Research Article

NUMERICAL ANALYSIS ON HEAT TRANSFER AND FLOW CHARACTERISTICTHROUGH ELLIPTICAL TWISTED TUBE

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ABSTRACT

In the present study, a circular smooth tube (CST), an elliptical smooth tube (EST) and various elliptical twisted tube (ETT) configurations are investigated according to heat transfer and flow characteristics by using a CFD software. The ETT configurations are considered as the aspect ratio (AR) of 1.5 and 2.0 and the twist pitch length (PL) of 50, 100 and 200 mm. The hydraulic diameter of the tubes is kept constant to ensure that the results are independentof the hydraulic diameter size. A constant heat flux of 50 kW/m^2 is applied to the test tube and the flow conditions are under turbulent flow conditions corresponding to Reynolds number 4060-26,998. Water is elected as a working fluid. The thermal and physical properties of the fluid are considered dependent on temperature. It is resulted that the ETT contributes to enhancing heat transfer, despite increasing the friction factor. As the AR increases and the PL decreases, the heat transfer is positively affected, while the friction factor is negatively affected. As a result, when the heat transfer and the hydraulic performance of all cases are simultaneously determined with the performance evaluation criteria (PEC) value, the highest PEC value is obtained as 1.39 for the case of ETT_AR=2.0_PL=50 at Reynolds number of 4524.

Keywords: *Heat transfer, flow characteristic, elliptical twisted tube, numerical analysis.*

1.INTRODUCTION

Due to rapidly increasing the population and running out of fossil fuels, the energy need has been increasing day by day. In this respect, it will be always valuable to use the energy efficiently. Since most of the engineering applications in the industry involve heat exchangers, studies aimed at increasing the efficiency of

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heat exchangers are important in terms of the energy efficiency. In addition, with increasing the efficiency of heat exchangers, smaller scale heat exchangers can be designed. Heat transfer enhancement techniques are divided into two techniques: active technique and passive technique. Active techniques require extra power for the system, while passive techniques do not require any power input to the system [1]. The passive techniques consist of using turbulator devices in the tube such as twisted tape [2,3] and coiled wire [4,5], extending surfaces [6] of the tube as corrugated tube [7], coiled tube [8], placing pin fins [9] and twisted tubes [10,11] and using nanofluid [12,13].

Dong et al. [14] experimentally investigated the thermal and hydraulic performance of the spiral twisted tube. Their results showed that the spirally twisted tube heat exchangers provide obvious heat transfer enhancement for both the laminar and the turbulent flow. They also present heat transfer correlations for the spiral tube heat exchanger with a deviation of ±10%. Yu et al. [10] investigated the turbulent heat transfer performance of twisted oval tubes with different cross-sectioned wire coil inserts. Their results show that using twisted tube and wire coil together could further enhance heat transfer rate and pump consumption, while the performance evaluation criteria decreased. Tan et al. [11] conductedan experimental study on twisted oval tube heat exchanger. They recommended that the twisted oval tube heat exchangers are to work at low tube side flow rate and high shell side flow rate. Tang et al. [15] experimentally and numerically investigated the circular tube, elliptical twisted tube and tri-lobed tube using water fluid with respect to thermal and hydraulic performance. They resulted in the twisted tri-lobed tube that is more suitable for substituting for a straight tube in heat exchanger according to performance evaluation criteria.

In the light of the literature review cited above, the twisted tube is a good choice to increase the heat transfer performance of the heat exchangers. However, an assessment on the aspect ratio (AR) and twist pitch length (PL) have not been presented yet according to heat transfer performance, flow characteristic and performance evaluation criteria.

With this motivation, a numerical study is carried out to investigate the effects of the AR and the PL on the heat transfer, the flow characteristic and performance evaluation criteria.

