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3D CFD Simulation of Heat Transfer and Friction Characteristics of Laminar Flow of Water through a Circular Duct with Centre-Cleared Twin Twisted Tape

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Abstract

Heat transfer behavior of laminar flow inside circular duct with centre-cleared twin twisted tape (CTTT) swirl generator was investigated numerically. The governing equations were solved with a finite-volume-based numerical method. A three-dimensional non uniform grid was generated, in order to critically examine the flow and heat transfer. The centre-cleared twin twisted tapes were tested in the current work; investigations were performed in the Reynolds number range of 100- 1,000 with four different CTTTs at twist ratio (TR) of H/W = 1.0, 2.0, 3.0 and 4.0. Water is used as working fluid (Pr = 7). The effects of Reynolds number, Nusselt number, friction factor, convective heat transfer coefficient were examined and discussed. The use of CTTT was found to increase the heat transfer considerably when compared with plain tube. The Nusselt number increased with the increase of Reynolds number. The overall enhancement ratio has been calculated in order to discuss the overall effect of CTTT and the working parameters. The present findings would be useful inputs for the design of solar thermal heaters and heat exchangers.

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1. Introduction

Laminar flow within tubes is encountered in many industrial applications such as solar thermal heaters and heat exchangers. In case of laminar flow, there is major thermal resistance in the bulk flow in addition to the dominant thermal resistance in the thin boundary layer adjacent to the flow. Therefore, twisted-tape inserts are used to mix the gross flow effectively to reduce the thermal resistance in the core flow through the channel, thereby reducing the thickness of boundary layer efficiently.

For several years, the heat transfer enhancement in heat exchanger tubes by using twisted tapes has been extensively studied. Twisted tapes are commonly installed in a tube heat exchanger to promote the fluid mixing between central and near-wall regions. Swirl flow devices form an important group of passive augmentation techniques, and twisted-tape is one of the most important members of this group. Low heat transfer co-efficient in tube-side laminar flow of viscous liquid is encountered in many tubular heat exchangers used in chemical process industry. The overall heat transfer coefficient, in all such cases, is controlled by the low heat transfer coefficient of laminar flow in the tube-side. Thus enhancement of inside heat transfer coefficient results in improving the overall performance of a heat exchanger. Many of the passive and active techniques available for augmentation of laminar flow heat transfer have been discussed in detail by Bergles and Webb (1985) and Bergles et al. (1983 & 1991).

Twisted tapes can be most beneficially used to augment laminar flow heat transfer since the improvement in heat transfer has been found to be much more than the corresponding increase in pumping power. Insertion of centre-cleared twisted tapes in tube is one such augmentation technique. Twisted tapes reduce the dominant thermal resistance of the viscous stream and reduce the required heat transfer surface area. However, the thermal improvements are accompanied by increased pressure drop. They are inserted in the circular duct leaving some gaps along the periphery and the corners of the duct cross-section. Full-length twisted-tape inside a circular duct increases the duct side surface heat transfer co-efficient of a heat exchanger.

The dimensionless geometric parameters that influence the heat transfer and friction characteristics are the rib height and rib pitch. Integral rib-roughness has been used for the enhancement of tube-side heat transfer coefficient in low-flow automotive radiators and in this case, the flow Reynolds number is small and may be < 2000. Farrel et al. (1991) tested one fully-ribbed and two broken ribbed flat radiator tube. They obtained friction factors for $200<$ Re $<11,000$. However, the heat transfer coefficients were obtained only for turbulent flow with Re > 2000. The broken-ribbed tube with the highest e/Dh yielded the highest heat transfer coefficient as well as the highest friction factor. Olsson and Sunden (1996) tested two ribbed radiator tubes with airflow. The air heat transfer data were taken with constant wall temperature and the data provided the axially averaged heat transfer coefficient over the tube length. The enhanced tubes showed higher friction factors than the smooth tube in both laminar and turbulent regions. However, as the Reynolds number decreased in the strictly laminar region, the friction factors tended to converge and approach the smooth tube value. Also, the laminar-turbulent transition Reynolds number decreased as the friction factor increased. Similar to the friction behavior, the Colburn j factors also tended to converge at low Reynolds numbers, and approached the smooth tube value. However, in contrast to the friction factors, the j factors did not show a clear laminar-turbulent transition. Olsson and Sunden (1998) investigated the effect of rib configurations for the multiple V-ribbed channel. Saha and Dutta (2001) observed that, for regularly spaced twisted-tape elements, thermohydraulic performance of twisted tapes with multiple twists in the tape module was not much different from that with single twist in the tape module. Twisted tapes with gradually decreasing pitch performed worse than their uniform-pitch counterparts. Patil (2000) worked with varying width twisted-tape inserts for which both friction factor and Nusselt number were lower than those with full-width twisted tapes. Saha et al. (1989, 1990) introduced regularly spaced twisted-tape elements which are better than full-length twisted tapes under certain circumstances. Li et al. (2005) designed an optimal multi-layer spacer with optimal non-woven nets in the outer layers and twisted tapes in the middle layer. Helical screw-tape inserts by Sivashanmugam et al. (2006) behaved the same way as the twisted tapes. Twin and triple twisted tapes are also effective enhancement devices, as studied by Chang et al. (2005). A detailed insight into the studies on twisted tapes has been provided by Dewan et al. (2004). Hong et al. (2007) employed evenly spaced twisted tapes in a convergent–divergent tube, while Yuxiang et al. (2012) investigated numerically converging-diverging tubes equipped with twin twisted tapes and obtained some promising results.

