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To cite this article: Zhiwei Wu and Caifu Qian 2020 *J. Phys.: Conf. Ser.* **1549** 042114

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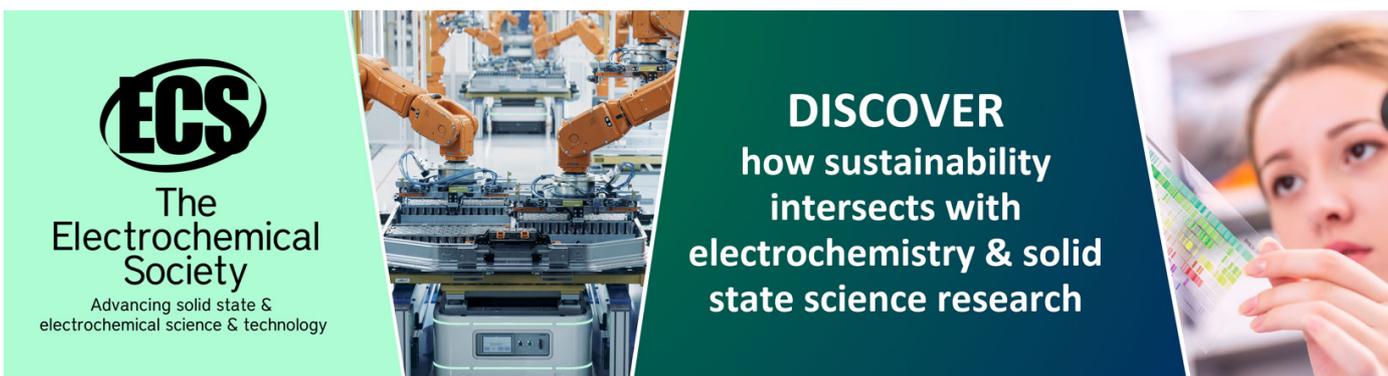
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# Numerical Investigation of Fluid Flow and Heat Transfer Characteristics in Helical Tube with Spiral Corrugations

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**Abstract.** The flow and heat transfer performance of helical tube with spiral corrugation are numerically studied and compared with helical smooth tube. The velocity field, temperature field and synergistic performance at different sections are analyzed. It is found show that the turbulence intensity is enhanced due to the existence of the spiral corrugations, and as a results, the helical tube with spiral corrugations has more efficient heat transfer capability. Compared with the helical smooth tube, the heat transfer performance of helical tube with spiral corrugations is improved by 20-50%, but the pressure drop increases by about 100%.

## 1. Introduction

Helical tube (HT) heat exchangers are widely used in the pharmaceutical, petroleum, food, chemical, metallurgical, power and textile industries due to their compact structure and high heat transfer coefficients [1-3]. Compared with the straight tube, the bending of the tube will lead to the formation of secondary flow, which increases the heat transfer coefficient. The research on the flow and thermal performance of HT has been paid more and more attention [4, 5]. Jayakumar et al. [6] numerically simulated the heat transfer performance of vertically oriented spiral tubes with different structural parameters, and established an association formula to predict the Nusselt number. Mahmoudi et al. [7] carried out experiments and simulations on the convective heat transfer performance of HT with TiO<sub>2</sub> / water as working medium. The results showed that the Nusselt value could be increased by 30% by using nanofluid instead of distilled water. Piazzade et al. [8] used different turbulence models to simulate the flow and heat transfer performance in HT. Ghorbani et al. [9] Studied experimentally influence of the pitch and diameter of the HT on the heat transfer performance of the shell side of the heat exchanger under laminar and turbulent flow. Beigzadeh et al. [10] used the artificial neural network model to estimate the heat transfer performance and resistance coefficient in the HT, and compared them with the existing empirical relationship. The results show that the model has excellent performance in predicting the Nusselt number and friction coefficient. Rough surface technology is a passive heat transfer enhancement method. Usually, the flow state of the fluid is changed by means of modifying the heat transfer surface and changing the shape of the flow passage, which will cause the fluid in the boundary layer to be disturbed greatly, so as to achieve the effect of enhancing heat transfer. In the aspect of flow and heat transfer, it is of great significance to design and improve the HT with enhanced heat transfer technology. Li et al. [11] analyzed the influence of spiral corrugation parameters and Reynolds number on the flow and heat transfer in HT by numerical simulation. The results show that the heat transfer can be greatly improved due to the additional rotational motion.