2.MATERIAL AND METHODOLOGY

2.1.PHYSICAL MODEL

In this study, a circular smooth tube (CST), an elliptical smooth tube (EST) and various configurated elliptical twisted tubes (ETT) are considered to investigate the heat transfer and flow characteristic. The ETTs are configurated with a different aspect ratios (AR) of 1.5 and 2.0 and twist pitch lengths (PL) of 50, 100 and 200 mm. The considered tubes are illustrated in Figure 1. The dimensions of the ellipse (a and b) are adjusted to keep the hydraulic diameter of 17 mm constant so that the hydraulic diameter does not affect the non-dimensional number such as Nusselt number (Nu), friction factor (f) and performance evaluation criteria (PEC). The solution domain consisting of main three sections: an entrance section, a test section and an exit section is depicted in Figure 2. In order to provide the hydraulic developedflow, the entrance tube before the test section which is a length of 1000 mm is placed with length of 250 mm [16]. The exit section with a length of 150 mm is placed to prevent the reserve flow effects.

Figure 1. Considered tube configurations in the study

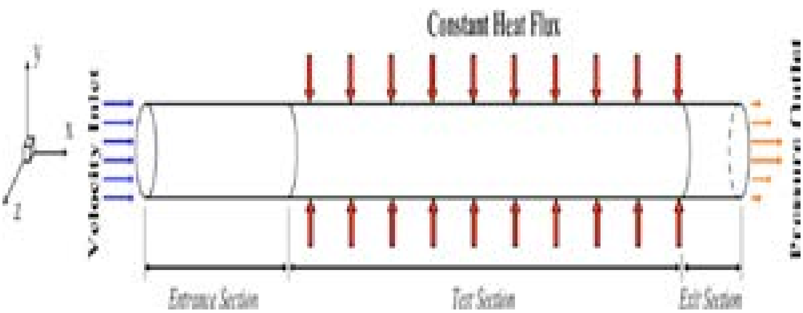

Figure 2. Solution domain with boundary typest

2.2.NUMERICAL METHOD

The thermal and hydraulic performance of the considered tubes are numerically investigated by using a CFD program. k-ω, the Standard turbulence model is used to simulate turbulent flow through the solution domain. Polyhedral mesh structure with boundary layer mesh is generated for the solution domain as shown in Figure 3.

Figure 3. Used mesh structure for the numerical analysis

2.3.GOVERNING EQUATIONS

The CFD program solves continuity (1), momentum (2) and energy (3) conservation equations [17].

$$\nabla(\rho\vec{V}) = 0 \quad (1)$$

$$\nabla(\rho\vec{V}\vec{V}) = -\Delta P + \nabla(\mu\nabla\vec{V}) \quad (2)$$

$$\nabla(\rho c_p\vec{V}T) = \nabla(k\nabla T) \quad (3)$$

Where ρ is fluid density, V is velocity, P is pressure, μ is dynamic viscosity,capacity, k is thermal conductivity, and T is temperature.

Semi Implicit Method for Pressure Linked Equations (SIMPLE) algorithm scheme is conducted to achieve the relationship between pressure and velocity coupling to enforce mass conservation and to obtain pressure field [17]. Quadratic Upstream Interpolation for Convective Kinematics (QUICK) scheme is used for discretion of convection terms and diffusion terms. The residual criteria of continuity, velocities, energy, k and ω are taken as 1 × 10−5 to ensure convergence of the solution.

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k V_i) = \frac{\partial}{\partial x_j}\left(\Gamma_k \frac{\partial k}{\partial x_j}\right) + G_k - Y_k + S_k \quad (4)$$

$$\frac{\partial}{\partial t}(\rho \omega) + \frac{\partial}{\partial x_i}(\rho \omega V_i) = \frac{\partial}{\partial x_j}\left(\Gamma_\omega \frac{\partial \omega}{\partial x_j}\right) + G_\omega - Y_\omega + S_\omega \quad (5)$$

In these equations, G_k represents the generation of turbulence kinetic energy due to the mean velocity gradients. G_ω is the generation of ω. Γ_k and Γ represent the effective diffusivity of k and ω, respectively. Y_k and Y_ω represent the dissipation of k and ω due to turbulence. All of the above terms are calculated as described below. S_k and S_ω are user-defined source terms. The effective diffusivities for the k- ω model are given by.