The literature review reveals a lack of focus on the centre-cleared twin twisted-tape (CTTT) inserts. The fluid flow due to centre-cleared twisted-tape generated swirl flow is likely to provide better mixing in the flow resulting in increased heat and momentum diffusion. This may increase heat transfer even if it may also give increased pressure drop. Accordingly, in this study, the laminar flow numerical heat transfer and pressure drop of full-length CTTT inserts in circular ducts are investigated.

2. Methodology

2.1 The Model

This work investigates transport phenomenon in tube with CTTT (with water as working fluid), for Re from 100 to 1,000. With twist, a swirling flow will be generated and with changing the twist ratio, the number and intensity of these swirling flows will changed along the flow direction. The width of the twisted-tapes of the present study is 15 cm. The twist ratios (TR), H/W= 1.0, 2.0, 3.0 and 4.0 (H is twist length, and W is the tape width, as shown in Fig. 1) were tested. The simplified form of continuity, momentum and energy equations for a three dimensional, steady state, incompressible flow and laminar forced convection heat transfer without viscous dissipation are given below.

Continuity equation:

$$\frac{\partial u_i}{\partial x_i} = 0 \tag{1}$$

Momentum equation:

$$\rho\left(u_i \frac{\partial u_i}{\partial x_i}\right) = \frac{\partial p}{\partial x_j} + \mu \left[\frac{\partial^2 u_i}{\partial x_i^2}\right] \tag{2}$$

Energy Equation:

$$\left[u_i \frac{\partial T_i}{\partial x_i}\right] = \alpha \left[\frac{\partial^2 T}{\partial x_i^2}\right] \tag{3}$$

The Reynolds number of water flow in the duct is calculated from,

$$Re = \frac{vD_h}{\nu} = \frac{\rho v D_h}{\mu} \tag{4}$$

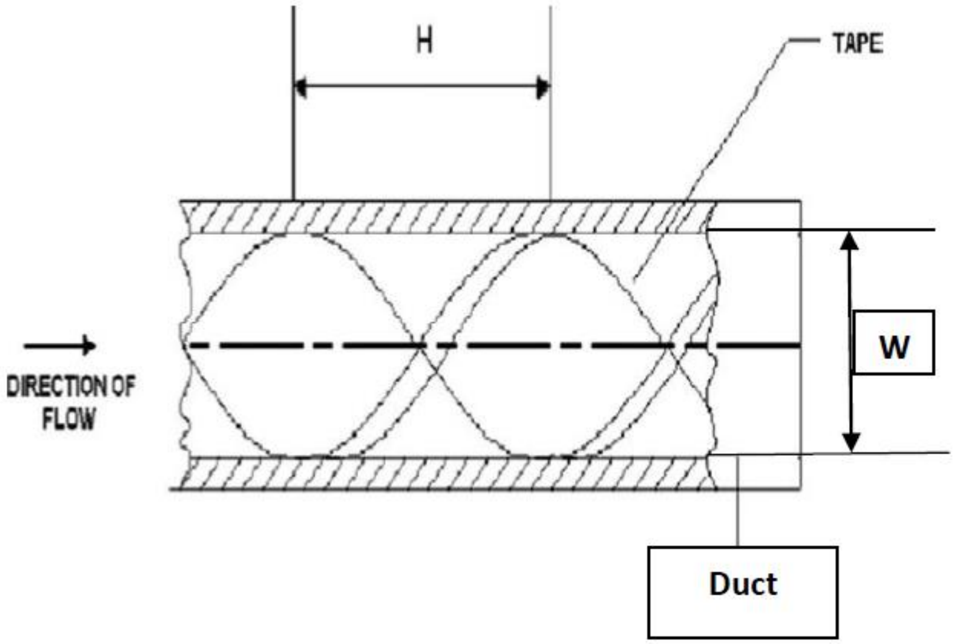


Fig. 1. Layout of the circular duct containing a full-length twisted-tape

Steady state values of the plate and water temperatures in the duct at various locations were used to determine the values of useful parameters such as heat supplied to the water (Q_u) and heat transfer coefficient (h), as given by:

$$Q_u = \dot{m}C_p(T_{ao} - T_{ai}) \tag{5}$$

$$h = \frac{Q_u}{A_p\,(T_p - T_f)} \tag{6}$$

The convective heat transfer coefficient was then used to obtain Nusselt number (Nu) as:

$$Nu = \frac{hD_h}{K} \tag{7}$$

The friction factor (f) was determined from the measured values of pressure drop (ΔP) across the test section length:

$$f = \frac{2\,\Delta PD_h}{4\rho LV^2}$$

$$\text{or, } f = \frac{2\Delta PD_h}{4LG^2} \tag{8}$$

where, $G = \frac{m}{WH}$ is the mass velocity of water.

