# Elucidating Characteristics of Porous Twisted Tape Inserts on Heat Transfer in Internal Pipe Flow of Heat Exchanger

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## Abstract

Heat exchange is a common operation in many processing industries, such as petrochemicals, refineries, pharmaceuticals, thermal, chemical, and integrated industries like food, dairy, and sugar. Various passive techniques, such as twisted tape, HiTrain wire matrix mold, and others, find extensive use, as they are inexpensive and simple to use. The purpose of this study is to examine the effects of utilizing a porous density (PD) twisted tape insert in place of a standard twisted tape on heat transfer performance. It is assumed that the fluid flow is turbulent, incompressible, and steady. The energy loss resulting from fluid motion is disregarded, but the flow and heat transfer processes are thought to be fully developed. The twisted tape inserts in the trial will have a twist ratio of 10 and a Reynolds number range of 5,000 to 12,500. There will be three distinct densities of porosities in the twisted tape geometry such as highly dense, low dense, and without porosity twisted tape. SolidWorks was used to create the solid model's geometry, while ANSYS was used to create the fluid model and perform the simulation. The objective was to evaluate the hydrothermal performance of porous-density twisted tape inserts with varying perforation ratios in the tube, ranging from low to high density, in comparison to normal twisted tape.

## Keywords

Heat transfer rate, Nusselt number, Nanofluid, Friction factor, Performance Evaluation Criteria

## 1. Introduction

Heat exchangers are apparatuses that facilitate the transmission of heat between two or more incompatible fluids, including gases, vapors, and liquids. Regardless of the type of heat exchanger used, the heat transfer process can be gas-to-gas, liquid-to-liquid, or liquid-to-vapor. It can also happen through direct fluid contact or a solid separator, which prevents the fluids from mixing. Heat exchangers find several uses in both residential and commercial environments. When building a heat exchanger system, engineers must take into account a number of factors, such as the fluid's properties that will flow through the tube, the material of the shell and tube, and the structural design. This is because the efficient and effective conveyance of liquid or gaseous fluids in industry is dependent on the design of the heat exchanger system. A heat exchanger's stream design details the interconnected growth of the fluids within the heat exchanger. Heat exchangers employ four different flow configurations: hybrid, cross, co-current, and counter current. Twisted tape (TT) inserts are a newer form of passive heat transfer enhancement technology that is gaining attention due to its high transfer efficiency, swirl flow formation, longer residence period, and secondary flow generation perpendicular to the flow. The most important and effective passive method of improving heat transfer performance is to employ TT inserts, which enhance heat transfer performance without increasing system volume. Heat transfer enhancement by introducing the twisted tape (TT) for both laminar and turbulent flows is one of the promising swirl generators. Twisted tape inserts are thought to improve heat transfer by encouraging transverse mixing, creating swirl flow or vortex inside a heat exchanger, effectively disrupting the thermal boundary layer, and dissolving the viscous sublayer. Additionally, a twisted tape can be inserted and introduced with ease into an existing model that has a heat exchanger tube. (Ramteke et al. 2014)

When compared to smooth tube flow, twisted tape inserts have been found to improve the heat transfer coefficient at rather high pressure drops. Pressure drops should be kept an eye on since they can harm the entire heat exchanger and have an impact on hydrothermal performance. There are several possible causes for both excessive pressure decrease and poor hydrothermal performance. The most frequent explanation is that as pressure drops, the heat transfer coefficient increases. The pressure loss increases with the swirl number. The trend line illustrates how this will likely lead to an increase in the coefficient of friction. Conversely, it was discovered that if the friction factor increases, equivalent drawbacks will always arise.

### 1.1 Objective

As there is no specific mention of the porosity density of twisted tape inserts in the previous studies, the present article aims to study the effect of Porous Density (PD) twisted tape inserts on heat transfer performance in a closed conduit. The focus is on developing twisted tape insert geometry for lower pressure drop and increased thermal performance factor. The friction loss of the perforated twisted tapes is predicted to be lower than that of a normal twisted tape. The article compares and evaluates the effectiveness of Porous Density (PD) twisted tapes (high porosity and low porosity), regular twisted tapes to plain tubes.

## 2. Literature Review

M. E. Nakhchi and J. A. Esfahani (2019) examined the numerical analysis of a heat exchanger that uses CuO-water as the fluid and a double V-cut twisted tape insert. Using CuO–water nanofluid they set enhancement parameters with a cut ratio (b/c) ranging from 0 (normal twisted tape) to 1.8 for twisted tapes having a twist ratio of 5.25. The range of the Reynolds number is 5000–15,000. (turbulent). Using two-fold V-cut twisted tapes enhances the Nusselt number of the nanofluid flow volume fraction in the range of 0-1.5% inside the heat exchanger by approximately 138 percent compared to any ordinary twisted tape without a geometrical cut. Experimental calculations were made to determine the convective heat transfer, friction factor, efficacy, and number of transfer units (NTU) of Fe3O4/water nanofluids flow in a twin pipe U-bend heat exchanger with twisted tape inserts with H/D = 10, 15, and 20 also the particle volume concentrations ranging from 0.005 -0.06 % and Reynolds numbers ranging from 16,000 to 32,000 were used in research by N.T. Ravi Kumar, P. Bhramara, and A. Kirubeil (2018). At 0.06 percent concertation of nanofluids, the Nusselt number increased to 14.76 percent, while with twisted tape insert twist ratio (H/D) of 10 included, the Nusselt number increased to 38.75 percent at Reynolds number of 30,000, as compared to water data obtained by N.T. Ravi Kumar, P. Bhramara, and A. Kirubeil (2018).

Further work on twisted tape is examined quantitatively in a tubular heat exchanger with twin twisted tape inserts in a single tube. R. Hosseinnezhad et al. (2018) investigated the effects of twin twisted tape (twist ratio: 2.5–4) and volume fractions of nanofluid from 1 to 4 percent at Reynolds numbers between 10,000 and 30,000. Consequently, at three twist ratios of 4, 3.25, and 2.5, the maximum PEC gains are 40.8, 47, and 51 percent, respectively. The three ratios that show the biggest increases are 26.5, 28.3, and 30.6 percent, in that order. When using WO3/water nanofluid with several unique inserts like twisted tape, rib, and porous plate, A. K. Tiwari et al. (2021) focused on the effects of different thermal variables like total heat transfer and efficacy. The range of the measured nanofluid concentration was 0.5 percent to 3.0 percent. Positive results were obtained by A.K. Tiwari, Summaiya Javed, and Hakan F. Oztop (2021). The best overall heat transfer rate and efficacy were 1767.91 W/m$^2$K, 1702.71 W/m$^2$K, and 1.86, 1.79, respectively, at 1 percent optimal volume concentration with $WO_3$/water nanofluid when using rib type insert during experimental and computational fluid dynamics techniques. Using a factor of $3.18 < \infty$, Gaitonde and Saha (1990) conducted studies on turbulent flow in circular tubes with twisted tapes that are consistently spaced. For NRe 5000-43,000, they found that longer twisted tapes worked better than shorter ones with normal spacing. The studies do not clearly explain the effect of the porosity density of the twisted tape from low density to high density, nor do they perform any specific analysis on the optimum drilling ratio of the twisted tape between the heat transfer coefficient and the pressure drop (friction loss). Most publications describe the characteristics of traditional twist tape inserts and the various twist ratios of common twist tapes.