$$\Gamma_k = \mu + \frac{\mu_t}{\sigma_k} \tag{6}$$

$$\Gamma_w = \mu + \frac{\mu_t}{\sigma_w} \tag{7}$$

Where σ_k and σ_ω are the turbulent Prandtl numbers for k and ω, respectively. The turbulent viscosity,
μ_t, is computed by combining k and ω as follows. a^* is assumed as 1.0 in the high Reynolds number formof the k-ω model equations. Detailed information is available in the Fluent user guide [17].

$$\mu_t = \alpha^* \frac{\rho k}{\omega} \tag{8}$$

2.4. DATA REDUCTION

Used data are exported from the software with area-weighted average by using surface integrals.The Reynolds number (Re), which is a ratio of inertial force to viscous force, is expressed as:

$$Re = \frac{\rho D_h V}{\mu} \tag{9}$$

where ρ density of the fluid, D_h is the hydraulic diameter of the tube, V velocity of the fluid and μ the is dynamic viscosity of the fluid.
The average Nusselt number (Nu), which is ratio of convective heat transfer rate to conductive heattransfer rate, is expressed as:

$$Nu = \frac{h D_h}{k} \tag{10}$$

k is thermal conductivity of the fluid, and the average convective heat transfer coefficient (h)along the test section is calculated as:

$$h = \frac{q}{\Delta T} \tag{11}$$

q is constant heat flux applied onto the wall surface of the entrance and test section. The averagetemperature difference (ΔT) in this equation is expressed as:

$$\Delta T = T_s - T_b \tag{12}$$

where T_S is wall surface of the test section temperature and T_b is the fluid bulk temperature between inlet and outlet of the test section. The friction factor (f) defined by Fanning friction factor is expressed as:

$$f = \frac{\Delta T}{1/2 \rho V^2 \frac{L}{D_h}} \tag{13}$$

where ΔP is the pressure difference between inlet and outlet of the test section. L is the length of the test section of the tube. Performance evaluation criteria (PEC) is used to evaluate heat transfer performance and hydraulic characteristics with Eq. (14) at the same Re number.

$$PEC = \frac{Nu/Nu_0}{(f/f_0)^{1/3}} \tag{14}$$

where Nu_0 and f_0 represent the Nusselt number and friction factor of the non-applied any passiveheat transfer enhancement technique, respectively.

3.RESULTS AND DISCUSSION

3.1.VALIDATION OF NUMERICAL METHODOLOGY

The validation of the numerical methodology should be ensured to prove the accuracy of the study. In this scope, the validation of the study for the (CST) is obtained according to the Nusselt number and the friction factor versus Reynolds number as given in Figure 4 by comparing Dittus and Boelter equation [18] and Blasius equation [19]. Obtained maximum deviations from the validation study are approximately ± 3.59% and ± 12.93% for the Nusselt number and the friction factor, respectively.

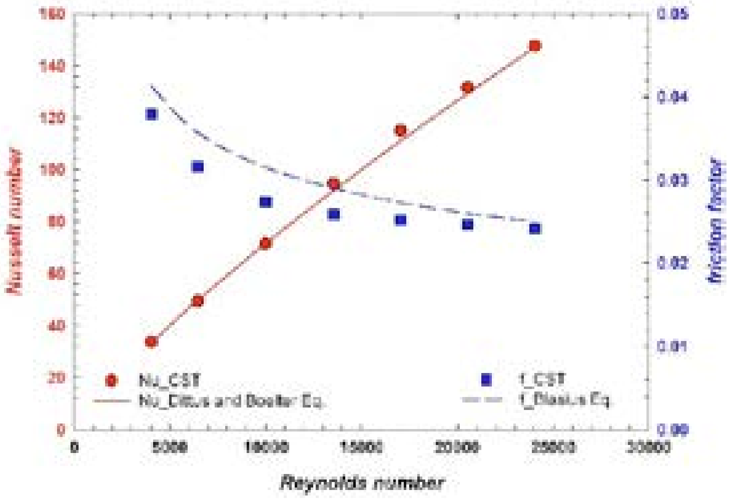

Figure 4. Validation of CST for both the Nusselt number and the friction factor versus the Reynolds number

3.2.RESULTS OF THE NUMERICAL STUDIES

In this study, the effect of the AR and the PL of the ETT is numerically investigated on the heat transfer, flow characteristics and PEC. The AR is adjusted as 1.5 and 2.0 by keeping constant the hydraulic diameter of 17.0 mm which belongs to the CST. The ETTs are created to have PL of 50, 100 and 200 mm.