The effect of heat transfer enhancement under given pumping power was evaluated in terms of the thermal performance factor (η), also known as enhancement efficiency, as defined by:

$$\eta = \frac{Nu}{Nu_0} / \left(\frac{f}{f_0}\right)^{0.33} \quad (9)$$

where Nu and Nu_0 are Nusselt numbers and for the enhanced tube and the smooth tube respectively, while f and f_0 denote the corresponding values of friction factor.

2.2. *Numerical Solution*

The three-dimensional (3D) governing equations shown above were discretized on a non-uniform structured grid by using finite volume method. Structured mesh was generated with fine mesh to obtain accurate results with less number of nodes. Different grid sizes were tested as part of grid independence study. After a rigorous grid independence check, a mesh consisting of 415,951 cells were used for the present computation. The pressures were calculated using semi implicit pressure linked equations (SIMPLE) scheme. The discretized equations were then linearized using an implicit technique and solved using third order accurate QUICK scheme. The convergence criteria for energy equation and momentum equations were 10^{-8} and 10^{-5} respectively.

3. Results and Discussion

The results of heat transfer, friction factor and thermal performance factor characteristics in the tube fitted with centre-cleared twin twisted (CTTT) tape and plain twisted tape (TT), under the laminar flow regime are described in the present section. The effect of twist ratio (TR) on the thermohydraulic performance is also presented. The results obtained from computations on heat transfer and friction factor characteristics in the plain circular smooth duct are verified in terms of Nusselt number and friction factor. The data obtained from the computational analysis for the plain circular smooth duct are in reasonable agreement with the predicted results (as shown in Figs. 2 and 3), from the proposed correlations for thermally developing, hydrodynamically developed laminar flow at Constant wall temperature:

$$Nu_0 = \mathbf{1.953} \times [\boldsymbol{Re.Pr} \times \left(\frac{\boldsymbol{D}}{\boldsymbol{L}}\right)]^{\mathbf{0.33}} \quad (10)$$

$$f = (64/Re) \quad (11)$$

Fig. 4 shows the variation of Nusselt number (Nu) with Reynolds number of the tube equipped with CTTT and TT with four different twist ratios (TR=1.0, 2.0, 3.0 and 4.0). Nusselt number considerably increases with increasing Reynolds number, reflecting the increase of convective heat transfer. TR = 1.0 has the highest Nusselt number and the base case (TT) has the lowest heat transfer. As the TR increases the Nusselt number decreases considerably.

The effectiveness of heat transfer enhancement in the tube equipped with CTTT compared to those of the plain twisted tape, in terms of Nusselt number ratio (Nu/Nu_0) is presented in Fig. 5. It is observed that the Nusselt number ratio of all the investigated cases are consistently higher than unity. This implies the beneficial gain for heat transfer enhancement of using the CTTT inserts over the plain twisted tape. It is also shown that the Nusselt number ratio tends to decrease with increasing Reynolds number. This is because of the influence of the CTTT insert on heat transfer enhancement, which is less significant for increasing Reynolds numbers.

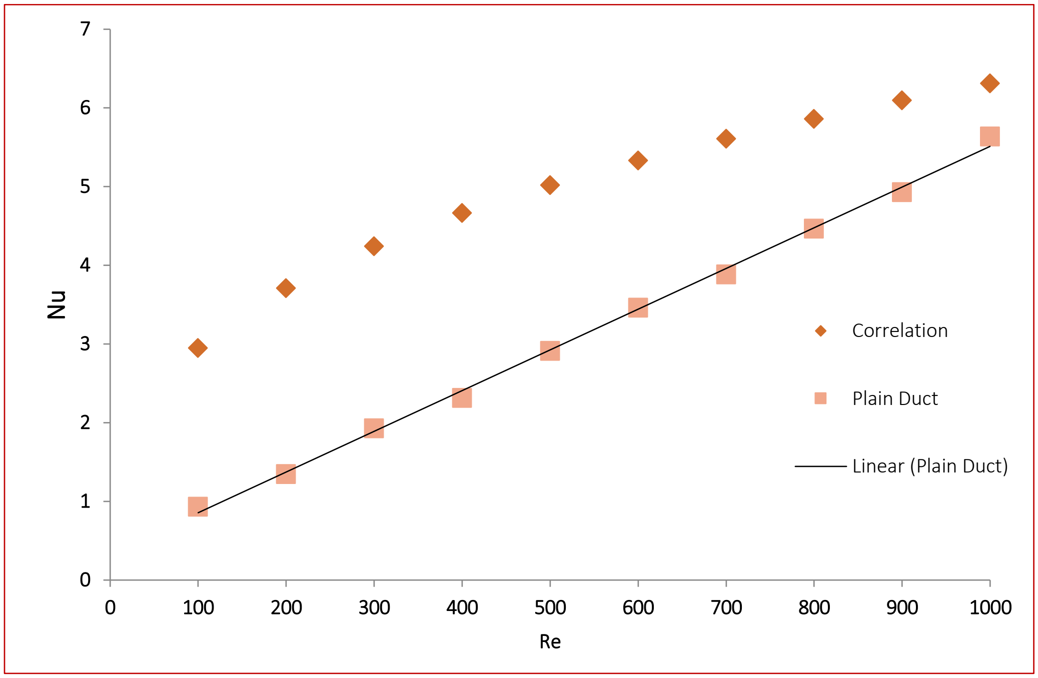


Fig. 2 Verification of the Nusselt number for the plain circular smooth duct.

