Field synergy analysis of six starts spiral corrugated tube under high Reynolds number

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Abstract. Coaxial heat exchanger is widely used in air conditioning, refrigeration etc., due to its highly efficient heat transfer performance. Spiral corrugated tube plays an important role in coaxial heat exchanger. In this paper, the numerical model of a six starts spiral corrugated tube and a smooth tube with the same size are developed. The temperature field and the velocity field of their streamline and longitudinal vortex are investigated respectively. Then, their heat transfer and pressure drop performance inside the spiral corrugated tube under different high Reynolds number is investigated by compared their Nusselt number and friction coefficient. Meanwhile, their field synergy performances with their field synergy angles are presented. The result shows that the Nusselt number and friction coefficient of spiral corrugated tube are always larger than the smooth tube, and with the increasing of Reynolds number, the heat transfer performance of SCT becomes better than smooth tube, however, the friction coefficient ratio also increases synchronously. And in spiral corrugated tube, the field synergy angel is smaller than in the smooth tube. This work can be referred by some who are also dealing with spiral corrugated tube and its heat performance research.

1. Introduction
Coaxial heat exchangers have been widely used in petroleum, chemical, energy and other process industries. With the development of industrial technology, enhanced heat transfer technology has made a great progress and it is also widely applied to the heat transfer devices, especially the application of enhanced heat transfer tubes like spiral corrugated tube (SCT).

Several researchers have done many meaningful research works on SCT. For single start spiral corrugated tubes, some scholars carried out their research from the perspective of different flow conditions. Some focused on laminar and transitional flow conditions. Rainieri \textit{et al.} \cite{1, 2} studied the convective heat transfer performance of high viscosity fluid and made a comparison with smooth tube, and the experimental results show that the heat transfer performance of spiral corrugated tube was better than the smooth tube. Bhattacharyya \textit{et al.} \cite{3} and Saha \textit{et al.} \cite{4} researched the heat transfer performance and hydraulic performance of heat exchange tube under laminar flow condition respectively. Pal \textit{et al.} \cite{5} carried out an experimental study on single start corrugated tube under laminar flow condition, and the results showed that both the flow resistance coefficient and heat transfer coefficient in tube increased. Garcia \textit{et al.} \cite{6} conducted the heat transfer and pressure drop experiments of three heat exchange tubes under laminar and transitional flow conditions. Meanwhile, others paid attention to the turbulent flow condition. Zimparov \textit{et al.} \cite{7, 8} analyzed the flow resistance coefficient and heat transfer coefficient of single start SCT under turbulent flow condition. Li \textit{et al.} \cite{9} studied the flow and heat transfer characteristics of spiral SCT by numerical simulation.
At the same time, some researchers also paid attention to the multi-head SCT. Lazim et al. [10] analyzed the enhanced heat transfer performance of four starts SCT by numerical simulation, and the results showed that the heat transfer coefficient of spiral corrugated tube was much larger than smooth tube. Ahn. S. W et al. [11] also carried out an experimental study on the heat transfer performance of four starts SCT. Meanwhile, Chen et al. [12] also investigated the hydrodynamic and heat transfer performance of four-start SCT focus on the effect of corrugation angles. Furthermore, Gao et al. [13] took a numerical study on multiple thread SCT focus on the pressure drop influenced by pitch length.

In this paper, the numerical model of a six starts spiral corrugated tube and a smooth tube with the same size are developed. Then, the temperature field and the velocity field of their streamline and longitudinal vortex are investigated respectively. Meanwhile, their heat transfer and pressure drop performance under different high Reynolds number is investigated, compared their Nusselt number and friction coefficient. At the same time, their field synergy performances with their field synergy angles are presented. This work can be referred by someone who deals with SCT and its heat performance research, or someone who concern about the field synergy analysis of heat exchangers.

2. Numerical model

In this part, the mathematical model and the geometric model of SCT and the smooth tube are carried out firstly. Then, the field synergy analysis is introduced. Furthermore, the calculation of Nusselt number and friction coefficient are presented respectively. Finally, the grid and bounding condition of SCT and smooth tube is introduced.

