressure drop penalties. TPF values remain above 1.0 across all configurations and flow rates, indicating enhanced thermal performance over the smooth tube baseline. The highest TPF is recorded for $P/D = 7.5$, showing the strongest enhancement at low flow due to larger pitch provided $P/D = 7.5$ consistently provides the highest TPF across all Re, especially significant at low Re ($Re = 4000-8000$). The spacing (P/D) strongly influences performance: greater pitch enables enhanced mixing and lower pressure loss, resulting in higher TPF. At higher Re, TPF converges across all configurations, suggesting diminishing return of ring inserts with increasing turbulence. ng more active surface contact and secondary flow generation.

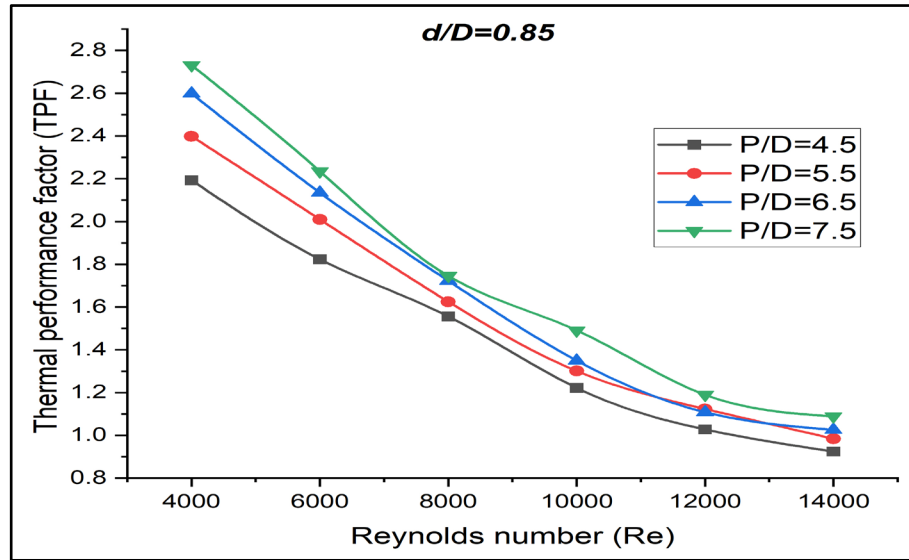


Figure 4.19. Variation of Thermal Performance Factor with Reynolds Number (Re) for $d/D=0.55$ and Different P/D Configurations Inside the Heat Exchanger Tube with Special Ring Inserts

PART 5

PERFORMANCE EVALUATION OF TWISTED TAPES AND NOVEL RING INSERTS WITH HYBRID NANOFLUIDS

This chapter investigates the influence of twist ratio and special ring inserts on the thermal and hydrodynamic performance of a heat exchanger tube. Through detailed numerical simulations, the effects of these geometric modifications are examined using both water and hybrid nanofluids ($\text{Al}_2\text{O}_3\text{--Cu/water}$) as working fluids. Temperature contours, velocity fields, and streamline patterns are analyzed to highlight the enhancement in heat transfer and the changes in flow behavior resulting from these passive techniques. The findings provide valuable insights into optimizing heat exchanger designs for improved thermal efficiency. The chapter further explores the influence of nanoparticle concentration on flow and temperature fields to determine optimal configurations that balance heat transfer enhancement and pressure loss.

5.1. IMPACT OF TWIST RATIO AND RING INSERTS ON THERMAL AND FLOW CHARACTERISTICS OF WATER AND HYBRID NANOFLUIDS

This section investigates the thermal and flow behavior inside a heat exchanger tube equipped with special ring inserts and a fixed twist ratio of $\text{TR} = 5.5$, at a Re of 12,000, with geometric parameters $d/D = 0.75$ and $P/D = 5.5$. Using numerical simulations, the performance is compared for two working fluids: water and an $\text{Al}_2\text{O}_3\text{--Cu/water}$ hybrid nanofluid. The analysis focuses on temperature distribution, velocity contours, and streamline patterns to evaluate the combined effect of ring inserts and swirling flow on heat transfer enhancement and fluid dynamics. The findings emphasize the role of hybrid nanofluids in improving surface wetting and thermal efficiency under this specific geometric configuration.

Figure 5.1 presents a comparative temperature distribution within the heat exchanger tube equipped with special ring and twist tape inserts at a Reynolds number of 12,000, with geometric parameters $d/D=0.75$, $TR=5.5$, and $P/D=5.5$. Two working fluids are used: (a) pure water and (b) a hybrid nanofluid composed of Al_2O_3 –Cu nanoparticles suspended in water. In both cases, the temperature decreases from the wall (red zones) toward the center of the tube (blue/cooler zones), indicating heat transfer from the wall to the fluid. Figure (a) with pure water shows a wider region of higher temperature (red and orange zones), suggesting slower heat absorption due to water's lower thermal conductivity. Figure (b) using Al_2O_3 –Cu/water nanofluid exhibits a sharper temperature gradient and narrower hot region near the wall, indicating enhanced heat transfer capability. This is due to the higher effective thermal conductivity and increased specific heat of the hybrid nanofluid compared to water. The tighter thermal boundary layer in part (b) suggests a higher convective heat transfer coefficient. Quantitatively, the convective heat transfer coefficient (h) for water reaches 3819 $W/m^2 \cdot K$, while for the nanofluid, it increases to 4588 $W/m^2 \cdot K$, marking an improvement of nearly 20%. This enhancement is attributed to the superior thermal conductivity of the hybrid nanofluid, as well as its enhanced surface wetting capability, which ensures closer contact with the tube surface and improved thermal interaction. The Al_2O_3 –Cu nanoparticles improve thermal conductivity via two mechanisms: Brownian motion enhancing micro-convection within the fluid, and higher surface area of nanoparticles increasing energy transport. These effects result in better surface wetting, which enhances fluid contact with the tube surface and promotes faster heat dissipation.

