Diyala Journal of Engineering Sciences

Journal homepage: https://djes.info/index.php/djes

ISSN: 1999-8716 (Print); 2616-6909 (Online)

# Numerical Evaluation of Using Metal Foam Twisted Tape in Outer Side of a Double-Pipe Heat Exchanger

Hassan Hikmat*, Issam M. Ali

Department of Mechanical Engineering, University of Baghdad, 10071 Baghdad, Iraq

**ARTICLE INFO**

***Article history:***
Received October 27, 2024
Revised February 13, 2025
Accepted February 15, 2025
Available online June 1, 2025

***Keywords:***
Metal foam
Twisted tape
Thermal performance
Heat Exchanger

**ABSTRACT**

This study presents a comprehensive numerical investigation into the thermal and hydraulic performance of a double-pipe heat exchanger equipped with various twisted tape configurations, including metal foam twisted tape, conventional (smooth) twisted tape, and porous media inserts. The analysis is conducted using Computational Fluid Dynamics (CFD) simulations in ANSYS Fluent under steady-state conditions, with air and water as the working fluids. Key parameters such as outlet temperature, pressure drop, Nusselt number, thermal performance factor (TPF), and the pitch-to-height ratio (Pt/W = 3.6, 5.2, and 6.8) are evaluated over a wide range of Reynolds numbers (500 < Re < 13,000). The study explores three distinct cases with different external fin configurations to assess the impact of geometric design on heat transfer enhancement. The results indicate a significant improvement in thermal performance when using twisted tape inserts. Conventional twisted tape enhances heat transfer by up to 33% compared to a plain tube; while metal foam twisted tape achieves a 37% enhancement. Furthermore, metal foam twisted tape demonstrates an 11.2% superior thermal performance over conventional twisted tape, particularly in the laminar flow regime. However, at higher velocities (i.e., higher Reynolds numbers), the difference in performance decreases due to flow choking within the metal foam matrix. Maximum heat transfer augmentation is observed before the transition to turbulent flow, after which the performance gradually declines.

## 1. Introduction

In industrial applications, the exchange of thermal energy between two fluids is facilitated through the use of a double-pipe heat exchanger. This crucial process involves two distinct fluids characterized by disparate temperatures within the heat exchanger, creating an opportune environment for efficient heat transfer between them. The double-pipe configuration serves as a fundamental apparatus for promoting thermal energy exchange, a vital component in numerous industrial processes. This type of heat exchanger enables the seamless transfer of heat from one fluid to another, contributing to the overall efficiency and functionality of diverse industrial systems. The interaction between fluids of varying temperatures within the heat exchanger sets the stage for the dynamic and essential heat exchange phenomena essential for myriad industrial applications [1].

Metal foams, known for their porous structures with high thermal conductivity and compact volume, have garnered attention for enhancing heat transfer in thermal systems. The incorporation of these materials improves thermal performance by maximizing heat transfer through an extensive surface area [2]. Metal foams, with their efficient thermal dissipation, significantly enhance conductive

* Corresponding author.
E-mail address: hekmat9@gmail.com
DOI: 10.24237/djes.2025.18206

heat transfer, leading to increased heat transfer rates [3]. However, it's essential to note that integrating foam structures may reduce fluid velocity, impacting natural convection streams. Despite this, the use of metal foams can promote a more uniform temperature distribution within the system [4].

Aljubury et al. [5] conducted an experimental investigation focusing on the thermo-hydraulic performance of Metal Foam Twisted Tape (MFTT) within a double pipe heat exchanger, with a comparison against Traditional Twisted Tape (TTT). Their objective was to achieve superior thermal enhancement factors compared to TTT under similar operating conditions, primarily assessing Nusselt number (Nu) and Nu/Nuo across various tape-twist ratios and Reynolds numbers. Their findings revealed an inverse correlation between twist ratio and heat transfer enhancement. MFTT surpassed TTT and the unmodified pipe, demonstrating 175% to 230% higher Nu values than TTT, and notably, a more substantial enhancement with Nu values ranging from 275% to 500% higher than the clear pipe. These results underscored remarkable advancements in heat transfer facilitated by MFTT. Aljubury et al. primarily studied the impact of tape-twist ratios and Reynolds numbers on the Nusselt number and thermal enhancement. In contrast, the current study introduces variations in the number of turns and number of fins as additional design parameters, exploring their influence on thermal and hydraulic performance.

Ahmed et al [6] conducted a comprehensive study employing a three-dimensional CFD model in an annular channel, researchers focused on a helical insert wrapped with wire twist ratio of (1.92). The investigation, under single-phase flow conditions Figure (1), explored Reynolds numbers from 200 to 2300. Simulation results demonstrated substantial improvements in both Nusselt number and frictional effects compared to a plain pipe, with Nusselt number augmentation ranging from 1.34 to 2.6 times and frictional effects increasing from 3.5 to 8 times. These findings highlight significant enhancements in heat transfer and, conversely, an elevated pressure drop associated with the wire-wrapped pipe design.

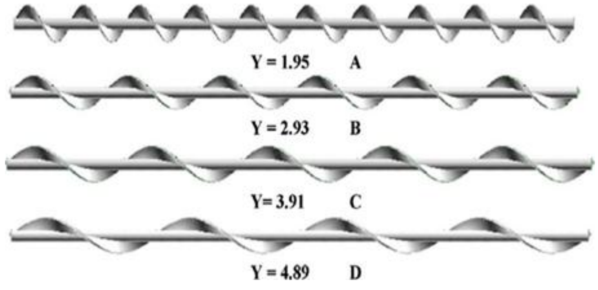



**Figure 1.** Wire-wrapped helical insert [6]

Li et al. [7] introduced the "central hollow narrow twisted tape," a new geometric configuration, to enhance thermal flow and efficiency in pipes under laminar flow. The design featured adjustable parameters-hollow width and clearance-creating a one-sided twisted tape Figure (2). Results indicated a substantial 28.1% improvement in overall heat transfer performance compared to traditional convolute tapes. The central hollow narrow twisted tape demonstrated significant potential for efficiently improving heat transfer and thermal efficiency in laminar flow pipes, offering valuable insights for innovative heat exchanger designs in various industrial applications.

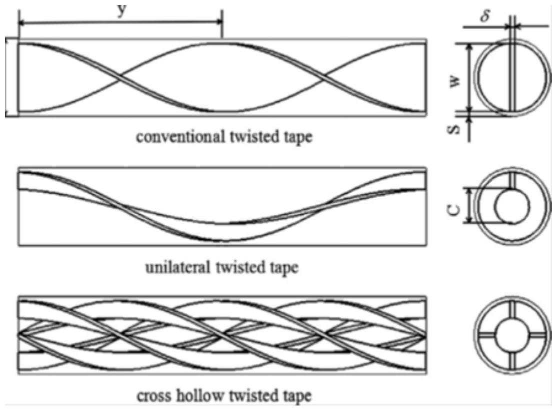



**Figure 2.** Typical, unilateral and cross hollow twisted tapes [7]

In an experimental investigation by Eiamsa-ard and Eiamsa-ard [8] under uniform heat flux conditions, perforated twisted tape inserts were studied with varying parameters (l/m, dia/m, and pitch/m ratios) depicted in Figure (3). Results revealed a noticeable increase in heat transfer rate as pitch/m and l/m ratios decreased, and dia/m ratio increased compared to a plain pipe. The perforated twisted tape outperformed plain pipes and those with conventional convolute strips, exhibiting a superior heat transfer rate of approximately 28.2% compared to a plain pipe and 27.4% compared to pipes with conventional torsion strips. It surpassed smooth pipes with an impressive 86.7% improvement, highlighting the significant potential of perforated twisted tape inserts for efficient heat exchange applications.

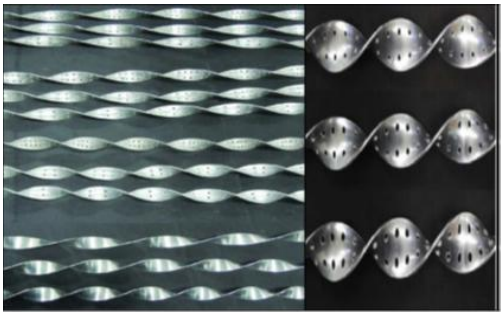


**Figure 3.** Perforated torsion strip with punches in diverse size [8]

According to a review of the literature, in recent years, there have been various attempts to discover various forms of twisted tape insert TTT by varying arrangement and geometry. The current study tries to investigate the significance of using metal foam twisted fins (MFTF) as a twisted tape inset and comparing its thermo-hydraulic performance with traditional twisted tape (TTT) to fill this knowledge gap by means of numerical work based on the performance evaluating factor (PEF). This technique used to storage thermal energy [9] and open-cell metal foam condenser [10] and [11]. The Nusselt number is heightened by the metal foam in a manner directly correlated with the rise in pore density [12]. The investigation goal of the innovatively designed MFTF is to improve the heat transfer rate, which provides a better PEF over those of TTT at the same dimensions and pumping power.

