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Enhancement of Heat Transfer in a Pipe Fitted with Rectangular Cut Twisted Tape Inserts

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ABSTRACT

This study has numerically analyzed heat transfer intensification in a pipe (U-loop) by incorporating rectangular-cut twisted tape inserts. This research uses a tubular pipe of 26.6 mm inner diameter and a 1935.62 mm long U-loop-shaped tubular pipe. The rectangular cuts are placed on one side of the twisted tape to create swirling along the flow region. From our simulation, we found the highest *Nu* number (357.59 at *Re*=17288.05) and better f (friction factor) for the tube installed with rectangular cut twisted tape insert compared to plain twisted tape and plain tube. It has also been yielded that the best thermal performance factor of 1.04 is established for rectangular-cut twisted tape inserts compared to other types. Since the rectangular cut twisted tape has increased the heat transfer rate, this concept can be used to facilitate the heat transfer rate in heat exchangers in various industrial sectors.

1. INTRODUCTION

The enhancement of heat transfer rate is a significant issue that needs to improve in the industrial area. The heat transfer rate can be amplified by expanding the efficacy of Heat Exchangers (HX). The device is usually used in refrigeration, air conditioning, power plants, food processing, and automobiles [1]. Process industries emphasize different techniques for augmenting the rate of heat transfer so that they can minimize the cost and maximize the performance of a HX. Three methods such as, active, passive, and combined are implemented to improve the heat transfer rate in a HX. Among these three the passive method is widely used in HX as it is operated without any external power source [2]. However, surface coating, wire coil inserts, surface roughness, and several varieties of Twisted Tape (TT) such as plain TT, full-length TT, short–length TT, square–cut TT, and V-cut TT are popularly employed in this method [3]. The flow region's twisted tapes (TT) produce eddies due to increased perturbation between the internal surface of the tube and the inserted twisted tape. These risen eddies, generated by the turbulators ultimately facilitate the convective heat transfer coefficient by disrupting the boundary layer (BL) [4]. Accordingly, the application of twisted tape inserts can lessen pressure loss, boost heat transfer coefficient, and optimize heat exchanger (HX) functionality [5]. So TT inserts technology has a great impact on the industrial area.

Numerous studies have investigated the augmentation of heat transfer rate by applying several inserts in a tube. Murugesan et al. have assessed the influence of heat transfer rate in a tube fitted with Square–cut TT (STT) insert for the twist ratio 2.0, 4.4, and 6.0. In this study, the highest Nusselt number (*Nu*), frictional factor, and thermal performance factor (TPF) are observed compared to a plain tube and tube inserted with plain twisted tape (PTT) [6]. In another experiment, Murugesan et al. investigated the implication of heat transfer and pressure loss using a circular pipe equipped with and without a V-cut TT. This outcome confirms an improvement in TPF, friction factor, and Nusselt (*Nu*) number at a lower twist ratio (y=2.0), width ratio (WR=0.34), and highest depth ratio (DR=0.43) relative to a simple tube [7]. Besides, the addition of Triangular Perforated Twisted Tape with V-cut greatly impacts the thermal performance factor (TPF), friction factor, *Nu* number, and overall heat transfer coefficient.

A recent study analyzed the effect of pitch (50 mm,100 mm, 120 mm) for TPTTV. In this research, the highest heat transfer coefficient is evaluated for TPTTV, also the best TPF (1.49), *Nu* number, and friction factor are reported with a reduced pitch value of 50 mm, higher than the PTT insert [8]. A computational study has been done to enhance the convective heat transfer coefficient by inserting single, double, and triple twisted tape in a heat exchanger (HX). The analysis found evidence that different types of TT inserts increase the heat transfer rate and the triple twisted tape inserts affect fluid resistance by 21.2 to 13.7 times more than the single and double TT inserts [9].