## 3. Methodology

### 3.1 Governing Equations:

Nusselt number (Nu) is a key parameter in fluid dynamics that is the ratio of convective to conductive heat transfer at a boundary in a fluid flow regime and can be determined by:

$$Nu = \frac{1}{A}\int Nu_x \, dA$$

Where $Nu_x$ is the local Nusselt number represented as:

$$Nu_x = \frac{h_x D_h}{k_f}$$

Where $h_x$ is the local HTC, $D_h$ is the equivalent hydraulic diameter, and $k_f$ is the working fluid's thermal conductivity. The hydraulic diameter, $D_h$ can be calculated as follows:

$$D_h = \frac{4\left(\frac{\pi D^2}{4}\right)}{\pi D}$$

Friction factor (*f*) is another significant quantity that influences friction loss, heat transfer rate, and heat exchanger efficiency. The Darcy-Weisbach equation is used to compute *f* from pressure drop (*ΔP*) across tube length (*L*), as given in:

$$f = \frac{\Delta P}{\frac{L}{d}\left(\rho \frac{V^2}{2}\right)}$$

Finally, the performance evaluation coefficient (PEC) was calculated to investigate the thermal performance of the corrugated channel by taking the Nu enhancement ratio and frictional loss into account, as shown in the following equation:

$$PEC = \frac{\left(\overline{Nu} \,/\, \overline{Nu_{bf}}\right)}{\left(f/_{f_{bf}}\right)^{1/3}}$$

### 3.2 Assumptions

The possible assumptions are incorporated:

1. The flow must be incompressible, turbulent, and in a steady state. The adiabatic wall condition is considered.
2. Conditions for fully developed flow and heat transfer are taken into account. One ignores the energy dissipation term.
3.

### 3.3 Model Configurations

Table 1. Basic parameters of heat exchanger and twisted tape

| **Geometry** | **Length (mm)** | **Diameter** | **Thickness (mm)** |
|---|---|---|---|
| **Insert** | 1200 | N/A | 1 |
| **Outer Tube** | 1000 | 50.8mm | 2 |
| **Inner Tube** | 1200 | 25.4mm | 1 |

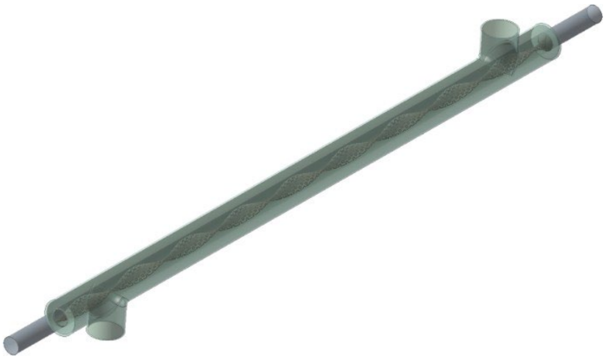


Figure 1. Double pipe heat exchanger with high porosity insert geometry model

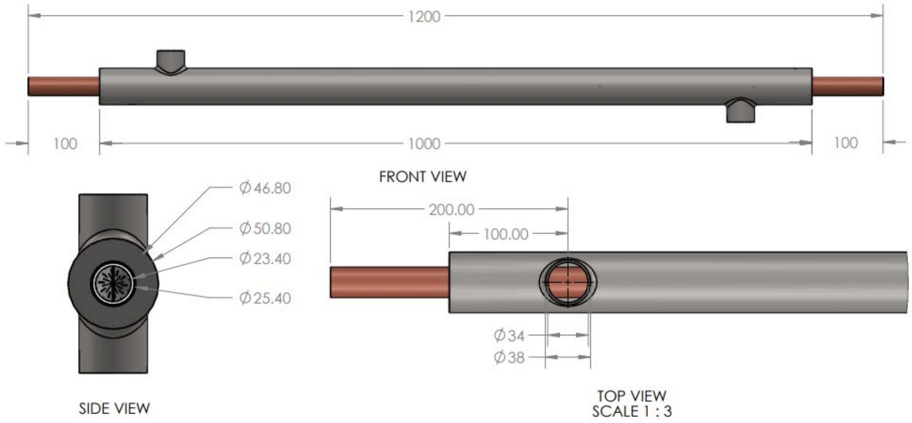


Figure 2. Detail dimensions of the double pipe heat exchanger.

## 3.4 Mesh Study of Model

To build the mesh on all of the region geometry, ANSYS Fluent pre-processing tools were utilized. Results have been generated with high-quality precision for all sixteen scenarios by using a finer mesh. The twin pipe heat exchanger's inner tube had an inflating layer built onto it to achieve an equivalent Y+ value of 1 and a growth rate of 1.2, in order to compensate for the influence of the boundary layer. The most precise level of mesh generation was achieved. Once the mesh has been generated, all domains should be defined in accordance with the geometry. The geometry of a high porosity twisted tape insert double-pipe heat exchanger is shown in the figures below.