3.3.HEAT TRANSFER

Nusselt number is used to interpret the heat transfer performance of the heat exchanger tube. Fig. 5 shows the distribution of Nusselt number results with respect to Reynolds number for all considered tube configurations. As widely known, as the Reynolds number increases the Nusselt number increases for all cases, since the turbulent flow plays a major role in enhancing the convective heat transfer rate in the internal tube flow.

It is clearly seen from Fig. 5 that the increase in the AR leads to an increase the Nusselt number for all Reynolds numbers. The reason of this result is the fact that as the AR increase, the diameter of the ETT is shrunk. This phenomenon leads to thin the thickness of the thermal boundary layer.

When the heat transfer performance of the ETT is examined in terms of the PL, as the PL decreases, the Nu increases for all Reynolds numbers. One of the reasons for this result is the decrease in the PL means to increase in the surface area of the ETT. As expected, the heat transfer surface area is directly proportional to the convective heat transfer. Another reason for that is as the PL decreases through the tube, the tendency of the disrupted boundary layers to affect each other has increased. As a result, the decrease in the PL inclines both to increase the heat transfer surface area and to disrupt the thermal boundary layer more.

The statements mentioned above can be supported by Fig. 6 where the temperature contours for all considered tube cases are shown. It is seen from the figure that the thermal boundary layer is disrupted and the bulk temperature is higher for the case having a higher Nusselt number result than the others. For instance, a zone having a minimum temperature that is 300 K is not observed for the ETT_AR=2.0_PL=50, while as the PL increases for the other cases, the zone having a minimum temperature expands. Furthermore, it is seen the intensification of the secondary flow from Fig.8. The secondary flow is another phenomenon causing to enhances the convective heat transfer. The velocity gradient at the cross-sectional area of the tube is observed in the Fig. 8. The more irregular velocity gradient occurs at the secondary flow, the fluid is more severely mixed. Therefore, it serves to enhance the convective heat transfer.

