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# CFD study of heat transfer enhancement and fluid flow characteristics of turbulent flow through tube with twisted tape inserts

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**Abstract**

A three-dimensional computational conjugate heat and mass transfer study has been carried out using computational fluid dynamics (CFD) software package ANSYS FLUENT to investigate the effect of insert's twist ratio on the heat transfer and fluid flow performance. Investigation was carried out for air flow at 300 Kelvin and Reynolds number ranging from 3642 to 21857 through a tube with constant wall heat flux of 8000 W/m$^2$. Validating against Gneilski and Petukhob models, the current model has been used to investigate the effect of insert with twist ratio 3.46 and 7.6 on Nusselt number, friction factor and thermal performance factor of the tube. Results show that for twist ratio of 3.46, Nusselt numbers and friction factors are increased by 20% to 62% and 185% to 245% respectively, and thermal performance factor ranged between 0.9 and 1.2. Those were observed to be increased by 10% to 30%, 128% to 183% and ranged between 0.95 to 1.05 respectively for twist ratio of 7.6. It is concluded that twisted tapes provide better heat transfer enhancement at relatively lower Reynolds number and twist ratio.



*Keywords:* Heat transfer enhancement; twisted tape insert; computational fluid dynamics; Fluent.

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## 1. Introduction

Energy efficiency of a thermal system like heat exchanger is a great concern for economic running of industry, cost saving and environment protection. For that reason, heat transfer and fluid flow characteristics improvement of heat exchangers are researched worldwide. To improve the heat transfer characteristics of tube side convection heat transfer, different types of active and passive techniques have been used. Among the passive technique's, implementation of inserts inside the tube as a swirl generator is of great interest. It reduces the thermal boundary thickness, increases residence time by swirling the flow and normal gradient of velocity. As a result, heat transfer coefficient, Nusselt number increases with a penalty from rise of pressure. Eiamsa-ard et al. [1] investigated the effect of helical tape inserts with centered rod, without centered rod and regularly spaced helical rod in tube side heat transfer for double pipe counter flow heat exchanger and found that helical tape with insert have higher heat transfer coefficient than others but greater friction increase. Garcia et al. [2] studied three different wire coils insert and twisted tape insert for solar collector and found with the increase of heat transfer coefficient, decrease of wall temperature. Bhuiya et al. [3] experimented heat transfer enhancement by helical tape inserts for different twist ratios in turbulent flow regime and found that with the decrease of twist ratio, increase in heat transfer coefficient with pressure drop rising but energy savings overall. Experimental investigation for heat transfer enhancement in u-tube with $Al_2O_3$ nanofluid using twisted tape insert was done by Prasad and Gupta [4]. Guo et al. [5] investigated the heat transfer enhancement by center cleared twisted tapes in laminar flow regime for different width ratios of twisted tape and reported increase of thermal performance factor with the increase of Reynolds Number.

Besides experimental study, numerical study was also carried out. Bellos et al. [6] numerically investigated the contribution of internally finned absorber, twisted tape inserts and perforated plate inserts for enhancement of heat transfer in parabolic troughs receiver and stated for internally finned absorbers best performance. Though the enhancement was little compared to the enhancement provided by these inserts at uniform heat flux condition at the tube profile (parabolic troughs has non-uniform heat flux at receiver surface). Salman et al. [7] did CFD study of heat transfer enhancement by classical twisted tape and parabolic cut twisted tape using Water-Copper Oxide nanofluid and reported Parabolic cut twisted tape insert best performance. In CFD study of full length, half-length upstream, half-length downstream twisted tape insert at turbulent flow by Yadav et al. [8] revealed that full length twisted tape insert has best performance, later half-length downstream twisted tape has better performance over half-length upstream twisted tape. Said et al. [9] numerically investigated helical flow duct plate heat exchanger by varying pitch ratios and channel cross section aspect ratios and stated highest effectiveness of 67.56 for 0.24 pitch ratio. Bhuyan et al. [10] numerically investigated heat transfer characteristics of full length and short length twisted tape inserts in tubular u-loop pipes at laminar flow for transient conditions using uniform heat flux and stated highest outlet temperature for full length insert then short length insert compared to plain tube. CFD study of loose fit twisted tape insert was done by Piriyarangroj et al. [11] and it showed better performance for 0.05 clearance ratio than any other clearance ratio. Sharifi et al. [12] in their CFD study investigated the performance of coiled wire insert for laminar flow using structured hexahedral mesh.

