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CFD study of heat transfer enhancement and fluid flow characteristics of laminar flow through tube with helical screw tape insert

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Abstract

In this study, a three-dimensional (3D) computational fluid dynamics (CFD) analysis is performed to investigate the heat transfer performance and fluid flow characteristics using a helical screw tape insert in pipe flow. An inserted tube geometry was improved using a helical coil i.e., wire-wrapped with 1.92 twist ratio. Single-phase, horizontal, and a laminar flow ($200 < Re < 2300$) through an annular channel is considered. A flow domain is created first, and then discretized, later all boundary conditions are applied and finally, simulated it using ANSYS FLUENT. The results are analyzed and processed further to check for mesh sensitivity and experimental validation. The simulated results showed that the heat transfer rate in terms of Nusselt number increased 1.34-2.6 times, whereas, the friction factor also increased 3.5-8 times for wire-wrapped-tube i.e., the tube with helical screw tape insert in comparison to the plain tube. The thermal performance factor was evaluated and the maximum value is found 3.79 at a constant pumping power. The pressure drop was estimated which was increased due to the increased flow restriction by inserted coils.

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1. Introduction

Heat exchangers thermal performance are of great importance for efficient economic running of industrial machineries. There are many active and passive techniques such as fluid passage modification, addition of swirl generators, mixing of nanofluid to enhance the thermal performance of tube side convection heat transfer of heat exchangers. These techniques reduce plant irreversibility, increase quality and amount of heat transfer and better fluid flow characteristics. Researchers around the world investigated the thermal performance enhancement of convective heat transfer in tubes using different types of insert as a fluid passage modifier [1-4]. Most of the insert geometry works as a swirl generator, inducer of mixing and turbulence and enhancer of residence time, normal gradients of velocity.

Bhuiya et al. [2] studied the heat and fluid flow performance of tube with helical tape inserts for turbulent flow at higher Reynolds number and reported increase of Nusselt number and friction factor up to 260 and 285% respectively than plain tubes. Thermal performance was reported to be 44% higher than plain tube at constant blower power. Eiamsa-ard et al. [3] used counter twisted tapes for counter flow, co-twisted tapes for co-swirl flow and single twisted tape. Experiments showed better performance for counter twisted tapes than others and single twisted tape have better performance over co-swirl flow. Salam et al. [4] experimentally investigated performance of rectangular cut twisted tape insert at turbulent flow using different flux conditions and concluded increase of Nusselt number 2.3 to 2.9 times with rise of friction factor 1.4 to 1.8 times, enhancing thermal performance factor 1.9 to 2.3 times. Gunes et al. [5] experimented tube with coiled wire insert for turbulent flow under uniform heat flux condition. It revealed higher heat transfer for lower pitch ratios of insert with increase in pressure drop.

Besides experimental investigation numerical and CFD investigation are also carried out by researchers. Sharifi et al. [6] in their CFD study investigated the performance of coiled wire insert for laminar flow using structured hexahedral mesh. CFD study of loose fit twisted tape insert was done by Piriyarangroj et al. [7] and it showed better performance for 0.05 clearance ratio than any other clearance ratio. Salman et al. [8] simulated plain twisted tape (2.93 and 3.91 twist ratios) and baffled twisted tape for laminar flow and observed lower twist ratio have higher heat transfer enhancement and enhancement increases with increase in Reynolds number. They also described of better performance for Baffled twisted tape than plain twisted tape. Salman et al. [9] simulated plain and alternate axis twisted tape insert for twist ratios= 2.93, 3.91, 4.89 and alternative angles= 30^o, 60^o, 90^o for laminar flow at uniform heat flux and concluded twist ratio of 2.93 and alternative angle of 90^o offered a maximum heat transfer enhancement. Bellos et al. [10] did numerical investigation on heat transfer enhancement of evacuated and non-evacuated parabolic trough collectors using internally finned absorber, twisted tape and perforated plate inserts. Results showed best thermal efficiency enhancement of internally finned absorber (2.1% for non-evacuated and 1.6% for evacuated tube collector) later twisted tape (1.8% for non-evacuated and 1.5% for evacuated tube collector) and last perforated plate inserts (1.4% for non-evacuated and 1.2% for evacuated tube collector). Internally finned absorber also showed best performance for pressure drop.