The objective of this study is to numerically analyze the impact of using a metal foam twisted tape on the thermal performance of a double-pipe heat exchanger using the fluent – CFD software. This includes evaluating its performance against conventional twisted tape and smooth tape under different fin configurations and flow conditions using air and water as working fluids. Key parameters such as outlet temperature, pressure drop, Nusselt number, and the coefficient of performance are examined across a range of Reynolds numbers (500 < Re < 13,000) at different cases of tape arrangement (one, two, and three tapes) and (one, two, and three turns). The goal is to identify the optimal conditions and configurations that maximize thermal enhancement while balancing the pressure drop, particularly in different flow regimes.

## 2. Numerical modeling approach

A counter-flow double pipe heat exchanger is employed, constructed from copper with an inner diameter of 21.4 mm and a thickness of 2 mm [13]. In this setup, the hot fluid enters the pipe, while the cold fluid flows outside the pipe. A twisted tape made of copper metal foam is placed in an annular shape around the outer surface of the pipe, which disrupts the flow of the cold fluid, as in Figure (4). The numerical study is conducted using Fluent CFD software for detailed analysis. The solution method is coupled, and gradient of discretization is least squares cell based. The Program Controlled mesh setting is used to let the software automatically generate an optimized mesh based on the geometry and physics of the problem. This approach saves time, reduces user errors, ensures compatibility with the solver, and adapts well to complex geometries. It provides a balance between accuracy and computational efficiency without requiring manual adjustments.

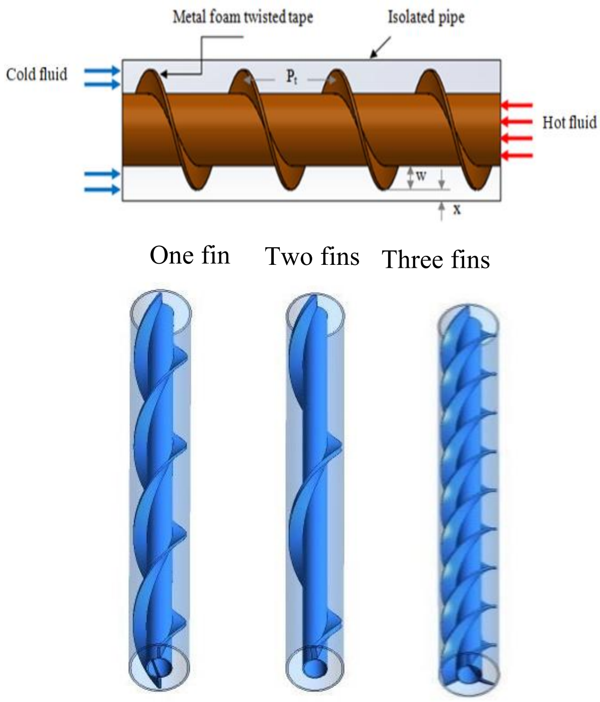



**Figure 4.** Schematic shape of metal foam inserted in an annular pipe

Where: $P_t$ : Pitch of twisted tape, w : tape height, x: distance between the twisted tape and the wall of outer pipe

A heat was applied to pipe by the hot air within the inner pipe, the porosity of the metal foam not exceed (0.75) and thickness of (3 mm), and compared this case with conventional twisted tape and plane pipe. The dimensions of the case study illustrated in table (1) and table (2).

**Table 1:** Values of constant geometrical parameters

| Parameters | Symbol | Value (mm) |
|---|---|---|
| Length of heat exchanger | L | 1000 |
| Diameter of inner pipe with pipe thickness | d | 25.4 |
| Diameter of outer pipe | D | 80 |
| Thickness of twisted tape | t | 3 |
| tape height | w | 25 |
| distance between the twisted tape and the wall of outer pipe | x | 2.8 |

**Table 2:** Values of variable parameters

| Parameters | Symbol | Value |
|---|---|---|
| Number of twist turn | Nt | 1,2,3 |
| Number of fins | Nf | 1,2,3 |
| Type of fin insert | Tf | Metal foam, typical |
| Reynolds number range | Re | 500-13000 |

## 3. Governing equations

The equations governing the issue of flow with heat transfer as in the case of heat exchangers represent the basic equations for fluid flow and the laws of conservation of mass [14], energy [15], and momentum [16].

The conservation of mass: -

$$\frac{\partial}{\partial Z}(\bar{u}) + \frac{1}{r}\frac{\partial}{\partial r}(r\bar{v}) + \frac{1}{r}\frac{\partial}{\partial \theta}(\bar{w}) = 0 \qquad (1)$$

where: $\bar{u}$ , $\bar{v}$ , $\bar{w}$ represented average velocity components in the directions r ,$\theta$ ,z.

The conservation of energy is the first law of thermodynamics, which, when applied to the moving fluid element [17]

$$\left\{\begin{matrix} The\ rate\ at\ which \\ energy\ changes\ within \\ a\ fluid\ element \end{matrix}\right\} = \left\{\begin{matrix} Net\ flux\ of \\ heat\ into \\ the\ element \end{matrix}\right\} + \left\{\begin{matrix} Rate\ of\ working\ done\ on \\ the\ element\ due\ to\ body \\ and\ surface\ forces \end{matrix}\right\} \qquad (2)$$

$$\frac{\partial T}{\partial t} + u_r\frac{\partial T}{\partial r} + \frac{u_\theta}{r}\frac{\partial T}{\partial \theta} + u_z\frac{\partial T}{\partial z} = \frac{\dot{q}_g}{cp} + \alpha\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) + \frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2} + \frac{\partial^2 T}{\partial z^2}\right] + \frac{\Phi}{\rho\, cp} \qquad (3)$$

Where the viscous rate of dissipation is

$$\Phi = 2\mu\left[\left(\frac{\partial u_r}{\partial r}\right)^2 + \left(\frac{1}{r}\frac{\partial u_\theta}{\partial \theta} + \frac{u_r}{r}\right)^2 + \left(\frac{\partial u_z}{\partial z}\right)^2\right] + \mu\left[\left(\frac{1}{r}\frac{\partial u_r}{\partial \theta} + \frac{\partial u_\theta}{\partial r} - \frac{u_\theta}{r}\right)^2 + \left(\frac{\partial u_\theta}{\partial z} + \frac{1}{r}\frac{\partial u_z}{\partial \theta}\right)^2 + \left(\frac{\partial u_z}{\partial r} + \frac{\partial u_r}{\partial z}\right)^2\right] \qquad (4)$$

The general momentum equations for three dimensions in cylindrical coordinate are

*In Z direction*

$$\rho\left(\bar{u}\frac{\partial}{\partial z}\left(\bar{u} - \mu_{eff}\frac{\partial \bar{u}}{\partial z}\right) + \bar{u}\frac{1}{r}\frac{\partial}{\partial r}\left(r\bar{v} - \mu_{eff}r\frac{\partial \bar{u}}{\partial r}\right) + \bar{u}\frac{1}{r}\frac{\partial}{\partial \theta}\left(\bar{w} - \mu_{eff}\frac{1}{r}\frac{\partial \bar{u}}{\partial \theta}\right)\right) = -\frac{\partial \bar{p}}{\partial z} + \frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial \bar{u}}{\partial z}\right) + \frac{1}{r}\frac{\partial}{\partial r}\left(r\mu_{eff}\left(\frac{\partial \bar{v}}{\partial z}\right)\right) + \frac{1}{r}\frac{\partial}{\partial \theta}\left(\mu_{eff}\left(\frac{\partial \bar{w}}{\partial z}\right)\right) \qquad \dots\dots\dots (5)$$

*In r direction*

$$\rho\left(\bar{v}\frac{\partial}{\partial z}\left(\bar{u} - \mu_{eff}\frac{\partial \bar{v}}{\partial z}\right) + \bar{v}\frac{1}{r}\frac{\partial}{\partial r}\left(r\bar{v} - \mu_{eff}r\frac{\partial \bar{v}}{\partial r}\right) + \bar{v}\frac{1}{r}\frac{\partial}{\partial \theta}\left(\bar{w} - \mu_{eff}\frac{1}{r}\frac{\partial \bar{v}}{\partial \theta}\right)\right) = -\frac{\partial \bar{p}}{\partial r} + \frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial \bar{u}}{\partial r}\right) + \frac{1}{r}\frac{\partial}{\partial r}\left(r\mu_{eff}\frac{\partial \bar{v}}{\partial r}\right) + \frac{1}{r}\frac{\partial}{\partial \theta}\left(\mu_{eff}\left(r\frac{\partial\left(\frac{\bar{w}}{r}\right)}{\partial r}\right)\right) + \rho\frac{\bar{w}^2}{r} - \frac{2\mu_{eff}}{r}\left(\frac{1}{r}\frac{\partial \bar{w}}{\partial \theta} + \frac{\bar{v}}{r}\right) \qquad (6)$$