From these it is visible that twisted tape has significant impact on heat transfer enhancement and properly validated CFD study is suitable tool for performing investigation on different types of insert. For this reason, CFD analysis of twisted tape insert for two different twist ratios is done for turbulent flow using air as working fluid. Heat transfer coefficient, friction factor and thermal performance factor is evaluated in this study.

## 2. Materials and methods

### *2.1. Physical model*

The designed model has a fluid domain of 900mm length and 26.6mm diameter where an insert of 800mm length is placed after 50mm from inlet and ends before 50 mm of outlet of the fluid domain as shown in Fig. 1(a). Fig. 1(b) shows only the fluid domain. Two Plain Twisted Tape (PTT) inserts of 3.46 and 7.6 twist ratio (TR = y/w see Fig. 1(c)) were investigated at Reynolds number range of 3642 to 21857. Both inserts have thickness of 1mm and width of 21mm. Here twisted tape acts as swirl generator. Conduction heat transfer through insert is neglected to reduce the complexity. So, both fluid domain and insert was designed in SOLIDWORKS and later imported to ANSYS for CFD study. At first insert domain is cutout using Boolean operation to reduce the computational domain size. Then boundary condition was imposed and solved for both plain tube and tube with insert. The properties of air used for this simulation is described in Table 1.

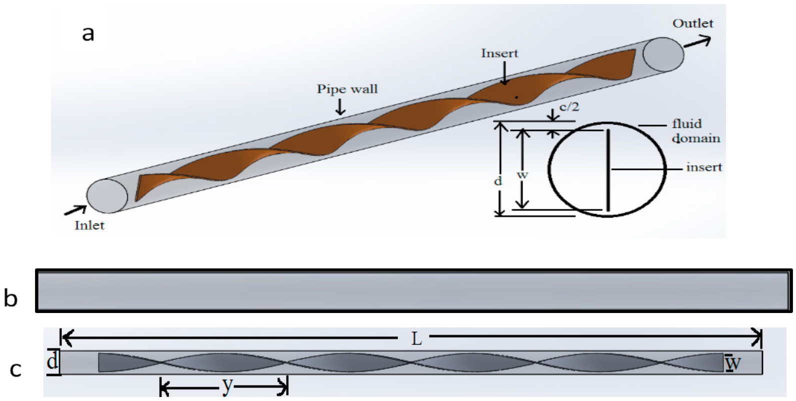


Table 1. Properties of air at simulated conditions

| Properties | Value |
|---|---|
| Prandtl number, Pr | 0.744 |
| Thermal conductivity, k | 0.0242 W/m-k |
| Specific heat, $C_p$ | 1006.43 J/kg-k |
| Air density, ρ | 1.225 kg/m$^3$ |
| Dynamic viscosity, μ | 1.7894×10^-5 kg/m-s |

Fig. 1. (a) Schematic of the 3D fluid domain with insert; (b) Plain tube without insert; (c) Plain tube with insert

## *2.2. Mathematical modelling and data reduction*

Governing equations like energy, continuity and momentum equations are similar to the study of Eiamsa-ard et al. [13]. Data reduction are described below.

Nusselt number, Nu is calculated using equation (1) where d is pipe diameter.

$$Nu = \frac{hd}{k} \quad (1)$$

Friction factor, f is calculated using equation (2) where Δp is pressure drop, V is the velocity of air and L is the pipe length.

$$f = \frac{\Delta p}{((L/d)(\rho V^2/2))} \quad (2)$$

Theoretical Nusselt number for plain tube is calculated using Gneilski correlation [14].

$$Nu = 0.0214 * (Re\text{^}0.8 - 100) * Pr\text{^}0.4 \quad (3)$$

Theoretical friction factor for plain tube is calculated using Petukhov correlation [15].

$$f = (0.79 ln\, Re - 1.64)^{-2} \quad (4)$$

## *2.3. Boundary conditions*

At inlet, velocity inlet condition was implemented where air enters with 300k temperature in correspondent velocity. At outlet, pressure outlet condition was given with zero-gauge pressure for all simulation. So, mass weighted average of pressure at inlet gives pressure drop in this system. Simulations were run at a constant heat flux of 8000 W/m2 at the pipe wall for both plain tube (with no insert) and tube with insert. The wall between fluid and insert is assumed adiabatic (no conduction heat transfer). Navier-Stokes equation coupled with energy equation and Shear Stress Transport (SST) k-ω two equations were modelled and solved.