CFD study of heat transfer enhancement using insert is a useful method to check for thermal and fluid flow characteristics of an insert prior to fabrication and experimental investigation. This assists in selection of suitable insert without fabrication and investigation of numerous inserts. So, resource and cost of research can be minimized and it gives accuracy close to experimental results. So, in this study CFD investigation of heat transfer and fluid flow characteristics of tube with helical screw tape inserts (HST) with 1.92 twist ratio (TR) for laminar flow is done. A commercial software package is used, proper model, boundary conditions are applied and solved. Later the data is analyzed and summarized to report the findings.

2. Materials and method

2.1. Physical model

The geometry of fluid domain and insert both were designed in a commercial software. The fluid domain has a length of 425 mm, diameter of 30 mm. The cylindrical portion of the insert has a dimeter of 19 mm and the helical insert swept around it has a height of 4 mm and width of 1 mm. The insert is 350 mm long and has 1.92 twist ratio. It is placed in 25 mm downwash of the inlet and ends 50 mm before the outlet. There is some clearance between the insert and the pipe wall meaning insert is loose fitted and convection heat transfer is the main mechanism of heat transfer to water from pipe wall. If insert is tight fitted then some conduction occurs through insert. To reduce complexity and computational effort insert is loose fitted. So, these two parts are then imported into ANSYS and insert domain was replaced by a Boolean cut to adiabatic wall thus minimizing computational domain. Here, inserts main purpose is to act only as a swirl generator (i.e. flow passage modifier). Later the domain is discretized using ICEM and solved. Schematic of the computational model and discretized domain is shown in Fig. 1. The physical and thermal properties of water at simulated conditions are shown in Table 1.

Table 1. Physical properties of water at simulation conditions

Properties	Value
Prandtl number, Pr	7
Thermal conductivity, k	0.6 w/m-k
Specific heat, C_p	4180 J/kg-k
Water density, ρ	998 kg/m^3
Dynamic viscosity, μ	1.003×10^-3 kg/m-s

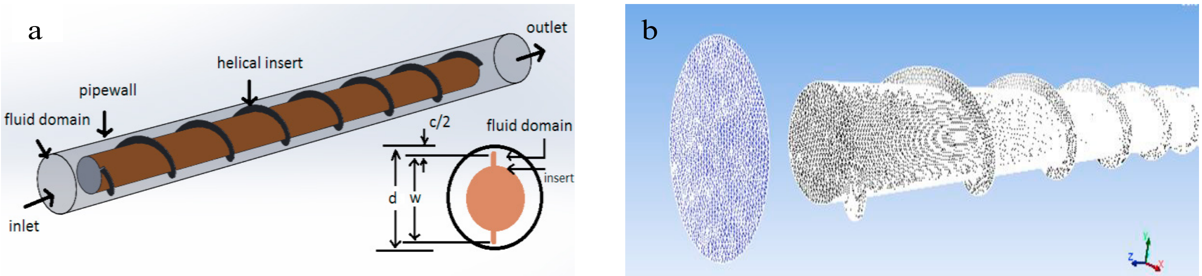


Fig. 1. (a) Schematic of the 3D fluid domain with insert; (b) mesh sample of inlet and insert

2.2. Data reduction

Amount of heat transferred to the fluid

$$Q = m' C_p (T_{out} - T_{in}) \tag{1}$$

This heat is transferred from pipe wall to the fluid using convective heat transfer, where A_s, T_s and T_b are area of the surface, temperature of the pipe wall and bulk fluid, respectively.

$$Q = A_s h (T_s - T_b) \tag{2}$$

Convective heat transfer coefficient is calculated as, where q is heat flux.

$$h = \frac{q}{T_s - T_b} \tag{3}$$

Simulated Nusselt number is calculated from

$$Nu = \frac{hd}{k} \tag{4}$$

Simulated friction factor is found using Darcy-Weisbach equation, where Δp is pressure drop.

$$f = \frac{\Delta p}{((L/d) * (\rho V^2)/2)} \tag{5}$$

For governing equations and boundary conditions momentum, continuity and energy equations are used from Guo et al. [11].