*In θ direction*

$$\rho\left(\bar{w}\frac{\partial}{\partial z}\left(\bar{u}-\mu_{eff}\frac{\partial \bar{w}}{\partial z}\right)+\bar{w}\frac{1}{r}\frac{\partial}{\partial r}\left(r\bar{v}-\mu_{eff}r\frac{\partial \bar{w}}{\partial r}\right)+\bar{w}\frac{1}{r}\frac{\partial}{\partial \theta}\left(\bar{w}-\mu_{eff}\frac{1}{r}\frac{\partial \bar{w}}{\partial \theta}\right)\right)=-\frac{1}{r}\frac{\partial \bar{p}}{\partial \theta}+\frac{\partial}{\partial z}\left(\mu_{eff}\left(\frac{1}{r}\frac{\partial \bar{u}}{\partial \theta}\right)\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(\mu_{eff}\left(\frac{\partial \bar{v}}{\partial \theta}-\bar{w}\right)\right)+\left(\frac{\mu_{eff}}{r}\left(r\frac{\partial\left(\frac{\bar{w}}{r}\right)}{\partial r}+\frac{1}{r}\frac{\partial \bar{v}}{\partial \theta}\right)\right)-\rho\frac{\bar{w}\bar{v}}{r}+\frac{1}{r}\frac{\partial}{\partial \theta}\left(\mu_{eff}\left(\frac{1}{r}\frac{\partial \bar{w}}{\partial \theta}+\frac{2\bar{v}}{r}\right)\right) \quad (7)$$

It is difficult to obtain an improvement in thermal performance directly because the increase in turbulence of the fluid flow causes a large loss of energy and an increase in the frictional pressure drop in the pipe, so it will consume additional energy for pumping, so the design process needs to be evaluated to obtain the optimal design, The thermal performance factor is calculated by the following equation [18].

$$\eta=\frac{Nu/Nu_b}{(f/f_b)^{1/3}} \quad (8)$$

Where (Nu) average Nusselt number with use twisted tape inside the pipe and $Nu_b$ is average Nusselt number at bared pipe. The friction factor obtained at smooth pipe and when the twisted tape inserted in the pipe.

$$Nu=\frac{h*D_h}{k} \quad (9)$$

$$Nu_b=\frac{h_b*D_h}{k} \quad (10)$$

$$f=\frac{2}{\rho}\frac{D_h}{u^2}\frac{dp}{dx} \quad (11)$$

$$f_b=\frac{2}{\rho}\frac{D_h}{u^2}\left(\frac{dp}{dx}\right)_b \quad (12)$$

where :- h : surface heat transfer coefficient of total area (pipe and twisted tape) [W/m$^2$.K], $h_b$ : surface heat transfer coefficient of bared pipe [W/m$^2$.K], f : friction factor of outer pipe with twisted tape insert, $f_b$ : friction factor of bared pipe

The focus of the present work is on thermal performance, not just pressure losses. While it is true that adding any geometry typically results in an increase in pressure losses, it is important to consider the simultaneous enhancement in heat transfer. If the increase in heat transfer is greater than the increase in pressure losses, the case can be considered improved, as indicated by a thermal performance factor greater than one. Therefore, the method is still feasible, as the overall thermal performance is the key factor.

## 4. Assumptions

To simplify the study, there are many assumptions proposed to the case study:

a) Constant properties, density and viscosity of the air, and thermal conductivity and specific heat of the pipe material
b) Steady state conditions.
c) The turbulence model proposed to this study is k-ω (SST) which is suit for inserting twisted tape inside pipe.
d) No slip at the wall.
e) Laminar and turbulent flow.

The k-ω (SST) turbulence model is particularly suitable for this study involving a twisted tape inserted into a pipe because it combines the strengths of both the k-ω and k-ε models, making it highly effective for capturing the complex flow dynamics and heat transfer characteristics of swirl-inducing devices like twisted tapes. The k-ω model is well-suited for accurately resolving the flow near the walls, which is critical in heat transfer studies where boundary layers significantly influence performance. Twisted tapes create strong secondary flows and disrupt the boundary layer, and the k-ω formulation effectively captures these effects, ensuring reliable predictions of heat transfer rates and flow behavior. Twisted tapes generate swirl flow and enhance turbulence intensity, particularly in the near-wall region. The Shear Stress Transport (SST) component of the model is designed to handle such flows where shear stresses dominate, providing accurate turbulence modeling in swirl-enhanced configurations. The insertion of a twisted tape can cause flow separation and complex recirculation zones. The SST model is particularly good at predicting these phenomena due to its ability to switch between k-ω and k-ε modes, improving accuracy in regions of adverse pressure gradients [19].

## 5. Boundary conditions

The boundary conditions applied on the case study as shown in the figure (5), can be summarized as follows

1. **Cold velocity inlet**
   - Boundary Type: Velocity inlet.
   - Parameters:
     - Velocity: Specifies the flow rate of cold fluid entering.
     - Temperature: Initial temperature of the cold fluid.
     - Turbulence Parameters: Intensity and hydraulic diameter.
2. **Hot velocity inlet**
   - Boundary Type: Velocity inlet.
   - Parameters:
     - Velocity: Specifies the flow rate of hot fluid entering.
     - Temperature: Initial temperature of the hot fluid.
     - Turbulence Parameters: Intensity and hydraulic diameter.

3. **Pressure outlet**

- Boundary Type: Pressure outlet.
- Parameters:
  - Gauge Pressure: Typically 0 Pa (atmospheric pressure)
  - Outlet Temperature: Calculated based on simulation results.

4. **Wall boundary**

- Boundary Type: Wall.
- Parameters:
  - No-Slip Condition: Zero velocity at the wall.
  - Heat Transfer:
  - Adiabatic (no heat transfer)
  - Conjugate heat transfer (with conduction at inner tube and twisted tape for typical tape)

5. **Porous media boundary**

- Boundary Type: Porous media.
- Parameters:
  - Permeability: Defines flow resistance due to porosity.
  - Inertial Resistance: Accounts for additional resistance.
  - Thermal Conductivity: Specifies heat conduction within the porous medium.

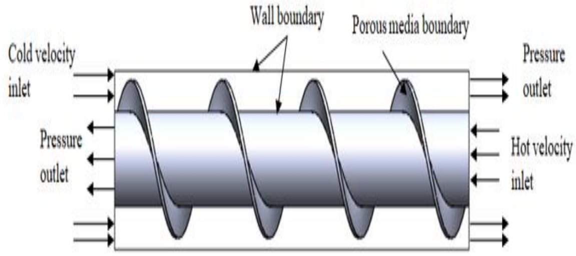
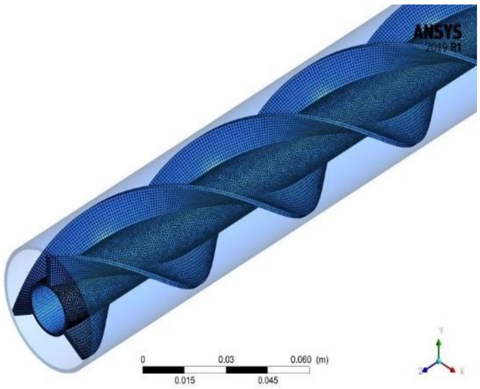


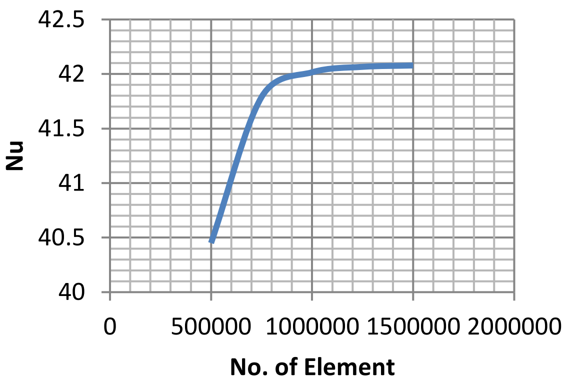

**Figure 5.** Boundary conditions of the case study

## 6. Mesh independency

The solution's accuracy improves as the mesh becomes finer, but beyond a certain point, further refinement does not significantly change the results. Figure (6) illustrates the grid employed for numerical simulations in the double-pipe heat exchanger study and mesh independency. The grid incorporates boundary layers at the inlet and outlet sections, enhancing mesh quality for improved simulation accuracy [15]. Using a turbulence model, even in scenarios covering laminar and turbulent flow ranges, ensures that the simulation can accurately capture the full flow physics, handle the transition to turbulence, and provide reliable results for practical engineering applications. The layout effectively illustrates the structural complexities of the heat exchanger [20], utilizing a controlled-order cell type selected for its capability to provide precise control over mesh characteristics, leading to a refined and well-structured grid [21]. The final mesh consists of 1390311 nodes, after evaluating various mesh sizes, including 520400, 910650, 1218800, 1390311, and 1590400 nodes.




a)




b)

**Figure 6.** Mesh of the case study, a) mesh of the geometry b) mesh independency of Nusselt number

In the analyzed heat exchanger, the internal pipe receives hot fluid at 50 degrees Celsius, flowing at a constant velocity of 3 m/s. Cold fluid is introduced at velocities ranging from 0.2 to 3 m/s, and the external pipe receives this cold fluid at 25 degrees Celsius. Both pipe flows are entirely turbulent, and copper is used for the construction materials of the twisted tapes and pipes [22]. Table 3 provides detailed boundary conditions, specifying temperatures, velocities, and turbulence characteristics for the hot and cold fluids circulating through the heat exchanger.