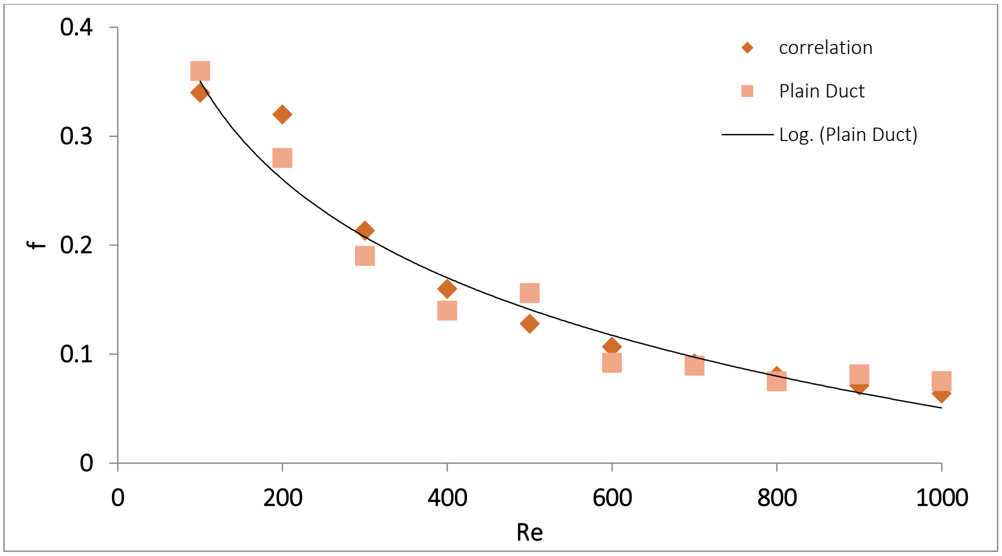


Fig. 3 Verification of the friction factor for the plain circular smooth duct.

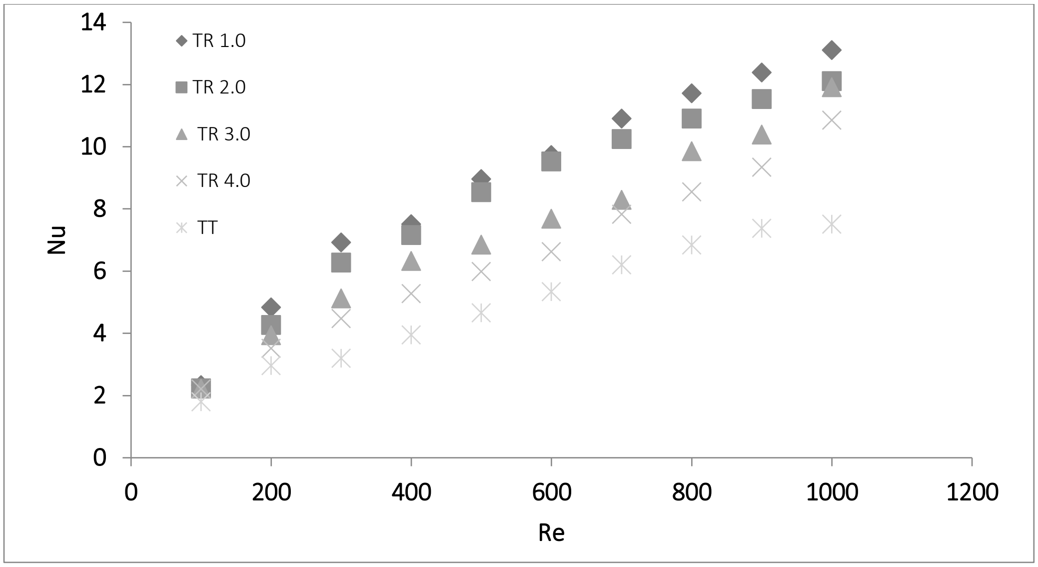


Fig. 4. Variation of Nusselt number with Reynolds number for different twist ratios.