2.1. Mathematical model

Supposing that the heat transfer process in SCT is single-phase fluid convection heat transfer, we can make the following assumptions as follows: working medium is supposed as incompressible fluid, and the fluid flow is supposed as steady flow; and heat generated from viscous dissipation in fluid flow process can be ignored; meanwhile, gravity is ignored; at the same time, the contact surface of wall and fluid medium is supposed as no slip and without deformation. Therefore, the continuity of flow and heat transfer processes in SCT can be described as Eq. (1) ~ (5):

\[ \nabla \cdot (\rho U) + \frac{\partial \rho}{\partial t} = 0 \]  
(1)

\[ \frac{\partial u_x}{\partial t} = f_x - \frac{1}{\rho} \frac{\partial p}{\partial x} + \nu \left( \frac{\partial^2 u_x}{\partial x^2} + \frac{\partial^2 u_x}{\partial y^2} + \frac{\partial^2 u_x}{\partial z^2} \right) \]  
(2)

\[ \frac{\partial u_y}{\partial t} = f_y - \frac{1}{\rho} \frac{\partial p}{\partial y} + \nu \left( \frac{\partial^2 u_y}{\partial x^2} + \frac{\partial^2 u_y}{\partial y^2} + \frac{\partial^2 u_y}{\partial z^2} \right) \]  
(3)

\[ \frac{\partial u_z}{\partial t} = f_z - \frac{1}{\rho} \frac{\partial p}{\partial z} + \nu \left( \frac{\partial^2 u_z}{\partial x^2} + \frac{\partial^2 u_z}{\partial y^2} + \frac{\partial^2 u_z}{\partial z^2} \right) \]  
(4)

\[ \frac{\partial (\rho T)}{\partial t} + \nabla \cdot (\rho \vec{u} T) = \nabla \cdot \left( \frac{h}{C_p} \text{grad}T \right) + S_T \]  
(5)

Here, \( \rho \) refers to density, \( t \) refers to time, \( u_x, u_y, u_z \) refer to velocity along direction x, y and z respectively, \( \nabla \cdot (\rho U) \) refers to the divergence of mass flux \( \rho U \), \( f_x, f_y, f_z \) refer to body face of micro unit, \( P \) refers to the pressure of micro unit. \( T \) refers to temperature, \( h \) refers to fluid heat transfer coefficient, \( S_T \) refers to the heat energy which is converted from mechanical energy.

2.2. Geometric model

In this paper, STC has six starts. The structural parameters of the spiral corrugated tube mainly include pitch \( p=35mm \), groove depth \( e=2.25mm \), and the tube external diameter \( D_i=8mm \). The total length of the model is 600mm. Fig.1 is the physical graph of spiral corrugated tube.
2.3. Field synergy analysis

The field synergy principle of boundary-layer flow has been systematically expounded by Guo and Tao et al. [14,15]. It shows that the convective heat transfer not only depends on the thermal physics properties, the temperature difference and the fluid flow velocity, but also depends on the angle between the velocity vector and the temperature gradient. That is to say, reducing the intersection angle between the velocity and the temperature gradient can enhance the convective heat transfer by optimizing the coordination of velocity and temperature gradient. In our previous work [16,17], we compared the dimple jacketed heat exchanger with conventional jacketed heat exchanger. By adding dimples to conventional jacketed heat exchanger, it can improve the heat transfer coefficient of jacketed heat exchanger.

Due to the spiral shape of the inner tube, SCT is much more effective than smooth tube, because the continuity of the fluid flow inside SCT generating significant secondary flow, so that the main direction to the vertical velocity component increases. Thus, the velocity and temperature fields get a good field synergistic effect and enhance the heat transfer performance.