In a numerical study, researchers have claimed that different twist ratios (TR) of PTT inserts can influence heat transfer parameters. The findings demonstrate that the heat transfer parameters increase with an increment in TR, and a TR of 3.5 provides the best heat transfer rate [10]. To achieve a better heat transfer rate, Salam et al. [11] experimented with a pipe inserted with rectangular cut twisted tape inserts. From the observations, the *Nu* number and frictional factor values (ranging from 39% to 80%) enhanced with the increment of Reynolds (*Re*) number. Another research work has been performed by Bhosale et al. [3] on improving a HX by incorporating rectangular-cut twisted tape inserts in a pipe. In that simulation, the highest *Nu* number and heat transfer coefficient are found at TR 3.426 in a pipe fitted with rectangular-cut TT inserts. Rubbi et al. [12] examined the heat transfer augmentation process in a tube employing twisted tape (TT) with hemispherical extruded surface (HES) for the turbulent flow area at twist ratio (TR) 4.0. Because of the enhanced swirl generation in the flow passage, they identified a 69.4% advancement in the heat transfer rate for the twisted tape (TT) with the hemispherical extruded surface (HES) compared to a plain tube. The positive influence of the thermal performance factor (TPF) was also observed in that research work.

Furthermore, the insertion of helical tape and nail rod in a heat exchanger substantially affects the heat transfer intensification process. Recently an experimental analysis has been done on the heat enhancement process by implanting TT and helical tapes in a double tube HX. A maximum TPF of 3.06 and 219% to 315% higher *Nu* numbers are reported for different twist ratios (TT insert's twist ratio and helical tape insert's twist ratio) in this study [13]. Marzouk and his team numerically and experimentally investigated heat transfer augmentation using nail rod insert in a double tube HX for various pitch lengths (100 mm, 50 mm, 25 mm). They established the highest *Nu* number, pressure drop, and heat transfer enhancement with a reduced pitch length of 25 mm. Due to the larger number of nails at a smaller pitch length (25 mm), better fluid mixing, and turbulence were observed, which significantly improved the heat transfer rate [14]. Hayat and his team conducted a numerical analysis using twisted tape inserts with trapezoid-shaped ribs. The inclination angle of the trapezoidal ribs was also taken into account (ranging from 30° to 60°) in the study with the Reynolds number covering the range of 4000 to 12000. At 30° angle, they noticed the highest thermal performance factor (η), which resulted in superior heat transfer performance [15]. Abdulhamed et al. [16] recently investigated the heat enhancement phenomenon by inserting TT inserts with large holes. They selected air as working fluid and observed the highest *Nu* number and thermal performance factor with the expansion of hole numbers.

From the above literature review, it is inferred that researchers have worked on several TT inserts to improve heat transfer enhancement. They have also clarified the heat augmentation phenomena using rectangular-cut TT inserts, compared to a plain tube. However, no previous study has been done on rectangular-cut TT inserts in a U-loop-shaped tube comparing the heat transfer rate with PTT and a plain tube. Accordingly, this paper aims to address the heat transfer intensification rate implanting rectangular-cut TT inserts in a U-loop pipe by comparing it with PTT and a plain tube. The rectangular cut twisted tape insert enhances heat transfer effectiveness by mixing the fluid with the pipe's closest wall. This rectangular cut TT insert may add a new dimension to the industrial application for its affordability i.e. uncomplicated manufacturing technique. In this research, the considered heat transfer fluid is water, and the Finite Element Method (FEM) is adopted to conduct this numerical simulation. The computational domain and the simulations are performed by the COMSOL Multiphysics software [17]. Besides, the Reynolds number range 5319.4 to 17,288.05 is determined for non-isothermal turbulent flow region. The impact of thermal performance factor (TPF), Nusselt number (*Nu*), and friction factor of rectangular cut TT, PTT, and plain tube are numerically analyzed. The results are evaluated against the experimental study of Salam et al. [11] and observed a significant similarity between the two results.

2. MATHEMATICAL MODEL

The prediction of flow pattern and heat transfer rate can be observed by Computational fluid dynamics (CFD), and that CFD depends on Navier-Stokes equations as they simply analyze the behavior of fluid in various fluid systems. Moreover, CFD is the field where FEM is applied for the discretization. As the Comsol multiphysics software is based on the FEM, we chose to simulate our result with this software. The equation of continuity and the Navier-Stokes equations are shown [18].