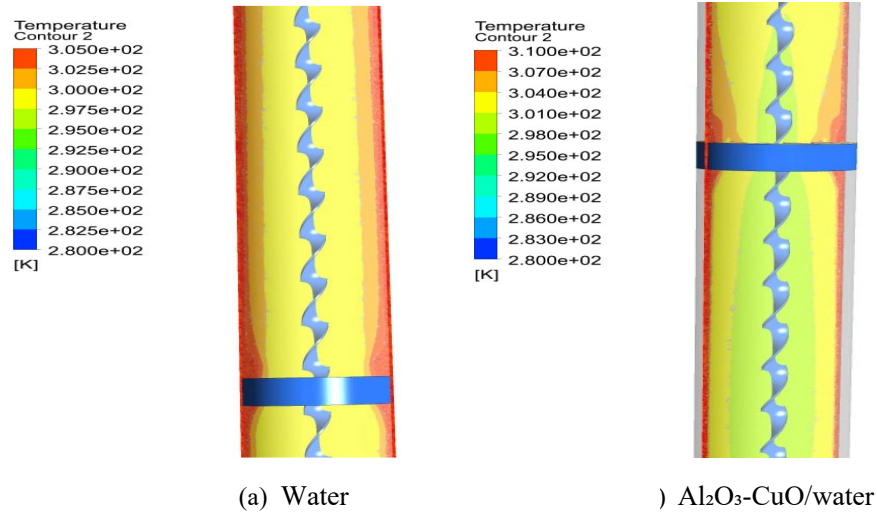


Figure 5.1. Temperature Distributions within the Heat Exchanger Tube with Special Ring and Twist Tap Inserts.at $Re = 12,000$, $d/D=0.75$ and $P/D = 5.5$.

Figure 5.5 compares the velocity contours inside the heat exchanger tube with special ring and twist tape inserts for two working fluids at $Re=12,000$, $d/D=0.75$, $TR=5.5$, and $P/D=5.5$: (a) Water, and(b) Al_2O_3-Cu /water nanofluid. The velocity distribution contours clearly highlight the superior performance of the Al_2O_3-Cu /water nanofluid. The combination of nanofluid properties and mechanical inserts leads to enhanced surface wetting, elevated wall shear, and increased turbulence. These features result in a thinner thermal boundary layer and better heat transfer performance, making this configuration highly effective for thermal system enhancement. In the case of water, the velocity profile shows moderate intensity and recirculation zones downstream of the twisted tape and ring. In contrast, the nanofluid case exhibits significantly more vigorous and widespread velocity contours. The maximum flow velocity reaches nearly 0.6 m/s, compared to around 0.31 m/s for water. This substantial increase is due to the enhanced thermal and momentum diffusivity of the Al_2O_3-Cu /water hybrid nanofluid.

For water, the velocity distribution is characterized by a maximum velocity of approximately 0.31 m/s, observed centrally along the core of the tube, moderate velocity gradients near the wall, showing limited radial mixing. The contour shows localized acceleration near the tape edges, but the effect decays quickly toward the wall, indicating incomplete surface wetting. The twist tape and rings induce flow

separation and recirculation zones, especially downstream of the rings, promoting partial mixing but with uneven distribution. Regions near the wall remain at relatively low velocities (< 0.1 m/s), which implies the thermal boundary layer is thicker and less disturbed.

In contrast, the nanofluid case displays a significantly altered and enhanced velocity field. The maximum velocity exceeds 0.6 m/s, nearly double that of the water case, indicating higher momentum exchange and energy utilization. The swirling flow is stronger and better sustained along the entire tube length. This is due to the enhanced inertia and thermal conductivity of the hybrid nanofluid. The flow is more uniform and symmetrical, reducing stagnant zones and enhancing convective heat transfer. The contour reveals an extended region of high velocity near the tube wall, unlike water, which indicates superior surface wetting. This leads to greater wall shear stress and thinner thermal boundary layers. The twist tape in the nanofluid configuration generates robust longitudinal vortices, which continue to interact with the tube wall much farther downstream.

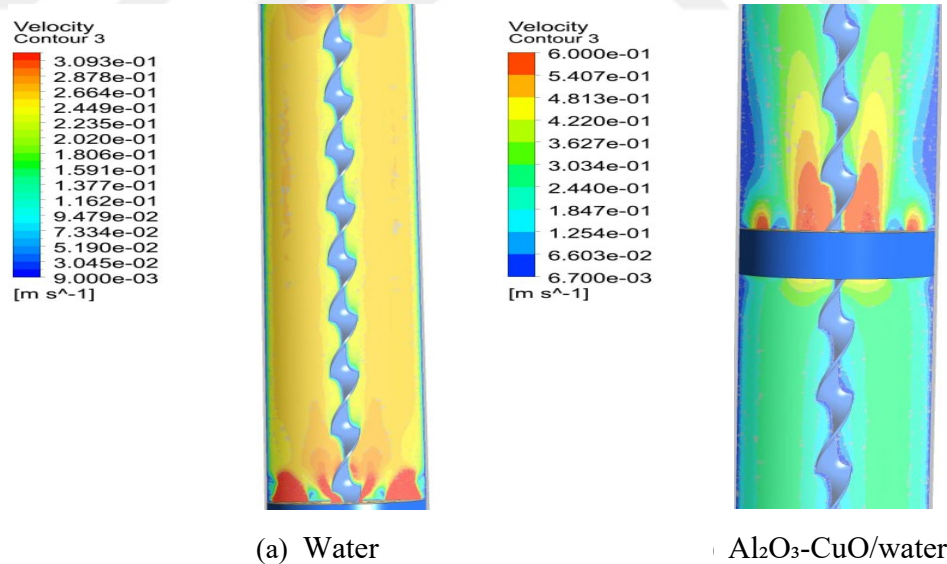


Figure 5.2. Axial Velocity Distribution within the Heat Exchanger Tube with Special Ring and Twist Tap Inserts.at $Re = 12,000$, $d/D=0.75$, $TR=5.5$, and $P/D = 5.5$.

Figure 5.3, which shows the flow streamline patterns inside a heat exchanger tube fitted with special ring inserts and twisted tape for two working fluids: water and an

$\text{Al}_2\text{O}_3\text{-CuO/water}$ hybrid nanofluid. This figure highlights that $\text{Al}_2\text{O}_3\text{-Cu/water}$ nanofluid, when used with twisted tape and special ring inserts, enhances surface wetting, ensures better flow uniformity, and sustains efficient heat transfer mechanisms. While peak axial velocity is slightly lower than with water, the broader and more uniform velocity distribution near the walls and across the tube length leads to superior thermal and hydraulic performance. In water, surface wetting is partial, and flow stagnation is evident in the lower corner zones near the wall. In contrast, the hybrid nanofluid ensures near-total wall coverage through enhanced turbulence, which is especially important for sustaining high convective heat transfer. This behavior correlates with earlier observations from Figures 5.1 and 5.2 where higher local heat transfer coefficients were recorded for nanofluid, due in part to this improved wall interaction.

For water, the flow streamlines exhibit a regular swirling motion generated by the twisted tape insert, with visible circulatory paths extending along the axial direction. Despite the swirl, the velocity gradient is steeper near the core and weakens near the tube wall, indicating limited surface wetting and boundary layer disruption. The streamlines cluster toward the centerline of the tube, and velocity contours show values up to ~ 0.95 m/s, concentrated in the high-energy core region. The flow near the wall is slower, leading to thermal inefficiencies and reduced heat transfer in peripheral regions.