*2.4. Solution method and convergence criteria*

Steady state, gravity independent (since model is horizontal) analysis was done. Semi Implicit Pressure Linked Equations (SIMPLE) was used as pressure velocity coupling method for this simulation. Second order upwinding was selected for continuity, momentum and energy to check the variables inside each cell. Under relaxation factor and other parameters were set to default value. Simulation were performed till continuity and energy residue reached 10-3 and 10-5. Computer with 8 GB DDR3 RAM, 4 thread core i5 processor and dedicated Graphics Card was used for calculation. Each solution procedure on average took 5 hours.

*2.5. Model validation*

*2.5.1. Grid Sensitivity Check*

Simulations are done for plain tube and tube with insert for different number of mesh elements using maximum face size of 1mm, 2mm and 3mm of mesh element with inflation layers at the boundary to check for grid sensitivity. The total number of mesh elements for these tests are 292877, 699068, 2516594 for plain tube and 231040, 478834, 3289401 for tube with insert. Grid sensitivity are investigated using numerical results of outlet temperature of the bulk air for inlet velocity of 12 m/s, temperature 300k, constant heat flux of 8000 W/m$^2$ at the pipe wall. The outlet temperature at this boundary condition and same flow rate should be same for plain tube or tube with insert due to equal amount of heat addition. The results in Fig. 2 showed very little variation for simulated temperature with theoretical temperature at different mesh sizes. Considering computation time and solution accuracy 2mm maximum face size for mesh element were selected for both plain tube and tube with insert.

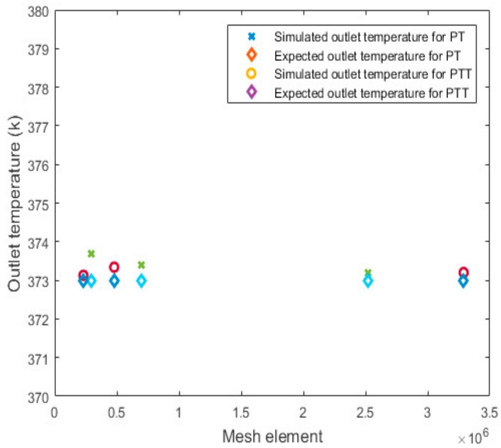


Fig. 2. Mesh sensitivity check

*2.5.2. Validation of simulation*

Validity of the simulated results are checked for plain tube using wide accepted Gneilski [14] correlation for comparison of Nusselt number and Petukhov correlation [15] for friction factor. Simulated Nusselt number showed satisfactory agreement data ranging between -20.29% to 10.19% with Gneilski 1976 correlation and r.m.s. error of 6.64% with its value. Simulated friction factors also showed satisfactory agreement with Petukhov 1970 model with data ranging between -12.88% to 3.9% and r.m.s. error of 3.93% to its value.

## 3. Heat transfer characteristics

Since constant heat flux on the pipe wall is maintained for plain tube (PT) and PTT equal amount heat is added for all the cases. In PT cases fluid near the pipe wall have no velocity and thermal boundary layer is thick where as in PTT, insert swirls the flow and dissipates heat from the surface quickly. It also reduces the thermal boundary layer thickness and gives rapid mixing of fluid. As a result, average surface temperature of the pipe wall for PTT is less then PT. Since heat addition is constant for all the cases, for same flow rate of air in PT and PTT, outlet temperature of the bulk fluid will be the same. Thus, temperature difference between pipe surface and bulk fluid is less for PTT case then PT which means increase in heat transfer coefficient. Fig. 3(a). represents heat transfer coefficient with respect to Reynolds number. It is observed that with increase of Reynolds number heat transfer coefficient increased. Fig. 3(b). represents heat transfer coefficient ratio with Reynolds number. Though the numerical value of heat transfer coefficient was increasing in both PTT and PT cases but ratio of PTT and PT case was decreasing with increasing

Reynolds number meaning reduction in heat transfer enhancement. Fig. 4(a) and 4(b) represents Nusselt number and Nusselt number ratio with respect to Reynolds number respectively. Increase in heat transfer coefficient also increases Nusselt number and the trends of Nusselt number and Nusselt number ratio with Reynolds number is similar to that of heat transfer coefficient. From Fig. 5(a) and 5(b), PT has higher temperature near to pipe wall and a higher portion in the middle of the cross section has lower temperature whereas, in PTT of 3.46 TR it has good mixing of fluid due to insert thus relatively uniform temperature distribution. In Fig. 5(c) and 5(d), PT pipe wall has higher temperature because thick thermal boundary layer of fluid has higher temperature adjacent to pipe wall but PTT pipe wall has relatively lower temperature than PT because it has thin thermal boundary layer thickness and good heat dissipation at boundary due to swirl. PTT pipe wall also have twisted temperature profile of higher and lower temperature depending upon insert geometry.