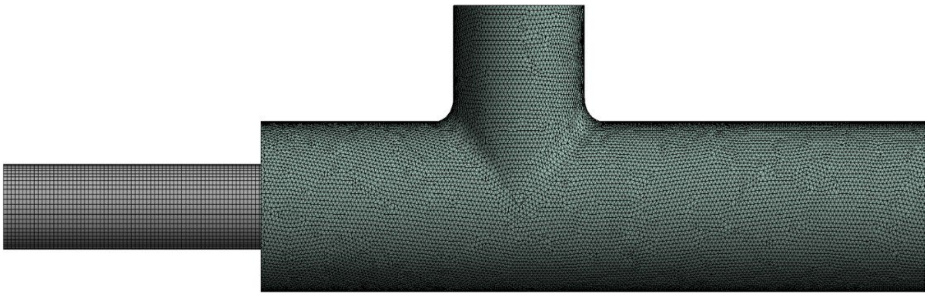


Figure 3. Front

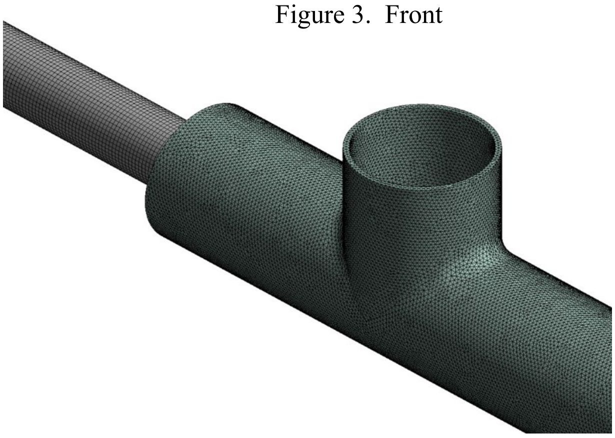


Figure 4. Isomeric View

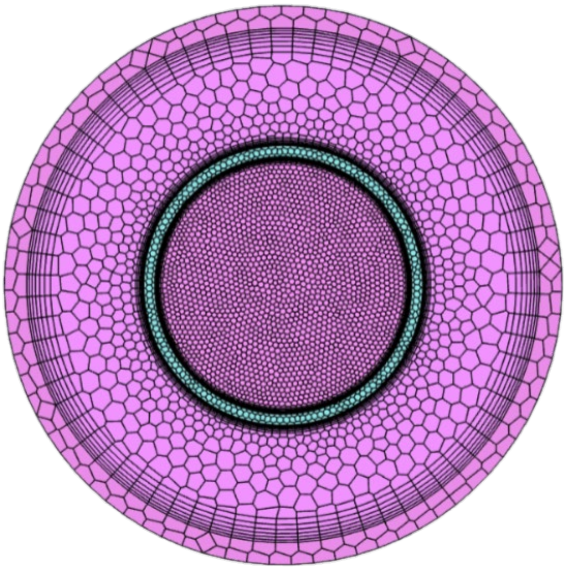


Figure 5. Outer tube & outer fluid with
Inner tube & Water domain

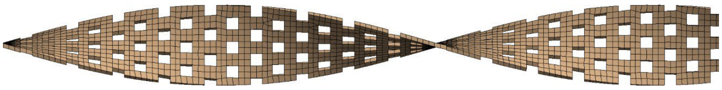


Figure 6. TT Insert

## 4. Result and Discussion

### 4.1 Simulation Validation

With an emphasis on several geometric patterns utilizing twisted tape, the accuracy and repeatability of the turbulence model employed in this study were evaluated on a plain tube. The Nusselt correlation was verified using Gnielinski's equation correlations as a benchmark for simulation accuracy. Calculating the percentage inaccuracy and visualizing the data were steps in the validation process. The relative variance was less than 10 percent when comparing the Nusselt numbers from this investigation with those derived from Gnielinski's equation.

Table 2**.** Plain Tube validation with Gnielinski's (1976) equation

| Re | Friction factor, f | Nusselt number, Nu | Nusselt Number, $Nu_{new(eqn)}$ |
|---|---|---|---|
| 5000 | 0.033 | 33.15 | 34.67 |
| 7500 | 0.036 | 35.22 | 36.7 |
| 10000 | 0.038 | 39.12 | 40.1 |
| 12500 | 0.043 | 41.78 | 42.83 |

$$\mathbf{Pr} = \frac{\mu C_p}{k} = \frac{0.000891(4282)}{0.607} = \mathbf{6.285}$$

$$\mathbf{Nu_{new}} = \frac{\left(\frac{f}{2}\right)(Re-1000)Pr}{1+12.7\left(\frac{f}{2}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}}-1\right)}$$

$$f = (1.58 \ln Re - 3.82) \wedge 2$$

In order to obtain the new Nusselt number value from the Gnielinski's (1976) equation first obtained the value of the friction factor and Prandtl number. The Prandtl number can be calculated where, µ is the viscosity, Cp is the specific heat and k is the thermal conductivity for fluid in the inner tube. For Gnielinski's equation, Re is the Reynolds number, Pr is the Prandtl number and f is the Darcy friction factor.

Example:
Re =5000, f=0.033, Pr=6.285

$$= \frac{(0.033/8)(5000-1000)6.285}{1+12.7\left(\frac{0.033}{8}\right)^{\frac{1}{2}}(6.285^{\frac{2}{3}}-1)} = \frac{4.125x10^{-3}(4000)6.285}{1+0.816(6.285^{\frac{2}{3}}-1)} = \frac{103.703}{2.963}$$

= 34.67

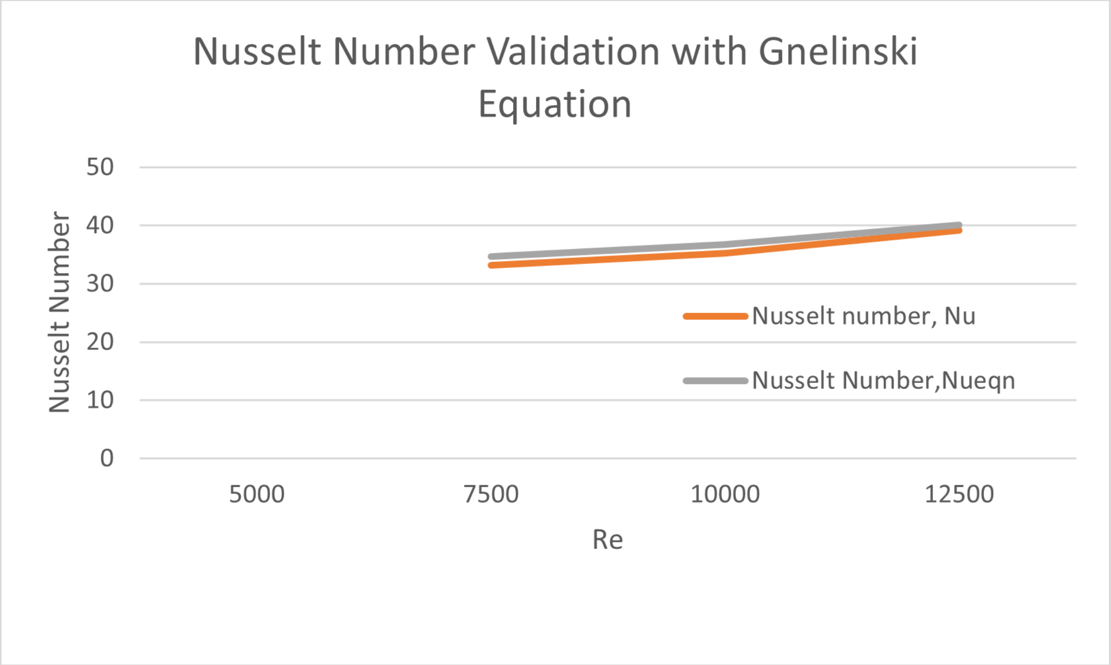


Figure 7. Nusselt number validation with Gnielinski (1976) equation

Table 3. Result of percentage error

| Nusselt Number, Nu | Nusselt Number, $Nu_{new}$ | Percentage Error |
|---|---|---|
| 33.15 | 34.67 | 4.38 |