2.4. Nusselt number and friction coefficient

Fluid under turbulent flow condition has a better heat transfer performance, thus the heat transfer process is always set under turbulent condition in industrial engineering, and in industrial engineering Zeder-Tate formula is widely adopted to predicate the heat transfer performance of tube heat exchanger [18]. Here, the applicable conditions are as follows: \( Re \geq 10000 \) and \( Pr=0.7~16700 \), and the aspect ratio of tube \( l/d \geq 60 \).

\[
Nu = 0.027Re^{0.8}Pr^{1/3} \left( \frac{\mu}{\mu_w} \right)^{0.14} \tag{6}
\]

The calculation of numerical simulation of total heat transfer and Nusselt number are as follows:

\[
Q = \frac{C_p q_m (t_i - t_o)}{\mu} \quad \tag{7}
\]

\[
Nu = \frac{h d}{\lambda} = \frac{d}{\lambda} \frac{Q_m}{A_i} (t_i - t_o) \quad \tag{8}
\]

Here, \( Q \) refers to the total heat transfer rate; \( q_m \) refers to mass flow; \( t_i \) refers to inlet temperature and \( t_o \) refers to outlet temperature. \( h_i \) refers to convective heat transfer coefficient. \( \lambda \) refers to the thermal conductivity and \( d \) refers to inner diameter.

The empirical formula of friction coefficient is Blasius formula [18]. The applicable condition of is \( Re \leq 100000 \). Therefore, the friction coefficient can also be obtained.

\[
f = 0.3164Re^{-0.25} \tag{9}
\]

\[
h_f = \frac{\Delta p}{\rho g} = f \frac{l}{d} \frac{v^2}{2g} \quad \tag{10}
\]

\[
f = \Delta p \frac{d}{l} \frac{2}{\rho v^2} \quad \tag{11}
\]
Here, \( \Delta p \) refers to the pressure difference of tube inlet and tube outlet; \( f \) refers to flow resistance coefficient. According to the above mentioned empirical formulas, we can calculate both the theoretical value and numerical simulation of resistance coefficient.

2.5. Grid and bounding condition

Fig.2 shows the mesh of SCT. We conduct a grid independence check by taking the smooth tube and SCT as examples and finally determine 0.6mm as the simulation mesh size in software Gambit.

![Fig.2 Mesh of SCT](image)

For boundary condition in the numerical work, the heat transfer medium is water. The specific values are described as follows: density is 997.8kg/m\(^3\), specific heat capacity is 4178J/kg·K, thermal conductivity is 0.667W/m·K, and viscosity is 0.4061mPa·s. The inlet boundary condition is set as velocity inlet and inlet temperature is 363K. The outlet boundary condition is set as pressure outlet and outlet pressure is 0Pa. The wall chooses no-slip wall condition. The paper deals with tube side, so the wall heat transfer is 0 and temperature of wall surface is set as 300K. The material of wall is copper.

Here the model is single-phase flow and heat transfer model. The Reynolds number in tube ranges from \( 1 \times 10^4 \) to \( 6 \times 10^4 \), while in SCT Reynolds number is achieved in the smooth tube with same size and inlet velocity. Due to that the internal heat exchange field of SCT is complex, so realizable turbulence model combines with wall function method are used for the tube. The calculation processes are set as follows: steady flow, 3D and pressure implicit solver is selected; SIMPLE algorithm is used to couple pressure and velocity; Pressure in discrete formatting chooses standard format. Momentum, turbulent kinetic energy, turbulent dissipation rate and energy all choose second-order upwind scheme. In the setting of convergence conditions, the absolute valve of energy residual is less than \( 1 \times 10^{-8} \), and the absolute valve of quality residual is less than \( 1 \times 10^{-6} \).

3. Results and discussion

In this part, the performances of SCT and smooth tube are detail compared. And the temperature field is analyzed firstly. Then, the velocity field of their streamline and longitudinal vortex are investigated respectively. Meanwhile, the heat transfer performances with their Nusselt number \( Nu \) and friction coefficient \( f \) are analyzed. Finally, their field synergy performances with their field synergy angles are presented.