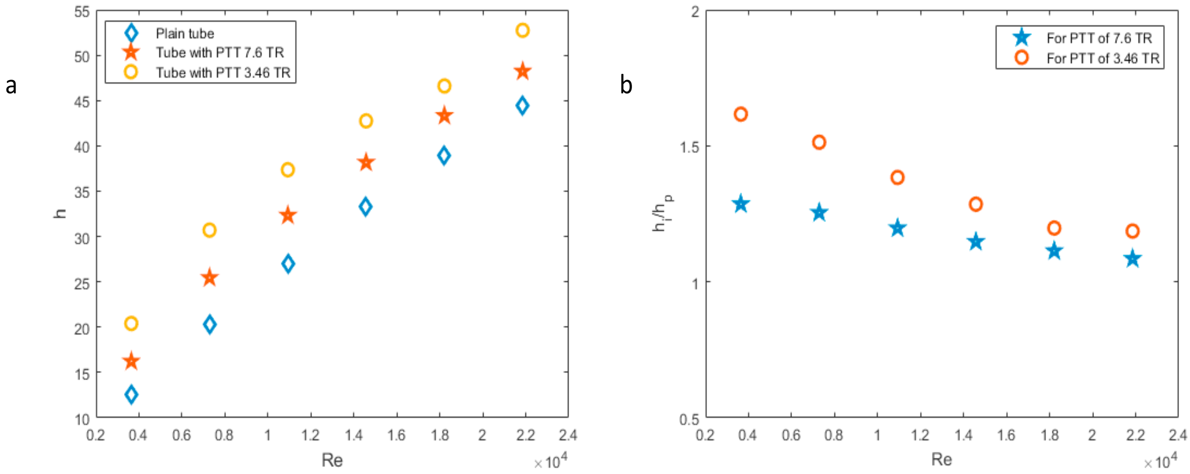


Fig. 3. Relationship between (a) Heat transfer coefficient vs. Re (b) heat transfer coefficient ratio vs Re

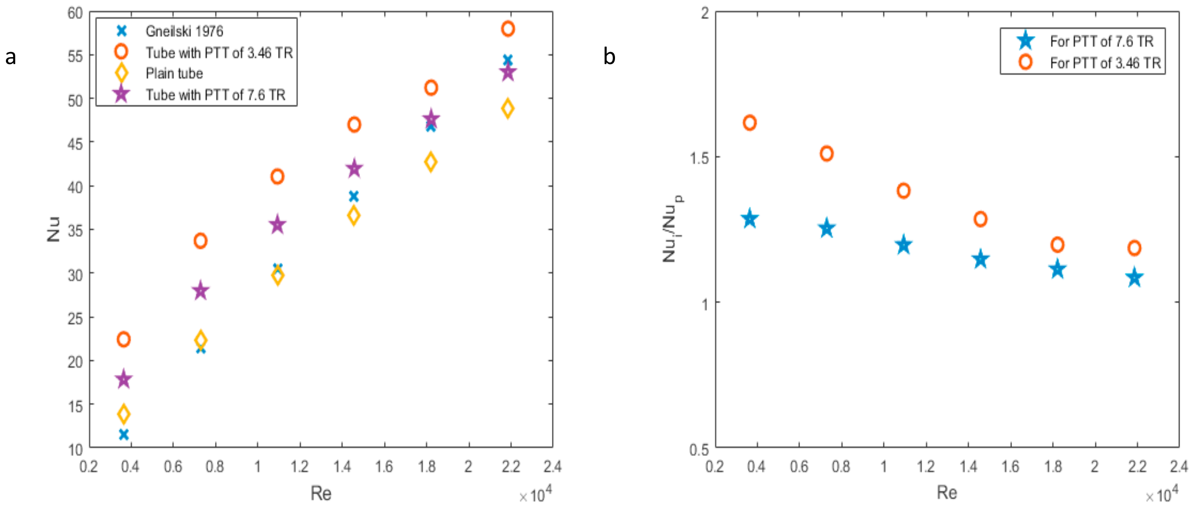


Fig. 4. Relationship between (a) Nusselt number vs. Re (b) Nusselt number ratio vs Re