The flow patterns become more expansive and uniformly distributed With the hybrid nanofluid ($\text{Al}_2\text{O}_3\text{-Cu/Water}$), reflecting increased turbulence and higher momentum exchange across the cross-section. Streamlines demonstrate greater adherence to the wall, suggesting improved surface wetting—a key contributor to better heat transfer. The velocity magnitude, while slightly lower in peak (up to ~ 0.82 m/s), is more uniformly distributed, reducing temperature gradients and enabling more balanced heat dissipation. The fluid's higher viscosity and thermal conductivity promote stronger secondary flow, suppressing dead zones and stabilizing recirculation behind the inserts.

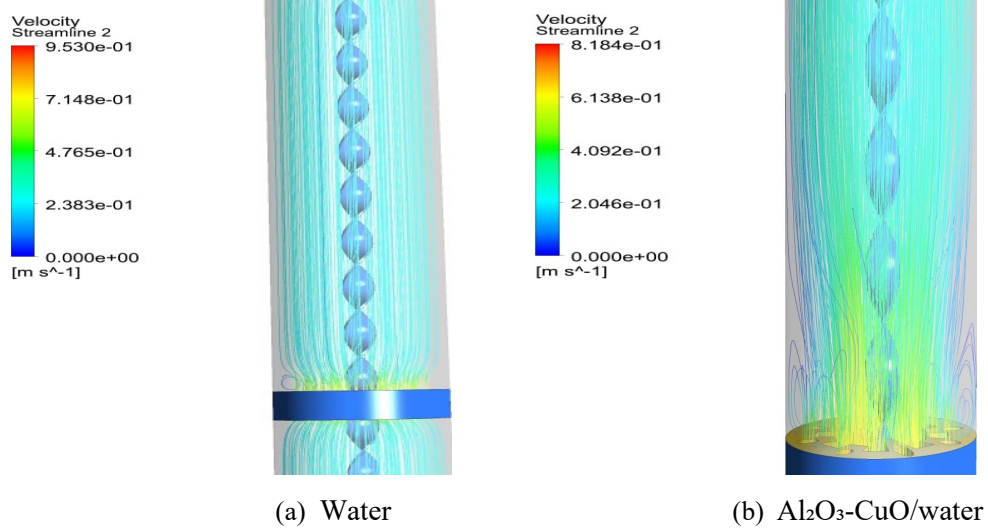


Figure 5.3. Flow Streamline Patterns Inside the Heat Exchanger Tube with Special Ring and Twist Tap Inserts at $Re = 12,000$, $d/D=0.75$, $TR=5.5$ and $P/D = 5.5$.

5.2. IMPACT OF TWIST RATIO AND RING INSERTS ON HEAT EXCHANGER PERFORMANCE

This section explores the impact of integrating special ring inserts and twist tape elements on the thermal and hydraulic performance of heat exchanger tubes. By varying the twist ratio ($Tr = 0-15.5$), the analysis covers key parameters including the Nusselt number, convective heat transfer coefficient, friction factor, and thermal performance factor across a range of Reynolds numbers.

Figure 5.4 illustrates the variation of the Nu with Re for different twist ratios ($Tr = 0, 5.5, 10.5$, and 15.5) inside a heat exchanger tube equipped with special ring inserts. The analysis is conducted at a fixed geometric configuration of $d/D = 0.75$ and $P/D = 5.5$. Across the full range of Reynolds numbers ($4000-14000$), it is evident that adding twisted tapes increases the Nusselt number significantly compared to the base case ($Tr = 0$, i.e., rings only). The twisted tape inserts effectively improve secondary flow generation, and when coupled with ring-induced recirculation zones, create a synergistic effect that enhances convective heat transfer. This is particularly noticeable at higher Reynolds numbers, where turbulence becomes dominant, and the Nu rises steeply. When comparing different twist ratios, a clear inverse relationship is observed

between twist ratio and thermal enhancement: the smaller the twist ratio, the greater the heat transfer performance. This is due to the stronger swirl flow and turbulence intensity induced by lower twist ratios, which promote boundary layer disruption and mixing. However, excessive turbulence can lead to higher pressure drops. Despite this, $Tr = 5.5$ consistently outperforms $Tr = 10.5$ and 15.5 , which show reduced Nu values due to less intense swirling flow.

The base case, $Tr = 0$, which incorporates only the special rings without twisted tape, shows a gradual rise in Nu with increasing Re . The Nusselt number increases from 171.12 at $Re = 4000$ to 206.85 at $Re = 14000$, representing a total enhancement of $\sim 20.9\%$. The configuration with $Tr = 5.5$ demonstrates the highest heat transfer enhancement. At $Re = 4000$, Nu jumps to 250.18, which is nearly 46.2% higher than the base case. As Re increases, this enhancement becomes even more pronounced. At $Re = 14000$, the Nu reaches 377.85, almost 82.6% greater than $Tr = 0$. The steady increase across the Re range indicates the effectiveness of the strong vortex and secondary flow created by tight twisting (short pitch length). With $Tr = 10.5$, a middle-ground twist intensity, the Nu values are significantly improved over $Tr = 0$ but lower than $Tr = 5.5$. At $Re = 4000$, Nu is 210.18, offering a 22.8% increase over the base case. At $Re = 14000$, Nu rises to 321.65, which is about 55.5% higher than $Tr = 0$, but still $\sim 14.9\%$ lower than $Tr = 5.5$ at the same Reynolds number. The largest twist ratio, $Tr = 15.5$, results in the lowest enhancement among all twisted configurations but still maintains better performance than $Tr = 0$. At $Re = 4000$, the Nu reaches 186.18, representing a $\sim 8.8\%$ gain over the base case. As the flow develops, the benefit becomes more noticeable. By $Re = 14000$, Nu rises to 295.16, showing a 42.7% enhancement compared to $Tr = 0$. However, this value is still 21.9% lower than the Nu achieved by $Tr = 5.5$.