$$\frac{\partial\rho}{\partial t}+\nabla\cdot(\rho\mathbf{u})=0 \tag{1}$$

$$\rho(\mathbf{u}\cdot\nabla)\mathbf{u}=\nabla\cdot\begin{bmatrix}-pI+(\mu+\mu_T)(\nabla\mathbf{u}+(\nabla\mathbf{u})^T)\\ -\frac{2}{3}(\mu+\mu_T)(\nabla\cdot\mathbf{u})I-\frac{2}{3}\rho KI\end{bmatrix}+F \tag{2}$$

The *k*-*ω* turbulent model is chosen as it gives better prediction in curved boundaries (U-loop pipe).

$$\rho(\mathbf{u}\cdot\nabla)k=\nabla\cdot[(\mu+\mu_T\sigma_k^*)\nabla_K]+P_k-\rho\beta^*\mathrm{k}\omega \tag{3}$$

$$\rho(\mathbf{u}\cdot\nabla)\omega=\nabla\cdot[(\mu+\mu_T\sigma_\omega)\nabla\omega]+\alpha\frac{\omega}{k}P_k\ \text{-}\rho\beta\omega^2 \tag{4}$$

$$\mu_T=\rho\frac{K}{\omega} \tag{5}$$

$$P_k=\mu_\mathrm{T}[\nabla\mathbf{u}{:}(\nabla\mathbf{u})+(\nabla\mathbf{u})^\mathrm{T}\text{-}\frac{2}{3}(\nabla\cdot\mathbf{u})^2]\text{-}\frac{2}{3}\rho\mathrm{k}\nabla\cdot\mathbf{u} \tag{6}$$

Eq. (7) represents the heat transfer rate along the flow:

$$\rho C_p\frac{\partial T}{\partial t}+\rho C_p\mathbf{u}\cdot\nabla T=\nabla\cdot(k\nabla T)+\mathbf{Q} \tag{7}$$

In Eq. (7), Q is a heat source, where, K is the thermal conductivity.

$$\rho C_p\frac{\partial T}{\partial t}+\nabla\cdot(-k\nabla T)=\mathbf{Q} \tag{8}$$

$$h=\frac{\mathbf{Q}}{T_w-T_b} \tag{9}$$

where, the wall and bulk temperature are denoted by T_w and T_b respectively $T_b=\frac{T_{out}+T_{in}}{2}$. The Nusselt number is determined as:

$$Nu=\frac{hD}{K} \tag{10}$$

D is the pipe diameter and the Darcy friction factor can be found as [19, 20], where pressure loss and head loss are Δp and h.

$$\Delta P = h\rho g \tag{11}$$

$$h = \frac{flu^2}{2gD} \tag{12}$$

Finally, we have (13) using Eqs. (11) and (12):

$$\Delta p = \frac{l}{D}\cdot\frac{fu^2\rho}{2} \tag{13}$$

Thermal performance factor calculated by using the equation:

$$\eta = \frac{\frac{Nu}{Nu_0}}{\left(\frac{f}{f_0}\right)^{\left(\frac{1}{3}\right)}} \tag{14}$$

The Nu number and the friction factor of the smooth pipe are expressed as Nu_0 and f_0 respectively [21].

2.1 Boundary conditions

This approach sets the following boundary conditions for various velocities. Initially, fluid temperature is set to $T=T_{in}=293.15K$, and the wall function conditions are assumed by:

$$\mathbf{u}\cdot\mathbf{n}=0 \tag{15}$$

$$\begin{bmatrix} (\mu+\mu_T)(\nabla\mathbf{u}+(\nabla\mathbf{u})^T) \\ -\frac{2}{3}(\mu+\mu_T)(\nabla\cdot\mathbf{u})I-\frac{2}{3}\rho kI \end{bmatrix}\mathbf{n} = -\rho\frac{u_\tau}{\delta_w}\mathbf{u}_{\tan g} \tag{16}$$

$$\mathbf{u_{tang}} = \mathbf{u}-(\mathbf{u}\cdot\mathbf{n})\mathbf{n} \tag{17}$$

$$\nabla k\cdot\mathbf{n}=0, \omega=-\rho\frac{c_\mu K^2}{K_v\delta_w\mu} \tag{18}$$

3. COMPUTATIONAL DOMAIN AND MESH DESIGN

A U-loop-shaped pipe installed with rectangular cut twisted tape (TT) insert and plain TT insert is used in this study whose length is 1935.62 mm and inner diameter is 26.597 mm. We have run our simulations by altering the mesh design to achieve better results. Accordingly, this computational model confirms the best result for finer mesh. The mesh design of a U-loop-shaped pipe, a pipe fitted with plain TT insert, and rectangular cut TT insert are demonstrated in Figure 1 (a), (b), (c). Moreover, the comparison of the finer mesh elements and temperature distributions of three different inserts is shown in Table 1.