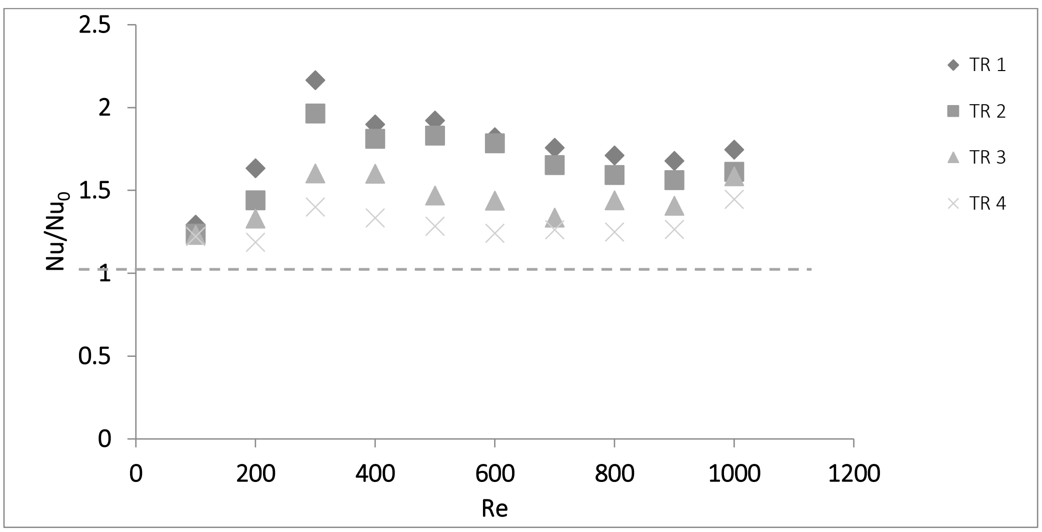


Fig 5. Variation of (Nu/Nu_0) with Reynolds number for different twist ratios.

The influence of the CTTT inserts on friction factor characteristics is represented in Fig. 6, which shows the relationship between the friction factor and Reynolds number at different twist ratios for tubes fitted with CTTT and plain TT. For all the cases, the friction factor conventionally decreases with increase in Reynolds number. It is also seen that the friction factor has similar trends for both CTTT and TT inserts. The Reynolds number increases with decreasing twist ratio; this is attributed to the use of CTTT inserts with a smaller twist ratio, which leads to a higher viscous loss near the tube wall regions caused by a stronger swirl flow or turbulence flow and long residence time in the tube.

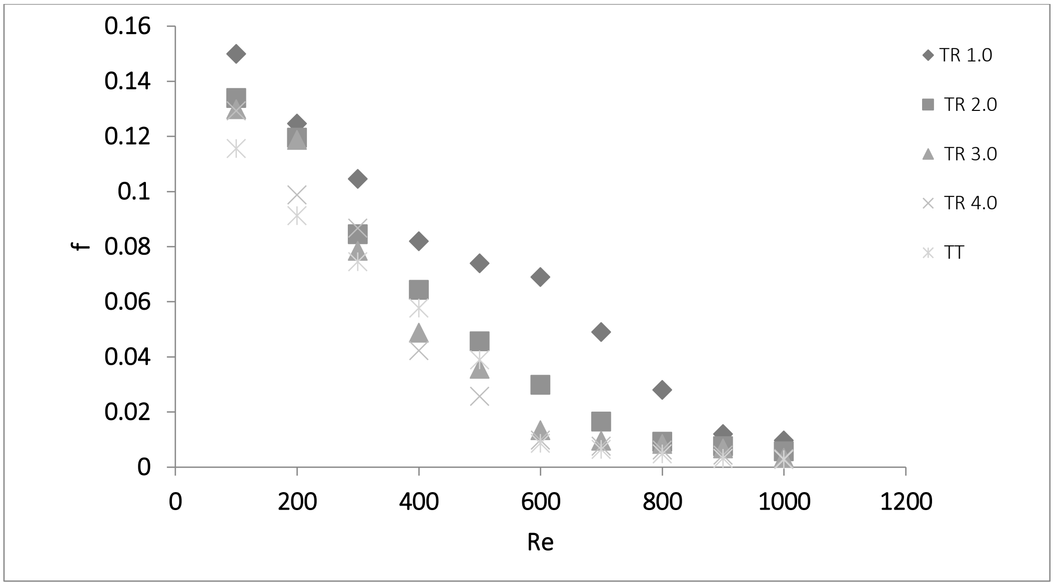


Fig. 6. Variation of friction factor characteristics with Reynolds number for different twist ratios.