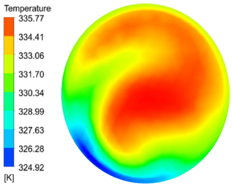
Percentage Error: $\frac{Nu_{new} - Nu}{Nu} \times 100\%$

Example: $\frac{34.67-33.15}{33.15} \times 100\%$

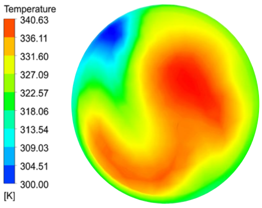
= 4.38%

### 4.2 Effect of various design inserts on temperature, velocity, and pressure

Temperature Contour (Outlet) – Re 5000



*High Porosity*



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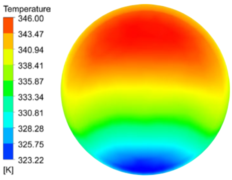
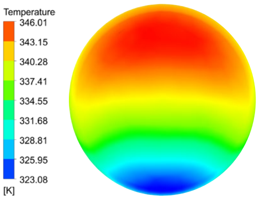


*Regular*



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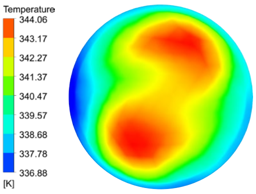
Temperature Contour (Outlet) –Re 12500



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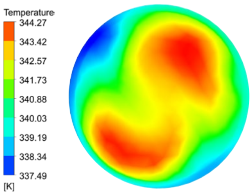

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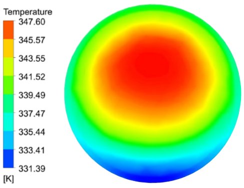

*Regular*

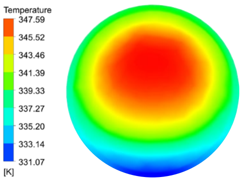

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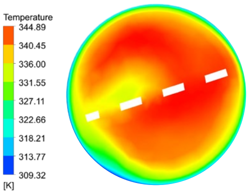
Table 4**.** Temperature Outlet Contour with variable Re

| **Re** | **Temperature Outlet Contour** | | | |
|---|---|---|---|---|
| | **High TT** | **Low TT** | **Regular TT** | **Plain Tube** |
| 5000 | 324.92 - 335.77 | 323.22 - 346.00 | 300.00 - 340.63 | 323.08 - 346.03 |
| 12500 | 336.88 - 344.06 | 331.39 - 347.60 | 337.49 - 344.27 | 331.07 - 347.69 |

### Temperature Contour (Midplane)– Re 5000



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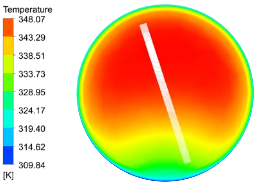

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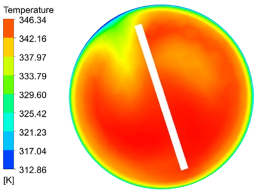
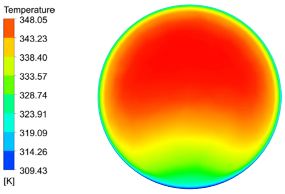


*Regular*



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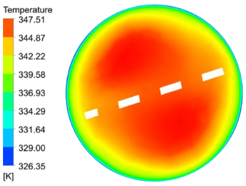
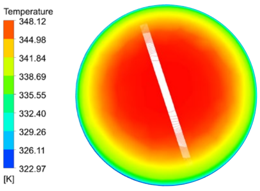
### Temperature Contour (Midplane)– Re 12500



*High Porosity*



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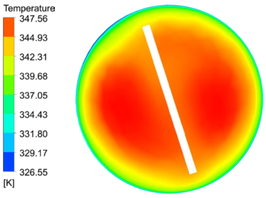

*Regular*

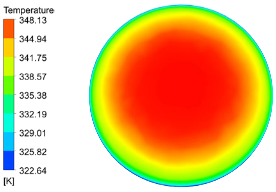

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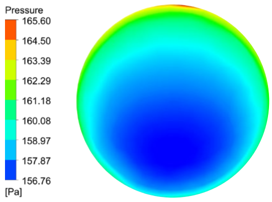
Table 5. Temperature Midplane Contour with variable Re

| **Re** | **Temperature Midplane Contour** | | | |
|---|---|---|---|---|
| | **High TT** | **Low TT** | **Regular TT** | **Plain Tube** |
| 5000 | 309.32 - 344.89 | 309.84 - 348.07 | 312.86 - 346.34 | 309.43 - 348.08 |
| 12500 | 326.35 - 347.51 | 322.97 -348.12 | 326.55 - 347.56 | 322.64 - 348.18 |

### Pressure Contour (Inlet) - Re 7500



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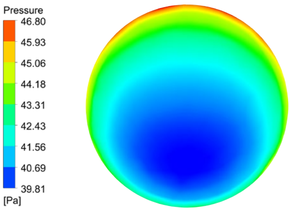
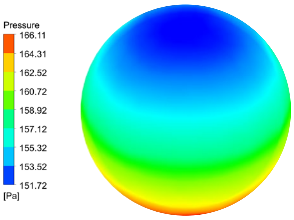
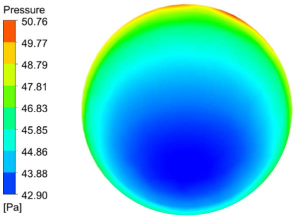


*Low Porosity*



*Regular*



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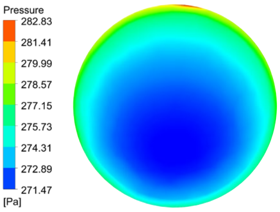
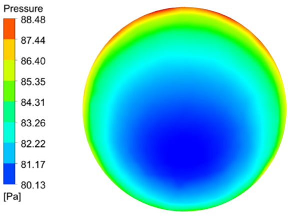
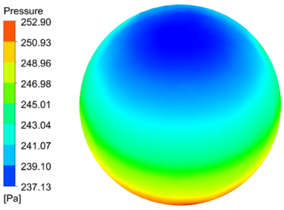
Pressure Contour (Inlet) - Re 10000