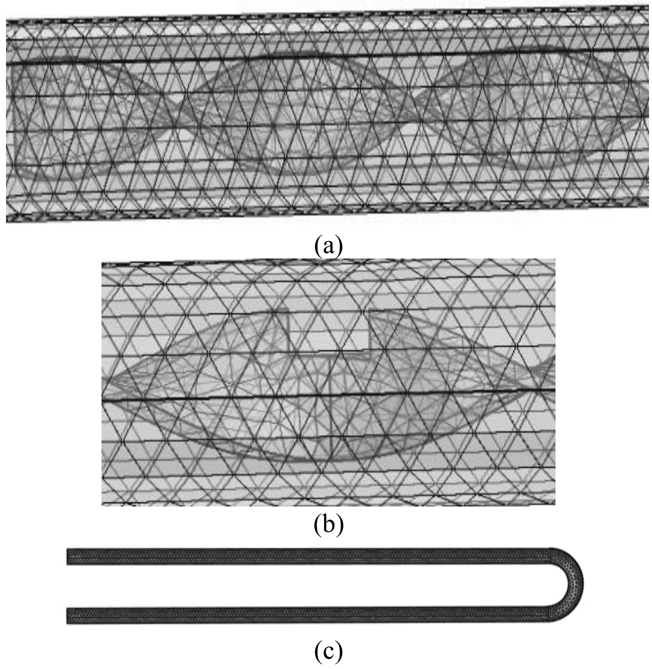


Figure 1. (a) Mesh design of a tube fitted with plain twisted tape (b) tube fitted with rectangular cut twisted tape (c) U-looped tube with insert

Table 1. Finer mesh comparison of three different inserts

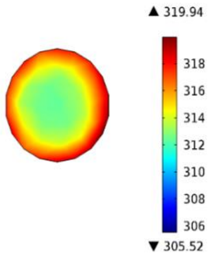
Property Name	Plain Tube	Plain Twist	Rectangular Cut
Tetrahedral elements	50273	87870	119455
Triangular elements	12842	23600	26550
Edge elements	764	2952	3472
Vertex elements	12	84	332
Minimum element quality	0.2337	0.07282	0.02637
Average element quality	0.7445	0.7048	0.7038
Element volume ratio	0.05427	0.01276	0.002795

4. NUMERICAL RESULTS

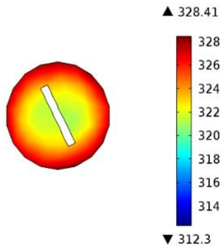
The result of this analysis is then compared with the experimental study of Chowdhury et al. [10]. To get a higher heat transfer rate, they analyzed their data over a range of Reynolds (Re) numbers from 10,000 to 19,070 and found the highest Nusselt (Nu) number to be 309 at Re=19,070 for a pipe incorporated with rectangular cut twisted tape insert. Accordingly, in this numerical analysis, the Reynolds number span is considered 5,319.4 to 17,288.05, and the best Nu number observed is 357.59 at Re=17,288.05 for a rectangular cut TT insert. Moreover, the friction factor influence is compared with the experiment of Salam et al. [11], where the friction factor reduces with the increment of the Re number in every case of TT insert. A better friction factor is found for the rectangular cut TT insert in Salam et al.'s [11] experimental and in our numerical studies. From these two studies, it is inferred that the rectangular cut TT insert is responsible for a higher heat transfer enhancement rate, which facilitates the vortex flow and fluid disturbance in the pipe's inner wall. On the other hand, the deviation of this numerical analysis could be easily understood if we performed this study experimentally.