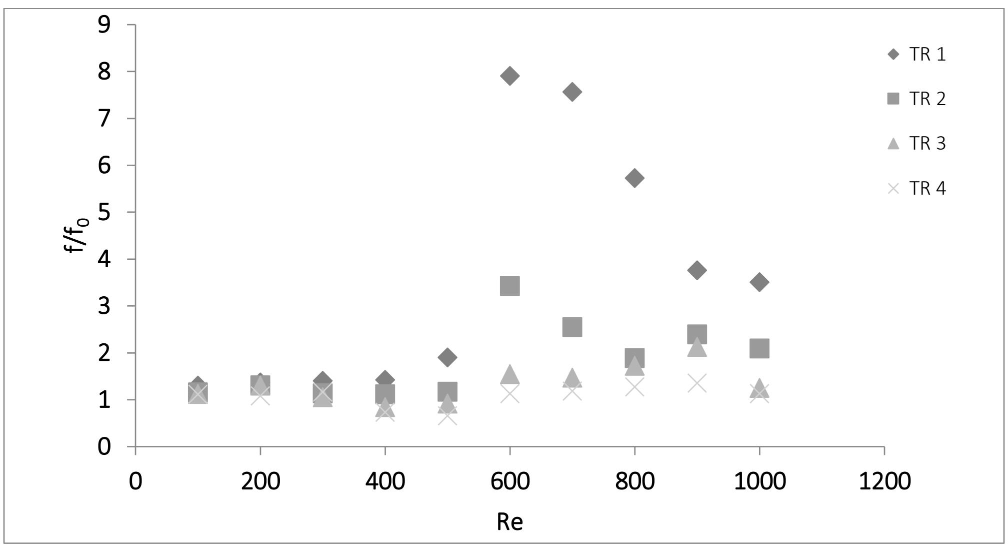


Fig. 7. Variation of (f/f_0) with Reynolds number for different twist ratios.

From Fig. 7 it is visible that both *f* and f/f_0 tend to decrease with the increase of *Re*, for all the cases. Similar to the *Nu* trends, the friction factor from the inserted twin twisted tape is considerably higher than that from the plain twisted tape alone at a given Re. The X- Vorticity contours at Re = 800 for different twist ratio is presented in Fig. 8. TR = 3.0 is found to produce a high X- Vorticity, and also two zones of high vorticity correspond to good mixing of fluids between them.

The Q-criterion is given by:

$$Q = \frac{1}{2}\left(\Omega_{ij}^{\ 2} - S_{ij}^{\ 2}\right) \tag{12}$$

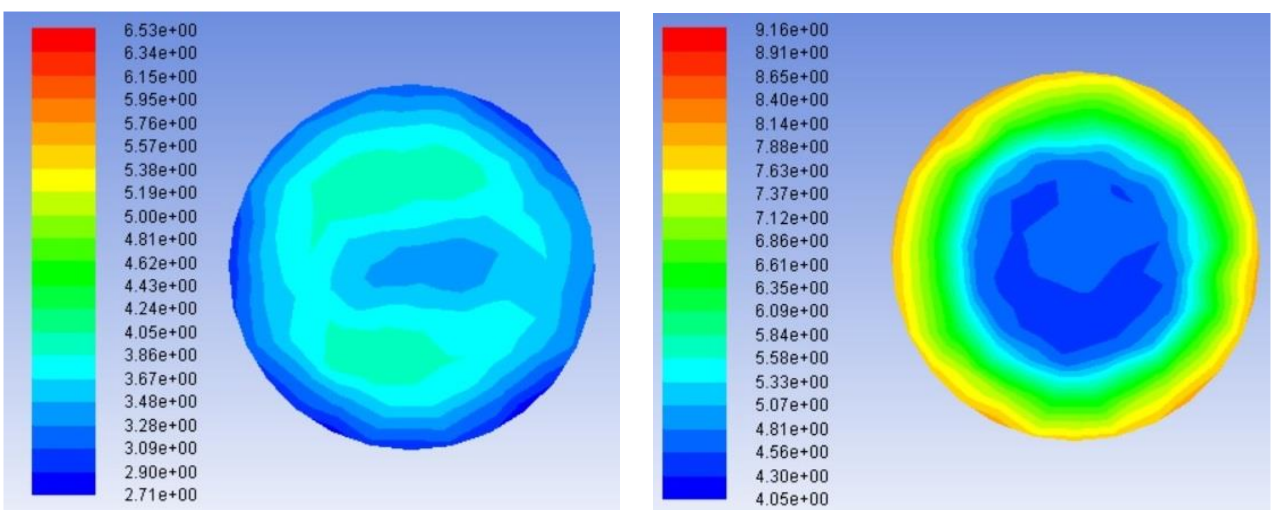
where,

$$\Omega_{ij} = \frac{1}{2}\left(\frac{\partial u_i}{\partial x_j} - \frac{\partial u_j}{\partial x_i}\right) \tag{13}$$

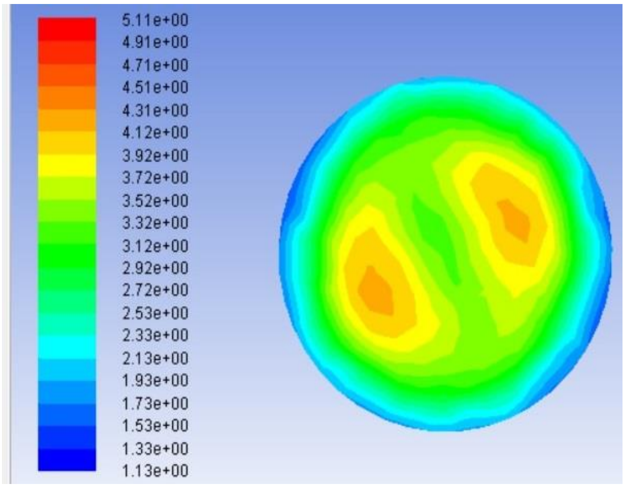
$$S_{ij} = \frac{1}{2}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) \tag{14}$$

The Reynolds numbers used in this computational study are from 100 to 1000. The iso-surfaces of Q – criterion for different twist ratios at Re = 1000 are presented in Fig. 9. There is a secondary flow vortex formation inside the tube, which might help in increasing the heat transfer rate. There is a clear view where the vortices are developing inside the tube. More eddies start appearing with increase in Reynolds number. At TR= 1.0, the vortices are more compared to higher twist ratios.