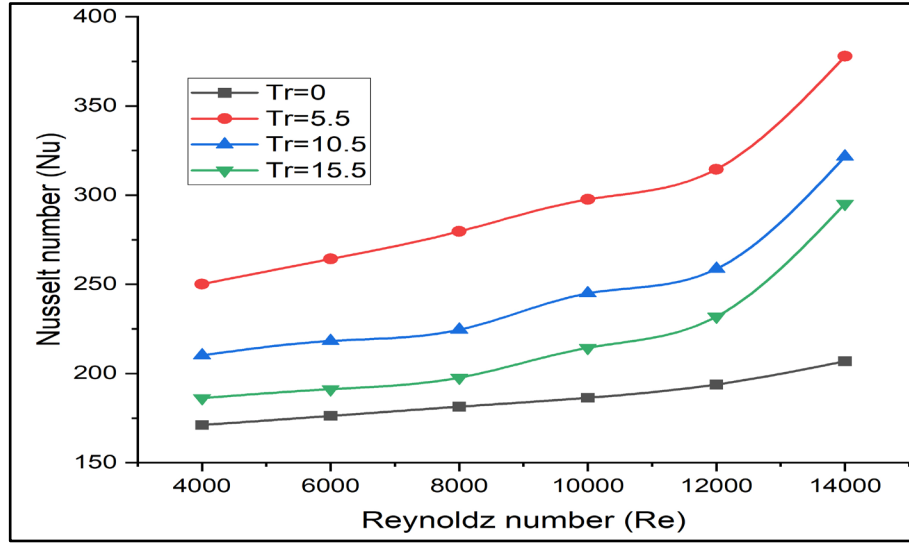


Figure 5.4. Variation of Nusselt Number with Reynolds Number for Different Twist Ratio Inside the Heat Exchanger Tube with Special Ring and Twist Tap Inserts.

Figure 5.4 illustrates the variation of the convective heat transfer coefficient (h) with Re for different twist ratios ($Tr = 0, 5.5, 10.5$, and 15.5) inside a heat exchanger tube equipped with special ring inserts. The analysis is conducted at a fixed geometric configuration of $d/D = 0.75$ and $P/D = 5.5$. The findings illustrate that the convective heat transfer coefficient (h) increases progressively with Reynolds number (Re) for all twist ratio configurations which as higher Reynolds numbers indicate increased flow velocity and turbulence, both of which enhance convective heat transfer by reducing the thermal boundary layer thickness. The data also confirms that even the lowest-performing twisted tape configuration ($Tr = 15.5$) significantly outperforms the plain tube with only ring inserts ($Tr = 0$). This validates the combined benefit of using twisted tapes alongside special ring inserts. Lower twist ratios (like $Tr = 5.5$) introduce a stronger centrifugal force and more effective secondary flows, which disturb the laminar sub-layer and enhance convective transport. As the twist ratio increases ($Tr = 10.5$ and 15.5), the swirling effect weakens, and boundary layer thinning is reduced, resulting in relatively lower h values. Still, all twisted configurations outperform the smooth ring setup, indicating the benefit of combined swirl and ring effect over rings alone. $Tr = 5.5$ consistently leads to the highest heat transfer coefficient at all Reynolds numbers. This is due to the strong swirl flow and aggressive fluid mixing promoted by the tighter twist, which continually disrupts the thermal boundary layer, while $Tr =$

10.5 and $Tr = 15.5$ still outperform the baseline ($Tr = 0$) but to a lesser extent. At $Re = 4000$, the h value for $Tr = 5.5$ reaches $3038.19 \text{ W/m}^2\cdot\text{K}$, which is approximately 46% higher than the baseline case of $Tr = 0$ ($2078.05 \text{ W/m}^2\cdot\text{K}$). As Re increases to 14000, this enhancement becomes even more significant, h for $Tr = 5.5$ reaches $4588.53 \text{ W/m}^2\cdot\text{K}$, compared to $2511.97 \text{ W/m}^2\cdot\text{K}$ for $Tr = 0$, representing an $\sim 82.6\%$ increase. For twist ratios of $Tr = 10.5$ and 15.5 , the heat transfer coefficients remain higher than the $Tr = 0$ case but consistently lower than $Tr = 5.5$. At $Re = 14000$, h values are $3906.08 \text{ W/m}^2\cdot\text{K}$ for $Tr = 10.5$ and $3584.32 \text{ W/m}^2\cdot\text{K}$ for $Tr = 15.5$.

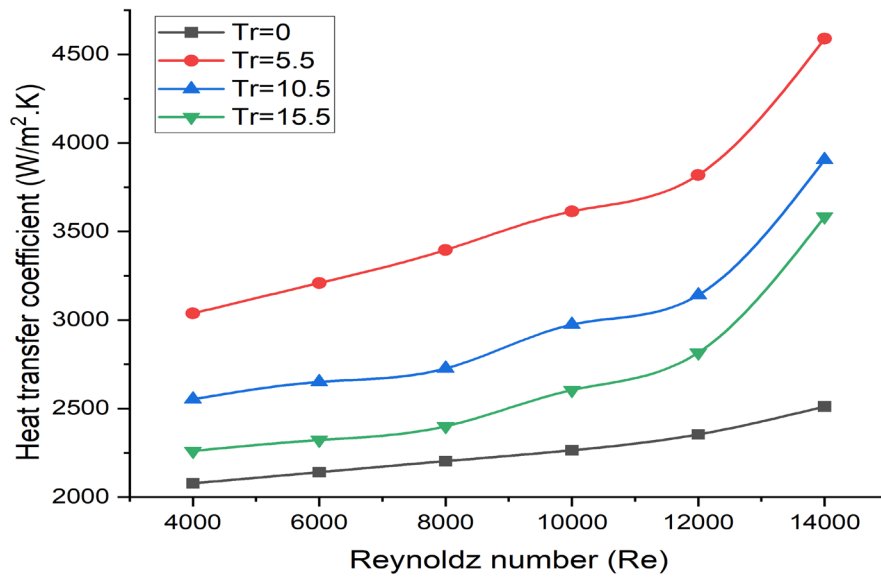


Figure 5.5. Variation of Convective Heat Transfer Coefficient with Reynolds Number for Different Twist Ratio Inside the Heat Exchanger Tube with Special Ring Inserts.