**Table 3:** Boundary conditions for the numerical study

| Parameter | Symbol | Value |
|---|---|---|
| Inlet temperature of hot flow (internal pipe) | $T_{i)hot}$ [°C] | 50 |
| Inlet temperature of cold flow (external pipe) | $T_{i)cold}$ [°C] | 25 |
| Inlet velocity of hot flow (internal pipe) | $V_{i)hot}$ [m/s] | 3 |
| Reynolds number of cold flow (external pipe) | Re[----] | 500-13000 |

## 7. Convergence criteria

Convergence criteria define the conditions that determine when an iterative process or algorithm has successfully reached a solution. These conditions ensure that the process yields results that are adequately close to the intended goal or target, see Figure (7). The exact criteria vary based on the problem's context, such as numerical optimization, machine learning, or solving differential equations.

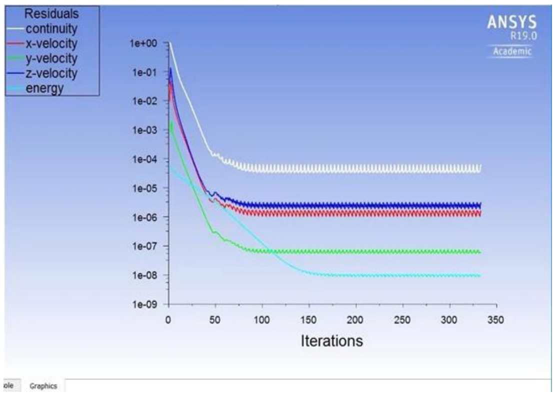



**Figure 7.** Convergence

## 8. Results and discussion

In this investigation, the thermal performance of a double-pipe heat exchanger is analyzed using numerical simulations to evaluate the effect of employing a modified twisted tape fin (MFTF). The study compares two configurations: one utilizing a conventional twisted tape in the outer pipe, and the other using the MFTF design. Both configurations are further compared to a bare pipe (without any enhancement), serving as a baseline to assess the improvements introduced by the modified fin geometry. The analysis considers two working fluids, air and water, to evaluate the thermal performance across varying fluid properties and flow conditions. The primary objective of this investigation is to enhance the heat transfer performance of a double-pipe heat exchanger by introducing a modified twisted tape fin (MFTF) and evaluating its effectiveness in comparison to both a traditional twisted tape and a bare pipe. The study aims to assess the overall impact of the MFTF on thermal and hydraulic performance under different operating conditions using numerical simulations.

*8.1 Validation of the current numerical investigation*

A comparative analysis between the current numerical results and equations from Aljubury et al. [5] was performed to assess simulation accuracy [23], as depicted in Figure (8) and (9). The findings revealed a percentage deviation below 2% for average Nusselt number in metal foam and 11% for typical twisted tape (TTT). Similarly, the average friction factor shows a 12% error for both metal foam twisted tape (MFTF) scenarios, the errors within 10%–15% are typically considered acceptable in heat transfer and fluid mechanics studies [19]. This validation affirms the reliability and fidelity of the numerical outcomes, indicating close agreement with established reference results. The reasonable explanations for the deviation that lead to discrepancy between the numerical and experiments results as follow:

- Modeling Assumptions and Simplifications
- Geometry and Boundary Conditions
- Numerical Method and Grid Resolution
- Experimental Uncertainties
- Heat Transfer and Flow Regimes
- Finite Computational Resources

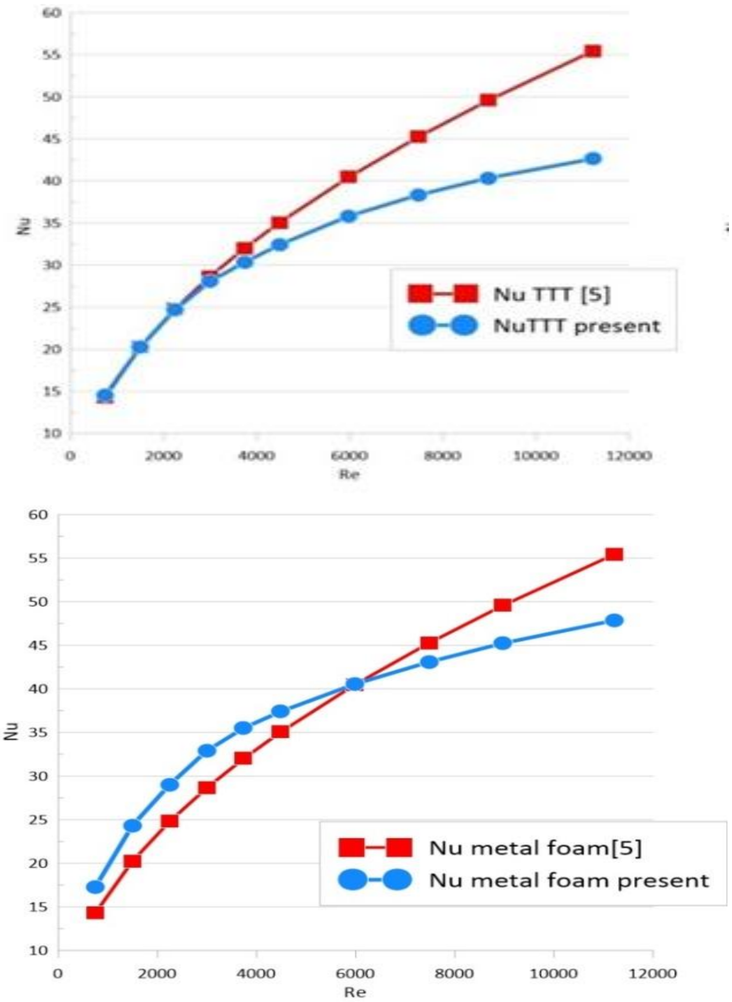



**Figure 8.** validation the Nusselt number of the current study and Aljubury et al. [5] of metal foam and typical twisted tape (TTT).

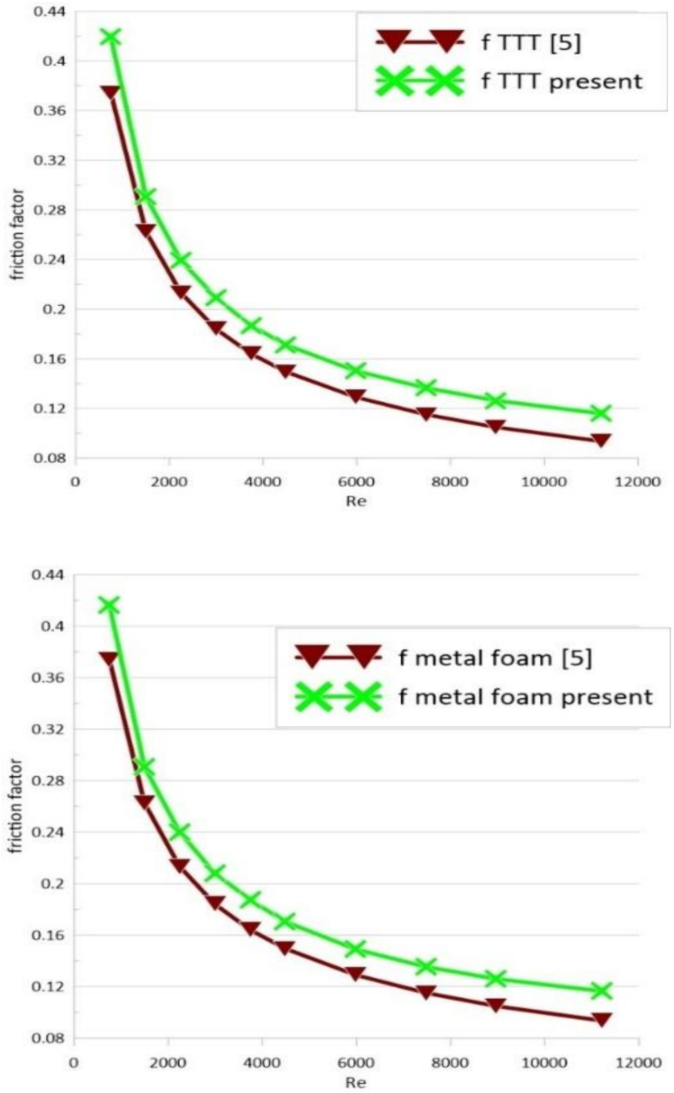



**Figure 9.** validation the friction factor of the current study and Aljubury et al. [5] of metal foam and typical twisted tape (TTT).