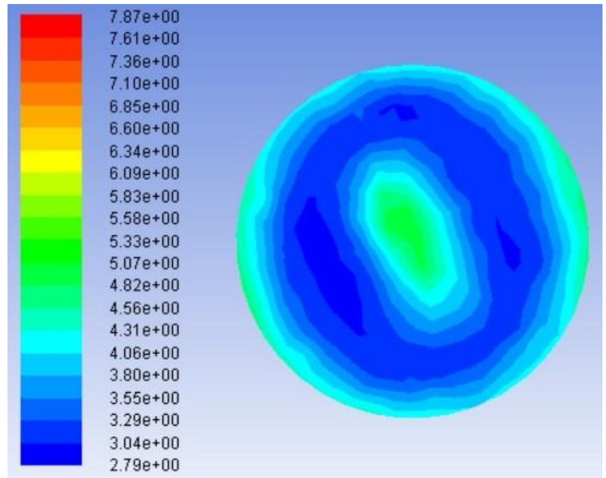
As enhancement in heat transfer is gradually accompanied by pressure drop, and thermal performance factor (η) is the appropriate parameter used for evaluating the practical use of the twisted tape. The factor is obtained by considering the effect of heat transfer enhancement and the increase of pressure drop, simultaneously. The variation of thermal performance factor with Reynolds number of the tubes equipped with CTTT is shown in Fig. 10. The CTTT is found efficient from the energy point of view because η is greater than unity in most of the varied Reynolds number ranges, which indicates that the effect of heat transfer enhancement due to the turbulator is more dominant than the effect of rising friction and vice versa. As expected from Fig. 11, the tube with inserts at lower Reynolds numbers up to about 500 provides higher η, and therefore TR 2 and TR 3 can be recommended.



(a) **(b)**

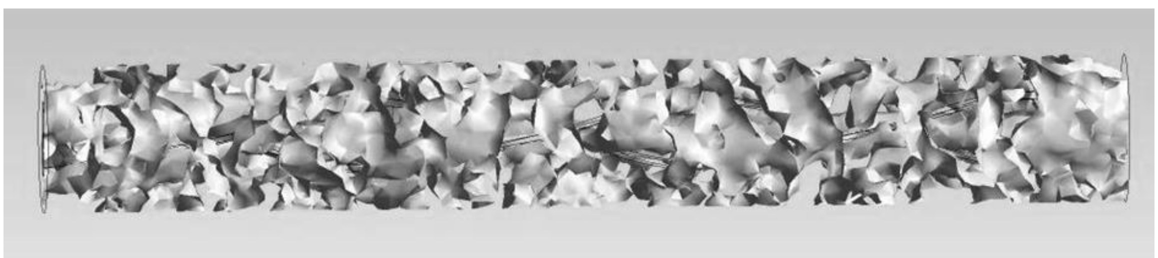


(c)

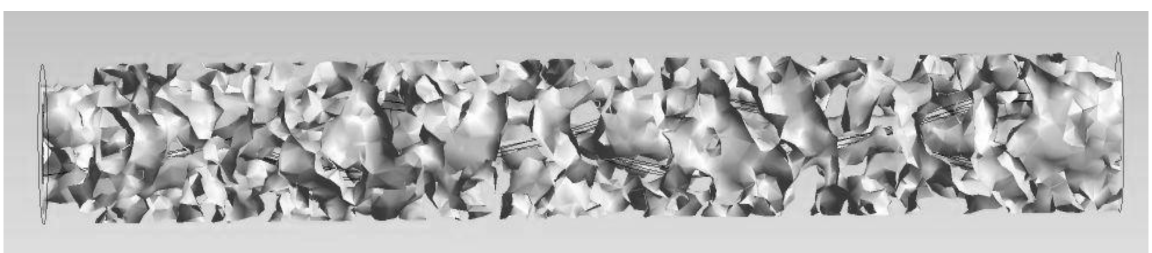


(d)

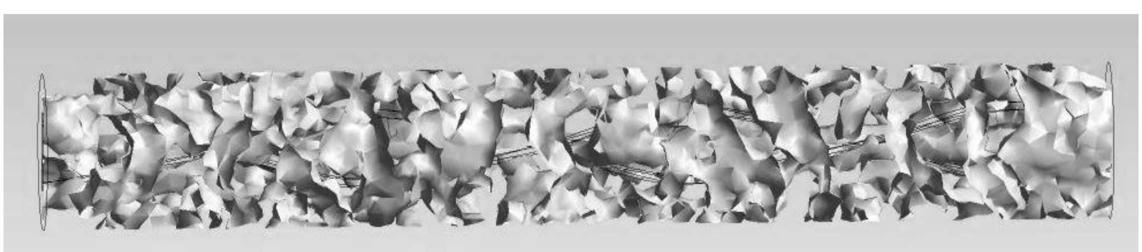
Fig. 8: X- Vorticity contours at outlet of the duct at Re = 800 for different twist ratios; (a) TR 1.0, (b) TR 2.0, (c) TR 3.0, and (d) TR 4.0.