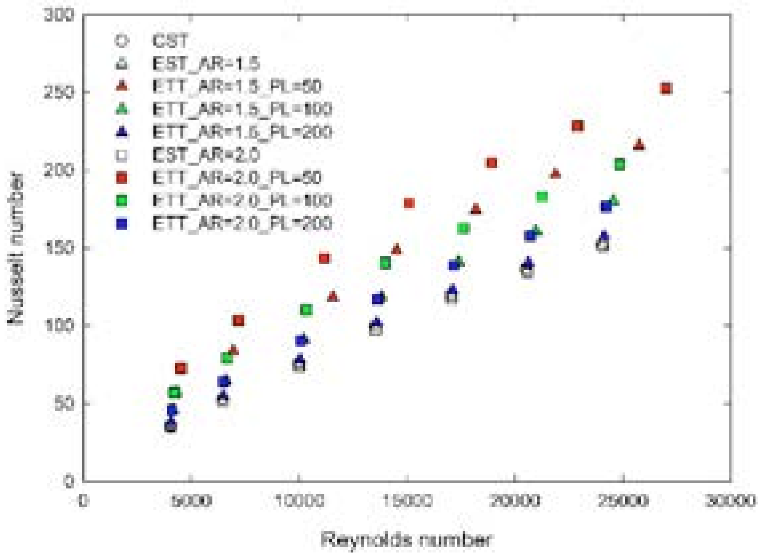

Figure 5. Distribution of Nusselt number results versus the Reynolds number

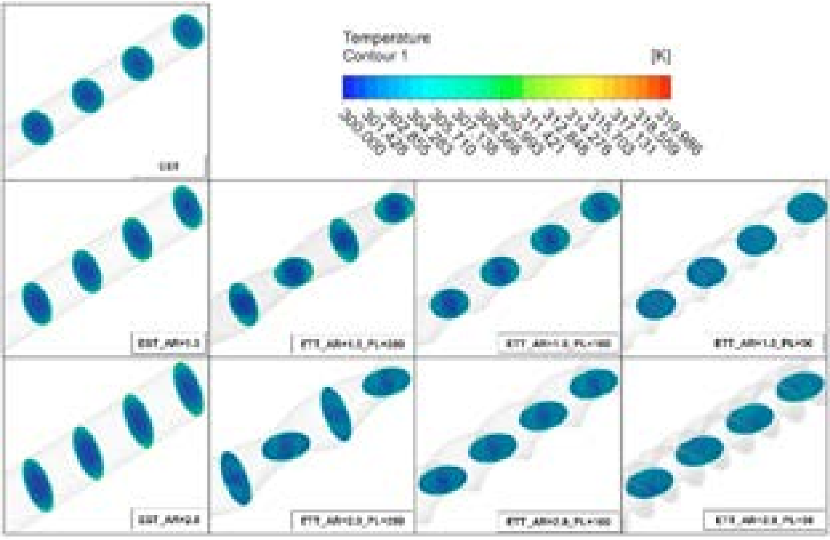

Figure 6. Temperature contours at different cross-sectional areas for the all cases at Reynolds number of 14,000.

3.4.FLOW CHARACTERISTIC

Friction factor (f) is used to discuss and interpret the flow characteristic occurring through the considered tube cases. Fig. 7 shows the distribution of friction factor results with respect to Reynolds number for the cases. It is widely known that as the Reynolds number increases, the friction factor decreases for the internal tube flow. Because, the inertial forces are dominant on the viscous forces adjacent the inner surface of the tube. The highest friction factor result is observed for the case of ETT_AR=2.0_PL=50 for all Reynolds numbers. There is a directly proportional among the various PL which is that as the PL decreases the friction factor increases since the size of the obstacle surface area increases through the ETT.

Moreover, as the AR increases, the friction factor increases. It is because the velocity magnitude increase and the pressure drop increases when the tube section is shrunk.