*8.2 The effect of adding twisted fins*

In the simulation comparing typical and copper foam twisted tapes for heat transfer enhancement, Figure (10 a) illustrates temperature contours. These contours provide visual representation of temperature distribution within the system. The presence of twisted tape, whether typical or made of copper foam, induces increased turbulence within the flow as shown in Figure (10 b). This heightened turbulence is crucial for enhancing heat transfer as it promotes better mixing of fluid layers, leading to improved thermal exchange between the fluid and the heat transfer surfaces. Additionally, the Nusselt number, which quantifies the convective heat transfer coefficient, is observed to increase significantly with the insertion of twisted tape. This increase in Nusselt number signifies enhanced convective heat transfer rates. The twisted tape serves as a disruptor to the flow, creating vortices and enhancing fluid mixing, which in turn leads to more efficient heat transfer.

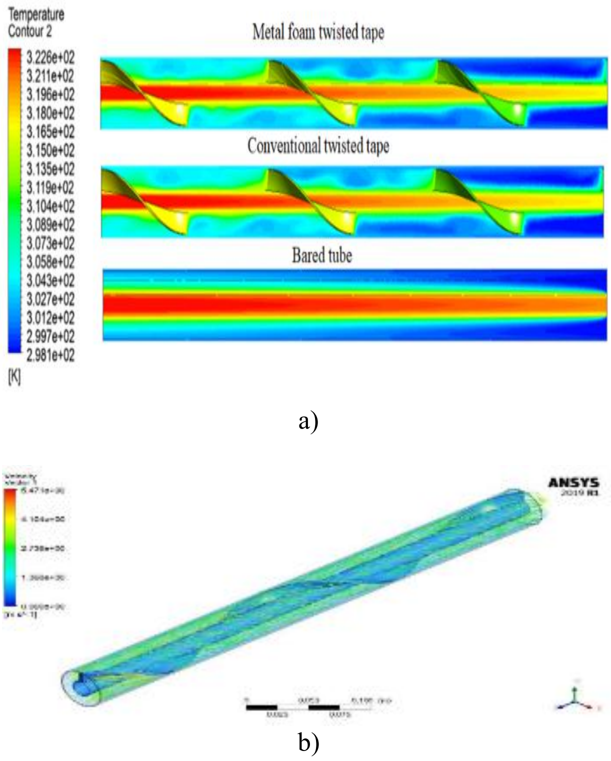



**Figure 10.** a) temperature contour of three cases b) velocity victor

*8.3 The effect of Re parameter for Air*

Figure 11 displays Nusselt number of air variations for one, two, and three fins round the pipe, revealing significant increases in laminar and transient flows. For three turns, typical fins exhibit improvements compared to bared pipe of (75.1%, 107.2%, 134.5%) for one, two, and three fins, while foam metal shows higher Nusselt numbers with increases of (86.2%, 118.4%, 135.6%) for the same configurations due to flow disturbances because the tape works as fin and fluid mixer at the same time with low stagnation flow near the wall.

Figure 12 illustrates the variation of the friction factor with Reynolds number. As Reynolds number increases, the friction factor decreases, given its inversely proportional relationship with the square of velocity according to Darcy's equation (11). The distinct difference between cases MFTF, TTT and the bare pipe is evident due to flow obstruction by the tape consequently increasing the friction factor. (MFT) increases the friction factor because the pore in the metal foam makes the surface rougher. For three turn is the worst of friction factor as compared to the bare pipe (185%, 278%, and 404%) for one, two, and three fins.

Figure (13) depicts thermal performance factor changes across Reynolds numbers for (MFTF) and (TTT). In laminar flow, the factor rises until the early transitional region, turning turbulent due to the twisted tape's geometry. With increasing Reynolds number, the factor decreases due to elevated pressure drop. Comparing one, two, and three fins, optimal performance factors occur in laminar flow of air. For the first and second cases, the two-turn MFTF (1.42 and 1.48) of thermal performance factor, while the one-turn MFTF performs best in laminar and transitional flow in the third case (1.61) of thermal performance factor. The porosity allows fluid to pass through the fins; this can increase the heat transfer efficiency while maintaining a low pressure drop that results in increasing TPF.