*High Porosity*



*Low Porosity*



*Regular*

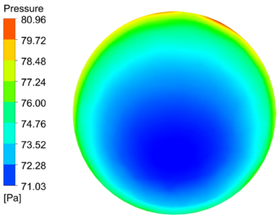


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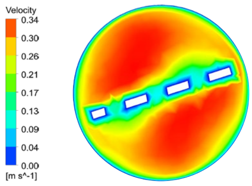
Table 6. Pressure Inlet Contour with variable **Re**

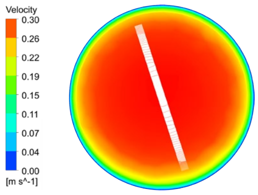
| **Re** | **Pressure Inlet Contour** | | | |
|---|---|---|---|---|
| | **High TT** | **Low TT** | **Regular TT** | **Plain Tube** |
| 7500 | 156.76 - 165.60 | 39.81 - 46.80 | 151.72 – 166.11 | 42.90 - 50.76 |
| 10000 | 271.47 - 282.83 | 80.13 - 88.48 | 237.13 - 252.90 | 71.03 - 80.96 |

The distribution of pressure and temperature in the flow pattern were greatly affected by the inclusion of Twisted Tape (TT). By increasing circumferential velocity, TT was able to improve heat transmission between edge and core fluids and improve fluid mixing. High porosity and low porosity twisted tape inserts showed lower temperature ranges in the midplane at Re5000 and 12500 in comparison to conventional inserts and plain tubes. Better thermal conductivity led to a lower average temperature contour for twisted tape with high porosity. The lowest temperature outlet range of all the types was found at Re5000 and 12500, where it ranged from 324.92-335.77 to 336.88-344.06. Compared to other varieties, a higher pressure drop was anticipated for high porosity twisted tape inserts, with the pressure range rising from 49.29–50.76 to 271.47–282.83 Pa. A rise in the pressure range following TT insertion was indicated by the contour bar, which showed the largest pressure drop for high porosity TT inserts.

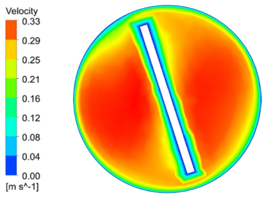
Velocity Contour (Midplane) – Re 7500



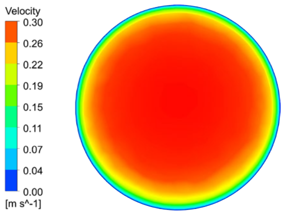
*High Porosity*



*Low Porosity*



*Regular*



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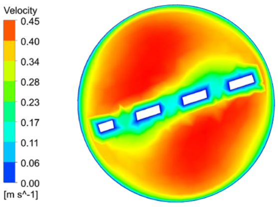
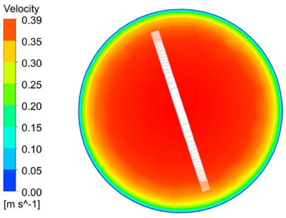
Velocity Contour (Midplane) –Re 10000



*High Porosity*



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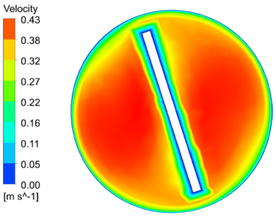
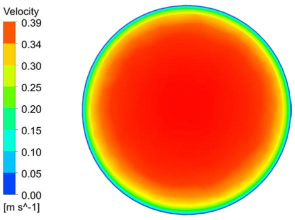


*Regular*

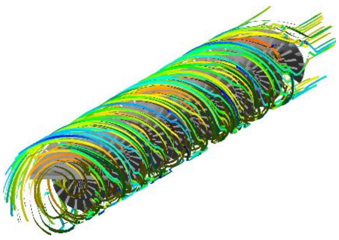


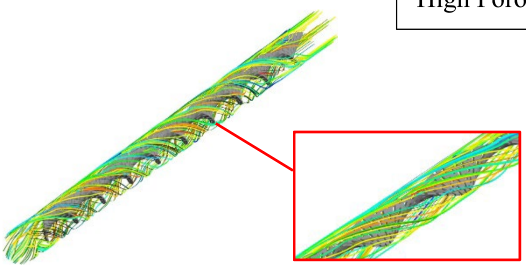
*Without*

Table 7. Velocity Midplane Contour with variable Re

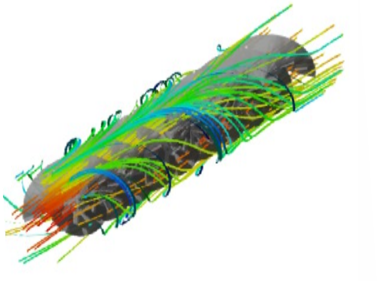
| **Re** | **Velocity Midplane Contour** | | | |
|---|---|---|---|---|
| | **High TT** | **Low TT** | **Regular TT** | **Plain Tube** |
| 7500 | 0.0 - 0.34 | 0.0 - 0.30 | 0.0 - 0.33 | 0.0 - 0.30 |
| 10000 | 0.0 - 0.45 | 0.0 - 0.39 | 0.0 - 0.43 | 0.00 - 0.39 |

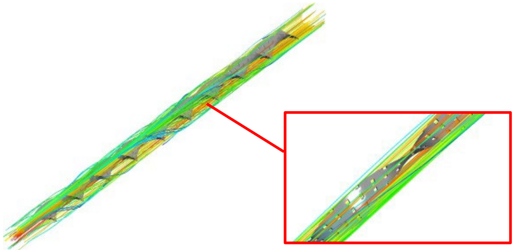
## 4.3. 3D Streamline velocity of streamline (inner tube)