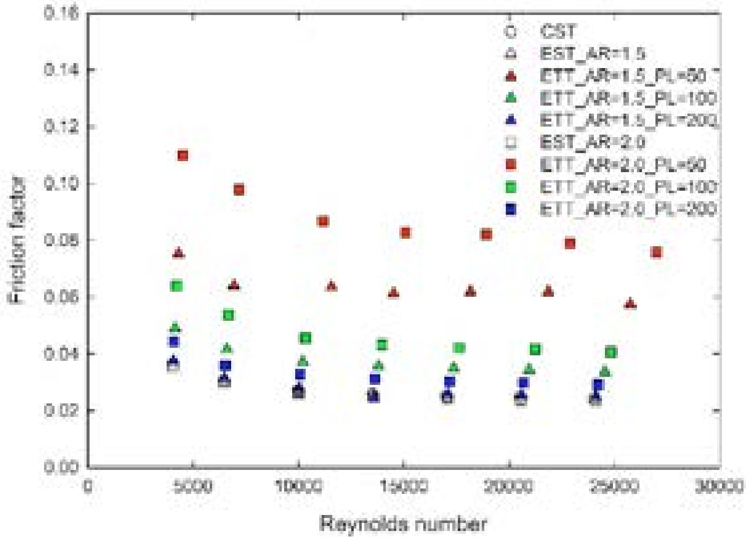

Figure 7. Distribution of friction factor results versus the Reynolds number

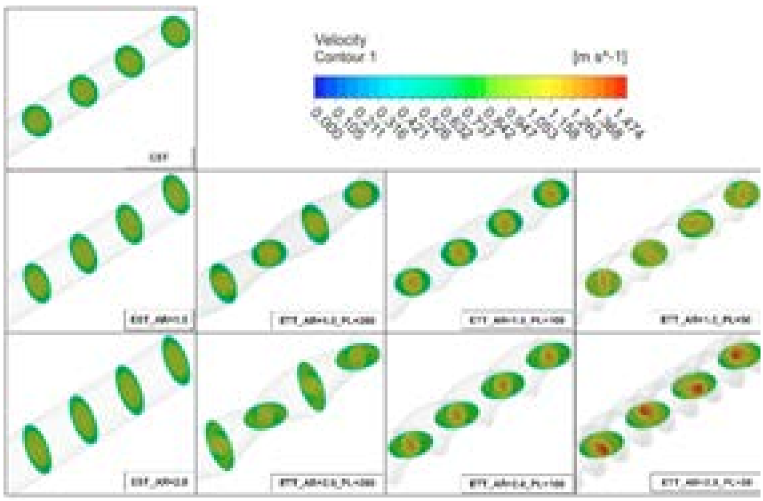

Figure 8. Velocity contours at different cross-sectional areas for the all cases

In addition to the results mentioned above, it should be emphasized that the kept the hydraulic diameter constant does not cause severe effects for neither the Nusselt number nor the friction factor. Therefore, comparing the ETTs having various AR and PL with each other is sensible and acceptable.

3.5.PERFORMANCE EVALUATION CRITERIA

Performance evaluation criteria (PEC) is widely used to determine the performance of the heat exchangers in terms of both heat transfer and hydraulic performance, simultaneously. It is accepted that if the PEC value is higher than 1.0, the case is beneficial to employ the related case in the application. For this purpose, Fig. 9 is given to compare the PEC results obtained from the considered cases versus the Reynolds number. It is resulted that the highest PEC value is obtained as 1.39 from the case of ETT_AR=2.0_PL=50 at a Reynolds number of approximately 4524. The second high PEC value is obtained from the case of ETT_AR=2.0_PL=100. It shows that the AR has more effect than PL on the PEC. It is because the increase in the heat transfer is greater than the decrease in the friction factor when the AR and the PL is compared with each other.

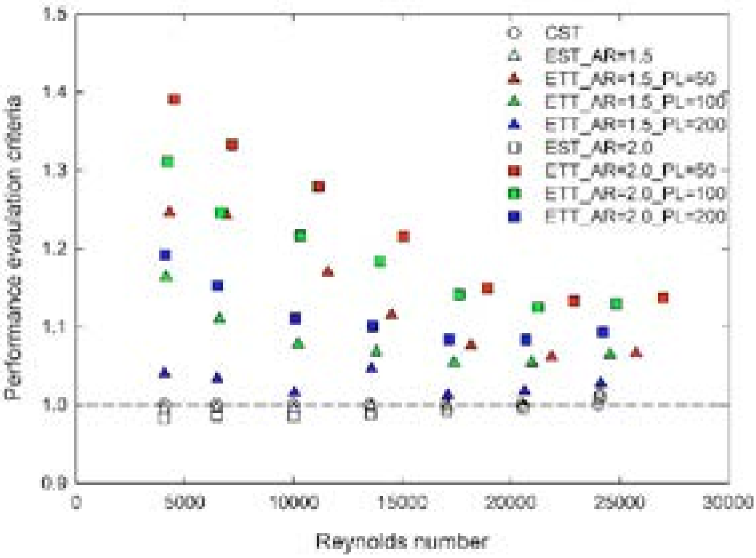

Figure 9. Distribution of performance evaluation criteria results versus the Reynolds number

4.CONCLUSION

This study presents a numerical study investigating the effects of the aspect ratio (AR) and the pitch length (PL) of the elliptical twisted tube (ETT) on heat transfer, flow characteristics and performance evaluation criteria (PEC). All tube cases are considered by keeping the hydraulic diameter constant. The flow conditions are turbulent corresponding the Reynolds number ranging from approximately 5000 to 25,000. Revealed conclusions from the study are given in the followings:

1. When the hydraulic diameter is kept constant for the ETTs, the Nusselt number, the friction factor and the PEC are not severely affected.
2. The increase in the AR leads to an increase in the Nu, the f and the PEC for all Reynolds numbers.
3. The decrease in the PL leads to an increase in the Nu, the f and the PEC for all Reynolds numbers.
4. The highest Nu is obtained as 252.52 at Re of 26998 for ETT_AR=2.0_PL=50. At the same time, the highest Nu corresponds to 1.5 times the smooth tube at the same the Re.
5. The highest PEC value is obtained as 1.39 for ETT_AR=2.0_PL=50 at Re of 4524.

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