As we all know, the inlet part can influence the simulation result. Thus, in order to eliminate the influence of the inlet part, we firstly analyze the temperature and pressure of SCT on \( z \) direction under \( Re=10000 \), which is shown in Fig.3. As we can see, in the inlet part, the pressure changes rapidly, especially in the part of 0~0.05m. In the part of 0.1~0.55m, the change of temperature and pressure seem steady. With the increase of Reynolds number, the scope of influenced region decreases. Therefore, we choose the part 0.1~0.5m as the research part.
3.1. Temperature field analysis

In order to investigate the temperature distributions inside SCT and its advantages compared to smooth tube, we set cross section z=0.3m as an example, which is shown in Fig.4. In Fig.4, the smooth tube and SCT are both under Re=20000. As we can see, in smooth tube, the temperature distributions with its isotherms are similar to a series of concentric circles, and as the distance from the center increases, the distance between two adjacent isotherms decreases. In other words, the temperature gradient becomes larger and larger when it is the farther away from the center.

For SCT, due to the changing of its structure, the fluid flow inside SCT is more complex than inside smooth tube, and the isotherms inside tend to be irregular circles, especially near the corner parts. Meanwhile, compared with smooth tube, the interval distances of isotherms are more uniform. Furthermore, in smooth tube, there are larger areas occupied by the high temperature zone compared with SCT, which indicates that SCT has a better heat transfer performance.

3.2. Velocity analysis

3.2.1. Streamline analysis. In order to observe the fluid flow inside the smooth tube and SCT, the streamlines inside are obtained under Re=20000 which are shown in Fig.5. As is shown in Fig.5,
Streamlines inside smooth tube are all straight lines parallel to the axis, while the streamlines inside SCT are helix lines. That is to say, inside smooth tube, there is little secondary flow or the secondary flow rate is very small, while secondary flow benefits heat transfer obviously.

Fig. 5 Streamline inside the smooth tube and SCT

Fig. 6 Velocity contour in cross section z=0.2, 0.3, 0.4 m of smooth tube and SCT

Fig. 7 Velocity vectors distribution in cross section z=0.3 m of SCT
In order to further investigate the fluid inside, Fig. 6 presents the velocity contour in the cross sections $z=0.2, 0.3, 0.4\,\text{m}$ of smooth tube and SCT under $Re=20000$. As we can see, inside smooth tube, the secondary flow velocity distributes in $0-0.0004\,\text{m/s}$. However, the secondary flow velocity distribution inside SCT is $0-0.6\,\text{m/s}$, which are thousands times of the smooth tube. Therefore, SCT pay a very significant role to turn out secondary flow.

Here, the velocity vectors distribution in cross section $z=0.3\,\text{m}$ of SCT under $Re=20000$ is presented as an example. As we can see in Fig. 7, the directions of velocity vectors are the same with the spiral direction, and these velocity vectors have same axis with SCT. Meanwhile, the values of velocity vectors increase with the increasing of the distances of the centre point, which can be also investigated in Fig. 6. However, due to the viscosity of the boundary wall, the velocity vectors decrease.

3.2.2. Longitudinal vortex analysis. Vortex is the physical movement to explain the rotating of fluid. It equals to rotation velocity or the twice of the angular velocity. And it can be obtained through Eq. (12) ~ (16). If $\Omega=0$, the fluid field is none rotational flow, while $\Omega\neq0$, the fluid field is the rotational flow. Vortex is a kind of vector, while the longitudinal vortexes in this part have the direction which perpendiculars to the flow direction.

$$\Omega = 2\omega = \nabla \times \nu$$

$$\Omega = \begin{pmatrix} \frac{\partial v_z}{\partial y} - \frac{\partial v_y}{\partial z} \\ \frac{\partial v_x}{\partial z} - \frac{\partial v_z}{\partial x} \\ \frac{\partial v_y}{\partial x} - \frac{\partial v_x}{\partial y} \end{pmatrix}$$

$$\omega_x = \frac{\partial v_y}{\partial z} - \frac{\partial v_z}{\partial y}$$

$$\omega_y = \frac{\partial v_x}{\partial z} - \frac{\partial v_z}{\partial x}$$

$$\omega_z = \frac{\partial v_y}{\partial x} - \frac{\partial v_x}{\partial y}$$

In order to investigate the longitudinal vortexes inside smooth tube and SCT, we choose cross sections $z=0.2, 0.3, 0.4\,\text{m}$ under $Re=20000$ as examples, which are shown in Fig. 8.