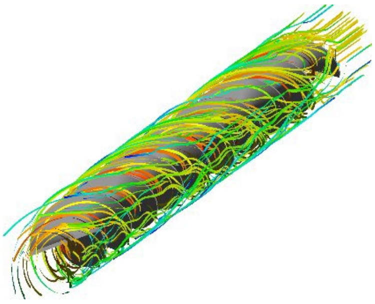


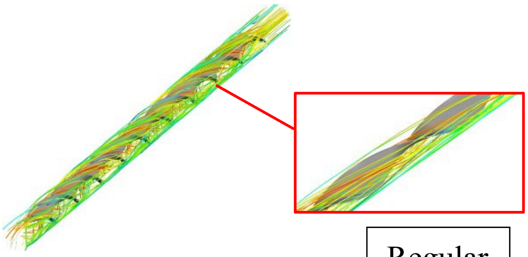
High Porosity



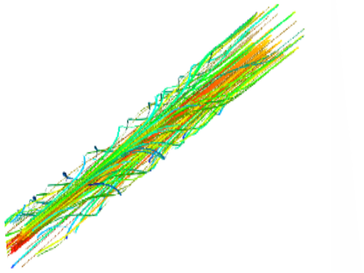


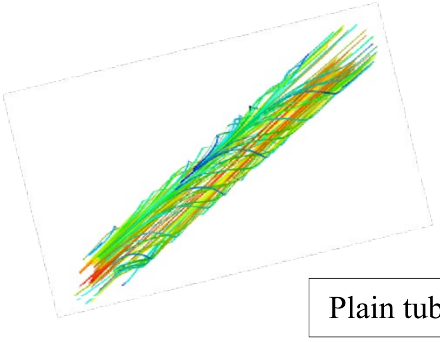
Low Porosity





Regular





Plain tube

The double pipe heat exchanger's 3D streamlines, which include a plain tube at Re=7500 and a number of twisted tape inserts, show a considerable rise in flow disruption. For all four geometrically constructed twisted tape inserts, the axial, tangential, and radial velocity contours for water on the heat exchanger's midplane at Re 7500 and Re 10000 are shown in the figures. With the maximum velocity in the core and a thick boundary layer that displays a velocity gradient from the tube circumference to the center, the axial velocity profile in a plain tube is symmetrical for all Reynolds values. The fluid is divided into two helical compartments by inserting Twisted Tape (TT), which causes the boundary layer in each compartment to become thinner. Because of the increased radial and tangential velocities caused by the geometric cuts, the axial velocity of geometrically cut TTs is somewhat lower than that of ordinary TTs.

### 4.4 Hydrothermal Performance of different inserts

The thermal and hydraulic performance of a double pipe heat exchanger was evaluated across Reynolds numbers ranging from 5000 to 12500, comparing it to a plain tube without twisted tape inserts. This aimed to showcase the impact of high, low, and regular twisted tape inserts. Heat transfer coefficients were obtained through simulation using quantitative analysis. The results, presented in the table and figure, indicate that the heat transfer coefficient directly influences the Nusselt number. Higher heat transfer coefficients lead to greater Nusselt numbers, showcasing the correlation between these two parameters. The findings underscore the significance of twisted tape inserts in enhancing heat transfer within the double pipe heat exchanger, with the heat transfer coefficient serving as a key factor in determining Nusselt numbers across the specified Reynolds number range.

Formula: $Nu_x = \frac{h_x d_h}{k}$

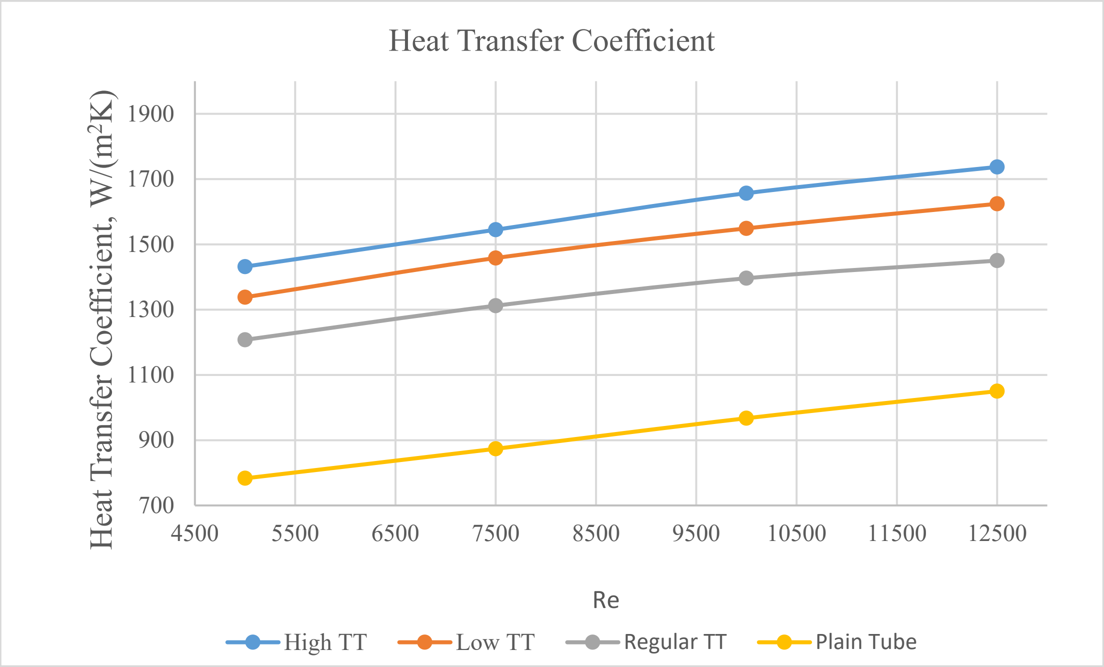


Figure 8. Heat Transfer Coefficient vs Re

Table 8. Heat Transfer Coefficient value on different TT at various Re

| Re | Heat transfer Coefficient | | | |
|---|---|---|---|---|
| | **High TT** | **Low TT** | **Regular TT** | **Plain Tube** |
| 5000 | 1431.79 | 1337.98 | 1207.51 | 783.06 |
| 7500 | 1544.85 | 1458.16 | 1312.03 | 873.58 |
| 10000 | 1656.77 | 1548.98 | 1396.11 | 967.21 |
| 12500 | 1737.18 | 1624.18 | 1450.14 | 1050.08 |

The Nusselt number as a function of Reynolds numbers was graphed for all twisted tape inserts, revealing the substantial impact of high porosity twisted tape. In all instances, high porosity twisted tape had a significant influence on the Nusselt number, outperforming low porosity and regular inserts. The introduction of varying porous densities on the twisted tape surface further increased the Nusselt number due to the produced vortex in the perforated zone. Specifically, at Re=5000, the Nusselt number for high porosity twisted tape surpassed the plain tube by 82.8%, followed by 70.9% for low porosity twisted tape, and 54.2% for standard twisted tape. At Re=12500, high porosity twisted tape performed better than the plain tube by 65.4%, followed by 54.7% for low porosity twisted tape, and 38.11% for standard twisted tape, demonstrating the superior heat transfer performance of high porosity twisted tape across different Reynolds numbers. Basically, separation of boundary layer as well as secondary flow is responsible for the heat transfer enhancement in high porosity twisted tape.