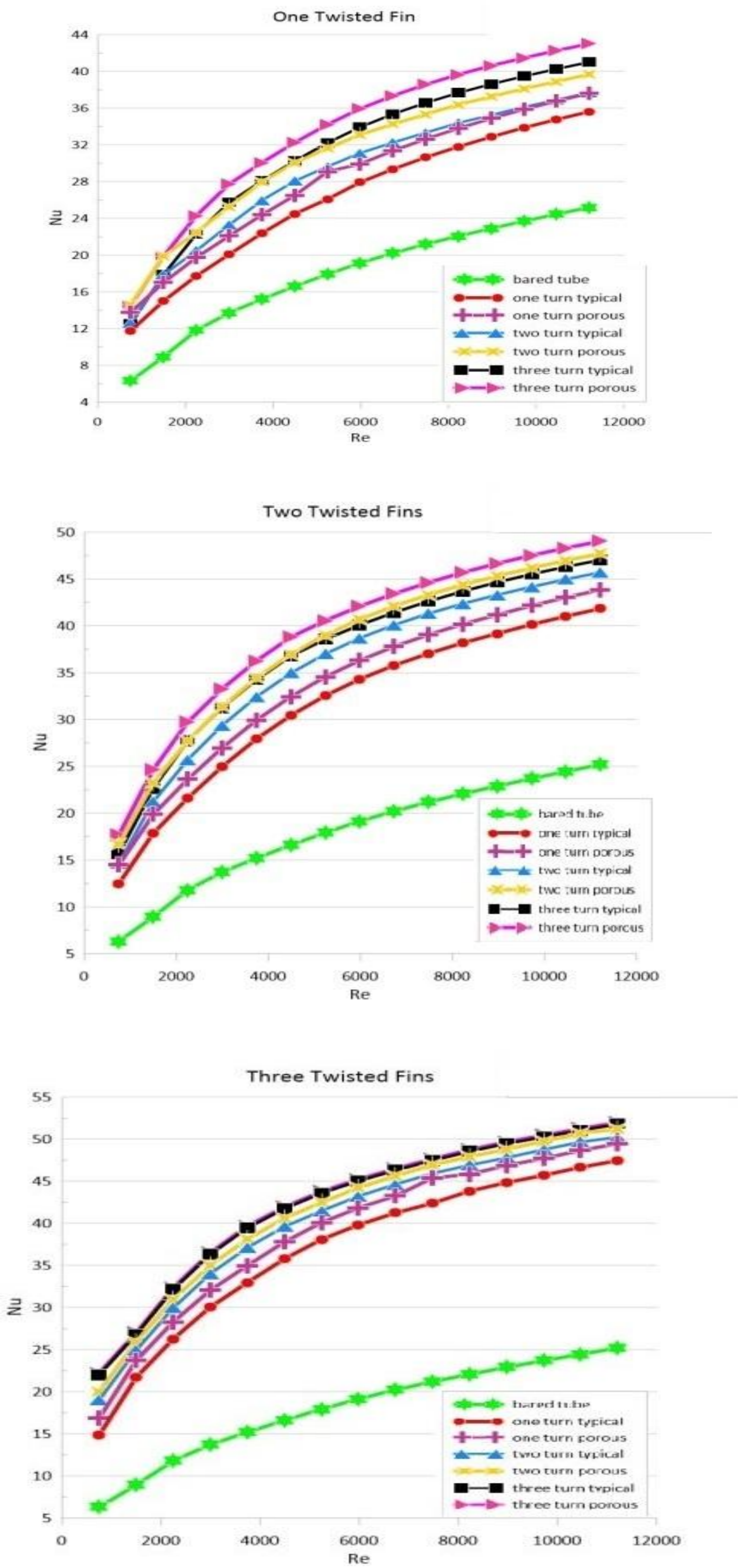



**Figure 11.** Nusselt number of air at different ranges of Reynolds number

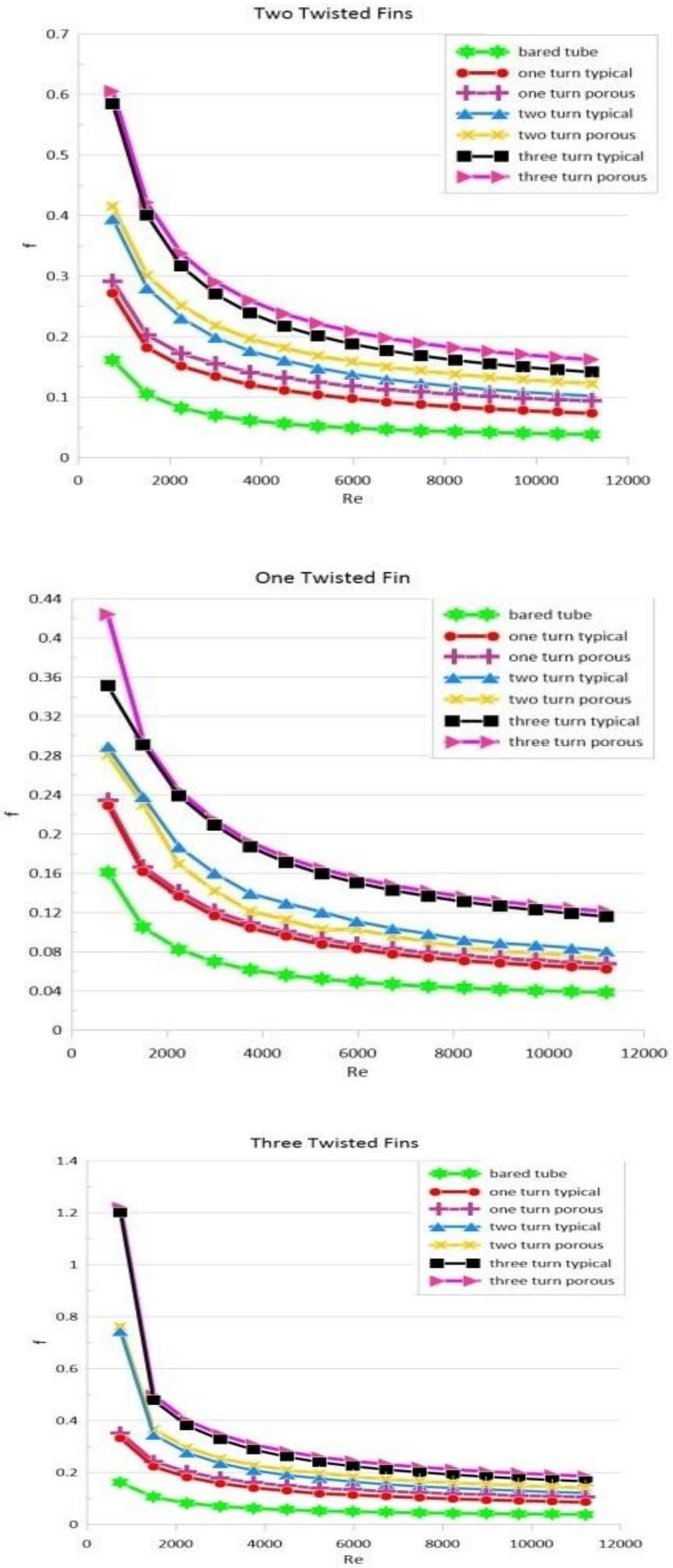



**Figure 12.** friction factor of air at different ranges of Reynolds number

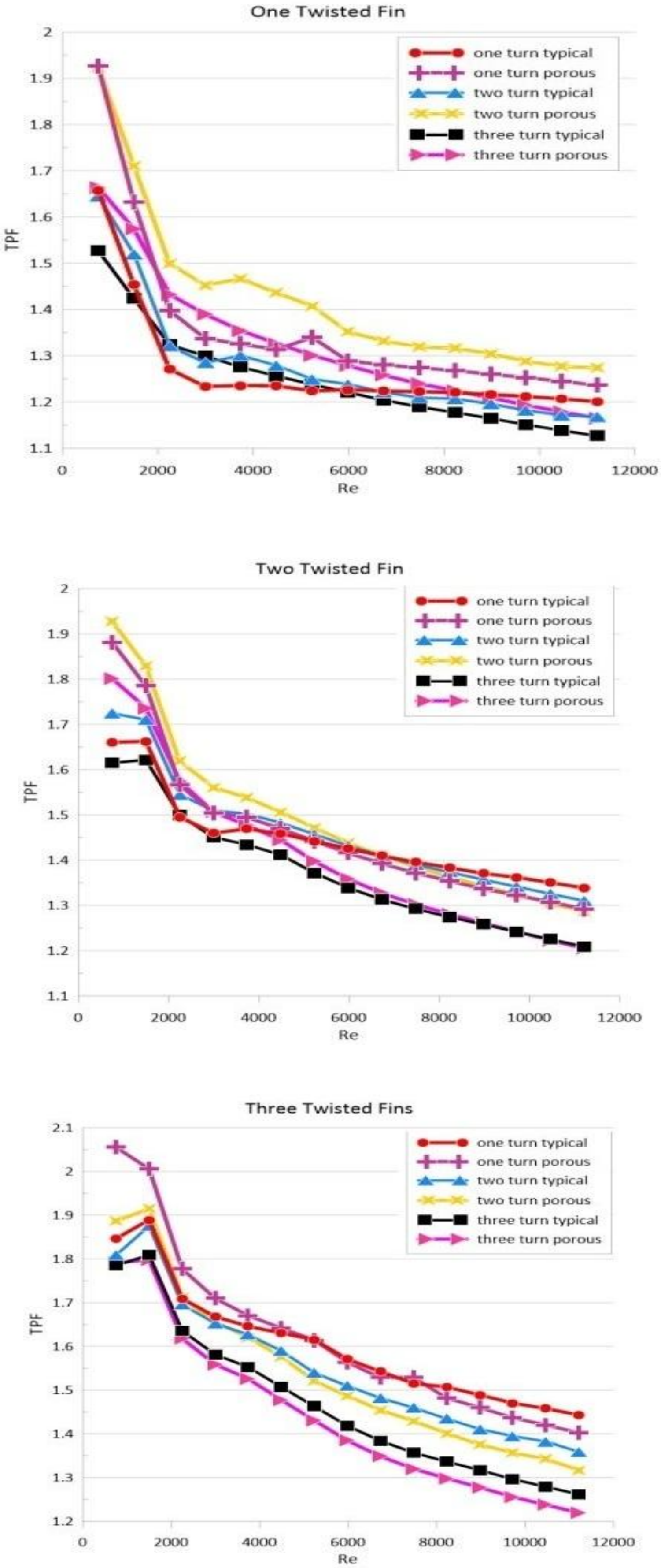


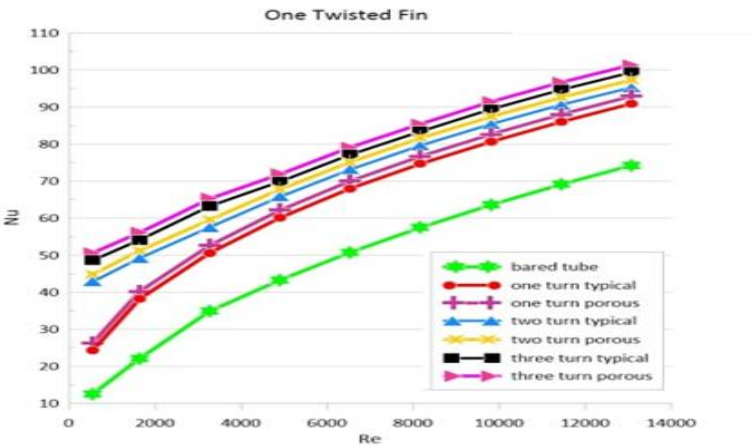

**Figure 13.** Thermal performance factor of air at different ranges of Reynolds number

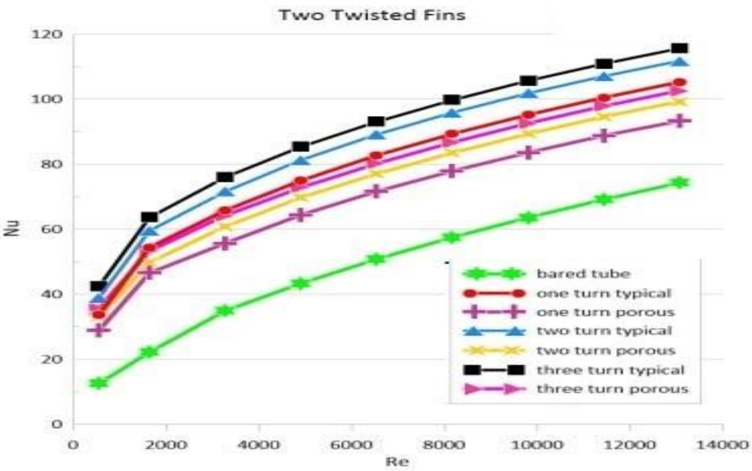
*8.4 The effect of Re parameter for water*

The Figure 14 illustrates the variations in Nusselt number of waters for different numbers of fins (one, two, and three) around the pipe. For three turns, typical fins show enhancements compared to a bare pipe, with improvements (59%, 85%, and 146%) for one, two, and three fins, respectively. On the other hand, metal foam fins exhibit even higher Nusselt numbers, with increases (63%, 60%, 113%) for the same configurations, attributed to flow turbulence caused by the foam structure. The twisted fins (both typical and porous) significantly improve the Nusselt number compared to the bare tube due to enhanced turbulence and secondary flows. Typical fins are more effective than porous fins because of their stronger obstruction of the flow, which induces more intense swirl.