Fig. 8 Longitudinal vortex in cross section $z=0.2, 0.3, 0.4\,\text{m}$ of smooth tube and SCT
As we can see from Fig.8, longitudinal vortexes inside SCT are much larger than longitudinal vortexes inside smooth tube, near four orders of magnitude. Thus, we regard there are obvious longitudinal vortexes inside SCT while there are little longitudinal vortexes inside smooth tube. Meanwhile, inside SCT, we can find that the maximum longitudinal vortexes mainly locate in the central parts of SCT while the minimum mainly locate near the boundary wall. This is because in the bottom part of SCT, the secondary flow velocity is weak while the viscosity of the boundary wall is large, which turn out the boundary layer detachment and fluid turbulence.

3.3. Heat transfer performance analysis
In this part, the heat transfer performances with their Nusselt number $Nu$ and friction coefficient $f$ are analyzed of the smooth tube and SCT. The smooth equation value in Fig.9 is achieved by Eq. (6) and Eq. (9), while the SCT simulation value and smooth simulation value are obtained by Eq. (7) ~ (8) and Eq. (10) ~ (11). As we can see, the simulation result can be verified since the error between the empirical formula and the simulation result are both less than 3%. Then, comparison between smooth tube and SCT is carried out. As we can see in Fig.9, under the same $Re$, $Nu$ and $f$ of SCT are always larger than $Nu$ and $f$ of smooth tube. Meanwhile, under turbulence condition, with the increasing of $Re$, the convective heat transfer is strengthen, which can also be seen from Fig.9. While for the friction coefficient, when $Re$ is larger than 30000, it becomes steady.

Fig.9 $Nu$ and $f$ comparison of smooth tube and SCT

Fig.10 $Nu_e/Nu_t$ and $f_e/f_t$ comparison of smooth tube and SCT
In order to further comparing the heat transfer performances of smooth tube and SCT, Fig.10 presents the $\frac{Nu_{SCT}}{Nu_{smooth}}$ and $\frac{f_{SCT}}{f_{smooth}}$ comparison of smooth tube and SCT. As we can see, when $Re=1\times10^{5}$--$6\times10^{5}$, $Nu$ of SCT is 1.32--1.363 times of $Nu$ of smooth tube, while the $f$ of SCT is 1.97--2.51 times of smooth tube $f$. And with the increasing of $Re$, the heat transfer performance of SCT becomes better than smooth tube, however, the friction coefficient ratio also increases synchronously.

3.4. Field synergy angle analysis

In order to further proving the advantage of SCT than smooth tube, the field synergy analysis is carried out comparing their field synergy angles. As is shown in Fig.10, it is the field synergy angle distributions in cross section $z=0.3m$ of smooth tube and SCT under $Re=10000$.

As we can see, in SCT, its field synergy angles distribute in the range of 70°--90°, while in the smooth tube, its synergy angles are all nearly 90°. Then is to say, the field synergy angle also indicate that SCT has a better heat transfer performance. Meanwhile, in SCT, we also find that in the central part, the field synergy angles are smaller than other parts. It is because of the existence of longitudinal vortex, which improves the field synergy performance of velocity field and temperature field.

![Field synergy angle distribution in cross section z=0.3m of smooth tube and SCT](image)

4. Conclusion

In this paper, a numerical study of a six starts spiral corrugated tube and a smooth tube with same size is carried out focus on their heat transfer performance and its field synergy. Firstly, the temperature field and the velocity field of their streamline and longitudinal vortex are investigated respectively. Then, by compared their Nusselt number and friction coefficient, their heat transfer and pressure drop performance inside SCT under different high Reynolds number is investigated. Meanwhile, their field synergy performances with their field synergy angles are presented. The result shows that the Nusselt number and friction coefficient of SCT are always larger than the smooth tube, and with the increasing of Reynolds number, the heat transfer performance of SCT becomes better than smooth tube, however, the friction coefficient ratio also increases synchronously. And in SCT, its field synergy angles distribute in the range of 70°--90°, while in the smooth tube, its synergy angles are all nearly 90°, which also indicate that SCT has a better heat transfer performance. This work can be referred by someone who deals with SCT and its heat performance research, or someone who concern about the field synergy analysis of heat exchangers.