2.3. Boundary conditions

In CFD study of heat transfer enhancement using insert, heat supply is implemented into pipe wall using two methods. One is uniform heat flux and another is temperature. In this study heat supply of 100000 W/m^2 is implemented in pipe wall using uniform heat flux condition. There is some assumption like steady state condition, fully developed incompressible laminar flow, no heat loss is used in this study. Velocity inlet and pressure outlet conditions are selected for inlet and outlet respectively. The study is done for different velocities of water for $0 < Re < 2000$. Uniform velocity is used in the inlet which is fully developed further in pipe and zero-gauge pressure at outlet to measure the pressure drop. Insert walls are set to adiabatic conditions assuming negligible heat transfer. For plain tube simulation only 425 mm long, 30 mm diameter fluid domain was used for same heat flux and boundary conditions as tube with insert.

2.4. Solution method and convergence criteria

Navier-Stokes equation coupled with energy equation and laminar viscosity model is solved for fully developed water flow. Semi Implicit Pressure Linked Equations (SIMPLE) a Pressure Velocity Coupling method is used for the simulation. Second order upwinding for pressure, momentum and energy is selected to determine the variables inside each cell keeping under relaxation factors to default values. Simulation is stopped when the residue for continuity and energy reached 10^{-3} and 10^{-5}. The computer used for this purpose is a core i5 processor (4 threads) 2.3 GHz CPU with 8 GB RAM. Each solution procedure took average time of 3 hours.

2.5. Model validation

2.5.1. Grid sensitivity check

Grid sensitivity is checked for plain tube and tube with insert. Simulation were performed for maximum mesh element face size of 0.0008m, 0.001m, 0.0012m for tube with insert and 0.0008m, 0.001m, 0.0015m for tube without insert resulting in 2072881, 1412699, 836838 and 1709381, 1314916, 507229 number of total elements for tube with insert and without insert respectively. Simulations were carried on for Re of 776 for all the element sizes while boundary conditions remained the same. Sensitivity is checked using outlet temperatures for different element sizes of both the cases. Since amount of heat addition is same for all the cases therefore the outlet temperature should be same. Results revealed as shown in Fig. 2, negligible difference in outlet temperature with theoretical temperature. Considering computation time and solution accuracy, 0.0015m and 0.0008m maximum face size for mesh element were selected for plain tube and tube with insert respectively.

2.5.2. Simulation validation

Results are validated using Nusselt number and friction factor for plain tube using modified Seider and Tate correlation developed by mills [12] and Eqn. 6 respectively. The root mean square (rms) error of simulated Nusselt number with theoretical Nusselt number is 32% and a negligible error of friction factor is obtained.

$$f = \frac{64}{Re} \tag{6}$$

3. Results and discussion

3.1. Heat transfer characteristics

Due to no slip condition fluid velocity adjacent to pipe wall is zero and relatively small near that region. So thermal boundary layer is thick. As a result, fluid inside thermal boundary layer has higher temperature. So, temperature gradient between pipe wall and adjacent fluid domain is less resulting in lower heat transfer coefficient. As velocity increases in plain tube thermal boundary layer thickness is reduced since fluid moves faster and heat transfer coefficient increases. When insert is placed in tube then it swirls the flow inducing the mixing of fluid and turbulence. It increases the residence time and normal gradients of velocity. As a result, pipe wall temperature drops that reduces temperature of heat addition minimizing irreversibility. Figs. 3 (a) and 3(b) describe that with increase of velocity, heat transfer coefficient increases and it increases up to 2.6 times for tube with insert then plain tube at Re= 1940. From Figs. 3(c) and 3(d), Nusselt number is also increasing with Reynolds number and it increases up to 2.6 times like heat transfer coefficient in comparison to plain tube. Bhuiya et al. [2] achieved 2.6 times increase in Nusselt number for helical screw tape at 50000 Reynolds number. Similar achievement in current study compared to their study is obtained at relatively lower Reynolds number because of higher thickness of rod. The higher the thickness of rod the more fluid passes adjacent to heating surface. These enhances the heat transfer characteristics.