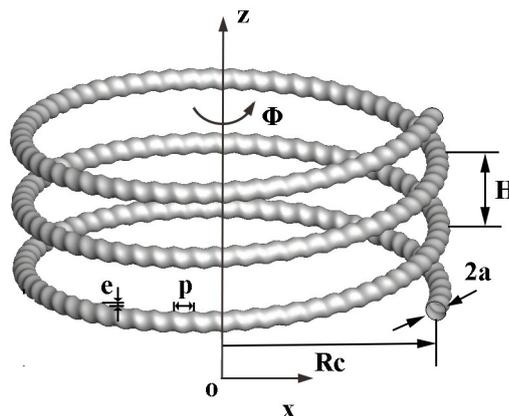
Zachár [12] compared the flow and temperature fields of HT and helical tube with spiral corrugated (HTSC) by numerical simulation. The results show that the heat transfer rate and pressure drop of HTSC are increased by 80-100% and 10-600% respectively. Zhang et al. [13] conducted a numerical study on the heat transfer and pressure drop of the spherical HTSC. The results show that the vortex caused by the corrugated structure destroys the flow boundary layer, which increases the turbulence intensity of the flow, and strengthens the heat transfer process. Rainieri et al. [14] carried out experimental research on forced convection in straight tube and HTSC in the range of low Reynolds number, in order to verify the effectiveness of passive reinforcement technology in processing highly viscous fluids.

In this paper, the flow and heat transfer performance of helical tube with spiral corrugation (HTSC) are numerically studied and compared with helical smooth tube (HT). The velocity field, temperature field and synergistic performance at different sections are analyzed.

## 2. Numerical simulation

### 2.1. Physical model

Fig. 1 shows the basic structure of HTSC. The spiral corrugation is located on both sides of the HT, forming a spiral tube wall. Geometric parameters of HTSC include curvature radius ( $R_c$ ), pitch of HT ( $H$ ), inner diameter ( $2a$ ), pitch of spiral corrugated ( $P$ ), helical angle ( $\Phi$ ) and depth of spiral tube ( $e$ ).



**Fig. 1** Physical model of HTSC

### 2.2. Boundary conditions

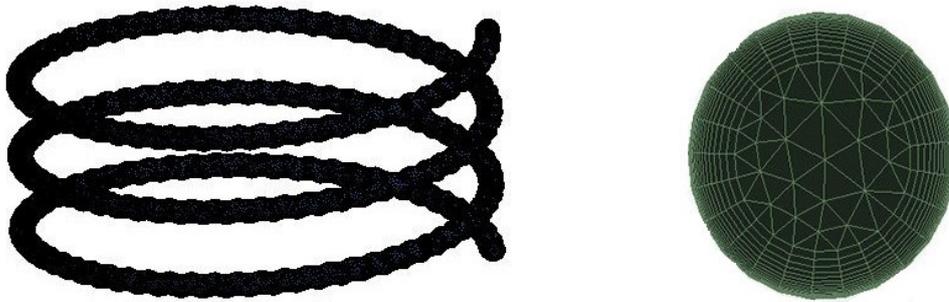
The boundary conditions are as follows: The inlet boundary condition is set as the velocity inlet which is related to Reynolds number. The calculated working fluid inlet temperature is 293 K, and the wall temperature is constant 323 K. The static pressure outlet boundary condition is used, and the relative pressure is set to zero. In this paper, the Realizable  $K-\epsilon$  turbulence model is adopted. The pressure-speed coupling is based on the SIMPLEC scheme. Bending moment, turbulence, and energy equations are discretized by a second-order upwind scheme. In addition, convergence criteria of  $10^{-5}$  for continuity and velocity components and  $10^{-6}$  for energy are used, respectively. The heat transfer medium in the tube is water. The physical properties of water are assumed to be constant which are listed in Table 1.

**Table 1** Physical properties parameter of water

Parameter	Value
$c_p$	4182, J kg <sup>-1</sup> K <sup>-1</sup>
$\mu$	0.001003, kg m <sup>-1</sup> s <sup>-1</sup>
$\rho$	998.2, kg m <sup>-3</sup>
$\lambda$	0.6, Wm <sup>-1</sup> K <sup>-1</sup>

### 2.3. Grid model and validation

Due to the twisted geometry of spiral corrugation, the flow near the wall is very complex, so unstructured meshes are generated in the fluid region. There are large temperature gradient and velocity gradient near the wall, so the boundary layer is divided into 15 layers, as shown in Fig. 2. In order to ensure the accuracy of the numerical results, the grid independence test is carried out, as shown in Table 2. It is seen that even the grid nodes is significantly increased, the difference of Nusselt number is less than 1%. Considering the limitation of computing resources, the grid model with 141737 nodes is employed in the following analysis.