Figure 5.6 presents the variation in the friction factor (f) with Reynolds number (Re) at $d/D = 0.75$, $P/D = 5.5$ with different twist ratios ($Tr = 0, 5.5, 10.5, 15.5$) and reveals important insights into the hydrodynamic behavior inside the heat exchanger tube enhanced with special ring inserts and twisted tapes. The findings present that the maximum friction factor observed 0.1838 at $Tr = 5.5$, $Re = 6000$, and the minimum friction factor 0.1681 at $Tr = 0$, $Re = 14000$. $Tr = 5.5$ offers the highest surface wetting, ideal for applications prioritizing heat transfer over pressure losses. $Tr = 15.5$ shows least resistance, beneficial in systems where pumping power is a primary concern. The f exhibits a non-monotonic trend with Reynolds number in the presence of twisted tape and special ring inserts. While a gradual decrease in friction factor is generally

observed at higher Reynolds numbers due to the dominance of inertial forces, slight initial increases occur at lower Re. For $Tr = 0$ (no twist), the friction factor shows slight fluctuations with Re, starting at 0.1699 (Re = 4000), peaking slightly at 0.1738 (Re = 6000), and then gradually reducing to 0.1681 (Re = 14000). For $Tr = 5.5$, f increases from 0.1828 (Re = 4000) to 0.1838 (Re = 6000), indicating increased turbulence, then decreases gradually to 0.176 at Re = 14000. This implies that while twisting enhances initial turbulence, its effect tapers with rising Re. For $Tr = 10.5$, a similar trend appears: the friction factor drops from 0.1809 to 0.1704, indicating better stabilization of the swirl flow at higher Re. For $Tr = 15.5$, the trend is smoother and more stable, beginning at 0.1710 and reducing slowly to 0.1690, suggesting lower turbulence intensity but steadier flow resistance. The behavior of friction factor also reflects the degree of wall-fluid interaction or surface wetting. High f values at lower Re and low Tr indicate that the fluid remains in more intimate contact with the pipe surface due to slow-moving, chaotic eddies. This enhanced surface wetting leads to improved thermal interaction but increases pressure loss. As Re increases, inertial forces dominate over viscous forces, thinning the boundary layer and reducing direct surface engagement, especially at higher Tr values where the swirl becomes less disruptive. This results in slightly decreased surface wetting and lower f values. Interestingly, for moderate Tr (10.5), surface wetting remains optimized even at mid-to-high Reynolds numbers, balancing thermal performance and pressure drop.

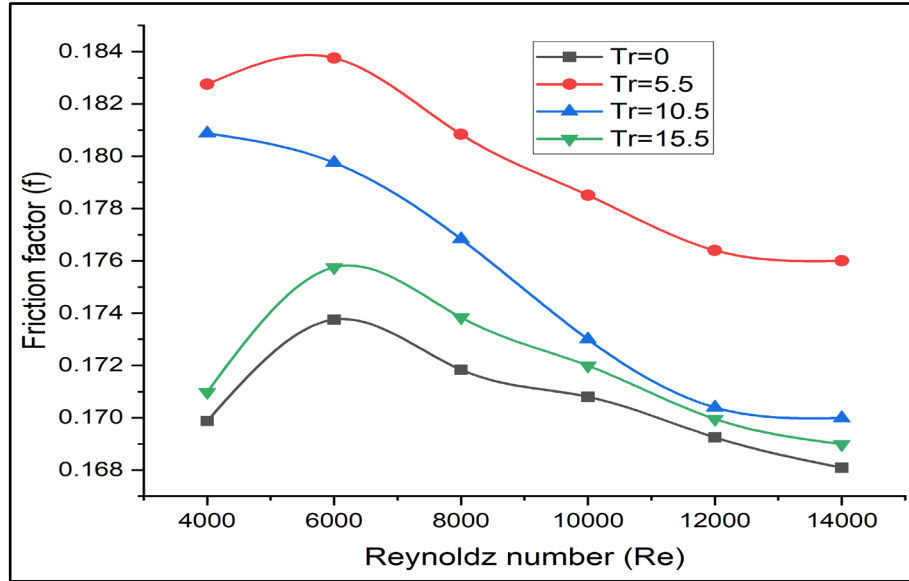


Figure 5.6. Variation of Friction Factor with Reynolds Number for Different Twist Ratio Inside the Heat Exchanger Tube with Special Ring Inserts.

Figure 5.7 shows how the Thermal Performance Factor (TPF) changes with Re for four different twist ratios ($Tr = 0, 5.5, 10.5, 15.5$), at a constant value of $d/D = 0.75$ and $P/D = 5.5$. The findings show a consistent decrease in TPF with increasing Re across all twist ratios. This trend reflects the well-known thermal behavior where thermal performance is more pronounced at lower Re values, due to more dominant viscous and secondary flow effects induced by inserts. However, even at higher Re, the twisted tape inserts configurations continue to show superior performance over the plain tube ($Tr = 0$). $Tr = 5.5$ is the optimal configuration across all Reynolds numbers for thermal performance. Lower twist ratios (tighter twists) produce stronger enhancement but may also contribute to higher pressure drops (as discussed in friction factor analysis). The performance gap between twisted configurations and plain tubes widens at lower Re. Surface wetting and boundary layer renewal are central to the enhanced TPF, particularly for $Tr = 5.5$. At the highest Reynolds numbers tested ($Re = 14000$), even though overall TPF values have dropped, $Tr = 5.5$ still achieves a value of 2.01, compared to 1.73 ($Tr = 10.5$), 1.59 ($Tr = 15.5$), and 1.12 for the smooth tube ($Tr = 0$). This proves that twisted tapes continue to offer thermal benefits even in the turbulent flow regime.

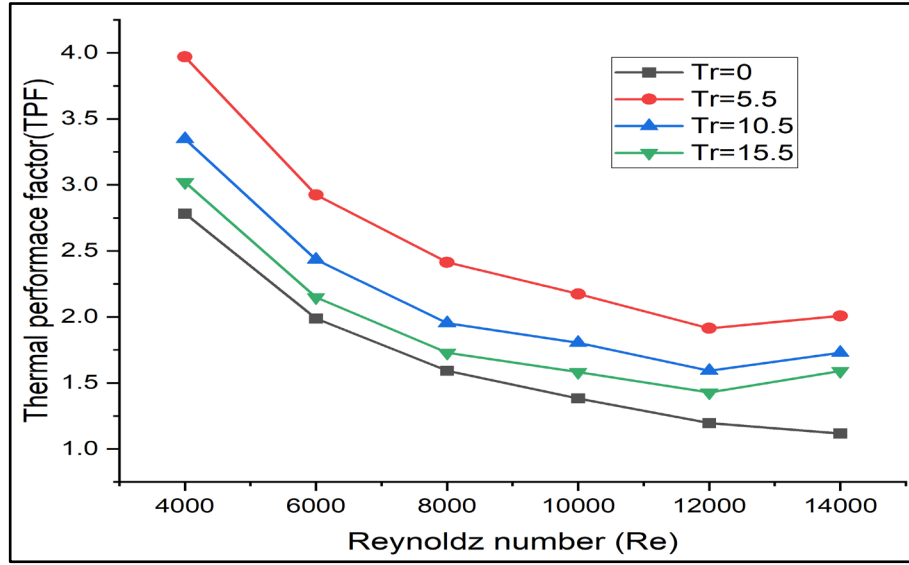


Figure 5.7. Variation of Thermal Performance Factor with Reynolds Number for Different Twist Ratio Inside the Heat Exchanger Tube with Special Ring Inserts.