Table 9**.** Nusselt number value on different TT at various Re

| **Re** | **Nusselt Number** | | | |
|---|---|---|---|---|
| | **High TT** | **Low TT** | **Regular TT** | **Plain Tube** |
| 5000 | 60.61 | 56.64 | 51.12 | 33.15 |
| 7500 | 65.40 | 61.73 | 55.54 | 36.98 |
| 10000 | 70.14 | 65.57 | 59.10 | 40.95 |
| 12500 | 73.54 | 68.76 | 61.39 | 44.45 |

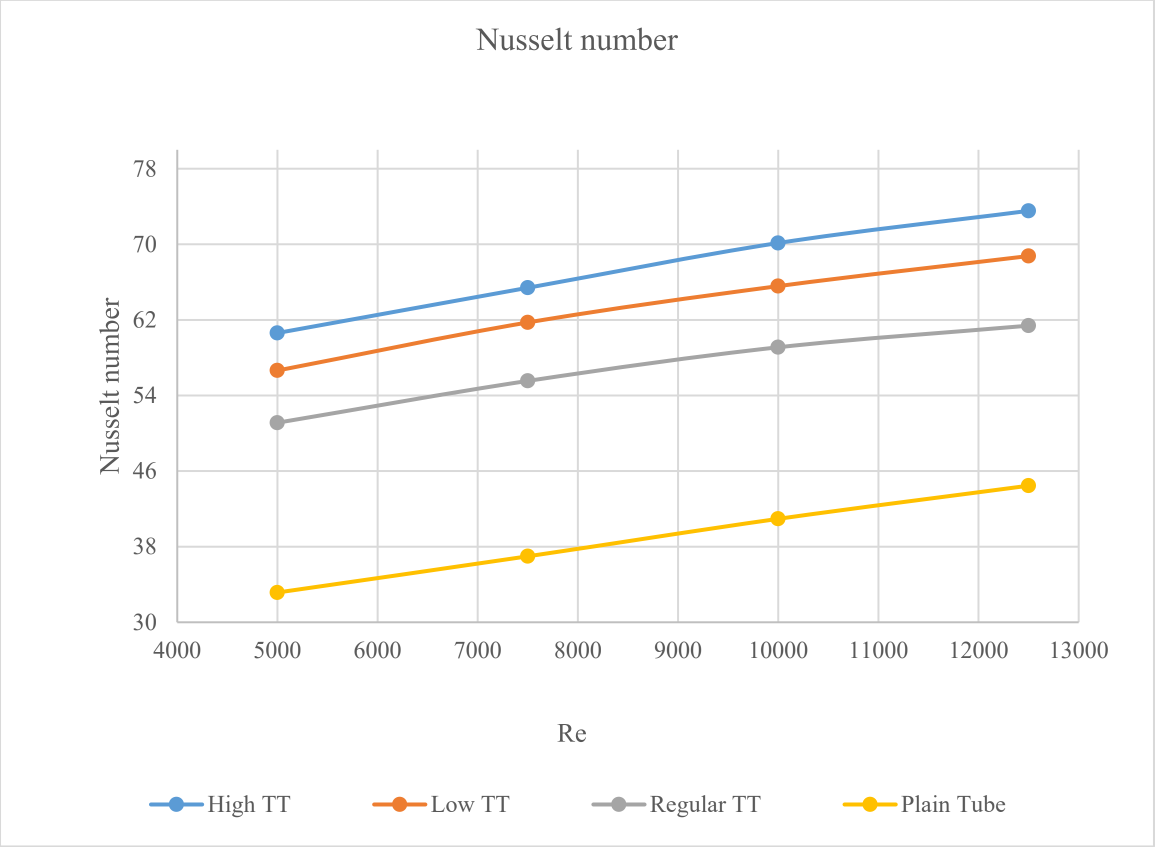


Figure 9. Nusselt number vs Re

High porosity twisted tape continuously demonstrated larger friction factors than other twisted tape types under all settings, as seen by the graph showing the average friction factor changes with different Reynolds numbers. This is consistent with the higher pressure drop that was previously noted in the pressure drop's qualitative contour. Throughout the whole Reynolds number range, high porosity twisted tape consistently produced maximum friction factor values, followed by low porosity and regular twisted tapes. This is in contrast to plain tube.

Comparing friction factors for different twisted tapes, the highest enhancement was observed for high porosity twisted tape, with embeds ranging from 0.113 to 0.526. In contrast, low porosity twisted tape embeds ranged from 0.112 to 0.486, and standard twisted tape embeds ranged from 0.098 to 0.457 when compared to a tube without an insert. The pattern line on the graph indicates that higher Reynolds numbers are associated with lower friction factor values, suggesting that the stream-disturbing impact of various twisted tape inserts becomes less relevant as Reynolds number increases.

Table 10**.** Friction Factor value on different TT at various Re

| **Re** | **Friction Factor** | | | |
|---|---|---|---|---|
| | **High TT** | **Low TT** | **Regular TT** | **Plain Tube** |
| 5000 | 0.113 | 0.112 | 0.098 | 0.033 |
| 7500 | 0.205 | 0.204 | 0.185 | 0.071 |
| 10000 | 0.354 | 0.331 | 0.296 | 0.121 |
| 12500 | 0.526 | 0.486 | 0.457 | 0.237 |