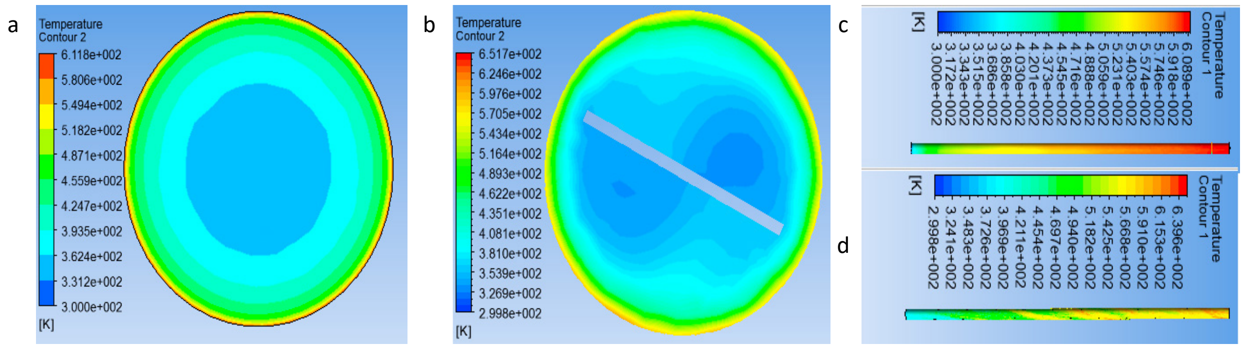


Fig. 5. Temperature profile at Re=18000 for (a) PT outlet (b) PTT of 3.46 TR outlet (c) pipe wall of PT (d) pipe wall pf PTT of 3.46 TR

## 4. Pressure drop characteristics

Fig. 6 explains the effect of twist ratio on friction factor with the increase of Reynolds number. From Fig. 6(a) it is observed that friction factor decreases with the increase of Reynolds number and the trend is similar for plain tube and both tube with insert. Tube with insert have higher friction factor than plain tubes. This is due to the flow restriction, higher surface area, turbulent kinetic energy and eddy viscosity. Friction factor increases with the decrease of twist ratio. This is due to the strong swirl generation and longer residence time in case of lower twist ratios [3]. Friction factor ratio also decreases with the increase of Reynolds number. Friction factor increased 185-245% and 128-183% for use of insert of 3.46 and 7.6 TR than plain tube.

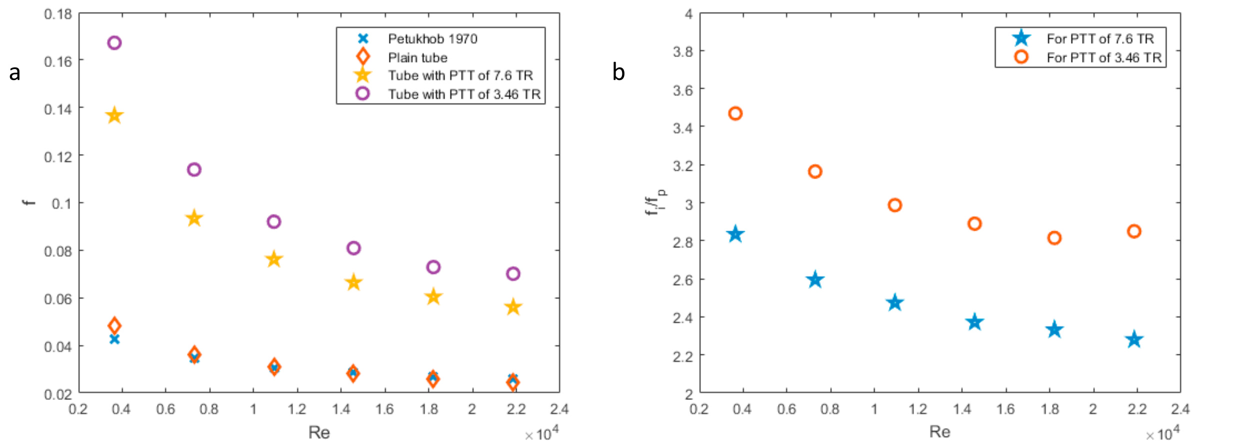


Fig. 6. Relationship between (a) friction factor vs. Re (b) friction factor ratio vs Re

## 5. Thermal performance factor

Thermal performance factor is the enhancement of heat transfer obtained by using insert and is the ratio of Nusselt number and friction factor. However, to quantify the energy savings by using inserts it needs to be compared for heat transfer enhancement at equal pumping power to that of plain tube following below procedure [16-18].