5.3. IMPACT OF CONCENTRATION OF HYBRID NANOPARTICLES ON HEAT EXCHANGER PERFORMANCE TUBE WITH SPECIAL RING AND TWIST TAP INSERTS

This section explores the influence of varying concentrations of $\text{Al}_2\text{O}_3\text{-Cu}$ /water hybrid nanofluids on the thermal and hydraulic performance of a heat exchanger tube enhanced with special ring and twist tape inserts. By analyzing parameters such as Nusselt number, friction factor, and thermal performance factor across different Reynolds numbers, the study assesses how increased nanoparticle concentration affects heat transfer enhancement and flow resistance.

Figure 5.8 illustrates the impact of hybrid nanoparticle concentration (ϕ) on the Nu in a heat exchanger tube equipped with special ring inserts and twist tape inserts at a twist ratio (Tr) of zero and 5.5, with fixed geometric parameters of $d/D = 0.75$ and $P/D = 5.5$. It is clear that the use of hybrid nanoparticles at increasing concentrations within a tube equipped with special rings and twisted ribbons significantly improves thermal performance (Nu), especially under high turbulence conditions. However, attention should be paid to the potential for increased pressure loss when using high concentrations, which require greater pumping. The Nu increases significantly with

increasing Re for all concentrations. Nu also increases with increasing hybrid nanoparticle concentration (ϕ) at each value of Re . The figure shows that the effect of particle concentration is stronger at higher Re . At $Re = 4000$: Nu increases from 250.18 to 347.61, an increase of approximately 39%, while at $Re = 14000$: Nu increases from 377.85 to 503.26, an increase of approximately 33%. The rate of improvement in Nu becomes lower when moving from 1% to 2% concentration compared to the first jumps, indicating a point of diminishing returns. At $Re = 14,000$, nanoparticles achieve the highest thermal efficiency compared to other cases, reflecting the stronger turbulence stimulation in turbulent flow. The interaction between the particle effect and the twisted ring and ribbon geometry generates vortex flows that enhance heat transfer by mixing and breaking the thermal adhesion layer.

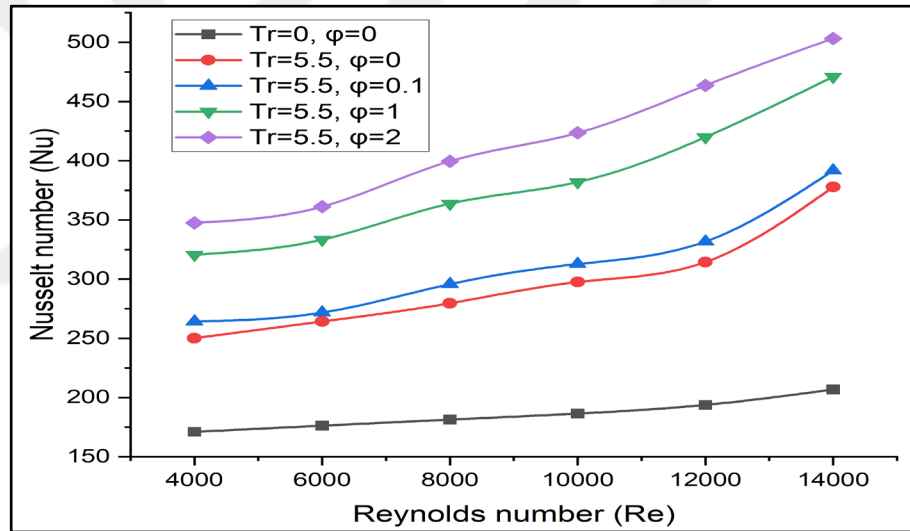


Figure 5.8. Variation of Nusselt Number with Reynolds Number for Different Concentration of Hybrid Nanoparticles Inside the Heat Exchanger Tube with Special Ring Inserts.

Figure 5.9 highlights the influence of hybrid nanoparticle concentration (ϕ) on convective heat transfer performance within a heat exchanger tube enhanced with special ring and twist tape inserts at a twist ratio (Tr) of zero and 5.5, with fixed geometric parameters of $d/D = 0.75$ and $P/D = 5.5$. The use of special rings with twisted bands with a Tr ratio of 5.5, along with hybrid nanofluids, significantly enhances the heat transfer coefficient inside the heat exchanger tube. The effect becomes more pronounced with higher nanoparticle concentrations and higher Re . The findings demonstrate that increasing the concentration of hybrid nanoparticles (ϕ)

significantly improves the heat transfer performance of the system. These improvements are consistent across the entire Re range and are particularly impactful at moderate to high flow rates. The configuration of $d/D = 0.75$, $P/D = 5.5$, and $Tr = 5.5$ proved to be highly effective in harnessing the thermal potential of nanofluids. Across all tested Re, a consistent and significant increase in the Nu is observed with the rise in nanoparticle volume concentration (ϕ). This trend underscores the positive role played by hybrid nanoparticles in augmenting thermal conductivity and promoting convective heat transfer through improved energy transport mechanisms and enhanced boundary layer disruption. At the same Re, increasing the nanoparticle concentration results in a significant thermal improvement due to higher thermal conductivity and increased convection capacity. The heat transfer coefficient increases nonlinearly with nanoparticle concentration, indicating a cumulative thermal effect. At $Tr = 5.5$, it is evident that the use of nanoparticles yields significantly higher thermal performance compared to the case without them ($\phi = 0$), enhancing the effectiveness of this configuration in heat exchangers.

At low Re, specifically $Re = 4000$, the baseline Nusselt number for $Tr = 5.5$ without nanoparticles ($\phi = 0$) is 250.18. With a minimal nanoparticle addition ($\phi = 0.1$), Nu increases to 264.17, a modest gain of around 5.6%. However, with a 1% volume concentration, Nu jumps to 320.74, and further rises to 347.61 at 2%, reflecting a significant 38.9% enhancement compared to the $\phi = 0$ case. As the Re increases, so does the Nu across all concentrations, indicating stronger convective heat transfer due to higher fluid velocities. At $Re = 8000$, Nu increases from 279.65 ($\phi = 0$) to 399.61 ($\phi = 2$), indicating a ~43% improvement. At $Re = 12000$, the maximum Nu reaches 463.52 for $\phi = 2$, which is 47.4% higher than the value at $\phi = 0$ (Nu = 314.52). At the highest Re analyzed ($Re = 14000$), the trend continues, with Nu values escalating significantly with nanoparticle concentration. For $\phi = 0$, Nu is 377.85, while for $\phi = 2$, it rises to 503.26, an improvement of approximately 33.2%.