Figure 15 illustrates the friction factor of water for different numbers of fins (one, two, and three) around the pipe. For three turns, it is as high as possible for the first case of the three turns fin foam. However, in the second and third cases, the typical three turns MFTF is the highest due to the large pressure losses that are caused by the vortices. The friction factor increases in the three turns as maximum as compared with bare pipe by (152%, 268%, 379%) for one, two, and three fins respectively, and for metal foam twisted fin by (223%, 267%, 319%) for one, two, and three fins respectively.

Figure (16) shows the Comparing one, two, and three fins of the performance factors occur in laminar and turbulent flow of water. It is noteworthy that in all cases, the typical fin outperforms the foam fin by a margin of (8.5%, 5.7%, 5%) for one, two, and three turns, respectively, when considering a single fin, and (17.6%, 16%, 16.6%) for two fins, and (25.4%, 19.1%, 11%) for three fins. The highest improvement in thermal performance is observed when the flow is water with three fins and one turn. At high Re, flow separation and energy losses due to excessive turbulence become more pronounced in systems with obstructions (like fins). This can decrease the heat transfer efficiency while maintaining a high pressure drop, further reducing TPF.



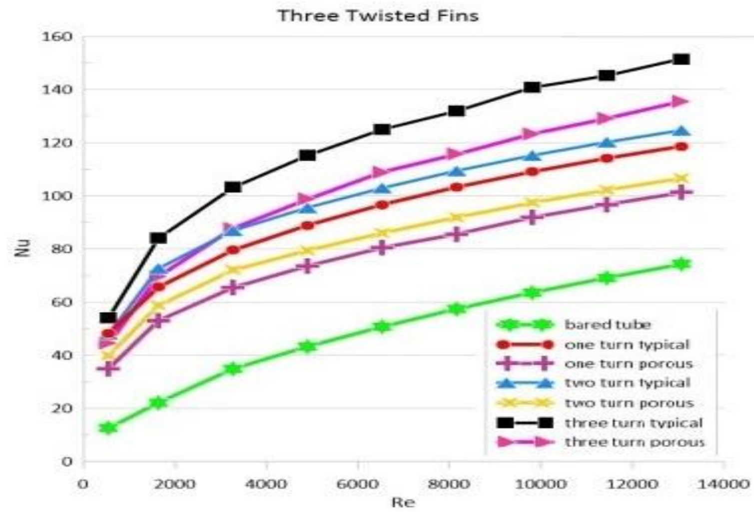
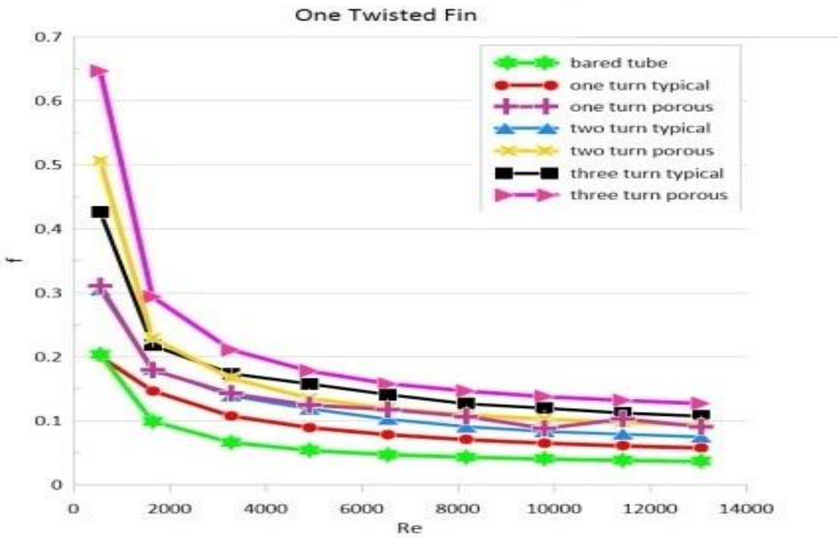
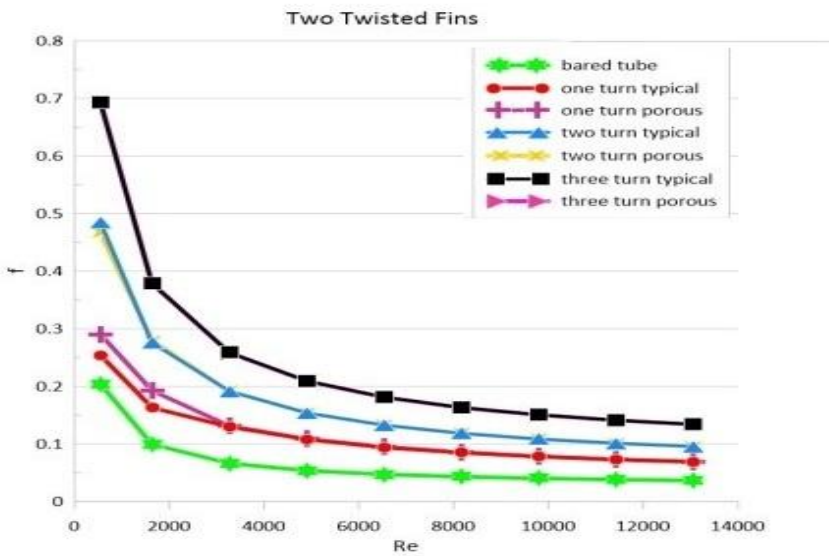



**Figure 14.** Nusselt number of waters at different ranges of Reynolds number



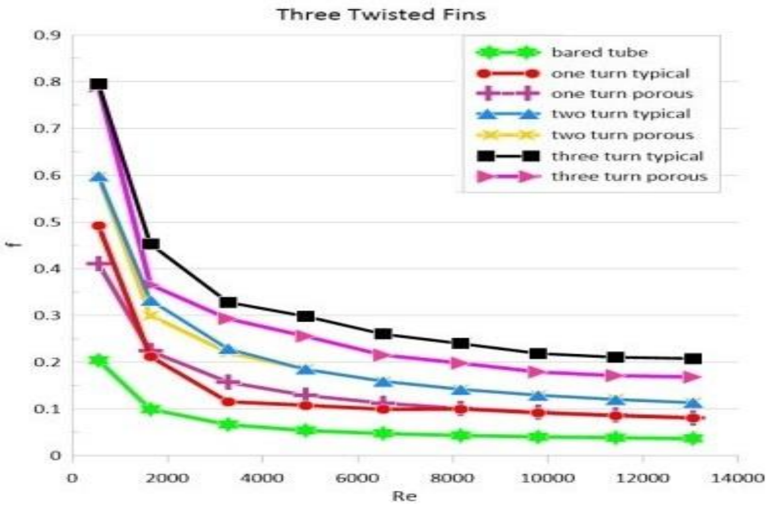
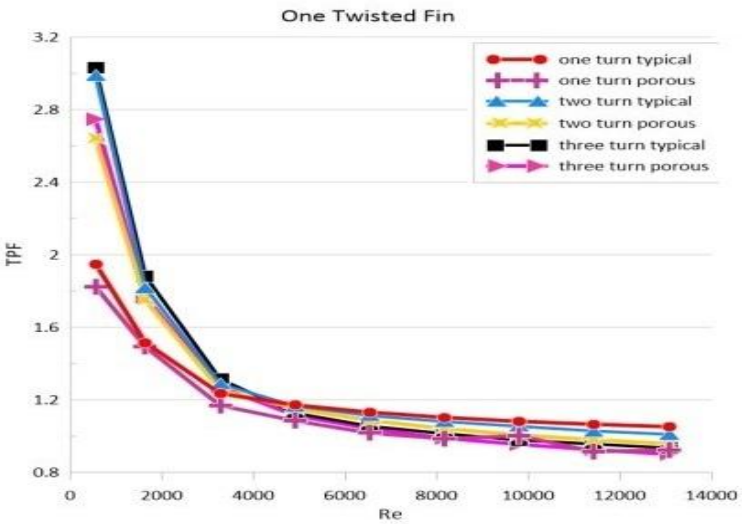
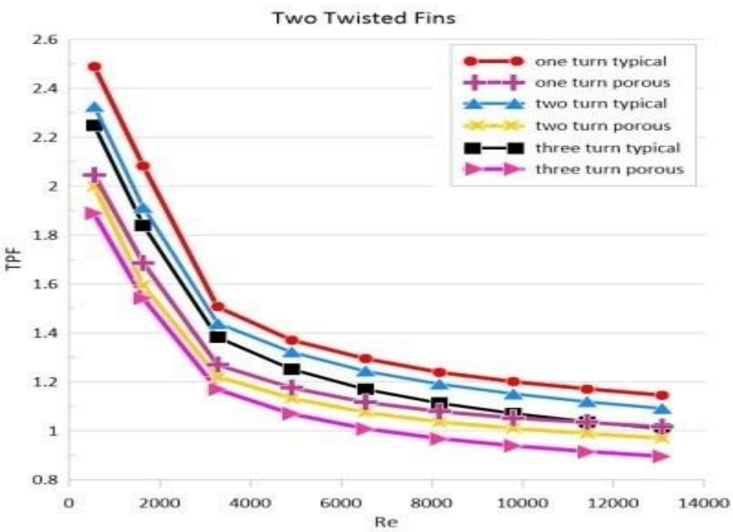