Figs. 4(a) and 4(b) show the temperature profile of pipe wall and outlet for plain tube while Figs. 4(c) and 4(d) show temperature profile of pipe wall and outlet for tube with helical screw tape inserts. From Fig. 4 it is visible that insert fitted tube's pipe wall temperature is lower than plain tubes pipe wall temperature for same heat flux condition. This happens because insert modifies the fluid passage providing swirl flow, thus greater effectiveness in dissipating heat from pipe wall. As a result, irreversibility is reduced and heat transfer characteristics is enhanced. Outlet bulk temperature is same in both the cases but better mixing in case of insert fitted tube.

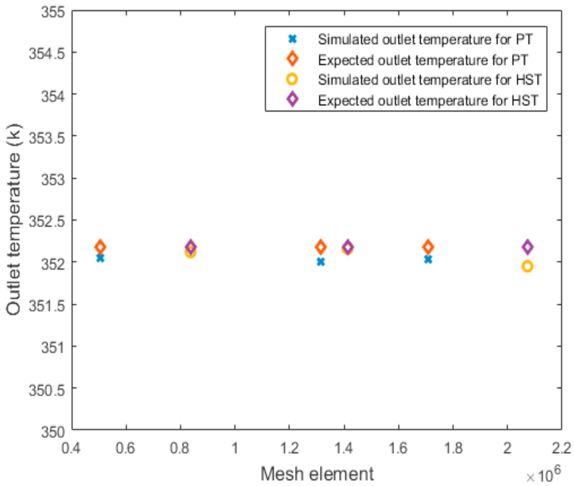


Fig. 2. Grid sensitivity test

3.2. Fluid flow characteristics

The effect of addition of insert and velocity for friction factor is plotted against Reynolds number in Fig. 5. It is visible that the simulated and theoretical model for friction factor has similarity in trend. Friction factor drops as velocity increases which is similar in nature reported by Sivashanmugam and Suresh [13]. The average increase in friction factor is 6.05 times in Reynolds number range from 200 to 2000. Maximum friction factor rise was 8 times than plain tube at Re=1940. Increase of friction factor in helical screw tape insert was due to the increase of surface area, turbulence and mixing of fluid.

3.3. Performance evaluation

Thermal performance factor is the ratio of Nusselt number ratio and friction factor ratio at a constant pumping power. It describes the performance enhancement by using insert. So, performance enhancement of tube with insert is evaluated at identical pumping power with plain tube used by [4, 14].

For constant pumping power evaluation criteria, where, ‘i’ denotes insert and smooth tube by ‘s’ and Q is discharge.