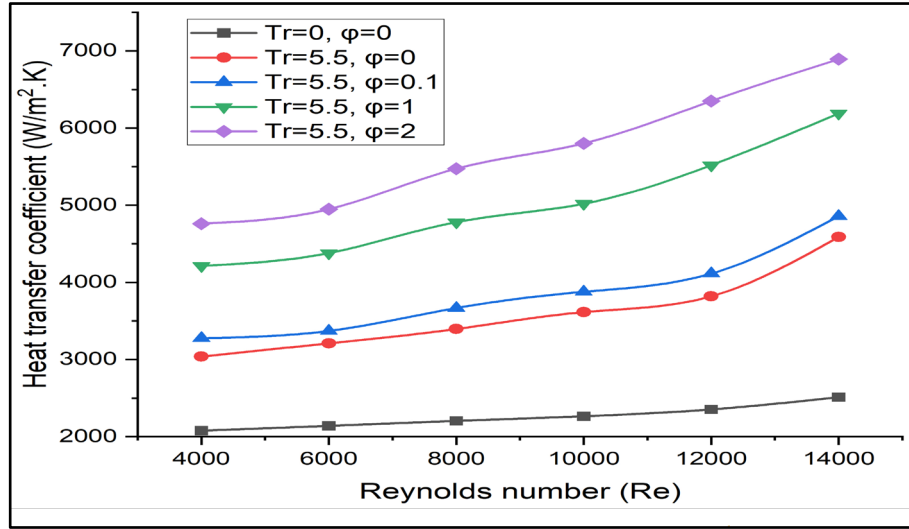


Figure 5.9. Variation of Convective Heat Transfer Coefficient with Reynolds Number for Different Concentration of Hybrid Nanoparticles Inside the Heat Exchanger Tube with Special Ring Inserts.

Figure 5.10 highlights the the variation in friction factor (f) with Reynolds number (Re) for a heat exchanger tube equipped with special ring inserts and twisted tape, operating under different hybrid nanoparticle concentrations ($\phi=0\%, 0.1\%, 1\%, 2\%$) at a twist ratio (Tr) of zero and 5.5, with fixed geometric parameters of $d/D = 0.75$ and $P/D = 5.5$. The introduction of hybrid nanofluids with increasing concentration leads to a consistent but controlled increase in friction factor across all Reynolds numbers. This increase is more pronounced at lower Re and diminishes as Re increases, indicating a saturation-like behavior. The presence of nanoparticles (especially at $\phi = 2\%$) enhances surface interaction, leading to greater wetting and boundary layer disturbance, thereby slightly increasing friction. Hybrid nanoparticles likely cause microscale eddies and localized velocity gradients that contribute to higher f . The enhancement in friction factor is moderate, especially at high Re , which may be an acceptable trade-off given the corresponding boost in heat transfer (as seen in Nu trends).

Across all tested Re (from 4000 to 14000), it is evident that the friction factor increases with nanoparticle concentration. For instance, at $Re = 4000$, the friction factor increases from 0.18275 ($\phi = 0\%$) to 0.19422 ($\phi = 2\%$), reflecting the added resistance to flow caused by higher particle concentrations and the consequent increase in fluid viscosity. The effect of concentration becomes less pronounced at higher Reynolds

numbers, though the trend remains consistent. At $Re = 14000$, the values increase from 0.176 ($\phi = 0\%$) to 0.18474 ($\phi = 2\%$), indicating that while turbulence becomes more dominant at high Re , the particles still contribute to flow resistance. In addition, surface wetting behavior likely plays a role in this trend. The hybrid nanofluids (especially at $\phi = 2\%$) improve surface interaction due to better thermal conductivity and adhesion properties of the nanoparticles. Furthermore, the lowest friction factor values are consistently seen with $\phi = 0\%$ across all Re , confirming that plain water with TT and ring inserts exhibits the least resistance, while the addition of nanoparticles enhances convective mixing at the cost of a moderate pressure drop.

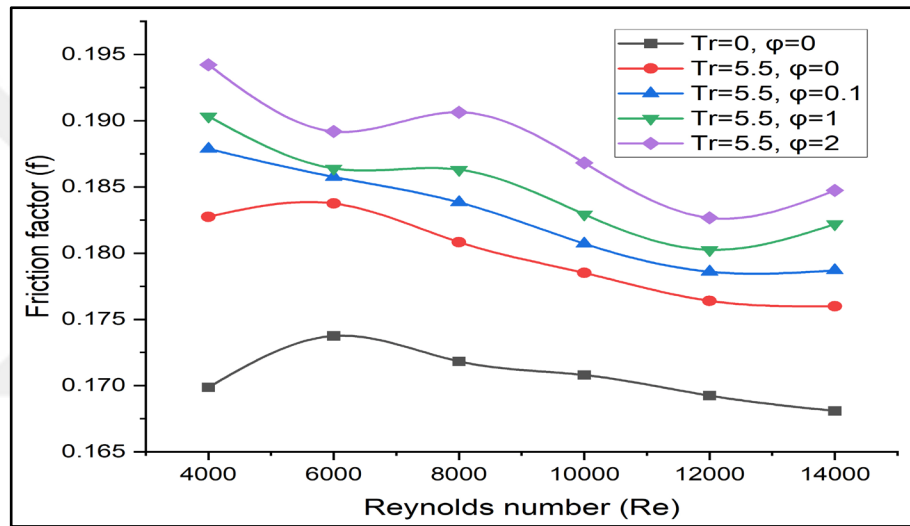


Figure 5.10. Variation of Friction Factor with Reynolds Number for Different Concentration of Hybrid Nanoparticles Inside the Heat Exchanger Tube with Special Ring Inserts.

Figure 5.11 highlights the thermal performance factor (TPF) variations for a heat exchanger tube with special ring inserts and twist tap inserts, considering different hybrid nanofluid concentrations (ϕ) at a twist ratio (Tr) of zero and 5.5, with fixed geometric parameters of $d/D = 0.75$ and $P/D = 5.5$. Integrating twist inserts with hybrid nanofluids (especially $\phi = 2\%$) significantly boosts thermal performance. This synergy is most effective at lower Reynolds numbers, making it ideal for low- to mid-flow industrial applications. The TPF decreases with increasing Re for all configurations. Higher nanoparticle volume fractions (ϕ) lead to greater TPF values, especially at lower Re . Enhancement is more pronounced at low to medium Re ranges (4000–8000), stabilizing at higher Re values. The results demonstrate that the TPF significantly

improves with the inclusion of hybrid nanoparticles compared to the base fluid (water). At all Re tested, the configuration with $\phi=2\%$ consistently achieves the highest TPF values, confirming the substantial benefit of increasing nanoparticle concentration. For instance, at Re=4000, TPF increases from 2.78 in the base case to 4.38 when $\phi=2\%$, reflecting an approximate 57% enhancement in thermal efficiency. This effect remains notable across the entire Re range.