**Figure 15.** Friction factor of water at different ranges of Reynolds number



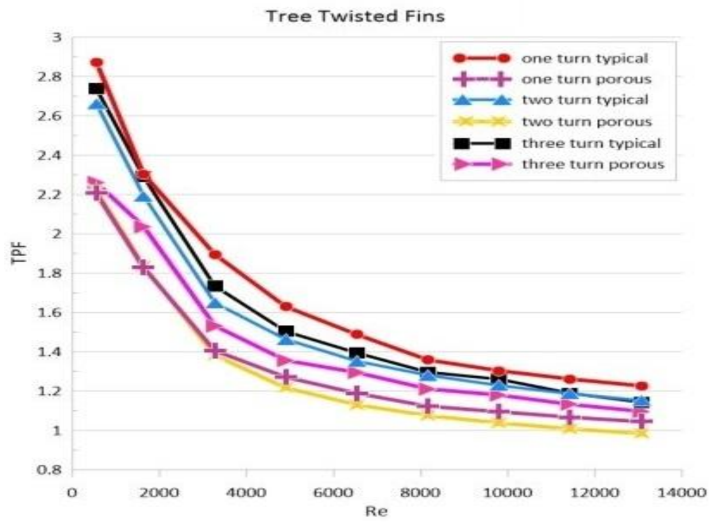



**Figure 16.** Thermal performance factor of water at different ranges of Reynolds number

## 9. Conclusions

From the numerical results of simulating the insertion of twisted tape: a conventional twisted tape and a metal foam twisted tape inside the outer surface of a double-pipe heat exchanger, the following was concluded:

1- When the twisted tape was inserted, the thermal performance factor increased for air flow by (7%, 11.2%, 5.7%) for one turn, two turns, and three turns, respectively of MFTF over TTT for one fin and for two fins (1.16%, 1.98%, 2.82%) for one turn, two turns, and three turns, respectively and for three fins (1.2%, -0.74%, -2%) for one turn, two turns, and three turns, respectively, the negative sign means the TTT better than MFTF.
2- Improving thermal performance is more effective in the laminar flow range for TTT about (27% , 17% , 18%) for one, two, and three fins respectively, and for MFTF about (34%, 29%, 25%) for one, two, and three fins respectively
3- When comparing typical twisted tape and MFTF, the maximum thermal performance improvement of MFTF is 11.2% for one fin with two turns around the pipe.
4- Foam fins demonstrate better scalability for air compared to water.
5- For the suggested future work, it could be focus on exploring different parameters, such as investigation of additional fluid types, optimization of metal foam and twisted tape geometries and cost-benefit analysis and practical implementation

In summary, the study indicates that MFTF provides better thermal performance than conventional twisted tape, particularly in air systems and in the laminar flow regime, with significant improvements seen in specific configurations.

## References


[1] Baragh, S.; Shokouhmand, H.; Ajarostaghi, S.S.M.; Nikian, M. An experimental investigation on forced convection heat transfer of single-phase flow in a channel with different arrangements of porous media. Int. J. Therm. Sci. 2018, 134, 370–379, DOI: 10.1016/j.ijthermalsci.2018.03.017.

[2] Hasan DJ, Farhan AA. The effect of staggered porous fins on the performance of photovoltaic panel in Baghdad. J Eng. 2020; 26:1-13. doi: 10.31026/j.eng.2020.08.01

[3] Baragh, S.; Shokouhmand, H.; Ajarostaghi, S.S.M. Experiments on mist flow and heat transfer in a pipe fitted with porous media. Int. J. Therm. Sci. 2019, 137, 388–398, DOI: 10.1016/j.ijthermalsci.2018.11.017.

[4] Ajarostaghi, S.S.M.; Delavar, M.A.; Poncet, S. Thermal mixing, cooling and entropy generation in a micromixer with a porous zone by the lattice Boltzmann method. J. Therm. Anal. Calorim. 2020, 140, 1321–1339, DOI: 10.1007/s10973-019-08956-2.

[5] Issam M. A. Aljubury, Raed G. Saihood, Ammar A. Farhan "Experimental study on thermo-hydraulic performance of metal foam twisted tape in a double pipe heat exchanger" Wiley Periodicals LLC. Heat

Transfer. 2022; 51:7910–7928, DOI: 10.1002/htj.22256.

[6] K. Ahmed, O. Le, I. Andri, A. Pina, and J. Fournier, "ScienceDirect ScienceDirect study heat transfer enhancement and fluid flow 15th of CFD study of heat enhancement and fluid flow characteristics of laminar flow through the pipe with helical helical screw tape t," Energy Procedia, vol. 160, pp. 699–706, 2019, doi: 10.1016/j.egypro.2019.02.190.

[7] P. Li, Z. Liu, W. Liu, and G. Chen, "Numerical study on heat transfer enhancement characteristics of pipe inserted with centrally hollow narrow twisted tapes," Int. J. Heat Mass Transf., vol. 88, pp. 481–491, 2015, doi: 10.1016/j.ijheatmasstransfer.2015.04.103.

[8] C. T. P. Eiamsa-ard and S. Eiamsa-ard, "Heat transfer and thermal performance characteristics of heat exchanger pipe fitted with perforated twisted-tapes," pp. 881–892, 2012, doi: 10.1007/s00231-011-0943-0.

[9] Jasim M. Mahdi, et al "Intensifying the thermal response of PCM via fin-assisted foam strips in the shell-and-tube heat storage system" Journal of Energy Storage.Volume 45, January 2022, 103733, DOI: 10.1016/j.est.2021.103733.

[10] Abdul Hadi N. Khalifa, Issam Mohammed Ali, Atheer Saleh Hassoon "Exergy Analysis of Refrigeration Cycle With Open-Cell Metal Foam Condenser" Journal of Multidisciplinary Engineering Science and Technology (JMEST) ISSN: 2458-9403 Vol. 3 Issue 4, April – 2016, DOI: 10.17561/jmest. v3i4.1060.

[11] ASH Abdul Hadi N. Khalifa, Issam M.Ali "Experimental Study the Performance of Aluminum Foams Condensers in the Vapor Compression Cycle" International Journal of Current Engineering and Technology 5 (4), 600-605, DOI: 10.14741/ijcet/v.5.4.600.

[12] Issam Mohammed Ali, et al "Experimental Investigation of Forced Convection Heat Transfer in Open Cell Copper Fins" Al-Nahrain Journal for Engineering Sciences (NJES) Vol.20 No.1, 0217 pp.272 – 280, DOI: 10.29194/ANJES.200100272.

[13] A. M. A. AL Kareem, "Study the Effect of Using Different Types of Twisted Tapes on Thermal Performance in Solar Collector," M.S. thesis, Dept. of Mechanical Engineering Techniques in Power, Technical College, July 2020, DOI: 10.13140/RG.2.2.35847.70564.

[14] Frank M. White "Fluid Mechanics" McGraw-Hill Series in Mechanical Engineering, Fourth Edition, 1991. PP: 219.

[15] Versteeg HK and Malalasekera V, 2007. "An Introduction to Computational Fluid Dynamics: The Finite Volume Method", Second Edition, Pearson Education Limited 1995, 2007

[16] R.L.Webb, N.H. Kim, Principles of Enhanced Heat Transfer, second ed., Taylor& Francis Group, New York, 2005.

[17] Frank M. White "Fluid Mechanics" McGraw-Hill Series in Mechanical Engineering, Fourth Edition, 1991. PP: 219.

[18] Rajendra Karwa "Performance Evaluation Criterion at Equal Pumping Power forEnhanced Performance Heat Transfer Surfaces" Hindawi Publishing Corporation Journal of Solar Energy Volume 2013, Article ID 370823, 9 pages http://dx.doi.org/10.1155/2013/370823,

[19] F. R. Menter, "Two-Equation Eddy-Viscosity Turbulence Models for Engineering Applications," AIAA Journal, vol. 32, no. 8, pp. 1598–1605, Aug. 1994. doi: 10.2514/3.12149.

[20] Brown, R., & Lee, S. (2018). Structural Analysis of Heat Exchanger Designs. International Journal of Heat Transfer, 25(3), 123-135.

[21] Johnson, T., & Smith, R. (2021). A Controlled Order Cell Type for Precise Mesh Control in Computational Fluid Dynamics. Journal of Computational Physics, 150(2), 245-259

[22] Smith, J., & Johnson, A. (2020). Experimental Study on Heat Transfer Enhancement Using Copper Twisted Tapes and Tubes. International Journal of Heat and Mass Transfer, 35(2), 123-135.

[23] Brown, A., & Lee, S. (2019). Assessing Simulation Accuracy: A Comparative Analysis between Numerical Results and Equations. In Proceedings of the International Conference on Computational Fluid Dynamics (pp. 120-135). Springer.