**Fig. 2** Grid model of HTSC ( $a=10\text{mm}$ ,  $P=20\text{mm}$ ,  $H=66.7\text{mm}$ ,  $e=1\text{mm}$ ,  $R_c=200\text{mm}$ )**Table 2.** Comparison of  $t_{\text{out}}$ , pressure drop, and Nu for different numbers of grid nodes

No. of nodes(million)	$t_{\text{out}}(\text{K})$	Pressure drop(Pa)	$Nu$
1.0	313.30	9468.59	202.03
1.4	313.25	8528.53	202.02
2.3	313.31	8462.08	203.67
5.5	313.24	8807.45	202.86

At the same time, it is necessary to verify the applicability of the turbulence model before analyzing the HTSC. Due to the lack of experimental data on turbulent flow and heat transfer characteristics in HTSC, this paper compares the numerical results of friction coefficient and Nusselt number in a smooth HT with the empirical formulas given by C.J. bolinder [15] and C.X. Lin [16] as shown in Fig.3. The relevant formula is as follows:

$$f = 0.076 Re^{-0.25} + 0.00725 \left( \frac{R_c}{a} \right)^{-0.5} \quad (1)$$

$$Nu = 0.023 Re^{0.85} Pr^{0.4} \left( \frac{a}{R_c} \right)^{0.1} \quad (2)$$

It is seen from Fig. 3 that the numerical results are in good agreement with the empirical formula data, and the maximum deviation of  $f$  is 7.8% and  $Nu$  is 6.9%, indicating that the simulation results are reliable.

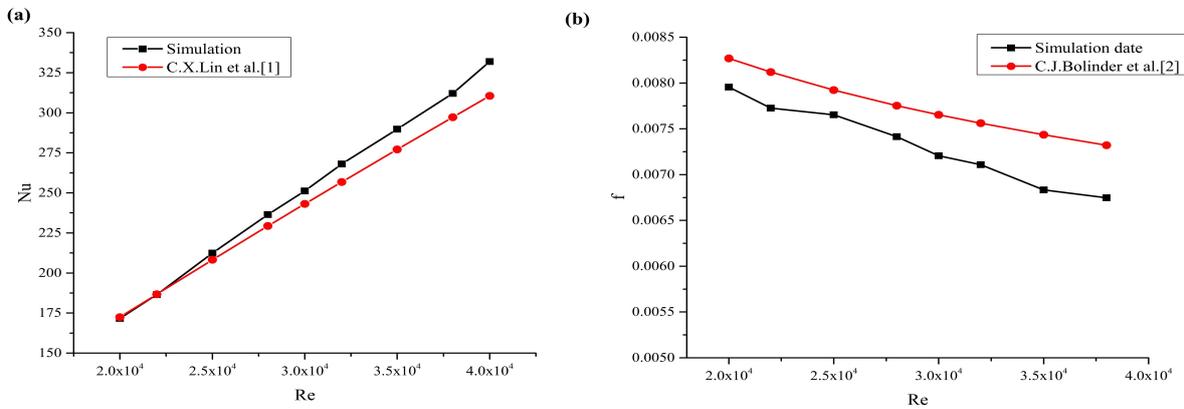


Fig.3 Comparison of the numerical results in HT with the empirical formulas: (a)for  $f$ , (b) for  $Nu$ .

### 3. Results and discussions

#### 3.1. Flow characteristics

In this paper, the flow and heat transfer performance of HTSC ( $H50, Rc100, e1, P20$ ) and HT( $H50, Rc100$ ) with the same helical pitch and helical diameter at  $Re = 20000$  are compared and analyzed. In addition, the influence of the structural parameters of spiral corrugations on the heat transfer and flow performance is studied.