Across all nanoparticle concentrations, the TPF exhibits a decreasing trend with increasing Re. This is primarily attributed to the increasing dominance of inertial forces at higher Re, which can reduce the relative effect of the enhanced thermal conductivity provided by the nanofluid. Despite this decline, the TPF for the nanofluid cases still remains substantially higher than that of water alone. At Re=14000, the TPF for $\phi=2\%$ is 2.24 compared to just 1.12 for the base case, indicating a doubled thermal performance even at high flow rates. As the nanoparticle volume fraction increases from $\phi=0\%$ to $\phi=2\%$, a clear and consistent improvement in TPF is observed. The transition from $\phi=0\%$ to $\phi=0.1\%$ yields a moderate enhancement, while increasing to $\phi=1\%$ and $\phi=2\%$ provides more substantial gains. At Re=10000, for instance, the TPF increases from 2.17 ($\phi = 0\%$) to 2.72 ($\phi = 2\%$), demonstrating the pronounced thermal benefit of nanoparticle addition.

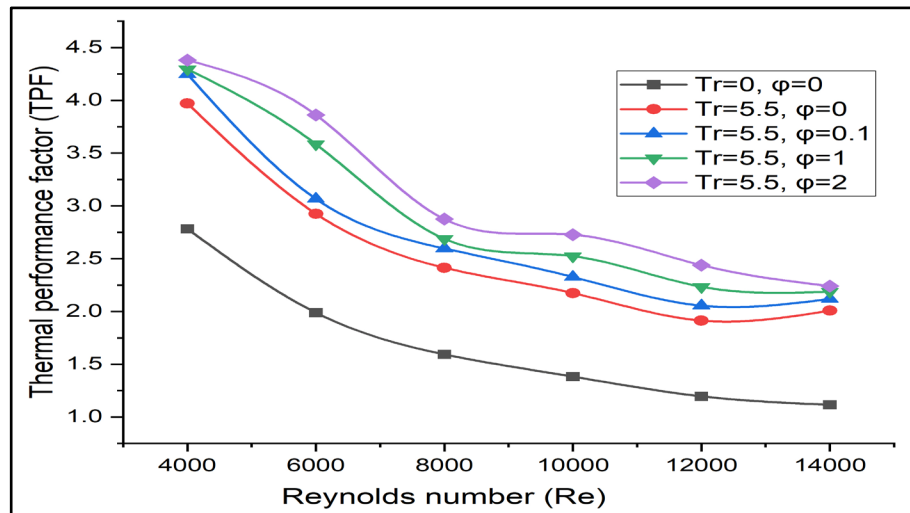


Figure 5.11. Variation of Thermal Performance Factor with Reynolds Number for Different Concentration of Hybrid Nanoparticles Inside the Heat Exchanger Tube with Special Ring Inserts.

PART 6

CONCLUSION AND FUTURE WORK

6.1. CONCLUSION

This study presents a significant advancement in the field of passive heat transfer enhancement by introducing a novel internal ring insert design within a turbulent-flow heat exchanger system. Unlike conventional symmetrical or evenly spaced ring inserts commonly found in literature, the current design features geometrically customized rings with variable diameter ratios (d/D) and spacing (P/D), specifically engineered to intensify localized turbulence and disrupt thermal boundary layers. This approach enabled a more effective convective heat transfer without inducing excessive pressure losses.

What distinguishes this work is not only the innovation in ring design but also the integration of these inserts with twisted tape elements and hybrid nanofluids, creating a hybrid enhancement system. By using $\text{Al}_2\text{O}_3\text{-Cu/water}$ nanofluid, the study leverages both improved thermal conductivity and the disturbance caused by the internal inserts. This integration was evaluated under turbulent flow conditions ($\text{Re} = 4000\text{--}12000$), using validated CFD models to simulate heat transfer, friction factor, and overall thermal performance.

The study demonstrated that the combined use of special ring inserts and twisted tapes, when paired with hybrid nanofluids, leads to a synergistic effect—enhancing heat transfer more than any component could achieve alone. The internal flow characteristics observed in simulations showed increased velocity gradients and thermal mixing near the wall region, leading to higher Nusselt numbers and efficient heat distribution. Overall, the study confirms that smart geometric modification,

when integrated with advanced fluids, can significantly improve heat exchanger performance in solar and industrial applications.

The main outcomes derived from this research are as follows:

- The new inner ring design achieved the highest heat transfer improvement (Nu/Nus) of 5.1, compared to the smooth tube.
- The hybrid system achieved the highest thermal performance factor (TPF) of 4.38, indicating the high efficiency of the proposed configuration.
- The relative friction coefficient (f/fs) ranged from 2.3 to 24.7, lower than some previous studies that recorded values as high as 75, reflecting a good design balance.
- The combined ring design with the twisted bands resulted in more effective internal turbulence than either technique alone.
- The ring geometry was particularly effective in disrupting the thermal boundary layer, leading to enhanced mixing and improved surface heat transfer.
- Increasing the ring diameter (d/D) and decreasing the pitch ratio (P/D) significantly improved the Nusselt number while maintaining the friction factor within a manageable range.
- Increasing the nanofluid concentration to 2% resulted in a significant improvement in heat transfer, with a slight increase in the friction coefficient.
- Decreasing the twist ratio (TR) to 5.5 led to an increase in the Nu value, but was accompanied by a greater increase in f .
- The best balance between thermal performance and pressure was achieved at $TR = 5.5$, $d/D = 0.75$, and a nanoparticle concentration of 0.2%.
- The proposed configurations outperformed most previous studies in terms of TPF, reducing pressure loss by up to 40% compared to similar systems.

These outcomes offer valuable guidance for the design and development of advanced, high-performance heat exchangers applicable in industrial processes, power generation, and solar thermal systems.

6.2. FUTURE WORK

While the present study has delivered valuable insights into hybrid passive enhancement techniques, several avenues remain open for future exploration:

- Exploring other hybrid nanofluid combinations (e.g., $\text{TiO}_2\text{-Ag}$, graphene–water) and different pipe materials could provide new insights into material-performance interactions.
- Future research could employ optimization algorithms (e.g., genetic algorithms, response surface methodology) to determine the best combination of TR , d/D , P/D , and nanofluid concentration for maximum performance.
- Extending the model to simulate transient heat loads, especially in solar applications where input varies with time, would offer more realistic design parameters.

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RESUME

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