$$(Q\Delta P)_i = (Q\Delta P)_s \tag{7}$$

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

$$(fRe^3)_i = (fRe^3)_s \tag{8}$$

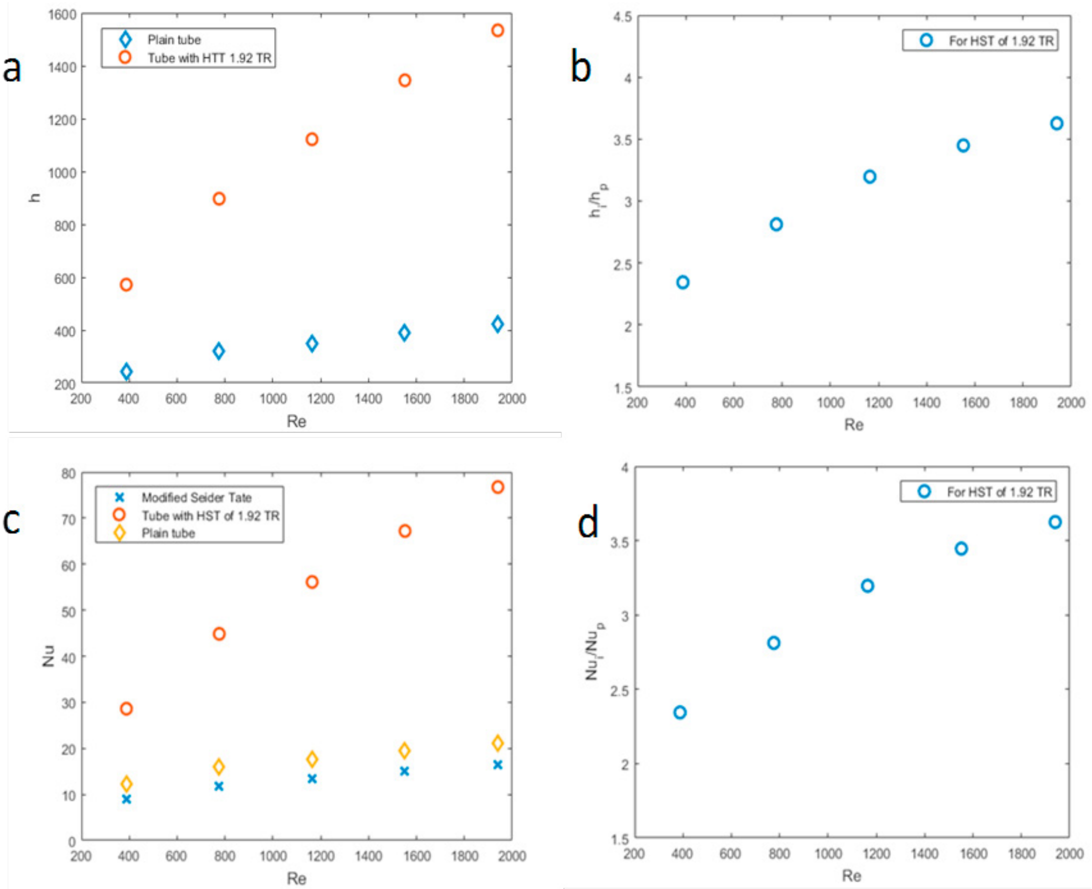
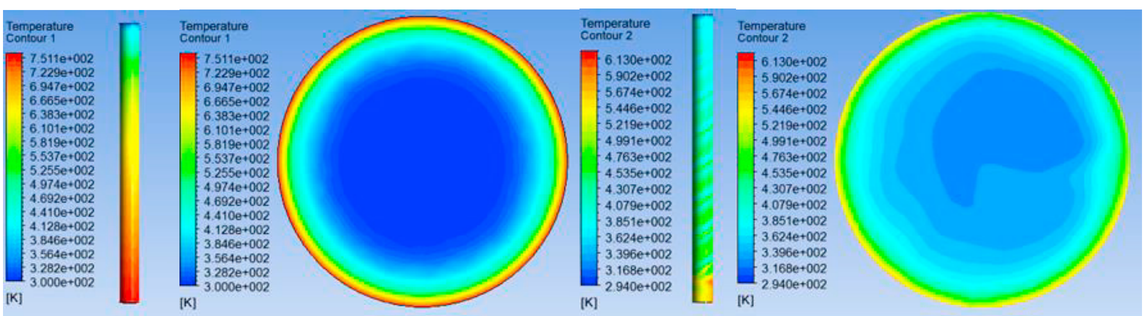


Fig. 3. Relationship between (a) Heat transfer coefficient vs. Reynolds number (b) heat transfer coefficient ratio (c) Nusselt number vs. Reynolds number (d) Nusselt number ratio



a b c d

Fig. 4. Temperature profile at 800 Re for (a) plain tube pipe wall (b) plain tube outlet (c) pipe wall of tube with HST (d) outlet of tube with HST