For constant pumping power assessment, where i and s denotes insert fitted and plain tube.

$$(Q\Delta P)_i = (Q\Delta P)_s \tag{5}$$

From the relationship between the friction factor and the Reynolds number, it is described as

$$(fRe^3)_i = (fRe^3)_s \tag{6}$$

$$Re_s = Re_i \left(\frac{f_i}{f_s}\right)^{1/3} \tag{7}$$

Simulated thermal performance factor or heat transfer enhancement factor is obtained using

$$\eta = \frac{\left(\frac{Nu_i}{Nu_s}\right)}{\left(\frac{f_i}{f_s}\right)^{1/3}} \tag{8}$$

Following the procedure above, thermal performance factor, $\eta_{TR}$ for PTT with 3.46 TR becomes

$$\eta_{TR=3.46} = Re^{-0.1544} pr^{3.9586} \left(\frac{y}{w}\right)^{2.1044} \tag{9}$$

thermal performance factor for PTT with 7.6 TR becomes

$$\eta_{TR=7.6} = Re^{-0.0657} pr^{7.1936} \left(\frac{y}{w}\right)^{1.3096} \tag{10}$$

From eqn. 9 and eqn. 10 with relative to Reynolds number thermal performance factor for these two twist ratios are calculated which are shown in Fig. 7. It is observed that thermal performance factor decreases with the increase of Reynolds number and increases with the decrease of twist ratio. Here, thermal performance factor is greater than unity for lower Reynolds number. It ranged between 0.9-1.2 and 0.95-1.05 for TR of 3.46 and 7.6 when compared to plain tube. Eiamsa-ard et al. [18] in a near similar study obtained close result for Nusselt number and thermal performance factor for short length twisted tape insert.

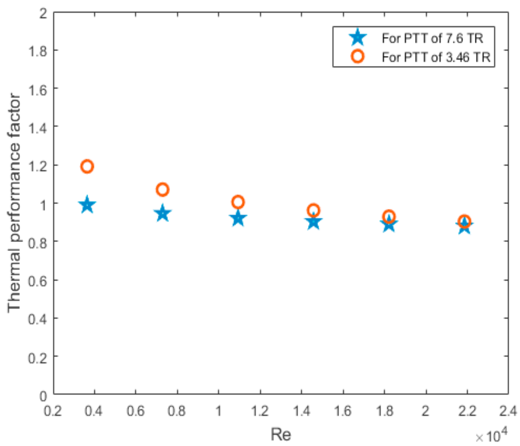


Fig. 7. Relationship between thermal performance factor vs. Re

## 6. Conclusions

A CFD study of plain twisted tape insert (PTT) of 3.46 and 7.6 twist ratio (TR) in tube was done for turbulent flow as heat transfer enhancer using a commercial software package. This insert augmented the rate of heat transfer with increase in pressure drop. The findings of the study are:

- Nusselt number and friction factor increased with the increase of Reynolds number (3600<Re<22000). Nusselt number increased 20 to 62% and 10 to 30 % for use of insert of 3.46 and 7.6 TR than plain tube. Friction factor increased 185 to 245%and 128 to 183% for use of insert of 3.46 and 7.6 TR than plain tube.
- Twisted tape with lower twist ratio has higher level of enhancement with corresponding increase in pressure drop.
- Thermal performance factor is greater than unity for lower Reynolds number. It ranged between 0.9-1.2 and 0.95-1.05 for TR of 3.46 and 7.6.
- Pipe wall temperature is reduced in case of insert fitted tube meaning lower irreversibility and higher heat transfer characteristics. It has twisted contour of relatively higher and lower temperature like the twist of the insert mainly caused by velocity in the region effected by twisted tape.

## References


[1] Eiamsa-ard S, Promvonge P. Enhancement of heat transfer in a tube with regularly-spaced helical tape swirl generators 2005;78:483–94. doi:10.1016/j.solener.2004.09.021.