4.1 Temperature distribution

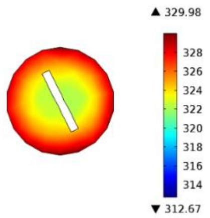
Figure 2 shows the cross-sectional view of temperature distribution and Figure 3 shows the temperature variation against the *Re* number for the pipe inserted with rectangular cut TT insert, plain TT, and plain tube. It is detected that the temperature diminishes with the increment of the *Re* number and the highest temperature is found for rectangular cut twisted tape (TT) insert than others.



(a)



(b)



(c)

Figure 2. Temperature distribution for (a) plain tube (b) plain twisted tape (c) rectangular cut twisted tape inserts

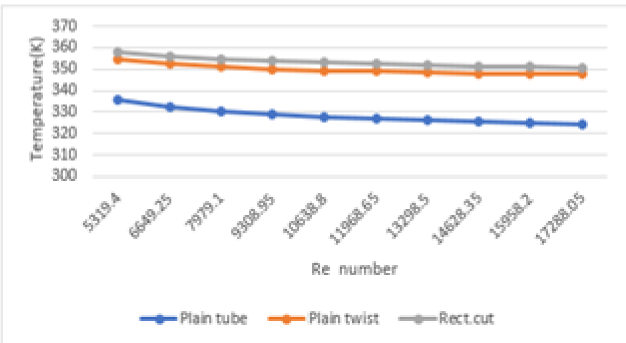


Figure 3. Temperature vs Reynolds number

4.2 Nusselt number

Figure 4 indicates the *Nu* number with respect to the *Re* number for the cases of rect. cut TT insert, plain TT insert,and plain tube. The outcomes show that the *Nu* number enhancess with the increment of Reynolds *(Re)* number. Superior results are observed for the tubular pipe fitted with rect. cut TT insert compared to the other two pipes used in this study. Generally, the dominance of convective heat transfer is characterized by the Nusselt number. So, due to higher convective heat transfer coefficient in the turbulent flow passage, the *Nu* number increases with the *Re* number.

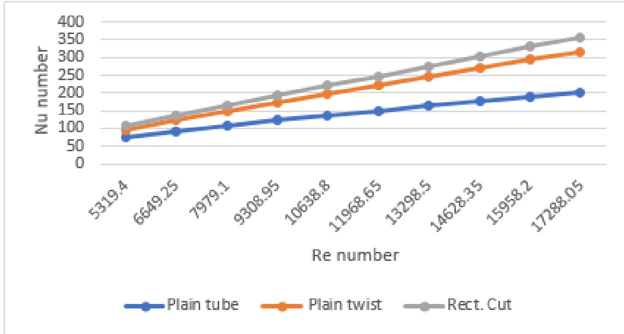


Figure 4. *Nu* number vs Reynolds number

4.3 Nusselt number enhancement index

In Figure 5, the *Nu* number enhancement index is observed against the *Re* number for rectangular cut, plain TT, and plain tube. It is found that the enhancement ratio for the rectangular cut TT insert is more significant than the plain tube and plain twisted tape also the *Nu* number enhancement ratio increases with the increment of Reynolds numbers. This indicates the positive influence of convective heat transfer in the turbulent flow region of the pipe arranged with rectangular cut TT insert.