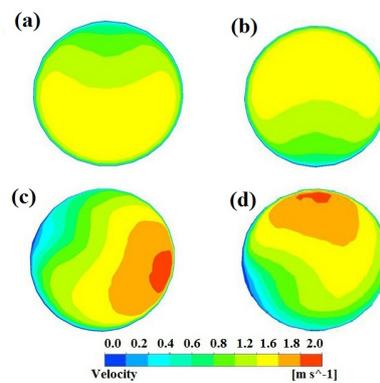
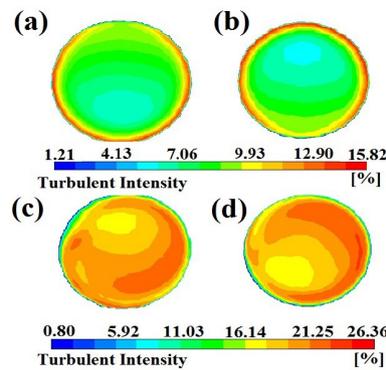


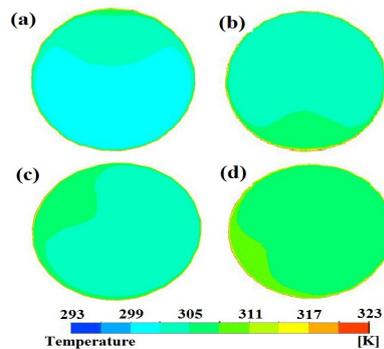
Fig.4 The velocity fields of cross section: (a) HT,  $\Phi=3\pi$ , (b) HT,  $\Phi=4\pi$ , (c) HTSC,  $\Phi=3\pi$ , (d) HTSC,  $\Phi=4\pi$ .(upper side of cross is the outside wall )



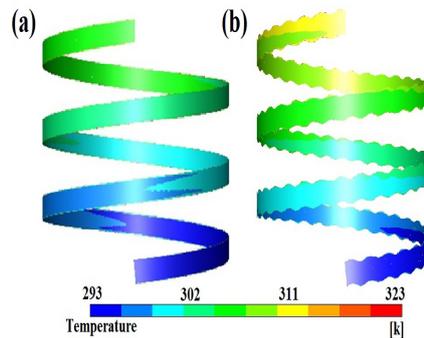
**Fig.5** The turbulent intensity of cross section: (a) HT,  $\Phi=3\pi$ , (b) HT,  $\Phi=4\pi$ , (c) HTSC,  $\Phi=3\pi$ , (d) HTSC,  $\Phi=4\pi$ . (upper side of cross is the outside wall )

Fig. 4 shows the velocity field of HT and HTSC on the cross section ( $\Phi=3\pi$ ,  $\Phi=4\pi$  and  $Re=20000$  ), and Fig. 5 shows the turbulence intensity on the cross section. It can be seen that the velocity gradient near the outer wall is much larger than that near the inner wall, which is mainly due to the centrifugal force caused by the bending of the tube. Under the action of centrifugal force, the HT produces secondary flow and vortex, which promotes the disturbance of fluid on the tube section. In addition, for the HT, the profile shape is saddle shape. In the HTSC, due to the existence of spiral corrugations, the fluid is periodically disturbed. The saddle shape of the profile is destroyed and the turbulence intensity is strengthened, which makes the temperature distribution of the fluid more uniform and plays the role of enhancing heat transfer.

### 3.2. Heat transfer performance

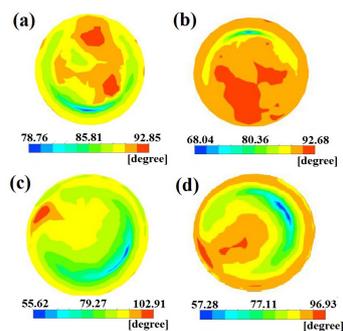


**Fig. 6** The temperature fields of cross section: (a) HT,  $\Phi=3\pi$ , (b) HT,  $\Phi=4\pi$ , (c) HTSC,  $\Phi=3\pi$ , (d) HTSC,  $\Phi=4\pi$ . (upper side of cross is the outside wall )



**Fig. 7** Distribution of fluid temperature in axial direction.

Fig. 6 shows the temperature field of HT and HTSC on the cross section ( $\Phi=3\pi$ ,  $\Phi=4\pi$  and  $Re=20000$ ). At the same position, the temperature of HTSC is much higher than that of HT. The temperature gradient near the outer wall is much higher than that near the tube wall. This is because the secondary flow caused by centrifugal force destroys the flow boundary layer and strengthens the heat transfer process. Fig. 7 shows the axial fluid temperature distribution. It is seen clearly that the outlet temperature of HTSC is higher than that of HT. Due to the existence of spiral corrugations, the fluid near the wall surface vortices along the corrugations, which is beneficial to destroy the boundary layer and enhance the turbulence intensity to enhance heat transfer. In addition, Guo et al. [17] analyzed the fluid flow in the boundary layer from the energy equation and proposed the field synergy principle of heat transfer enhancement, that is, the effect of the degree of synergy between the velocity vector and the temperature gradient vector on the heat transfer. Fig. 8 shows the synergy angle in the cross section of the tube. It is seen that the synergy angle of the tube center is large and the synergy is poor. And the synergy angle of HTSC is obviously smaller than that of HT, indicating that HTSC has better heat transfer performance.