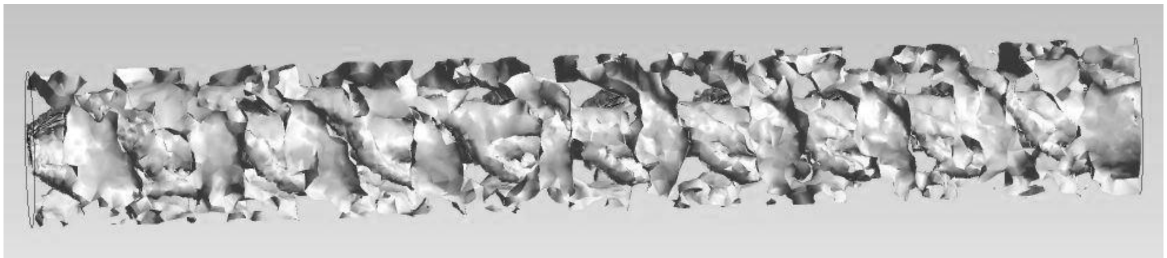
(a)



(b)



(c)



(d)

Fig. 9 Iso surfaces of Q - criterion (Q = 1200) depending on surface velocity at Re = 1000 for different twist ratios; (a) TR 1.0, (b) TR 2.0, (c) TR = 3.0, and (d) TR = 4.0

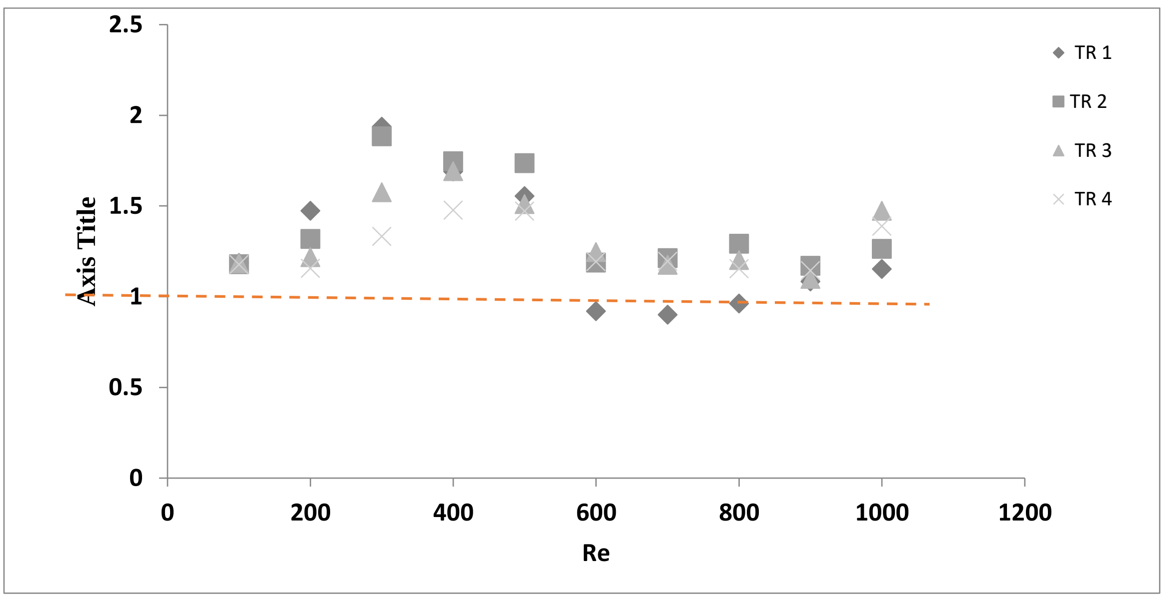


Fig. 10. Variation of η with Reynolds number for different twist ratios.

4. Conclusion

A numerical study has been conducted to examine the heat transfer and flow friction characteristics in a circular duct fitted with centre cleared twin twisted tape at different twist ratios for laminar flow (Re from 100 to 1000). The 3D governing equations were solved by using a finite volume method. The current predictions provide promising inputs for the design of solar thermal heaters and heat exchangers. The key findings could be summarized as follows:

- The heat transfer, friction factor and thermal enhancement efficiency increased with decrease of twist ratio.
- Nusselt number increased with increase in Reynolds number while it was reverse for the case of friction factor.
- The Nu/Nu_0 augmentation tended to slightly decrease with the rise in Re.
- The thermal enhancement efficiency for all the cases was more than unity, indicating that the effect of heat transfer enhancement due to the turbulator was more dominant than the effect of rising friction and vice versa.

The present findings would be useful inputs for the design of solar thermal heaters and heat exchangers. Performing similar analysis with other working fluids and comparing with the present results would be an interesting extension to the current study.

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