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References

[1] Rainieri S, Bozzioli F, Cattani L and Pagliarini G 2012 Experimental investigation on the convective heat transfer enhancement for highly viscous fluids in helical coiled corrugated tubes. J. Phys.: Conf. Series. IOP Publishing. 395(1) 012032.

[2] Bhattacharyya S, Saha S K 2012 Thermohydraulics of laminar flow through a circular tube having integral helical rib roughness and fitted with centre-cleared twisted-tape. Exp. Therm Fluid Sci. 42 154-162.

[3] Bhattacharyya S, Saha S K 2012 Thermohydraulics of laminar flow through a circular tube having integral helical rib roughness and fitted with centre-cleared twisted-tape. Exp. Therm Fluid Sci. 42: 154-162.

[4] Saha S K 2013 Thermohydraulics of laminar flow through a circular tube having integral helical corrugations and fitted with helical screw-tape insert. Chem. Eng. Commun. 200(3) 418-436.

[5] Pal P K, Saha S K 2014 Experimental investigation of laminar flow of viscous oil through a circular tube having integral spiral corrugation roughness and fitted with twisted tapes with oblique teeth. Exp. Therm Fluid Sci. 57 301-309.

[6] Garcia A, Solano J P, Vicente P G 2012 The influence of artificial roughness shape on heat transfer enhancement: Corrugated tubes, dimpled tubes and wire coils. Appl. Therm. Eng. 35 196-201.

[7] Zimparov V. 2004 Prediction of friction factors and heat transfer coefficients for turbulent flow in corrugated tubes combined with twisted tape inserts. Part 1: friction factors. Int. J. Heat Mass Transfer. 47(3) 589-599.

[8] Zimparov V. 2004 Prediction of friction factors and heat transfer coefficients for turbulent flow in corrugated tubes combined with twisted tape inserts. Part 2: heat transfer coefficients. Int. J. Heat Mass Transfer. 47(2) 385-393.

[9] Li Y, Wu J, Wang H, et al. 2014 Fluid flow and heat transfer characteristics in helical tubes cooperating with spiral corrugation. Energy Procedia, 2012, 17: 791-800.

[10] Lazim T M, Kareem Z S, Jaafar M N M 2014 Heat Transfer Enhancement in Spirally Corrugated Tube. I.RE.MO.S. 970-978.

[11] Ahn S W 2003 Experimental studies on heat transfer in the annuli with corrugated inner tubes. KSME Int. J. 17(8) 1226-1233.

[12] Chen X D, Xu X Y, Nguang S K. 2001 Characterization of the effect of corrugation angles on hydrodynamic and heat transfer performance of four-start spiral tubes. J. Heat Transfer. 123(6) 1149-1158.

[13] Gao X F, Qian J Y, Jin, Z. J., et al. 2014 Numerical study of pressure drop influenced by pitch length inside multiple thread spiral-grooved tube. Int. Conf. on Flow Dynamic. Sendai, Japan. October 8-10..

[14] Guo Z Y, Tao W Q, Shah R K 2005 The field synergy (coordination) principle and its applications in enhancing single phase convective heat transfer. Int. J. Heat Mass Transfer 48(9) 1797-1807.

[15] Tao W Q, Guo Z Y, Wang B X 2002 Field synergy principle for enhancing convective heat transfer—its extension and numerical verifications. Int. J. Heat Mass Transfer. 45(18) 3849-3856.

[16] Qian J Y, Jin Z J, Gao X F 2014 Numerical investigation on the heat transfer characteristics of dimple jacketed heat exchanger in the chemical post-processing integrated equipment. Int. Conf. on heat transfer & fluid flow. Prague, Czech. August 11-12

[17] Qian J Y, Jin Z J, Gao X F 2015 CFD analysis on pressure drop of dimple jacketed heat exchanger in chemical post-processing integrated equipment. 6th Int. Sym. on Advances in Computational Heat Transfer. Rutgers University, USA. May 25-29

[18] He C H, Feng X. 2007 Principles of Chemical Engineering. Beijing: Science Press