**Fig. 8** The Synergy angle of cross section: (a) HT,  $\Phi=3\pi$ , (b) HT,  $\Phi=4\pi$ , (c) HTSC,  $\Phi=3\pi$ , (d) HTSC,  $\Phi=4\pi$ .

3.3. Nusselt Number and Pressure drop

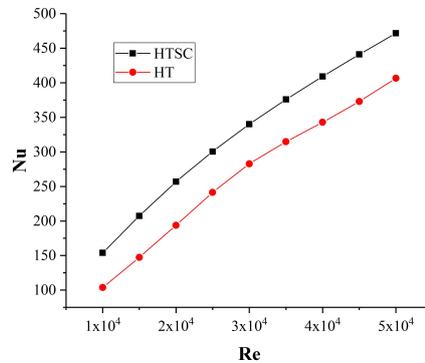


Fig. 9 The variations of Nu with Re

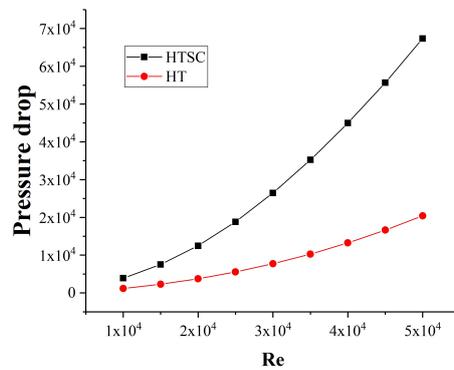


Fig. 10 The variations of pressure drop with Re

Fig. 9 and Fig. 10 respectively show variations of Nu and pressure drop with Re for HTSC as well as HT. Clearly, as shown in Fig. 9, the Nusselt numbers of HTSC and HT increase with the increase of Re. The main reason is that with the increase of Re, the intensity of turbulence in the tube increases, which is conducive to heat exchange. At the same time, as shown in Fig. 10, the pressure drop of HTSC and HT also increases with the increase of Re, indicating that when the turbulence intensity increases, the pressure loss in the tube increases. Compared with HT, the heat transfer performance of HTSC is improved by 20-50%. However, the pressure drop increases by about 100%.

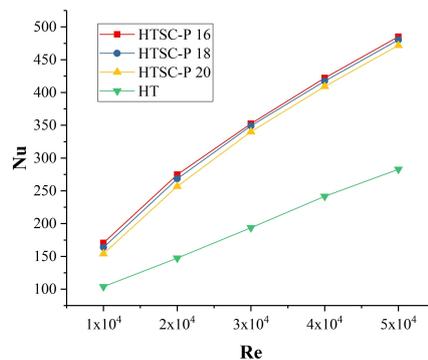
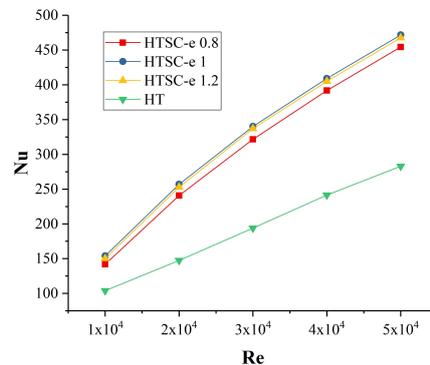


Fig. 11 The variations of Nu with Re



**Fig. 12** The variations of  $Nu$  with  $Re$

Fig.11 and Fig. 12 show the influence of pitch and height of spiral corrugation on heat transfer performance. It is seen that the increase of pitch will reduce the Nusselt number. In addition, when the height of the spiral corrugations is close to 1, it has better heat transfer performance.

#### 4. Conclusion

(1) For the helical tube with spiral corrugations, vortexes near the wall surface exist and the turbulence intensity is enhanced due to the existence of the spiral corrugations.

(2) Compared with the helical smooth tube, the heat transfer performance of helical tube with spiral corrugations is improved by 20-50%, but the pressure drop increases by about 100%.

(3) For the helical tube with spiral corrugations studied here, increase of pitch will reduce the Nusselt number and the best heat transfer performance occurs when the height of the spiral corrugations is close to 1.

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