$$Re_s = Re_i \left(\frac{f_i}{f_s}\right)^{1/3} \tag{9}$$

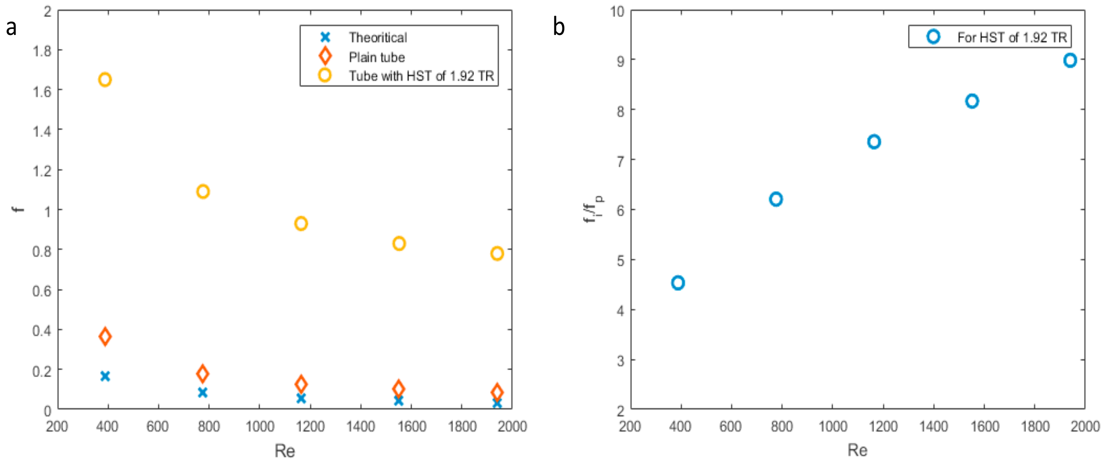


Fig. 5. (a) Friction factor vs. Reynolds number (b) Friction factor ratio

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

$$\eta = \frac{\frac{Nu_i}{Nu_s}}{\left(\frac{f_i}{f_s}\right)^{\frac{1}{3}}} \tag{10}$$

Following this procedure, the resultant thermal performance factor equation is

$$\eta = Re^{0.2988} Pr^{-2.457} \left(\frac{y}{w}\right)^{5.9043} \tag{11}$$

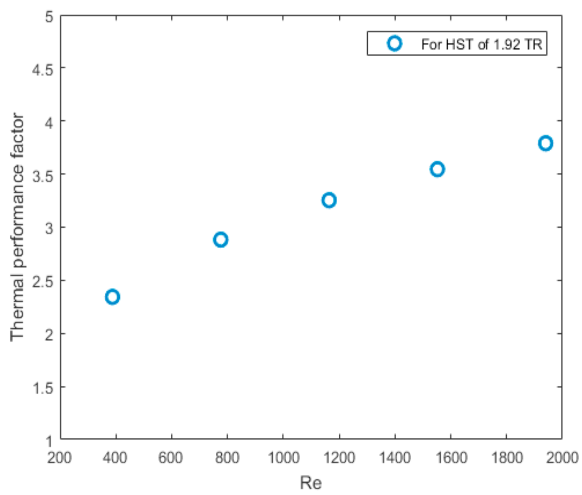


Fig. 6. Thermal performance factor vs. Reynolds number

Where, y/w stands for twist ratio. From Eqn. 11 and consecutive Re, Fig. 6 is obtained. From Fig. 6 it is observed that thermal performance factor increases with the increase in Reynolds number and it is found 3.79 for Reynolds number 1940. For laminar flow increase of thermal performance factor with the increase in Reynolds number for circular tube fitted with center-cleared twisted tape is reported by Guo et al. [11] whereas, decrease is reported for helical tape inert in turbulent flow by Bhuiya et al. [2]. Thermal performance factor for all simulated Reynolds number is greater than unity meaning that the enhancement of heat transfer is greater than frictional energy loss. So, it is clear that helical screw tape insert can cause energy savings.

4. Conclusion

A CFD study of helical screw tape insert of 1.92 twist ratio in tube was done for laminar flow as heat transfer enhancer using a commercial software package. This insert augmented the rate of heat transfer with significant increase in pressure drop. The findings of the study are:

- Nusselt number and friction factor are increased with increase in Reynolds number and it is 1.34-2.6 times and 3.5-8 times higher than the plain tube, respectively.
- Thermal performance factor is greater than unity for all the cases. It ranged between 2.34-3.79 and achieved the value of 3.79 at Re=1940.
- Heat transfer enhancement is superior to the frictional pressure drops. So, helical screw tape inserts cause energy savings.
- Pipe wall temperature is reduced in case of insert fitted tube meaning lower irreversibility and the higher heat transfer characteristics.
- It has helical contour of relatively higher and lower temperature in pipe wall like the twist of the insert.

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