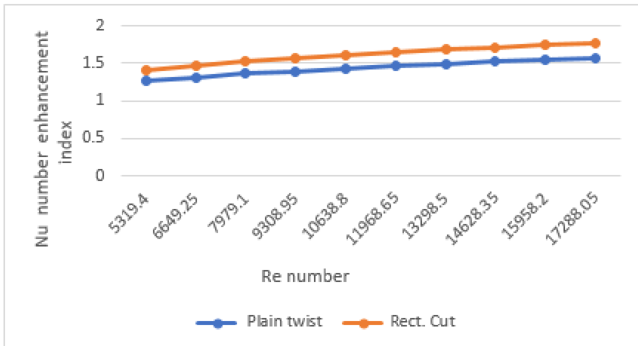


Figure 5. Nusselt number enhancement index vs Reynolds number

4.4 Friction factor

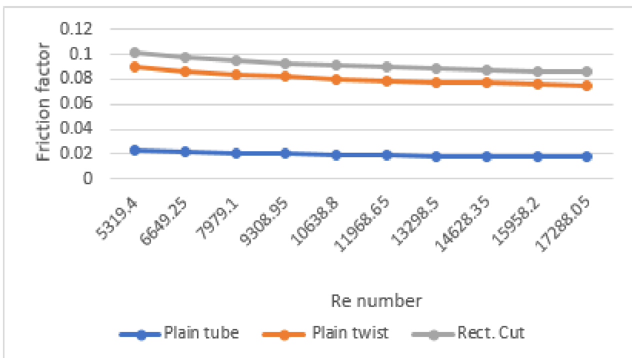


Figure 6. Friction factor vs Reynolds number

Figure 6 illustrates the decline in the friction factor (*f*) as the Reynolds (*Re*) number enhances for TT (twisted tape) without and with rect. cut and plain tube. The friction factor, 0.1 is observed at *Re* = 5319.4 for the rectangular cut twisted tape insert which yields increasingly effective results than PTT (plain twisted tape) and plain tube. It is apparent from the graph that the numerical value of friction factor (*f*) lessens with the enhancement of Reynolds numbers. This is because intense fluid disturbance in the pipe's inner wall that decreases the fluid resistance.

4.5 Friction factor enhancement index

The same trend for the friction factor enhancement index is observed in Figure 7. We found that the effect of the friction factor enhancement index for rectangular cut twisted tape insert is more dominant than the other two inserts (plain twisted tape and plain tube). The friction factor enhancement index indicates less pressure loss in the flow area which is crucial for better heat transfer enhancement.

Figure 7. Friction factor enhancement index vs Reynolds number

4.6 Thermal performance factor (TPF)

The thermal performance comparison at different Reynolds numbers for various inserts (rectangular cut, plain TT, and plain tube) are plotted in Figure 8. The present finding confirms that the TPF value is significantly higher for rectangular cut TT inserts than plain TT and plain tube. Moreover, compared to the other inserts, the best efficiency (η =1.04) is observed at *Re* = 17,288.05 for the tube incorporated with rectangular cut TT insert. This indicates that the swirl flow in the rectangular cut TT insert's flow region enhanced the heat transfer and improved device efficiency.

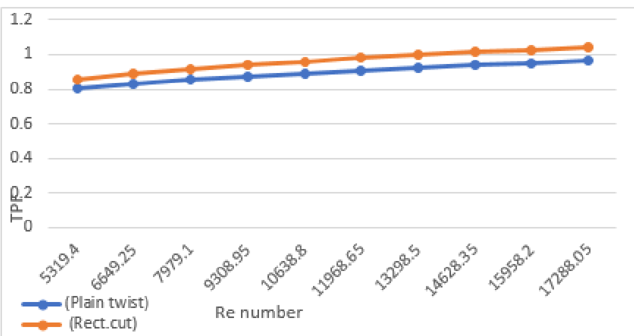


Figure 8. Thermal performance factor vs *Re* number

5. CONCLUSIONS

The heat transfer enhancement phenomena were conducted for the cases of U-loop tubes fitted with rectangular cut TT insert, plain TT, and plain tube insert in the turbulent flow region. Key findings from our simulation are:

i. The presence of rectangular cut TT inserts significantly enhanced the heat transfer rate than plain TT and plain tube. This is due to the occurrence of excessive turbulence and vortex flows in the flow channel of the pipe.
ii. The *Nu* number is observed as 357.59 for the rectangular cut TT insert which is higher than plain TT (*Nu* = 317.23) and plain tube (*Nu* = 202.06).
iii. The friction factor value is found at *Re* = 5319.4 for the rectangular cut TT insert compared to plain TT and plain tube.
iv. The highest TPF value η = 1.04 is tested for the rectangular cut TT insert than other inserts (plain TT and plain tube).

By observing our findings we may conclude that one can enhance the heat transfer rate by considering other turbulent models together with different shapes of inserts for the same pipe geometry.

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NOMENCLATURE

l	length of the pipe
C_P	specific heat, J. kg^{-1}. K^{-1}
ΔP	pressure drop
f	friction factor (-)
Nu	Nusselt number (-)
Re	Reynolds number (-)
Ti	inlet temperature (K)
To	outlet temperature (K)
Tb	bulk temperature (K)

Greek symbols

η	thermal enhancement factor
ρ	density of water