[2] García A, Herrero-Martin R, Solano JP, Pérez-García J. The role of insert devices on enhancing heat transfer in a flat-plate solar water collector. Appl Therm Eng 2018;132:479–89. doi:10.1016/j.applthermaleng.2017.12.090.

[3] Bhuiya MMK, Chowdhury MSU, Ahamed JU, Azad AK. Heat transfer performance evaluation and prediction of correlation for turbulent flow through a tube with helical tape inserts at higher Reynolds number. Heat Mass Transf 2016;52:1219–30. doi:10.1007/s00231-015-1643-y.

[4] Durga Prasad P V., Gupta AVSSKS. Experimental investigation on enhancement of heat transfer using Al2O3/water nanofluid in a u-tube with twisted tape inserts. Int Commun Heat Mass Transf 2016;75:154–61. doi:10.1016/j.icheatmasstransfer.2016.03.019.

[5] Guo J, Fan A, Zhang X, Liu W. A numerical study on heat transfer and friction factor characteristics of laminar flow in a circular tube fitted with center-cleared twisted tape. Int J Therm Sci 2011;50:1263–70. doi:10.1016/j.ijthermalsci.2011.02.010.

[6] Bellos E, Tzivanidis C. Enhancing the performance of evacuated and non-evacuated parabolic trough collectors using twisted tape inserts, perforated plate inserts and internally finned absorber. Energies 2018;11. doi:10.3390/en11051129.

[7] Salman SD, Kadhum AAH, Takriff MS, Mohamad AB. Heat transfer enhancement of laminar nanofluids flow in a circular tube fitted with parabolic-cut twisted tape inserts. Sci World J 2014;2014. doi:10.1155/2014/543231.

[8] Yadav RJ, Padalkar AS. CFD Analysis for Heat Transfer Enhancement inside a Circular Tube with Half-Length Upstream and Half-Length Downstream Twisted Tape. J Thermodyn 2012;2012. doi:10.1155/2012/580593.

[9] El-said EMS, Abdulaziz M, Awad MM. Plate Heat Exchanger Using Helical Flow Duct. Cogent Eng 2017.

[10] Bhuyan M, Deb UK, Shahriar M, Acherjee S. Simulation of Heat Transfer in a Tubular Pipe Using Different Twisted Tape Inserts 2017:397–409. doi:10.4236/ojfd.2017.73027.

[11] Piriyarungroj N, Eiamsa-ard S, Promvonge P. Simulation of Turbulent Heat Transfer in a Tube Fitted with Twisted Tape Placed Separately from the Tube Walls 2015;751:245–50. doi:10.4028/www.scientific.net/AMM.751.245.

[12] Sharifi K, Sabeti M, Rafiei M, Mohammadi AH, Shirazi L. Computational Fluid Dynamics ( CFD ) Technique to Study the Effects of Helical Wire Inserts on Heat Transfer and Pressure Drop in a Double Pipe Heat Exchanger. Appl Therm Eng 2017. doi:10.1016/j.applthermaleng.2017.08.146.

[13] Eiamsa-ard S, Wongcharee K, Sripattanapipat S. 3-D Numerical simulation of swirling fl ow and convective heat transfer in a circular tube induced by means of loose- fi t twisted tapes ☆. Int Commun Heat Mass Transf 2009;36:947–55. doi:10.1016/j.icheatmasstransfer.2009.06.014.

[14] Gneilski V. New equations for heat and mass transfer in turbulent pipe flow and channel flow. Int Chem Eng 1976;16:359–68.

[15] Petukhov BS. Heat Transfer and Friction in Turbulent Pipe Flow with Variable Physical Properties. Adv Heat Transf 1970;6:503–64. doi:10.1016/S0065-2717(08)70153-9.

[16] Salam B, Biswas S, Saha S, Bhuiya MMK. Heat transfer enhancement in a tube using rectangular-cut twisted tape insert. Procedia Eng 2013;56:96–103. doi:10.1016/j.proeng.2013.03.094.

[17] Webb RL, Kim N. Principles of Enhanced Heat Transfer. 2nd Edition. Boca Raton: Taylor & Francis.

[18] Eiamsa-ard S, Thianpong C, Eiamsa-ard P, Promvonge P. Convective heat transfer in a circular tube with short-length twisted tape insert ☆. Int Commun Heat Mass Transf 2009;36:365–71. doi:10.1016/j.icheatmasstransfer.2009.01.006.