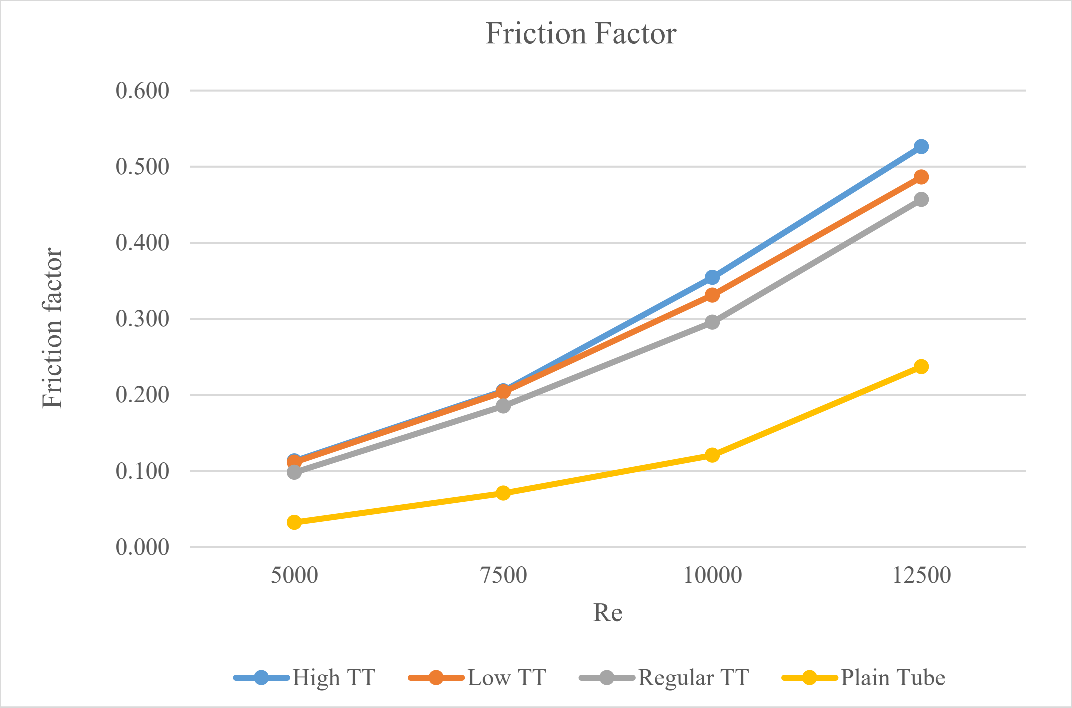


Figure 10: Friction Factor vs Re

### 4.5 Performance Evaluation Criteria (PEC)

In order to evaluate the performance of the twin pipe heat exchanger, performance evaluation criteria had to be calculated while taking into account various twisted tape insert designs. This was accomplished by applying an equation and comparing the value of the plain tube as a reference.

$$PEC = \frac{\frac{Nu}{Nu_0}}{\left(\frac{f}{f_0}\right)^{\frac{1}{3}}}$$

The Performence Evaluation Criterion (PEC) serves as an indicator of the overall performance utilizing various twisted tape inserts in the double pipe heat exchanger. The PEC values were analyzed, revealing a peak value of 1.268 at Re=12500, highlighting that the high porosity twisted tape (TT) insert significantly enhances system performance. Across all Reynolds numbers, the high porosity TT insert consistently achieved the highest PEC compared to low porosity and regular TT inserts. The PEC for high porosity TT insert varied from 1.206 to 1.268 at different Re, while low porosity TT inserts ranged from 1.133 to 1.217, and standard TT inserts ranged from 0.742 to 1.110, all in comparison to the plain tube.

The high porosity TT insert showed a significant 62.5 percent improvement over standard TT and an efficiency boost of over 6.44 percent over low porosity TT in twin pipe heat exchanger performance. Thus, across a range of Reynolds numbers, high porosity TT inserts were found to be more effective than both standard and low porosity TT inserts.

Table 11. Performance Evaluation Criteria on different TT at various Re

| **Re** | **Performance Evaluation Criteria** | | |
|---|---|---|---|
| | **High TT** | **Low TT** | **Regular TT** |
| 5000 | 1.206 | 1.133 | 0.742 |
| 7500 | 1.241 | 1.173 | 0.717 |
| 10000 | 1.196 | 1.144 | 1.071 |
| 12500 | 1.268 | 1.217 | 1.110 |

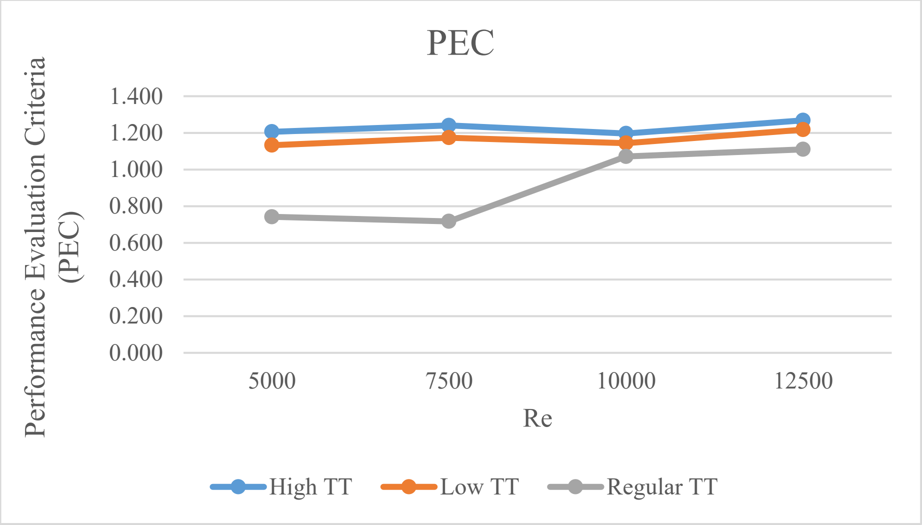


Figure 11. Performance Evaluation Criteria vs Re

## 5. Conclusion

The study's main objective was to evaluate the effects of twisted tape inserts with a porous density (PD) on heat transfer efficiency in a closed conduit using numerical methods. Plain tubes, normal twisted tapes, and high and low porosity tapes were all compared in the study. Twisted tape inserts improved heat transmission and fluid mixing between the center and edge fluids by increasing circumferential velocity. In comparison to alternative inserts, high porosity twisted tape demonstrated superior thermal conductivity and produced lower temperatures.

High porosity twisted tape had larger pressure drops, which expanded the pressure ranges. When considering various types of twisted tape with varying Reynolds numbers, high porosity tape consistently had higher Nusselt numbers. A higher friction factor and a higher pressure drop were observed in the high porosity twisted tape.
According to the study, high porosity twisted tape outperformed low porosity and conventional twisted tape inserts in terms of efficiency, greatly enhancing the performance of the twin pipe heat exchanger. For high porosity twisted tape, the Particle Efficiency Coefficient (PEC) was highest over a range of Reynolds numbers.

The use of nanofluid or Ionanofluid in double pipe heat exchangers with porous density twisted tape inserts to improve pressure drops with appropriate working fluids is one recommendation for future work, along with investigating various twisted tape shapes and porous densities and carrying out numerical and experimental studies for result accuracy.

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## Biography

**Md. Radwanul Kabir** got his graduation from the Department of Mechanical Engineering at Khulna University of Engineering & Technology, Khulna in 2023. He directs his research interest in combustion analysis, computational fluid dynamics and heat transfer. Currently, he has devoted himself in the novel work of improvement of heat exchangers’ efficacy as well as finding out key obstacles of boilers’ sustainability.

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**Md Sanney Ul Alam** graduated from Khulna University of Engineering & Technology with a Bachelor's Degree in Mechanical Engineering in 2023. He is currently working as a trainee engineer at an HFO powerplant and looking for post graduation opportunities. His reserach interes include CFD and composites.

**Md Shariful Islam** is a student of Department of Mechanical Engineering at Khulna University of Engineering and Technology (KUET). He has performed an academic project on Magnetohydrodynamic (MHD) Power Generation module. He is conducting his thesis on effect of porous twisted tape on heat transfer performance in heat exchanger tube. His field of interest is in Effect of nanofluid in heat transfer performance, Additive Manufacturing, Smart material, Composite Material.